

INTRODUCTION to FINITE ELEMENT METHODS

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Material assembled from Lecture Notes for the course **Introduction to Finite Elements Methods** (ASEN 5007) offered from 1986 to date at the Aerospace Engineering Sciences Department of the University of Colorado at Boulder.

Preface

This textbook presents an Introduction to the computer-based simulation of linear structures by the Finite Element Method (FEM). It assembles the “converged” lecture notes of **Introduction to Finite Element Methods** or IFEM. This is a core graduate course offered in the Department of Aerospace Engineering Sciences of the University of Colorado at Boulder.

IFEM was first taught on the Fall Semester 1986 and has been repeated every year since. It is taken by both first-year graduate students as part of their M.S. or M.E. requirements, and by senior undergraduates as technical elective. Selected material in Chapters 1 through 3 is used to teach a two-week introduction of Matrix Structural Analysis and Finite Element concepts to junior undergraduate students who are taking their first Mechanics of Materials course.

Prerequisites for the graduate-level course are multivariate calculus, linear algebra, a basic knowledge of structural mechanics at the Mechanics of Materials level, and some familiarity with programming concepts learnt in undergraduate courses.

The course originally used Fortran 77 as computer implementation language. This has been gradually changed to *Mathematica* since 1995. The changeover is now complete. No prior knowledge of *Mathematica* is required because that language, unlike Fortran or similar low-level programming languages, can be picked up while “going along.” Inasmuch as *Mathematica* supports both symbolic and numeric computation, as well as direct use of visualization tools, the use of the language is interspersed throughout the book.

Book Objectives

“In science there is only physics; all the rest is stamp collecting” (Lord Kelvin). The quote reflects the values of the mid-XIX century. Even now, at the dawn of the XXIth, progress and prestige in the natural sciences favors fundamental knowledge. By contrast, engineering knowledge consists of three components:¹

1. Conceptual knowledge: understanding the framework of the physical world.
2. Operational knowledge: methods and strategies for formulating, analyzing and solving problems, or “which buttons to push.”
3. Integral knowledge: the synthesis of conceptual and operational knowledge for technology development.

The language that connects conceptual and operational knowledge is mathematics, and in particular the use of mathematical models. Most engineering programs in the USA correctly emphasize both conceptual and operational components. They differ, however, in how well the two are integrated. The most successful curricula are those that address the tendency to “horizontal disconnection” that bedevils engineering students suddenly exposed to a vast array of subjects.

Integral knowledge is unique to the engineering profession. Synthesis ability is a personal attribute that cannot be coerced, only encouraged and cultivated, the same as the best music programs do not

¹ Extracted from: B. M. Argrow, Pro-active teaching and learning in the Aerospace Engineering Sciences Curriculum 2000, internal report, University of Colorado, February 2001.

automatically produce Mozarts. Studies indicate no correlation between good engineers and good students.² The best that can be done is to provide an adequate (and integrated) base of conceptual and operational knowledge to potentially good engineers.

Where does the Finite Element Method (FEM) fit in this framework?

FEM was developed initially, and prospered, as a computer-based simulation method for the analysis of aerospace structures. Then it found its way into both design and analysis of complex structural systems, not only in Aerospace but in Civil and Mechanical Engineering. In the late 1960s it expanded to the simulation of non-structural problems in fluids, thermomechanics and electromagnetics. This “Physical FEM” is an *operational tool*, which fits primarily the operational knowledge component of engineering, and draws from the mathematical models of the real world. It is the form emphasized in the first part of this book.

The success of FEM as a general-purpose simulation method attracted attention in the 1970s from two quarters beyond engineering: mathematicians and software entrepreneurs. The world of FEM eventually split into applications, mathematics, and commercial software products. The former two are largely housed in the comfortable obscurity of academia. There is little cross-talk between these communities. They have different perspectives. They have separate constituencies, conferences and publication media, which slows down technology transfer. As of this writing, the three-way split seems likely to continue, as long as there is no incentive to do otherwise.

This book aims to keep a presentation balance: the physical and mathematical interpretations of FEM are used eclectically, with none overshadowing the other. Key steps of the computer implementation are presented in sufficient detail so that a student can understand what goes on behind the scenes of a “black box” commercial product. The goal is that students navigating this material can eventually feel comfortable with any of the three “FEM communities” they come in contact during their professional life, whether as engineers, managers, researchers or teachers.

Book Organization

The book is divided into three Parts of roughly similar length.

Part I: The Direct Stiffness Method. This part comprises Chapters 1 through 11. It covers major aspects of the Direct Stiffness Method (DSM). This is the most important realization of FEM, and the one implemented in general-purpose commercial finite element codes used by practicing engineers. Following an introductory first chapter, Chapters 2-4 present the fundamental steps of the DSM as a matrix method of structural analysis. A plane truss structure is used as motivating example. This is followed by Chapters 5-10 on programming, element formulation, modeling issues, and techniques for application of boundary conditions. Chapter 11 deals with relatively advanced topics including condensation and global-local analysis. Throughout these chapters the physical interpretation is emphasized for pedagogical convenience, as unifying vision of this “horizontal” framework.

Part II: Formulation of Finite Elements. This part extends from Chapters 12 through 19. It is more focused than Part I. It covers the development of elements from the more general viewpoint of the variational (energy) formulation. The presentation is inductive, always focusing on specific elements and progressing from the simplest to more complex cases. Thus Chapter 12 rederives the

² As evaluated by conventional academic metrics, which primarily test operational knowledge. One difficulty with teaching synthesis is that good engineers and designers are highly valued in industry but rarely comfortable in academia.

plane truss (bar) element from a variational formulation, while Chapter 13 presents the plane beam element. Chapter 14 introduces the plane stress problem, which serves as a testbed for the derivation of two-dimensional isoparametric elements in Chapter 15 through 18. This part concludes with an overview of requirements for convergence.

Part III: Computer Implementation. Chapters 21 through 28 deal with the computer implementation of the finite element method. Experience has indicated that students profit from doing computer homework early. This begins with Chapter 5, which contains an Introduction to *Mathematica*, and continues with homework assignments in Parts I and II. The emphasis changes in Part III to a systematic description of components of FEM programs, and the integration of those components to do problem solving.

Exercises

Most Chapters are followed by a list of homework exercises that pose problems of varying difficulty. Each exercise is labeled by a tag of the form

[type:rating]

The type is indicated by letters A, C, D or N for exercises to be answered primarily by analytical work, computer programming, descriptive narration, and numerical calculations, respectively. Some exercises involve a combination of these traits, in which case a combination of letters separated by + is used; for example A+N indicates analytical derivation followed by numerical work. For some problems heavy analytical work may be helped by the use of a computer-algebra system, in which case the type is identified as A/C.

The rating is a number between 5 and 50 that estimates the degree of difficulty of an Exercise, in the following “logarithmic” scale:

- 5 A simple question that can be answered in seconds, or is already answered in the text if the student has read and understood the material.
- 10 A straightforward question that can be answered in minutes.
- 15 A relatively simple question that requires some thinking, and may take on the order of half to one hour to answer.
- 20 Either a problem of moderate difficulty, or a straightforward one requiring lengthy computations or some programming, normally taking one to six hours of work.
- 25 A scaled up version of the above, estimated to require six hours to one day of work.
- 30 A problem of moderate difficulty that normally requires on the order of one or two days of work. Arriving at the answer may involve a combination of techniques, some background or reference material, or lengthy but straightforward programming.
- 40 A difficult problem that may be solvable only by gifted and well prepared individual students, or a team. Difficulties may be due to the need of correct formulation, advanced mathematics, or high level programming. With the proper preparation, background and tools these problems may be solved in days or weeks, while remaining inaccessible to unprepared or average students.
- 50 A research problem, worthy of publication if solved.

Most Exercises have a rating of 15 or 20. Assigning three or four per week puts a load of roughly 5-10 hours of solution work, plus the time needed to prepare the answer material. Assignments of difficulty

25 or 30 are better handled by groups, or given in take-home exams. Assignments of difficulty beyond 30 are never assigned in the course, but listed as a challenge for an elite group.

Occasionally an Exercise has two or more distinct but related parts identified as items. In that case a rating may be given for each item. For example: [A/C:15+20]. This does not mean that the exercise as a whole has a difficulty of 35, because the scale is roughly logarithmic; the numbers simply rate the expected effort per item.

Selecting Course Material

The number of chapters has been coordinated with the 28 lectures and two midterm exams of a typical 15-week semester course offered with two 75-minute lectures per week. The expectation is to cover one chapter per lecture. Midterm exams cover selective material in Parts I and II, whereas a final exam covers the entire course. It is recommended to make this final exam a one-week take-home to facilitate computer programming assignments. Alternatively a final term project may be considered. The experience of the writer, however, is that term projects are not useful at this level, since most first-year graduate students lack the synthesis ability that develops in subsequent years.

The writer has assigned weekly homeworks by selecting exercises from the two Chapters covered in the week. Choices are often given. The rating may be used by graders to weight scores. Unlike exams, group homeworks with teams of two to four students are recommended. Teams are encouraged to consult other students, as well as the instructor and teaching assistants, to get over gaps and hurdles. This group activity also lessens schedule conflicts common to working graduate students.

Feedback from course offerings as well as advances in topics such as programming languages resulted in new material being incorporated at various intervals. To keep within course coverage constraints, three courses of action were followed in revising the book.

Deleted Topics. All advanced analysis material dealing with variational calculus and direct approximation methods such as Rayleigh-Ritz, Galerkin, least squares and collocation, was eliminated by 1990. The few results needed for Part II are stated therein as recipes. That material was found to be largely a waste of time for engineering students, who typically lack the mathematical background required to appreciate the meaning and use of these methods in an application-independent context.³ Furthermore, there is abundant literature that interested students may consult should they decide to further their knowledge in those topics for self-study or thesis work.

Appendices. “Refresher” material on vector and matrix algebra has been placed on Appendices A through D. This is material that students are supposed to know as a prerequisite. Although most of it is covered by a vast literature, it was felt advisable to help students in collecting key results for quick reference in one place, and establishing a consistent notational system.

Starred Material. Chapter-specific material that is not normally covered in class is presented in fine print sections marked with an asterisk. This material belongs to two categories. One is extension to the basic topics, which suggest the way it would be covered in a more advanced FEM course. The other includes general exposition or proofs of techniques presented as recipes in class for expedience or time constraints. Starred material may be used as source for term projects or take-home exams.

The book organization presents flexibility to instructors in organizing the coverage for shorter courses, for example in a quarter system, as well as fitting a three-lectures-per-week format. For the latter case

³ This is a manifestation of the disconnection difficulty noted at the start of this Preface.

it is recommended to cover two Chapters per week, while maintaining weekly homework assignments. In a quarter system a more drastic condensation would be necessary; for example much of Part I may be left out if the curriculum includes a separate course in Matrix Structural Analysis, as is common in Civil and Architectural Engineering.

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Overview

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This book is an introduction to the analysis of linear elastic structures by the Finite Element Method (FEM). This Chapter presents an overview of where the book fits, and what finite elements are.

§1.1. WHERE THIS MATERIAL FITS

The field of Mechanics can be subdivided into three major areas:

$$\text{Mechanics} \begin{cases} \textit{Theoretical} \\ \textit{Applied} \\ \textit{Computational} \end{cases} \quad (1.1)$$

Theoretical mechanics deals with fundamental laws and principles of mechanics studied for their intrinsic scientific value. *Applied mechanics* transfers this theoretical knowledge to scientific and engineering applications, especially as regards the construction of mathematical models of physical phenomena. *Computational mechanics* solves specific problems by simulation through numerical methods implemented on digital computers.

REMARK 1.1

Paraphrasing an old joke about mathematicians, one may define a computational mechanician as a person who searches for solutions to given problems, an applied mechanician as a person who searches for problems that fit given solutions, and a theoretical mechanician as a person who can prove the existence of problems and solutions.

§1.1.1. Computational Mechanics

Several branches of computational mechanics can be distinguished according to the *physical scale* of the focus of attention:

$$\text{Computational Mechanics} \begin{cases} \textit{Nanomechanics and micromechanics} \\ \textit{Continuum mechanics} \begin{cases} \text{Solids and Structures} \\ \text{Fluids} \\ \text{Multiphysics} \end{cases} \\ \textit{Systems} \end{cases} \quad (1.2)$$

Nanomechanics deals with phenomena at the molecular and atomic levels of matter. As such it is closely to connected particle physics and chemistry. Micromechanics looks primarily at the crystallographic and granular levels of matter. Its main technological application is the design and fabrication of materials and microdevices.

Continuum mechanics studies bodies at the macroscopic level, using continuum models in which the microstructure is homogenized by phenomenological averages. The two traditional areas of application are solid and fluid mechanics. The former includes *structures* which, for obvious reasons, are fabricated with solids. Computational solid mechanics takes an applied sciences approach, whereas computational structural mechanics emphasizes technological applications to the analysis and design of structures.

Computational fluid mechanics deals with problems that involve the equilibrium and motion of liquid and gases. Well developed subsidiaries are hydrodynamics, aerodynamics, acoustics, atmospheric physics, shock and combustion.

Multiphysics is a more recent newcomer. This area is meant to include mechanical systems that transcend the classical boundaries of solid and fluid mechanics, as in interacting fluids and structures. Phase change problems such as ice melting and metal solidification fit into this category, as do the interaction of control, mechanical and electromagnetic systems.

Finally, *system* identifies mechanical objects, whether natural or artificial, that perform a distinguishable function. Examples of man-made systems are airplanes, buildings, bridges, engines, cars, microchips, radio telescopes, robots, roller skates and garden sprinklers. Biological systems, such as a whale, amoeba, inner ear, or pine tree are included if studied from the viewpoint of biomechanics. Ecological, astronomical and cosmological entities also form systems.¹

In the progression of (1.2) the *system* is the most general concept. A system is studied by *decomposition*: its behavior is that of its components plus the interaction between the components. Components are broken down into subcomponents and so on. As this hierarchical process continues the individual components become simple enough to be treated by individual disciplines, but their interactions may get more complex. Consequently there is a tradeoff art in deciding where to stop.²

§1.1.2. Statics vs. Dynamics

Continuum mechanics problems may be subdivided according to whether inertial effects are taken into account or not:

$$\text{Continuum mechanics} \begin{cases} \text{Statics} \\ \text{Dynamics} \end{cases} \quad (1.3)$$

In dynamics the time dependence is explicitly considered because the calculation of inertial (and/or damping) forces requires derivatives respect to actual time to be taken.

Problems in statics may also be time dependent but the inertial forces are ignored or neglected. Static problems may be classified into strictly static and quasi-static. For the former time need not be considered explicitly; any historical time-like response-ordering parameter (if one is needed) will do. In quasi-static problems such as foundation settlement, creep deformation, rate-dependent plasticity or fatigue cycling, a more realistic estimation of time is required but inertial forces are still neglected.

§1.1.3. Linear vs. Nonlinear

A classification of static problems that is particularly relevant to this book is

$$\text{Statics} \begin{cases} \text{Linear} \\ \text{Nonlinear} \end{cases} \quad (1.4)$$

Linear static analysis deals with static problems in which the *response* is linear in the cause-and-effect sense. For example: if the applied forces are doubled, the displacements and internal stresses also double. Problems outside this domain are classified as nonlinear.

¹ Except that their function may not be clear to us. “The usual approach of science of constructing a mathematical model cannot answer the questions of why there should be a universe for the model to describe. Why does the universe go to all the bother of existing?” (Stephen Hawking).

² Thus in breaking down a car engine, say, the decomposition does not usually proceed beyond the components you can buy at a parts shop.

§1.1.4. Discretization methods

A final classification of CSM static analysis is based on the discretization method by which the continuum mathematical model is *discretized* in space, *i.e.*, converted to a discrete model of finite number of degrees of freedom:

$$\text{Spatial discretization method} \left\{ \begin{array}{l} \text{Finite Element (FEM)} \\ \text{Boundary Element (BEM)} \\ \text{Finite Difference (FDM)} \\ \text{Finite Volume (FVM)} \\ \text{Spectral} \\ \text{Meshfree} \end{array} \right. \quad (1.5)$$

In CSM *linear* problems finite element methods currently dominate the scene. Boundary element methods post a strong second choice in specific application areas. For *nonlinear* problems the dominance of finite element methods is overwhelming.

Classical finite difference methods in solid and structural mechanics have virtually disappeared from practical use. This statement is not true, however, for fluid mechanics, where finite difference discretization methods are still important. Finite-volume methods, which directly address the discretization of conservation laws, are important in difficult problems of fluid mechanics, for example high-Re gas dynamics. Spectral methods are based on transforms that map space and/or time dimensions to spaces where the problem is easier to solve.

A recent newcomer to the scene are the meshfree methods. These are finite difference methods on arbitrary grids constructed through a subset of finite element techniques and tools.

§1.1.5. FEM Variants

The term *Finite Element Method* actually identifies a broad spectrum of techniques that share common features outlined in §1.3 and §1.4. Two subclassifications that fit well applications to structural mechanics are³

$$\text{FEM Formulation} \left\{ \begin{array}{l} \text{Displacement} \\ \text{Equilibrium} \\ \text{Mixed} \\ \text{Hybrid} \end{array} \right. \quad \text{FEM Solution} \left\{ \begin{array}{l} \text{Stiffness} \\ \text{Flexibility} \\ \text{Mixed (a.k.a. Combined)} \end{array} \right. \quad (1.6)$$

Using the foregoing classification, we can state the topic of this book more precisely: the *computational analysis of linear static structural problems* by the Finite Element Method. Of the variants listed in (1.6), emphasis is placed on the *displacement* formulation and *stiffness* solution. This combination is called the *Direct Stiffness Method* or DSM.

³ The distinction between these subclasses require advanced technical concepts, which cannot be covered in an introductory treatment such as this book.

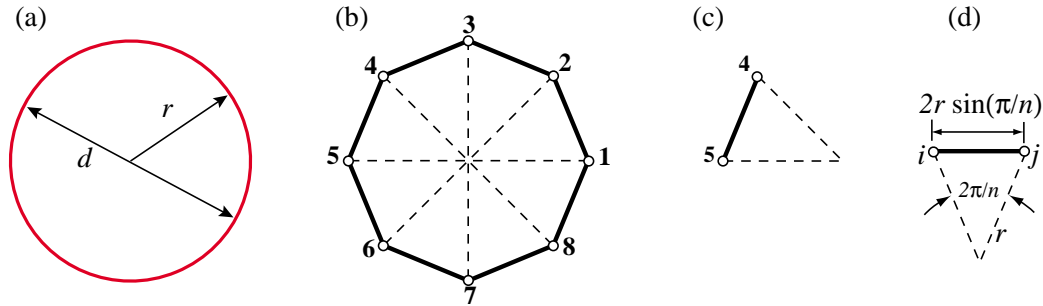


Figure 1.1. The “find π ” problem treated with FEM concepts: (a) continuum object, (b) a discrete approximation (inscribed regular polygon), (c) disconnected element, (d) generic element.

§1.2. WHAT DOES A FINITE ELEMENT LOOK LIKE?

The subject of this book is FEM. But what is a finite element? The concept will be partly illustrated through a truly ancient problem: find the perimeter L of a circle of diameter d . Since $L = \pi d$, this is equivalent to obtaining a numerical value for π .

Draw a circle of radius r and diameter $d = 2r$ as in Figure figOverViewArchimedes(a). Inscribe a regular polygon of n sides, where $n = 8$ in Figure 1.1(b). Rename polygon sides as *elements* and vertices as *nodes*. Label nodes with integers $1, \dots, 8$. Extract a typical element, say that joining nodes 4–5, as shown in Figure 1.1(c). This is an instance of the *generic element* i – j shown in Figure 1.1(d). The element length is $L_{ij} = 2r \sin(\pi/n)$. Since all elements have the same length, the polygon perimeter is $L_n = nL_{ij}$, whence the approximation to π is $\pi_n = L_n/d = n \sin(\pi/n)$.

Table 1.1. Rectification of Circle by Inscribed Polygons (“Archimedes FEM”)

n	$\pi_n = n \sin(\pi/n)$	Extrapolated by Wynn- ϵ	Exact π to 16 places
1	0.0000000000000000		
2	2.0000000000000000		
4	2.828427124746190	3.414213562373096	
8	3.061467458920718		
16	3.121445152258052	3.141418327933211	
32	3.136548490545939		
64	3.140331156954753	3.141592658918053	
128	3.141277250932773		
256	3.141513801144301	3.141592653589786	3.141592653589793

Values of π_n obtained for $n = 1, 2, 4, \dots, 256$ are listed in the second column of Table 1.1. As can be seen the convergence to π is fairly slow. However, the sequence can be transformed by Wynn’s ϵ algorithm⁴ into that shown in the third column. The last value displays 15-place accuracy.

Some key ideas behind the FEM can be identified in this example. The circle, viewed as a *source mathematical object*, is replaced by polygons. These are *discrete approximations* to the circle.

⁴ A widely used extrapolation algorithm that speeds up the convergence of many sequences. See, e.g., [1.54].

The sides, renamed as *elements*, are specified by their end *nodes*. Elements can be separated by disconnecting nodes, a process called *disassembly* in the FEM. Upon disassembly a *generic element* can be defined, *independently of the original circle*, by the segment that connects two nodes i and j . The relevant element property: length L_{ij} , can be computed in the generic element independently of the others, a property called *local support* in the FEM. The target property: the polygon perimeter, is obtained by reconnecting n elements and adding up their length; the corresponding steps in the FEM being *assembly* and *solution*, respectively. There is of course nothing magic about the circle; the same technique can be used to rectify any smooth plane curve.⁵

This example has been offered in the FEM literature to aduce that finite element ideas can be traced to Egyptian mathematicians from *circa* 1800 B.C., as well as Archimedes' famous studies on circle rectification by 250 B.C. But comparison with the modern FEM, as covered in Chapters 2–3, shows this to be a stretch. The example does not illustrate the concept of degrees of freedom, conjugate quantities and local-global coordinates. It is guilty of circular reasoning: the compact formula $\pi = \lim_{n \rightarrow \infty} n \sin(\pi/n)$ uses the unknown π in the right hand side.⁶ Reasonable people would argue that a circle is a simpler object than, say, a 128-sided polygon. Despite these flaws the example is useful in one respect: showing a fielder's choice in the replacement of one mathematical object by another. This is at the root of the simulation process described in the next section.

§1.3. THE FEM ANALYSIS PROCESS

Processes using FEM in some way involve carrying out a sequence of steps. Those sequences take two canonical configurations, depending on the environment in which FEM is used and the main objective: model-based physical simulation, or mathematical derivations. These are reviewed next to introduce terminology often used in the sequel.

§1.3.1. The Physical FEM

A canonical use of FEM is simulation of physical systems. This has to be done by using models. Thus the process is often called *model-based simulation*. The process is illustrated in Figure 1.2. The centerpiece is the *physical system* to be modeled. Accordingly, this sequence is called the *Physical FEM*. The processes of idealization and discretization are carried out *concurrently* to produce the discrete model. The solution is computed as before.

Figure 1.2 depicts a *ideal mathematical model*. This may be presented as a *continuum limit* or “continuification” of the discrete model. For some physical systems, notably those well modeled by continuum fields, this step is useful. For others, such as complex engineering systems, it makes no sense. Indeed FEM discretizations may be constructed and adjusted without reference to mathematical models, simply from experimental measurements.

The concept of *error* arises in the Physical FEM in two ways. These are known as *verification* and *validation*, respectively. Verification is done by replacing the discrete solution is into the discrete model to get the solution error. This error is not generally important. Substitution in the ideal

⁵ A similar limit process, however, may fail in three or more dimensions.

⁶ This objection is bypassed if n is advanced as a power of two, as in Table 1.1, by using the half-angle recursion

$$\sqrt{2} \sin \alpha = \sqrt{1 - \sqrt{1 - \sin^2 2\alpha}}, \text{ started from } 2\alpha = \pi \text{ for which } \sin \pi = -1.$$

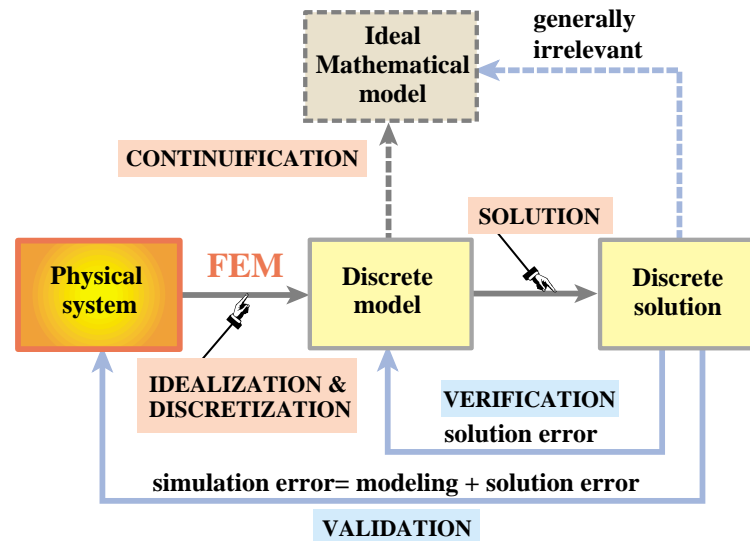


Figure 1.2. The Physical FEM. The physical system (left) is the source of the simulation process. The ideal mathematical model (should one go to the trouble of constructing it) is inessential.

mathematical model in principle provides the discretization error. This step is rarely useful in complex engineering systems, however, because there is no reason to expect that the mathematical model exists, and even if it does, that it is more physically relevant than the discrete model. Validation tries to compare the discrete solution against observation by computing the *simulation error*, which combines modeling and solution errors. As the latter is typically insignificant, the simulation error in practice can be identified with the modeling error.

One way to adjust the discrete model so that it represents the physics better is called *model updating*. The discrete model is given free parameters. These are determined by comparing the discrete solution against experiments, as illustrated in Figure 1.3. Inasmuch as the minimization conditions are generally nonlinear (even if the model is linear) the updating process is inherently iterative.

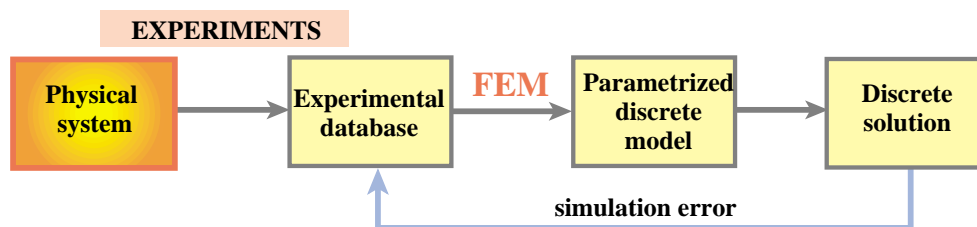


Figure 1.3. Model updating process in the Physical FEM.

§1.3.2. The Mathematical FEM

The other canonical way of using FEM centers on the mathematics. The process steps are illustrated in Figure 1.4. Spotlight now falls on the *mathematical model*. This is often an ordinary or

partial differential equation in space and time. A discrete finite element model is generated from a variational or weak form of the mathematical model.⁷ This is the *discretization* step. The FEM equations are processed by an equation solver, which delivers a discrete solution (or solutions).

On the left Figure 1.4 shows an *ideal physical system*. This may be presented as a *realization* of the mathematical model; conversely, the mathematical model is said to be an *idealization* of this system. For example, if the mathematical model is the Poisson's equation, realizations may be a heat conduction or a electrostatic charge distribution problem. This step is inessential and may be left out. Indeed FEM discretizations may be constructed without any reference to physics.

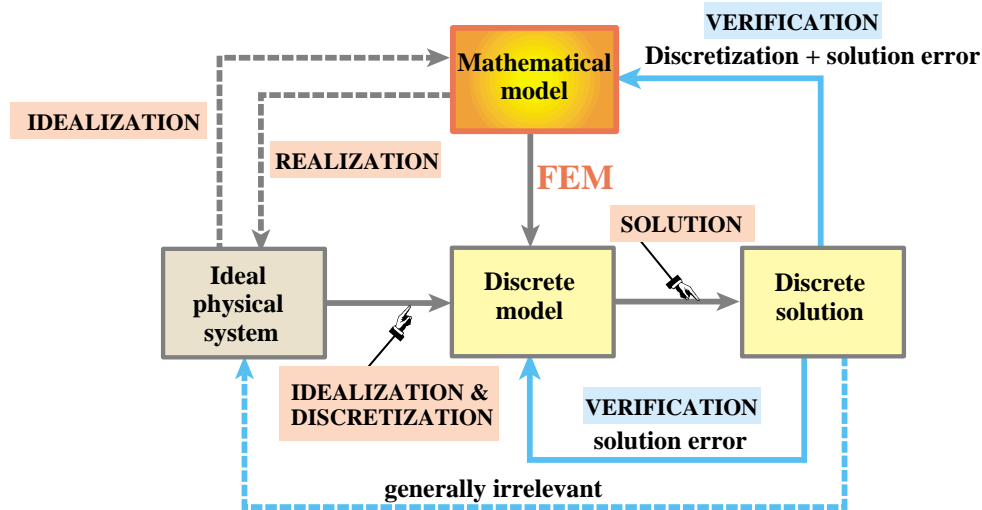


Figure 1.4. The Mathematical FEM. The mathematical model (top) is the source of the simulation process. Discrete model and solution follow from it. The ideal physical system (should one go to the trouble of exhibiting it) is inessential.

The concept of *error* arises when the discrete solution is substituted in the “model” boxes. This replacement is generically called *verification*. As in the Physical FEM, the *solution error* is the amount by which the discrete solution fails to satisfy the discrete equations. This error is relatively unimportant when using computers, and in particular direct linear equation solvers, for the solution step. More relevant is the *discretization error*, which is the amount by which the discrete solution fails to satisfy the mathematical model.⁸ Replacing into the ideal physical system would in principle quantify modeling errors. In the mathematical FEM this is largely irrelevant, however, because the ideal physical system is merely that: a figment of the imagination.

§1.3.3. Synergy of Physical and Mathematical FEM

The foregoing canonical sequences are not exclusive but complementary. This synergy⁹ is one of the reasons behind the power and acceptance of the method. Historically the Physical FEM was

⁷ The distinction between strong, weak and variational forms is discussed in advanced FEM courses. In the present book such forms will be stated as recipes.

⁸ This error can be computed in several ways, the details of which are of no importance here.

⁹ Such interplay is not exactly a new idea: “The men of experiment are like the ant, they only collect and use; the reasoners

the first one to be developed to model complex physical systems such as aircraft, as narrated in Appendix H. The Mathematical FEM came later and, among other things, provided the necessary theoretical underpinnings to extend FEM beyond structural analysis.

A glance at the schematics of a commercial jet aircraft makes obvious the reasons behind the Physical FEM. There is no simple differential equation that captures, at a continuum mechanics level,¹⁰ the structure, avionics, fuel, propulsion, cargo, and passengers eating dinner.

There is no reason for despair, however. The time honored *divide and conquer* strategy, coupled with *abstraction*, comes to the rescue. First, separate the structure and view the rest as masses and forces, most of which are time-varying and nondeterministic.

Second, consider the aircraft structure as built of *substructures* (a part of a structure devoted to a specific function): wings, fuselage, stabilizers, engines, landing gears, and so on. Take each substructure, and continue to break it down into *components*: rings, ribs, spars, cover plates, actuators, etc, continuing through as many levels as necessary. Eventually those components become sufficiently simple in geometry and connectivity that they can be reasonably well described by the continuum mathematical models provided, for instance, by Mechanics of Materials or the Theory of Elasticity. At that point, *stop*. The component level discrete equations are obtained from a FEM library based on the mathematical model. The system model is obtained by going through the reverse process: from component equations to substructure equations, and from those to the equations of the complete aircraft.

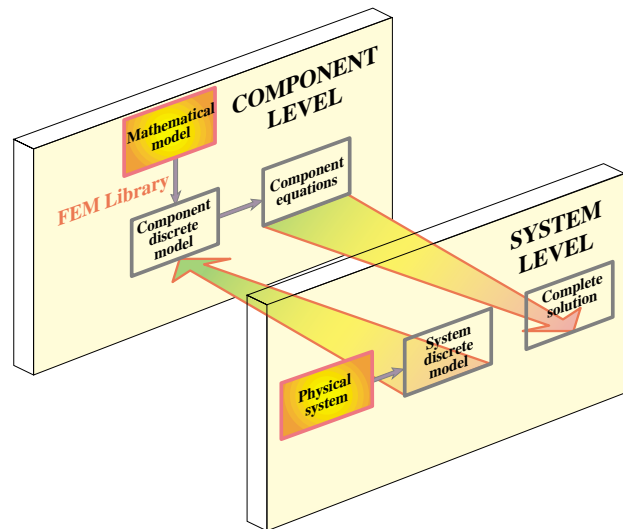


Figure 1.5. Combining physical and mathematical modeling through multilevel FEM. Only two levels (system and component) are shown for simplicity; intermediate substructure levels are omitted.

This *system assembly* process is governed by the classical principles of Newtonian mechanics expressed in conservation form. The multilevel decomposition process is diagramed in Figure 1.5, in which the intermediate substructure level is omitted for simplicity.

REMARK 1.2

More intermediate decomposition levels are used in some systems, such as offshore and ship structures, which are characterized by a modular fabrication process. In that case the decomposition mimics the way the system is actually constructed. The general technique, called *superelements*, is discussed in Chapter 11.

resemble spiders, who make cobwebs out of their own substance. But the bee takes the middle course: it gathers its material from the flowers of the garden and field, but transforms and digests it by a power of its own.” (Francis Bacon, 1620).

¹⁰ Of course at the atomic and subatomic level quantum mechanics works for everything, from landing gears to passengers. But it would be slightly impractical to model the aircraft by 10^{36} interacting particles.

REMARK 1.3

There is no point in practice in going beyond a certain component level while considering the complete model, since the level of detail can become overwhelming without adding significant information. Further refinement of particular components is done by the so-called global-local analysis technique outlined in Chapter 11. This technique is an instance of multiscale analysis.

For sufficiently simple structures, passing to a discrete model is carried out in a single *idealization and discretization* step, as illustrated for the truss roof structure shown in Figure 1.6. Multiple levels are unnecessary here. Of course the truss may be viewed as a substructure of the roof, and the roof as a substructure of a building.

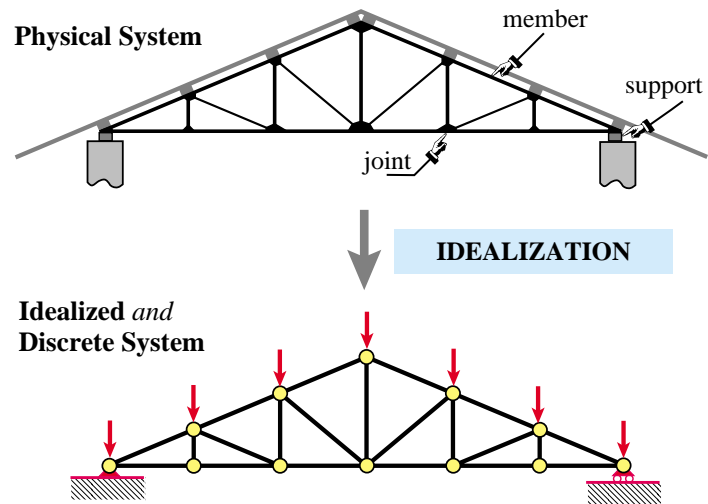


Figure 1.6. The idealization process for a simple structure. The physical system, here a roof truss, is directly idealized by the mathematical model: a pin-jointed bar assembly. For this particular structure idealized and discrete models coalesce.

§1.4. INTERPRETATIONS OF THE FINITE ELEMENT METHOD

Just like there are two complementary ways of using the FEM, there are two complementary interpretations for teaching it. One interpretation stresses the physical significance and is aligned with the Physical FEM. The other focuses on the mathematical context, and is aligned with the Mathematical FEM.

§1.4.1. Physical Interpretation

The physical interpretation focuses on the flowchart of Figure 1.2. This interpretation has been shaped by the discovery and extensive use of the method in the field of structural mechanics. This relationship is reflected in the use of structural terms such as “stiffness matrix”, “force vector” and “degrees of freedom.” This terminology carries over to non-structural applications.

The basic concept in the physical interpretation is the *breakdown* (\equiv disassembly, tearing, partition, separation, decomposition) of a complex mechanical system into simpler, disjoint components called finite elements, or simply *elements*. The mechanical response of an element is characterized in terms of a finite number of degrees of freedom. These degrees of freedoms are represented as the values of the unknown functions as a set of node points. The element response is defined by algebraic equations constructed from mathematical or experimental arguments. The response of the original system is considered to be approximated by that of the *discrete model* constructed by *connecting* or *assembling* the collection of all elements.

The breakdown-assembly concept occurs naturally when an engineer considers many artificial and natural systems. For example, it is easy and natural to visualize an engine, bridge, aircraft or skeleton as being fabricated from simpler parts.

As discussed in §1.3, the underlying theme is *divide and conquer*. If the behavior of a system is too complex, the recipe is to divide it into more manageable subsystems. If these subsystems are still too complex the subdivision process is continued until the behavior of each subsystem is simple enough to fit a mathematical model that represents well the knowledge level the analyst is interested in. In the finite element method such “primitive pieces” are called *elements*. The behavior of the total system is that of the individual elements plus their interaction. A key factor in the initial acceptance of the FEM was that the element interaction can be physically interpreted and understood in terms that were eminently familiar to structural engineers.

§1.4.2. Mathematical Interpretation

This interpretation is closely aligned with the flowchart of Figure 1.4. The FEM is viewed as a procedure for obtaining numerical approximations to the solution of boundary value problems (BVPs) posed over a domain Ω . This domain is replaced by the union \cup of disjoint subdomains $\Omega^{(e)}$ called finite elements. In general the geometry of Ω is only approximated by that of $\cup\Omega^{(e)}$.

The unknown function (or functions) is locally approximated over each element by an interpolation formula expressed in terms of values taken by the function(s), and possibly their derivatives, at a set of *node points* generally located on the element boundaries. The states of the assumed unknown function(s) determined by unit node values are called *shape functions*. The union of shape functions “patched” over adjacent elements form a *trial function basis* for which the node values represent the generalized coordinates. The trial function space may be inserted into the governing equations and the unknown node values determined by the Ritz method (if the solution extremizes a variational principle) or by the Galerkin, least-squares or other weighted-residual minimization methods if the problem cannot be expressed in a standard variational form.

REMARK 1.4

In the mathematical interpretation the emphasis is on the concept of *local (piecewise) approximation*. The concept of element-by-element breakdown and assembly, while convenient in the computer implementation, is not theoretically necessary. The mathematical interpretation permits a general approach to the questions of convergence, error bounds, trial and shape function requirements, etc., which the physical approach leaves unanswered. It also facilitates the application of FEM to classes of problems that are not so readily amenable to physical visualization as structures; for example electromagnetics and thermal conduction.

REMARK 1.5

It is interesting to note some similarities in the development of Heaviside’s operational methods, Dirac’s delta-function calculus, and the FEM. These three methods appeared as ad-hoc computational devices created by engineers and physicists to deal with problems posed by new science and technology (electricity, quantum mechanics, and delta-wing aircraft, respectively) with little help from the mathematical establishment. Only some time after the success of the new techniques became apparent were new branches of mathematics (operational calculus, distribution theory and piecewise-approximation theory, respectively) constructed to justify that success. In the case of the finite element method, the development of a formal mathematical theory started in the late 1960s, and much of it is still in the making.

§1.5. KEEPING THE COURSE

The first Part of this book, which is the subject of Chapters 2 through 11, stresses the physical interpretation in the framework of the Direct Stiffness Method (DSM) on account of its instructional advantages. Furthermore the computer implementation becomes more transparent because the sequence of computer operations can be placed in close correspondence with the DSM steps.

Subsequent Chapters incorporate ingredients of the mathematical interpretation when it is felt convenient to do so. However, the exposition avoids excessive entanglement with the mathematical theory when it may obfuscate the physics.

A historical outline of the evolution of Matrix Structural Analysis into the Finite Element Method is given in Appendix H, which provides appropriate references.

In Chapters 2 through 6 the time is frozen at about 1965, and the DSM presented as an aerospace engineer of that time would have understood it. This is not done for sentimental reasons, although that happens to be the year in which the writer began his thesis work on FEM under Ray Clough.

Virtually all finite element codes are now based on the DSM and the computer implementation has not essentially changed since the late 1960s.¹¹

§1.6. *WHAT IS NOT COVERED

The following topics are not covered in this book:

1. Elements based on equilibrium, mixed and hybrid variational formulations.
2. Flexibility and mixed solution methods of solution.
3. Kirchhoff-based plate and shell elements.
4. Continuum-based plate and shell elements.
5. Variational methods in mechanics.
6. General mathematical theory of finite elements.
7. Vibration analysis.
8. Buckling analysis.
9. General stability analysis.
10. General nonlinear response analysis.
11. Structural optimization.
12. Error estimates and problem-adaptive discretizations.
13. Non-structural and coupled-system applications of FEM.
14. Structural dynamics.
15. Shock and wave-propagation dynamics.
16. Designing and building production-level FEM software and use of special hardware (*e.g.* vector and parallel computers)

Topics 1–7 pertain to what may be called “Advanced Linear FEM”, whereas 9–11 pertain to “Nonlinear FEM”. Topics 12–15 pertain to advanced applications, whereas 16 is an interdisciplinary topic that interweaves with computer science.

§1.7. *HISTORICAL SKETCH AND BIBLIOGRAPHY

This section summarizes the history of structural finite elements since 1950 to date. It functions as a hub for dispersed historical references.

For exposition convenience, structural “finitelementology” may be divided into four generations that span 10 to 15 years each. There are no sharp intergenerational breaks, but noticeable change of emphasis. The following summary does not cover the conjoint evolution of Matrix Structural Analysis into the Direct Stiffness Method from 1934 through 1970. This was the subject of a separate essay [1.15], which is also given in Appendix H.

§1.7.1. G1: The Pioneers

The 1956 paper by Turner, Clough, Martin and Topp [1.50], henceforth abbreviated to TCMT, is recognized as the start of the current FEM, as used in the overwhelming majority of commercial codes. Along with Argyris’ serial [1.1] they prototype the first generation, which spans 1950 through 1962. A panoramic picture of this period is available in two textbooks [1.34,1.41]. Przemieniecki’s text is still reprinted by Dover. The survey by Gallagher [1.19] was influential at the time but is now difficult to access outside libraries.

The pioneers were structural engineers, schooled in classical mechanics. They followed a century of tradition in regarding structural elements as a device to transmit forces. This “element as force transducer” was the

¹¹ With the gradual disappearance of Fortran as a “live” programming language, noted in §1.7.5, changes at the computer implementation level have recently accelerated.

standard view in pre-computer structural analysis. It explains the use of flux assumptions to derive stiffness equations. Element developers worked in, or interacted closely with, the aircraft industry. (One reason is that only large aerospace companies were then able to afford mainframe computers.) Accordingly they focused on thin structures built up with bars, ribs, spars, stiffeners and panels. Although the Classical Force method dominated stress analysis during the 1950s [1.15], stiffness methods were kept alive by use in dynamics and vibration.

§1.7.2. G2: The Golden Age

The next period spans the golden age of FEM: 1962–1972. This is the “variational generation.” Melosh [1.31] showed that conforming displacement models are a form of Rayleigh-Ritz based on the minimum potential energy principle. This influential paper marks the confluence of three lines of research: Argyris’ dual formulation of energy methods [1.1], the Direct Stiffness Method (DSM) of Turner [1.51–1.53], and early ideas of interelement compatibility as basis for error bounding and convergence [1.30,1.17]. G1 workers thought of finite elements as idealizations of structural components. From 1962 onward a two-step interpretation emerges: discrete elements approximate continuum models, which in turn approximate real structures.

By the early 1960s FEM begins to expand into Civil Engineering through Clough’s Boeing-Berkeley connection [1.10] and had been named [1.8,1.9]. Reading de Veubeke’s famous article [1.18] side by side with TCMT [1.50] one can sense the ongoing change in perspective opened up by the variational framework. The first book devoted to FEM appears in 1967 [1.56]. Applications to nonstructural problems had started in 1965 [1.55], and were treated in some depth by Martin and Carey [1.29].

From 1962 onwards the displacement formulation dominates. This was given a big boost by the invention of the isoparametric formulation and related tools (numerical integration, fitted natural coordinates, shape functions, patch test) by Irons and coworkers [1.23–1.25]. Low order displacement models often exhibit disappointing performance. Thus there was a frenzy to develop higher order elements. Other variational formulations, notably hybrids [1.37–1.40], mixed [1.20,1.49] and equilibrium models [1.18] emerged. G2 can be viewed as closed by the monograph of Strang and Fix [1.47], the first book to focus on the mathematical foundations.

§1.7.3. G3: Consolidation

The post-Vietnam economic doldrums are mirrored during this post-1972 period. Gone is the youthful exuberance of the golden age. This is consolidation time. Substantial effort is put into improving the stock of G2 displacement elements by tools initially labeled “variational crimes” [1.46], but later justified. Textbooks by Hughes [1.22] and Bathe [1.3] reflect the technology of this period. Hybrid and mixed formulations record steady progress [1.2]. Assumed strain formulations appear [1.26]. A booming activity in error estimation and mesh adaptivity is fostered by better understanding of the mathematical foundations [1.48].

Commercial FEM codes gradually gain importance. They provide a reality check on what works in the real world and what doesn’t. By the mid-1980s there was gathering evidence that complex and high order elements were commercial flops. Exotic gadgetry interweaved amidst millions of lines of code easily breaks down in new releases. Complexity is particularly dangerous in nonlinear and dynamic analyses conducted by novice users. A trend back toward simplicity starts [1.27,1.28].

§1.7.4. G4: Back to Basics

The fourth generation begins by the early 1980s. More approaches come on the scene, notably the Free Formulation [1.6,1.7], orthogonal hourglass control [1.16], Assumed Natural Strain methods [1.4–1.45], stress hybrid models in natural coordinates [1.35–1.42], as well as variants and derivatives of those approaches: ANDES [1.32,1.14], EAS [1.43,1.44] and others. Although technically diverse the G4 approaches share two common objectives:

- (i) Elements must fit into DSM-based programs since that includes the vast majority of production codes, commercial or otherwise.

- (ii) Elements are kept simple but should provide answers of engineering accuracy with relatively coarse meshes. These were collectively labeled “high performance elements” in 1989 [1.13].

“Things are always at their best in the beginning,” said Pascal. Indeed. By now FEM looks like an aggregate of largely disconnected methods and recipes. The blame should not be placed on the method itself, but the communities split noted in the book Preface.

§1.7.5. Recommended Books for Linear FEM

The literature is vast: over 200 textbooks and monographs have appeared since 1967. Some recommendations for readers interested in further studies within linear FEM are offered below.

Basic level (reference): Zienkiewicz and Taylor [1.58]. This two-volume set is a comprehensive upgrade of the previous edition [1.57]. Primarily an encyclopædic reference work that gives a panoramic coverage of FEM applications, as well as a comprehensive list of references. Not a textbook or monograph. Prior editions suffered from loose mathematics, largely fixed in this one. A three-volume fifth edition has appeared recently.

Basic level (textbook): Cook, Malkus and Plesha [1.11]. This third edition is comprehensive in scope and fairly up to date although the coverage is more superficial than Zienkiewicz and Taylor. A fourth edition has appeared recently.

Intermediate level: Hughes [1.22]. It requires substantial mathematical expertise on the part of the reader. Recently reprinted as Dover edition.

Mathematically oriented: Strang and Fix [1.47]. Still the most readable mathematical treatment for engineers, although outdated in several subjects. Out of print.

Best value for the \$\$\$: Przemieniecki’s Dover edition [1.41], list price \$15.95 (2003). A reprint of a 1966 McGraw-Hill book. Although woefully outdated in many respects (the word “finite element” does not appear except in post-1960 references), it is a valuable reference for programming simple elements. Contains a fairly detailed coverage of substructuring, a highly practical topic missing from the other books. Comprehensive bibliography in Matrix Structural Analysis up to 1966.

Most fun (if you appreciate British “humor”): Irons and Ahmad [1.25]. Out of print.

For buying out-of-print books through web services, check the search engine in www3.addall.com as well as that of www.amazon.com

§1.7.6. Hasta la Vista, Fortran

Most FEM books that include programming samples or even complete programs use Fortran. Those face an uncertain future. Since the mid-1990s, Fortran is gradually disappearing as a programming language taught in USA engineering undergraduate programs. (It still survives in Physics and Chemistry departments because of large amounts of legacy code.) So one end of the pipeline is drying up. Low-level scientific programming is moving to C and C++, mid-level to Java, Perl and C#, high-level to Matlab, Mathematica and their free-source Linux equivalents. How attractive can a book teaching in a dead language be?

To support this argument with some numbers, here is a September-2003 snapshot of ongoing open source software projects listed in <http://freshmeat.net>. This conveys the relative importance of various languages (a mixed bag of newcomers, going-strongs, and have-beens) in the present environment.

Lang	Projects	Perc	Lang	Projects	Perc	Lang	Projects	Perc
Ada	38	0.20%	APL	3	0.02%	ASP	25	0.13%
Assembly	170	0.89%	Awk	40	0.21%	Basic	15	0.08%
C	5447	28.55%	C#	41	0.21%	C++	2443	12.80%
Cold Fusion	10	0.05%	Common Lisp	27	0.14%	Delphi	49	0.26%
Dylan	2	0.01%	Eiffel	20	0.10%	Emacs-Lisp	33	0.17%

Erlang	11	0.06%	Euler	1	0.01%	Euphoria	2	0.01%
Forth	15	0.08%	Fortran	45	0.24%	Haskell	28	0.15%
Java	2332	12.22%	JavaScript	236	1.24%	Lisp	64	0.34%
Logo	2	0.01%	ML	26	0.14%	Modula	7	0.04%
Object Pascal	9	0.05%	Objective C	131	0.69%	Ocaml	20	0.10%
Other	160	0.84%	Other Scripting Engines	82	0.43%			
Pascal	38	0.20%	Perl	2752	14.42%	PHP	2020	10.59%
Pike	3	0.02%	PL/SQL	58	0.30%	Pliant	1	0.01%
PROGRESS	2	0.01%	Prolog	8	0.04%	Python	1171	6.14%
Rexx	7	0.04%	Ruby	127	0.67%	Scheme	76	0.40%
Simula	1	0.01%	Smalltalk	20	0.10%	SQL	294	1.54%
Tcl	356	1.87%	Unix Shell	550	2.88%	Vis Basic	15	0.08%
Xbasic	1	0.01%	YACC	11	0.06%	Zope	34	0.18%
Total Projects: 19079								

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Homework Exercises for Chapter 1

Overview

EXERCISE 1.1

[A:15] Work out Archimedes' problem using a circumscribed regular polygon, with $n = 1, 2, 4, \dots 256$. Does the sequence converge any faster?

EXERCISE 1.2

[D:20] Select one of the following vehicles: truck, car, motorcycle, or bicycle. Draw a two level decomposition of the structure into substructures, and of selected components of some substructures.

EXERCISE 1.3

[D:30] In one of the earliest articles on the FEM, Clough [1.9] writes:

“When idealized as an assemblage of appropriately shaped two- and three-dimensional elements in this manner, an elastic continuum can be analyzed by standard methods of structural analysis. It should be noted that the approximation which is employed in this case is of physical nature; a modified structural system is substituted for the actual continuum. There need be no approximation in the mathematical analysis of this structural system. This feature distinguishes the finite element technique from finite difference methods, in which the exact equations of the actual physical system are solved by approximate mathematical procedures.”

Discuss critically the contents of this paragraph while placing it in the context of time of writing (early 1960s). Is the last sentence accurate?

2

The Direct Stiffness Method: Breakdown

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This Chapter begins the exposition of the Direct Stiffness Method (DSM) of structural analysis. The DSM is by far the most common implementation of the Finite Element Method (FEM). In particular, all major commercial FEM codes are based on the DSM.

The exposition is done by following the DSM steps applied to a simple plane truss structure.

§2.1. WHY A PLANE TRUSS?

The simplest structural finite element is the bar (also called linear spring) element, which is illustrated in Figure 2.1(a). Perhaps the most complicated finite element (at least as regards number of degrees of freedom) is the curved, three-dimensional “brick” element depicted in Figure 2.1(b).

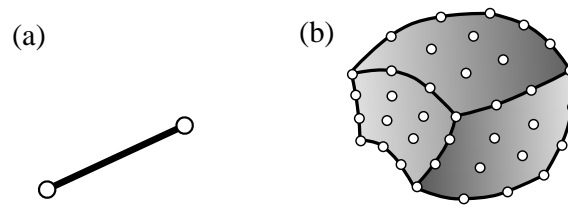


Figure 2.1. From the simplest to a highly complex structural finite element:
(a) 2-node bar element for trusses, (b) 64-node tricubic, curved “brick” element for three-dimensional solid analysis.

Yet the remarkable fact is that, in the DSM, the simplest and most complex elements are treated alike! To illustrate the basic steps of this democratic method, it makes educational sense to keep it simple and use a structure composed of bar elements. A simple yet nontrivial structure is the *pin-jointed plane truss*.¹

Using a plane truss to teach the stiffness method offers two additional advantages:

- (a) Computations can be entirely done by hand as long as the structure contains just a few elements. This allows various steps of the solution procedure to be carefully examined and understood before passing to the computer implementation. Doing hand computations on more complex finite element systems rapidly becomes impossible.
- (b) The computer implementation on any programming language is relatively simple and can be assigned as preparatory computer homework.

§2.2. TRUSS STRUCTURES

Plane trusses, such as the one depicted in Figure 2.2, are often used in construction, particularly for roofing of residential and commercial buildings, and in short-span bridges. Trusses, whether two or three dimensional, belong to the class of *skeletal structures*. These structures consist of elongated structural components called *members*, connected at *joints*. Another important subclass of skeletal structures are frame structures or *frameworks*, which are common in reinforced concrete construction of building and bridges.

¹ A one dimensional bar assembly would be even simpler. That kind of structure would not adequately illustrate some of the DSM steps, however, notably the back-and-forth transformations from global to local coordinates.

Skeletal structures can be analyzed by a variety of hand-oriented methods of structural analysis taught in beginning Mechanics of Materials courses: the Displacement and Force methods. They can also be analyzed by the computer-oriented FEM. That versatility makes those structures a good choice to illustrate the transition from the hand-calculation methods taught in undergraduate courses, to the fully automated finite element analysis procedures available in commercial programs.

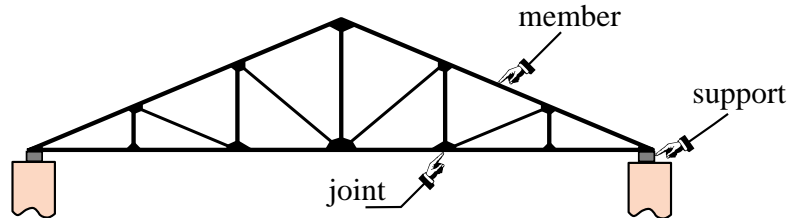


Figure 2.2. An actual plane truss structure. That shown is typical of a roof truss used in residential building construction.

In this and the following Chapter we will go over the basic steps of the DSM in a “hand-computer” calculation mode. This means that although the steps are done by hand, whenever there is a procedural choice we shall either adopt the way which is better suited towards the computer implementation, or explain the difference between hand and computer computations. The actual computer implementation using a high-level programming language is presented in Chapter 5.

To keep hand computations manageable in detail we use just about the simplest structure that can be called a plane truss, namely the three-member truss illustrated in Figure 2.3. The *idealized* model of the example truss as a pin-jointed assemblage of bars is shown in Figure 2.4(a), which also gives its geometric and material properties. In this idealization truss members carry only axial loads, have no bending resistance, and are connected by frictionless pins. Figure 2.4(b) displays support conditions as well as the applied forces applied to the truss joints.

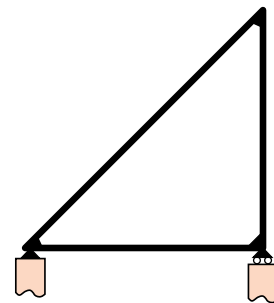


Figure 2.3. The three-member example truss.

It should be noted that as a practical structure the example truss is not particularly useful — the one depicted in Figure 2.2 is far more common in construction. But with the example truss we can go over the basic DSM steps without getting mired into too many members, joints and degrees of freedom.

§2.3. IDEALIZATION

Although the pin-jointed assemblage of bars (as depicted in Figure 2.4) is sometimes presented as an actual problem, it actually represents an *idealization* of a true truss structure. The axially-carrying members and frictionless pins of this structure are only an approximation of a real truss. For example, building and bridge trusses usually have members joined to each other through the use of gusset plates, which are attached by nails, bolts, rivets or welds. See Figure 2.2. Consequently members will carry some bending as well as direct axial loading.

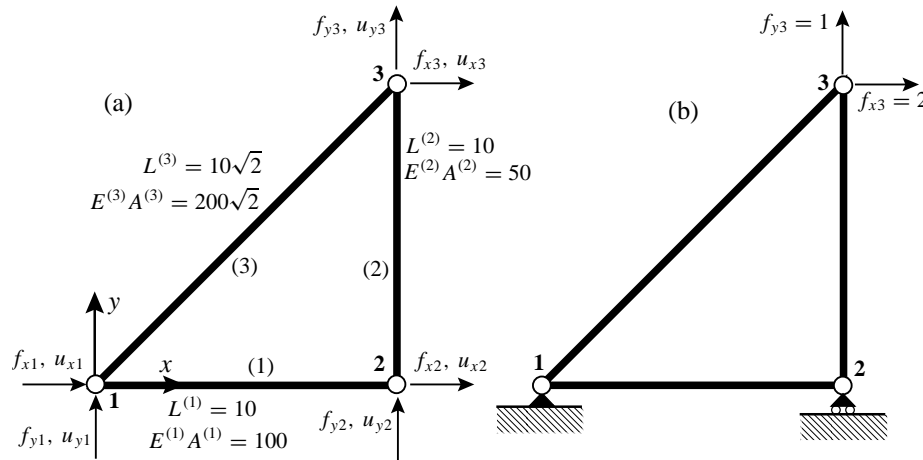


Figure 2.4. Pin-jointed idealization of example truss: (a) geometric and elastic properties, (b) support conditions and applied loads.

Experience has shown, however, that stresses and deformations calculated for the simple idealized problem will often be satisfactory for overall-design purposes; for example to select the cross section of the members. Hence the engineer turns to the pin-jointed assemblage of axial force elements and uses it to carry out the structural analysis.

This replacement of true by idealized is at the core of the *physical interpretation* of the finite element method discussed in §1.4.

§2.4. JOINT FORCES AND DISPLACEMENTS

The idealization of the example truss, pictured in Figure 2.4, has three *joints*, which are labeled 1, 2 and 3, and three *members*, which are labeled (1), (2) and (3). These members connect joints 1–2, 2–3, and 1–3, respectively. The member lengths are denoted by $L^{(1)}$, $L^{(2)}$ and $L^{(3)}$, their elastic moduli by $E^{(1)}$, $E^{(2)}$ and $E^{(3)}$, and their cross-sectional areas by $A^{(1)}$, $A^{(2)}$ and $A^{(3)}$. Both E and A are assumed to be constant along each member.

Members are generically identified by index e (because of their close relation to finite elements, see below), which is usually enclosed in parentheses to avoid confusion with exponents. For example, the cross-section area of a generic member is $A^{(e)}$. Joints are generically identified by indices such as i , j or n . In the general FEM, the name “joint” and “member” is replaced by *node* and *element*, respectively. The dual nomenclature is used in the initial Chapters to stress the physical interpretation of the FEM.

The geometry of the structure is referred to a common Cartesian coordinate system $\{x, y\}$, which is called the *global coordinate system*. Other names for it in the literature are *structure coordinate system* and *overall coordinate system*.

The key ingredients of the stiffness method of analysis are the *forces* and *displacements* at the joints.

In a idealized pin-jointed truss, externally applied forces as well as reactions *can act only at the joints*. All member axial forces can be characterized by the x and y components of these forces, which we call f_x and f_y , respectively. The components at joint i will be denoted as f_{xi} and f_{yi} , respectively. The set of all joint forces can be arranged as a 6-component column vector called \mathbf{f} .

The other key ingredient is the displacement field. Classical structural mechanics tells us that the displacements of the truss *are completely defined by the displacements of the joints*. This statement is a particular case of the more general finite element theory.

The x and y displacement components will be denoted by u_x and u_y , respectively. The *values* of u_x and u_y at joint i will be called u_{xi} and u_{yi} and, like the joint forces, they are arranged into a 6-component vector called \mathbf{u} . Here are the two vectors, shown side by side:

$$\mathbf{f} = \begin{bmatrix} f_{x1} \\ f_{y1} \\ f_{x2} \\ f_{y2} \\ f_{x3} \\ f_{y3} \end{bmatrix}, \quad \mathbf{u} = \begin{bmatrix} u_{x1} \\ u_{y1} \\ u_{x2} \\ u_{y2} \\ u_{x3} \\ u_{y3} \end{bmatrix}. \quad (2.1)$$

In the DSM these six displacements are the primary unknowns. They are also called the *degrees of freedom* or *state variables* of the system.²

How about the displacement boundary conditions, popularly called support conditions? This data will tell us which components of \mathbf{f} and \mathbf{u} are true unknowns and which ones are known *a priori*. In structural analysis procedures of the pre-computer era such information was used *immediately* by the analyst to discard unnecessary variables and thus reduce the amount of bookkeeping that had to be carried along by hand.

The computer oriented philosophy is radically different: *boundary conditions can wait until the last moment*. This may seem strange, but on the computer the sheer volume of data may not be so important as the efficiency with which the data is organized, accessed and processed. The strategy “save the boundary conditions for last” will be followed here for the hand computations.

§2.5. THE MASTER STIFFNESS EQUATIONS

The *master stiffness equations* relate the joint forces \mathbf{f} of the complete structure to the joint displacements \mathbf{u} of the complete structure *before* specification of support conditions.

Because the assumed behavior of the truss is linear, these equations must be linear relations that connect the components of the two vectors. Furthermore it will be assumed that if all displacements vanish, so do the forces.³

If both assumptions hold the relation must be homogeneous and be expressible in component form as follows:

$$\begin{bmatrix} f_{x1} \\ f_{y1} \\ f_{x2} \\ f_{y2} \\ f_{x3} \\ f_{y3} \end{bmatrix} = \begin{bmatrix} K_{x1x1} & K_{x1y1} & K_{x1x2} & K_{x1y2} & K_{x1x3} & K_{x1y3} \\ K_{y1x1} & K_{y1y1} & K_{y1x2} & K_{y1y2} & K_{y1x3} & K_{y1y3} \\ K_{x2x1} & K_{x2y1} & K_{x2x2} & K_{x2y2} & K_{x2x3} & K_{x2y3} \\ K_{y2x1} & K_{y2y1} & K_{y2x2} & K_{y2y2} & K_{y2x3} & K_{y2y3} \\ K_{x3x1} & K_{x3y1} & K_{x3x2} & K_{x3y2} & K_{x3x3} & K_{x3y3} \\ K_{y3x1} & K_{y3y1} & K_{y3x2} & K_{y3y2} & K_{y3x3} & K_{y3y3} \end{bmatrix} \begin{bmatrix} u_{x1} \\ u_{y1} \\ u_{x2} \\ u_{y2} \\ u_{x3} \\ u_{y3} \end{bmatrix}. \quad (2.2)$$

² *Primary unknowns* is the correct mathematical term whereas *degrees of freedom* has a mechanics flavor. The term *state variables* is used more often in nonlinear analysis.

³ This assumption implies that the so-called *initial strain* effects, also known as *prestress* or *initial stress* effects, are neglected. Such effects are produced by actions such as temperature changes or lack-of-fit fabrication, and are studied in Chapter 4.

In matrix notation:

$$\mathbf{f} = \mathbf{K}\mathbf{u}. \quad (2.3)$$

Here \mathbf{K} is the *master stiffness matrix*, also called *global stiffness matrix*, *assembled stiffness matrix*, or *overall stiffness matrix*. It is a 6×6 square matrix that happens to be symmetric, although this attribute has not been emphasized in the written-out form (2.2). The entries of the stiffness matrix are often called *stiffness coefficients* and have a physical interpretation discussed below.

The qualifiers (“master”, “global”, “assembled” and “overall”) convey the impression that there is another level of stiffness equations lurking underneath. And indeed there is a *member level* or *element level*, into which we plunge in the **Breakdown** section.

REMARK 2.1

Interpretation of Stiffness Coefficients. The following interpretation of the entries of \mathbf{K} is highly valuable for visualization and checking. Choose a displacement vector \mathbf{u} such that all components are zero except the i^{th} one, which is one. Then \mathbf{f} is simply the i^{th} column of \mathbf{K} . For instance if in (2.3) we choose u_{x2} as unit displacement,

$$\mathbf{u} = \begin{bmatrix} 0 \\ 0 \\ 1 \\ 0 \\ 0 \\ 0 \end{bmatrix}, \quad \mathbf{f} = \begin{bmatrix} K_{x1x2} \\ K_{y1x2} \\ K_{x2x2} \\ K_{y2x2} \\ K_{x3x2} \\ K_{y3x2} \end{bmatrix}. \quad (2.4)$$

Thus K_{y1x2} , say, represents the y-force at joint 1 that would arise on prescribing a unit x -displacement at joint 2, while all other displacements vanish.

In structural mechanics the property just noted is called *interpretation of stiffness coefficients as displacement influence coefficients*, and extends unchanged to the general finite element method.

§2.6. BREAKDOWN

The first three DSM steps are: (1) disconnection, (2) localization, and (3) computation of member stiffness equations. These are collectively called *breakdown steps* and are described below.

§2.6.1. Disconnection

To carry out the first step of the DSM we proceed to *disconnect* or *disassemble* the structure into its components, namely the three truss members. This step is illustrated in Figure 2.5.

To each member $e = 1, 2, 3$ is assigned a Cartesian system $\{\bar{x}^{(e)}, \bar{y}^{(e)}\}$. Axis $\bar{x}^{(e)}$ is aligned along the axis of the e^{th} member. See Figure 2.5. Actually $\bar{x}^{(e)}$ runs along the member longitudinal axis; it is shown offset in that Figure for clarity.

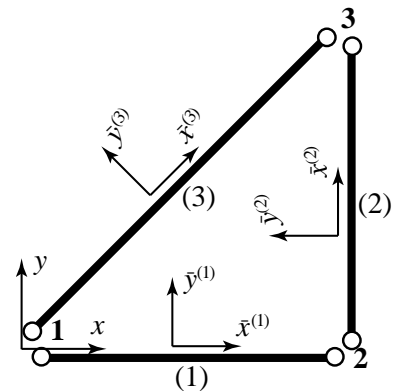


Figure 2.5. Breakdown of example truss into individual members (1), (2) and (3), and selection of local coordinate systems.

By convention the positive direction of $\bar{x}^{(e)}$ runs from joint i to joint j , where $i < j$. The angle formed by $\bar{x}^{(e)}$ and x is called $\varphi^{(e)}$. The axes origin is arbitrary and may be placed at the member midpoint or at one of the end joints for convenience.

These systems are called *local coordinate systems* or *member-attached coordinate systems*. In the general finite element method they receive the name *element coordinate systems*.

§2.6.2. Localization

Next, we drop the member identifier (e) so that we are effectively dealing with a *generic* truss member as illustrated in Figure 2.6. The local coordinate system is $\{\bar{x}, \bar{y}\}$. The two end joints are called i and j .

As shown in Figure 2.6, a generic truss member has four joint force components and four joint displacement components (the member degrees of freedom). The member properties include the length L , elastic modulus E and cross-section area A .

§2.6.3. Computation of Member Stiffness Equations

The force and displacement components of Figure 2.6(a) are linked by the *member stiffness relations*

$$\bar{\mathbf{f}} = \bar{\mathbf{K}} \bar{\mathbf{u}}, \quad (2.5)$$

which written out in full is

$$\begin{bmatrix} \bar{f}_{xi} \\ \bar{f}_{yi} \\ \bar{f}_{xj} \\ \bar{f}_{yj} \end{bmatrix} = \begin{bmatrix} \bar{K}_{xixi} & \bar{K}_{xiyi} & \bar{K}_{xixj} & \bar{K}_{xiyj} \\ \bar{K}_{yixi} & \bar{K}_{yiyi} & \bar{K}_{yixj} & \bar{K}_{yiyj} \\ \bar{K}_{xjxi} & \bar{K}_{xjyi} & \bar{K}_{xjxj} & \bar{K}_{xjyj} \\ \bar{K}_{yjxi} & \bar{K}_{yjyi} & \bar{K}_{yjxj} & \bar{K}_{yjyj} \end{bmatrix} \begin{bmatrix} \bar{u}_{xi} \\ \bar{u}_{yi} \\ \bar{u}_{xj} \\ \bar{u}_{yj} \end{bmatrix}. \quad (2.6)$$

Vectors $\bar{\mathbf{f}}$ and $\bar{\mathbf{u}}$ are called the *member joint forces* and *member joint displacements*, respectively, whereas $\bar{\mathbf{K}}$ is the *member stiffness matrix* or *local stiffness matrix*. When these relations are interpreted from the standpoint of the FEM, “member” is replaced by “element” and “joint” by “node.”

There are several ways to construct the stiffness matrix $\bar{\mathbf{K}}$ in terms of the element properties L , E and A . The most straightforward technique relies on the Mechanics of Materials approach covered in undergraduate courses. Think of the truss member in Figure 2.6(a) as a linear spring of equivalent stiffness k_s , an interpretation depicted in Figure 2.6(b). If the member properties are *uniform* along its length, Mechanics of Materials bar theory tells us that⁴

$$k_s = \frac{EA}{L}, \quad (2.7)$$

Consequently the force-displacement equation is

$$F = k_s d = \frac{EA}{L} d, \quad (2.8)$$

⁴ See for example, Chapter 2 of [2.1].

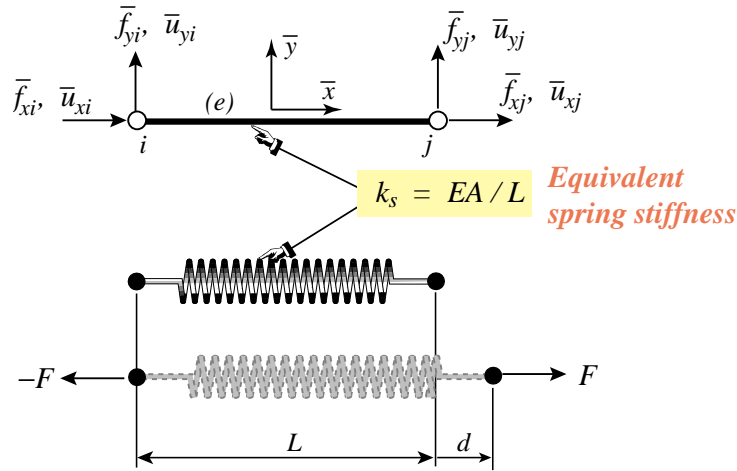


Figure 2.6. Generic truss member referred to its local coordinate system $\{\bar{x}, \bar{y}\}$:
(a) idealization as bar element, (b) interpretation as equivalent spring.

where F is the internal axial force and d the relative axial displacement, which physically is the bar elongation.

The axial force and elongation can be immediately expressed in terms of the joint forces and displacements as

$$F = \bar{f}_{xj} = -\bar{f}_{xi}, \quad d = \bar{u}_{xj} - \bar{u}_{xi}, \quad (2.9)$$

which express force equilibrium⁵ and kinematic compatibility, respectively.

Combining (2.8) and (2.9) we obtain the matrix relation⁶

$$\bar{\mathbf{f}} = \begin{bmatrix} \bar{f}_{xi} \\ \bar{f}_{yi} \\ \bar{f}_{xj} \\ \bar{f}_{yj} \end{bmatrix} = \frac{EA}{L} \begin{bmatrix} 1 & 0 & -1 & 0 \\ 0 & 0 & 0 & 0 \\ -1 & 0 & 1 & 0 \\ 0 & 0 & 0 & 0 \end{bmatrix} \begin{bmatrix} \bar{u}_{xi} \\ \bar{u}_{yi} \\ \bar{u}_{xj} \\ \bar{u}_{yj} \end{bmatrix} = \bar{\mathbf{K}} \bar{\mathbf{u}}, \quad (2.10)$$

Hence

$$\bar{\mathbf{K}} = \frac{EA}{L} \begin{bmatrix} 1 & 0 & -1 & 0 \\ 0 & 0 & 0 & 0 \\ -1 & 0 & 1 & 0 \\ 0 & 0 & 0 & 0 \end{bmatrix}. \quad (2.11)$$

This is the truss stiffness matrix in local coordinates.

Two other methods for obtaining the local force-displacement relation (2.8) are covered in Exercises 2.6 and 2.7.

In the following Chapter we will complete the main DSM steps by putting the truss back together and solving for the unknown forces and displacements.

⁵ Equations $F = \bar{f}_{xj} = -\bar{f}_{xi}$ follow by considering the free body diagram (FBD) of each joint. For example, take joint i as a FBD. Equilibrium along x requires $-F - \bar{f}_{xi} = 0$ whence $F = -\bar{f}_{xi}$. Doing this on joint j yields $F = \bar{f}_{xj}$.

⁶ The detailed derivation of (2.10) is the subject of Exercise 2.3.

Notes and Bibliography

The Direct Stiffness Method has been the dominant FEM version since the mid-1960s, and is the procedure followed by all major commercial production codes. It was invented and developed at Boeing in the 1950s, particularly through the leadership of Jon Turner [2.4–2.7]. All applications-oriented FEM books treat it, although the procedural steps are sometimes not clearly identified. In particular, the textbooks recommended in §1.7.5 contain adequate expositions.

Trusses, also called bar assemblies, are usually the first structures treated in Mechanics of Materials books written for undergraduate courses. Two widely used books at this level are Beer and Johnston [2.1] and Popov [2.2].

Steps in the derivation of stiffness matrices for truss elements are well covered in a number of early treatment of finite element books, of which Chapter 5 of Przemieniecki [2.3] is an good example.

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Homework Exercises for Chapter 2

The Direct Stiffness Method: Breakdown

EXERCISE 2.1

[D:5] Explain why *arbitrarily oriented* mechanical loads on an *idealized* pin-jointed truss structure must be applied at the joints. [Hint: idealized truss members have no bending resistance.] How about actual trusses: can they take loads applied between joints?

EXERCISE 2.2

[A:15] Show that the sum of the entries of each row of the master stiffness matrix \mathbf{K} of any plane truss, before application of any support conditions, must be zero. [Hint: think of translational rigid body modes.] Does the property hold also for the columns of that matrix?

EXERCISE 2.3

[A:15] Using matrix algebra derive (2.10) from (2.8) and (2.9).

EXERCISE 2.4

[A:15] By direct matrix multiplication verify that for the generic truss member $\bar{\mathbf{f}}^T \bar{\mathbf{u}} = F d$. Can you interpret this result physically? (Interpretation hint: look at (E2.3) below)

EXERCISE 2.5

[A:20] The transformation equations between the 1-DOF spring and the 4-DOF generic truss member may be written in compact matrix form as

$$d = \mathbf{T}_d \bar{\mathbf{u}}, \quad \bar{\mathbf{f}} = F \mathbf{T}_f, \quad (\text{E2.1})$$

where \mathbf{T}_d is 1×4 and \mathbf{T}_f is 4×1 . Starting from the identity $\bar{\mathbf{f}}^T \bar{\mathbf{u}} = F d$ proven in the previous exercise, and using *compact matrix notation*, show that $\mathbf{T}_f = \mathbf{T}_d^T$. Or in words: *the displacement transformation matrix and the force transformation matrix are the transpose of each other.* (This is a general result.)

EXERCISE 2.6

[A:20] Derive the equivalent spring formula $F = (EA/L) d$ of (2.8) by the Theory of Elasticity relations $e = d\bar{u}(\bar{x})/d\bar{x}$ (strain-displacement equation), $\sigma = Ee$ (Hooke's law) and $F = A\sigma$ (axial force definition). Here e is the axial strain (independent of \bar{x}) and σ the axial stress (also independent of \bar{x}). Finally, $\bar{u}(\bar{x})$ denotes the axial displacement of the cross section at a distance \bar{x} from node i , which is linearly interpolated as

$$\bar{u}(\bar{x}) = \bar{u}_{xi} \left(1 - \frac{\bar{x}}{L}\right) + \bar{u}_{xj} \frac{\bar{x}}{L} \quad (\text{E2.2})$$

Justify that (E2.2) is correct since the bar differential equilibrium equation: $d[A(d\sigma/d\bar{x})]/d\bar{x} = 0$, is verified for all \bar{x} if A is constant along the bar.

EXERCISE 2.7

[A:20] Derive the equivalent spring formula $F = (EA/L) d$ of (2.8) by the principle of Minimum Potential Energy (MPE). In Mechanics of Materials it is shown that the total potential energy of the axially loaded bar is

$$\Pi = \frac{1}{2} \int_0^L A \sigma e d\bar{x} - Fd, \quad (\text{E2.3})$$

where symbols have the same meaning as the previous Exercise. Use the displacement interpolation (E2.2), the strain-displacement equation $e = d\bar{u}/d\bar{x}$ and Hooke's law $\sigma = Ee$ to express Π as a function $\Pi(d)$ of the relative displacement d only. Then apply MPE by requiring that $\partial \Pi / \partial d = 0$.

3

The Direct Stiffness Method: Assembly and Solution

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§3.1. INTRODUCTION

Chapter 2 explained the breakdown of a truss structure into components called *members* or *elements*. Upon deriving the stiffness relations at the element level in terms of the local coordinate system, we are now ready to go back up to the original structure. This process is called *assembly*.

Assembly involves two substeps: *globalization*, through which the member stiffness equations are transformed back to the global coordinate system, and *merging* of those equations into the global stiffness equations. On the computer these steps are done concurrently, member by member. After all members are processed we have the *free-free master stiffness equations*.

Next comes the *solution*. This process also embodies two substeps: *application of boundary conditions* and *solution* for the unknown joint displacements. To apply the BCs, the free-free master stiffness equations are modified by taking into account which components of the joint displacements and forces are given and which are unknown.

The modified equations are submitted to a linear equation solver, which returns the unknown joint (node) displacements. As discussed under Notes and Bibliography (§3.6) on some FEM implementations, particularly in programs written in the 1960s and 1970s, one or more of the foregoing operations are done concurrently.

The solution step completes the DSM proper. *Postprocessing* steps may follow, in which derived quantities such as internal forces and stresses are recovered from the displacement solution. The various DSM steps are summarized in Figure 3.1 for the convenience of the reader.

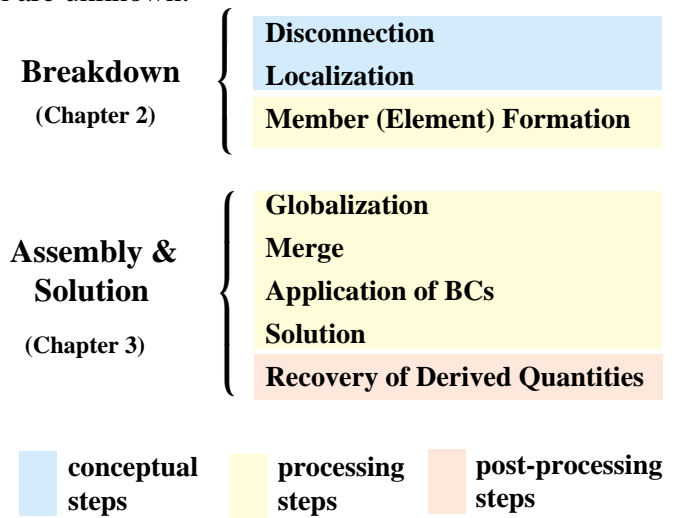


Figure 3.1. Steps of the Direct Stiffness Method (DSM).

§3.2. ASSEMBLY

§3.2.1. Coordinate Transformations

Before describing the globalization step, we must establish matrix relations that connect joint displacements and forces in the global and local coordinate systems. The necessary transformations are easily obtained by inspection of Figure 3.2. For the displacements

$$\begin{aligned}
 \bar{u}_{xi} &= u_{xi}c + u_{yi}s, & \bar{u}_{yi} &= -u_{xi}s + u_{yi}c, \\
 \bar{u}_{xj} &= u_{xj}c + u_{yj}s, & \bar{u}_{yj} &= -u_{xj}s + u_{yj}c,
 \end{aligned} \tag{3.1}$$

where $c = \cos \varphi$, $s = \sin \varphi$ and φ is the angle formed by \bar{x} and x , measured positive counterclockwise from x . The matrix form of this relation is

$$\begin{bmatrix} \bar{u}_{xi} \\ \bar{u}_{yi} \\ \bar{u}_{xj} \\ \bar{u}_{yj} \end{bmatrix} = \begin{bmatrix} c & s & 0 & 0 \\ -s & c & 0 & 0 \\ 0 & 0 & c & s \\ 0 & 0 & -s & c \end{bmatrix} \begin{bmatrix} u_{xi} \\ u_{yi} \\ u_{xj} \\ u_{yj} \end{bmatrix}. \tag{3.2}$$

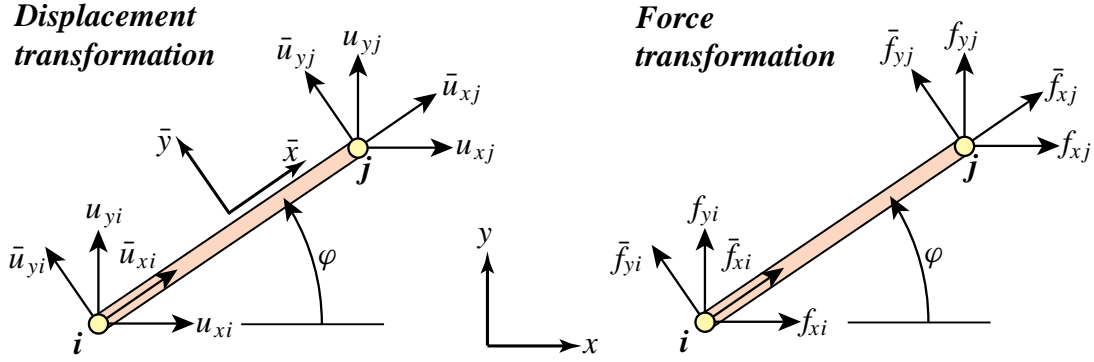


Figure 3.2. The transformation of node displacement and force components from the local system $\{\bar{x}, \bar{y}\}$ to the global system $\{x, y\}$.

The 4×4 matrix that appears above is called a *displacement transformation matrix* and is denoted¹ by \mathbf{T} . The node forces transform as $f_{xi} = \bar{f}_{xi}c - \bar{f}_{yi}s$, etc., which in matrix form become

$$\begin{bmatrix} f_{xi} \\ f_{yi} \\ f_{xj} \\ f_{yj} \end{bmatrix} = \begin{bmatrix} c & -s & 0 & 0 \\ s & c & 0 & 0 \\ 0 & 0 & c & -s \\ 0 & 0 & s & c \end{bmatrix} \begin{bmatrix} \bar{f}_{xi} \\ \bar{f}_{yi} \\ \bar{f}_{xj} \\ \bar{f}_{yj} \end{bmatrix}. \quad (3.3)$$

The 4×4 matrix that appears above is called a *force transformation matrix*. A comparison of (3.2) and (3.3) reveals that the force transformation matrix is the *transpose* \mathbf{T}^T of the displacement transformation matrix \mathbf{T} . This relation is not accidental and can be proved to hold generally.²

REMARK 3.1

Note that in (3.2) the local system (barred) quantities appear on the left-hand side, whereas in (3.3) they show up on the right-hand side. The expressions (3.2) and (3.3) are discrete counterparts of what are called covariant and contravariant transformations, respectively, in continuum mechanics. The counterpart of the transposition relation is the *adjointness* property.

REMARK 3.2

For this particular structural element \mathbf{T} is square and orthogonal, that is, $\mathbf{T}^T = \mathbf{T}^{-1}$. But this property does not extend to more general elements. Furthermore in the general case \mathbf{T} is not even a square matrix, and does not possess an ordinary inverse. However the congruential transformation relations (3.4)–(3.6) do hold generally.

§3.2.2. Globalization

From now on we reintroduce the member (element) index, e . The member stiffness equations in global coordinates will be written

$$\mathbf{f}^{(e)} = \mathbf{K}^{(e)} \mathbf{u}^{(e)}. \quad (3.4)$$

¹ This matrix will be called \mathbf{T}_d when its association with displacements is to be emphasized, as in Exercise 2.5.

² A simple proof that relies on the invariance of external work is given in Exercise 2.5. However this invariance was only checked by explicit computation for a truss member in Exercise 2.4. The general proof relies on the Principle of Virtual Work, which is discussed later.

The compact form of (3.2) and (3.3) for the e^{th} member is

$$\bar{\mathbf{u}}^{(e)} = \mathbf{T}^{(e)} \mathbf{u}^{(e)}, \quad \mathbf{f}^{(e)} = (\mathbf{T}^{(e)})^T \bar{\mathbf{f}}^{(e)}. \quad (3.5)$$

Inserting these matrix expressions into (2.5): $\bar{\mathbf{f}}^{(e)} = \bar{\mathbf{K}}^{(e)} \bar{\mathbf{u}}^{(e)}$ and comparing with (3.4) we find that the member stiffness in the global system $\{x, y\}$ can be computed from the member stiffness $\bar{\mathbf{K}}^{(e)}$ in the local system $\{\bar{x}, \bar{y}\}$ through the congruential transformation

$$\boxed{\mathbf{K}^{(e)} = (\mathbf{T}^{(e)})^T \bar{\mathbf{K}}^{(e)} \mathbf{T}^{(e)}}. \quad (3.6)$$

Carrying out the matrix multiplications in full we get

$$\mathbf{K}^{(e)} = \frac{E^{(e)} A^{(e)}}{L^{(e)}} \begin{bmatrix} c^2 & sc & -c^2 & -sc \\ sc & s^2 & -sc & -s^2 \\ -c^2 & -sc & c^2 & sc \\ -sc & -s^2 & sc & s^2 \end{bmatrix}, \quad (3.7)$$

in which $c = \cos \varphi^{(e)}$, $s = \sin \varphi^{(e)}$, with superscripts of c and s suppressed to reduce clutter. If the angle is zero we recover (2.10), as may be expected. $\mathbf{K}^{(e)}$ is called a *member stiffness matrix in global coordinates*. The proof of (3.6) and verification of (3.7) is left as Exercise 3.1.

The globalized member stiffness matrices for the example truss can now be easily obtained by inserting appropriate values into (3.7). For member (1), with end joints 1–2, angle $\varphi = 0^\circ$ and the member data listed in Figure 2.4(a) we get

$$\begin{bmatrix} f_{x1}^{(1)} \\ f_{y1}^{(1)} \\ f_{x2}^{(1)} \\ f_{y2}^{(1)} \end{bmatrix} = 10 \begin{bmatrix} 1 & 0 & -1 & 0 \\ 0 & 0 & 0 & 0 \\ -1 & 0 & 1 & 0 \\ 0 & 0 & 0 & 0 \end{bmatrix} \begin{bmatrix} u_{x1}^{(1)} \\ u_{y1}^{(1)} \\ u_{x2}^{(1)} \\ u_{y2}^{(1)} \end{bmatrix}. \quad (3.8)$$

For member (2), with end joints 2–3, and angle $\varphi = 90^\circ$:

$$\begin{bmatrix} f_{x2}^{(2)} \\ f_{y2}^{(2)} \\ f_{x3}^{(2)} \\ f_{y3}^{(2)} \end{bmatrix} = 5 \begin{bmatrix} 0 & 0 & 0 & 0 \\ 0 & 1 & 0 & -1 \\ 0 & 0 & 0 & 0 \\ 0 & -1 & 0 & 1 \end{bmatrix} \begin{bmatrix} u_{x2}^{(2)} \\ u_{y2}^{(2)} \\ u_{x3}^{(2)} \\ u_{y3}^{(2)} \end{bmatrix}. \quad (3.9)$$

Finally, for member (3), with end joints 1–3, and angle $\varphi = 45^\circ$:

$$\begin{bmatrix} f_{x1}^{(3)} \\ f_{y1}^{(3)} \\ f_{x3}^{(3)} \\ f_{y3}^{(3)} \end{bmatrix} = 20 \begin{bmatrix} 0.5 & 0.5 & -0.5 & -0.5 \\ 0.5 & 0.5 & -0.5 & -0.5 \\ -0.5 & -0.5 & 0.5 & 0.5 \\ -0.5 & -0.5 & 0.5 & 0.5 \end{bmatrix} \begin{bmatrix} u_{x1}^{(3)} \\ u_{y1}^{(3)} \\ u_{x3}^{(3)} \\ u_{y3}^{(3)} \end{bmatrix}. \quad (3.10)$$

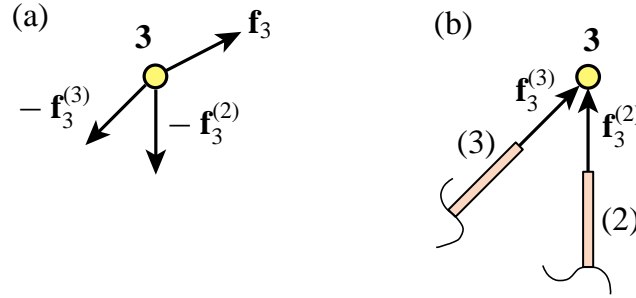


Figure 3.3. The force equilibrium of joint 3 of the example truss, depicted as a free body diagram in (a). Here \mathbf{f}_3 is the known external joint force applied on the joint. Joint forces $\mathbf{f}_3^{(2)}$ and $\mathbf{f}_3^{(3)}$ are applied *by the joint on the members*, as illustrated in (b). Consequently the forces applied *by the members on the joint* are $-\mathbf{f}_3^{(2)}$ and $-\mathbf{f}_3^{(3)}$. These forces would act in the directions shown if both members (2) and (3) were in tension. The free-body equilibrium statement is $\mathbf{f}_3 - \mathbf{f}_3^{(2)} - \mathbf{f}_3^{(3)} = \mathbf{0}$ or $\mathbf{f}_3 = \mathbf{f}_3^{(2)} + \mathbf{f}_3^{(3)}$. This translates into the two component equations: $f_{x3} = f_{x3}^{(2)} + f_{x3}^{(3)}$ and $f_{y3} = f_{y3}^{(2)} + f_{y3}^{(3)}$, of (3.11).

§3.2.3. Assembly Rules

The key operation of the assembly process is the “placement” of the contribution of each member to the master stiffness equations. The process is technically called *merging* of individual members. The merge operation can be physically interpreted as reconnecting that member in the process of fabricating the complete structure. For a truss structure, reconnection means inserting the pins back into the joints. Merge is mathematically governed by two rules of structural mechanics:

1. *Compatibility of displacements:* The joint displacements of all members meeting at a joint are the same.
2. *Force equilibrium:* The sum of forces exerted by all members that meet at a joint balances the external force applied to that joint.

The first rule is physically obvious: reconnected joints must move as one entity. The second one can be visualized by considering a joint as a free body, although care is required in the interpretation of joint forces and their signs. Notational conventions to this effect are explained in Figure 3.3 for joint 3 of the example truss, at which members (2) and (3) meet. Application of the foregoing rules at this particular joint gives

$$\text{Rule 1: } u_{x3}^{(2)} = u_{x3}^{(3)}, \quad u_{y3}^{(2)} = u_{y3}^{(3)}.$$

$$\text{Rule 2: } f_{x3} = f_{x3}^{(2)} + f_{x3}^{(3)} = f_{x3}^{(1)} + f_{x3}^{(2)} + f_{x3}^{(3)}, \quad f_{y3} = f_{y3}^{(2)} + f_{y3}^{(3)} = f_{y3}^{(1)} + f_{y3}^{(2)} + f_{y3}^{(3)}. \quad (3.11)$$

The addition of $f_{x3}^{(1)}$ and $f_{y3}^{(1)}$ to $f_{x3}^{(2)} + f_{x3}^{(3)}$ and $f_{y3}^{(2)} + f_{y3}^{(3)}$, respectively, changes nothing because member (1) is not connected to joint 3; we are simply adding zeros. But this augmentation enables us to write the matrix relation: $\mathbf{f} = \mathbf{f}^{(1)} + \mathbf{f}^{(2)} + \mathbf{f}^{(3)}$, which will be used in (3.19).

§3.2.4. Hand Assembly by Augmentation and Merge

To directly visualize how the two assembly rules translate to member merging rules, we first *augment* the member stiffness relations by adding zero rows and columns as appropriate to *complete* the force and displacement vectors.

For member (1):

$$\begin{bmatrix} f_{x1}^{(1)} \\ f_{y1}^{(1)} \\ f_{x2}^{(1)} \\ f_{y2}^{(1)} \\ f_{x3}^{(1)} \\ f_{y3}^{(1)} \end{bmatrix} = \begin{bmatrix} 10 & 0 & -10 & 0 & 0 & 0 \\ 0 & 0 & 0 & 0 & 0 & 0 \\ -10 & 0 & 10 & 0 & 0 & 0 \\ 0 & 0 & 0 & 0 & 0 & 0 \\ 0 & 0 & 0 & 0 & 0 & 0 \\ 0 & 0 & 0 & 0 & 0 & 0 \end{bmatrix} \begin{bmatrix} u_{x1}^{(1)} \\ u_{y1}^{(1)} \\ u_{x2}^{(1)} \\ u_{y2}^{(1)} \\ u_{x3}^{(1)} \\ u_{y3}^{(1)} \end{bmatrix}. \quad (3.12)$$

For member (2):

$$\begin{bmatrix} f_{x1}^{(2)} \\ f_{y1}^{(2)} \\ f_{x2}^{(2)} \\ f_{y2}^{(2)} \\ f_{x3}^{(2)} \\ f_{y3}^{(2)} \end{bmatrix} = \begin{bmatrix} 0 & 0 & 0 & 0 & 0 & 0 \\ 0 & 0 & 0 & 0 & 0 & 0 \\ 0 & 0 & 0 & 0 & 0 & 0 \\ 0 & 0 & 0 & 5 & 0 & -5 \\ 0 & 0 & 0 & 0 & 0 & 0 \\ 0 & 0 & 0 & -5 & 0 & 5 \end{bmatrix} \begin{bmatrix} u_{x1}^{(2)} \\ u_{y1}^{(2)} \\ u_{x2}^{(2)} \\ u_{y2}^{(2)} \\ u_{x3}^{(2)} \\ u_{y3}^{(2)} \end{bmatrix}. \quad (3.13)$$

For member (3):

$$\begin{bmatrix} f_{x1}^{(3)} \\ f_{y1}^{(3)} \\ f_{x2}^{(3)} \\ f_{y2}^{(3)} \\ f_{x3}^{(3)} \\ f_{y3}^{(3)} \end{bmatrix} = \begin{bmatrix} 10 & 10 & 0 & 0 & -10 & -10 \\ 10 & 10 & 0 & 0 & -10 & -10 \\ 0 & 0 & 0 & 0 & 0 & 0 \\ 0 & 0 & 0 & 0 & 0 & 0 \\ -10 & -10 & 0 & 0 & 10 & 10 \\ -10 & -10 & 0 & 0 & 10 & 10 \end{bmatrix} \begin{bmatrix} u_{x1}^{(3)} \\ u_{y1}^{(3)} \\ u_{x2}^{(3)} \\ u_{y2}^{(3)} \\ u_{x3}^{(3)} \\ u_{y3}^{(3)} \end{bmatrix}. \quad (3.14)$$

According to the first rule, we can *drop the member identifier* in the displacement vectors that appear in the foregoing matrix equations. Hence the reconnected member equations are

$$\begin{bmatrix} f_{x1}^{(1)} \\ f_{y1}^{(1)} \\ f_{x2}^{(1)} \\ f_{y2}^{(1)} \\ f_{x3}^{(1)} \\ f_{y3}^{(1)} \end{bmatrix} = \begin{bmatrix} 10 & 0 & -10 & 0 & 0 & 0 \\ 0 & 0 & 0 & 0 & 0 & 0 \\ -10 & 0 & 10 & 0 & 0 & 0 \\ 0 & 0 & 0 & 0 & 0 & 0 \\ 0 & 0 & 0 & 0 & 0 & 0 \\ 0 & 0 & 0 & 0 & 0 & 0 \end{bmatrix} \begin{bmatrix} u_{x1} \\ u_{y1} \\ u_{x2} \\ u_{y2} \\ u_{x3} \\ u_{y3} \end{bmatrix}, \quad (3.15)$$

$$\begin{bmatrix} f_{x1}^{(2)} \\ f_{y1}^{(2)} \\ f_{x2}^{(2)} \\ f_{y2}^{(2)} \\ f_{x3}^{(2)} \\ f_{y3}^{(2)} \end{bmatrix} = \begin{bmatrix} 0 & 0 & 0 & 0 & 0 & 0 \\ 0 & 0 & 0 & 0 & 0 & 0 \\ 0 & 0 & 0 & 0 & 0 & 0 \\ 0 & 0 & 0 & 5 & 0 & -5 \\ 0 & 0 & 0 & 0 & 0 & 0 \\ 0 & 0 & 0 & -5 & 0 & 5 \end{bmatrix} \begin{bmatrix} u_{x1} \\ u_{y1} \\ u_{x2} \\ u_{y2} \\ u_{x3} \\ u_{y3} \end{bmatrix}, \quad (3.16)$$

$$\begin{bmatrix} f_{x1}^{(3)} \\ f_{y1}^{(3)} \\ f_{x2}^{(3)} \\ f_{y2}^{(3)} \\ f_{x3}^{(3)} \\ f_{y3}^{(3)} \end{bmatrix} = \begin{bmatrix} 10 & 10 & 0 & 0 & -10 & -10 \\ 10 & 10 & 0 & 0 & -10 & -10 \\ 0 & 0 & 0 & 0 & 0 & 0 \\ 0 & 0 & 0 & 0 & 0 & 0 \\ -10 & -10 & 0 & 0 & 10 & 10 \\ -10 & -10 & 0 & 0 & 10 & 10 \end{bmatrix} \begin{bmatrix} u_{x1} \\ u_{y1} \\ u_{x2} \\ u_{y2} \\ u_{x3} \\ u_{y3} \end{bmatrix}. \quad (3.17)$$

These three equations can be represented in direct matrix notation as

$$\mathbf{f}^{(1)} = \mathbf{K}^{(1)} \mathbf{u}, \quad \mathbf{f}^{(2)} = \mathbf{K}^{(2)} \mathbf{u}, \quad \mathbf{f}^{(3)} = \mathbf{K}^{(3)} \mathbf{u}. \quad (3.18)$$

According to the second rule

$$\mathbf{f} = \mathbf{f}^{(1)} + \mathbf{f}^{(2)} + \mathbf{f}^{(3)} = (\mathbf{K}^{(1)} + \mathbf{K}^{(2)} + \mathbf{K}^{(3)}) \mathbf{u} = \mathbf{K} \mathbf{u}, \quad (3.19)$$

so all we have to do is add the three stiffness matrices that appear above, and we arrive at the master stiffness equations:

$$\begin{bmatrix} f_{x1} \\ f_{y1} \\ f_{x2} \\ f_{y2} \\ f_{x3} \\ f_{y3} \end{bmatrix} = \begin{bmatrix} 20 & 10 & -10 & 0 & -10 & -10 \\ 10 & 10 & 0 & 0 & -10 & -10 \\ -10 & 0 & 10 & 0 & 0 & 0 \\ 0 & 0 & 0 & 5 & 0 & -5 \\ -10 & -10 & 0 & 0 & 10 & 10 \\ -10 & -10 & 0 & -5 & 10 & 15 \end{bmatrix} \begin{bmatrix} u_{x1} \\ u_{y1} \\ u_{x2} \\ u_{y2} \\ u_{x3} \\ u_{y3} \end{bmatrix}. \quad (3.20)$$

Using this technique *member merging* becomes simply *matrix addition*.

This explanation of the assembly process is conceptually the easiest to follow and understand. It is virtually foolproof for hand computations. However, this is *not* the way the process is carried out on the computer because it would be enormously wasteful of storage for large systems. A computer-oriented procedure is discussed in §3.5.

§3.3. SOLUTION

Having formed the master stiffness equations we can proceed to the solution phase. To prepare the equations for an equation solver we need to separate known and unknown components of \mathbf{f} and \mathbf{u} . In this Section a technique suitable for hand computation is described.

§3.3.1. Applying Displacement BCs by Reduction

If one attempts to solve the system (3.20) numerically for the displacements, surprise! The solution “blows up” because the coefficient matrix (the master stiffness matrix) is singular. The mathematical interpretation of this behavior is that rows and columns of \mathbf{K} are linear combinations of each other (see Remark 3.4 below). The physical interpretation of singularity is that there are unsuppressed *rigid body motions*: the truss still “floats” in the $\{x, y\}$ plane.

To eliminate rigid body motions and render the system nonsingular we must apply the *support conditions* or *displacement boundary conditions*. From Figure 2.4(b) we observe that the support conditions for the example truss are

$$u_{x1} = u_{y1} = u_{y2} = 0, \quad (3.21)$$

whereas the known applied forces are

$$f_{x2} = 0, \quad f_{x3} = 2, \quad f_{y3} = 1. \quad (3.22)$$

When solving the stiffness equations by hand, the simplest way to account for support conditions is to *remove* equations associated with known joint displacements from the master system. To apply (3.21) we have to remove equations 1, 2 and 4. This can be systematically accomplished by *deleting* or “striking out” rows and columns number 1, 2 and 4 from \mathbf{K} and the corresponding components from \mathbf{f} and \mathbf{u} . The reduced three-equation system is

$$\begin{bmatrix} 10 & 0 & 0 \\ 0 & 10 & 10 \\ 0 & 10 & 15 \end{bmatrix} \begin{bmatrix} u_{x2} \\ u_{x3} \\ u_{y3} \end{bmatrix} = \begin{bmatrix} f_{x2} \\ f_{x3} \\ f_{y3} \end{bmatrix} = \begin{bmatrix} 0 \\ 2 \\ 1 \end{bmatrix}. \quad (3.23)$$

Equation (3.23) is called the *reduced master stiffness system*. The coefficient matrix of this system is no longer singular.

REMARK 3.3

In mathematical terms, the free-free master stiffness matrix \mathbf{K} in (3.20) has order $N = 6$, rank $r = 3$ and a rank deficiency of $d = N - r = 6 - 3 = 3$ (these concepts are summarized in Appendix C.) The dimension of the null space of \mathbf{K} is $d = 3$. This space is spanned by three independent rigid body motions: the two rigid translations along x and y and the rigid rotation about z .

REMARK 3.4

Conditions (3.21) represent the simplest type of support conditions, namely specified zero displacements. Chapters 4, 9 and 10 discuss how more general constraint forms, such as prescribed nonzero displacements and multifreedom constraints, are handled.

§3.3.2. Solving for Displacements

Solving the reduced system by hand (for example, via Gauss elimination) yields

$$\begin{bmatrix} u_{x2} \\ u_{x3} \\ u_{y3} \end{bmatrix} = \begin{bmatrix} 0 \\ 0.4 \\ -0.2 \end{bmatrix}. \quad (3.24)$$

This is called a *partial displacement solution* (also *reduced displacement solution*) because it excludes suppressed displacement components. This solution vector is *expanded* to six components by including the three specified values (3.21):

$$\mathbf{u} = \begin{bmatrix} u_{x1} \\ u_{y1} \\ u_{x2} \\ u_{y2} \\ u_{x3} \\ u_{y3} \end{bmatrix} = \begin{bmatrix} 0 \\ 0 \\ 0 \\ 0 \\ 0.4 \\ -0.2 \end{bmatrix}. \quad (3.25)$$

This is called the *complete displacement solution*, or simply the *displacement solution*.

§3.4. POSTPROCESSING

The last major processing step of the DSM is the solution for joint displacements. But often the analyst needs information on other mechanical quantities; for example the reaction forces at the supports, or the internal member forces. Such quantities are said to be *derived* because they are *recovered* from the displacement solution. The recovery of derived quantities is part of the so-called *postprocessing steps* of the DSM. Two such steps are described below.

§3.4.1. Recovery of Reaction Forces

Premultiplying the complete displacement solution (3.25) by \mathbf{K} we get

$$\mathbf{f} = \mathbf{Ku} = \begin{bmatrix} 20 & 10 & -10 & 0 & -10 & -10 \\ 10 & 10 & 0 & 0 & -10 & -10 \\ -10 & 0 & 10 & 0 & 0 & 0 \\ 0 & 0 & 0 & 5 & 0 & -5 \\ -10 & -10 & 0 & 0 & 10 & 10 \\ -10 & -10 & 0 & -5 & 10 & 15 \end{bmatrix} \begin{bmatrix} 0 \\ 0 \\ 0 \\ 0 \\ 0.4 \\ -0.2 \end{bmatrix} = \begin{bmatrix} -2 \\ -2 \\ 0 \\ 1 \\ 2 \\ 1 \end{bmatrix} \quad (3.26)$$

This vector recovers the known applied forces (3.22) as can be expected. Furthermore we get three *reaction forces*: $f_{x1} = f_{y1} = -2$ and $f_{y2} = 1$, which are associated with the support conditions (3.21). It is easy to check that the complete force system is in equilibrium; this is the topic of Exercise 3.2.

§3.4.2. Recovery of Internal Forces and Stresses

Frequently the structural engineer is not primarily interested in displacements but in *internal forces* and *stresses*. These are in fact the most important quantities for preliminary structural design.

In trusses the only internal forces are the *axial member forces*. These are pictured in Figure 3.4. These forces are denoted by $p^{(1)}$, $p^{(2)}$ and $p^{(3)}$ and collected in a vector \mathbf{p} . The average axial stress $\sigma^{(e)}$ is easily obtained on dividing $p^{(e)}$ by the cross-sectional area of the member.

The axial force $p^{(e)}$ in member (e) can be obtained as follows. Extract the displacements of member (e) from the displacement solution \mathbf{u} to form $\mathbf{u}^{(e)}$. Then recover local joint displacements from $\bar{\mathbf{u}}^{(e)} = \mathbf{T}^{(e)} \mathbf{u}^{(e)}$.

Compute the deformation d (relative displacement) and recover the axial force from the equivalent spring constitutive relation:

$$d^{(e)} = \bar{u}_{xj}^{(e)} - \bar{u}_{xi}^{(e)}, \quad p^{(e)} = \frac{E^{(e)} A^{(e)}}{L^{(e)}} d^{(e)}. \quad (3.27)$$

An alternative interpretation of (3.27) is to regard $e^{(e)} = d^{(e)}/L^{(e)}$ as the (average) member axial strain, $\sigma^{(e)} = E^{(e)} e^{(e)}$ as (average) axial stress, and $p^{(e)} = A^{(e)} \sigma^{(e)}$ as the axial force. This is more in tune with the Theory of Elasticity viewpoint discussed in Exercise 2.6.

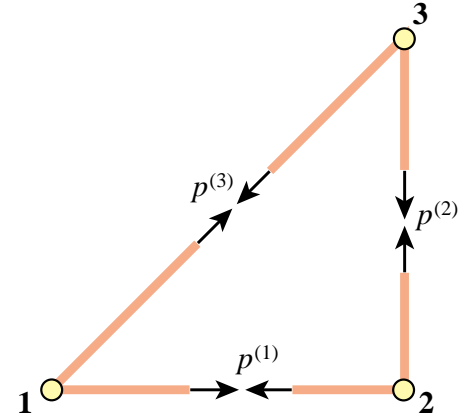


Figure 3.4. The internal forces in the example truss are the axial forces $p^{(1)}$, $p^{(2)}$ and $p^{(3)}$ in the members. Directions shown pertain to tension.

§3.5. *COMPUTER ORIENTED ASSEMBLY AND SOLUTION

§3.5.1. *Assembly by Freedom Pointers

The practical computer implementation of the DSM assembly process departs significantly from the “augment and add” technique described in §3.2.4. There are two major differences:

- (I) Member stiffness matrices are *not* expanded. Their entries are directly merged into those of \mathbf{K} through the use of a “freedom pointer array” called the *Element Freedom Table* or EFT.
- (II) The master stiffness matrix \mathbf{K} is stored using a special format that takes advantage of symmetry and sparseness.

Difference (II) is a more advanced topic that is deferred to the last part of the book. For simplicity we shall assume here that \mathbf{K} is stored as a full square matrix, and study only (I). For the example truss the freedom-pointer technique expresses the entries of \mathbf{K} as the sum

$$K_{pq} = \sum_{e=1}^3 K_{ij}^{(e)} \quad \text{for } i = 1, \dots, 4, \quad j = 1, \dots, 4, \quad p = \text{EFT}^{(e)}(i), \quad q = \text{EFT}^{(e)}(j). \quad (3.28)$$

Here $K_{ij}^{(e)}$ denote the entries of the 4×4 globalized member stiffness matrices in (3.8) through (3.10). Entries K_{pq} that do not get any contributions from the right hand side remain zero. $\text{EFT}^{(e)}$ denotes the Element Freedom Table for member (e) . For the example truss these tables are

$$\text{EFT}^{(1)} = \{1, 2, 3, 4\}, \quad \text{EFT}^{(2)} = \{3, 4, 5, 6\}, \quad \text{EFT}^{(3)} = \{1, 2, 5, 6\}. \quad (3.29)$$

Physically these tables map local freedom indices to global ones. For example, freedom number 3 of member (2) is u_{x3} , which is number 5 in the master equations; consequently $\text{EFT}^{(2)}(3) = 5$. Note that (3.28) involves three nested loops: over e (outermost), over i , and over j . The ordering of the last two is irrelevant. Advantage may be taken of the symmetry of $\mathbf{K}^{(e)}$ and \mathbf{K} to roughly halve the number of additions. Exercise 3.6 follows the scheme (3.28) by hand.

§3.5.2. *Applying Displacement BCs by Modification

In §3.3.1 the support conditions (3.21) were applied by reducing (3.20) to (3.23). Reduction is convenient for hand computations because it cuts down on the number of equations to solve. But it has a serious flaw for computer implementation: the equations must be rearranged. It was previously noted that on the computer the number of equations is not the only important consideration. Rearrangement can be as or more expensive than solving the equations, particularly if the coefficient matrix is stored in sparse form or in secondary storage.

To apply support conditions without rearranging the equations we clear (set to zero) rows and columns corresponding to prescribed zero displacements as well as the corresponding force components, and place ones on the diagonal to maintain non-singularity. The resulting system is called the *modified* set of master stiffness equations. For the example truss this approach yields

$$\begin{bmatrix} 1 & 0 & 0 & 0 & 0 & 0 \\ 0 & 1 & 0 & 0 & 0 & 0 \\ 0 & 0 & 10 & 0 & 0 & 0 \\ 0 & 0 & 0 & 1 & 0 & 0 \\ 0 & 0 & 0 & 0 & 10 & 10 \\ 0 & 0 & 0 & 0 & 10 & 15 \end{bmatrix} \begin{bmatrix} u_{x1} \\ u_{y1} \\ u_{x2} \\ u_{y2} \\ u_{x3} \\ u_{y3} \end{bmatrix} = \begin{bmatrix} 0 \\ 0 \\ 0 \\ 0 \\ 2 \\ 1 \end{bmatrix}, \quad (3.30)$$

in which rows and columns for equations 1, 2 and 4 have been cleared. Solving this modified system yields the complete displacement solution (3.25).

REMARK 3.5

In a “smart” stiffness equation solver the modified system need not be explicitly constructed by storing zeros and ones. It is sufficient to *mark* the equations that correspond to displacement BCs. The solver is then programmed to skip those equations. However, if one is using a standard solver from, say, a library of scientific routines or a commercial program such as *Matlab* or *Mathematica*, such intelligence cannot be expected, and the modified system must be set up explicitly.

Notes and Bibliography

The coverage of the assembly and solution steps of the DSM, along with globalization and application of BCs, is not uniform across the wide spectrum of FEM books. Authors have introduced little “quirks” although the overall concepts are not affected. The most common variations arise in two contexts:

- (1) Some treatments apply support conditions *before merge*, explicitly eliminating known displacement freedoms as the elements are processed and merged into **K**. The output of the assembly process is what is called here a reduced stiffness matrix.³
- (2) In the *frontal solution method* of Irons [3.1,3.2], assembly and solution are done concurrently. More precisely, as elements are formed and merged, displacement boundary conditions are applied, and Gauss elimination and reduction of the right hand side starts once the assembler senses (by tracking an “element wavefront”) that no more elements contribute to a certain node.

Both variants appeared in FEM programs written during the 1960s and 1970s. They were motivated by computer resource limitations of the time: memory was scarce⁴ and computing time expensive. On the negative side, interweaving steps leads to unmodular code (which may easily become “spaghetti code” in low-level languages such as Fortran). On present computers those storage and time savings are of less significance. Modularity, which simplifies scripting in higher order languages such as *Matlab*, is currently desirable because

³ For the example truss, the coefficient matrix in (3.23) is a reduced stiffness whereas that in (3.30) is a modified one.

⁴ Which required heavy use of slower secondary storage such as disk, drum or tape.

it increases “plug-in” operational flexibility and reduces the chance for errors. These two attributes respond to economic reality: human time is nowadays far more expensive than computer time.

Both the hand-oriented and computer-oriented application of boundary conditions have been presented here, although the latter is still considered an advanced topic. While hand computations become unfeasible once the FEM model gets beyond a few elements, they are important from an instructional standpoint.

References

- [3.1] Irons, B. M., A frontal solution program for finite element analysis, *Int. J. Numer. Meth. Engrg.*, **12**, 5–32, 1970.
- [3.2] Irons, B. M., Ahmad, S., *Techniques of Finite Elements*, Ellis Horwood Ltd, Chichester, UK, 1980.

Homework Exercises for Chapter 3

The Direct Stiffness Method: Assembly and Solution

EXERCISE 3.1

[A:20] Derive (3.6) from $\bar{\mathbf{K}}^{(e)} \bar{\mathbf{u}}^{(e)} = \bar{\mathbf{f}}^{(e)}$, (3.4) and (3.6). (*Hint*: premultiply both sides of $\bar{\mathbf{K}}^{(e)} \bar{\mathbf{u}}^{(e)} = \bar{\mathbf{f}}^{(e)}$ by an appropriate matrix). Then check by hand that using that formula you get (3.7). Use Falk's scheme for the multiplications.⁵

EXERCISE 3.2

[A:15] Draw a free body diagram of the nodal forces (3.26) acting on the free-free truss structure, and verify that this force system satisfies translational and rotational (moment) equilibrium.

EXERCISE 3.3

[A:15] Using the method presented in §3.4.2 compute the axial forces in the three members of the example truss. Partial answer: $p^{(3)} = 2\sqrt{2}$.

EXERCISE 3.4

[A:20] Describe an alternative method that recovers the axial member forces of the example truss from consideration of joint equilibrium, without going through the computation of member deformations.

EXERCISE 3.5

[A:20] Suppose that the third support condition in (3.21) is $u_{x2} = 0$ instead of $u_{y2} = 0$. Rederive the reduced system (3.23) for this case. Verify that this system cannot be solved for the joint displacements u_{y2} , u_{x3} and u_{y3} because the reduced stiffness matrix is singular.⁶ Offer a physical interpretation of this failure.

EXERCISE 3.6

[N:20] Construct by hand the free-free master stiffness matrix of (3.20) using the freedom-pointer technique (3.28). Note: start from \mathbf{K} initialized to the null matrix, then cycle over $e = 1, 2, 3$.

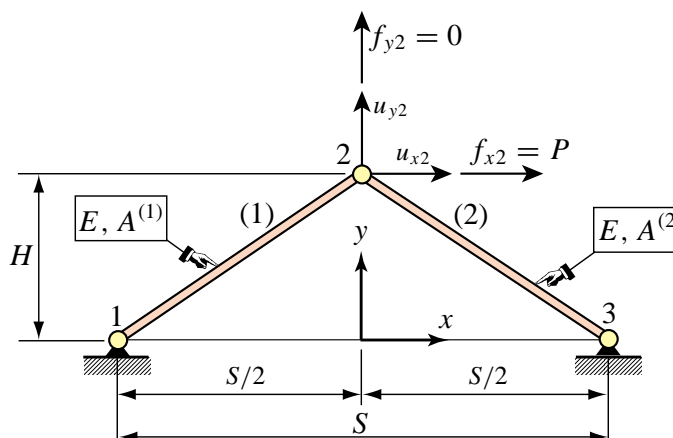


Figure E3.1. Truss structure for Exercise 3.7.

⁵ This scheme is recommended to do matrix multiplication by hand. It is explained in §B.3.2 of Appendix B.

⁶ A matrix is singular if its determinant is zero; cf. §C.2 of Appendix C for a “refresher” in that topic.

EXERCISE 3.7

[N:25] Consider the two-member arch-truss structure shown in Figure E3.1. Take span $S = 8$, height $H = 3$, elastic modulus $E = 1000$, cross section areas $A^{(1)} = 2$ and $A^{(2)} = 4$, and horizontal crown force $P = f_{x2} = 12$. Using the DSM carry out the following steps:

- (a) Assemble the master stiffness equations. Any method: augment-and-add, or the more advanced “freedom pointer” technique explained in §3.5.1, is acceptable.
- (b) Apply the displacement BCs and solve the reduced system for the crown displacements u_{x2} and u_{y2} . Partial result: $u_{x2} = 9/512 = 0.01758$.
- (c) Recover the node forces at all joints including reactions. Verify that overall force equilibrium (x forces, y forces, and moments about any point) is satisfied.
- (d) Recover the axial forces in the two members. Result should be $p^{(1)} = -p^{(2)} = 15/2$.

EXERCISE 3.8

[D:15] Why are disconnection and localization labeled as “conceptual steps” in Figure 3.1? Hint: do they have to be actually programmed?

4

The Direct Stiffness Method: Additional Topics

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Chapters 2 and 3 outlined the “core” steps of the Direct Stiffness Method (DSM). Those steps were illustrated with the hand analysis of a plane truss structure. This Chapter covers some topics that were left out from Chapters 2–3 for clarity. These include: the imposition of prescribed nonzero displacements, and the treatment of thermal effects.

§4.1. PRESCRIBED NONZERO DISPLACEMENTS

The support conditions considered in the example truss of Chapters 2–3 resulted in the specification of zero displacement components; for example $u_{y2} = 0$. There are cases, however, where the known value is nonzero. This happens, for example, in the study of settlement of foundations of ground structures such as buildings and bridges, and in the analysis of motion-driven machinery components. Mathematically these are called non-homogenous boundary conditions. The treatment of this generalization of the FEM equations is studied in the following subsections.

§4.1.1. Application of DBCs by Reduction

We describe first a reduction technique, analogous to that covered in §3.2.1, which is suitable for hand computations. Recall the master stiffness equations (3.20) for the example truss:

$$\begin{bmatrix} 20 & 10 & -10 & 0 & -10 & -10 \\ 10 & 10 & 0 & 0 & -10 & -10 \\ -10 & 0 & 10 & 0 & 0 & 0 \\ 0 & 0 & 0 & 5 & 0 & -5 \\ -10 & -10 & 0 & 0 & 10 & 10 \\ -10 & -10 & 0 & -5 & 10 & 15 \end{bmatrix} \begin{bmatrix} u_{x1} \\ u_{y1} \\ u_{x2} \\ u_{y2} \\ u_{x3} \\ u_{y3} \end{bmatrix} = \begin{bmatrix} f_{x1} \\ f_{y1} \\ f_{x2} \\ f_{y2} \\ f_{x3} \\ f_{y3} \end{bmatrix} \quad (4.1)$$

Suppose that the applied forces are as for the example truss but the prescribed displacements are

$$u_{x1} = 0, \quad u_{y1} = -0.5, \quad u_{y2} = 0.4 \quad (4.2)$$

This means that joint 1 goes down vertically whereas joint 2 goes up vertically, as depicted in Figure 4.1. Inserting the known data into (4.1) we get

$$\begin{bmatrix} 20 & 10 & -10 & 0 & -10 & -10 \\ 10 & 10 & 0 & 0 & -10 & -10 \\ -10 & 0 & 10 & 0 & 0 & 0 \\ 0 & 0 & 0 & 5 & 0 & -5 \\ -10 & -10 & 0 & 0 & 10 & 10 \\ -10 & -10 & 0 & -5 & 10 & 15 \end{bmatrix} \begin{bmatrix} 0 \\ -0.5 \\ u_{x2} \\ 0.4 \\ u_{x3} \\ u_{y3} \end{bmatrix} = \begin{bmatrix} f_{x1} \\ f_{y1} \\ 0 \\ f_{y2} \\ 2 \\ 1 \end{bmatrix} \quad (4.3)$$

The first, second and fourth rows of (4.3) are removed, leaving only

$$\begin{bmatrix} -10 & 0 & 10 & 0 & 0 & 0 \\ -10 & -10 & 0 & 0 & 10 & 10 \\ -10 & -10 & 0 & -5 & 10 & 15 \end{bmatrix} \begin{bmatrix} 0 \\ -0.5 \\ u_{x2} \\ 0.4 \\ u_{x3} \\ u_{y3} \end{bmatrix} = \begin{bmatrix} 0 \\ 2 \\ 1 \end{bmatrix} \quad (4.4)$$

Columns 1, 2 and 4 are removed by transferring all known terms from the left to the right hand side:

$$\begin{bmatrix} 10 & 0 & 0 \\ 0 & 10 & 10 \\ 0 & 10 & 15 \end{bmatrix} \begin{bmatrix} u_{x2} \\ u_{x3} \\ u_{y3} \end{bmatrix} = \begin{bmatrix} 0 \\ 2 \\ 1 \end{bmatrix} - \begin{bmatrix} (-10) \times 0 + 0 \times (-0.5) + 0 \times 0.4 \\ (-10) \times 0 + (-10) \times (-0.5) + 0 \times 0.4 \\ (-10) \times 0 + (-10) \times (-0.5) + (-5) \times 0.4 \end{bmatrix} = \begin{bmatrix} 0 \\ -3 \\ -2 \end{bmatrix}. \quad (4.5)$$

These are the *reduced stiffness equations*. Note that its coefficient matrix of (4.5) is exactly the same as in the reduced system (3.23) for prescribed zero displacements.

The right hand side, however, is different. It consists of the applied joint forces *modified by the effect of known nonzero displacements*. They are called the *modified node forces* or *effective node forces*. Solving the reduced system yields

$$\begin{bmatrix} u_{x2} \\ u_{x3} \\ u_{y3} \end{bmatrix} = \begin{bmatrix} 0 \\ -0.5 \\ 0.2 \end{bmatrix}. \quad (4.6)$$

Filling the missing entries with the known values (4.2) yields the complete displacement solution (listed as row vector to save space):

$$\mathbf{u} = [0 \quad -0.5 \quad 0 \quad 0.4 \quad -0.5 \quad 0.2]^T. \quad (4.7)$$

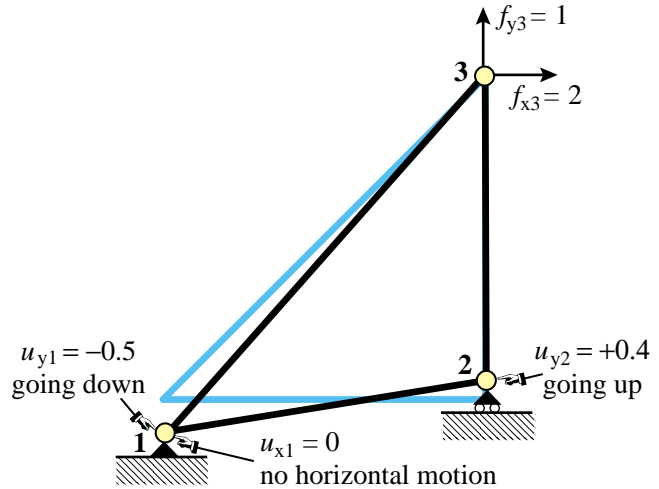


Figure 4.1. The example truss with prescribed nonzero vertical displacements at joints 1 and 2.

Taking the solution (4.7) and going through the postprocessing steps discussed in §3.3, we can find that the reaction forces and the internal member forces do not change. This is a consequence of the fact that the example truss is *statically determinate*. The force systems (internal and external) in such structures are insensitive to movements such as foundation settlements.

§4.1.2. *Application of DBCs by Modification

The computer-oriented modification approach follows the same idea outlined in §3.5.2. As there, the main objective is to avoid rearranging the master stiffness equations. To understand the process it is useful to think of being done in two stages. First equations 1, 2 and 4 are modified so that they become trivial equations, as illustrated for the example truss and the support conditions (4.2):

$$\begin{bmatrix} 1 & 0 & 0 & 0 & 0 & 0 \\ 0 & 1 & 0 & 0 & 0 & 0 \\ -10 & 0 & 10 & 0 & 0 & 0 \\ 0 & 0 & 0 & 1 & 0 & 0 \\ -10 & -10 & 0 & 0 & 10 & 10 \\ -10 & -10 & 0 & -5 & 10 & 15 \end{bmatrix} \begin{bmatrix} u_{x1} \\ u_{x2} \\ u_{x3} \\ u_{y1} \\ u_{y2} \\ u_{y3} \end{bmatrix} = \begin{bmatrix} 0 \\ -0.5 \\ 0 \\ 0.4 \\ 2 \\ 1 \end{bmatrix} \quad (4.8)$$

The solution of this system recovers (4.2) by construction (for example, the fourth equation is simply $1 \times u_{y2} = 0.4$). In the next stage, columns 1, 2 and 4 of the coefficient matrix are cleared by transferring all known terms

to the right hand side, following the same procedure explained in (4.5). We thus arrive at

$$\begin{bmatrix} 1 & 0 & 0 & 0 & 0 & 0 \\ 0 & 1 & 0 & 0 & 0 & 0 \\ 0 & 0 & 10 & 0 & 0 & 0 \\ 0 & 0 & 0 & 1 & 0 & 0 \\ 0 & 0 & 0 & 0 & 10 & 10 \\ 0 & 0 & 0 & 0 & 10 & 15 \end{bmatrix} \begin{bmatrix} u_{x1} \\ u_{x2} \\ u_{x2} \\ u_{y2} \\ u_{x3} \\ u_{y3} \end{bmatrix} = \begin{bmatrix} 0 \\ -0.5 \\ 0 \\ 0.4 \\ -3 \\ -2 \end{bmatrix} \quad (4.9)$$

As before, these are called the *modified master stiffness equations*. Observe that the equations retain the original number and order. Solving this system yields the complete displacement solution (4.7).

Note that if all prescribed displacements are zero, forces on the right hand side are not modified, and one would recover (3.30).

REMARK 4.1

The modification is not actually programmed as discussed above. First the applied forces in the right-hand side are modified for the effect of nonzero prescribed displacements, and the prescribed displacements stored in the reaction-force slots. This is called the *force modification* step. Second, rows and columns of the stiffness matrix are cleared as appropriate and ones stored in the diagonal positions. This is called the *stiffness modification* step. It is essential that the procedural steps be executed in the indicated order, because stiffness terms must be used to modify forces before they are cleared.

§4.1.3. *Matrix Forms of DBC Application Methods

The reduction and modification techniques for applying DBCs can be presented in compact matrix form. First, the free-free master stiffness equations $\mathbf{K}\mathbf{u} = \mathbf{f}$ are partitioned as follows:

$$\begin{bmatrix} \mathbf{K}_{11} & \mathbf{K}_{12} \\ \mathbf{K}_{21} & \mathbf{K}_{22} \end{bmatrix} \begin{bmatrix} \mathbf{u}_1 \\ \mathbf{u}_2 \end{bmatrix} = \begin{bmatrix} \mathbf{f}_1 \\ \mathbf{f}_2 \end{bmatrix}. \quad (4.10)$$

In this matrix equation, subvectors \mathbf{u}_2 and \mathbf{f}_1 collect displacement and force components, respectively, that are *known, given or prescribed*. Subvectors \mathbf{u}_1 and \mathbf{f}_2 collect force and displacement components, respectively, that are *unknown*. Forces in \mathbf{f}_2 represent reactions on supports; consequently \mathbf{f}_2 is called the *reaction vector*.

On transferring the known terms to the right hand side the first matrix equation becomes

$$\mathbf{K}_{11}\mathbf{u}_1 = \mathbf{f}_1 - \mathbf{K}_{12}\mathbf{u}_2. \quad (4.11)$$

This is the *reduced master equation system*. If the support B.C.s are homogeneous (that is, all prescribed displacements are zero), $\mathbf{u}_2 = \mathbf{0}$, and we do not need to change the right-hand side:

$$\mathbf{K}_{11}\mathbf{u}_1 = \mathbf{f}_1. \quad (4.12)$$

Examples that illustrate (4.11) and (4.12) are (4.5) and (3.23), respectively.

The computer-oriented modification technique retains the same joint displacement vector as in (4.10) through the following rearrangement:

$$\begin{bmatrix} \mathbf{K}_{11} & \mathbf{0} \\ \mathbf{0} & \mathbf{I} \end{bmatrix} \begin{bmatrix} \mathbf{u}_1 \\ \mathbf{u}_2 \end{bmatrix} = \begin{bmatrix} \mathbf{f}_1 - \mathbf{K}_{12}\mathbf{u}_2 \\ \mathbf{u}_2 \end{bmatrix}. \quad (4.13)$$

This *modified system* is simply the reduced equation (4.11) augmented by the trivial equation $\mathbf{I}\mathbf{u}_2 = \mathbf{u}_2$. This system is often denoted as

$$\hat{\mathbf{K}}\mathbf{u} = \hat{\mathbf{f}}. \quad (4.14)$$

Solving (4.14) yields the complete displacement solution including the specified displacements \mathbf{u}_2 .

For the computer implementation it is important to note that the partitioned form (4.10) is only used to permit the use of compact matrix notation. In actual programming the equations are *not* explicitly rearranged: they retain their original numbers. For instance, in the example truss

$$\mathbf{u}_1 = \begin{bmatrix} u_{x1} \\ u_{y1} \\ u_{y2} \end{bmatrix} \equiv \begin{bmatrix} \text{DOF \#1} \\ \text{DOF \#2} \\ \text{DOF \#4} \end{bmatrix}, \quad \mathbf{u}_2 = \begin{bmatrix} u_{x2} \\ u_{x3} \\ u_{y3} \end{bmatrix} \equiv \begin{bmatrix} \text{DOF \#3} \\ \text{DOF \#5} \\ \text{DOF \#6} \end{bmatrix}. \quad (4.15)$$

The example shows that \mathbf{u}_1 and \mathbf{u}_2 are generally interspersed throughout \mathbf{u} . Thus, matrix operations such as $\mathbf{K}_{12}\mathbf{u}_2$ involve indirect (pointer) addressing so as to avoid explicit array rearrangement.

§4.2. THERMOMECHANICAL EFFECTS

The assumptions invoked in Chapters 2-3 for the example truss result in zero external forces under zero displacements. This is implicit in the linear-homogeneous expression of the master stiffness equation $\mathbf{f} = \mathbf{K}\mathbf{u}$. If \mathbf{u} vanishes, so does \mathbf{f} . This behavior does not apply, however, if there are *initial force effects*.¹ If those effects are present, there can be displacements without external forces, and internal forces without displacements.

A common source of initial force effects are temperature changes. Imagine that a plane truss structure is *unloaded* (that is, not subjected to external forces) and is held at a *uniform reference temperature*. External displacements are measured from this environment, which is technically called a *reference state*. Now suppose that the temperature of some members changes with respect to the reference temperature while the applied external forces remain zero. Because the length of members changes on account of thermal expansion or contraction, the joints will displace. If the structure is statically indeterminate those displacements will induce strains and stresses and thus internal forces. These are distinguished from mechanical effects by the qualifier “thermal.” For many structures, particularly in aerospace and mechanical engineering, such effects have to be considered in the analysis and design.

There are other physical sources of initial force effects, such as moisture (hygrosteric) effects,² member prestress, residual stresses, or lack of fit. For linear structural models *all* such sources may be algebraically treated in the same way as thermal effects. The treatment results in an *initial force* vector that has to be added to the applied mechanical forces. This subject is outlined in §4.3 from a general perspective. However, to describe the main features of the matrix analysis procedure it is sufficient to consider the case of temperature changes.

In this Section we go over the analysis of a plane truss structure whose members undergo temperature changes from a reference state. It is assumed that the disconnection and localization steps of the DSM have been carried out. Therefore we begin with the derivation of the matrix stiffness equations of a generic truss member.

¹ Called *initial stress* or *initial strain* effects by many authors. The different names reflect what is viewed as the physical source of initial force effects at the continuum mechanics level.

² These are important in composite materials and geomechanics.

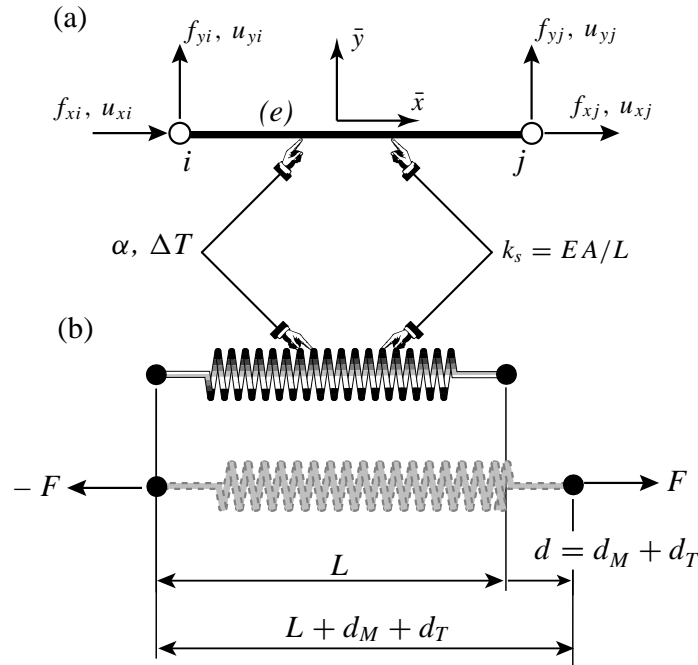


Figure 4.2. Generic truss member subjected to mechanical and thermal effects:
(a) idealization as bar, (b) idealization as equivalent linear spring.

§4.2.1. Thermomechanical Behavior

Consider the generic plane-truss member shown in Figure 4.2. The member is prismatic and uniform. The temperature T is also uniform. To reduce clutter the member identification subscript will be omitted in the following development until the globalization and assembly steps.

We introduce the concept of *reference temperature* T_{ref} . This is conventionally chosen to be the temperature throughout the structure at which the displacements, strains and stresses are zero if no mechanical forces are applied. In structures such as buildings and bridges T_{ref} is often taken to be the mean temperature during the construction period. Those zero displacements, strains and stresses, together with T_{ref} , define the *thermomechanical reference state* for the structure.

The member temperature variation from that reference state is $\Delta T = T - T_{ref}$. This may be positive or negative. If the member is *disassembled* or *disconnected*, under this variation the member length is free to change from L to $L + d_T$. If the thermoelastic constitutive behavior is linear³ then d_T is proportional to L and ΔT :

$$d_T = \alpha L \Delta T. \quad (4.16)$$

Here α is the coefficient of thermal expansion, which has physical units of one over temperature. This coefficient will be assumed to be uniform over the generic member. It may, however, vary from member to member. The *thermal strain* is defined as

$$e_T = d_T/L = \alpha \Delta T. \quad (4.17)$$

³ An assumption justified if the temperature changes are small enough so that α is approximately constant through the range of interest, and no material phase change effects occur.

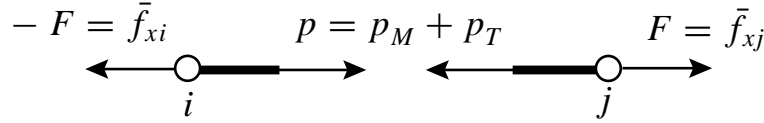


Figure 4.3. Equilibrium of truss member under thermomechanical forces.

Now suppose that the member is also subject to mechanical forces, more precisely the applied axial force F shown in Figure 4.2. The member axial stress is $\sigma = F/A$. In response to this stress the length changes by d_M . The *mechanical strain* is $e_M = d_M/L$. The total strain $e = d/L = (d_M + d_T)/L$ is the sum of the mechanical and the thermal strains:

$$e = e_M + e_T = \frac{\sigma}{E} + \alpha \Delta T \quad (4.18)$$

This superposition of deformations is the basic assumption made in the thermomechanical analysis. It is physically obvious for an unconstrained member such as that depicted in Figure 4.2.

At the other extreme, suppose that the member is completely blocked against axial elongation; that is, $d = 0$ but $\Delta T \neq 0$. Then $e = 0$ and $e_M = -e_T$. If $\alpha > 0$ and $\Delta T > 0$ the blocked member goes in *compression* because $\sigma = Ee_M = -Ee_T = -E\alpha \Delta T < 0$. This *thermal stress* is further discussed in Remark 4.2.

§4.2.2. Thermomechanical Stiffness Relations

Because $e = d/L$ and $d = \bar{u}_{xj} - \bar{u}_{xi}$, (4.18) can be developed as

$$\frac{\bar{u}_{xj} - \bar{u}_{xi}}{L} = \frac{\sigma}{E} + \alpha \Delta T, \quad (4.19)$$

To pass to internal forces (4.19) is multiplied through by EA :

$$\frac{EA}{L}(\bar{u}_{xj} - \bar{u}_{xi}) = A\sigma + EA\alpha \Delta T = p_M + p_T = p = F. \quad (4.20)$$

Here $p_M = A\sigma$ denotes the mechanical axial force, and $p_T = EA\alpha \Delta T$, which has the dimension of a force, is called (not surprisingly) the *internal thermal force*. The sum $p = p_M + p_T$ is called the *effective internal force*. The last relation in (4.20), $F = p = p_M + p_T$ follows from free-body member equilibrium; see Figure 4.3. Passing to matrix form:

$$F = \frac{EA}{L} \begin{bmatrix} -1 & 0 & 1 & 0 \end{bmatrix} \begin{bmatrix} \bar{u}_{xi} \\ \bar{u}_{yi} \\ \bar{u}_{xj} \\ \bar{u}_{yj} \end{bmatrix}. \quad (4.21)$$

Noting that $F = \bar{f}_{xj} = -\bar{f}_{xi}$ while $\bar{f}_{yi} = \bar{f}_{yj} = 0$, we can relate joint forces to joint displacements as

$$\begin{bmatrix} \bar{f}_{xi} \\ \bar{f}_{yi} \\ \bar{f}_{xj} \\ \bar{f}_{yj} \end{bmatrix} = \begin{bmatrix} -F \\ 0 \\ F \\ 0 \end{bmatrix} = \begin{bmatrix} \bar{f}_{Mxi} \\ \bar{f}_{Myi} \\ \bar{f}_{Mxj} \\ \bar{f}_{Myj} \end{bmatrix} + EA\alpha\Delta T \begin{bmatrix} -1 \\ 0 \\ 1 \\ 0 \end{bmatrix} = \frac{EA}{L} \begin{bmatrix} 1 & 0 & -1 & 0 \\ 0 & 0 & 0 & 0 \\ -1 & 0 & 1 & 0 \\ 0 & 0 & 0 & 0 \end{bmatrix} \begin{bmatrix} \bar{u}_{xi} \\ \bar{u}_{yi} \\ \bar{u}_{xj} \\ \bar{u}_{yj} \end{bmatrix}. \quad (4.22)$$

In compact matrix form this is $\bar{\mathbf{f}} = \bar{\mathbf{f}}_M + \bar{\mathbf{f}}_T = \bar{\mathbf{K}}\bar{\mathbf{u}}$, or

$$\boxed{\bar{\mathbf{K}}\bar{\mathbf{u}} = \bar{\mathbf{f}}_M + \bar{\mathbf{f}}_T.} \quad (4.23)$$

Here $\bar{\mathbf{K}}$ is the same member stiffness matrix derived in §2.6.3. The new ingredient that appears is the vector

$$\bar{\mathbf{f}}_T = EA\alpha\Delta T \begin{bmatrix} -1 \\ 0 \\ 1 \\ 0 \end{bmatrix}, \quad (4.24)$$

This is called the vector of *thermal joint forces* in local coordinates. It is an instance of an *initial force* vector at the element level.

REMARK 4.2

A useful physical interpretation of (4.23) is as follows. Suppose that the member is precluded from joint (node) motions so that $\bar{\mathbf{u}} = \mathbf{0}$. Then $\bar{\mathbf{f}}_M + \bar{\mathbf{f}}_T = \mathbf{0}$ or $\bar{\mathbf{f}}_M = -\bar{\mathbf{f}}_T$. It follows that $\bar{\mathbf{f}}_T$ contains the negated joint forces (internal forces) that develop in a heated or cooled bar if joint motions are precluded. Because for most materials $\alpha > 0$, rising the temperature of a blocked bar: $\Delta T > 0$, produces an internal compressive thermal force $p_T = A\sigma_T = -EA\alpha\Delta T$, in accordance with the expected physics. The quantity $\sigma_T = -E\alpha\Delta T$ is the *thermal stress*. This stress can cause buckling or cracking in severely heated structural members that are not allowed to expand or contract. This motivates the use of expansion joints in pavements, buildings and rails, and roller supports in long bridges.

§4.2.3. Globalization

At this point we restore the member superscript so that the member stiffness equations(4.22) are rewritten as

$$\bar{\mathbf{K}}^{(e)}\bar{\mathbf{u}}^{(e)} = \bar{\mathbf{f}}_M^{(e)} + \bar{\mathbf{f}}_T^{(e)}. \quad (4.25)$$

Use of the transformation rules developed in §3.1 to change displacements and forces to the global system $\{x, y\}$ yields

$$\mathbf{K}^{(e)}\mathbf{u}^{(e)} = \mathbf{f}_M^{(e)} + \mathbf{f}_T^{(e)}, \quad (4.26)$$

where $\mathbf{T}^{(e)}$ is the displacement transformation matrix (3.1), and the transformed quantities are

$$\mathbf{K}^{(e)} = (\mathbf{T}^{(e)})^T \bar{\mathbf{K}}^{(e)} \mathbf{T}^{(e)}, \quad \mathbf{f}_M^{(e)} = (\mathbf{T}^{(e)})^T \bar{\mathbf{f}}_M^{(e)}, \quad \mathbf{f}_T^{(e)} = (\mathbf{T}^{(e)})^T \bar{\mathbf{f}}_T^{(e)}. \quad (4.27)$$

These globalized member equations are used to assemble the free-free master stiffness equations by a member merging process.

§4.2.4. Merge

The merge process is based on the same assembly rules stated in §3.1.3 with only one difference: thermal forces are added to the right hand side. The member by member merge is carried out much as described as in §3.1.4, the main difference being that the thermal force vectors $\mathbf{f}_T^{(e)}$ are also merged into a master thermal force vector. Force merge can be done by augmentation-and-add (for hand work) or via freedom pointers (for computer work). Illustrative examples are provided below. Upon completion of the assembly process we arrive at the free-free master stiffness equations

$$\mathbf{Ku} = \mathbf{f}_M + \mathbf{f}_T = \mathbf{f}. \quad (4.28)$$

§4.2.5. Solution

The master system (4.28) has formally the same configuration as the master stiffness equations (2.3). The only difference is that the *effective* joint force vector \mathbf{f} contains a superposition of mechanical and thermal forces. Displacement boundary conditions can be applied by reduction or modification of these equations, simply by using effective joint forces in the descriptions of §3.2.1, §3.4.1 and §4.1. Processing the reduced or modified system by a linear equation solver yields the displacement solution \mathbf{u} .

§4.2.6. Postprocessing

The postprocessing steps described in §3.4 require some modifications because the derived quantities of interest to the structural engineer are *mechanical* reaction forces and internal forces. Effective forces by themselves are of little use in design. Mechanical joint forces including reactions are recovered from

$$\mathbf{f}_M = \mathbf{Ku} - \mathbf{f}_T \quad (4.29)$$

To recover mechanical internal forces in member (e), compute $p^{(e)}$ by the procedure outlined in §3.4.2, and subtract the thermal component:

$$p_M^{(e)} = p^{(e)} - E^{(e)} A^{(e)} \alpha^{(e)} \Delta T^{(e)}. \quad (4.30)$$

This equation comes from solving (4.20) for p_M . The mechanical axial stress is $\sigma^{(e)} = p_M^{(e)} / A^{(e)}$.

§4.2.7. Worked-Out Example 1

The first worked out problem is defined in Figure 4.4. Two truss members are connected in series as shown and fixed at the ends. Properties $E = 1000$, $A = 5$ and $\alpha = 0.0005$ are common to both members. The member lengths are 4 and 6. A mechanical load $P = 90$ acts on the roller node. The temperature of member (1) increases by $\Delta T^{(1)} = 25^\circ$ while that of member (2) drops by $\Delta T^{(2)} = -10^\circ$. Find the stress in both members.

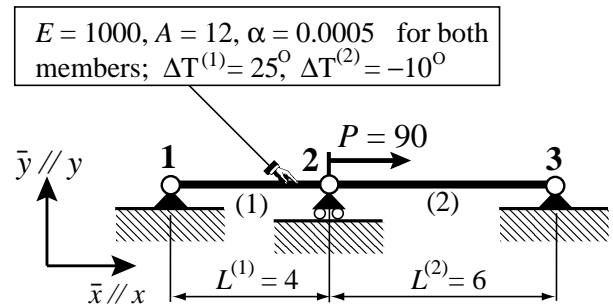


Figure 4.4. Structure for worked-out Example 1.

To reduce clutter note that all y motions are suppressed so only the x freedoms are kept: $u_{x1} = u_1$, $u_{x2} = u_2$ and $u_{x3} = u_3$. The corresponding node forces are denoted by $f_{x1} = f_1$, $f_{x2} = f_2$ and $f_{x3} = f_3$. The thermal force vectors, stripped to their $\bar{x} \equiv x$ components, are

$$\bar{\mathbf{f}}_T^{(1)} = \begin{bmatrix} \bar{f}_{T1}^{(1)} \\ \bar{f}_{T2}^{(1)} \end{bmatrix} = E^{(1)} A^{(1)} \alpha^{(1)} \Delta T^{(1)} \begin{bmatrix} -1 \\ 1 \end{bmatrix} = \begin{bmatrix} -150 \\ 150 \end{bmatrix}, \quad \bar{\mathbf{f}}_T^{(2)} = \begin{bmatrix} \bar{f}_{T2}^{(2)} \\ \bar{f}_{T3}^{(2)} \end{bmatrix} = E^{(2)} A^{(2)} \alpha^{(2)} \Delta T^{(2)} \begin{bmatrix} -1 \\ 1 \end{bmatrix} = \begin{bmatrix} 60 \\ -60 \end{bmatrix}. \quad (4.31)$$

The element stiffness equations are:

$$3000 \begin{bmatrix} 1 & -1 \\ -1 & 1 \end{bmatrix} \begin{bmatrix} \bar{u}_1^{(1)} \\ \bar{u}_2^{(1)} \end{bmatrix} = \begin{bmatrix} \bar{f}_{M1}^{(1)} \\ \bar{f}_{M2}^{(1)} \end{bmatrix} + \begin{bmatrix} -150 \\ 150 \end{bmatrix}, \quad 2000 \begin{bmatrix} 1 & -1 \\ -1 & 1 \end{bmatrix} \begin{bmatrix} \bar{u}_2^{(2)} \\ \bar{u}_3^{(2)} \end{bmatrix} = \begin{bmatrix} \bar{f}_{M2}^{(2)} \\ \bar{f}_{M3}^{(2)} \end{bmatrix} + \begin{bmatrix} 60 \\ -60 \end{bmatrix}, \quad (4.32)$$

No globalization is needed because the equations are already in the global system, and thus we can get rid of the localization marker symbols: $\bar{f} \rightarrow f$, $\bar{u} \rightarrow u$. Assembling by any method yields

$$1000 \begin{bmatrix} 3 & -3 & 0 \\ -3 & 5 & -2 \\ 0 & -2 & 2 \end{bmatrix} \begin{bmatrix} u_1 \\ u_2 \\ u_3 \end{bmatrix} = \begin{bmatrix} f_{M1} \\ f_{M2} \\ f_{M3} \end{bmatrix} + \begin{bmatrix} -150 \\ 150 + 60 \\ -60 \end{bmatrix} = \begin{bmatrix} f_{M1} \\ f_{M2} \\ f_{M3} \end{bmatrix} + \begin{bmatrix} -150 \\ 210 \\ -60 \end{bmatrix}. \quad (4.33)$$

The displacement boundary conditions are $u_1 = u_3 = 0$. The mechanical force boundary condition is $f_{M2} = 90$. On removing the first and third equations, the reduced system is $5000 u_2 = f_{M2} + 210 = 90 + 210 = 300$, which yields $u_2 = 300/5000 = +0.06$. The mechanical internal forces in the members are recovered from

$$p_M^{(1)} = \frac{E^{(1)} A^{(1)}}{L^{(1)}} (u_2 - u_1) - E^{(1)} A^{(1)} \alpha^{(1)} \Delta T^{(1)} = 3000 \times 0.06 - 12000 \times 0.0005 \times 25 = 60, \\ p_M^{(2)} = \frac{E^{(2)} A^{(2)}}{L^{(2)}} (u_3 - u_2) - E^{(2)} A^{(2)} \alpha^{(2)} \Delta T^{(2)} = 2000 \times (-0.06) - 12000 \times 0.0005 \times (-10) = -72, \quad (4.34)$$

whence the stresses are $\sigma^{(1)} = 60/12 = 5$ and $\sigma^{(2)} = -72/12 = -6$. Member (1) is in tension and member (2) in compression.

§4.2.8. Worked-Out Example 2

The second example concerns the example truss of Chapters 2-3. The truss is mechanically *unloaded*, that is, $f_{Mx2} = f_{Mx3} = f_{My3} = 0$. However the temperature of members (1) (2) and (3) changes by ΔT , $-\Delta T$ and $3\Delta T$, respectively, with respect to T_{ref} . The thermal expansion coefficient of all three members is assumed to be α . We will perform the analysis keeping α and ΔT as variables.

The thermal forces for each member in global coordinates are obtained by using (4.25) and the third of (4.27):

$$\mathbf{f}_T^{(1)} = E^{(1)} A^{(1)} \alpha^{(1)} \Delta T^{(1)} \begin{bmatrix} 1 & 0 & 0 & 0 \\ 0 & 1 & 0 & 0 \\ 0 & 0 & 1 & 0 \\ 0 & 0 & 0 & 1 \end{bmatrix} \begin{bmatrix} -1 \\ 0 \\ 1 \\ 0 \end{bmatrix} = 100\alpha \Delta T \begin{bmatrix} -1 \\ 0 \\ 1 \\ 0 \end{bmatrix}, \\ \mathbf{f}_T^{(2)} = E^{(2)} A^{(2)} \alpha^{(2)} \Delta T^{(2)} \begin{bmatrix} 0 & -1 & 0 & 0 \\ 1 & 0 & 0 & 0 \\ 0 & 0 & 0 & -1 \\ 0 & 0 & 1 & 0 \end{bmatrix} \begin{bmatrix} -1 \\ 0 \\ 1 \\ 0 \end{bmatrix} = 50\alpha \Delta T \begin{bmatrix} 0 \\ -1 \\ 0 \\ 1 \end{bmatrix}, \quad (4.35) \\ \mathbf{f}_T^{(3)} = E^{(3)} A^{(3)} \alpha^{(3)} \Delta T^{(3)} \frac{1}{\sqrt{2}} \begin{bmatrix} 1 & -1 & 0 & 0 \\ 1 & 1 & 0 & 0 \\ 0 & 0 & 1 & -1 \\ 0 & 0 & 1 & 1 \end{bmatrix} \begin{bmatrix} -1 \\ 0 \\ 1 \\ 0 \end{bmatrix} = 200\alpha \Delta T \begin{bmatrix} -1 \\ -1 \\ 1 \\ 1 \end{bmatrix}.$$

Merging the contribution of these 3 members gives the master thermal force vector

$$\mathbf{f}_T = \alpha \Delta T \begin{bmatrix} -100 + 0 - 200 \\ 0 + 0 - 200 \\ 100 + 0 + 0 \\ 0 - 50 + 0 \\ 0 + 0 + 200 \\ 0 + 50 + 200 \end{bmatrix} = \alpha \Delta T \begin{bmatrix} -300 \\ -200 \\ 100 \\ -50 \\ 200 \\ 250 \end{bmatrix} \quad (4.36)$$

The master stiffness matrix \mathbf{K} does not change. Consequently the master stiffness equations are

$$\begin{bmatrix} 20 & 10 & -10 & 0 & -10 & -10 \\ 10 & 10 & 0 & 0 & -10 & -10 \\ -10 & 0 & 10 & 0 & 0 & 0 \\ 0 & 0 & 0 & 5 & 0 & -5 \\ -10 & -10 & 0 & 0 & 10 & 10 \\ -10 & -10 & 0 & -5 & 10 & 15 \end{bmatrix} \begin{bmatrix} u_{x1} = 0 \\ u_{y1} = 0 \\ u_{x2} \\ u_{y2} = 0 \\ u_{x3} \\ u_{y3} \end{bmatrix} = \begin{bmatrix} f_{Mx1} \\ f_{My1} \\ f_{Mx2} = 0 \\ f_{My2} \\ f_{Mx3} = 0 \\ f_{My3} = 0 \end{bmatrix} + \alpha \Delta T \begin{bmatrix} -300 \\ -200 \\ 100 \\ -50 \\ 200 \\ 250 \end{bmatrix} \quad (4.37)$$

in which f_{Mx1} , f_{My1} and f_{My2} are the unknown mechanical reaction forces, and the known forces and displacements have been marked. Since the prescribed displacements are zero, the reduced system is simply

$$\begin{bmatrix} 10 & 0 & 0 \\ 0 & 10 & 10 \\ 0 & 10 & 15 \end{bmatrix} \begin{bmatrix} u_{x2} \\ u_{x3} \\ u_{y3} \end{bmatrix} = \begin{bmatrix} 0 \\ 0 \\ 0 \end{bmatrix} + \alpha \Delta T \begin{bmatrix} 100 \\ 200 \\ 250 \end{bmatrix} = \alpha \Delta T \begin{bmatrix} 100 \\ 200 \\ 250 \end{bmatrix}. \quad (4.38)$$

Solving (4.38) gives $u_{x2} = u_{x3} = u_{y3} = 10\alpha \Delta T$. Completing \mathbf{u} with the prescribed zero displacements and premultiplying by \mathbf{K} gives the complete effective force vector:

$$\mathbf{f} = \mathbf{K}\mathbf{u} = \begin{bmatrix} 20 & 10 & -10 & 0 & -10 & -10 \\ 10 & 10 & 0 & 0 & -10 & -10 \\ -10 & 0 & 10 & 0 & 0 & 0 \\ 0 & 0 & 0 & 5 & 0 & -5 \\ -10 & -10 & 0 & 0 & 10 & 10 \\ -10 & -10 & 0 & -5 & 10 & 15 \end{bmatrix} \begin{bmatrix} 0 \\ 0 \\ 10 \\ 0 \\ 10 \\ 10 \end{bmatrix} \alpha \Delta T = \alpha \Delta T \begin{bmatrix} -300 \\ -200 \\ 100 \\ -50 \\ 200 \\ 250 \end{bmatrix}. \quad (4.39)$$

But this vector is exactly \mathbf{f}_T . Consequently

$$\mathbf{f}_M = \mathbf{K}\mathbf{u} - \mathbf{f}_T = \mathbf{0}. \quad (4.40)$$

All mechanical joint forces, including reactions, vanish, and so do the internal mechanical forces. This is a consequence of the example frame being statically determinate.⁴ Such structures *do not develop thermal stresses under any combination of temperature changes*.

⁴ For the definition of static determinacy, see any textbook on Mechanics of Materials; e.g. [4.1,4.4].

§4.3. INITIAL FORCE EFFECTS

As previously noted, a wide spectrum of mechanical and non-mechanical effects can be accommodated under the umbrella of the *initial force* concept. The stiffness equations at the local (member) level are

$$\bar{\mathbf{K}}^{(e)} \bar{\mathbf{u}}^{(e)} = \bar{\mathbf{f}}_M^{(e)} + \bar{\mathbf{f}}_I^{(e)} = \bar{\mathbf{f}}^{(e)}, \quad (4.41)$$

and at the global (assembled structure) level:

$$\mathbf{K} \mathbf{u} = \mathbf{f}_M + \mathbf{f}_I = \mathbf{f}. \quad (4.42)$$

In these equations subscripts M and I identify mechanical and initial node forces, respectively. The sum of the two: $\bar{\mathbf{f}}$ at the local member level and \mathbf{f} at the global structure level, are called *effective* forces.

A physical interpretation of (4.42) can be obtained by considering that the structure is blocked against all motions: $\mathbf{u} = \mathbf{0}$. Then $\mathbf{f}_M = -\mathbf{f}_I$, and the undeformed structure experiences mechanical forces. These translate into internal forces and stresses. Engineers also call these *prestresses*.

Local effects that lead to initial forces at the member level are: temperature changes (studied in §4.2, in which $\mathbf{f}_I \equiv \mathbf{f}_T$), moisture diffusion, residual stresses, lack of fit in fabrication, and in-member prestressing. Global effects include prescribed nonzero joint displacements (studied in §4.1) and multimember prestressing (for example, by cable pretensioning of concrete structures).

As can be seen there is a wide variety of physical effects, whether natural or artificial, that lead to nonzero initial forces. The good news is that once the member equations (4.41) are formulated, the remaining DSM steps (globalization, merge and solution) are identical. This nice property extends to the general Finite Element Method.

§4.4. PSEUDOTHERMAL INPUTS

Some commercial FEM programs do not have a way to handle directly effects such as moisture, lack of fit, or prestress. But all of them can handle temperature variation inputs. Since in linear analysis all such effects can be treated as initial forces, it is possible (at least for bar elements) to model them as fictitious thermomechanical effects, by inputting phony temperature changes. The following example indicate that this is done for a bar element.

Suppose that a prestress force F_P is present in a bar. The total elongation is $d = d_M + d_P$ where $d_P = F_P L / (EA)$ is due to prestress. Equate to a thermal elongation: $d_T = \alpha \Delta T_P L$ and solve for $\Delta T_P = F_P / (E A \alpha)$. This is input to the program as a fictitious temperature change. If in addition there is a real temperature change ΔT one would of course specify $\Delta T + \Delta T_P$.

If this device is used, care should be exercised in interpreting results for internal forces and stresses given by the program. The trick is not necessary for personal or open-source codes over which you have full control.

Notes and Bibliography

The additional DSM topics treated in this Chapter are covered in virtually all books on Matrix Structural Analysis, such as the often quoted one by Przemieniecki [4.5]. Several recent FEM books ignore these topics as too elementary.

The physics of thermomechanics and the analysis of thermal stresses is covered adequately in textbooks such as Boley and Wiener [4.2], or manuals such as the widely used [4.6].

For the separate problems of heat conduction and heat transfer, the book by Özişik [4.3] provides a comprehensive classic treatment. There is a vast literature on prestressed structures; search under “prestress” in <http://www3.addall.com>.

The concepts of static determinacy and its counterpart: static indeterminacy, are important in skeletal structures such as trusses and frameworks. The pertinent design tradeoff is: insensitivity to initial force effects versus redundant safety. A discussion of this topic is beyond the scope of the book. Once going past skeletal structural systems, however, indeterminacy is the rule.

References

- [4.1] Beer, F. P. and Johnston E. R., *Mechanics of Materials*, McGraw-Hill, 2nd ed. 1992.
- [4.2] Boley, B. A. and Wiener, J. H., *Theory of Thermal Stresses*, Wiley, New York, 1960.
- [4.3] Özişik, M. N., *Boundary Value Problems of Heat Conduction*, Dover edition, 1989.
- [4.4] Popov, E. P., *Engineering Mechanics of Solids*, Prentice Hall, Englewood Cliffs, N. J., 2nd ed., 1991.
- [4.5] Przemieniecki, J. S., *Theory of Matrix Structural Analysis*, McGraw-Hill, New York, 1968; Dover edition 1986.
- [4.6] Roark, R. J., Budynas, R. G. and Young, W. C., *Roark's Formulas for Stress and Strain*, McGraw-Hill, New York, 7th ed., 2001.

Homework Exercises for Chapter 4
The Direct Stiffness Method: Additional Topics

EXERCISE 4.1

[N:20] Resolve items (a) through (c) — omitting (d) — of the problem of Exercise 3.7 if the vertical right support “sinks” so that the displacement u_{y3} is now prescribed to be -0.5 . Everything else is the same. Use the matrix reduction scheme of §4.1.1 to apply the displacement BCs.

EXERCISE 4.2

[N:20] Use the same data of Exercise 3.7, except that $P = 0$ and consequently there are no applied mechanical forces. Both members have the same dilatation coefficient $\alpha = 10^{-6} \text{ 1/}^\circ\text{F}$. Find the crown displacements u_{x2} and u_{y2} and the member stresses $\sigma^{(1)}$ and $\sigma^{(2)}$ if the temperature of member (1) rises by $\Delta T = 120^\circ\text{F}$ above T_{ref} , whereas member (2) stays at T_{ref} .

Shortcut: the element stiffnesses and master stiffness matrix are the same as in Exercise 3.7, so if that Exercise has been previously assigned no stiffness recomputations are necessary.

EXERCISE 4.3

[A:15] Consider the generic truss member of §2.4, reproduced in Figure E4.1 for convenience.

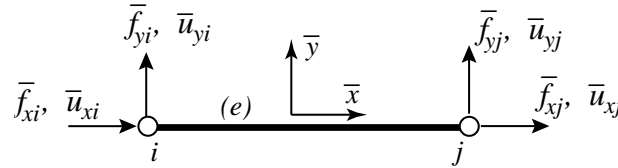


Figure E4.1. Generic truss member.

The disconnected member was supposed to have length L , but because of lack of quality control it was fabricated with length $L + \delta$, where δ is called the “lack of fit.” Determine the initial force vector $\bar{\mathbf{f}}_I$ to be used in (4.41). *Hint:* find the mechanical forces that would compensate for δ and restore the desired length.

EXERCISE 4.4

[A:10] Show that the lack of fit of the foregoing exercise can be viewed as equivalent to a prestress force of $-(EA/L)\delta$.

EXERCISE 4.5

[A:20] Show that prescribed nonzero displacements can, albeit somewhat artificially, be placed under the umbrella of initial force effects. Work this out for the example of §4.1.1. *Hint:* split node displacements into $\mathbf{u} = \mathbf{u}_H + \mathbf{u}_N$, where \mathbf{u}_N (the “nonhomogeneous” part) carries the nonzero displacement values.

EXERCISE 4.6

[A:40]. (Research paper level). Prove that any statically determinate truss structure is free of thermal stresses.

5

Analysis of Example Truss by a CAS

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§5.1. COMPUTER ALGEBRA SYSTEMS

Computer algebra systems, known by the acronym CAS, are programs designed to perform symbolic and numeric manipulations following the rules of mathematics.¹ The development of such programs began in the mid 1960s. The first comprehensive system — the “granddaddy” of them all, called *Macsyma* (an acronym for Project **Mac** Symbolic **Manipulator**) — was developed using the programming language Lisp at MIT’s famous Artificial Intelligence Laboratory over the period 1967 to 1980.

The number and quality of symbolic-manipulation programs has expanded dramatically since the availability of graphical workstations and personal computers has encouraged interactive and experimental programming. As of this writing the leading general-purpose contenders are *Maple* and *Mathematica*.² In addition there are a dozen or so more specialized programs, some of which are available free or at very reasonable cost.

§5.1.1. Why Mathematica?

In the present book *Mathematica* will be used for Chapters and Exercises that develop symbolic and numerical computation for matrix structural analysis and FEM implementations. *Mathematica* is a commercial product developed by Wolfram Research, web site: <http://www.wolfram.com>. The version used to construct the code fragments presented in this Chapter is 4.1, which was commercially released in 2001. (The latest version, released summer 2003, is 5.0.) The main advantages of *Mathematica* for technical computing are:

1. Availability on a wide range of platforms that range from PCs and Macs through Unix workstations. Its competitor *Maple* is primarily used on Unix systems.
2. Up-to-date user interface with above average graphics. On all machines *Mathematica* offers a graphics user interface called the Notebook front-end. This is mandatory for serious work.
3. A powerful programming language.
4. Good documentation and abundance of application books at all levels.

One common disadvantage of CAS, and *Mathematica* is not exception, is computational inefficiency in numerical calculations compared with a low-level implementation in, for instance, C or Fortran. The relative penalty can reach several orders of magnitude. For instructional use, however, the penalty is acceptable when compared to *human efficiency*. This means the ability to get FEM programs up and running in very short time, with capabilities for symbolic manipulation and graphics as a bonus.

§5.1.2. Programming Style and Prerequisites

The following material assumes that you are a moderately experienced user of *Mathematica*, or are willing to learn to be one. See **Notes and Bibliography** for a brief discussion of tutorial and reference materials.

Practice with the program until you reach the level of writing functions, modules and scripts with relative ease. With the Notebook interface and a good primer it takes only a few hours.

When approaching that level you may notice that functions in *Mathematica* display many aspects similar

¹ Some vendors call that kind of activity “doing mathematics by computer.” It is more appropriate to regard such programs as enabling tools that help humans with complicated and error-prone manipulations. As of now, only humans can do mathematics.

² Another commonly used program for engineering computations: *Matlab*, does only numerical computations although an interface to *Maple* can be purchased as a toolbox.

to C.³ You can exploit this similarity if you are proficient in that language. But *Mathematica* functions do have some unique aspects, such as matching arguments by pattern, and the fact that internal variables are global unless otherwise made local.⁴

Although function arguments can be modified, in practice this should be avoided because it may be difficult to trace side effects. The programming style enforced here outlaws output arguments and a function can only return its name. But since the name can be a list of arbitrary objects the restriction is not serious.⁵

Our objective is to develop a symbolic program written in *Mathematica* that solves the example plane truss as well as some symbolic versions thereof. The program will rely heavily on the development and use of *functions* implemented using the `Module` construct of *Mathematica*. Thus the style will be one of procedural programming.⁶ The program will not be particularly modular (in the computer science sense) because *Mathematica* is not suitable for that programming style.⁷

The code presented in Sections 5.2–5.7 uses a few language constructs that may be deemed as advanced, and these are briefly noted in the text so that appropriate reference to the *Mathematica* reference manual can be made.

§5.1.3. Class Demo Scripts

The cell scripts shown in Figures 5.1 and 5.2 will be used to illustrate the organization of a Notebook file and the “look and feel” of some basic *Mathematica* commands. These scripts will be demonstrated in class from a laptop.

§5.2. THE ELEMENT STIFFNESS MODULE

As our first FEM code segment, the top box of Figure 5.3 shows a module that evaluates and returns the 4×4 stiffness matrix of a plane truss member (two-node bar) in global coordinates. The text in that box of that figure is supposed to be placed on a Notebook cell. Executing the cell, by clicking on it and hitting an appropriate key (<Enter> on a Mac), gives the output shown in the bottom box. The contents of the figure is described in further detail below.

§5.2.1. Module Description

The stiffness module is called `ElemStiff2DTwoNodeBar`. Such descriptive names are permitted by the language. This reduces the need for detailed comments.

³ Simple functions can be implemented in *Mathematica* directly, for instance `DotProduct[x_,y_]:=x.y`; more complex ones are handled by the `Module` construct emphasized here.

⁴ In *Mathematica* everything is a function, including programming constructs. Example: in C `for` is a loop-opening keyword; in *Mathematica* `For` is a function that runs a loop according to its arguments.

⁵ Such restrictions on arguments and function returns are closer in spirit to C than Fortran although you can of course modify C-function arguments using pointers — exceedingly dangerous but often unavoidable.

⁶ The name `Module` should not be taken too seriously: it is far away from the concept of modules in Ada, Modula, Oberon or Fortran 90. But such precise levels of interface control are rarely needed in symbolic languages.

⁷ And indeed none of the CAS packages in popular use is designed for strong modularity because of historical and interactivity constraints.

Integration example

```
f[x_,α_,β_]:= (1+β*x^2)/(1+α*x+x^2);
F=Integrate[f[x,-1,2],{x,0,5}];
F=Simplify[F];
Print[F]; Print[N[F]];
F=NIntegrate[f[x,-1,2],{x,0,5}];
Print["F=",F/InputForm];
```

10 + Log[21]

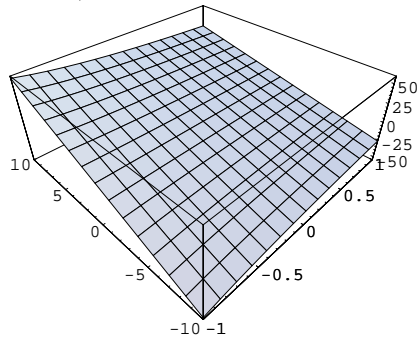
13.0445

F=13.044522437723455

Figure 5.1. Example cell for class demo.

```
Fa=Integrate[f[z,a,b],{z,0,5}];
Fa=Simplify[Fa]; Print[Fa];
Plot3D[Fa,{a,-1,1},{b,-10,10},ViewPoint->{-1,-1,1}];
Print["Fa=",Fa/InputForm]
```

$$\frac{1}{2\sqrt{4-a^2}} \left(10\sqrt{4-a^2} b - a\sqrt{4-a^2} b \log[26+5a] - i(2+(-2+a^2)b) \log\left[1 - \frac{ia}{\sqrt{4-a^2}}\right] + \right. \\ \left. 2i \log\left[1 + \frac{ia}{\sqrt{4-a^2}}\right] - 2ib \log\left[1 + \frac{ia}{\sqrt{4-a^2}}\right] + ia^2 b \log\left[1 + \frac{ia}{\sqrt{4-a^2}}\right] + \right. \\ \left. 2i \log\left[\frac{-10i - ia + \sqrt{4-a^2}}{\sqrt{4-a^2}}\right] - 2ib \log\left[\frac{-10i - ia + \sqrt{4-a^2}}{\sqrt{4-a^2}}\right] + ia^2 b \log\left[\frac{-10i - ia + \sqrt{4-a^2}}{\sqrt{4-a^2}}\right] - \right. \\ \left. 2i \log\left[\frac{10i + ia + \sqrt{4-a^2}}{\sqrt{4-a^2}}\right] + 2ib \log\left[\frac{10i + ia + \sqrt{4-a^2}}{\sqrt{4-a^2}}\right] - ia^2 b \log\left[\frac{10i + ia + \sqrt{4-a^2}}{\sqrt{4-a^2}}\right] \right)$$



$$\begin{aligned} Fa = & (10\sqrt{4-a^2} b - a\sqrt{4-a^2} b \log[26+5a] - \\ & i(2+(-2+a^2)b) \log[1 - (Ia)/\sqrt{4-a^2}] + (2I) \log[1 + (Ia)/\sqrt{4-a^2}] - \\ & (2I) b \log[1 + (Ia)/\sqrt{4-a^2}] + Ia^2 b \log[1 + (Ia)/\sqrt{4-a^2}] + \\ & (2I) \log[(-10I - Ia + \sqrt{4-a^2})/\sqrt{4-a^2}] - \\ & (2I) b \log[(-10I - Ia + \sqrt{4-a^2})/\sqrt{4-a^2}] + \\ & Ia^2 b \log[(-10I - Ia + \sqrt{4-a^2})/\sqrt{4-a^2}] - \\ & (2I) \log[(10I + Ia + \sqrt{4-a^2})/\sqrt{4-a^2}] + \\ & (2I) b \log[(10I + Ia + \sqrt{4-a^2})/\sqrt{4-a^2}] - \\ & Ia^2 b \log[(10I + Ia + \sqrt{4-a^2})/\sqrt{4-a^2}]) / (2\sqrt{4-a^2}) \end{aligned}$$

Figure 5.2. Another example cell for class demo.

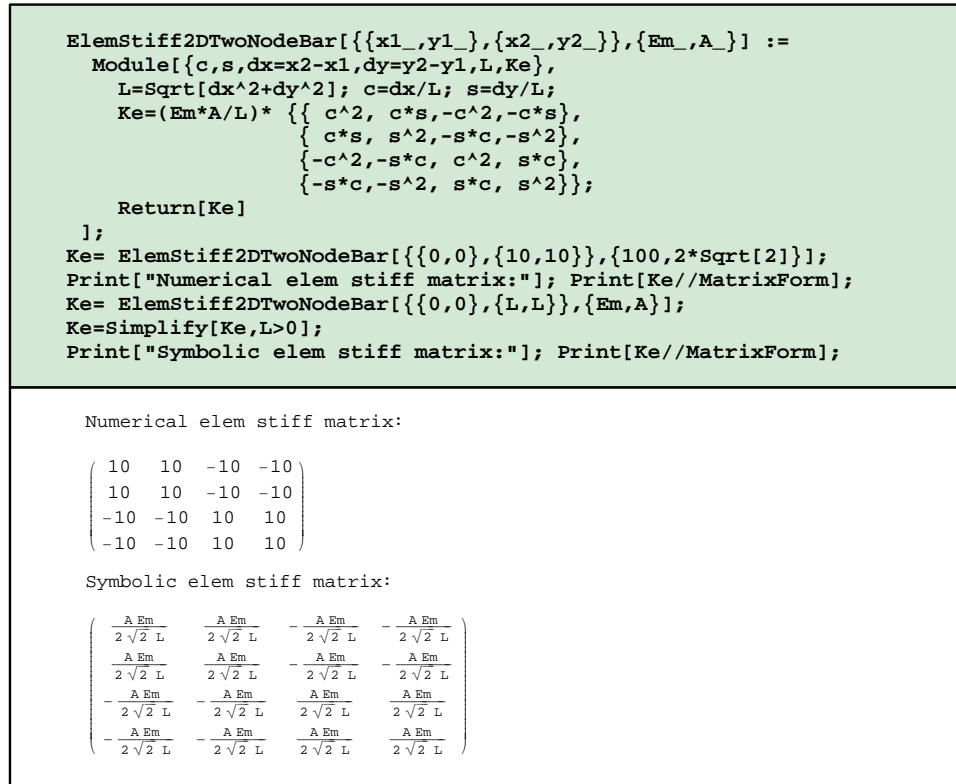


Figure 5.3. Module `ElemStiff2DTwoNodeBar` to form the element stiffness of a 2D bar element in global coordinates, test program and its output.

The module takes two arguments:

$\{\{x_1, y_1\}, \{y_1, y_2\}\}$ A two-level list⁸ containing the $\{x, y\}$ coordinates of the bar end nodes labelled as 1 and 2.⁹

$\{Em, A\}$ A level-one list containing the bar elastic modulus, E and the member cross section area, A . See §5.2.3 as to why name E cannot be used.

The use of the underscore after argument item names in the declaration of the `Module` is a requirement for pattern-matching in *Mathematica*. If, as recommended, you have learned functions and modules this aspect should not come as a surprise.

The module name returns the 4×4 member stiffness matrix internally called `Ke`. The logic that leads to the formation of that matrix is straightforward and need not be explained in detail. Note, however, the elegant direct declaration of the matrix `Ke` as a level-two list, which eliminates the fiddling around with array indices typical of standard programming languages. The format in fact closely matches the mathematical expression (3.4).

⁸ A level-one list is a sequence of items enclosed in curly braces. For example: $\{x_1, y_1\}$ is a list of two items. A level-two list is a list of level-one lists. An important example of a level-two list is a matrix.

⁹ These are called the *local node numbers*, and replace the i, j of previous Chapters. This is a common FEM programming practice.

§5.2.2. Programming Remarks

The function in Figure 5.3 uses several intermediate variables with short names: dx , dy , s , c and L . It is strongly advisable to make these symbols *local* to avoid potential names clashes somewhere else.¹⁰ In the `Module[...]` construct this is done by listing those names in a list immediately after the opening bracket. Local variables may be *initialized* when they are constants or simple functions of the argument items; for example on entry to the module $dx = x_2 - x_1$ initializes variable dx to be the difference of x node coordinates, namely $\Delta x = x_2 - x_1$.

The use of the `Return` statement fulfills the same purpose as in C or Fortran 90. *Mathematica* guides and textbooks advise against the use of that and other C-like constructs. The writer strongly disagrees: the `Return` statement makes clear what the `Module` gives back to its invoker and is self-documenting.

§5.2.3. Case Sensitivity

Mathematica, like most recent computer languages, is case sensitive so that for instance E is not the same as e . This is fine. But the language designer decided that names of system-defined objects such as built-in functions and constants must begin with a capital letter. Consequently the liberal use of names beginning with a capital letter may run into clashes. If, for example, you cannot use E because of its built-in meaning as the base of natural logarithms.¹¹

In the code fragments presented throughout this book, identifiers beginning with upper case are used for objects such as stiffness matrices, modulus of elasticity, and cross section area, following established usage in Mechanics. When there is danger of clashing with a protected system symbol, additional lower case letters are used. For example, E_m is used for the elastic modulus instead of E because the latter is a reserved symbol.

§5.2.4. Testing the Member Stiffness Module

Following the definition of `ElemStiff2DTwoNodeBar` in Figure 5.3 there are several statements that constitute the *module test program* that call the module and print the returned results. Two cases are tested. First, the stiffness of member (3) of the example truss, using all-numerical values. Next, some of the input arguments for the same member are given symbolic names so they stand for variables; for example the elastic module is given as E_m instead of 100 as in the foregoing test. The print output of the test is shown in the lower portion of Figure 5.3.

The first test returns the member stiffness matrix (3.10) as may be expected. The second test returns a symbolic form in which three symbols appear: the coordinates of end node 2, which is taken to be located at $\{L, L\}$ instead of $\{10, 10\}$, A , which is the cross-section area and E_m , which is the elastic modulus. Note that the returning matrix K_e is subject to a `Simplify` step before printing it, which is the subject of an Exercise. The ability to carry along variables is of course a fundamental capability of any CAS and the main reason for which such programs are used.

§5.3. MERGING A MEMBER INTO THE MASTER STIFFNESS

The next fragment of *Mathematica* code, listed in Figure 5.4, is used in the assembly step of the DSM. `Module MergeElemIntoMasterStiff` receives the 4×4 element stiffness matrix formed by

¹⁰ The “global by default” choice is the worst one, but we must live with the rules of the language.

¹¹ In retrospect this appears to have been a highly questionable decision. System defined names should have been identified by a reserved prefix or postfix to avoid surprises, as done in *Macsyma* or *Maple*. *Mathematica* issues a warning message, however, if an attempt to redefine a “protected symbol” is made.

```

MergeElemIntoMasterStiff[Ke_,eftab_,Kin_]:=Module[
  {i,j,ii,jj,K=Kin},
    For [i=1, i<=4, i++, ii=eftab[[i]],
      For [j=i, j<=4, j++, jj=eftab[[j]],
        K[[jj,ii]]=K[[ii,jj]]+Ke[[i,j]]
      ]
    ]; Return[K]
];

K=Table[0,{6},{6}];
Print["Initialized master stiffness matrix:"];
Print[K//MatrixForm]
Ke=ElemStiff2DTwoNodeBar[{0,0},{10,10}},{100,2*Sqrt[2]};
Print["Member stiffness matrix:"]; Print[Ke//MatrixForm];
K=MergeElemIntoMasterStiff[Ke,{1,2,5,6},K];
Print["Master stiffness after member merge:"];
Print[K//MatrixForm];

```

Initialized master stiffness matrix:

$$\begin{pmatrix} 0 & 0 & 0 & 0 & 0 & 0 \\ 0 & 0 & 0 & 0 & 0 & 0 \\ 0 & 0 & 0 & 0 & 0 & 0 \\ 0 & 0 & 0 & 0 & 0 & 0 \\ 0 & 0 & 0 & 0 & 0 & 0 \\ 0 & 0 & 0 & 0 & 0 & 0 \end{pmatrix}$$

Member stiffness matrix:

$$\begin{pmatrix} 10 & 10 & -10 & -10 \\ 10 & 10 & -10 & -10 \\ -10 & -10 & 10 & 10 \\ -10 & -10 & 10 & 10 \end{pmatrix}$$

Master stiffness after member merge:

$$\begin{pmatrix} 10 & 10 & 0 & 0 & -10 & -10 \\ 10 & 10 & 0 & 0 & -10 & -10 \\ 0 & 0 & 0 & 0 & 0 & 0 \\ 0 & 0 & 0 & 0 & 0 & 0 \\ -10 & -10 & 0 & 0 & 10 & 10 \\ -10 & -10 & 0 & 0 & 10 & 10 \end{pmatrix}$$

Figure 5.4. Module MergeElemIntoMasterStiff to merge a 4×4 bar element stiffness into the master stiffness matrix, test program and its output.

FormElemStiff2DNodeBar and “merges” it into the master stiffness matrix. The module takes three arguments:

- Ke The 4×4 member stiffness matrix to be merged. This is a level-two list.
- eftab The column of the Element Freedom Table, defined in §3.4.1, appropriate to the member being merged; cf. (3.29). Recall that the EFT lists the global equation numbers for the four member degrees of freedom. This is a level-one list consisting of 4 integers.
- Kinp The incoming 6×6 master stiffness matrix. This is a level-two list.

MergeElemIntoMasterStiff returns, as module name, the updated master stiffness matrix internally called K with the member stiffness merged in. Thus we encounter here a novelty: an input-output argument. Because a formal argument cannot be modified, the situation is handled by copying the incoming Kin into K on entry. It is the copy which is updated and returned via the Return statement. The implementation has a strong C flavor with two nested For loops. Because the iterators are very simple, nested Do loops could have been used as well.

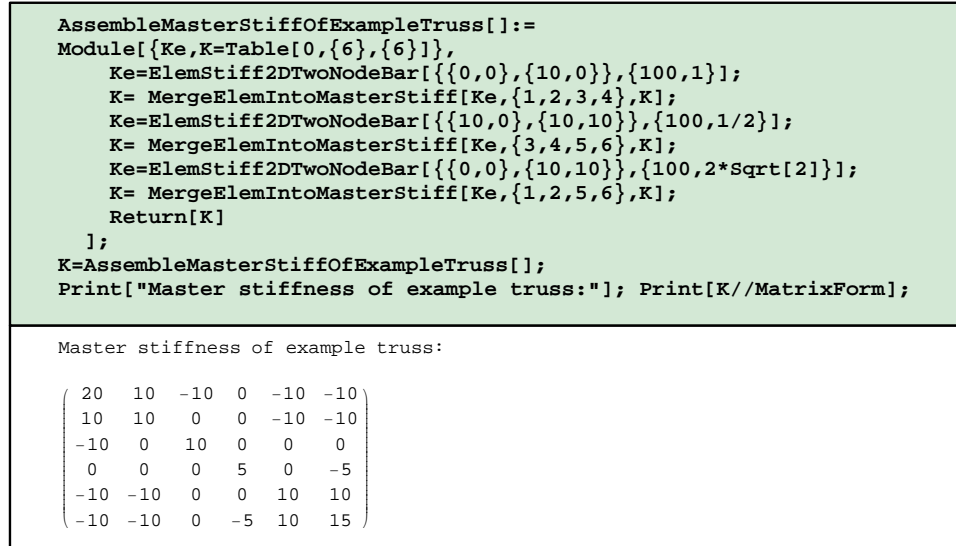


Figure 5.5. Module `AssembleMasterStiffOfExampleTruss` that forms the 6×6 master stiffness matrix of the example truss, test program and its output.

The statements after the module provide a simple test. Before the first call to this function, the master stiffness matrix must be initialized to a zero 6×6 array. This is done in the first test statement using the `Table` function. The test member stiffness matrix is that of member (3) of the example truss, and is obtained by calling `ElemStiff2DTwoNodeBar`. The EFT is $\{1,2,5,6\}$ since element freedoms 1,2,3,4 map into global freedoms 1,2,5,6. Running the test statements yields the listing given in Figure 5.4. The result is as expected.

§5.4. ASSEMBLING THE MASTER STIFFNESS

The module `AssembleMasterStiffOfExampleTruss`, listed in the top box of Figure 5.5, makes use of the foregoing two modules: `ElemStiff2DTwoNodeBar` and `MergeElemIntoMasterStiff`, to form the master stiffness matrix of the example truss. The initialization of the stiffness matrix array in `K` to zero is done by the `Table` function of Mathematica, which is handy for initializing lists. The remaining statements are self explanatory. The module is similar in style to argumentless Fortran or C functions. It takes no arguments. All the example truss data is “wired in.”

The output from the test program in is shown in the lower box of Figure 5.5. The output stiffness matches that in Equation (3.20), as can be expected if all fragments used so far work correctly.

§5.5. MODIFYING THE MASTER SYSTEM

Following the assembly process the master stiffness equations $\mathbf{Ku} = \mathbf{f}$ must be modified to account for single-freedom displacement boundary conditions. This is done through the computer-oriented equation modification process described in §3.4.2.

Module `ModifiedMasterStiffForDBC` carries out this process for the master stiffness matrix \mathbf{K} , whereas `ModifiedMasterForcesForDBC` does this for the nodal force vector \mathbf{f} . These two modules are listed in the top box of Figure 5.6, along with test statements. The logic of both functions, but especially that of `ModifiedMasterForcesForBC`, is considerably simplified by assuming that *all*

```

ModifiedMasterStiffForDBC[pdof_,K_] := Module[
  {i,j,k,nk=Length[K],np=Length[pdof],Kmod=K},
  For [k=1,k<=np,k++, i=pdof[[k]]];
    For [j=1,j<=nk,j++, Kmod[[i,j]]=Kmod[[j,i]]=0];
      Kmod[[i,i]]=1];
  Return[Kmod]
];
ModifiedMasterForcesForDBC[pdof_,f_] := Module[
  {i,k,np=Length[pdof],fmod=f},
  For [k=1,k<=np,k++, i=pdof[[k]]]; fmod[[i]]=0];
  Return[fmod]
];
K=Array[Kij,{6,6}]; Print["Assembled master stiffness:"];
Print[K//MatrixForm];
K=ModifiedMasterStiffForDBC[{1,2,4},K];
Print["Master stiffness modified for displacement B.C.:"];
Print[K//MatrixForm];
f=Array[fi,{6}]; Print["Force vector:"]; Print[f];
f=ModifiedMasterForcesForDBC[{1,2,4},f];
Print["Force vector modified for displacement B.C.:"]; Print[f];

```

Assembled master stiffness:

$$\begin{pmatrix} K_{ij}[1,1] & K_{ij}[1,2] & K_{ij}[1,3] & K_{ij}[1,4] & K_{ij}[1,5] & K_{ij}[1,6] \\ K_{ij}[2,1] & K_{ij}[2,2] & K_{ij}[2,3] & K_{ij}[2,4] & K_{ij}[2,5] & K_{ij}[2,6] \\ K_{ij}[3,1] & K_{ij}[3,2] & K_{ij}[3,3] & K_{ij}[3,4] & K_{ij}[3,5] & K_{ij}[3,6] \\ K_{ij}[4,1] & K_{ij}[4,2] & K_{ij}[4,3] & K_{ij}[4,4] & K_{ij}[4,5] & K_{ij}[4,6] \\ K_{ij}[5,1] & K_{ij}[5,2] & K_{ij}[5,3] & K_{ij}[5,4] & K_{ij}[5,5] & K_{ij}[5,6] \\ K_{ij}[6,1] & K_{ij}[6,2] & K_{ij}[6,3] & K_{ij}[6,4] & K_{ij}[6,5] & K_{ij}[6,6] \end{pmatrix}$$

Master stiffness modified for displacement B.C.:

$$\begin{pmatrix} 1 & 0 & 0 & 0 & 0 & 0 \\ 0 & 1 & 0 & 0 & 0 & 0 \\ 0 & 0 & K_{ij}[3,3] & 0 & K_{ij}[3,5] & K_{ij}[3,6] \\ 0 & 0 & 0 & 1 & 0 & 0 \\ 0 & 0 & K_{ij}[5,3] & 0 & K_{ij}[5,5] & K_{ij}[5,6] \\ 0 & 0 & K_{ij}[6,3] & 0 & K_{ij}[6,5] & K_{ij}[6,6] \end{pmatrix}$$

Force vector:

$$\{fi[1], fi[2], fi[3], fi[4], fi[5], fi[6]\}$$

Force vector modified for displacement B.C.:

$$\{0, 0, fi[3], 0, fi[5], fi[6]\}$$

Figure 5.6. Modules ModifiedMasterStiff and ModifiedMasterForce that modify the master stiffness matrix and force vector of a truss to impose displacement BCs.

prescribed displacements are zero, that is, the BCs are homogeneous. The more general case of nonzero prescribed values is treated in Part III of the book.

Function ModifiedMasterStiffnessForDBC has two arguments:

pdof A list of the prescribed degrees of freedom identified by their global number. For the example truss this list contains three entries: {1, 2, 4}.

K The master stiffness matrix **K** produced by the assembly process.

The function clears appropriate rows and columns of **K**, places ones on the diagonal, and returns the modified **K** as function value. The only slightly fancy thing in this module is the use of the *Mathematica* function Length to extract the number of prescribed displacement components: Length[pdof] here will return the value 3, which is the length of the list pdof. Similarly nk=Length[K] assigns 6 to nk, which is the order of matrix **K**. Although for the example truss these values are known *a priori*, the use of Length serves to illustrate a technique that is heavily used in more general code.

Module ModifiedMasterForcesForDBC has similar structure and logic and need not be described in


```

IntForce2DTwoNodeBar[{{x1_,y1_},{x2_,y2_}},{Em_,A_},eftab_,u_]:=
Module[{c,s,dx=x2-x1,dy=y2-y1,L,ix,iy,jx,jy,ubar,e},
L=Sqrt[dx^2+dy^2]; c=dx/L; s=dy/L; {ix,iy,jx,jy}=eftab;
ubar={c*u[[ix]]+s*u[[iy]],-s*u[[ix]]+c*u[[iy]],
      c*u[[jx]]+s*u[[jy]],-s*u[[jx]]+c*u[[jy]]};
e=(ubar[[3]]-ubar[[1]])/L; Return[Em*A*e]
];
p =IntForce2DTwoNodeBar[{{0,0},{10,10}},{100,2*Sqrt[2]},
{1,2,5,6},{0,0,0,0,0.4,-0.2}];
Print["Member int force (numerical):"]; Print[N[p]];
p =IntForce2DTwoNodeBar[{{0,0},{L,L}},{Em,A},
{1,2,5,6},{0,0,0,0,ux3,uy3}];
Print["Member int force (symbolic):"]; Print[Simplify[p]];

```

```

Member int force (numerical):
2.82843
Member int force (symbolic):
A Em (ux3 + uy3)
2 L

```

Figure 5.7. Module IntForce2DTwoNodeBar for computing the internal force in a bar element.

detail. It is important to note, however, that for homogeneous BCs the modules are independent of each other and may be called in any order. On the other hand, if there were nonzero prescribed displacements present the force modification must be done *before* the stiffness modification. This is because stiffness coefficients that are cleared in the latter are needed for modifying the force vector.

The test statements are purposely chosen to illustrate another feature of *Mathematica*: the use of the Array function to generate subscripted symbolic arrays of one and two dimensions. The test output is shown in the bottom box of Figure 5.6, which should be self explanatory. The force vector and its modified form are printed as row vectors to save space.

§5.6. RECOVERING INTERNAL FORCES

Mathematica provides built-in matrix operations for solving a linear system of equations and multiplying matrices by vectors. Thus we do not need to write application functions for the solution of the modified stiffness equations and for the recovery of nodal forces. Consequently, the last application functions we need are those for internal force recovery.

Function IntForce2DTwoNodeBar listed in the top box of Figure 5.7 computes the internal force in an individual bar element. It is somewhat similar in argument sequence and logic to ElemStiff2DTwoNodeBar (Figure 5.3). The first two arguments are identical. Argument eftab provides the Element Freedom Table array for the element. The last argument, u, is the vector of computed node displacements.

The logic of IntForce2DTwoNodeBar is straightforward and follows the method outlined in §3.2.1. Member joint displacements $\bar{\mathbf{u}}^{(e)}$ in local coordinates $\{\bar{x}, \bar{y}\}$ are recovered in array ubar, then the longitudinal strain $e = (\bar{u}_{xj} - \bar{u}_{xi})/L$ and the internal (axial) force $p = EAe$ is returned as function value. As coded the function contains redundant operations because entries 2 and 4 of ubar (that is, components \bar{u}_{yi} and \bar{u}_{yj}) are not actually needed to get p , but were kept to illustrate the general backtransformation of global to local displacements.

Running this function with the test statements shown after the module produces the output shown in the bottom box of Figure 5.7. The first test is for member (3) of the example truss using the actual nodal displacements (3.24). It also illustrates the use of the *Mathematica* built in function N to produce output

```

IntForcesOfExampleTruss[u_]:= Module[{f=Table[0,{3}]},
  f[[1]]=IntForce2DTwoNodeBar[{{0,0},{10,0}}, {100,1}, {1,2,3,4},u];
  f[[2]]=IntForce2DTwoNodeBar[{{10,0},{10,10}}, {100,1/2}, {3,4,5,6},u];
  f[[3]]=IntForce2DTwoNodeBar[{{0,0},{10,10}}, {100,2*Sqrt[2]},
    {1,2,5,6},u];
  Return[f]
];
f=IntForcesOfExampleTruss[{0,0,0,0,0.4,-0.2}];
Print["Internal member forces in example truss:"];Print[N[f]];

Internal member forces in example truss:
{0., -1., 2.82843}

```

Figure 5.8. Module `IntForceOfExampleTruss` that computes internal forces in the 3 members of the example truss.

in floating-point form. The second test does a symbolic calculation in which several argument values are fed in variable form.

The top box of Figure 5.8 lists a higher-level function, `IntForcesOfExampleTruss`, which has a single argument: `u`. This is the complete 6-vector of joint displacements \mathbf{u} . This function calls `IntForce2DTwoNodeBar` three times, once for each member of the example truss, and returns the three member internal forces thus computed as a 3-component list.

The test statements listed after `IntForcesOfExampleTruss` feed the actual node displacements (3.24) to `IntForcesOfExampleTruss`. Running the functions with the test statements produces the output shown in the bottom box of Figure 5.8. The internal forces are $p^{(1)} = 0$, $p^{(2)} = -1$ and $p^{(3)} = 2\sqrt{2} = 2.82843$.

§5.7. PUTTING THE PIECES TOGETHER

After all this development and testing effort documented in Figures 5.3 through 5.8 we are ready to make use of all these bits and pieces of code to analyze the example plane truss. This is actually done with the logic shown in Figure 5.9. This *driver program* uses the previously described modules

```

ElemStiff2DTwoNodeBar
MergeElemIntoMasterStiff
AssembleMasterStiffOfExampleTruss
ModifiedMasterStiffForDBC
ModifiedMasterForcesForDBC
IntForce2DTwoNodeTruss
IntForcesOfExampleTruss

```

(5.1)

These functions must have been defined ("compiled") at the time the driver programs described below are run. A simple way to making sure that all of them are defined is to put all these functions in the same Notebook file and to mark them as *initialization cells*. These cells may be executed by picking up Kernel → Initialize → Execute Initialization. (An even simpler procedure would to group them all in one cell, but that would make placing separate test statements difficult.)

For a hierarchical version of (5.1), see last CATECS slide.

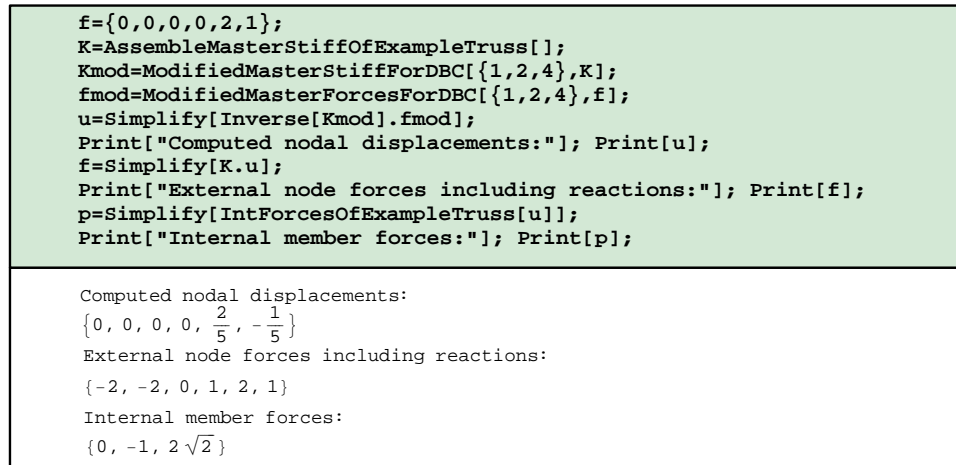


Figure 5.9. Driver program for numerical analysis of example truss and its output.

§5.7.1. The Driver Program

The program listed in the top box of Figure 5.9 first assembles the master stiffness matrix through `AssembleMasterStiffOfExampleTruss`. Next, it applies the displacement boundary conditions through `ModifiedMasterStiffForDBC` and `ModifiedMasterForcesForDBC`. Note that the modified stiffness matrix is placed into `Kmod` rather than `K` to save the original form of the master stiffness for the reaction force recovery later. The complete displacement vector is obtained by the matrix calculation

$$u = \text{Inverse}[Kmod] \cdot fmod$$

which takes advantage of two built-in *Mathematica* functions. `Inverse` returns the inverse of its matrix argument¹² The dot operator signifies matrix multiply (here, matrix-vector multiply.) The enclosing `Simplify` function is placed to simplify the expression of vector `u` in case of symbolic calculations; it is actually redundant if all computations are numerical as in Figure 5.9.

The remaining calculations recover the node vector including reactions by the matrix-vector multiply $f = K \cdot u$ (recall that `K` contains the unmodified master stiffness matrix) and the member internal forces `p` through `IntForcesOfExampleTruss`. The program prints `u`, `f` and `p` as row vectors to conserve space.

Running the program of the top box of Figure 5.9 produces the output shown in the bottom box of that figure. The results confirm the hand calculations of Chapter 3.

§5.7.2. Is This Worth the Trouble?

At this point you may wonder whether all of this work is worth the trouble. After all, a hand calculation (typically helped by a programmable calculator) would be quicker in terms of flow time. Writing and debugging the *Mathematica* fragments displayed here took the writer about six hours (although about two thirds of this was spent in editing and getting the fragment listings into the Chapter.) For larger problems, however, *Mathematica* would certainly beat hand-plus-calculator computations, the cross-over typically appearing for 10 to 20 equations. For up to about 500 equations and using floating-point

¹² This is a highly inefficient way to solve $Ku = f$ if this system becomes large. It is done here to keep simplicity.

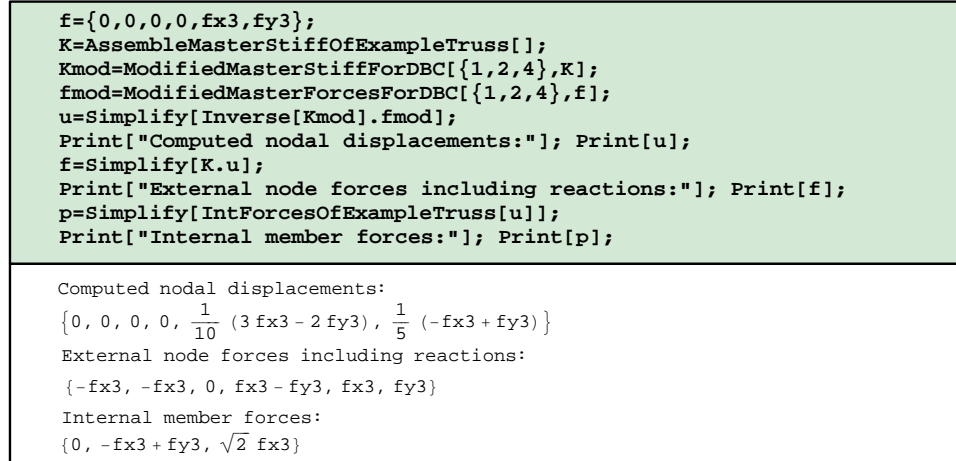


Figure 5.10. Driver program for symbolic analysis of example truss and its output.

arithmetic, *Mathematica* gives answers within minutes on a fast PC or Mac with sufficient memory but eventually runs out of steam at about 1000 equations. For a range of 1000 to about 10000 equations, *Matlab* would be the best compromise between human and computer flow time. Beyond 10000 equations a program in a low-level language, such as C or Fortran, would be most efficient in terms of computer time.¹³

One distinct advantage of computer algebra systems appear when you need to *parametrize* a small problem by leaving one or more problem quantities as variables. For example suppose that the applied forces on node 3 are to be left as f_{x3} and f_{y3} . You replace the last two components of array p as shown in the top box of Figure 5.10, execute the cell and shortly get the symbolic answer shown in the bottom box of Figure 5.10. This is the answer to an infinite number of numerical problems. Although one may try to undertake such studies by hand, the likelihood of errors grows rapidly with the complexity of the system. Symbolic manipulation systems can amplify human abilities in this regard, as long as the algebra “does not explode” because of combinatorial complexity. Examples of such nontrivial calculations will appear throughout the following Chapters.

REMARK 5.1

The “combinatorial explosion” danger of symbolic computations should be always kept in mind. For example, the numerical inversion of a $N \times N$ matrix is a $O(N^3)$ process, whereas symbolic inversion goes as $O(N!)$. For $N = 48$ the floating-point numerical inverse will be typically done in a fraction of a second. But the symbolic adjoint will have $48! = 12413915592536072670862289047373375038521486354677760000000000$ terms, or $O(10^{61})$. There may be enough electrons in this Universe to store that, but barely ...

Notes and Bibliography

As noted in §5.1.2 the 1350-page *Mathematica Book* [5.1] is just a reference manual. Since the contents are available online (click on **Help** in topbar) as part of purchase of the full system¹⁴, buying the printed book (list: \$53 but heavily discounted as used) is not necessary. In fact the latest edition is 1999; updates since have gone directly to the online version.

¹³ The current record for FEM structural applications is about 50 million equations, done on a massively parallel supercomputers. Fluid mechanics problems with over 200 million equations have been solved.

¹⁴ The student version comes with limited help files, but is otherwise functionally complete.

There is a nice tutorial available by Glynn and Gray [5.2], list: \$35, dated 1999. (Theodore Gray invented the Notebook front-end that appeared in version 2.2.) It is also available on CDROM from MathWare, Ltd, P. O. Box 3025, Urbana, IL 61208, e-mail: info@mathware.com. The CDROM is a hyperlinked version of the book that can be installed on the same directory as *Mathematica*.

Beyond these, there is a large number of books at all levels that expound on the use of *Mathematica* for various applications ranging from pure mathematics to physics and engineering. A web search (September 2003) on www3.addall.com hit 150 book titles containing *Mathematica*, compared to 111 for *Maple* and 148 for *Matlab*. A google search hits 1,550,000 pages containing *Mathematica*, but here *Matlab* wins with 1,770,000 hits.

References

- [5.1] Wolfram, S., *The Mathematica Book*, Wolfram Media Inc., 4th ed. 1999
- [5.2] Glynn, J. and Gray, T. H., *The Beginner's Guide to Mathematica Version 4*, Cambridge Univ. Press, 1999.

Homework Exercises for Chapter 5

Analysis of Example Truss by a CAS

Before doing any of these Exercises, download the *Mathematica* Notebook file `ExampleTruss.nb` from the course web site. (Go to Chapter 5 Index and click on the link). Open this Notebook file using version 4.0 or a later one. The first eight cells contain the modules and test statements listed in the top boxes of Figures 5.3–10. The first six of these are marked as *initialization cells*. Before running driver programs, they should be executed by picking up Kernel → Evaluation → Execute Initialization. Verify that the output of those six cells agrees with that shown in the bottom boxes of Figures 5.3–6. Then execute the driver programs in Cells 7–8 by clicking on each cell and pressing the appropriate key: <Enter> on a Mac, <Shift-Enter> on a Windows PC. Compare the output with that shown in Figures 5.9–10. If the output checks out, you may proceed to the Exercises.

EXERCISE 5.1

[C:10] Explain why the `Simplify` command in the test statements of Figure 5.3 says $L > 0$. (One way to figure this out is to just say `Ke=Simplify[Ke]` and look at the output. Related question: why does *Mathematica* refuse to simplify `Sqrt[L^2]` to `L` unless one specifies the sign of `L` in the `Simplify` command?

EXERCISE 5.2

[C:10] Explain the logic of the `For` loops in the merge function `MergeElemIntoMasterStiff` of Figure 5.4. What does the operator `+=` do?

EXERCISE 5.3

[C:10] Explain the reason behind the use of `Length` in the modules of Figure 5.6. Why not simply set `nk` and `np` to 6 and 3, respectively?

EXERCISE 5.4

[C:15] Of the seven modules listed in Figures 5.3 through 5.8, with names collected in (5.1), two can be used only for the example truss, three can be used for any plane truss, and two can be used for other structures analyzed by the DSM. Identify which ones and briefly state the reasons for your classification.

EXERCISE 5.5

[C:20] Modify the modules `AssembleMasterStiffOfExampleTruss`, `IntForcesOfExampleTruss` and the driver program of Figure 5.9 to solve numerically the three-node, two-member truss of Exercise 3.7. Verify that the output reproduces the solution given for that problem. Hint: modify cells but keep a copy of the original Notebook handy in case things go wrong.

EXERCISE 5.6

[C:25] Expand the logic of `ModifiedMasterForcesForDBC` to permit specified nonzero displacements. Specify these in a second argument called `pval`, which contains a list of prescribed values paired with `pdof`.

```

xynode={{0,0},{10,0},{10,10}}; elenod={{1,2},{2,3},{3,1}};
unode={{0,0},{0,0},{2/5,-1/5}}; amp=5; p={};
For [t=0,t<=1,t=t+1/5,
  For [e=1,e<=Length[elenod],e++, {i,j}=elenod[[e]];
    xyi=xynode[[i]];ui=unode[[i]];xyj=xynode[[j]];uj=unode[[j]];
    p=AppendTo[p,Graphics[Line[{xyi+amp*t*ui,xyj+amp*t*uj}]]];
  ];
];
Show[p,Axes->False,AspectRatio->Automatic];

```

Figure E5.1. Mystery program for Exercise 5.7.

EXERCISE 5.7

[C:20] Explain what the program of Figure E5.1 does, and the logic behind what it does. (You may want to put it in a cell and execute it.) What modifications would be needed so it can be used for any plane struss?

6

Constructing MoM Members

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§6.1. INTRODUCTION

The truss member used as example in Chapters 2–5 is an instance of a *structural element*. Such elements may be formulated directly using concepts and modeling techniques developed in Mechanics of Materials (MoM).¹ The construction does not involve the more advanced tools that are required for the continuum finite elements that appear in Part II.

This Chapter presents an overview of the technique to construct the element stiffness equations of “MoM members” using simple matrix operations. These simplified equations come in handy for a surprisingly large number of applications, particularly in skeletal structures. Focus is on *simplex elements*, which may be formed directly as a sequence of matrix operations. Non-simplex elements are presented as a recipe because their proper formulation requires work theorems not yet studied.

The physical interpretation of the FEM is still emphasized. Consequently we continue to speak of structures built up of *members* (elements) connected at *joints* (nodes).

§6.2. FORMULATION OF MOM MEMBERS

§6.2.1. What They Look Like

MoM-based formulations are largely restricted to *intrinsically one-dimensional members*. These are structural components one of whose dimensions, called the *longitudinal dimension*, is significantly larger than the other two, which are called the *transverse dimensions*. Such members are amenable to the simplified structural theories developed in MoM textbooks. We shall study only *straight members* with geometry defined by the two end joints. The member *cross sections* are defined by the intersection of planes normal to the longitudinal dimension with the member. See Figure 6.1. Note that although the individual member will be idealized as being one-dimensional in its intrinsic or local coordinate system, it is generally component of a two- or three-dimensional structure.

This class of structural components embodies bars, beams, beam-columns, shafts and spars. Although geometrically similar, the names distinguish the main kind of internal forces the member resists and transmits: axial forces for bars, bending and shear forces for beams, axial compression and bending for beam-columns, torsion forces for shafts, and shear forces for spars.

Members are connected at their end joints by displacement degrees of freedom. For truss (bar) members those freedoms are the translational components of the joint displacements. For other types, notably beams and shafts, nodal rotations are chosen as additional degrees of freedom.

Structures fabricated with MoM members are generally three-dimensional. Their geometry is defined with respect to a global Cartesian coordinate system $\{x, y, z\}$. Two-dimensional idealizations are useful simplifications should the nature of the geometry and loading allow the reduction of the structural model to one plane of symmetry, which is chosen to be the $\{x, y\}$ plane. Plane trusses and plane frameworks are examples of such simplifications.

¹ Mechanics of Materials was called Strength of Materials in older texts. The scope of this subject includes bars, beams, shafts, arches, thin plates and shells, but only one-dimensional models are covered in introductory undergraduate courses. MoM involves *ab initio* phenomenological assumptions such as “plane sections remain plane” or “shear effects can be neglected in thin beams.” These are stated as the byproduct of two centuries of engineering modeling practice, justified by success. A similar acronym (MOM) is used in Electrical Engineering for something completely different: the Method of Moments.

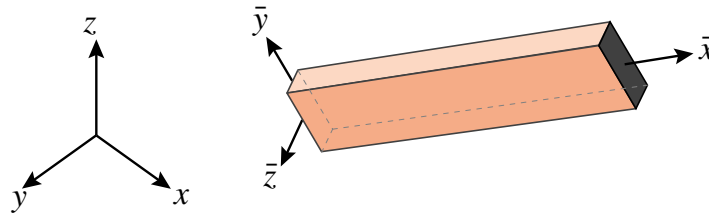


Figure 6.1. A Mechanics of Materials (MoM) member is a structural element one of whose dimensions (the longitudinal dimension) is significantly larger than the other two. Local axes $\{\bar{x}, \bar{y}, \bar{z}\}$ are chosen as indicated. Although the depicted member is prismatic, some applications utilize tapered or stepped members, the cross section of which varies as a function of \bar{x} .

In this Chapter we study generic structural members that fit the preceding class. An individual member is identified by (e) but this superscript will be usually suppressed in the equations below to reduce clutter. The local axes are denoted by $\{\bar{x}, \bar{y}, \bar{z}\}$, with \bar{x} along the longitudinal direction. See Figure 6.1.

The mathematical model of a MoM member is obtained by an idealization process. The model represents the member as a line segment that connects the two end joints, as depicted in Figure 6.2.

§6.2.2. End Quantities, Degrees of Freedom, Joint Forces

The set of mathematical variables used to link members are called *end quantities* or *connectors*. In the Direct Stiffness Method (DSM) these are joint displacements (the degrees of freedom) and the joint forces. These quantities are related by the member stiffness equations.

The degrees of freedom at the end joints i and j are collected in the joint displacement vector $\bar{\mathbf{u}}$. This may include translations only, rotations only, or a combination of translations and rotations.

The vector of joint forces $\bar{\mathbf{f}}$ groups components in one to one correspondence with $\bar{\mathbf{u}}$. Component pairs must be *conjugate* in the sense of the Principle of Virtual Work. For example if the \bar{x} -translation at joint i : \bar{u}_{xi} appears in $\bar{\mathbf{u}}$, the corresponding entry in $\bar{\mathbf{f}}$ is the \bar{x} -force \bar{f}_{xi} at i . If the rotation about \bar{z} at joint j : $\bar{\theta}_{zj}$ appears in $\bar{\mathbf{u}}$, the corresponding entry in $\bar{\mathbf{f}}$ is the \bar{z} -moment \bar{m}_{zj} .

§6.2.3. Internal Quantities

Internal quantities are mechanical actions that take place within the member. Those actions involve stresses and deformations. Accordingly two types of internal quantities appear:

Internal member forces form a finite set of stress-resultant quantities collected in an array \mathbf{p} . They are obtained by integrating stresses over each cross section, and thus are also called generalized stresses in structural mechanics. This set characterizes the forces resisted by the material. Stresses at any point in a section may be recovered if \mathbf{p} is known.

Member deformations form a finite set of quantities, chosen in one-to one correspondence with internal member forces, and collected in an array \mathbf{v} . This set characterizes the deformations experienced by the material. They are also called generalized strains in structural theory. Strains at any point in the member can be recovered if \mathbf{v} is known.

As in the case of end quantities, internal forces and deformations are paired in one to one correspondence. For example, the axial force in a bar member must be paired either with an average

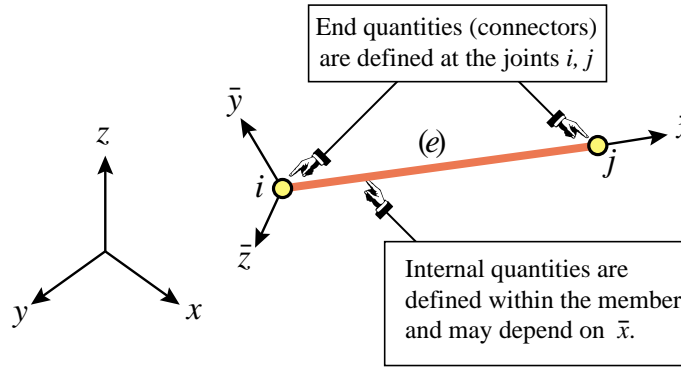


Figure 6.2. The FE mathematical idealization of a MoM member. The model is one-dimensional in \bar{x} . The two end joints are the site of end quantities: joint forces and displacements, that interconnect members. The internal quantities characterize the stresses and deformations in the member.

axial deformation, or with the total elongation. Pairs that mutually correspond in the sense of the Principle of Virtual Work are called *conjugate*. Unlike the case of end quantities, conjugacy is not a mandatory requirement although it simplifies some expressions.

§6.2.4. Discrete Field Equations, Tonti Diagram

The matrix equations that connect $\bar{\mathbf{u}}$, \mathbf{v} , \mathbf{p} and $\bar{\mathbf{f}}$ are called the *discrete field equations*. There are three of them.

The member deformations \mathbf{v} are linked to the joint displacements $\bar{\mathbf{u}}$ by the kinematic compatibility conditions, also called the deformation-displacement or strain-displacement equations:

$$\mathbf{v} = \mathbf{B} \bar{\mathbf{u}}. \quad (6.1)$$

The internal member forces are linked to the member deformations by the constitutive equations. In the absence of initial strain effects those equations are homogeneous:

$$\mathbf{p} = \mathbf{S} \mathbf{v}. \quad (6.2)$$

Finally, the internal member forces are linked to the joint forces by the equilibrium equations. If the internal forces \mathbf{p} are *constant over the member*, the relation is simply

$$\bar{\mathbf{f}} = \mathbf{A}^T \mathbf{p}. \quad (6.3)$$

In (6.3) the transpose of \mathbf{A} is used for convenience.²

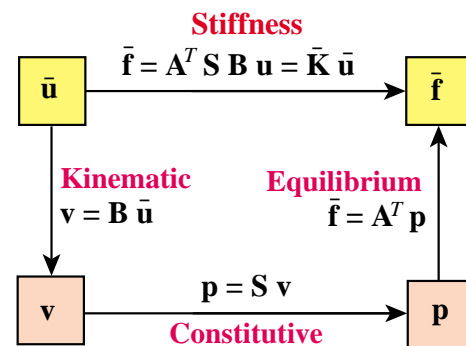


Figure 6.3. Tonti diagram of the three discrete field equations (6.1)–(6.3) and the stiffness equation (6.4) for a *simplex* MoM member. Internal and end quantities appear inside the orange and yellow boxes, respectively.

² If \mathbf{p} is a function of \bar{x} the relation is of differential type. This form is studied in §6.4.

These equations can be presented graphically as shown in Figure 6.3. This is a discrete variant of the so-called *Tonti diagrams*, which represent the governing equations as arrows linking boxes containing kinematic and static quantities. Tonti diagrams for field equations are introduced in Chapter 12.

Matrices **B**, **S** and **A** receive the following names in the literature:

- A** Equilibrium
- S** Rigidity, material, constitutive³
- B** Compatibility, deformation-displacement, strain-displacement

If the element is sufficiently simple, the determination of these three matrices can be carried out through MoM techniques. If the construction requires more advanced tools, however, recourse to the general methodology of finite elements and variational principles is necessary.

§6.3. SIMPLEX MOM MEMBERS

Throughout this section we assume that the *internal quantities are constant over the member length*. Such members are called *simplex elements*. If so the matrices **A**, **B** and **S** are independent of member cross section. The derivation of the element stiffness equations reduces to a straightforward sequence of matrix multiplications.

Under the constancy-along- \bar{x} assumption, elimination of the interior quantities **p** and **v** from (6.1)-(6.3) yields the element stiffness relation

$$\bar{\mathbf{f}} = \mathbf{A}^T \mathbf{S} \mathbf{B} \bar{\mathbf{u}} = \bar{\mathbf{K}} \bar{\mathbf{u}}, \quad (6.4)$$

whence the element stiffness matrix is

$$\boxed{\bar{\mathbf{K}} = \mathbf{A}^T \mathbf{S} \mathbf{B}.} \quad (6.5)$$

The four preceding matrix equations are diagrammed in the diagram of Figure 6.3.

If $\{\mathbf{p}, \mathbf{v}\}$ and $\{\bar{\mathbf{f}}, \bar{\mathbf{u}}\}$ are conjugate in the sense of the Principle of Virtual Work, it can be shown that $\mathbf{A} = \mathbf{B}$ and that **S** is symmetric. Then

$$\boxed{\bar{\mathbf{K}} = \mathbf{B}^T \mathbf{S} \mathbf{B}.} \quad (6.6)$$

is a symmetric matrix. Symmetry is computationally desirable for reasons outlined in Part III.

REMARK 6.1

If $\bar{\mathbf{f}}$ and $\bar{\mathbf{u}}$ are conjugate (as required in §6.2.2) but **p** and **v** are not, $\bar{\mathbf{K}}$ must come out to be symmetric even if **S** is unsymmetric and $\mathbf{A} \neq \mathbf{B}$. However there are more opportunities to go wrong.

³ The name *rigidity matrix* for **S** is preferable. It is a member integrated version of the cross section constitutive equations. The latter are usually denoted by symbol **R**, as in §6.4.

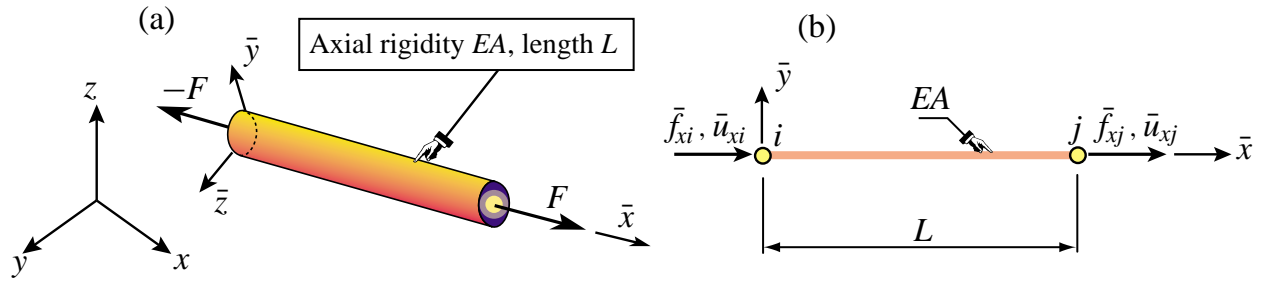


Figure 6.4. The prismatic bar (also called truss) member: (a) individual member shown in 3D space, (b) idealization as generic member.

§6.3.1. The Bar Element Revisited

The simplest MoM element is the prismatic bar or truss member already derived in Chapter 2. See Figure 6.4. This qualifies as simplex because all internal quantities are constant. One minor difference in the derivation below is that the joint displacements and forces in the \bar{y} direction are omitted in the generic element since they contribute nothing to the stiffness equations. In the FEM terminology, freedoms associated with zero stiffness are called *inactive*.

Three choices for internal deformation and force variables are considered. The results confirm that the element stiffness equations coalesce, as expected, since the external quantities are the same.

Derivation Using Axial Elongation and Axial Force. The member axial elongation d is taken as deformation measure, and the member axial force F as internal force measure. Hence \mathbf{v} and \mathbf{p} reduce to the scalars $v = d$ and $p = F$, respectively. The chain of discrete field equations is easily constructed:

$$d = [-1 \quad 1] \begin{bmatrix} \bar{u}_{xi} \\ \bar{u}_{xj} \end{bmatrix} = \mathbf{B}\bar{\mathbf{u}}, \quad F = \frac{EA}{L}d = Sd, \quad \bar{\mathbf{f}} = \begin{bmatrix} \bar{f}_{xi} \\ \bar{f}_{xj} \end{bmatrix} = \begin{bmatrix} -1 \\ 1 \end{bmatrix} F = \mathbf{A}^T F. \quad (6.7)$$

Consequently

$$\bar{\mathbf{K}} = \mathbf{A}^T S \mathbf{B} = S \mathbf{B}^T \mathbf{B} = S [-1 \quad 1] \begin{bmatrix} -1 \\ 1 \end{bmatrix} = \frac{EA}{L} \begin{bmatrix} 1 & -1 \\ -1 & 1 \end{bmatrix}. \quad (6.8)$$

Note that $\mathbf{A} = \mathbf{B}$ because F and d are conjugate.

Derivation Using Mean Axial Strain and Axial Force. Instead of d we may use the mean axial strain $\bar{\epsilon} = d/L$ as deformation measure whereas F is kept as internal force measure. The only change is that \mathbf{B} becomes $[-1 \quad 1]/L$ whereas S becomes EA . Matrix \mathbf{A} does not change. The product $\mathbf{A}^T S \mathbf{B}$ gives the same $\bar{\mathbf{K}}$ as in (6.8), as can be expected. Now \mathbf{A}^T is not equal to \mathbf{B} because F and $\bar{\epsilon}$ are not conjugate, but they differ only by a factor $1/L$.

Derivation Using Mean Axial Strain and Axial Stress. We keep the mean axial strain $\bar{\epsilon} = d/L$ as deformation measure but the mean axial stress $\bar{\sigma} = F/A$, (which is *not* conjugate to $\bar{\epsilon}$) is taken as internal force measure. Now $\mathbf{B} = [-1 \quad 1]/L$, $S = E$ and $\mathbf{A}^T = A [-1 \quad 1]$. The product $\mathbf{A}^T S \mathbf{B}$ gives again the same $\bar{\mathbf{K}}$ shown in (6.8).

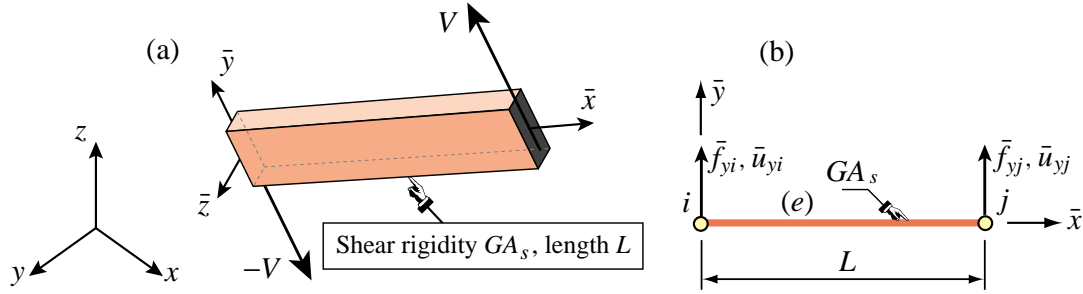


Figure 6.5. The prismatic spar (also called shear-web) member: (a) individual member shown in 3D space, (b) idealization as generic member.

Transformation to Global Coordinates. Since \bar{u}_{yi} and \bar{u}_{yj} are not part of (6.7) and (6.8) the displacement transformation matrix from local to global $\{x, y\}$ coordinates is 2×4 , instead of 4×4 as in §3.1. On restoring the element identifier (e) the appropriate local-to-global transformation is

$$\bar{\mathbf{u}}^{(e)} = \begin{bmatrix} \bar{u}_{xi}^{(e)} \\ \bar{u}_{xj}^{(e)} \end{bmatrix} = \begin{bmatrix} c & s & 0 & 0 \\ 0 & 0 & c & s \end{bmatrix} \begin{bmatrix} u_{xi}^{(e)} \\ u_{yi}^{(e)} \\ u_{xj}^{(e)} \\ u_{yj}^{(e)} \end{bmatrix} = \mathbf{T}^{(e)} \mathbf{u}^{(e)}, \quad (6.9)$$

where $c = \cos \varphi^{(e)}$, $s = \sin \varphi^{(e)}$, and $\varphi^{(e)}$ is the angle from x to \bar{x} , cf. Figure 3.2. The 4×4 globalized element stiffness matrix $\mathbf{K}^{(e)} = (\mathbf{T}^{(e)})^T \bar{\mathbf{K}}^{(e)} \mathbf{T}^{(e)}$ agrees with (3.7).

§6.3.2. The Spar Element

The *spar* or *shear-web* member has two joints (end nodes): i and j . This member can only resist and transmit a *constant shear force* V in the plane of the web, which is chosen to be the $\{\bar{x}, \bar{y}\}$ plane. See Figure 6.5. It is often used for modeling high-aspect aircraft wing structures, as illustrated in Figure 6.6. We consider here only *prismatic* spar members of uniform material and constant cross section, which thus qualify as simplex.

The active degrees of freedom for a generic spar member of length L , as depicted in Figure 6.5(b), are \bar{u}_{yi} and \bar{u}_{yj} . Let G be the shear modulus and A_s the effective shear area. (A concept developed in Mechanics of Materials; for a narrow rectangular cross section, $A_s = 5A/6$.) The shear rigidity is GA_s . As deformation measure the mean shear strain $\gamma = V/(GA_s)$ is chosen.

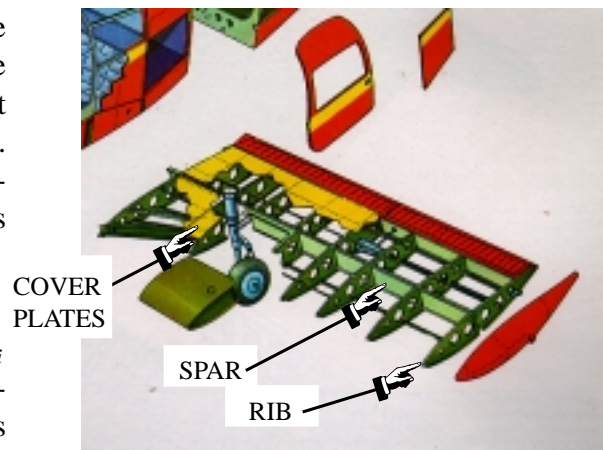


Figure 6.6. Spar members in aircraft wing (Piper Cherokee). For more impressive aircraft structures see CATECS slides.

The kinematic, constitutive, and equilibrium equations are

$$\gamma = \frac{1}{L} \begin{bmatrix} -1 & 1 \end{bmatrix} \begin{bmatrix} \bar{u}_{yi} \\ \bar{u}_{yj} \end{bmatrix} = \mathbf{B}\bar{\mathbf{u}}, \quad V = GA_s \gamma = S\gamma, \quad \bar{\mathbf{f}} = \begin{bmatrix} \bar{f}_{yi} \\ \bar{f}_{yj} \end{bmatrix} = \begin{bmatrix} -1 \\ 1 \end{bmatrix} V = \mathbf{A}^T V, \quad (6.10)$$

whence the local stiffness equations are

$$\bar{\mathbf{f}} = \begin{bmatrix} \bar{f}_{yi} \\ \bar{f}_{yj} \end{bmatrix} = \mathbf{A}^T S \mathbf{B}\bar{\mathbf{u}} = \frac{GA_s}{L} \begin{bmatrix} 1 & -1 \\ -1 & 1 \end{bmatrix} \begin{bmatrix} \bar{u}_{yi} \\ \bar{u}_{yj} \end{bmatrix} = \bar{\mathbf{K}}\bar{\mathbf{u}}. \quad (6.11)$$

Note that $\mathbf{A} \neq \mathbf{B}$ as V and γ are not work-conjugate. This difference is easily adjusted for, however; see Exercise 6.1.

If the spar member is used in a two dimensional context, the displacement transformation from local to global coordinates $\{x, y\}$ is

$$\bar{\mathbf{u}}^{(e)} = \begin{bmatrix} \bar{u}_{yi}^{(e)} \\ \bar{u}_{yj}^{(e)} \end{bmatrix} = \begin{bmatrix} -s & c & 0 & 0 \\ 0 & 0 & -s & c \end{bmatrix} \begin{bmatrix} u_{xi}^{(e)} \\ u_{yi}^{(e)} \\ u_{xj}^{(e)} \\ u_{yj}^{(e)} \end{bmatrix} = \mathbf{T}^{(e)} \mathbf{u}^{(e)}, \quad (6.12)$$

where $c = \cos \varphi^{(e)}$, $s = \sin \varphi^{(e)}$, and $\varphi^{(e)}$ is the angle from x to \bar{x} , cf. Figure 3.2. The 4×4 globalized spar stiffness matrix is then $\mathbf{K}^{(e)} = (\mathbf{T}^{(e)})^T \bar{\mathbf{K}}^{(e)} \mathbf{T}^{(e)}$.

More often, however, the spar member will be a component in a three-dimensional structural model, such as the aircraft wing depicted in Figure 6.6. If so the determination of the 2×6 transformation matrix is more involved. This is the topic of Exercise 6.5.

§6.3.3. The Shaft Element

The *shaft*, also called *torque member*, has two joints (end nodes): i and j . A shaft can only resist and transmit a *constant torque* or *twisting moment* T along its longitudinal axis \bar{x} , as pictured in Figure 6.7(a).

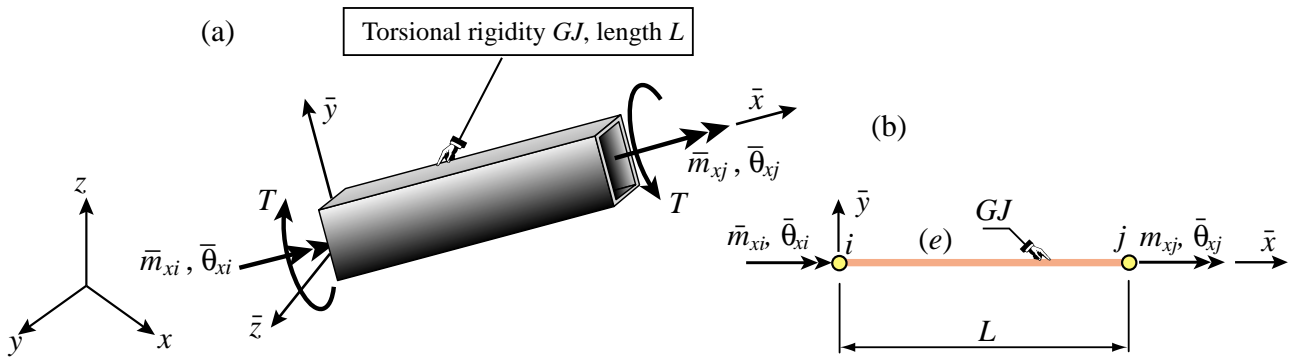


Figure 6.7. The prismatic shaft (also called torque member): (a) individual member shown in 3D space, (b) idealization as generic member.

We consider here only *prismatic* shaft members with uniform material and constant cross section, which thus qualify as simplex. The active degrees of freedom of a generic shaft member of length L , depicted in Figure 6.7(b), are $\bar{\theta}_{xi}$ and $\bar{\theta}_{xj}$. These are the infinitesimal end rotations about \bar{x} , positive according to the right-hand rule. The associated joint (node) forces are end moments denoted as \bar{m}_{xi} and \bar{m}_{xj} .

The only internal force is the torque T , positive if acting as pictured in Figure 6.7(a). Let G be the shear modulus and GJ the effective torsional rigidity.⁴ As deformation measure pick the relative twist angle $\phi = \bar{\theta}_{xj} - \bar{\theta}_{xi}$. The kinematic, constitutive, and equilibrium equations provided by Mechanics of Materials are

$$\phi = \begin{bmatrix} -1 & 1 \end{bmatrix} \begin{bmatrix} \bar{\theta}_{yi} \\ \bar{\theta}_{yj} \end{bmatrix} = \mathbf{B}\bar{\mathbf{u}}, \quad T = \frac{GJ}{L}\phi = S\gamma, \quad \bar{\mathbf{f}} = \begin{bmatrix} \bar{m}_{xi} \\ \bar{m}_{xj} \end{bmatrix} = \begin{bmatrix} -1 \\ 1 \end{bmatrix} T = \mathbf{B}^T T. \quad (6.13)$$

From these the local stiffness equations follow as

$$\bar{\mathbf{f}} = \begin{bmatrix} \bar{m}_{yi} \\ \bar{m}_{yj} \end{bmatrix} = \mathbf{B}^T S \mathbf{B} \bar{\mathbf{u}} = \frac{GJ}{L} \begin{bmatrix} 1 & -1 \\ -1 & 1 \end{bmatrix} \begin{bmatrix} \bar{\theta}_{yi} \\ \bar{\theta}_{yj} \end{bmatrix} = \bar{\mathbf{K}} \bar{\mathbf{u}}. \quad (6.14)$$

If the shaft is used in a two-dimensional context, the displacement transformation to global coordinates $\{x, y\}$ within the framework of infinitesimal rotations, is

$$\bar{\mathbf{u}}^{(e)} = \begin{bmatrix} \bar{\theta}_{xi}^{(e)} \\ \bar{\theta}_{xj}^{(e)} \end{bmatrix} = \begin{bmatrix} c & s & 0 & 0 \\ 0 & 0 & c & s \end{bmatrix} \begin{bmatrix} \theta_{xi}^{(e)} \\ \theta_{yi}^{(e)} \\ \theta_{xj}^{(e)} \\ \theta_{yj}^{(e)} \end{bmatrix} = \mathbf{T}^{(e)} \boldsymbol{\theta}^{(e)}, \quad (6.15)$$

where as usual $c = \cos \varphi^{(e)}$, $s = \sin \varphi^{(e)}$, and $\varphi^{(e)}$ is the angle from x to \bar{x} . Note that $\boldsymbol{\theta}^{(e)}$ collects only global node rotations components. This operation is elaborated further in Exercise 6.3.

§6.4. *NON-SIMPLEX MOM MEMBERS

The straightforward formulation of simplex MoM elements does not immediately carry over to the case in which internal quantities \mathbf{p} and \mathbf{v} vary over the member; that is, depend on \bar{x} . The dependence may be due to element type, varying cross section, or both. Thus one or more of the matrices \mathbf{A} , \mathbf{B} and \mathbf{S} depend on \bar{x} . Such members are called *non-simplex*.

The matrix multiplication recipe (6.4) cannot be used to construct the element stiffness matrix $\bar{\mathbf{K}}$ of non-simplex members. This can be grasped by observing that $\mathbf{A}(\bar{x})^T \mathbf{S}(\bar{x}) \mathbf{B}(\bar{x})$ would depend on \bar{x} . On the other hand, $\bar{\mathbf{K}}$ must be independent of \bar{x} because it relates the end quantities $\bar{\mathbf{u}}$ and $\bar{\mathbf{f}}$.

⁴ J has dimension of (length)⁴. For a circular or annular cross section it reduces to the polar moment of inertia about \bar{x} . The determination of J for noncircular cross sections is covered in Mechanics of Materials textbooks.

§6.4.1. *Formulation Rules

The derivation of non-simplex MoM elements requires use of the work principles of mechanics, for example the Principle of Virtual Work or PVW. Thus, more care must be exercised in the choice of conjugate internal quantities. The following rules can be justified through the arguments presented in Part II. They are stated here as recipe, and apply only to displacement-assumed elements.

Rule 1. Select internal deformations $\mathbf{v}(\bar{x})$ and internal forces $\mathbf{p}(\bar{x})$ that are conjugate in the PVW sense. Link deformations to node displacements by $\mathbf{v}(\bar{x}) = \mathbf{B}(\bar{x})\mathbf{u}$.

Rule 2. From the PVW it may be shown⁵ that the force equilibrium equation exists only in a differential sense:

$$\mathbf{B}^T d\mathbf{p} = d\bar{\mathbf{f}}. \quad (6.16)$$

Here d denotes differentiation with respect to \bar{x} .⁶

The interpretation of $d\bar{\mathbf{f}}$ is less immediate because $\bar{\mathbf{f}}$ is not a function of \bar{x} . It actually means the contribution of that member slice to the building of the node force vector $\bar{\mathbf{f}}$. See (6.18) and (6.19) below.

Rule 3. The constitutive relation is

$$\mathbf{p} = \mathbf{R}\mathbf{v}, \quad (6.17)$$

in which \mathbf{R} , which may depend on \bar{x} , must be symmetric. Note that symbol \mathbf{R} in (6.17) replaces the \mathbf{S} of (6.2). \mathbf{R} pertains to a specific cross section whereas \mathbf{S} applies to the entire member. This distinction is further elaborated in Exercise 6.9.

The discrete relations supplied by the three foregoing rules are displayed in the discrete Tonti diagram of Figure 6.8.

Internal quantities are now eliminated starting from the differential equilibrium relation (6.16):

$$d\bar{\mathbf{f}} = \mathbf{B}^T d\mathbf{p} = \mathbf{B}^T \mathbf{p} d\bar{x} = \mathbf{B}^T \mathbf{R} \mathbf{v} d\bar{x} = \mathbf{B}^T \mathbf{R} \mathbf{B} \bar{\mathbf{u}} d\bar{x} = \mathbf{B}^T \mathbf{R} \mathbf{B} d\bar{x} \bar{\mathbf{u}}. \quad (6.18)$$

Integrating both sides over the member length L yields

$$\bar{\mathbf{f}} = \int_0^L d\bar{\mathbf{f}} = \int_0^L \mathbf{B}^T \mathbf{R} \mathbf{B} d\bar{x} \bar{\mathbf{u}} = \bar{\mathbf{K}} \bar{\mathbf{u}}, \quad (6.19)$$

because $\bar{\mathbf{u}}$ does not depend on \bar{x} . Consequently the local element stiffness matrix is

$$\bar{\mathbf{K}} = \int_0^L \mathbf{B}^T \mathbf{R} \mathbf{B} d\bar{x} \quad (6.20)$$

The recipe (6.20) will be justified in Part II through energy methods. It will be seen that it generalizes to arbitrary displacement-assumed finite elements in any number of space dimensions. It is used in the derivation of the stiffness equations of the plane beam element in Chapter 13. The reduction of (6.20) to (6.6) when the dependence on \bar{x} disappears is the subject of Exercise 6.8.

⁵ The proof follows by equating the virtual work of a slice of length $d\bar{x}$: $d\bar{\mathbf{f}}^T \cdot \delta \bar{\mathbf{u}} = d\mathbf{p}^T \cdot \delta \mathbf{v} = d\mathbf{p}^T \cdot (\mathbf{B} \bar{\mathbf{u}}) = (\mathbf{B}^T \cdot d\mathbf{p})^T \cdot \delta \bar{\mathbf{u}}$. Since $\delta \bar{\mathbf{u}}$ is arbitrary, $\mathbf{B}^T d\mathbf{p} = d\bar{\mathbf{f}}$.

⁶ The meaning of $d\mathbf{p}$ is simply $\mathbf{p}(\bar{x}) d\bar{x}$. That is, the differential of internal forces as one passes from cross-section \bar{x} to a neighboring one $\bar{x} + d\bar{x}$.

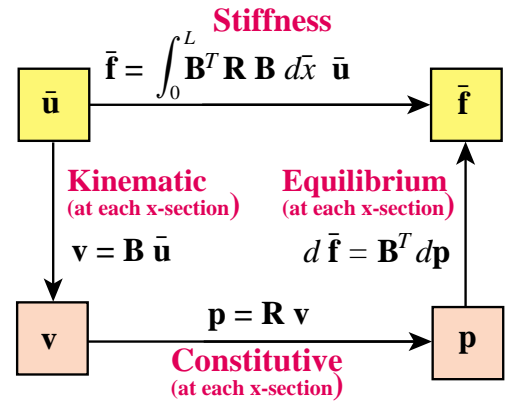


Figure 6.8. Discrete Tonti diagram of the discrete field equations for a non-simplex MoM member.

§6.4.2. *Example: Bar with Variable Cross Section

A two-node bar element has constant E but a continuously varying cross section area: A_i at joint i , A_j at joint j and A_m at the midpoint between i and j . This can be parabolically fit by as $A(\bar{x}) = A_i N_i(\bar{x}) + A_j N_j(\bar{x}) + A_m N_m(\bar{x})$, where $N_i(\bar{x}) = -\frac{1}{2}\xi(1 - \xi)$, $N_j(\bar{x}) = \frac{1}{2}\xi(1 + \xi)$ and $N_m(\bar{x}) = 1 - \xi^2$, with $\xi = 2x/L - 1$, are interpolating polynomials. As internal quantities take the strain e and the axial force $p = EAe$, which are conjugate quantities.

Assuming the strain e to be uniform over the element⁷ the MoM equations are

$$e = \mathbf{B}\bar{\mathbf{u}}, \quad p = EA(\bar{x})e = R(\bar{x})e, \quad d\bar{\mathbf{f}} = \mathbf{B}^T d\mathbf{p}, \quad \mathbf{B} = \frac{1}{L} \begin{bmatrix} -1 & 1 \end{bmatrix}. \quad (6.21)$$

Inserting into (6.20) and carrying out the integration yields

$$\bar{\mathbf{K}} = \frac{E\bar{A}}{L} \begin{bmatrix} 1 & -1 \\ -1 & 1 \end{bmatrix}, \quad \text{with} \quad \bar{A} = \frac{1}{6}(A_i + A_j) + \frac{2}{3}A_m. \quad (6.22)$$

Notes and Bibliography

The derivation of MoM elements using straightforward matrix algebra is typical of pre-1960 Matrix Structural Analysis (MSA). The book of Pestel and Leckie [6.3], unfortunately out of print, epitomizes that approach, which historically interweaved with Generation 1 of FEM.⁸ By 1970 simplified derivations had fallen out of favor as yokelish. But these elements do not need improvement. They still work fine: a bar stiffness today is the same as 40 years ago.

The Mechanics of Materials books by Beer-Johnston [6.1] and Popov [6.4] can be cited as being widely used in US undergraduate courses. But they are not the only ones. A September 2003 in-print book search through www3.addall.com on “Mechanics of Materials” returns 99 hits whereas one on “Strength of Materials” (the older name) compiles 112. Folding editions and paper/softback variants one gets about 60 books; by all accounts an impressive number.

Spar members are discussed only in MoM books focusing on aircraft structures, since they are primarily used in modeling shear web action. On the other hand, bars and shafts are standard fare.

The framework presented here is a small part of MSA. A panoramic view, including linkage to continuum formulations from the MSA viewpoint, is presented in [6.2].

The source of Tonti diagrams is discussed in Chapter 12.

References

- [6.1] Beer, F. P. and Johnston E. R., *Mechanics of Materials*, McGraw-Hill, 2nd ed. 1992.
- [6.2] Felippa, C. A., Parametrized unification of matrix structural analysis: classical formulation and d-connected mixed elements, *Finite Elements Anal. Des.*, **21**, 45–74, 1995.
- [6.3] Pestel, E. C., Leckie, F. A., *Matrix Methods in Elastomechanics*, McGraw-Hill, New York, 1963.
- [6.4] Popov, E. P., *Engineering Mechanics of Solids*, Prentice Hall, Englewood Cliffs, N. J., 2nd ed., 1991.

⁷ This is characteristic of a displaceant assumed element and is justified through the method of shape functions explained in Part II.

⁸ A generational outline is provided in §1.5.2.

Homework Exercises for Chapter 6

Constructing MoM Members

EXERCISE 6.1

[A:10] Explain how to select the deformation variable v (paired to V) of the spar member formulated in §6.3.2, so that $\mathbf{A} = \mathbf{B}$. Draw the Tonti diagram, as in Figure 6.3 but with the actual equations, for this choice.

EXERCISE 6.2

[A:15] Obtain the 4×4 global element stiffness matrix of a prismatic spar member in a two dimensional Cartesian system $\{x, y\}$. Start from (6.11). Justify the transformation (6.12). Evaluate $\mathbf{K}^{(e)} = (\mathbf{T}^{(e)})^T \bar{\mathbf{K}}^{(e)} \mathbf{T}^{(e)}$ in closed form.

EXERCISE 6.3

[A:15] Obtain the 4×4 global element stiffness matrix of a prismatic shaft element in a two dimensional Cartesian system $\{x, y\}$. Include only node rotation freedoms in the global displacement vector. Start from (6.14). Justify the transformation (6.15) (Hint: infinitesimal rotations transform as vectors). Evaluate $\mathbf{K}^{(e)} = (\mathbf{T}^{(e)})^T \bar{\mathbf{K}}^{(e)} \mathbf{T}^{(e)}$ in closed form.

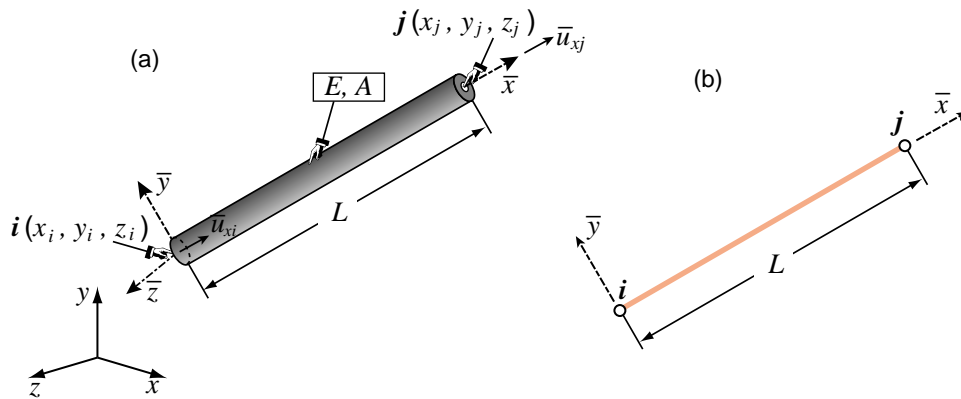


Figure E6.1. Bar element in 3D for Exercise 6.4.

EXERCISE 6.4

[A+N:15(10+5)] A bar element moving in three dimensional space is completely defined by the global coordinates $\{x_i, y_i, z_i\}, \{x_j, y_j, z_j\}$ of its end nodes i and j , as illustrated in Figure E6.1. The 2×6 displacement transformation matrix \mathbf{T} , with superscript (e) dropped for brevity, links $\bar{\mathbf{u}}^{(e)} = \mathbf{T}\mathbf{u}^{(e)}$. Here $\bar{\mathbf{u}}^{(e)}$ contains the two local displacements \bar{u}_{xi} and \bar{u}_{xj} whereas $\mathbf{u}^{(e)}$ contains the six global displacements $u_{xi}, u_{yi}, u_{zi}, u_{xj}, u_{yj}, u_{zj}$.

(a) From vector mechanics show that

$$\mathbf{T} = \frac{1}{L} \begin{bmatrix} x_{ji} & y_{ji} & z_{ji} & 0 & 0 & 0 \\ 0 & 0 & 0 & x_{ji} & y_{ji} & z_{ji} \end{bmatrix} = \begin{bmatrix} c_{xji} & c_{yji} & c_{zji} & 0 & 0 & 0 \\ 0 & 0 & 0 & c_{xji} & c_{yji} & c_{zji} \end{bmatrix} \quad (\text{E6.1})$$

in which L is the element length, $x_{ji} = x_j - x_i$, etc., and $c_{xji} = x_{ji}/L$, etc., are the direction cosines of the vector going from i to j .

(b) Evaluate \mathbf{T} for a bar going from node i at $\{1, 2, 3\}$ to node j at $\{3, 8, 6\}$.

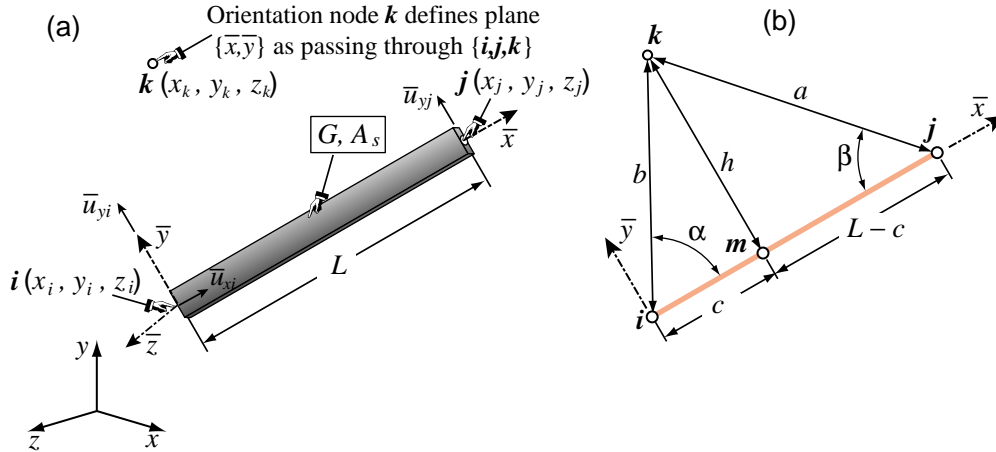


Figure E6.2. Spar element in 3D for Exercise 6.5.

EXERCISE 6.5

[A+N:30 (10+15+5)] A spar element in three dimensional space is only partially defined by the global coordinates $\{x_i, y_i, z_i\}$, $\{x_j, y_j, z_j\}$ of its end nodes i and j , as illustrated in Figure E6.2. The problem is that axis \bar{y} , which defines the direction of shear force transmission, is not uniquely defined by i and j .⁹ Most FEM programs use the *orientation node* method to complete the definition. A third node k , not colinear with i and j , is provided by the user. Nodes $\{i, j, k\}$ define the $\{\bar{x}, \bar{y}\}$ plane and thus \bar{z} . The projection of k on line ij is point m . The distance $h > 0$ from m to k is called h as shown in Figure E6.2(b). The 2×6 displacement transformation matrix \mathbf{T} , with superscript (e) omitted to reduce clutter, relates $\bar{\mathbf{u}}^{(e)} = \mathbf{T}\mathbf{u}^{(e)}$. Here $\bar{\mathbf{u}}^{(e)}$ contains the local transverse displacements \bar{u}_{yi} and \bar{u}_{yj} whereas $\mathbf{u}^{(e)}$ contains the global displacements $u_{xi}, u_{yi}, u_{zi}, u_{xj}, u_{yj}, u_{zj}$.

(a) Show that

$$\mathbf{T} = \frac{1}{h} \begin{bmatrix} x_{km} & y_{km} & z_{km} & 0 & 0 & 0 \\ 0 & 0 & 0 & x_{km} & y_{km} & z_{km} \end{bmatrix} = \begin{bmatrix} c_{xkm} & c_{ykm} & c_{zkm} & 0 & 0 & 0 \\ 0 & 0 & 0 & c_{xkm} & c_{ykm} & c_{zkm} \end{bmatrix} \quad (\text{E6.2})$$

in which $x_{km} = x_k - x_m$, etc., and $c_{xkm} = x_{km}/h$, etc., are the direction cosines of the vector going from m to k .

- (b) Work out the formulas to compute the coordinates of point m in terms of the coordinates of $\{i, j, k\}$. Using the notation of Figure E6.2(b) and elementary trigonometry, show that $h = 2A/L$, where $A = \sqrt{p(p-a)(p-b)(p-L)}$ with $p = \frac{1}{2}(L+a+b)$ (Heron's formula), $\cos \alpha = (L^2 + b^2 - a^2)/(2bL)$, $\cos \beta = (L^2 + a^2 - b^2)/(2aL)$, $c = b \cos \alpha$, $L - c = a \cos \beta$, $x_m = x_i(L-c)/L + x_j c/L$, etc.¹⁰
- (c) Evaluate \mathbf{T} for a spar member going from node i at $\{1, 2, 3\}$ to node j at $\{3, 8, 6\}$. with k at $\{4, 5, 6\}$.

EXERCISE 6.6

[A:20] Explain how thermal effects can be generally incorporated in the constitutive equation (6.2) to produce an initial force vector for a simplex element.

⁹ The same ambiguity arises in 3D beam elements, which are treated in Part III.

¹⁰ An alternative and more elegant procedure, found by a student in 1999, can be sketched as follows. From Figure E6.2(b) obviously the two subtriangles imk and jkm are right-angled at m and share side km of length h . Apply Pythagoras' theorem twice, and subtract so as to cancel out h^2 and c^2 , getting a linear equation for c that can be solved directly.

EXERCISE 6.7

[A:15] Draw the discrete Tonti diagram for the prismatic shaft element.

EXERCISE 6.8

[A:15] If the matrices \mathbf{B} and \mathbf{R} are constant over the element length L , show that expression (6.20) of the element stiffness matrix for a non-simplex member reduces to (6.6), in which $\mathbf{S} = L\mathbf{R}$.

EXERCISE 6.9

[A:20] Explain in detail the quickie derivation of footnote 5. (Knowledge of the Principle of Virtual Work is required to do this exercise.)

EXERCISE 6.10

[A:25(10+5+10)] Consider a non-simplex element in which \mathbf{R} varies with \bar{x} but $\mathbf{B} = \mathbf{A}$ is constant.

(a) From (6.20) prove that

$$\bar{\mathbf{K}} = L\mathbf{B}^T \bar{\mathbf{R}} \mathbf{B}, \quad \text{with} \quad \bar{\mathbf{R}} = \frac{1}{L} \int_0^L \mathbf{R}(\bar{x}) d\bar{x} \quad (\text{E6.3})$$

(b) Apply (E6.3) to obtain $\bar{\mathbf{K}}$ for a tapered bar with area defined by the linear law $A = A_i(1 - \bar{x}/L) + A_j\bar{x}/L$, where A_i and A_j are the end areas at i and j , respectively. Take $\mathbf{B} = [-1 \quad 1]/L$.

(c) Apply (E6.3) to verify the result (6.22) for a bar with parabolically varying cross section.

EXERCISE 6.11

[A/C+N:30(25+5)] A prismatic bar element in 3D space is referred to a global coordinate system $\{x, y, z\}$, as in Figure E6.1. The end nodes are located at $\{x_1, y_1, z_1\}$ and $\{x_2, y_2, z_2\}$.¹¹ The elastic modulus E and the cross section area A are constant along the length. Denote $x_{21} = x_2 - x_1$, $y_{21} = y_2 - y_1$, $z_{21} = z_2 - z_1$ and $L = \sqrt{x_{21}^2 + y_{21}^2 + z_{21}^2}$.

(a) Show that the element stiffness matrix in *global* coordinates can be compactly written¹²

$$\mathbf{K}^{(e)} = \frac{EA}{L^3} \mathbf{B}^T \mathbf{B}, \quad \text{in which} \quad \mathbf{B} = [-x_{21} \quad -y_{21} \quad -z_{21} \quad x_{21} \quad y_{21} \quad z_{21}]. \quad (\text{E6.4})$$

(b) Compute $\mathbf{K}^{(e)}$ if the nodes are at $\{1, 2, 3\}$ and $\{3, 8, 6\}$, with elastic modulus $E = 343$ and cross section area $A = 1$. Note: the computation can be either done by hand or with the help of a program such as the following *Mathematica* module, which is used in Part III:

¹¹ End nodes are labeled 1 and 2 instead of i and j to agree with the code listed below.

¹² There are several ways of arriving at this result. Some are faster and more elegant than others. Here is a sketch of one of the ways. Denote by L_0 and L the lengths of the bar in the undeformed and deformed configurations, respectively. Then

$$\frac{1}{2}(L^2 - L_0^2) = x_{21}(u_{x2} - u_{x1}) + y_{21}(u_{y2} - u_{y1}) + z_{21}(u_{z2} - u_{z1}) + Q \approx \mathbf{B}\mathbf{u},$$

in which Q is a quadratic function of node displacements which is therefore dropped in the small-displacement linear theory. Also on account of small displacements

$$\frac{1}{2}(L^2 - L_0^2) = \frac{1}{2}(L + L_0)(L - L_0) \approx L \Delta L.$$

Hence the small axial strain is $e = \Delta L/L = (1/L^2)\mathbf{B}\mathbf{u}^{(e)}$, which begins the Tonti diagram. Next is $F = EAe$. Finally you must show that force equilibrium at nodes requires $\mathbf{f}^{(e)} = (1/L)\mathbf{B}^T F$. Multiplying through gives (E6.4). A plodding way is to start from the local stiffness (6.8) and transform to global using (E6.1).

```

Stiffness3DBar[ncoor_,mprop_,fprop_,opt_] := Module[
  {x1,x2,y1,y2,z1,z2,x21,y21,z21,Em,Gm,rho,alpha,A,
   num,L,LL,LLL,B,Ke},  {{x1,y1,z1},{x2,y2,z2}}=ncoor;
  {x21,y21,z21}={x2-x1,y2-y1,z2-z1};
  {Em,Gm,rho,alpha}=mprop; {A}=fprop; {num}=opt;
  If [num,{x21,y21,z21,Em,A}=N[{x21,y21,z21,Em,A}]];
  LL=x21^2+y21^2+z21^2; L=PowerExpand[Sqrt[LL]];
  LLL=Simplify[LL*L]; B={{-x21,-y21,-z21,x21,y21,z21}};
  Ke=(Em*A/LLL)*Transpose[B].B;
Return[Ke]];

ClearAll[Em,A]; Em=343; A=1;
ncoor={{0,0,0},{2,6,3}}; mprop={Em,0,0,0}; fprop={A}; opt={False};
Ke=Stiffness3DBar[ncoor,mprop,fprop,opt];
Print["Stiffness of 3D Bar Element:"];
Print[Ke//MatrixForm];
Print["eigs of Ke: ",Eigenvalues[Ke]];

```

As a check, the six eigenvalues of this particular $\mathbf{K}^{(e)}$ should be 98 and five zeros.

7

FEM Modeling: Introduction

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Chapters 2 through 6 cover material technically known as Matrix Structural Analysis or MSA. This is a subject that historically preceded the Finite Element Method (FEM), as chronicled in Appendix H. Here we begin the coverage of the FEM proper. This is technically distinguished from MSA by the more dominant role of continuum and variational mechanics.

This Chapter introduces terminology used in FEM modeling, and surveys the attributes and types of finite elements used in structural mechanics. The next Chapter gives more specific rules for defining meshes, forces and boundary conditions.

§7.1. FEM TERMINOLOGY

The ubiquitous term “degrees of freedom,” often abbreviated to either freedom or DOF, has figured prominently in the preceding Chapters. This term, as well as “stiffness matrix” and “force vector,” originated in structural mechanics, the application for which FEM was invented. These names have carried over to non-structural applications. This “terminology overspill” is discussed next.

Classical analytical mechanics is that invented by Euler and Lagrange in the XVIII century and further developed by Hamilton and Jacobi as a systematic formulation of Newtonian mechanics. Its objects of attention are models of mechanical systems ranging from material particles composed of sufficiently large number of molecules, through airplanes, to the Solar System.¹ The spatial configuration of any such system is described by its *degrees of freedom*. These are also called *generalized coordinates*. The terms *state variables* and *primary variables* are also used, particularly in mathematically oriented treatments.

If the number of degrees of freedom is finite, the model is called *discrete*, and *continuous* otherwise. Because FEM is a discretization method, the number of degrees of freedom of a FEM model is necessarily finite. The freedoms are collected in a column vector called \mathbf{u} . This vector is called the *DOF vector* or *state vector*. The term *nodal displacement vector* for \mathbf{u} is reserved to mechanical applications.

In analytical mechanics, each degree of freedom has a corresponding “conjugate” or “dual” term, which represents a generalized force.² In non-mechanical applications, there is a similar set of conjugate quantities, which for want of a better term are also called *forces* or *forcing terms*. They are the agents of change. These forces are collected in a column vector called \mathbf{f} . The inner product $\mathbf{f}^T \mathbf{u}$ has the meaning of external energy or work.

Just as in the truss problem, the relation between \mathbf{u} and \mathbf{f} is assumed to be of linear and homogeneous. The last assumption means that if \mathbf{u} vanishes so does \mathbf{f} . The relation is then expressed by the master stiffness equations:

$$\boxed{\mathbf{K}\mathbf{u} = \mathbf{f}.} \quad (7.1)$$

\mathbf{K} is universally called the *stiffness matrix* even in non-structural applications because no consensus has emerged on different names.

The physical significance of the vectors \mathbf{u} and \mathbf{f} varies according to the application being modeled, as illustrated in Table 7.1.

¹ For cosmological scales, such as galaxy clusters, the general theory of relativity is necessary. For the atomic and sub-particle world, quantum mechanics is appropriate.

² In variational mathematics this is called a duality pairing.

Table 7.1. Significance of \mathbf{u} and \mathbf{f} in Miscellaneous FEM Applications

<i>Application Problem</i>	<i>State (DOF) vector \mathbf{u} represents</i>	<i>Conjugate vector \mathbf{f} represents</i>
Structures and solid mechanics	Displacement	Mechanical force
Heat conduction	Temperature	Heat flux
Acoustic fluid	Displacement potential	Particle velocity
Potential flows	Pressure	Particle velocity
General flows	Velocity	Fluxes
Electrostatics	Electric potential	Charge density
Magnetostatics	Magnetic potential	Magnetic intensity

If the relation between forces and displacements is linear but not homogeneous, equation (7.1) generalizes to

$$\mathbf{K}\mathbf{u} = \mathbf{f}_M + \mathbf{f}_I. \quad (7.2)$$

Here \mathbf{f}_I is the initial node force vector introduced in Chapter 4 for effects such as temperature changes, and \mathbf{f}_M is the vector of mechanical forces.

The basic steps of FEM are discussed below in more generality. Although attention is focused on structural problems, most of the steps translate to other applications problems as noted above. The role of FEM in numerical simulation is schematized in Figure 7.1, which is a merged simplification of Figures 1.2 and 1.3. Although this diagram oversimplifies the way FEM is actually used, it serves to illustrate terminology. The three key simulation steps shown are: *idealization*, *discretization* and *solution*. Each step is a source of errors. For example, the discretization error is the discrepancy that appears when the discrete solution is substituted in the mathematical model. The reverse steps: continuification and realization, are far more difficult and ill-posed problems.

The idealization and discretization steps, briefly mentioned in Chapter 1, deserve further discussion. The solution step is dealt with in more detail in Part III of this book.

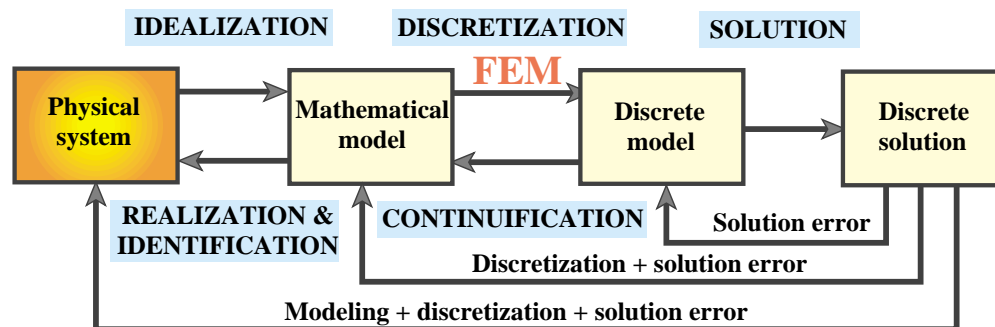


Figure 7.1. A simplified view of the physical simulation process, reproduced to illustrate modeling terminology.

§7.2. IDEALIZATION

Idealization passes from the physical system to a mathematical model. This is the most important step in engineering practice, because it cannot be “canned.” It must be done by a human.

§7.2.1. Models

The word “model” has the traditional meaning of a scaled copy or representation of an object. And that is precisely how most dictionaries define it. We use here the term in a more modern sense, which has become increasingly common since the advent of computers:

A model is a symbolic device built to simulate and predict aspects of behavior of a system.

(7.3)

Note the distinction made between *behavior* and *aspects of behavior*. To predict everything, in all physical scales, you must deal with the actual system. A model *abstracts* aspects of interest to the modeler. The qualifier *symbolic* means that a model represents a system in terms of the symbols and language of another discipline. For example, engineering systems may be (and are) modeled with the symbols of mathematics and/or computer sciences.³

§7.2.2. Mathematical Models

Mathematical modeling, or *idealization*, is a process by which an engineer or scientist passes from the actual physical system under study, to a *mathematical model* of the system, where the term *model* is understood in the sense of (7.3).

The process is called *idealization* because the mathematical model is necessarily an abstraction of the physical reality — note the phrase *aspects of behavior* in (7.3). The analytical or numerical results produced by the mathematical model are physically re-interpreted only for those aspects.⁴

To give an example on the choices an engineer may face, suppose that the structure is a flat plate structure subjected to transverse loading. Here is a non-exhaustive list of four possible mathematical models:

1. A *very thin* plate model based on Von Karman’s coupled membrane-bending theory.
2. A *thin* plate model, such as the classical Kirchhoff’s plate theory.
3. A *moderately thick* plate model, for example Mindlin-Reissner plate theory.
4. A *very thick* plate model based on three-dimensional elasticity.

The person responsible for this kind of decision is supposed to be familiar with the advantages, disadvantages, and range of applicability of each model. Furthermore the decision may be different in static analysis than in dynamics.

Why is the mathematical model an abstraction of reality? Engineering systems, particularly in Aerospace and Mechanical, tend to be highly complex. For simulation it is necessary to reduce that

³ A problem-definition input file, a digitized earthquake record, or a stress plot are examples of the latter.

⁴ Whereas idealization can be reasonably taught in advanced design courses, the converse process of “realization” or “identification” — see Figure 7.1 — generally requires considerable physical understanding and maturity that can only be gained through professional experience.

complexity to manageable proportions. Mathematical modeling is an abstraction tool by which complexity can be controlled.

This is achieved by “filtering out” physical details that are not relevant to the analysis process. For example, a continuum material model filters out the aggregate, crystal, molecular and atomic levels of matter. Engineers are typically interested in a few integrated quantities, such as the maximum deflection of a bridge or the fundamental periods of an airplane. Although to a physicist this is the result of the interaction of billions and billions of molecules, such details are weeded out by the modeling process. Consequently, picking a mathematical model is equivalent to choosing an information filter.

§7.2.3. Implicit vs. Explicit Modeling

As noted the diagram of Figure 7.1 is an oversimplification of engineering practice. The more common scenario is that pictured in Figures 1.2, 1.4 and 1.5. The latter is reproduced in Figure 7.2 for convenience.

A common scenario in industry is: you have to analyze a structure or a substructure, and at your disposal is a “black box” general-purpose finite element program. Those programs offer a *catalog* of element types; for example, bars, beams, plates, shells, axisymmetric solids, general 3D solids, and so on. The moment you choose specific elements from the catalog you automatically accept the mathematical models on which the elements are based. This is *implicit modeling*. Ideally you should be fully aware of the implications of your choice. Providing such “finite element literacy” is one of the objective of this book. Unfortunately many users of commercial programs are unaware of the implied-consent aspect of implicit modeling and their legal implications.

The other extreme happens when you select a mathematical model of the physical problem with your eyes wide open and *then* either shop around for a finite element program that implements that model, or write the program yourself. This is *explicit modeling*. It requires far more technical expertise, resources, experience and maturity than implicit modeling. But for problems that fall out of the ordinary it could be the right thing to do.

In practice a combination of implicit and explicit modeling is common. The physical problem to be simulated is broken down into subproblems. Those subproblems that are conventional and fit available programs may be treated with implicit modeling, whereas those that require special handling may only submit to explicit modeling.

§7.3. DISCRETIZATION

Mathematical modeling is a simplifying step. But models of physical systems are not necessarily simple to solve. They often involve coupled partial differential equations in space and time subject to boundary and/or interface conditions. Such models have an *infinite* number of degrees of freedom.

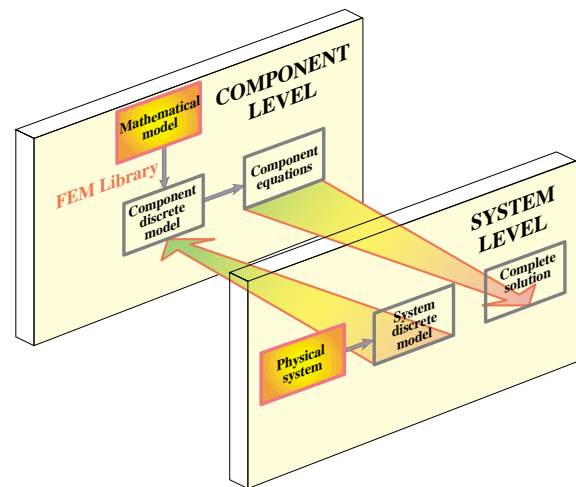


Figure 7.2. A reproduction of Figure 1.5 with some relabeling. Illustrates implicit modeling: picking elements from an existing FEM code consents to an idealization.

§7.3.1. Decisions

At this point one faces the choice of going for analytical or numerical solutions. Analytical solutions, also called “closed form solutions,” are more intellectually satisfying, particularly if they apply to a wide class of problems, so that particular instances may be obtained by substituting the values of free parameters. Unfortunately they tend to be restricted to regular geometries and simple boundary conditions. Moreover some closed-form solutions, expressed for example as inverses of integral transforms, often have to be numerically evaluated to be useful.

Most problems faced by the engineer either do not yield to analytical treatment or doing so would require a disproportionate amount of effort.⁵ The practical way out is numerical simulation. Here is where finite element methods enter the scene.

To make numerical simulations practical it is necessary to reduce the number of degrees of freedom to a *finite* number. The reduction is called *discretization*. The product of the discretization process is the *discrete model*. As discussed in Chapter 1, for complex engineering systems this model is the product of a multilevel decomposition.

Discretization can proceed in space dimensions as well as in the time dimension. Because the present book deals only with static problems, we need not consider the time dimension and are free to concentrate on *spatial discretization*.

§7.3.2. Error Sources and Approximation

Figure 7.1 tries to convey graphically that each simulation step introduces a source of error. In engineering practice modeling errors are by far the most important. But they are difficult and expensive to evaluate, because *model validation* requires access to and comparison with experimental results. These may be either scarce, or unavailable in the case of a new product in the design stage.

Next in order of importance is the *discretization error*. Even if solution errors are ignored — and usually they can — the computed solution of the discrete model is in general only an approximation in some sense to the exact solution of the mathematical model. A quantitative measurement of this discrepancy is called the *discretization error*. The characterization and study of this error is addressed by a branch of numerical mathematics called approximation theory.

Intuitively one might suspect that the accuracy of the discrete model solution would improve as the number of degrees of freedom is increased, and that the discretization error goes to zero as that number goes to infinity. This loosely worded statement describes the *convergence* requirement of discrete approximations. One of the key goals of approximation theory is to make the statement as precise as it can be expected from a branch of mathematics.⁶

⁵ This statement has to be tempered in two respects. First, the wider availability and growing power of computer algebra systems, introduced in Chapter 5, has widened the realm of analytical solutions than can be obtained within a practical time frame. Second, a combination of analytical and numerical techniques is often effective to reduce the dimensionality of the problem and to facilitate parameter studies. Important examples are provided by Fourier analysis, perturbation and boundary-element methods.

⁶ The discretization error is often overhyped in the FEM literature, since it provides an inexhaustible source of publishable poppycock. If the mathematical model is way off reducing the discretization error buys nothing; only a more accurate answer to the wrong problem.

§7.3.3. Other Discretization Methods

It was stated in the first chapter that the most popular discretization techniques in structural mechanics are finite element methods and boundary element methods. The finite element method (FEM) is by far the most widely used. The boundary element method (BEM) has gained in popularity for special types of problems, particularly those involving infinite domains, but remains a distant second, and seems to have reached its natural limits.

In non-structural application areas such as fluid mechanics and electromagnetics, the finite element method is gradually making up ground but faces stiff competition from both the classical and energy-based *finite difference* methods. Finite difference and finite volume methods are particularly well entrenched in computational fluid dynamics spanning moderate to high Reynolds numbers.

§7.4. THE FINITE ELEMENT METHOD

§7.4.1. Interpretation

The finite element method (FEM) is the dominant discretization technique in structural mechanics. As discussed in Chapter 1, the FEM can be interpreted from either a physical or mathematical standpoint. The treatment has so far emphasized the former.

The basic concept in the physical FEM is the subdivision of the mathematical model into disjoint (non-overlapping) components of simple geometry called *finite elements* or *elements* for short. The response of each element is expressed in terms of a finite number of degrees of freedom characterized as the value of an unknown function, or functions, at a set of nodal points. The response of the mathematical model is then considered to be approximated by that of the discrete model obtained by connecting or assembling the collection of all elements.

The disconnection-assembly concept occurs naturally when examining many artificial and natural systems. For example, it is easy to visualize an engine, bridge, building, airplane, or skeleton as fabricated from simpler components.

Unlike finite difference models, finite elements *do not overlap* in space. In the mathematical interpretation of the FEM, this property goes by the name *disjoint support* or *local support*.

§7.4.2. Element Attributes

Just like members in the truss example, one can take finite elements of any kind one at a time. Their local properties can be developed by considering them in isolation, as individual entities. This is the key to the modular programming of element libraries.

In the Direct Stiffness Method, elements are isolated by the disconnection and localization steps, which were described for the truss example in Chapter 2. The procedure involves the separation of elements from their neighbors by disconnecting the nodes, followed by referral of the element to a convenient local coordinate system.⁷ After that we can consider *generic* elements: a bar element, a beam element, and so on. From the standpoint of the computer implementation, it means that you can write one subroutine or module that constructs, by suitable parametrization, all elements of one type, instead of writing one module for each element instance.

⁷ Both steps are only carried out in the modeler's mind. They are placed as part of the DSM for instructional convenience. In practice the analysis begins directly at the element level.

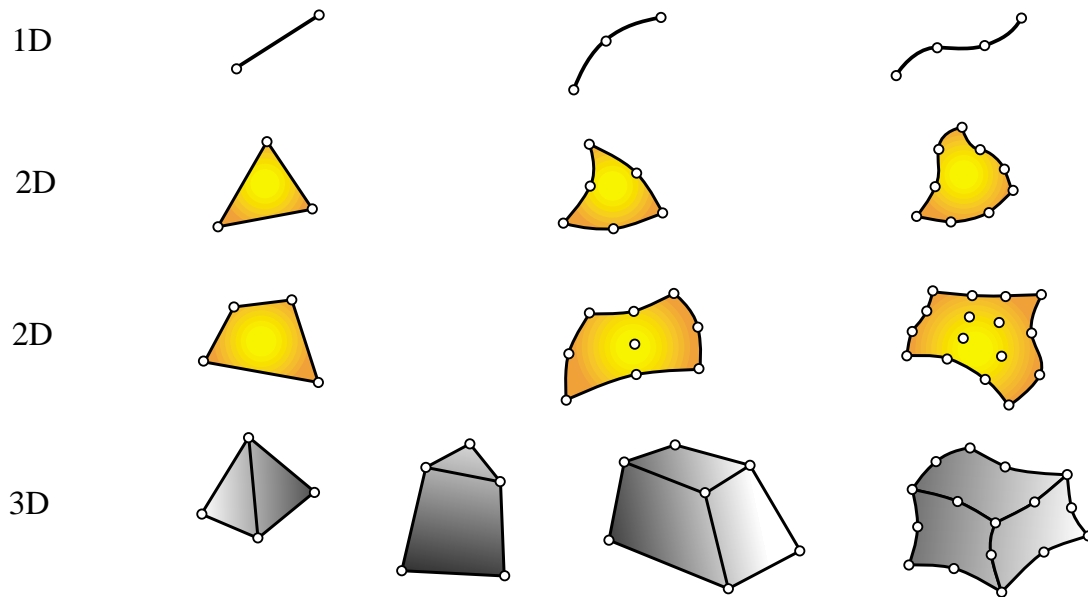


Figure 7.3. Typical finite element geometries in one through three dimensions.

Following is a summary of the data associated with an individual finite element. This data is used in finite element programs to carry out element level calculations.

Intrinsic Dimensionality. Elements can have one, two or three space dimensions.⁸ There are also special elements with zero dimensionality, such as lumped springs or point masses.

Nodal points. Each element possesses a set of distinguishing points called *nodal points* or *nodes* for short. Nodes serve a dual purpose: definition of element geometry, and home for degrees of freedom. They are usually located at the corners or end points of elements, as illustrated in Figure 7.3. In the so-called refined or higher-order elements nodes are also placed on sides or faces, as well as perhaps the interior of the element.

Geometry. The geometry of the element is defined by the placement of the nodal points. Most elements used in practice have fairly simple geometries. In one-dimension, elements are usually straight lines or curved segments. In two dimensions they are of triangular or quadrilateral shape. In three dimensions the most common shapes are tetrahedra, pentahedra (also called wedges or prisms), and hexahedra (also called cuboids or “bricks”). See Figure 7.3.

Degrees of freedom. The degrees of freedom (DOF) specify the *state* of the element. They also function as “handles” through which adjacent elements are connected. DOFs are defined as the values (and possibly derivatives) of a primary field variable at nodal points. The actual selection depends on criteria studied at length in Part II. Here we simply note that the key factor is the way in which the primary variable appears in the mathematical model. For mechanical elements, the primary variable is the displacement field and the DOF for many (but not all) elements are the displacement components at the nodes.

Nodal forces. There is always a set of nodal forces in a one-to-one correspondence with degrees of

⁸ In dynamic analysis, time appears as an additional dimension.

freedom. In mechanical elements the correspondence is established through energy arguments.

Constitutive properties. For a mechanical element these are relations that specify the material behavior. For example, in a linear elastic bar element it is sufficient to specify the elastic modulus E and the thermal coefficient of expansion α .

Fabrication properties. For mechanical elements these are fabrication properties which have been integrated out from the element dimensionality. Examples are cross sectional properties of MoM elements such as bars, beams and shafts, as well as the thickness of a plate or shell element.

This data is used by the element generation subroutines to compute element stiffness relations in the local system.

§7.5. CLASSIFICATION OF MECHANICAL ELEMENTS

The following classification of finite elements in structural mechanics is loosely based on the “closeness” of the element with respect to the original physical structure.

It is given here because it clarifies points that recur in subsequent sections, as well as providing insight into advanced modeling techniques such as hierarchical breakdown and global-local analysis.

§7.5.1. Primitive Structural Elements

These resemble fabricated structural components, and are often drawn as such; see Figure 7.4. The qualifier *primitive* is used to distinguish them from macroelements, which is another element class described below. Primitive means that they are not decomposable into simpler elements.

These elements are usually derived from Mechanics-of-Materials simplified theories and are better understood from a physical, rather than mathematical, standpoint. Examples are the elements discussed in Chapter 6: bars, cables, beams, shafts, spars.

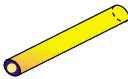

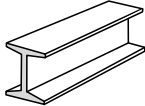

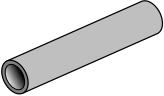





Physical Structural Component	Mathematical Model Name	Finite Element Discretization
	bar	
	beam	
	tube, pipe	
	spar (web)	
	shear panel (2D version of above)	

Figure 7.4. Examples of primitive structural elements.

§7.5.2. Continuum Elements

These do not resemble fabricated structural components at all. They result from the subdivision of “blobs” of continua, or of structural components viewed as continua.

Unlike structural elements, continuum elements are better understood in terms of their mathematical interpretation. Examples: plates, slices, shells, axisymmetric solids, general solids. See Figure 7.5.

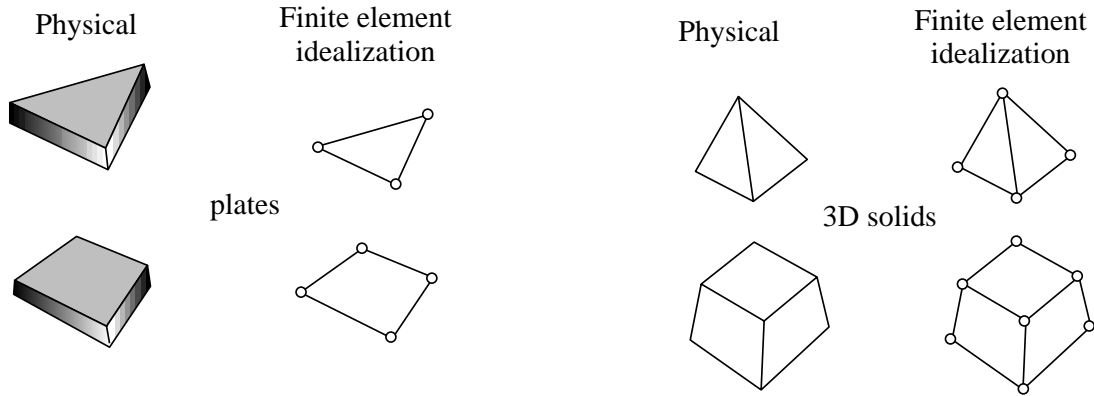


Figure 7.5. Continuum element examples.

§7.5.3. Special Elements

Special elements partake of the characteristics of structural and continuum elements. They are derived from a continuum mechanics standpoint but include features closely related to the physics of the problem. Examples: crack elements for fracture mechanics applications, shear panels, infinite and semi-infinite elements, contact and penalty elements, rigid-body elements. See Figure 7.6.

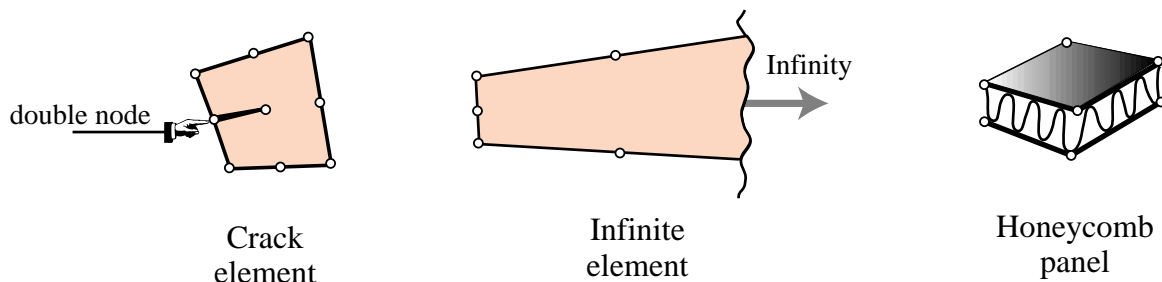


Figure 7.6. Special element examples.

§7.5.4. Macroelements

Macroelements are also called mesh units and superelements, although the latter term overlaps with substructures (defined below). These often resemble structural components, but are fabricated with simpler elements. See Figure 7.7.

The main reason for introducing macroelements is to simplify preprocessing tasks. For example, it may be simpler to define a regular 2D mesh using quadrilaterals rather than triangles. The fact that the quadrilateral is really a macroelement may not be important for the majority of users.

Similarly a box macroelement can save modeling times for structures that are built by such components; for example box-girder bridges.

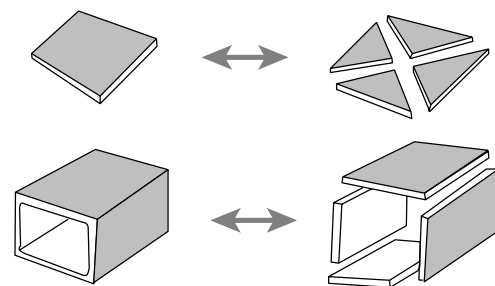


Figure 7.7. Macroelement examples.

§7.5.5. Substructures

Also called structural modules and superelements. These are sets of elements with a well defined structural function, typically obtained by cutting the complete structure into functional components. Examples: the wings and fuselage in an airplane, the deck and cables in a suspension bridge.

The distinction between substructures and macroelements is not clear-cut. The main conceptual distinction is that substructures are defined “top down” as parts of a complete structure, whereas macroelements are built “bottom up” from primitive elements. The term *superelement* is often used in a collective sense to embrace all element groupings. This topic is further covered in Chapter 11.

§7.6. ASSEMBLY

The assembly procedure of the Direct Stiffness Method for a general finite element model follows rules identical in principle to those discussed for the truss example. As in that case the process involves two basic steps:

Globalization. The element equations are transformed to a common *global* coordinate system.

Merge. The element stiffness equations are merged into the master stiffness equations by appropriate indexing and matrix-entry addition.

The hand calculations for the example truss conceal the implementation complexity. The master stiffness equations in practical cases may involve thousands (or even millions) of freedoms. The use of sparse matrix techniques, and possibly peripheral storage, becomes essential. But this inevitably increases the programming complexity. The topic is elaborated upon in Part III.

§7.7. BOUNDARY CONDITIONS

A key strength of the FEM is the ease and elegance with which it handles arbitrary boundary and interface conditions. This power, however, has a down side. A big hurdle faced by FEM newcomers is the understanding and proper handling of boundary conditions. Below is a simple recipe for treating boundary conditions. The following Chapter provides specific rules and examples.

§7.7.1. Essential and Natural B.C.

The key thing to remember is that boundary conditions (BCs) come in two basic flavors: essential and natural.

Essential BCs directly affect DOFs, and are imposed on the left-hand side vector **u**.

Natural BCs do not directly affect DOFs and are imposed on the right-hand side vector **f**.

The mathematical justification for this distinction requires use of concepts from variational calculus, and is consequently relegated to Part II. For the moment, the basic recipe is:

1. If a boundary condition involves one or more degrees of freedom in a *direct* way, it is essential. An example is a prescribed node displacement.
2. Otherwise it is natural.

The term “direct” is meant to exclude derivatives of the primary function, unless those derivatives also appear as degrees of freedom, such as rotations in beams and plates.

§7.7.2. Boundary Conditions in Structural Problems

Essential boundary conditions in mechanical problems involve *displacements* (but not strain-type displacement derivatives). Support conditions for a building or bridge problem furnish a particularly simple example. But there are more general boundary conditions that occur in practice. A structural engineer must be familiar with displacement B.C. of the following types.

Ground or support constraints. Directly restraint the structure against rigid body motions.

Symmetry conditions. To impose symmetry or antisymmetry restraints at certain points, lines or planes of structural symmetry. This allows the discretization to proceed only over part of the structure with a consequent savings in modeling effort and number of equations to be solved.

Ignorable freedoms. To suppress displacements that are irrelevant to the problem.⁹ Even experienced users of finite element programs are sometimes baffled by this kind. An example are rotational degrees of freedom normal to smooth shell surfaces.

Connection constraints. To provide connectivity to adjoining structures or substructures, or to specify relations between degrees of freedom. Many conditions of this type can be subsumed under the label *multipoint constraints* or *multifreedom constraints*. These can be notoriously difficult to handle from a numerical standpoint, and are covered in Chapters 9 and 10.

Notes and Bibliography

Most FEM textbooks do not provide a systematic treatment of modeling. This is no accident: few academic authors have experience with complex engineering systems. Good engineers are too busy (and in demand) to have time for writing books. This gap has been particularly acute since FEM came on the scene because of generational gaps: “real engineers” tend to mistrust the computer, and often for good reasons. The notion of explicit versus implicit modeling, which has deep legal and professional implications, is rarely mentioned.

FEM terminology is by now standard, and so is a majority of the notation. But that is not so in early publications. E.g. \mathbf{K} is universally used¹⁰ for stiffness matrix in virtually all post-1960 books. There are a few exceptions: Przemieniecki [7.1] uses \mathbf{S} . There is less unanimity on \mathbf{u} and \mathbf{f} for node displacement and force vectors, respectively; some books such as Zienkiewicz and Taylor [7.2] still use different symbols.

The element classification given here attempts to systematize dispersed references. In particular, the distinction between macroelements, substructures and superelements is an ongoing source of confusion, particularly since massively parallel computation popularized the notion of “domain decomposition” in the computer science community. The all-encompassing term “superelement” emerged in Norway by 1968 as part of the implementation of the computer program SESAM.

The topic of BC classification and handling is a crucial one in practice. More mistakes are done in this aspect of FEM application than anywhere else.

References

- [7.1] Przemieniecki, J. S., *Theory of Matrix Structural Analysis*, McGraw-Hill, New York, 1968; Dover edition 1986.
- [7.2] Zienkiewicz, O. C., Taylor, R. E., *The Finite Element Method*, 4th ed., McGraw-Hill, London, Vol. I: 1988, Vol II: 1993.

⁹ In classical dynamics these are called *ignorable coordinates*.

¹⁰ A symbol derived from the “spring constant” k that measures the stiffness of a spring.

8

FEM Modeling: Mesh, Loads and BCs

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This Chapter continues the exposition of finite element modeling principles. After some general recommendations, it provides guidelines on layout of finite element meshes, conversion of distributed loads to node forces, and how to handle the simplest forms of support boundary conditions. The following Chapters deal with more complicated forms of boundary conditions called multifreedom constraints.

The presentation is “recipe oriented” and illustrated by specific examples. All examples are from structural mechanics; most of them are two-dimensional. No attempt is given at a rigorous justification of rules and recommendations, because that would require mathematical tools beyond the scope of this course.

§8.1. GENERAL RECOMMENDATIONS

The general rules that should guide you in the use of commercial or public FEM packages, are:

- Use the *simplest* type of finite element that will do the job.
- *Never, never, never* mess around with complicated or special elements, unless you are *absolutely sure* of what you are doing.
- Use the *coarsest mesh* you think will capture the dominant physical behavior of the physical system, particularly in *design* applications.

Three word summary: *keep it simple*. Initial FE models may have to be substantially revised to accommodate design changes, and there is little point in using complicated models that will not survive design iterations. The time for refined models is when the design has stabilized and you have a better view picture of the underlying physics, possibly reinforced by experiments or observation.

§8.2. GUIDELINES ON ELEMENT LAYOUT

The following guidelines are stated for structural applications. As noted above, they will be often illustrated for two-dimensional meshes of continuum elements for ease of visualization.

§8.2.1. Mesh Refinement

Use a relatively fine (coarse) discretization in regions where you expect a high (low) *gradient* of strains and/or stresses. Regions to watch out for high gradients are:

- Near entrant corners, or sharply curved edges.
- In the vicinity of concentrated (point) loads, concentrated reactions, cracks and cutouts.
- In the interior of structures with abrupt changes in thickness, material properties or cross sectional areas.

The examples in Figure 8.1 illustrate some of these “danger regions.” Away from such regions one can use a fairly coarse discretization within constraints imposed by the need of representing the structural geometry, loading and support conditions reasonably well.

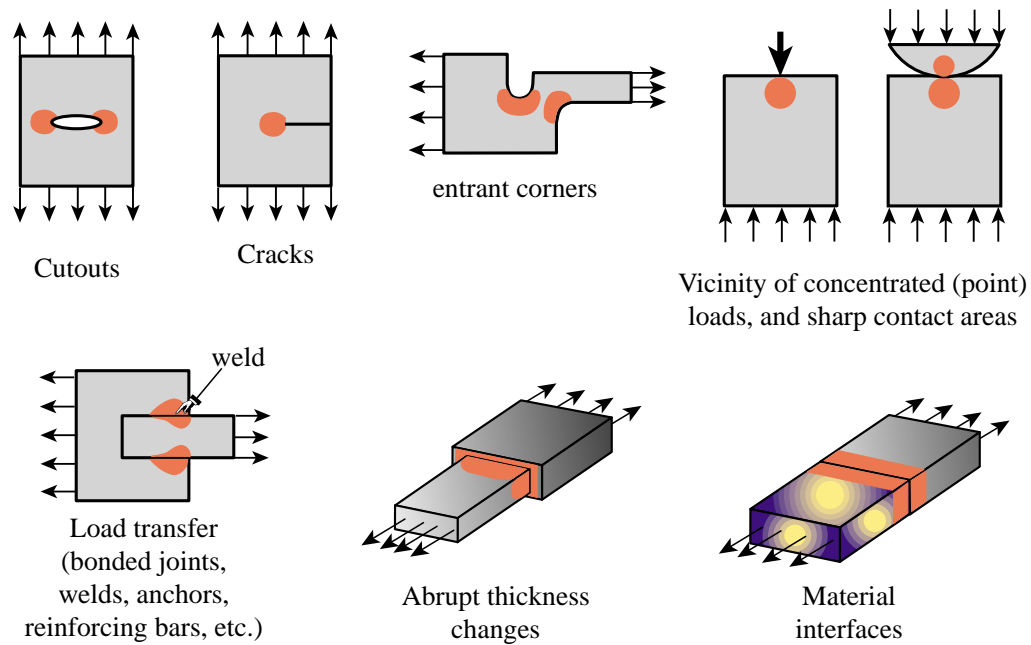


Figure 8.1. Some situations where a locally refined finite element discretization (in the red-colored areas) is recommended.

§8.2.2. Element Aspect Ratios

When discretizing two and three dimensional problems, try to avoid finite elements of high aspect ratios: elongated or “skinny” elements, such as the ones illustrated on the right of Figure 8.2. (The aspect ratio of a two- or three-dimensional element is the ratio between its largest and smallest dimension.)

As a rough guideline, elements with aspect ratios exceeding 3 should be viewed with caution and those exceeding 10 with alarm. Such elements will not necessarily produce bad results — that depends on the loading and boundary conditions of the problem — but do introduce the potential for trouble.

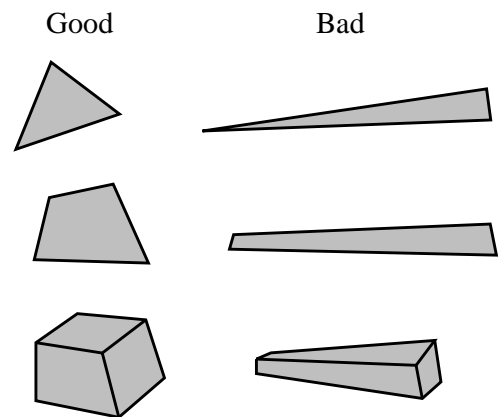


Figure 8.2. Elements with good and bad aspect ratios.

REMARK 8.1

In many “thin” structures modeled as continuous bodies the appearance of “skinny” elements is inevitable on account of computational economy reasons. An example is provided by the three-dimensional modeling of layered composites in aerospace and mechanical engineering problems.

§8.2.3. Physical Interfaces

A physical interface, resulting from example from a change in material, should also be an interelement boundary. That is, *elements must not cross interfaces*. See Figure 8.3.

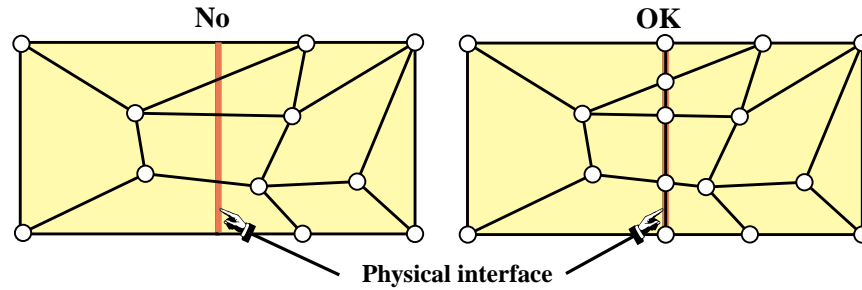


Figure 8.3. Illustration of the rule that elements should not cross material interfaces.

§8.2.4. Preferred Shapes

In two-dimensional FE modeling, if you have a choice between triangles and quadrilaterals with similar nodal arrangement, prefer quadrilaterals. Triangles are quite convenient for mesh generation, mesh transitions, rounding up corners, and the like. But sometimes triangles can be avoided altogether with some thought. One of the homework exercises is oriented along these lines.

In three dimensional FE modeling, prefer strongly bricks over wedges, and wedges over tetrahedra. The latter should be used only if there is no viable alternative.¹ The main problem with tetrahedra and wedges is that they can produce wrong stress results even if the displacement solution looks reasonable.

§8.3. DIRECT LUMPING OF DISTRIBUTED LOADS

In practical structural problems, distributed loads are more common than concentrated (point) loads.² Distributed loads may be of surface or volume type.

Distributed surface loads (called surface tractions in continuum mechanics) are associated with actions such as wind or water pressure, lift in airplanes, live loads on bridges, and the like. They are measured in force per unit area.

Volume loads (called body forces in continuum mechanics) are associated with own weight (gravity), inertial, centrifugal, thermal, prestress or electromagnetic effects. They are measured in force per unit volume.

A derived type: line loads, result from the integration of surface loads along one transverse direction, or of volume loads along two transverse directions. Line loads are measured in force per unit length.

Whatever their nature or source, distributed loads *must be converted to consistent nodal forces* for FEM analysis. These forces eventually end up in the right-hand side of the master stiffness equations.

The meaning of “consistent” can be made precise through variational arguments, by requiring that the distributed loads and the nodal forces produce the same external work. Since this requires

¹ Unfortunately, many existing space-filling automatic mesh generators in three dimensions produce tetrahedral meshes. There are generators that try to produce bricks, but these often fail in geometrically complicated regions.

² In fact, one of the objectives of a good structural design is to avoid or alleviate stress concentrations produced by concentrated forces.

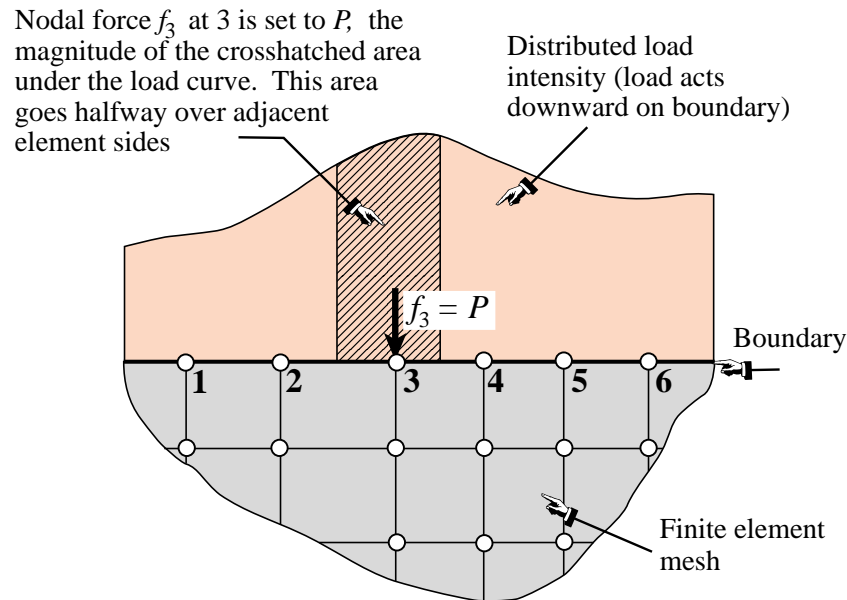


Figure 8.4. NbN direct lumping of distributed load, illustrated for a 2D problem.

the introduction of external work functionals, the topic is deferred to Part II. However, a simpler approach called *direct load lumping*, or simply *load lumping*, is often used by structural engineers in lieu of the more mathematically impeccable but complicated variational approach. Two variants of this technique are described below for distributed surface loads.

§8.3.1. Node by Node (NbN) Lumping

The node by node (NbN) lumping method is graphically explained in Figure 8.4. This example shows a distributed surface loading acting normal to the straight boundary of a two-dimensional FE mesh. (The load is assumed to have been integrated through the thickness normal to the figure, so it is actually a line load measured as force per unit length.)

The procedure is also called *tributary region* or *contributing region* method. For the example of Figure 8.4, each boundary node is assigned a *tributary region* around it that extends halfway to the adjacent nodes. The force contribution P of the cross-hatched area is directly assigned to node 3.

This method has the advantage of not requiring the computation of centroids, as required in the EbE technique discussed in the next subsection. For this reason it is often preferred in hand computations. It can be extended to three-dimensional meshes as well as volume loads.³ It should be avoided, however, when the applied forces vary rapidly (within element length scales) or act only over portions of the tributary regions.

§8.3.2. Element by Element (EbE) Lumping

In this variant the distributed loads are divided over element domains. The resultant load is assigned to the centroid of the load diagram, and apportioned to the element nodes by statics. A node force

³ The computation of tributary areas and volumes can be done through the so-called Voronoi diagrams. This is an advanced topic in computational geometry and thus not treated here.

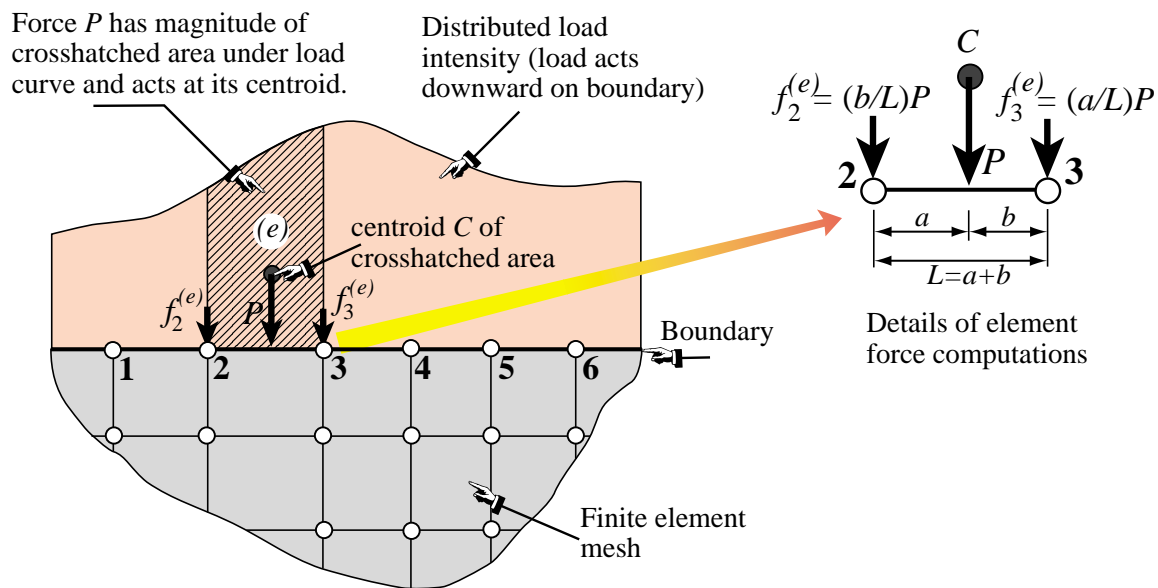


Figure 8.5. EbE direct lumping of distributed load, illustrated for a 2D problem.

is obtained by adding the contributions from all elements meeting at that node. The procedure is illustrated in Figure 8.5, which shows details of the computation over segment 2-3. The total force at node 3, for instance, would be that contributed by segments 2-3 and 3-4.

If applicable, the EbE procedure is more accurate than NbN lumping. In fact it agrees with the consistent node lumping for simple elements that possess only corner nodes. In those cases it is not affected by the sharpness of the load variation and can be even used for point loads that are not applied at the nodes.

The procedure is not applicable if the centroidal resultant load cannot be apportioned by statics. This happens if the element has midside faces or internal nodes in addition to corner nodes, or if it has rotational degrees of freedom. For those elements the variational-based consistent approach covered in Part II is preferable.

§8.4. BOUNDARY CONDITIONS

The key distinction between *essential* and *natural* boundary conditions (BC) was introduced in the previous Chapter. The distinction is explained in Part II from a variational standpoint. In this Chapter we discuss next the simplest *essential* boundary conditions in structural mechanics from a physical standpoint. This makes them relevant to problems with which a structural engineer is familiar. Because of the informal setting, the ensuing discussion relies heavily on examples.

In structural problems formulated by the DSM, the recipe of §7.7.1 that distinguishes between essential and natural BC is: if it directly involves the nodal freedoms, such as displacements or rotations, it is essential. Otherwise it is natural. Conditions involving applied loads are natural. Essential BCs take precedence over natural BCs.

The simplest essential boundary conditions are support and symmetry conditions. These appear in many practical problems. More exotic types, such as multifreedom constraints, require more

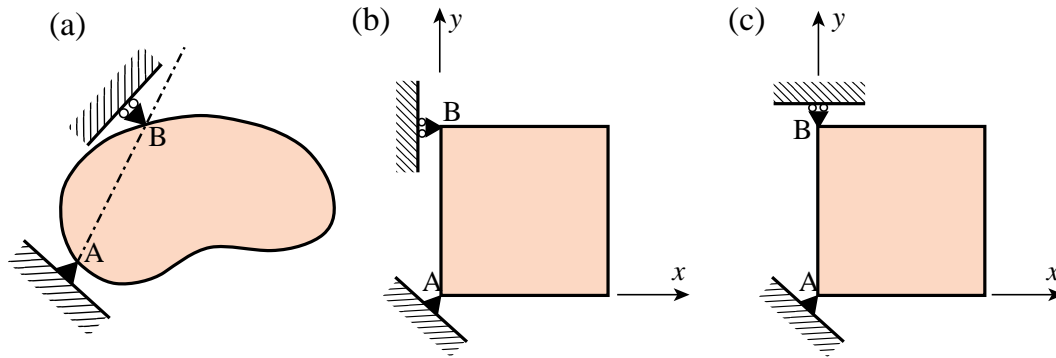


Figure 8.6. Examples of restraining a body against two-dimensional rigid body motions.

advanced mathematical tools and are covered in the next two Chapters.

§8.5. SUPPORT CONDITIONS

Supports are used to restrain structures against relative rigid body motions. This is done by attaching them to Earth ground (through foundations, anchors or similar devices), or to a “ground structure” which is viewed as the external environment.⁴ The resulting boundary conditions are often called *motion constraints*. In what follows we analyze two- and three-dimensional motions separately.

§8.5.1. Supporting Two Dimensional Bodies

Figure 8.6 shows two-dimensional bodies that displace in the plane of the paper. If a body is not restrained, an applied load will cause infinite displacements. Regardless of the loading conditions, the structure must be restrained against two translations and one rotation. Consequently the minimum number of constraints that has to be imposed in two dimensions is *three*.

In Figure 8.6, support A provides *translational* restraint, whereas support B, together with A, provides *rotational* restraint. In finite element terminology, we say that we *delete* (fix, remove, preclude) all translational displacements at point A, and that we delete the translational degree of freedom directed along the normal to the AB direction at point B. This body is free to distort in any manner without the supports imposing any displacement constraints.

Engineers call A and B *reaction-to-ground points*. This means that if the supports are conceptually removed, the applied loads are automatically balanced by reactive forces at points A and B, in accordance with Newton’s third law. Additional freedoms may be removed to model greater restraint by the environment. However, Figure 8.6(a) does illustrate the *minimal* number of constraints.

Figure 8.6(b) shows a simplified version of Figure 8.6(a). Here the line AB is parallel to the global y axis. We simply delete the x and y translations at point A, and the x translation at point B. If the roller support at B is modified as in 8.6(c), it becomes ineffective in constraining the infinitesimal rotational motion about point A because the rolling direction is normal to AB. The configuration of 8.6(c) is called a *kinematic mechanism*, and will be flagged by a singular modified stiffness matrix.

⁴ For example, the engine of a car is attached to the vehicle frame through mounts. The car frame becomes the “ground structure,” which moves with respect to Earth ground.

§8.5.2. Supporting Three Dimensional Bodies

Figure 8.7 illustrates the extension of the freedom deletion concept to three dimensions. The minimal number of freedoms that have to be deleted is now *six* and many combinations are possible. In the example of Figure 8.7, all three degrees of freedom at point *A* have been deleted to prevent rigid body translations. The *x* displacement component at point *B* is deleted to prevent rotation about *z*, the *z* component is deleted at point *C* to prevent rotation about *y*, and the *y* component is deleted at point *D* to prevent rotation about *x*.

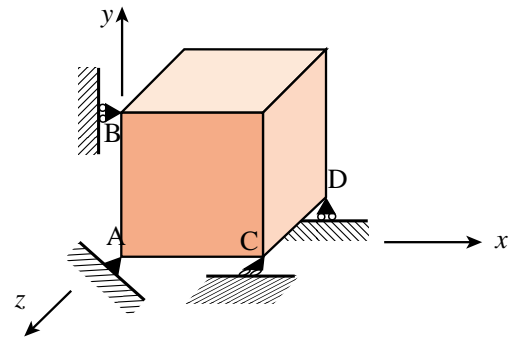


Figure 8.7. Suppressing rigid body motions in a three-dimensional body.

§8.6. SYMMETRY AND ANTISYMMETRY CONDITIONS

Engineers doing finite element analysis should be on the lookout for conditions of *symmetry* or *antisymmetry*. Judicious use of these conditions allows only a portion of the structure to be analyzed, with a consequent saving in data preparation and computer processing time.⁵

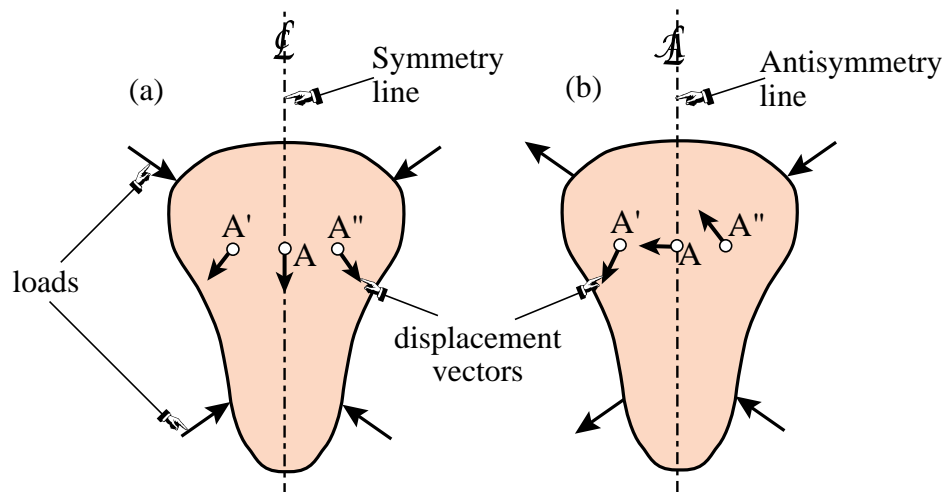


Figure 8.8. Visualizing symmetry and antisymmetry lines.

§8.6.1. Visualization

Recognition of symmetry and antisymmetry conditions can be done by either visualization of the displacement field, or by imagining certain rotational or reflection motions. Both techniques are illustrated for the two-dimensional case.

A *symmetry line* in two-dimensional motion can be recognized by remembering the “mirror” displacement pattern shown in Figure 8.8(a). Alternatively, a 180° rotation of the body about the symmetry line reproduces exactly the original problem.

⁵ Even if the conditions are not explicitly applied through BCs, they provide valuable checks on the computed solution.

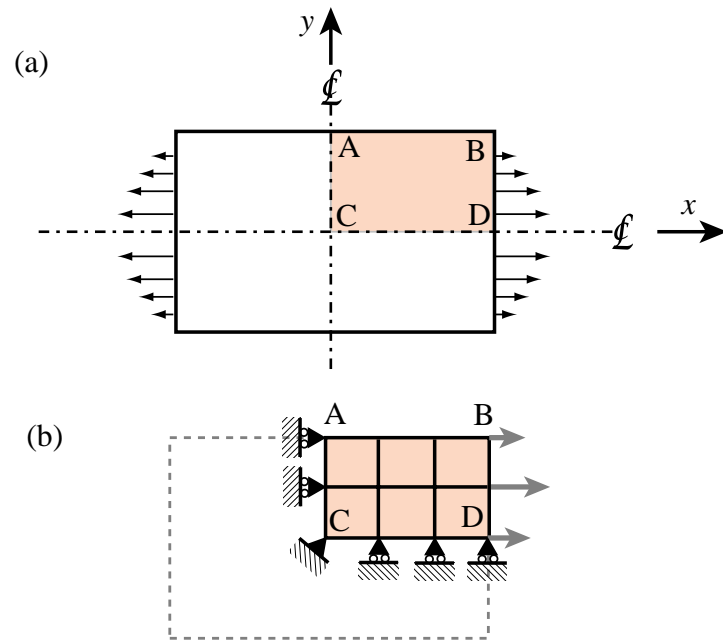


Figure 8.9. A doubly symmetric structure under symmetric loading.

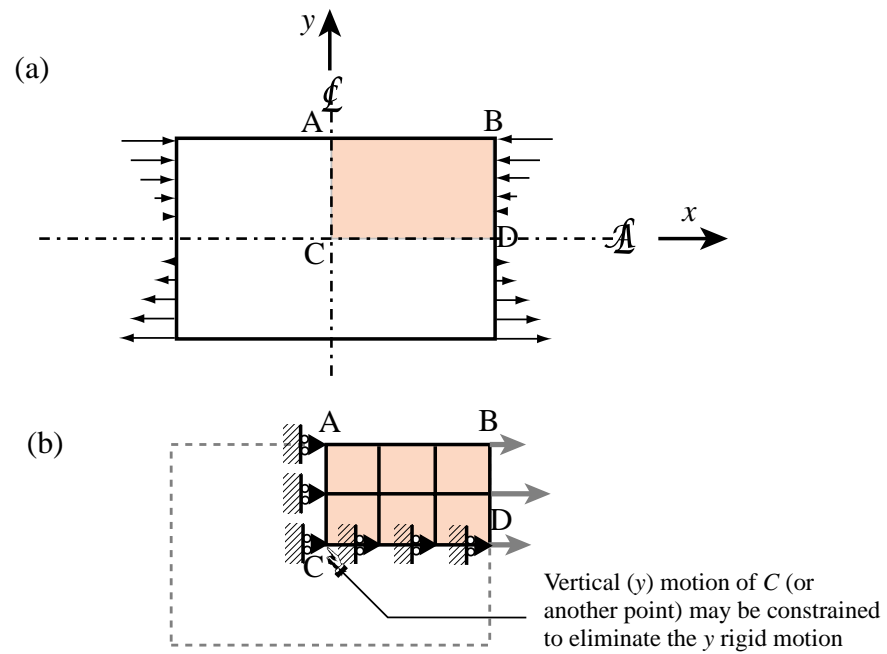


Figure 8.10. A doubly symmetric structure under antisymmetric loading.

An *antisymmetry line* can be recognized by the displacement pattern illustrated in Figure 8.8(b). Alternatively, a 180° rotation of the body about the antisymmetry line reproduces exactly the original problem except that all applied loads are reversed.

Similar recognition patterns can be drawn in three dimensions to help visualization of *planes* of symmetry or antisymmetry. More complex regular patterns associated with *sectorial* symmetry (also called *harmonic* symmetry) and *rotational symmetry* can be treated in a similar manner, but will not be discussed here.

§8.6.2. Effect of Loading Patterns

Although the structure may look symmetric in shape, it must be kept in mind that model reduction can be used only if the loading conditions are also symmetric or antisymmetric.

Consider the plate structure shown in Figure 8.9(a). This structure is symmetrically loaded on the x - y plane. Applying the recognition patterns stated above one concludes that the structure is *doubly symmetric* in both geometry and loading. It is evident that no displacements in the x -direction are possible for any point on the y -axis, and that no y displacements are possible for points on the x axis. A finite element model of this structure may look like that shown in Figure 8.9(b).

On the other hand if the loading is *antisymmetric*, as shown in Figure 8.10(a), then the x axis becomes an *antisymmetry line* because none of the $y = 0$ points can move along the x direction. The boundary conditions to be imposed on the finite element model are also different, as shown in Figure 8.10(b).

REMARK 8.2

For the antisymmetric loading case, one node point has to be constrained against vertical motion. If there are no actual physical supports, the choice is arbitrary and amounts only to an adjustment on the overall (rigid-body) vertical motion. In Figure 8.10(b) the center point C has been chosen to be that vertically-constrained node. But any other node could be selected as well; for example A or D. The important thing is not to overconstrain the structure by applying more than one y constraint.

Notes and Bibliography

FEM modeling rules in most textbooks are “diffuse” if given at all. As noted in Chapter 7, most authors lack practical experience and view FEM as a way to solve BVPs of their own choosing. The rule collection at the start of this Chapter attempts to place key recommendations in one place.

The treatment of BCs tends to be also flaky. A notable exception is Irons and Ahmad [8.1], which is understandable since Irons worked in industry (Rolls-Royce Ltd) before moving to academia.

References

- [8.1] Irons, B. M., Ahmad, S., *Techniques of Finite Elements*, Ellis Horwood Ltd, Chichester, UK, 1980.

Homework Exercises for Chapters 7 and 8

FEM Modeling

EXERCISE 8.1

[D:10] The plate structure shown in Figure E8.1 is loaded and deforms in the plane of the figure. The applied load at D and the supports at I and N extend over a fairly narrow area. Give a list of what you think are the likely “trouble spots” that would require a locally finer finite element mesh to capture high stress gradients.

Identify those spots by its letter and a reason. For example, D : vicinity of point load.

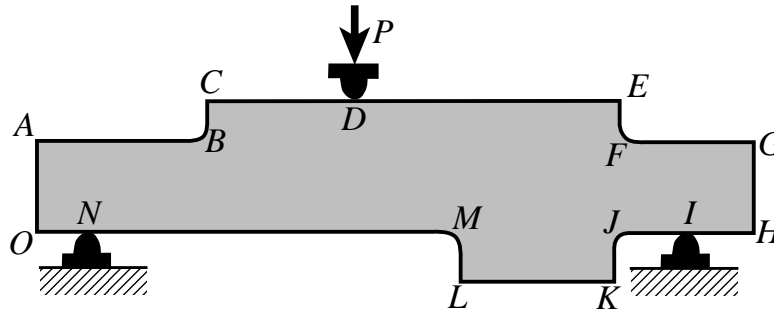


Figure E8.1. Plate structure for Exercise 8.1.

EXERCISE 8.2

[D:15] Part of a two-dimensional FE mesh has been set up as indicated in Figure E8.2. Region $ABCD$ is still unmeshed. Draw a *transition mesh* within that region that correctly merges with the regular grids shown, uses 4-node quadrilateral elements (quadrilaterals with corner nodes only), and *avoids triangles*. *Note*: There are several (equally acceptable) solutions.

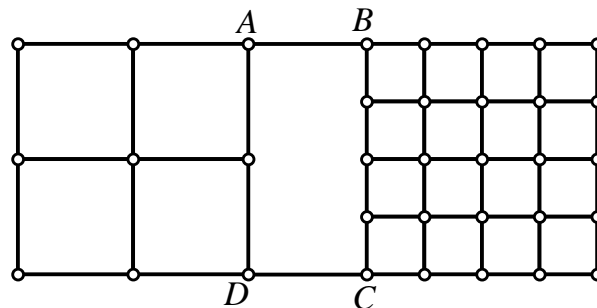


Figure E8.2. Plate structure for Exercise 8.2.

EXERCISE 8.3

[A:15] Compute the “lumped” nodal forces f_1 , f_2 , f_3 and f_4 equivalent to the linearly-varying distributed surface load q for the finite element layout defined in Figure E8.3. Use both NbN and EbE lumping. For example, $f_1 = 3q/8$ for NbN. Check that $f_1 + f_2 + f_3 + f_4 = 6q$ for both schemes (why?). Note that q is given as a force per unit of vertical length.

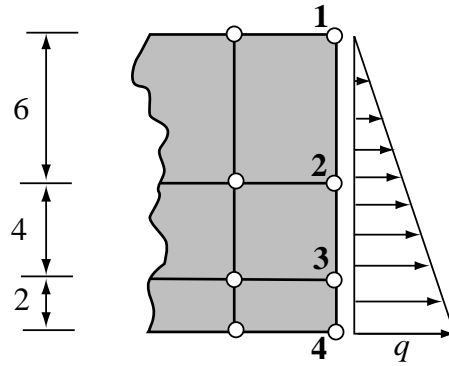


Figure E8.3. Mesh layout for Exercise 8.3.

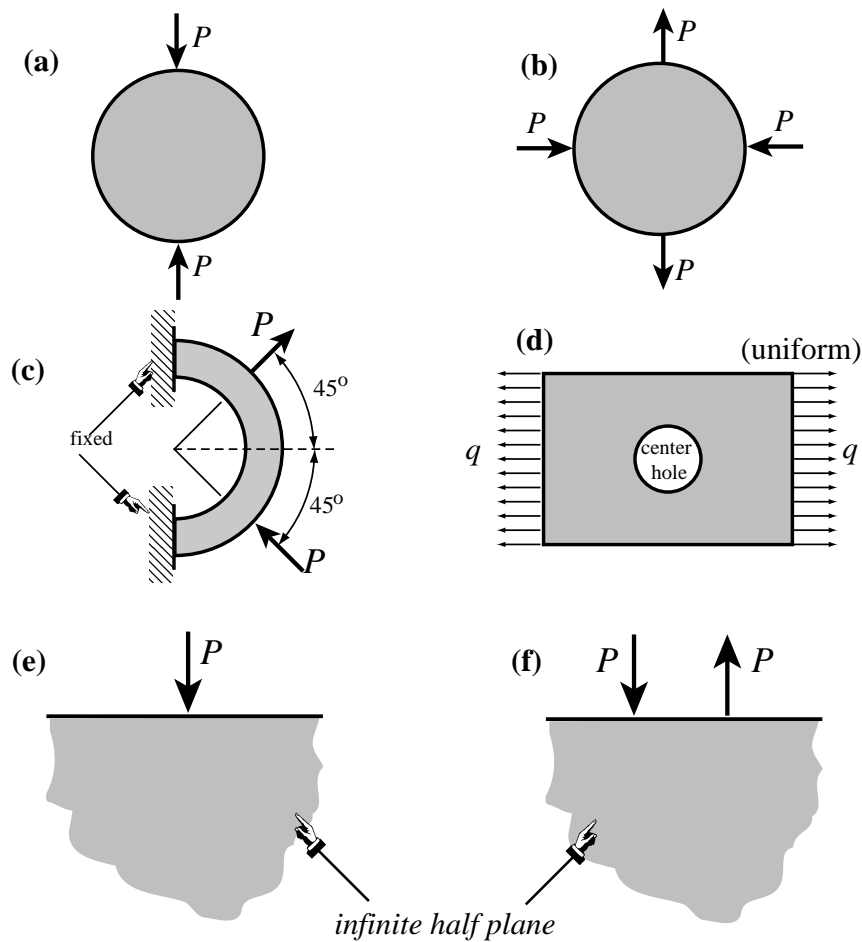


Figure E8.4. Problems for Exercise 8.4.

EXERCISE 8.4

[D:15] Identify the symmetry and antisymmetry lines in the two-dimensional problems illustrated in Figure E8.4. They are: (a) a circular disk under two diametrically opposite point forces (the famous “Brazilian test” for concrete); (b) the same disk under two diametrically opposite force pairs; (c) a clamped semiannulus under

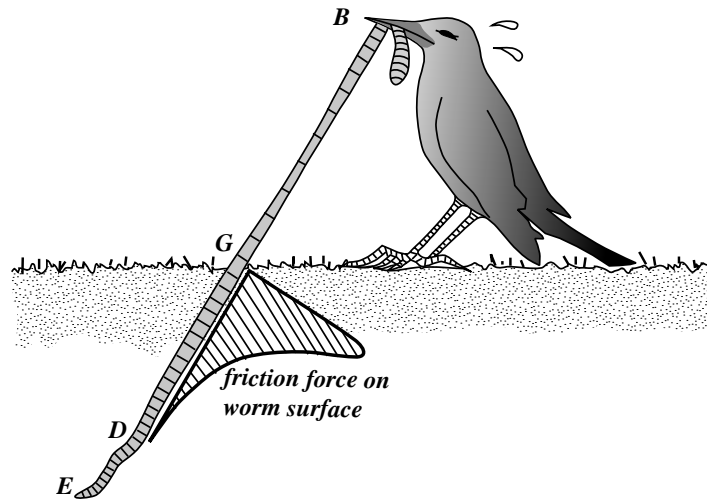


Figure E8.5. The hungry bird.

a force pair oriented as shown; (d) a stretched rectangular plate with a central circular hole. Finally (e) and (f) are half-planes under concentrated loads.⁶

Having identified those symmetry/antisymmetry lines, state whether it is possible to cut the complete structure to one half or one quarter before laying out a finite element mesh. Then draw a coarse FE mesh indicating, with rollers or fixed supports, which kind of displacement BCs you would specify on the symmetry or antisymmetry lines. *Note: Do all sketches on your paper, not on the printed figures.*

EXERCISE 8.5

[D:20] You (a finite element guru) pass away and come back to the next life as an intelligent but hungry bird. Looking around, you notice a succulent big worm taking a peek at the weather. You grab one end and pull for dinner; see Figure E8.5.

After a long struggle, however, the worm wins. While hungrily looking for a smaller one your thoughts wonder to FEM and how the bird extraction process might be modeled so you can pull it out more efficiently. Then you wake up to face this homework question. Try your hand at the following “worm modeling” points.

- The worm is simply modeled as a string of one-dimensional (bar) elements. The “worm axial force” is of course constant from the beak B to ground level G , then decreases rapidly because of soil friction (which varies roughly as plotted in the figure above) and drops to nearly zero over DE . Sketch how a good “worm-element mesh” should look like to capture the axial force well.
- On the above model, how would you represent boundary conditions, applied forces and friction forces?
- Next you want a more refined analysis of the worm that distinguishes skin and insides. What type of finite element model would be appropriate?
- (Advanced) Finally, point out what need to be added to the model of (c) to include the soil as an elastic medium.

Briefly explain your decisions. Don't write equations.

EXERCISE 8.6

[A/D:20] Explain from kinematics why two antisymmetry lines in 2D cannot cross at a finite point. As a

⁶ Note that (e) is the famous Flamant's problem, which is important in the 2D design of foundations of civil structures. The analytical solution of (e) and (f) may be found, for instance, in Timoshenko-Goodier's *Theory of Elasticity*, 2nd Edition, page 85ff.

corollary, investigate whether it is possible to have more than one antisymmetry line in a 2D elasticity problem.

EXERCISE 8.7

[A/D:15] Explain from kinematics why a symmetry line and an antisymmetry line must cross at right angles.

EXERCISE 8.8

[A/D:15] A 2D body has $n > 1$ symmetry lines passing through a point C and spanning an angle π/n from each other. This is called *sectorial symmetry* if $n \geq 3$. Draw a picture for $n = 5$, say for a car wheel. Explain why C is fixed.

EXERCISE 8.9

[A/D:25, 5 each] A body is in 3D space. The analogs of symmetry and antisymmetry lines are symmetry and antisymmetry planes, respectively. The former are also called mirror planes.

- State the kinematic properties of symmetry and antisymmetric planes, and how they can be identified.
- Two symmetry planes intersect. State the kinematic properties of the intersection line.
- A symmetry plane and an antisymmetry plane intersect. State the kinematic properties of the intersection line. Can the angle between the planes be arbitrary?
- Can two antisymmetry planes intersect?
- Three symmetry planes intersect. State the kinematic properties of the intersection point.

EXERCISE 8.10

[A:25] A 2D problem is called *periodic* in the x direction if all fields, in particular displacements, repeat upon moving over a distance $a > 0$: $u_x(x + a, y) = u_x(x, y)$ and $u_y(x + a, y) = u_y(x, y)$. Can this situation be treated by symmetry and/or antisymmetry lines?

EXERCISE 8.11

[A:25] Extend the previous exercise to *antiperiodicity*, in which $u_x(x + a, y) = u_x(x, y)$ and $u_y(x + a, y) = -u_y(x, y)$.

EXERCISE 8.12

[A:40] If the world were spatially n -dimensional (meaning it has elliptic metric), how many independent rigid body modes would a body have? (Prove by induction)

9

MultiFreedom Constraints I

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§9.1. CLASSIFICATION OF CONSTRAINT CONDITIONS

In previous Chapters we have considered structural support conditions that are mathematically expressible as constraints on individual degrees of freedom:

$$\boxed{\text{nodal displacement component} = \text{prescribed value.}} \quad (9.1)$$

These are called *single-freedom constraints*. Chapters 3 and 4 explain how to incorporate constraints of this form into the master stiffness equations, using hand- or computer-oriented techniques. The displacement boundary conditions studied in Chapter 8, including the modeling of symmetry and antisymmetry, lead to constraints of this form.

For example:

$$u_{x4} = 0, \quad u_{y9} = 0.6. \quad (9.2)$$

These are two single-freedom constraints. The first one is homogeneous while the second one is non-homogeneous; these qualifiers are defined below.

§9.1.1. MultiFreedom Constraints

The next step up in complexity involves *multifreedom equality constraints*, or *multifreedom constraints* for short, the last name being acronymed to MFC. These are functional equations that connect *two or more* displacement components:

$$\boxed{F(\text{nodal displacement components}) = \text{prescribed value,}} \quad (9.3)$$

where function F vanishes if all its nodal displacement arguments do. Equation (9.3), where all displacement components are in the left-hand side, is called the *canonical form* of the constraint.

An MFC of this form is called *multipoint* or *multinode* if it involves displacement components at different nodes. The constraint is called *linear* if all displacement components appear linearly on the left-hand-side, and *nonlinear* otherwise.

The constraint is called *homogeneous* if, upon transferring all terms that depend on displacement components to the left-hand side, the right-hand side — the “prescribed value” in (9.3) — is zero. It is called *non-homogeneous* otherwise.

In this and next Chapter only linear constraints will be studied. Furthermore more attention is devoted to the homogeneous case, because it arises more frequently in practice.

REMARK 9.1

The most general constraint class is that of inequality constraints, such as $u_{y5} - 2u_{x2} \geq 0.5$. These constraints are relatively infrequent in linear structural analysis, except in problems that involve contact conditions. They are of paramount importance, however, in other fields such as optimization and control.

§9.1.2. MFC Examples

Here are three examples of MFCs:

$$u_{x2} = \frac{1}{2}u_{y2}, \quad u_{x2} - 2u_{x4} + u_{x6} = 0.25, \quad (x_5 + u_{x5} - x_3 - u_{x3})^2 + (y_5 + u_{y5} - y_3 - u_{y3})^2 = 0. \quad (9.4)$$

The first one is linear and homogeneous. It is not a multipoint constraint because it involves the displacement components of one node: 2.

The second one is multipoint because it involves three nodes: 2, 4 and 6. It is linear and non-homogeneous.

The last one is multipoint, nonlinear and homogeneous. Geometrically it expresses that the distance between nodes 3 and 5 in two-dimensional motions on the $\{x, y\}$ plane remains constant. This kind of constraint appears in geometrically nonlinear analysis of structures, which is a topic beyond the scope of this course.

§9.1.3. *MFC Matrix Forms

Matrix forms of linear MFCs are often convenient for compact notation. An individual constraint such as the second one in (9.4) can be written

$$[1 \quad -2 \quad 1] \begin{bmatrix} u_{x2} \\ u_{x4} \\ u_{x6} \end{bmatrix} = 0.25. \quad (9.5)$$

In direct matrix notation:

$$\bar{\mathbf{a}}_i \bar{\mathbf{u}}_i = b_i, \quad (\text{no sum on } i) \quad (9.6)$$

in which index i ($i = 1, 2, \dots$) identifies the constraint, $\bar{\mathbf{a}}_i$ is a row vector, $\bar{\mathbf{u}}_i$ collects the set of degrees of freedom that participate in the constraint, and b_i is the right hand side scalar (0.25 above). The bar over \mathbf{a} and \mathbf{u} distinguishes (9.6) from the expanded form (9.8) discussed below.

For method description and general proofs it is often convenient to expand matrix forms so that they embody *all* degrees of freedom. For example, if (9.5) is part of a two-dimensional finite element model with 12 freedoms: $u_{x1}, u_{y1}, \dots, u_{y6}$, the left-hand side row vector may be expanded with nine zeros as follows

$$[0 \quad 0 \quad 1 \quad 0 \quad 0 \quad 0 \quad -2 \quad 0 \quad 0 \quad 0 \quad 1 \quad 0] \begin{bmatrix} u_{x1} \\ u_{y1} \\ u_{x2} \\ \vdots \\ u_{y6} \end{bmatrix} = 0.25, \quad (9.7)$$

in which case the matrix notation

$$\mathbf{a}_i \mathbf{u} = g_i \quad (9.8)$$

is used. Finally, all multifreedom constraints expressed as (9.8) may be collected into a single matrix relation:

$$\mathbf{A} \mathbf{u} = \mathbf{g}, \quad (9.9)$$

where rectangular matrix \mathbf{A} is formed by stacking the \mathbf{a}_i 's as rows and column vector \mathbf{g} is formed by stacking the g_i s as entries. If there are 12 degrees of freedom in \mathbf{u} and 5 multifreedom constraints then \mathbf{A} will be 5×12 .

§9.2. METHODS FOR IMPOSING MULTIFREEDOM CONSTRAINTS

Accounting for multifreedom constraints is done — at least conceptually — by changing the assembled master stiffness equations to produce a modified system of equations. The modification process is also called *constraint application* or *constraint imposition*. The modified system is the one submitted to the equation solver.

Three methods for treating MFCs are discussed in this and the next Chapter:

1. *Master-Slave Elimination*. The degrees of freedom involved in each MFC are separated into master and slave freedoms. The slave freedoms are then explicitly eliminated. The modified equations do not contain the slave freedoms.
2. *Penalty Augmentation*. Also called the *penalty function method*. Each MFC is viewed as the presence of a fictitious elastic structural element called *penalty element* that enforces it approximately. This element is parametrized by a numerical *weight*. The exact constraint is recovered if the weight goes to infinity. The MFCs are imposed by augmenting the finite element model with the penalty elements.
3. *Lagrange Multiplier Adjunction*. For each MFC an additional unknown is adjoined to the master stiffness equations. Physically this set of unknowns represent *constraint forces* that would enforce the constraints exactly should they be applied to the unconstrained system.

For each method the exposition tries to give first the basic flavor by working out the same example for each method. The general technique is subsequently presented in matrix form for completeness but is considered an advanced topic.

Conceptually, imposing MFCs is not different from the procedure discussed in Chapters 3 and 4 for single-freedom constraints. The master stiffness equations are assembled ignoring all constraints. Then the MFCs are imposed by appropriate modification of those equations. There are, however, two important practical differences:

1. The modification process is not unique because there are alternative constraint imposition methods, namely those listed above. These methods offer tradeoffs in generality, programming implementation complexity, computational effort, numerical accuracy and stability.
2. In the implementation of some of these methods — particularly penalty augmentation — constraint imposition and assembly are carried out simultaneously. In that case the framework “first assemble, then modify,” is not strictly respected in the actual implementation.

REMARK 9.2

The three methods are also applicable, as can be expected, to the simpler case of a single-freedom constraint such as (9.2). For most situations, however, the generality afforded by the penalty function and Lagrange multiplier methods are not warranted. The hand-oriented reduction process discussed in Chapters 3 and 4 is in fact a special case of the master-slave elimination method in which “there is no master.”

REMARK 9.3

Often both multifreedom and single-freedom constraints are prescribed. The modification process then involves two stages: apply multifreedom constraints and apply single freedom constraints. The order in which these are carried out is implementation dependent. Most implementations do the MFCs first, either after the assembly is completed or during the assembly process. The reason is practical: single-freedom constraints are often automatically taken care of by the equation solver itself.

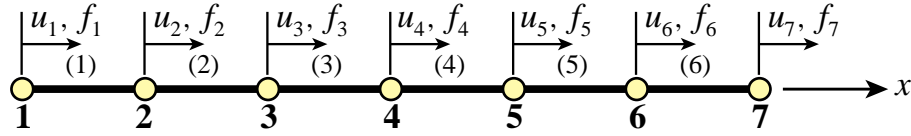


Figure 9.1. A one-dimensional problem discretized with six bar finite elements. The seven nodes may move only along the x direction. Subscript x is omitted from the u 's and f 's for simplicity.

§9.3. THE EXAMPLE STRUCTURE

The one-dimensional finite element discretization shown in Figure 9.1 will be used throughout Chapters 8–9 to illustrate the three MFC application methods. This structure consists of six bar elements connected by seven nodes that can only displace in the x direction.

Before imposing various multifreedom constraints discussed below, the master stiffness equations for this problem are assumed to be

$$\begin{bmatrix} K_{11} & K_{12} & 0 & 0 & 0 & 0 & 0 \\ K_{12} & K_{22} & K_{23} & 0 & 0 & 0 & 0 \\ 0 & K_{23} & K_{33} & K_{34} & 0 & 0 & 0 \\ 0 & 0 & K_{34} & K_{44} & K_{45} & 0 & 0 \\ 0 & 0 & 0 & K_{45} & K_{55} & K_{56} & 0 \\ 0 & 0 & 0 & 0 & K_{56} & K_{66} & K_{67} \\ 0 & 0 & 0 & 0 & 0 & K_{67} & K_{77} \end{bmatrix} \begin{bmatrix} u_1 \\ u_2 \\ u_3 \\ u_4 \\ u_5 \\ u_6 \\ u_7 \end{bmatrix} = \begin{bmatrix} f_1 \\ f_2 \\ f_3 \\ f_4 \\ f_5 \\ f_6 \\ f_7 \end{bmatrix}, \quad (9.10)$$

or

$$\mathbf{K}\mathbf{u} = \mathbf{f}. \quad (9.11)$$

The nonzero stiffness coefficients K_{ij} in (9.10) depend on the bar rigidity properties. For example, if $E^{(e)}A^{(e)}/L^{(e)} = 100$ for each element $e = 1, \dots, 6$, then $K_{11} = K_{77} = 100$, $K_{22} = \dots = K_{66} = 200$, $K_{12} = K_{23} = \dots = K_{67} = -100$. However, for the purposes of the following treatment the coefficients may be kept arbitrary. The component index x in the nodal displacements u and nodal forces f has been omitted for brevity.

Now let us specify a multifreedom constraint that states that nodes 2 and 6 are to move by the same amount:

$$u_2 = u_6. \quad (9.12)$$

Passing all node displacements to the right hand side gives the canonical form:

$$\boxed{u_2 - u_6 = 0.} \quad (9.13)$$

Constraint conditions of this type are sometimes called *rigid links* because they can be mechanically interpreted as forcing node points 2 and 6 to move together as if they were tied by a rigid member.¹

¹ This physical interpretation is exploited in the penalty method described in the next Chapter. In two and three dimensions rigid link constraints are more complicated.

We now study the imposition of constraint (9.13) on the master equations (9.11) by the methods mentioned above. In this Chapter the master-slave method is treated. The other two methods: penalty augmentation and Lagrange multiplier adjunction, are discussed in the following Chapter.

§9.4. THE MASTER-SLAVE METHOD

To apply this method *by hand*, the MFCs are taken one at a time. For each constraint a *slave* degree of freedom is chosen. The freedoms remaining in that constraint are labeled *master*. A new set of degrees of freedom $\hat{\mathbf{u}}$ is established by removing all slave freedoms from \mathbf{u} . This new vector contains master freedoms as well as those that do not appear in the MFCs. A matrix transformation equation that relates \mathbf{u} to $\hat{\mathbf{u}}$ is generated. This equation is used to apply a congruential transformation to the master stiffness equations. This procedure yields a set of modified stiffness equations that are expressed in terms of the new freedom set $\hat{\mathbf{u}}$. Because the modified system does not contain the slave freedoms, these have been effectively eliminated.

§9.4.1. A One-Constraint Example

The mechanics of the process is best seen by going through an example. To impose (9.13) choose u_6 as slave and u_2 as master. Relate the original unknowns u_1, \dots, u_7 to the new set in which u_6 is missing:

$$\begin{bmatrix} u_1 \\ u_2 \\ u_3 \\ u_4 \\ u_5 \\ u_6 \\ u_7 \end{bmatrix} = \begin{bmatrix} 1 & 0 & 0 & 0 & 0 & 0 \\ 0 & 1 & 0 & 0 & 0 & 0 \\ 0 & 0 & 1 & 0 & 0 & 0 \\ 0 & 0 & 0 & 1 & 0 & 0 \\ 0 & 0 & 0 & 0 & 1 & 0 \\ 0 & 1 & 0 & 0 & 0 & 0 \\ 0 & 0 & 0 & 0 & 0 & 1 \end{bmatrix} \begin{bmatrix} u_1 \\ u_2 \\ u_3 \\ u_4 \\ u_5 \\ u_7 \end{bmatrix}, \quad (9.14)$$

This is the required transformation relation. In compact form:

$$\boxed{\mathbf{u} = \mathbf{T}\hat{\mathbf{u}}}. \quad (9.15)$$

Replacing (9.15) into (9.11) and premultiplying by \mathbf{T}^T yields the modified system

$$\boxed{\hat{\mathbf{K}}\hat{\mathbf{u}} = \hat{\mathbf{f}}, \quad \text{in which} \quad \hat{\mathbf{K}} = \mathbf{T}^T \mathbf{K} \mathbf{T}, \quad \hat{\mathbf{f}} = \mathbf{T}^T \mathbf{f}. \quad (9.16)}$$

Carrying out the indicated matrix multiplications yields

$$\begin{bmatrix} K_{11} & K_{12} & 0 & 0 & 0 & 0 \\ K_{12} & K_{22} + K_{66} & K_{23} & 0 & K_{56} & K_{67} \\ 0 & K_{23} & K_{33} & K_{34} & 0 & 0 \\ 0 & 0 & K_{34} & K_{44} & K_{45} & 0 \\ 0 & K_{56} & 0 & K_{45} & K_{55} & 0 \\ 0 & K_{67} & 0 & 0 & 0 & K_{77} \end{bmatrix} \begin{bmatrix} u_1 \\ u_2 \\ u_3 \\ u_4 \\ u_5 \\ u_7 \end{bmatrix} = \begin{bmatrix} f_1 \\ f_2 + f_6 \\ f_3 \\ f_4 \\ f_5 \\ f_7 \end{bmatrix}, \quad (9.17)$$

Equation (9.17) is a new linear system containing 6 equations in the remaining 6 unknowns: u_1, u_2, u_3, u_4, u_5 and u_7 . Upon solving it, u_6 is recovered from the constraint (9.12).

REMARK 9.4

The form of modified system (9.16) can be remembered by a simple mnemonic rule: premultiply both sides of $\mathbf{u} = \mathbf{T} \hat{\mathbf{u}}$ by $\mathbf{T}^T \mathbf{K}$, and replace $\mathbf{K} \mathbf{u}$ by \mathbf{f} on the right hand side.

REMARK 9.5

For a simple freedom constraint such as $u_4 = 0$ the only possible choice of slave is of course u_4 and there is no master. The congruential transformation is then nothing more than the elimination of u_4 by striking out rows and columns from the master stiffness equations.

REMARK 9.6

For a simple MFC such as $u_2 = u_6$, it does not matter which degree of freedom is chosen as master or unknown. Choosing u_2 as slave produces a system of equations in which now u_2 is missing:

$$\begin{bmatrix} K_{11} & 0 & 0 & 0 & K_{12} & 0 \\ 0 & K_{33} & K_{34} & 0 & K_{23} & 0 \\ 0 & K_{34} & K_{44} & K_{45} & 0 & 0 \\ 0 & 0 & K_{45} & K_{55} & K_{56} & 0 \\ K_{12} & K_{23} & 0 & K_{56} & K_{22} + K_{66} & K_{67} \\ 0 & 0 & 0 & 0 & K_{67} & K_{77} \end{bmatrix} \begin{bmatrix} u_1 \\ u_3 \\ u_4 \\ u_5 \\ u_6 \\ u_7 \end{bmatrix} = \begin{bmatrix} f_1 \\ f_3 \\ f_4 \\ f_5 \\ f_2 + f_6 \\ f_7 \end{bmatrix}, \quad (9.18)$$

Although (9.17) and (9.18) are algebraically equivalent, the latter would be processed faster if a skyline solver (Part III of course) is used for the modified equations.

§9.4.2. Several Homogeneous MFCs

The matrix equation (9.16) in fact holds for the general case of multiple homogeneous linear constraints. Direct establishment of the transformation equation, however, is more complicated if slave freedoms in one constraint appear as masters in another. To illustrate this point, suppose that for the example system we have three homogeneous multifreedom constraints:

$$u_2 - u_6 = 0, \quad u_1 + 4u_4 = 0, \quad 2u_3 + u_4 + u_5 = 0, \quad (9.19)$$

Picking as slave freedoms u_6 , u_4 and u_3 from the first, second and third constraint, respectively, we can solve for them as

$$u_6 = u_2, \quad u_4 = -\frac{1}{4}u_1, \quad u_3 = -\frac{1}{2}(u_4 + u_5) = \frac{1}{8}u_1 - \frac{1}{2}u_5. \quad (9.20)$$

Observe that solving for u_3 from the third constraint brings u_4 to the right-hand side. But because u_4 is also a slave freedom (it was chosen as such for the second constraint) it is replaced in favor of u_1 using $u_4 = -\frac{1}{4}u_1$. The matrix form of the transformation (9.20) is

$$\begin{bmatrix} u_1 \\ u_2 \\ u_3 \\ u_4 \\ u_5 \\ u_6 \\ u_7 \end{bmatrix} = \begin{bmatrix} 1 & 0 & 0 & 0 \\ 0 & 1 & 0 & 0 \\ \frac{1}{8} & 0 & -\frac{1}{2} & 0 \\ -\frac{1}{4} & 0 & 0 & 0 \\ 0 & 0 & 1 & 0 \\ 0 & 1 & 0 & 0 \\ 0 & 0 & 0 & 1 \end{bmatrix} \begin{bmatrix} u_1 \\ u_2 \\ u_5 \\ u_7 \end{bmatrix}, \quad (9.21)$$

The modified system is now formed through the congruential transformation (9.16). Note that the slave freedoms selected from each constraint must be distinct; for example the choice u_6, u_4, u_4 would be inadmissible as long as the constraints are independent. This rule is easy to enforce when slave freedoms are chosen by hand, but can lead to implementation and numerical difficulties when it is programmed as an automated procedure, as further discussed later.

REMARK 9.7

The three MFCs (9.20) with u_6, u_4 and u_2 chosen as slaves and u_1, u_2 and u_5 chosen as masters, may be presented in the partitioned matrix form:

$$\begin{bmatrix} 0 & 0 & 1 \\ 0 & 4 & 0 \\ 2 & 1 & 0 \end{bmatrix} \begin{bmatrix} u_3 \\ u_4 \\ u_6 \end{bmatrix} = \begin{bmatrix} 0 & 1 & 0 \\ -1 & 0 & 0 \\ 0 & 0 & -1 \end{bmatrix} \begin{bmatrix} u_1 \\ u_2 \\ u_5 \end{bmatrix} \quad (9.22)$$

This may be compactly written $\mathbf{A}_s \mathbf{u}_s + \mathbf{A}_m \mathbf{u}_m = \mathbf{0}$. Solving for the slave freedoms gives $\mathbf{u}_s = -\mathbf{A}_s^{-1} \mathbf{A}_m \mathbf{u}_m$. Expanding with zeros to fill out \mathbf{u} and $\hat{\mathbf{u}}$ produces (9.21). This general matrix form is considered in §9.4.4. Note that non-singularity of \mathbf{A}_s is essential for this method to work.

§9.4.3. Nonhomogeneous MFCs

Extension to non-homogeneous constraints is immediate. In such a case the transformation equation becomes non-homogeneous. For example suppose that (9.15) contains a nonzero prescribed value:

$$\boxed{u_2 - u_6 = 0.2} \quad (9.23)$$

Nonzero RHS values such as 0.2 in (9.23) may be often interpreted physically as “gaps” (thus the use of the symbol \mathbf{g} in the matrix form). Chose u_6 again as slave: $u_6 = u_2 - 0.2$, and build the transformation

$$\begin{bmatrix} u_1 \\ u_2 \\ u_3 \\ u_4 \\ u_5 \\ u_6 \\ u_7 \end{bmatrix} = \begin{bmatrix} 1 & 0 & 0 & 0 & 0 & 0 \\ 0 & 1 & 0 & 0 & 0 & 0 \\ 0 & 0 & 1 & 0 & 0 & 0 \\ 0 & 0 & 0 & 1 & 0 & 0 \\ 0 & 0 & 0 & 0 & 1 & 0 \\ 0 & 1 & 0 & 0 & 0 & 0 \\ 0 & 0 & 0 & 0 & 0 & 1 \end{bmatrix} \begin{bmatrix} u_1 \\ u_2 \\ u_3 \\ u_4 \\ u_5 \\ u_6 \\ u_7 \end{bmatrix} + \begin{bmatrix} 0 \\ 0 \\ 0 \\ 0 \\ 0 \\ -0.2 \\ 0 \end{bmatrix}. \quad (9.24)$$

In compact matrix notation,

$$\boxed{\mathbf{u} = \mathbf{T} \hat{\mathbf{u}} + \mathbf{g}.} \quad (9.25)$$

Here the constraint gap vector \mathbf{g} is nonzero and \mathbf{T} is the same as before. To get the modified system applying the shortcut rule of Remark 9.4, premultiply both sides of (9.25) by $\mathbf{T}^T \mathbf{K}$, replace $\mathbf{K} \mathbf{u}$ by \mathbf{f} , and pass the data to the RHS:

$$\boxed{\hat{\mathbf{K}} \hat{\mathbf{u}} = \hat{\mathbf{f}}, \quad \text{in which} \quad \hat{\mathbf{K}} = \mathbf{T}^T \mathbf{K} \mathbf{T}, \quad \hat{\mathbf{f}} = \mathbf{T}^T (\mathbf{f} - \mathbf{K} \mathbf{g}).} \quad (9.26)$$

Upon solving (9.26) for $\hat{\mathbf{u}}$, the complete displacement vector is recovered from (9.25).

For the MFC (9.22) this technique gives the system

$$\begin{bmatrix} K_{11} & K_{12} & 0 & 0 & 0 & 0 \\ K_{12} & K_{22} + K_{66} & K_{23} & 0 & K_{56} & K_{67} \\ 0 & K_{23} & K_{33} & K_{34} & 0 & 0 \\ 0 & 0 & K_{34} & K_{44} & K_{45} & 0 \\ 0 & K_{56} & 0 & K_{45} & K_{55} & 0 \\ 0 & K_{67} & 0 & 0 & 0 & K_{77} \end{bmatrix} \begin{bmatrix} u_1 \\ u_2 \\ u_3 \\ u_4 \\ u_5 \\ u_7 \end{bmatrix} = \begin{bmatrix} f_1 \\ f_2 + f_6 - 0.2K_{66} \\ f_3 \\ f_4 \\ f_5 - 0.2K_{56} \\ f_7 - 0.2K_{67} \end{bmatrix}. \quad (9.27)$$

See Exercise 9.2 for multiple non-homogeneous MFCs.

§9.4.4. *The General Case

For implementation in general-purpose programs the master-slave method can be described as follows. The degrees of freedoms in \mathbf{u} are classified into three types: independent or uncommitted, masters and slaves. (The uncommitted freedoms are those that do not appear in any MFC.) Label these sets as \mathbf{u}_u , \mathbf{u}_m and \mathbf{u}_s , respectively, and partition the stiffness equations accordingly:

$$\begin{bmatrix} \mathbf{K}_{uu} & \mathbf{K}_{um} & \mathbf{K}_{us} \\ \mathbf{K}_{um}^T & \mathbf{K}_{mm} & \mathbf{K}_{ms} \\ \mathbf{K}_{us}^T & \mathbf{K}_{ms}^T & \mathbf{K}_{ss} \end{bmatrix} \begin{bmatrix} \mathbf{u}_u \\ \mathbf{u}_m \\ \mathbf{u}_s \end{bmatrix} = \begin{bmatrix} \mathbf{f}_u \\ \mathbf{f}_m \\ \mathbf{f}_s \end{bmatrix} \quad (9.28)$$

The MFCs may be written in matrix form as

$$\mathbf{A}_m \mathbf{u}_m + \mathbf{A}_s \mathbf{u}_s = \mathbf{g}, \quad (9.29)$$

where \mathbf{A}_s is assumed square and nonsingular. If so we can solve for the slave freedoms:

$$\mathbf{u}_s = -\mathbf{A}_s^{-1} \mathbf{A}_m \mathbf{u}_m + \mathbf{A}_s^{-1} \mathbf{g} = \mathbf{T} \mathbf{u}_m + \mathbf{g}, \quad (9.30)$$

Inserting into the partitioned stiffness matrix and symmetrizing

$$\begin{bmatrix} \mathbf{K}_{uu} & \mathbf{K}_{um} \mathbf{T} \\ \mathbf{T}^T \mathbf{K}_{um}^T & \mathbf{T}^T \mathbf{K}_{mm} \mathbf{T} \end{bmatrix} \begin{bmatrix} \mathbf{u}_u \\ \mathbf{u}_m \end{bmatrix} = \begin{bmatrix} \mathbf{f}_u - \mathbf{K}_{us} \mathbf{g} \\ \mathbf{f}_m - \mathbf{K}_{ms} \mathbf{g} \end{bmatrix} \quad (9.31)$$

It is seen that the misleading simplicity of the handworked examples is gone.

§9.4.5. *Retaining the Original Freedoms

A potential disadvantage of the master-slave method in computer work is that it requires a rearrangement of the original stiffness equations because $\hat{\mathbf{u}}$ is a subset of \mathbf{u} . The disadvantage can be annoying when sparse matrix storage schemes are used for the stiffness matrix, and becomes intolerable if secondary storage is used for that purpose.

With a bit of trickery it is possible to maintain the original freedom ordering. Let us display it for the example problem under (9.13). Instead of (9.14), use the *square* transformation

$$\begin{bmatrix} u_1 \\ u_2 \\ u_3 \\ u_4 \\ u_5 \\ u_6 \\ u_7 \end{bmatrix} = \begin{bmatrix} 1 & 0 & 0 & 0 & 0 & 0 & 0 \\ 0 & 1 & 0 & 0 & 0 & 0 & 0 \\ 0 & 0 & 1 & 0 & 0 & 0 & 0 \\ 0 & 0 & 0 & 1 & 0 & 0 & 0 \\ 0 & 0 & 0 & 0 & 1 & 0 & 0 \\ 0 & 1 & 0 & 0 & 0 & 0 & 0 \\ 0 & 0 & 0 & 0 & 0 & 0 & 1 \end{bmatrix} \begin{bmatrix} u_1 \\ u_2 \\ u_3 \\ u_4 \\ u_5 \\ \tilde{u}_6 \\ u_7 \end{bmatrix}, \quad (9.32)$$

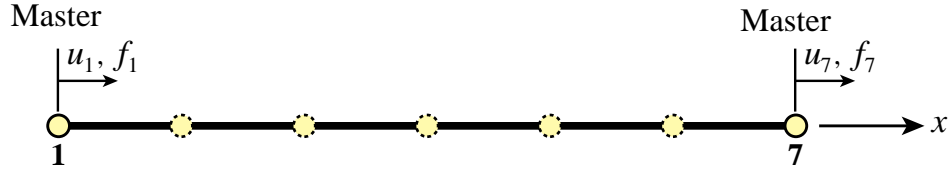


Figure 9.2. Model reduction of the example structure of Figure 9.1 to the end freedoms.

where \tilde{u}_6 is a *placeholder* for the slave freedom u_6 . The modified equations are

$$\begin{bmatrix} K_{11} & K_{12} & 0 & 0 & 0 & 0 & 0 & 0 \\ K_{12} & K_{22} + K_{66} & K_{23} & 0 & 0 & K_{56} & 0 & K_{67} \\ 0 & K_{23} & K_{33} & K_{34} & 0 & 0 & 0 & 0 \\ 0 & 0 & K_{34} & K_{44} & K_{45} & 0 & 0 & 0 \\ 0 & K_{56} & 0 & K_{45} & K_{55} & 0 & 0 & 0 \\ 0 & 0 & 0 & 0 & 0 & 0 & 0 & 0 \\ 0 & K_{67} & 0 & 0 & 0 & 0 & 0 & K_{77} \end{bmatrix} \begin{bmatrix} u_1 \\ u_2 \\ u_3 \\ u_4 \\ u_5 \\ \tilde{u}_6 \\ u_7 \end{bmatrix} = \begin{bmatrix} f_1 \\ f_2 + f_6 \\ f_3 \\ f_4 \\ f_5 \\ 0 \\ f_7 \end{bmatrix}, \quad (9.33)$$

which are submitted to the equation solver. If the solver is not trained to skip zero row and columns, a one should be placed in the diagonal entry for the \tilde{u}_6 (sixth) equation. The solver will return $\tilde{u}_6 = 0$, and this placeholder value is replaced by u_2 . Note the points in common with the computer-oriented placeholder technique described in §3.4 to handle single-freedom constraints.

§9.4.6. Model Reduction by Kinematic Constraints

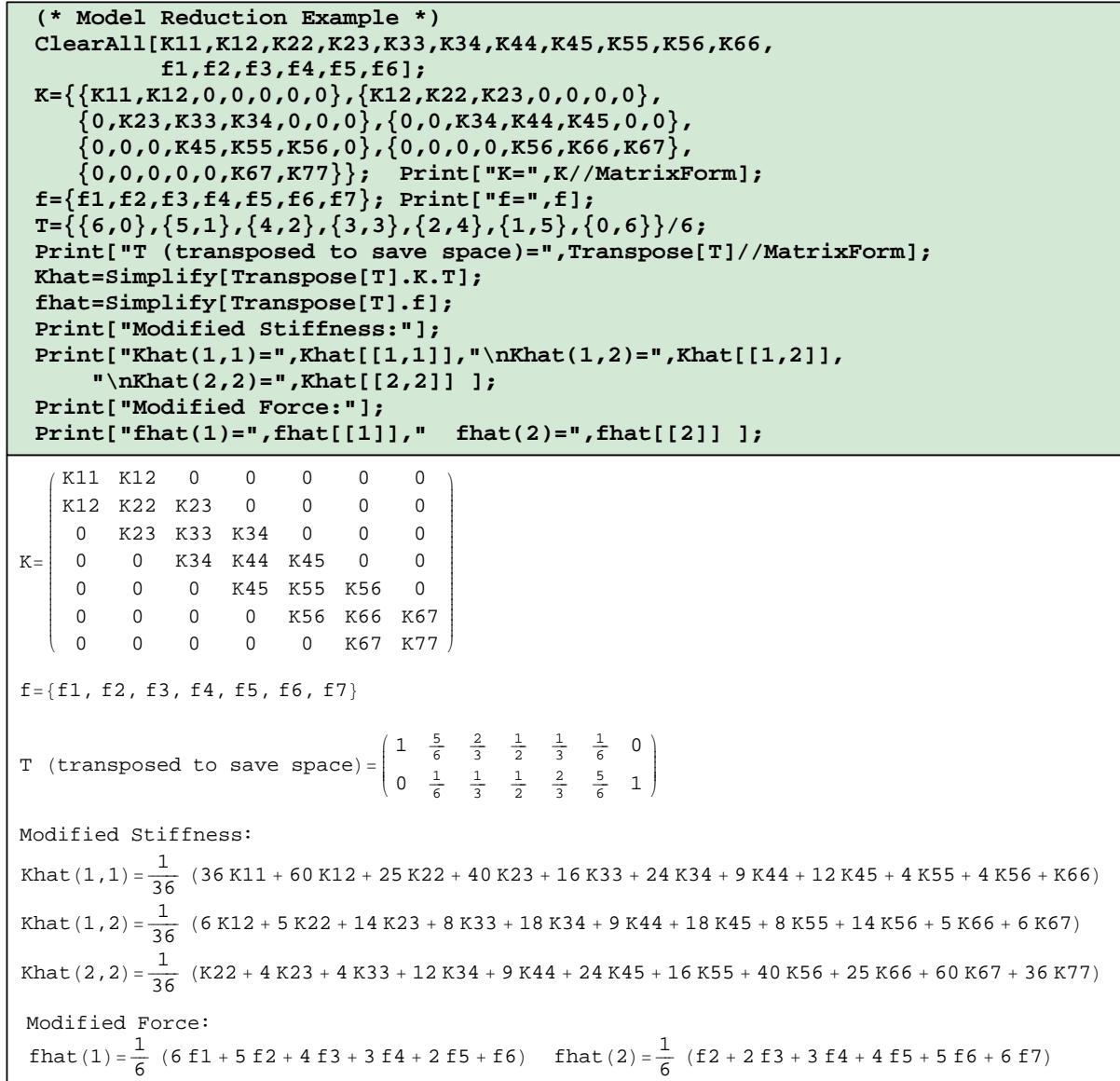
The congruential transformation equations (9.16) and (9.26) have additional applications beyond the master-slave method. An important one is *model reduction by kinematic constraints*. Through this procedure the number of DOF of a static or dynamic FEM model is reduced by a significant number, typically to 1% – 10% of the original number. This is done by taking a lot of slaves and a few masters. Only the masters are left after the transformation. The reduced model is commonly used in subsequent calculations as component of a larger system, particularly during design or in parameter identification.

To illustrate the method for a static model, consider the bar assembly of Figure 9.1. Assume that the only masters are the end motions u_1 and u_7 , as illustrated in Figure 9.2, and interpolate all freedoms linearly:

$$\begin{bmatrix} u_1 \\ u_2 \\ u_3 \\ u_4 \\ u_5 \\ u_6 \\ u_7 \end{bmatrix} = \begin{bmatrix} 1 & 0 \\ 5/6 & 1/6 \\ 4/6 & 2/6 \\ 3/6 & 3/6 \\ 2/6 & 4/6 \\ 1/6 & 5/6 \\ 0 & 1 \end{bmatrix} \begin{bmatrix} u_1 \\ u_7 \end{bmatrix}, \quad \text{or} \quad \mathbf{u} = \mathbf{T} \hat{\mathbf{u}}. \quad (9.34)$$

The reduced-model equations are $\hat{\mathbf{K}} \hat{\mathbf{u}} = \mathbf{T}^T \mathbf{K} \mathbf{T} \hat{\mathbf{u}} = \mathbf{T}^T \mathbf{f} = \hat{\mathbf{f}}$, or in detail

$$\begin{bmatrix} \hat{K}_{11} & \hat{K}_{17} \\ \hat{K}_{17} & \hat{K}_{77} \end{bmatrix} \begin{bmatrix} u_1 \\ u_7 \end{bmatrix} = \begin{bmatrix} \hat{f}_1 \\ \hat{f}_7 \end{bmatrix}, \quad (9.35)$$

Figure 9.3. *Mathematica* script for the model reduction example of Figure 9.2.

where

$$\begin{aligned} \hat{K}_{11} &= \frac{1}{36} (36K_{11} + 60K_{12} + 25K_{22} + 40K_{23} + 16K_{33} + 24K_{34} + 9K_{44} + 12K_{45} + 4K_{55} + 4K_{56} + K_{66}), \\ \hat{K}_{17} &= \frac{1}{36} (6K_{12} + 5K_{22} + 14K_{23} + 8K_{33} + 18K_{34} + 9K_{44} + 18K_{45} + 8K_{55} + 14K_{56} + 5K_{66} + 6K_{67}), \\ \hat{K}_{77} &= \frac{1}{36} (K_{22} + 4K_{23} + 4K_{33} + 12K_{34} + 9K_{44} + 24K_{45} + 16K_{55} + 40K_{56} + 25K_{66} + 60K_{67} + 36K_{77}), \\ \hat{f}_1 &= \frac{1}{6} (6f_1 + 5f_2 + 4f_3 + 3f_4 + 2f_5 + f_6), \quad \hat{f}_7 = \frac{1}{6} (f_2 + 2f_3 + 3f_4 + 4f_5 + 5f_6 + 6f_7). \end{aligned} \quad (9.36)$$

This reduces the order of the FEM model from 7 to 2. A *Mathematica* script to do the reduction is shown in Figure 9.3. The key feature is that the masters are picked *a priori*, as the freedoms to be retained in the model for further use.

REMARK 9.8

Model reduction can also be done by the static condensation method explained in Chapter 11. As its name indicates, condensation is restricted to static analysis. On the other hand, for such problems it is exact whereas model reduction by kinematic constraints generally introduces approximations.

§9.4.7. Assessment of the Master-Slave Method

What are the good and bad points of this constraint imposition method? It enjoys the advantage of being exact (except for inevitable solution errors) and of reducing the number of unknowns. The concept is also easy to explain. The main implementation drawback is the complexity of the general case as can be seen by studying (9.28) through (9.31). The complexity is due to three factors:

1. The equations may have to be rearranged because of the disappearance of the slave freedoms. This drawback can be alleviated, however, through the placeholder trick outlined in §9.4.5.
2. An auxiliary linear system, namely (9.30), has to be assembled and solved to produce the transformation matrix \mathbf{T} and vector \mathbf{g} .
3. The transformation process may generate many additional matrix terms. If a sparse matrix storage scheme is used for \mathbf{K} , the logic for allocating memory and storing these entries can be difficult and expensive.

The level of complexity depends on the generality allowed as well as on programming decisions. At one extreme, if \mathbf{K} is stored as full matrix and slave freedom coupling in the MFCs is disallowed the logic is simple.² At the other extreme, if arbitrary couplings are permitted and \mathbf{K} is placed in secondary (disk) storage according to some sparse scheme, the complexity can become overwhelming.

Another, more subtle, drawback of this method is that it requires decisions as to which degrees of freedom are to be treated as slaves. This can lead to implementation and numerical stability problems. Although for disjointed constraints the process can be programmed in reliable form, in more general cases of coupled constraint equations it can lead to incorrect decisions. For example, suppose that in the example problem you have the following two MFCs:

$$\frac{1}{6}u_2 + \frac{1}{2}u_4 = u_6, \quad u_3 + 6u_6 = u_7 \quad (9.37)$$

For numerical stability reasons it is usually better to pick as slaves the freedoms with larger coefficients. If this is done, the program would select u_6 as slave freedoms from both constraints. This leads to a contradiction because having two constraints we must eliminate two slave degrees of freedom, not just one. The resulting modified system would in fact be inconsistent. Although this defect can be easily fixed by the program logic in this case, one can imagine the complexity burden if faced with hundreds or thousands of MFCs.

Serious numerical problems can arise if the MFCs are not independent. For example:

$$\frac{1}{6}u_2 = u_6, \quad \frac{1}{5}u_3 + 6u_6 = u_7, \quad u_2 + u_3 - u_7 = 0. \quad (9.38)$$

² This is the case in model reduction, since each slave freedom appears in one and only one MFC.

The last constraint is an exact linear combination of the first two. If the program blindly chooses u_2 , u_3 and u_7 as slaves, the modified system is incorrect because we eliminate three equations when in fact there are only two independent constraints.

Exact linear dependence, as in (9.38), can be recognized by a rank analysis of the \mathbf{A}_s matrix defined in (9.29). In inexact floating-point arithmetic, however, such detection may fail.³

The complexity of slave selection is in fact equivalent to that of automatically selecting kinematic redundancies in the force method. It has led implementors of programs that use this method to require masters and slaves be prescribed in the input data, thus transferring the burden to users.

The method is not generally extendible to nonlinear constraints without extensive reworking.

In conclusion, the master-slave method is useful when a few simple linear constraints are imposed by hand. As a general purpose technique for finite element analysis it suffers from complexity and lack of robustness. It is worth learning this method, however, because of its great importance of the congruential transformation technique in *model reduction* for static and dynamic problems.

Notes and Bibliography

Multifreedom constraints are treated in several of the FEM books recommended in §1.7.5, notably Zienkiewicz and Taylor [9.4]. The master-slave method was incorporated to treat MFCs as part of the DSM developed at Boeing during the 1950s. It is first summarily described in DSM-overview by Turner, Martin and Weikel [9.3, p. 212]; however the implementation differs from the one described here since the relation of FEM to energy methods was not clear at the time.

The MS method was popularized through its adoption by the general-purpose NASTRAN code developed in the late 1960s [9.1] and early assessments [9.2]. The implementation unfortunately relied on user inputs to identify slave DOFs. Through this serious blunder the method gained a reputation for unreliability that persists to the present day.

The important application of MS to model reduction, which by itself justifies teaching the method, is largely ignored in FEM textbooks.

References

- [9.1] Anonymous, The NASTRAN Theoretical Manual, NASA SP-221 (1970); The NASTRAN User's Manual, NASA SP-222 (1970); The NASTRAN Programmer's Manual, NASA SP-223 (1970); The NASTRAN Demonstration Problem Manual, NASA SP-223 (1970).
- [9.2] Tocher, J. L. and Herness, E. D., A critical view of NASTRAN, in: *Numerical and Computer Codes in Structural Mechanics*, ed. by S. J. Fenes, N. Perrone, A. R. Robinson and W. C. Schnobrich, Academic Press, New York, 1973, pp. 151–173.
- [9.3] Turner, M. J., Martin, H. C., Weikel, R. C., Further development and applications of the stiffness method, in *AGARDograph 72: Matrix Methods of Structural Analysis*, ed. by B. M. Fraeijs de Veubeke, Pergamon Press, New York, 203–266, 1964.
- [9.4] Zienkiewicz, O. C., Taylor, R. E., *The Finite Element Method*, 4th ed., McGraw-Hill, London, Vol. I: 1988, Vol II: 1993.

³ The safest technique to identify dependencies is to do a singular-value decomposition (SVD) of \mathbf{A}_s . This can be, however, prohibitively expensive if one is dealing with hundreds or thousands of constraints.

Homework Exercises for Chapter 9

MultiFreedom Constraints I

EXERCISE 9.1

[C+N:20] The example structure of Figure 9.1 has $E^{(e)}A^{(e)}/L^{(e)} = 100$ for each element $e = 1, \dots, 6$. Consequently $K_{11} = K_{77} = 100$, $K_{22} = \dots = K_{66} = 200$, $K_{12} = K_{23} = \dots = K_{67} = -100$. The applied node forces are taken to be $f_1 = 1$, $f_2 = 2$, $f_3 = 3$, $f_4 = 4$, $f_5 = 5$, $f_6 = 6$ and $f_7 = 7$, which are easy to remember. The structure is subjected to one support condition: $u_1 = 0$ (a fixed left end), and to one MFC: $u_2 - u_6 = 1/5$.

Solve this problem using the master-slave method to process the MFC, taking u_6 as slave. Upon forming the modified system (9.30) apply the left-end support $u_1 = 0$ using the placeholder method of §3.4. Solve the equations and verify that the displacement solution and the recovered node forces including reactions are

$$\begin{aligned} \mathbf{u} &= [0 \quad 0.270 \quad 0.275 \quad 0.250 \quad 0.185 \quad 0.070 \quad 0.140]^T \\ \mathbf{Ku} &= [-27 \quad 26.5 \quad 3 \quad 4 \quad 5 \quad -18.5 \quad 7]^T \end{aligned} \quad (\text{E9.1})$$

Use *Mathematica* or *Matlab* to do the algebra is recommended. For example, the following *Mathematica* script solves this Exercise:

```
(* Exercise 9.1 - Master-Slave Method *)
(* MFC: u2-u6 = 1/5 - slave: u6 *)
MasterStiffnessOfSixElementBar[kbar_] := Module[
  {K=Table[0,{7},{7}]}, K[[1,1]]=K[[7,7]]=kbar;
  For [i=2,i<=6,i++,K[[i,i]]=2*kbar];
  For [i=1,i<=6,i++,K[[i,i+1]]=K[[i+1,i]]=-kbar];
  Return[K]];
FixLeftEndOfSixElementBar[Khat_,fhat_] := Module[
  {Kmod=Khat,fmod=fhat}, fmod[[1]]=0; Kmod[[1,1]]=1;
  Kmod[[1,2]]=Kmod[[2,1]]=0; Return[{Kmod,fmod}]];

K=MasterStiffnessOfSixElementBar[100];
Print["Stiffness K=",K//MatrixForm];
f={1,2,3,4,5,6,7}; Print["Applied forces=",f];
T={{1,0,0,0,0,0},{0,1,0,0,0,0},{0,0,1,0,0,0},
  {0,0,0,1,0,0},{0,0,0,0,1,0},{0,1,0,0,0,0},
  {0,0,0,0,0,1}};
Print["Transformation matrix T=",T//MatrixForm];
g={0,0,0,0,0,-1/5,0};
Print["Constraint gap vector g=",g];
Khat=Simplify[Transpose[T].K.T]; fhat=Simplify[Transpose[T].(f-K.g)];
{Kmod,fmod}=FixLeftEndOfSixElementBar[Khat,fhat]; (* fix left end *)
Print["Modified Stiffness upon fixing node 1:",Kmod//MatrixForm];
Print["Modified RHS upon fixing node 1:",fmod];
umod=LinearSolve[Kmod,fmod];
Print["Computed umod (lacks slave u6)=",umod];
u=T.umod+g; Print["Complete solution u=",u];
Print["Numerical u=",N[u]];
fu=K.u; Print["Recovered forces K.u with reactions=",fu];
Print["Numerical K.u=",N[fu]];
```

EXERCISE 9.2

[C+N:25] As in the previous Exercise but applying the following three MFCs, two of which are non-homogeneous:

$$u_2 - u_6 = 1/5, \quad u_3 + 2u_4 = -2/3, \quad 2u_3 - u_4 + u_5 = 0. \quad (\text{E9.2})$$

Hints. Chose u_4, u_5 and u_6 as slaves. Much of the script shown for Exercise 9.1 can be reused. The main changes are in the formation of \mathbf{T} and \mathbf{g} . If you are a *Mathematica* wizard (or willing to be one) those can be automatically formed by saying

```
sol=Simplify[Solve[{u2-u6==1/5, u3+2*u4==-2/3, 2*u3-u4+u5==0},{u4,u5,u6}]];
ums={u1,u2,u3,u4,u5,u6,u7}/.sol[[1]]; um={u1,u2,u3,u7};
T=Table[Coefficient[ums[[i]],um[[j]]],{i,1,7},{j,1,4}];
g=ums/.{u1->0,u2->0,u3->0,u4->0,u5->0,u6->0,u7->0};
Print["Transformation matrix T=",T//MatrixForm];
Print["Gap vector g=",g]
```

If you do this, explain what it does and why it works. Otherwise form and enter \mathbf{T} and \mathbf{g} by hand. The numerical results (shown to 5 places) should be

$$\begin{aligned} \mathbf{u} &= [0. \quad 0.043072 \quad -0.075033 \quad -0.29582 \quad -0.14575 \quad -0.15693 \quad -0.086928]^T, \\ \mathbf{Ku} &= [-4.3072 \quad 16.118 \quad 10.268 \quad -37.085 \quad 16.124 \quad -8.1176 \quad 7.]^T. \end{aligned} \quad (\text{E9.3})$$

EXERCISE 9.3

[A:25] Can the MFCs be pre-processed to make sure that no slave freedom in a MFC appears as master in another?

EXERCISE 9.4

[A:25] In the general case discussed in §9.4.4, under which condition is the matrix \mathbf{A}_s of (9.32) diagonal and thus trivially invertible?

EXERCISE 9.5

[A:25] Work out the general technique by which the unknowns need not be rearranged, that is, \mathbf{u} and $\hat{\mathbf{u}}$ are the same. Use “placeholders” for the slave freedoms. (Hint: use ideas of §3.4).

EXERCISE 9.6

[A/C:35] Is it possible to establish a slave selection strategy that makes \mathbf{A}_s diagonal or triangular? (This requires knowledge of matrix techniques such as pivoting.)

EXERCISE 9.7

[A/C:40] Work out a strategy that produces a well conditioned \mathbf{A}_s by selecting new slaves as linear combinations of finite element freedoms if necessary. (Requires background in numerical analysis and advanced programming experience in matrix algebra).

10

MultiFreedom Constraints II

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In this Chapter we continue the discussion of methods to treat multifreedom constraints (MFCs). The master-slave method described previously was found to exhibit serious shortcomings for treating arbitrary constraints, although the method has important applications to model reduction.

We now pass to the study of two other methods: penalty augmentation and Lagrange multiplier adjunction. Both of these techniques are better suited to general implementations of the Finite Element Method.

§10.1. THE PENALTY METHOD

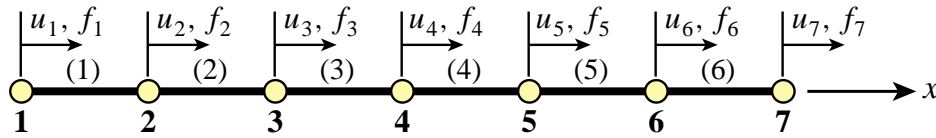


Figure 10.1. The example structure of Chapter 9, repeated for convenience.

§10.1.1. Physical Interpretation

The penalty method will be first presented using a physical interpretation, leaving the mathematical formulation to a subsequent section. Consider again the example structure of Chapter 9, which is reproduced in Figure 10.1 for convenience. To impose $u_2 = u_6$ imagine that nodes 2 and 6 are connected with a “fat” bar of axial stiffness w , labeled with element number 7, as shown in Figure 10.2. This bar is called a *penalty element* and w is its *penalty weight*.

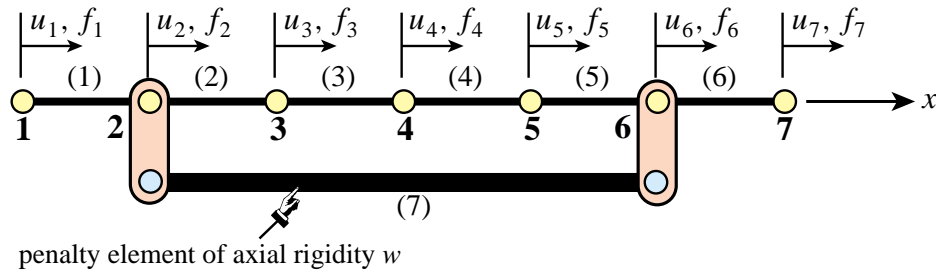


Figure 10.2. Adjunction of a fictitious penalty bar of axial stiffness w , identified as element 7, to enforce the MFC $u_2 = u_6$.

Such an element, albeit fictitious, can be treated exactly like another bar element insofar as continuing the assembly of the master stiffness equations. The penalty element stiffness equations, $\mathbf{K}^{(7)} \mathbf{u}^{(7)} = \mathbf{f}^{(7)}$, are¹

$$w \begin{bmatrix} 1 & -1 \\ -1 & 1 \end{bmatrix} \begin{bmatrix} u_2 \\ u_6 \end{bmatrix} = \begin{bmatrix} f_2^{(7)} \\ f_6^{(7)} \end{bmatrix} \quad (10.1)$$

Because there is one freedom per node, the two local element freedoms map into global freedoms 2 and 6, respectively. Using the assembly rules of Chapter 3 we obtain the following modified master

¹ The general method to get these equations is described in §10.1.4.

stiffness equations: $\hat{\mathbf{K}}\hat{\mathbf{u}} = \hat{\mathbf{f}}$, which shown in detail are

$$\begin{bmatrix} K_{11} & K_{12} & 0 & 0 & 0 & 0 & 0 \\ K_{12} & K_{22} + w & K_{23} & 0 & 0 & -w & 0 \\ 0 & K_{23} & K_{33} & K_{34} & 0 & 0 & 0 \\ 0 & 0 & K_{34} & K_{44} & K_{45} & 0 & 0 \\ 0 & 0 & 0 & K_{45} & K_{55} & K_{56} & 0 \\ 0 & -w & 0 & 0 & K_{56} & K_{66} + w & K_{67} \\ 0 & 0 & 0 & 0 & 0 & K_{67} & K_{77} \end{bmatrix} \begin{bmatrix} u_1 \\ u_2 \\ u_3 \\ u_4 \\ u_5 \\ u_6 \\ u_7 \end{bmatrix} = \begin{bmatrix} f_1 \\ f_2 \\ f_3 \\ f_4 \\ f_5 \\ f_6 \\ f_7 \end{bmatrix}. \quad (10.2)$$

This system can now be submitted to the equation solver. Note that $\hat{\mathbf{u}} \equiv \mathbf{u}$, and only \mathbf{K} has changed.

§10.1.2. Choosing the Penalty Weight

What happens when (10.2) is solved numerically? If a *finite* weight w is chosen the constraint $u_2 = u_6$ is approximately satisfied in the sense that one gets $u_2 - u_6 = e_g$, where $e_g \neq 0$. The “gap error” e_g is called the *constraint violation*. The magnitude $|e_g|$ of this violation depends on the weight: the larger w , the smaller the violation. More precisely, it can be shown that $|e_g|$ becomes proportional to $1/w$ as w gets to be sufficiently large (see Exercises). For example, raising w from, say, 10^6 to 10^7 can be expected to cut the constraint violation by roughly 10 if the physical stiffnesses are small compared to w .

Therefore it seems as if the proper strategy should be: try to make w as large as possible while respecting computer overflow limits. However, this is misleading. As the penalty weight w tends to ∞ the modified stiffness matrix in (10.2) becomes more and more *ill-conditioned with respect to inversion*.

To make this point clear, suppose for definiteness that the rigidities $E^{(e)}A^{(e)}/L^{(e)}$ of the actual bars $e = 1, \dots, 6$ are unity, that $w \gg 1$, and that the computer solving the stiffness equations has a floating-point precision of 16 decimal places. Numerical analysts characterize such precision by saying that $\epsilon_f = O(10^{-16})$, where $|\epsilon_f|$ is the smallest power of 10 that perceptibly adds to 1 in floating-point arithmetic.² The modified stiffness matrix of (10.2) becomes

$$\hat{\mathbf{K}} = \begin{bmatrix} 1 & -1 & 0 & 0 & 0 & 0 & 0 \\ -1 & 2 + w & -1 & 0 & 0 & -w & 0 \\ 0 & -1 & 2 & -1 & 0 & 0 & 0 \\ 0 & 0 & -1 & 2 & -1 & 0 & 0 \\ 0 & 0 & 0 & -1 & 2 & -1 & 0 \\ 0 & -w & 0 & 0 & -1 & 2 + w & -1 \\ 0 & 0 & 0 & 0 & 0 & -1 & 1 \end{bmatrix} \quad (10.3)$$

Clearly as $w \rightarrow \infty$ rows 2 and 6, as well as columns 2 and 6, tend to become linearly dependent; in fact the negative of each other. Linear dependence means singularity; hence $\hat{\mathbf{K}}$ approaches singularity as $w \rightarrow \infty$. In fact, if w exceeds $1/\epsilon_f = 10^{16}$ the computer will not be able to distinguish $\hat{\mathbf{K}}$ from an exactly singular matrix. If $w \ll 10^{16}$ but $w \gg 1$, the effect will be seen in increasing solution errors affecting the computed displacements $\hat{\mathbf{u}}$ returned by the equation solver. These errors, however, tend to be more of a random nature than the constraint violation error.

² Such definitions are more rigorously done by working with binary numbers and base-2 arithmetic but for the present conceptual discussion the use of decimal powers is sufficient.

§10.1.3. The Square Root Rule

Obviously we have two effects at odds with each other. Making w larger reduces the constraint violation error but increases the solution error. The best w is that which makes both errors roughly equal in absolute value. This tradeoff value is difficult to find aside of systematically running numerical experiments. In practice the heuristic *square root rule* is often followed.

This rule can be stated as follows. Suppose that the largest stiffness coefficient, before adding penalty elements, is of the order of 10^k and that the working machine precision is p digits.³ Then choose penalty weights to be of order $10^{k+p/2}$ with the proviso that such a choice would not cause arithmetic overflow.⁴

For the above example in which $k \approx 0$ and $p \approx 16$, the optimal w given by this rule would be $w \approx 10^8$. This w would yield a constraint violation and a solution error of order 10^{-8} . Note that there is no simple way to do better than this accuracy aside from using more floating-point precision. This is not easy to do when using standard low-level programming languages.

The name “square root” arises because the recommended w is in fact $10^k \sqrt{10^p}$. It is seen that picking the weight by this rule requires knowledge of both stiffness magnitudes and floating-point hardware properties of the computer used, as well as the precision selected by the program.

§10.1.4. Penalty Elements for General MFCs

For the constraint $u_2 = u_6$ the physical interpretation of the penalty element is clear. Points 2 and 6 must move in lockstep along x , which can be approximately enforced by the heavy bar device shown in Figure 10.2. But how about $3u_3 + u_5 - 4u_6 = 1$? Or just $u_2 = -u_6$?

The treatment of more general constraints is linked to the theory of *Courant penalty functions*, which in turn is a topic in variational calculus. Because the necessary theory given in §10.1.5 is viewed as an advanced topic, the procedure used for constructing a penalty element is stated here as a recipe. Consider the homogeneous constraint

$$3u_3 + u_5 - 4u_6 = 0. \quad (10.4)$$

Rewrite this equation in matrix form

$$\begin{bmatrix} 3 & 1 & -4 \end{bmatrix} \begin{bmatrix} u_3 \\ u_5 \\ u_6 \end{bmatrix} = 0, \quad (10.5)$$

and premultiply both sides by the transpose of the coefficient matrix:

$$\begin{bmatrix} 3 \\ 1 \\ -4 \end{bmatrix} \begin{bmatrix} 3 & 1 & -4 \end{bmatrix} \begin{bmatrix} u_3 \\ u_5 \\ u_6 \end{bmatrix} = \begin{bmatrix} 9 & 3 & -12 \\ 3 & 1 & -4 \\ -12 & -4 & 16 \end{bmatrix} \begin{bmatrix} u_3 \\ u_5 \\ u_6 \end{bmatrix} = \tilde{\mathbf{K}}^{(e)} \mathbf{u}^{(e)} = 0. \quad (10.6)$$

³ Such order-of-magnitude estimates can be readily found by scanning the diagonal of \mathbf{K} because the largest stiffness coefficient of the actual structure is usually a diagonal entry.

⁴ If overflows occurs, the master stiffness should be scaled throughout or a better choice of physical units made.

Here $\bar{\mathbf{K}}^{(e)}$ is the *unscaled* stiffness matrix of the penalty element. This is now multiplied by the penalty weight w and assembled into the master stiffness matrix following the usual rules. For the example problem, augmenting (10.2) with the scaled penalty element (10.6) yields

$$\begin{bmatrix} K_{11} & K_{12} & 0 & 0 & 0 & 0 & 0 \\ K_{12} & K_{22} & K_{23} & 0 & 0 & 0 & 0 \\ 0 & K_{23} & K_{33} + 9w & K_{34} & 3w & -12w & 0 \\ 0 & 0 & K_{34} & K_{44} & K_{45} & 0 & 0 \\ 0 & 0 & 3w & K_{45} & K_{55} + w & K_{56} - 4w & 0 \\ 0 & 0 & -12w & 0 & K_{56} - 4w & K_{66} + 16w & K_{67} \\ 0 & 0 & 0 & 0 & 0 & K_{67} & K_{77} \end{bmatrix} \begin{bmatrix} u_1 \\ u_2 \\ u_3 \\ u_4 \\ u_5 \\ u_6 \\ u_7 \end{bmatrix} = \begin{bmatrix} f_1 \\ f_2 \\ f_3 \\ f_4 \\ f_5 \\ f_6 \\ f_7 \end{bmatrix}, \quad (10.7)$$

If the constraint is nonhomogeneous the force vector is also modified. To illustrate this effect, consider the MFC: $3u_3 + u_5 - 4u_6 = 1$. Rewrite in matrix form as

$$\begin{bmatrix} 3 & 1 & -4 \end{bmatrix} \begin{bmatrix} u_3 \\ u_5 \\ u_6 \end{bmatrix} = 1. \quad (10.8)$$

Premultiply both sides by the transpose of the coefficient matrix:

$$\begin{bmatrix} 9 & 3 & -12 \\ 3 & 1 & -4 \\ -12 & -4 & 16 \end{bmatrix} \begin{bmatrix} u_3 \\ u_5 \\ u_6 \end{bmatrix} = \begin{bmatrix} 3 \\ 1 \\ -4 \end{bmatrix}. \quad (10.9)$$

Scaling by w and assembling yields

$$\begin{bmatrix} K_{11} & K_{12} & 0 & 0 & 0 & 0 & 0 \\ K_{12} & K_{22} & K_{23} & 0 & 0 & 0 & 0 \\ 0 & K_{23} & K_{33} + 9w & K_{34} & 3w & -12w & 0 \\ 0 & 0 & K_{34} & K_{44} & K_{45} & 0 & 0 \\ 0 & 0 & 3w & K_{45} & K_{55} + w & K_{56} - 4w & 0 \\ 0 & 0 & -12w & 0 & K_{56} - 4w & K_{66} + 16w & K_{67} \\ 0 & 0 & 0 & 0 & 0 & K_{67} & K_{77} \end{bmatrix} \begin{bmatrix} u_1 \\ u_2 \\ u_3 \\ u_4 \\ u_5 \\ u_6 \\ u_7 \end{bmatrix} = \begin{bmatrix} f_1 \\ f_2 \\ f_3 + 3w \\ f_4 \\ f_5 + w \\ f_6 - 4w \\ f_7 \end{bmatrix}, \quad (10.10)$$

§10.1.5. *The Theory Behind the Recipe

The rule comes from the following mathematical theory. Suppose we have a set of m linear MFCs. Using the matrix notation introduced in §9.1.3, these will be stated as

$$\mathbf{a}_p \mathbf{u} = b_p, \quad p = 1, \dots, m \quad (10.11)$$

where \mathbf{u} contains all degrees of freedom and each \mathbf{a}_p is a row vector with same length as \mathbf{u} . To incorporate the MFCs into the FEM model one selects a weight $w_p > 0$ for each constraints and constructs the so-called Courant quadratic penalty function or “penalty energy”

$$P = \sum_{p=1}^m P_p, \quad \text{with} \quad P_p = \mathbf{u}^T \left(\frac{1}{2} \mathbf{a}_p^T \mathbf{a}_p \mathbf{u} - w_p \mathbf{a}_p^T b_p \right) = \frac{1}{2} \mathbf{u}^T \mathbf{K}^{(p)} \mathbf{u} - \mathbf{u}^T \mathbf{f}^{(p)}, \quad (10.12)$$

where we have called $\mathbf{K}^{(p)} = w_p \mathbf{a}_p^T \mathbf{a}_p$ and $\mathbf{f}^{(p)} = w_p \mathbf{a}_p^T \mathbf{b}_i$. P is added to the potential energy function $\Pi = \frac{1}{2} \mathbf{u}^T \mathbf{K} \mathbf{u} - \mathbf{u}^T \mathbf{f}$ to form the *augmented potential energy* $\Pi_a = \Pi + P$. Minimization of Π_a with respect to \mathbf{u} yields

$$(\mathbf{K} \mathbf{u} + \sum_{p=1}^m \mathbf{K}^{(p)}) \mathbf{u} = \mathbf{f} + \sum_{p=1}^m \mathbf{f}^{(p)}. \quad (10.13)$$

Each term of the sum on p , which derives from term P_p in (10.12), may be viewed as contributed by a penalty element with globalized stiffness matrix $\mathbf{K}^{(p)} = w_p \mathbf{a}_p^T \mathbf{a}_p$ and globalized added force term $\mathbf{f}^{(p)} = w_p \mathbf{a}_p^T \mathbf{b}_p$.

To use a even more compact form we may write the set of multifreedom constraints as $\mathbf{A} \mathbf{u} = \mathbf{b}$. Then the penalty augmented system can be written compactly as

$$(\mathbf{K} + \mathbf{A}^T \mathbf{W} \mathbf{A}) \mathbf{u} = \mathbf{f} + \mathbf{W} \mathbf{A}^T \mathbf{b}, \quad (10.14)$$

where \mathbf{W} is a diagonal matrix of penalty weights. This compact form, however, conceals the structure of the penalty elements.

§10.1.6. Assessment of the Penalty Method

The main advantage of the penalty function method is its straightforward computer implementation. Looking at modified systems such as (10.2), (10.7) or (10.10) it is obvious that master equations need not be rearranged. That is, \mathbf{u} and $\hat{\mathbf{u}}$ are the same. Constraints may be programmed as “penalty elements,” and stiffness and force contributions of these elements merged through the standard assembler. In fact using this method there is no need to distinguish between unconstrained and constrained equations! Once all elements — regular and penalty — are assembled, the system can be passed to the equation solver.⁵

An important advantage with respect to the master-slave (elimination) method is its lack of sensitivity with respect to whether constraints are linearly dependent. To give a simplistic example, suppose that the constraint $u_2 = u_6$ appears twice. Then two penalty bar elements connecting 2 and 6 will be inserted, doubling the intended weight but otherwise not causing undue harm.

An advantage with respect to the Lagrange multiplier method described in §10.2 is that positive definiteness is not lost. Such loss can affect the performance of certain numerical processes.⁶ Finally, it is worth noting that the the penalty method is easily extendible to nonlinear constraints although such extension falls outside the scope of this book.

The main disadvantage, however, is a serious one: the choice of weight values that balance solution accuracy with the violation of constraint conditions. For simple cases the square root rule previously described often works, although its effective use calls for knowledge of the magnitude of stiffness coefficients. Such knowledge may be difficult to extract from a general purpose “black box” program. For difficult cases selection of appropriate weights may require extensive numerical experimentation, wasting the user time with numerical games that have no bearing on the actual objective, which is getting a solution.

The deterioration of the condition number of the penalty-augmented stiffness matrix can have serious side effects in some solution procedures such as eigenvalue extraction or iterative solvers.

⁵ Single freedom constraints, such as those encountered in Chapters 3–4, are usually processed separately for efficiency.

⁶ For example, solving the master stiffness equations by Cholesky factorization or conjugate-gradients.

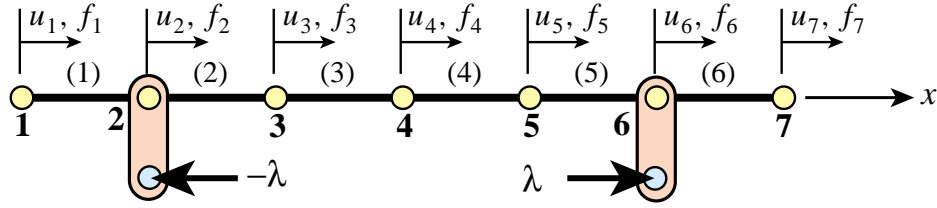


Figure 10.3. Physical interpretation of Lagrange multiplier adjunction to enforce the MFC $u_2 = u_6$.

Finally, even if optimal weights are selected, the combined solution error cannot be lowered beyond a threshold value.

From this assessment it is evident that penalty augmentation, although superior to the master-slave method from the standpoint of generality and ease of implementation, is no panacea.

§10.2. LAGRANGE MULTIPLIER ADJUNCTION

§10.2.1. Physical Interpretation

As in the case of the penalty function method, the method of Lagrange multipliers can be given a rigorous justification within the framework of variational calculus. But in the same spirit it will be introduced for the example structure from a physical standpoint that is particularly illuminating.

Consider again the constraint $u_2 = u_6$. Borrowing some ideas from the penalty method, imagine that nodes 2 and 6 are connected now by a *rigid* link rather than a flexible one. Thus the constraint is imposed exactly. But of course the penalty method with an infinite weight would “blow up.”

We may remove the link if it is replaced by an appropriate reaction force pair $(-\lambda, +\lambda)$, as illustrated in Figure 10.3. These are called the *constraint forces*. Incorporating these forces into the original stiffness equations (9.10) we get

$$\begin{bmatrix} K_{11} & K_{12} & 0 & 0 & 0 & 0 & 0 \\ K_{12} & K_{22} & K_{23} & 0 & 0 & 0 & 0 \\ 0 & K_{23} & K_{33} & K_{34} & 0 & 0 & 0 \\ 0 & 0 & K_{34} & K_{44} & K_{45} & 0 & 0 \\ 0 & 0 & 0 & K_{45} & K_{55} & K_{56} & 0 \\ 0 & 0 & 0 & 0 & K_{56} & K_{66} & K_{67} \\ 0 & 0 & 0 & 0 & 0 & K_{67} & K_{77} \end{bmatrix} \begin{bmatrix} u_1 \\ u_2 \\ u_3 \\ u_4 \\ u_5 \\ u_6 \\ u_7 \end{bmatrix} = \begin{bmatrix} f_1 \\ f_2 - \lambda \\ f_3 \\ f_4 \\ f_5 \\ f_6 + \lambda \\ f_7 \end{bmatrix}. \quad (10.15)$$

This λ is called a *Lagrange multiplier*. Because λ is an unknown, let us transfer it to the *left hand side* by *appending* it to the vector of unknowns:

$$\begin{bmatrix} K_{11} & K_{12} & 0 & 0 & 0 & 0 & 0 & 0 \\ K_{12} & K_{22} & K_{23} & 0 & 0 & 0 & 0 & 1 \\ 0 & K_{23} & K_{33} & K_{34} & 0 & 0 & 0 & 0 \\ 0 & 0 & K_{34} & K_{44} & K_{45} & 0 & 0 & 0 \\ 0 & 0 & 0 & K_{45} & K_{55} & K_{56} & 0 & 0 \\ 0 & 0 & 0 & 0 & K_{56} & K_{66} & K_{67} & -1 \\ 0 & 0 & 0 & 0 & 0 & K_{67} & K_{77} & 0 \end{bmatrix} \begin{bmatrix} u_1 \\ u_2 \\ u_3 \\ u_4 \\ u_5 \\ u_6 \\ u_7 \\ \lambda \end{bmatrix} = \begin{bmatrix} f_1 \\ f_2 \\ f_3 \\ f_4 \\ f_5 \\ f_6 \\ f_7 \end{bmatrix}. \quad (10.16)$$

But now we have 7 equations in 8 unknowns. To render the system determinate, the constraint condition $u_2 - u_6 = 0$ is appended as eighth equation:

$$\begin{bmatrix} K_{11} & K_{12} & 0 & 0 & 0 & 0 & 0 & 0 \\ K_{12} & K_{22} & K_{23} & 0 & 0 & 0 & 0 & 1 \\ 0 & K_{23} & K_{33} & K_{34} & 0 & 0 & 0 & 0 \\ 0 & 0 & K_{34} & K_{44} & K_{45} & 0 & 0 & 0 \\ 0 & 0 & 0 & K_{45} & K_{55} & K_{56} & 0 & 0 \\ 0 & 0 & 0 & 0 & K_{56} & K_{66} & K_{67} & -1 \\ 0 & 0 & 0 & 0 & 0 & K_{67} & K_{77} & 0 \\ 0 & 1 & 0 & 0 & 0 & -1 & 0 & 0 \end{bmatrix} \begin{bmatrix} u_1 \\ u_2 \\ u_3 \\ u_4 \\ u_5 \\ u_6 \\ u_7 \\ \lambda \end{bmatrix} = \begin{bmatrix} f_1 \\ f_2 \\ f_3 \\ f_4 \\ f_5 \\ f_6 \\ f_7 \\ 0 \end{bmatrix}, \quad (10.17)$$

This is called the *multiplier-augmented* system. Its coefficient matrix, which is symmetric, is called the *bordered stiffness matrix*. The process by which λ is appended to the vector of original unknowns is called *adjunction*. Solving this system provides the desired solution for the degrees of freedom while also characterizing the constraint forces through λ .

§10.2.2. Lagrange Multipliers for General MFCs

The general procedure will be stated first as a recipe. Suppose that we want to solve the example structure subjected to three MFCs

$$u_2 - u_6 = 0, \quad 5u_2 - 8u_7 = 3, \quad 3u_3 + u_5 - 4u_6 = 1, \quad (10.18)$$

Adjoin these MFCs as the eighth, ninth and tenth equations:

$$\begin{bmatrix} K_{11} & K_{12} & 0 & 0 & 0 & 0 & 0 & 0 \\ K_{12} & K_{22} & K_{23} & 0 & 0 & 0 & 0 & 0 \\ 0 & K_{23} & K_{33} & K_{34} & 0 & 0 & 0 & 0 \\ 0 & 0 & K_{34} & K_{44} & K_{45} & 0 & 0 & 0 \\ 0 & 0 & 0 & K_{45} & K_{55} & K_{56} & 0 & 0 \\ 0 & 0 & 0 & 0 & K_{56} & K_{66} & K_{67} & 0 \\ 0 & 0 & 0 & 0 & 0 & K_{67} & K_{77} & 0 \\ 0 & 1 & 0 & 0 & 0 & -1 & 0 & 0 \\ 0 & 5 & 0 & 0 & 0 & 0 & -8 & 0 \\ 0 & 0 & 3 & 0 & 1 & -4 & 0 & 0 \end{bmatrix} \begin{bmatrix} u_1 \\ u_2 \\ u_3 \\ u_4 \\ u_5 \\ u_6 \\ u_7 \\ \lambda \end{bmatrix} = \begin{bmatrix} f_1 \\ f_2 \\ f_3 \\ f_4 \\ f_5 \\ f_6 \\ f_7 \\ 0 \\ 3 \\ 1 \end{bmatrix}, \quad (10.19)$$

Three Lagrange multipliers: λ_1 , λ_2 and λ_3 , are required to take care of three MFCs. Adjoin those unknowns to the nodal displacement vector. Symmetrize the coefficient matrix by adjoining 3 columns that are the transpose of the 3 last rows, and filling the bottom right-hand corner with

zeros:

$$\begin{bmatrix} K_{11} & K_{12} & 0 & 0 & 0 & 0 & 0 & 0 & 0 & 0 \\ K_{12} & K_{22} & K_{23} & 0 & 0 & 0 & 0 & 1 & 5 & 0 \\ 0 & K_{23} & K_{33} & K_{34} & 0 & 0 & 0 & 0 & 0 & 3 \\ 0 & 0 & K_{34} & K_{44} & K_{45} & 0 & 0 & 0 & 0 & 0 \\ 0 & 0 & 0 & K_{45} & K_{55} & K_{56} & 0 & 0 & 0 & 1 \\ 0 & 0 & 0 & 0 & K_{56} & K_{66} & K_{67} & -1 & 0 & -4 \\ 0 & 0 & 0 & 0 & 0 & K_{67} & K_{77} & 0 & -8 & 0 \\ 0 & 1 & 0 & 0 & 0 & -1 & 0 & 0 & 0 & 0 \\ 0 & 5 & 0 & 0 & 0 & 0 & -8 & 0 & 0 & 0 \\ 0 & 0 & 3 & 0 & 1 & -4 & 0 & 0 & 0 & 0 \end{bmatrix} \begin{bmatrix} u_1 \\ u_2 \\ u_3 \\ u_4 \\ u_5 \\ u_6 \\ u_7 \\ \lambda_1 \\ \lambda_2 \\ \lambda_3 \end{bmatrix} = \begin{bmatrix} f_1 \\ f_2 \\ f_3 \\ f_4 \\ f_5 \\ f_6 \\ f_7 \\ 0 \\ 3 \\ 1 \end{bmatrix}, \quad (10.20)$$

§10.2.3. *The Theory Behind the Recipe

The recipe illustrated by (10.20) comes from a well known technique of variational calculus. Using the matrix notation introduced in §9.1.3, compactly denote the set of m MFCs by $\mathbf{A}\mathbf{u} = \mathbf{b}$, where \mathbf{A} is $m \times n$. The potential energy of the unconstrained finite element model is $\Pi = \frac{1}{2}\mathbf{u}^T \mathbf{K}\mathbf{u} - \mathbf{u}^T \mathbf{f}$. To impose the constraint, adjoin m Lagrange multipliers collected in vector $\boldsymbol{\lambda}$ and form the Lagrangian

$$L(\mathbf{u}, \boldsymbol{\lambda}) = \Pi + \boldsymbol{\lambda}^T (\mathbf{A}\mathbf{u} - \mathbf{b}) = \frac{1}{2}\mathbf{u}^T \mathbf{K}\mathbf{u} - \mathbf{u}^T \mathbf{f} + \boldsymbol{\lambda}^T (\mathbf{A}\mathbf{u} - \mathbf{b}). \quad (10.21)$$

Extremization of L with respect to \mathbf{u} and $\boldsymbol{\lambda}$ yields the multiplier-augmented form

$$\begin{bmatrix} \mathbf{K} & \mathbf{A}^T \\ \mathbf{A} & \mathbf{0} \end{bmatrix} \begin{bmatrix} \mathbf{u} \\ \boldsymbol{\lambda} \end{bmatrix} = \begin{bmatrix} \mathbf{f} \\ \mathbf{b} \end{bmatrix} \quad (10.22)$$

The master stiffness matrix \mathbf{K} in (10.22) is said to be *bordered* with \mathbf{A} and \mathbf{A}^T . Solving this system provides \mathbf{u} and $\boldsymbol{\lambda}$. The latter can be interpreted as forces of constraint in the following sense: a removed constraint can be replaced by a system of forces characterized by $\boldsymbol{\lambda}$ multiplied by the constraint coefficients. More precisely, the constraint forces are $-\mathbf{A}^T \boldsymbol{\lambda}$.

§10.2.4. Assessment of the Lagrange Multiplier Method

In contrast to the penalty method, the method of Lagrange multipliers has the advantage of being exact (aside from computation errors). It provides directly the constraint forces, which are of interest in many applications. It does not require any guesses as regards weights. As the penalty method, it can be extended without difficulty to nonlinear constraints.

It is not free of disadvantages. It introduces additional unknowns, requiring expansion of the original stiffness method. It renders the augmented stiffness matrix indefinite, an effect that may cause grief with some linear equation solving methods that rely on positive definiteness. Finally, as the master-slave method, it is sensitive to the degree of linear independence of the constraints: if the constraint $u_2 = u_6$ is specified twice, the bordered stiffness is obviously singular.

On the whole the Lagrangian multiplier method appear to be the most elegant for a general-purpose finite element program that is supposed to work as a “black box” by minimizing guesses and choices from its users. Its implementation, however, is not simple. Special care must be exercised to detect singularities due to constraint dependency and to account for the effect of loss of positive definiteness of the bordered stiffness on equation solvers.

§10.3. *THE AUGMENTED LAGRANGIAN METHOD

The general matrix forms of the penalty function and Lagrangian multiplier methods are given by expressions (10.13) and (10.22), respectively. A useful connection between these methods can be established as follows.

Because the lower diagonal block of the bordered stiffness matrix in (10.22) is null, it is not possible to directly eliminate λ . To make this possible, replace this block by $\epsilon \mathbf{S}^{-1}$, where \mathbf{S} is a constraint-scaling diagonal matrix of appropriate order and ϵ is a small number. The reciprocal of ϵ is a large number called $w = 1/\epsilon$. To maintain exactness of the second equation, $\epsilon \mathbf{S}^{-1} \lambda$ is added to the right-hand side:

$$\begin{bmatrix} \mathbf{K} & \mathbf{A}^T \\ \mathbf{A} & \epsilon \mathbf{S}^{-1} \end{bmatrix} \begin{bmatrix} \mathbf{u} \\ \lambda \end{bmatrix} = \begin{bmatrix} \mathbf{f} \\ \epsilon \mathbf{S}^{-1} \lambda^P \end{bmatrix} \quad (10.23)$$

Here superscript P (for “predicted value”) is attached to the λ on the right-hand side as a “tracer.” We can now formally solve for λ and subsequently for \mathbf{u} . The results may be presented as

$$\begin{aligned} (\mathbf{K} + w \mathbf{A}^T \mathbf{S} \mathbf{A}) \mathbf{u} &= \mathbf{f} + w \mathbf{A}^T \mathbf{S} \mathbf{b} - \mathbf{A}^T \lambda^P, \\ \lambda &= \lambda^P + w \mathbf{S}(\mathbf{b} - \mathbf{A} \mathbf{u}), \end{aligned} \quad (10.24)$$

Setting $\lambda^P = \mathbf{0}$ in the first matrix equation yields

$$(\mathbf{K} + w \mathbf{A}^T \mathbf{S} \mathbf{A}) \mathbf{u} = \mathbf{f} + w \mathbf{A}^T \mathbf{S} \mathbf{b}. \quad (10.25)$$

On taking $\mathbf{W} = w \mathbf{S}$, the general matrix equation (10.13) of the penalty method is recovered.

This relation suggests the construction of *iterative procedures* in which one tries to *improve the accuracy of the penalty function method while w is kept constant* [10.2]. This strategy circumvents the aforementioned ill-conditioning problems when the weight w is gradually increased. One such method is easily constructed by inspecting (10.24). Using superscript k as an iteration index and keeping w fixed, solve equations (10.24) in tandem as follows:

$$\begin{aligned} (\mathbf{K} + \mathbf{A}^T \mathbf{W} \mathbf{A}) \mathbf{u}^k &= \mathbf{f} + \mathbf{A}^T \mathbf{W} \mathbf{b} - \mathbf{A}^T \lambda^k, \\ \lambda^{k+1} &= \lambda^k + \mathbf{W}(\mathbf{b} - \mathbf{A} \mathbf{u}^k), \end{aligned} \quad (10.26)$$

for $k = 0, 1, \dots$, beginning with $\lambda^0 = \mathbf{0}$. Then \mathbf{u}^0 is the penalty solution. If the process converges one recovers the exact Lagrangian solution without having to solve the Lagrangian system (10.23) directly.

The family of iterative procedures that may be precipitated from (10.24) collectively pertains to the class of *augmented Lagrangian methods*.

§10.4. SUMMARY

The treatment of linear MFCs in finite element systems can be carried out by several methods. Three of these: master-slave elimination, penalty augmentation and Lagrange multiplier adjunction, have been discussed. It is emphasized that no method is uniformly satisfactory in terms of generality, robustness, numerical behavior and simplicity of implementation.

Figure 10.4 provides a tabular assessment of the three techniques in terms of seven attributes.

	Master-Slave Elimination	Penalty Function	Lagrange Multipliers
Generality	fair	excellent	excellent
Ease of implementation	poor to fair	good	fair
Sensitivity to user decisions	high	high	small to none
Accuracy	variable	mediocre	excellent
Sensitivity as regards constraint dependence	high	none	high
Retains positive definiteness	yes	yes	no
Modifies unknown vector	yes	no	yes

Figure 10.4. Assessment summary of three MFC application methods.

For a general purpose program that tries to attain “black box” behavior (that is, minimal decisions on the part of users) the method of Lagrange multipliers has the edge. This edge is unfortunately blunted by a fairly complex computer implementation and by the loss of positive definiteness in the bordered stiffness matrix.

Notes and Bibliography

A form of the penalty function method was first proposed by Courant [10.1]. It entered the FEM through the work of numerous people in the 1960s.

The Lagrange Multiplier method is much older. Multipliers (called initially “coefficients”) were introduced by Lagrange in his famous *Mécanique Analytique* monograph [10.4], as part of the procedure for forming the function now called the Lagrangian. Its use in FEM is more recent than penalty methods.

Augmented Lagrangian methods have received much attention since the late 1960s, when they originated in the field of constrained optimization [10.3,10.5]. The use of the Augmented Lagrangian Multiplier method for FEM application is discussed in [10.2], wherein the iterative algorithm (10.26) for the master stiffness equations is derived.

References

- [10.1] Courant, R., Variational methods for the study of problems of equilibrium and vibrations, *Bull. Amer. Math. Soc.*, Vol. 49, 1–23, 1943.
- [10.2] Felippa, C. A., Iterative procedures for improving penalty function solutions of algebraic systems, *Int. J. Numer. Meth. Engrg.*, **12**, 821–836, 1978.
- [10.3] M. R. Hestenes, Multiplier and gradient methods, *J. Optim. Theory Appl.*, **4**, 303–320, 1969
- [10.4] Lagrange, J. L., *Mécanique Analytique*, Chez la veuve Desaint, Paris, 1788; Édition complète, 2 vols., Blanchard, Paris, 1965.
- [10.5] Powell, M. J. D., A method for nonlinear constraints in optimization problems, in *Optimization*, ed. by R. Fletcher, Academic Press, London, 283–298, 1969.

Homework Exercises for Chapter 10

MultiFreedom Constraints II

EXERCISE 10.1

[C+N:20] This is identical to Exercise 9.1, except that the MFC $u_2 - u_6 = 1/5$ is to be treated by the penalty function method. Take the weight w to be 10^k , in which k varies as $k = 3, 4, 5, \dots, 16$. For each sample w compute the Euclidean-norm solution error $e(w) = \|\mathbf{u}^p(w) - \mathbf{u}^{ex}\|_2$, where \mathbf{u}^p is the computed solution and \mathbf{u}^{ex} is the exact solution listed in (E9.1). Plot $k = \log_{10} w$ versus $\log_{10} e$ and report for which weight e attains a minimum. (See Slide #5 for a check). Does it roughly agree with the square root rule (§10.1.3) if the computations carry 16 digits of precision?

As in Exercise 9.1, use *Mathematica*, *Matlab* (or similar) to do the algebra. For example, the following *Mathematica* script solves this Exercise:

```
(* Exercise 10.1 - Penalty Method *)
(* MFC: u2-u6=1/5 variable w *)
K=MasterStiffnessOfSixElementBar[100];
Print["Stiffness K=",K/MatrixForm];
f={1,2,3,4,5,6,7}; Print["Applied forces=",f];
uexact= {0,0.27,0.275,0.25,0.185,0.07,0.14}; ew={};
For [w=100, w<=10^16, w=10*w; (* increase w by 10 every pass *)
    Khat=K; fhat=f;
    Khat[[2,2]]+=w; Khat[[6,6]]+=w; Khat[[6,2]]=Khat[[2,6]]-=w;
    fhat[[2]]+=(1/5)*w; fhat[[6]]-=(1/5)*w; (*insert penalty *)
    {Kmod,fmod}=FixLeftEndOfSixElementBar[Khat,fhat];
    u=LinearSolve[N[Kmod],N[fmod]];
    Print["Weight w=",N[w]//ScientificForm," u=",u//InputForm];
    e=Sqrt[(u-uexact).(u-uexact)];
    (*Print["L2 solution error=",e//ScientificForm]; *)
    AppendTo[ew,{Log[10,w],Log[10,e]}];
];
ListPlot[ew,AxesOrigin->{5,-8},Frame->True, PlotStyle->
  {AbsolutePointSize[4],AbsoluteThickness[2],RGBColor[1,0,0]},
  PlotJoined->True,AxesLabel->{"Log10(w)","Log10(u error)"}];
```

Here `MasterStiffnessOfSixElementBar` and `FixLeftEndOfSixElementBar` are the same modules listed in Exercise 9.1.

Note: If you run the above program, you may get several beeps from *Mathematica* as it is processing some of the systems with very large weights. Don't be alarmed: those are only warnings. The `LinearSolve` function is alerting you that the coefficient matrices $\hat{\mathbf{K}}$ for weights of order 10^{12} or bigger are ill-conditioned.

EXERCISE 10.2

[C+N:15] Again identical to Exercise 9.1, except that the MFC $u_2 - u_6 = 1/5$ is to be treated by the Lagrange multiplier method. The results for the computed \mathbf{u} and the recovered force vector $\mathbf{K}\mathbf{u}$ should agree with (E9.1). Use *Mathematica*, *Matlab* (or similar) to do the algebra. For example, the following *Mathematica* script solves this Exercise:

```
(* Exercise 10.2 - Lagrange Multiplier Method *)
(* MFC: u2-u6=1/5 *)
K=MasterStiffnessOfSixElementBar[100];
Khat=Table[0,{8},{8}]; f={1,2,3,4,5,6,7}; fhat=AppendTo[f,0];
```

```

For [i=1,i<=7,i++, For[j=1,j<=7,j++, Khat[[i,j]]=K[[i,j]] ]];
{Kmod,fmod}=FixLeftEndOfSixElementBar[Khat,fhat];
Kmod[[2,8]]=Kmod[[8,2]]= 1;
Kmod[[6,8]]=Kmod[[8,6]]=-1; fmod[[8]]=1/5;
Print["Kmod=",Kmod//MatrixForm];
Print["fmod=",fmod];
umod=LinearSolve[N[Kmod],N[fmod]]; u=Take[umod,7];
Print["Solution u=",u ,",   lambda=",umod[[8]]];
Print["Recovered node forces=",K.u];

```

Here `MasterStiffnessOfSixElementBar` and `FixLeftEndOfSixElementBar` are the same modules listed in Exercise 9.1.

Does the computed solution agree with (E9.1)?

EXERCISE 10.3

[A:10] For the example structure, show which penalty elements would implement the following MFCs:

$$\begin{aligned}
 \text{(a)} \quad u_2 + u_6 &= 0, \\
 \text{(b)} \quad u_2 - 3u_6 &= 1/3.
 \end{aligned}
 \tag{E10.1}$$

As answer, show the stiffness equations of those two elements in a manner similar to (10.1).

EXERCISE 10.4

[A/C+N:15+15+10] Suppose that the assembled stiffness equations for a one-dimensional finite element model before imposing constraints are

$$\begin{bmatrix} 2 & -1 & 0 \\ -1 & 2 & -1 \\ 0 & -1 & 2 \end{bmatrix} \begin{bmatrix} u_1 \\ u_2 \\ u_3 \end{bmatrix} = \begin{bmatrix} 1 \\ 0 \\ 2 \end{bmatrix}.
 \tag{E10.2}$$

This system is to be solved subject to the multipoint constraint

$$u_1 = u_3. \tag{E10.3}$$

- Impose the constraint (E10.3) by the master-slave method taking u_1 as master, and solve the resulting 2×2 system of equations by hand.
- Impose the constraint (E10.3) by the penalty function method, leaving the weight w as a free parameter. Solve the equations by hand or CAS (Cramer's rule is recommended) and verify analytically that as $w \rightarrow \infty$ the solution approaches that found in (a). Tabulate the values of u_1, u_2, u_3 for $w = 0, 1, 10, 100$. *Hint 1:* the value of u_2 should not change. *Hint 2:* the solution for u_1 should be $(6w + 5)/(4w + 4)$.
- Impose the constraint (E10.3) by the Lagrange multiplier method. Show the 4×4 multiplier-augmented system of equations analogous to (10.13) and solve it by computer or calculator.

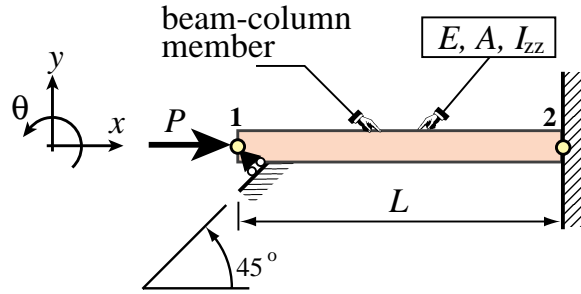


Figure E10.1. Beam-column member for Exercise 10.5.

EXERCISE 10.5

[A/C:10+15+10] The beam-column member shown in Figure E10.1 rests on a skew roller that forms a 45° angle with the horizontal axis x , and is loaded axially by a force P .

The finite element equations upon removing the fixed right end, but before imposing the skew-roller MFC, are

$$\begin{bmatrix} EA/L & 0 & 0 \\ 0 & 12EI_{zz}/L^3 & 6EI_{zz}/L^2 \\ 0 & 6EI_{zz}/L^2 & 4EI_{zz}/L \end{bmatrix} \begin{bmatrix} u_{x1} \\ u_{y1} \\ \theta_1 \end{bmatrix} = \begin{bmatrix} P \\ 0 \\ 0 \end{bmatrix}, \quad (\text{E10.4})$$

where E , A , and I_{zz} are member properties, θ_1 is the left end rotation, and L is the member length.⁷ To simplify the calculations set $P = \alpha EA$, and $I_{zz} = \beta AL^2$, in which α and β are dimensionless parameters, and express the following solutions in terms of α and β .

- Apply the skew roller constraint by the master-slave method (make u_{y1} slave) and solve for u_{x1} and θ_1 in terms of L , α and β . This may be done by hand or a CAS. Partial solution: $u_{x1} = \alpha L/(1 + 3\beta)$.
- Apply the skew roller constraint by the penalty method by adjoining a penalty truss member of axial stiffness $k = wEA$ normal to the roller, and compute u_{x1} (Cramer's rule is recommended if solved by hand). Verify that as $w \rightarrow \infty$ the answer obtained in (a) is recovered. Partial solution: $u_{x1} = \alpha L(3\beta + wL)/(3\beta + wL(1 + 3\beta))$.
- Apply the skew roller constraint by Lagrangian multiplier adjunction, and solve the resulting 4×4 system of equations using a CAS. Verify that you get the same solution as in (a).

EXERCISE 10.6

[A:30] Show that the master-slave transformation method $\mathbf{u} = \mathbf{T}\hat{\mathbf{u}}$ can be written down as a special form of the method of Lagrange multipliers. Start from the augmented functional

$$\Pi_{MS} = \frac{1}{2} \mathbf{u}^T \mathbf{K} \mathbf{u} - \mathbf{u}^T \mathbf{f} + \boldsymbol{\lambda}^T (\mathbf{u} - \mathbf{T}\hat{\mathbf{u}}) \quad (\text{E10.5})$$

and write down the stationarity conditions of Π_{MS} with respect to \mathbf{u} , $\boldsymbol{\lambda}$ and $\hat{\mathbf{u}}$ in matrix form.

EXERCISE 10.7

[A:35] Check the matrix equations (10.23) through (10.26) quoted for the Augmented Lagrangian method.

EXERCISE 10.8

[A:40] (Advanced, close to a research problem) Show that the master-slave transformation method $\mathbf{u} = \mathbf{T}\hat{\mathbf{u}}$ can be expressed as a limit of the penalty function method as the weights go to infinity. Start from the augmented

⁷ The stiffness equations for a beam column are derived in Part III of this book. For now consider (E10.4) as a recipe.

functional

$$\Pi_p = \frac{1}{2} \mathbf{u}^T \mathbf{K} \mathbf{u} - \mathbf{u}^T \mathbf{f} + \frac{1}{2} w (\mathbf{u} - \mathbf{T} \hat{\mathbf{u}})^T (\mathbf{u} - \mathbf{T} \hat{\mathbf{u}}) \quad (\text{E10.6})$$

Write down the matrix stationarity conditions with respect to \mathbf{u} and $\hat{\mathbf{u}}$ and eliminate \mathbf{u} . *Hint:* using Woodbury's formula (Appendix C, §C.5.2)

$$(\mathbf{K} + w \mathbf{T}^T \mathbf{S} \mathbf{T})^{-1} = \mathbf{K}^{-1} - \mathbf{K}^{-1} \mathbf{T}^T (\bar{\mathbf{K}} + w^{-1} \mathbf{S}^{-1})^{-1} \mathbf{T} \mathbf{K}^{-1}. \quad (\text{E10.7})$$

show that

$$\bar{\mathbf{K}} = \mathbf{T} \mathbf{K}^{-1} \mathbf{T}^T \quad (\text{E10.8})$$

11

Superelements and Global-Local Analysis

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§11.1. SUPERELEMENT CONCEPT

Superelements are groupings of finite elements that, upon assembly, may be considered as an *individual element* for computational purposes. These purposes may be driven by modeling or solution needs.

A random assortment of elements does not necessarily make up a superelement. To be considered as such, a grouping must meet certain conditions. Informally we can say that it must form a structural component on its own. This imposes certain conditions stated mathematically in §11.1.3. Inasmuch as these conditions involve advanced concepts such as rank sufficiency, which are introduced in later Chapters, the restrictions are not dwelled upon here.

As noted in Chapter 7, superelements may originate from two overlapping contexts: “bottom up” or “top down.” In a bottom up context one thinks of superelements as built from simpler elements. In a top-down context, superelements may be thought as being large pieces of a complete structure. This dual viewpoint motivates the following classification:

Macroelements. These are superelements assembled with a few primitive elements. Also called *mesh units* when they are presented to program users as individual elements.

Substructures. Complex assemblies of elements that result on breaking up a structure into distinguishable portions.

When does a substructure becomes a macroelement or vice-versa? There are no precise rules. In fact the generic term *superelement* was coined to cover the entire spectrum, ranging from *individual elements* to *complete structures*. This universality is helped by the common features noted below.

Both macroelements and substructures are treated exactly the same way as regards matrix processing. The basic rule is that associated with *condensation* of internal degrees of freedom. The process is illustrated in the following section with a simple example. The reader should note, however, that the technique applies to *any* superelement, whether composed of two or a million elements.¹

§11.1.1. Where Does the Idea Comes From?

Substructuring was invented by aerospace engineers in the early 1960s² to carry out a first-level breakdown of complex systems such as a complete airplane, as depicted in Figure 11.1. The decomposition may continue hierarchically through additional levels as illustrated in Figure 11.2. The concept is also natural for space vehicles operating in stages, such as the Apollo short stack depicted in Figure 11.3.

Three original motivating factors for substructuring can be cited.

1. **Facilitate division of labor.** Substructures with different functions are done by separate design groups with specialized knowledge and experience. Groups are thus protected from “hurry up and wait” constraints. More specifically: a wing design group can keep on working on refinements and improvements as long as the interface information (the wing-fuselage intersection) stays sensibly unchanged.

¹ Of course the computer implementation is totally different as one goes from macroelements to substructures, because efficient processing for large matrix systems requires exploitation of sparsity.

² See Notes and Bibliography at the end of this Chapter.

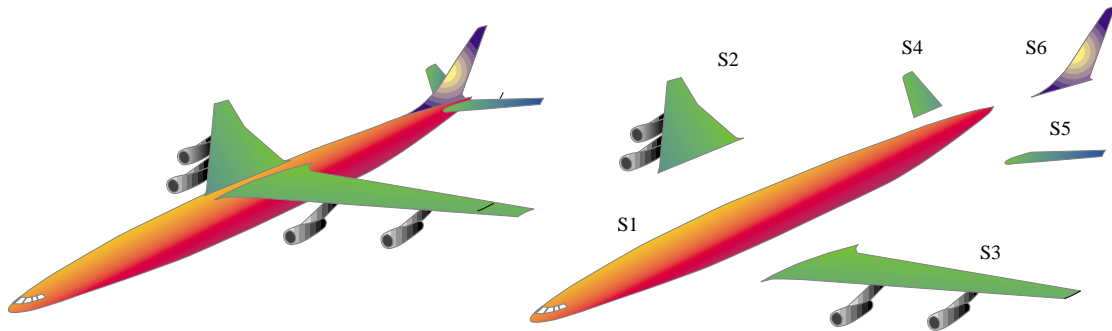


Figure 11.1. Complete airplane broken down into six level one substructures identified as S_1 through S_6 .

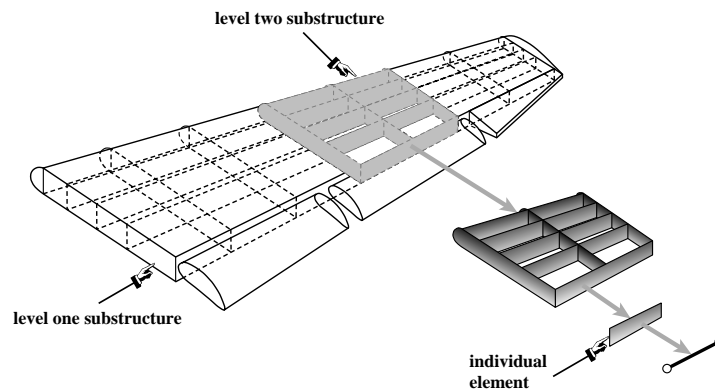


Figure 11.2. Further breakdown of wing structure. The decomposition process may continue down to the individual element level.

2. **Take advantage of repetition.** Often structures are built of several identical or nearly identical units. For instance, the wing substructures S_2 and S_3 of Figure 11.1 are mirror images on reflection about the fuselage midplane, and so are the stabilizers S_4 and S_5 . Even if the loading is not symmetric about that midplane, recognizing repetitions saves model preparation time.
3. **Overcome computer limitations.** The computers of the 1960s operated under serious memory limitations. (For example, the first supercomputer: the Control Data 6600, had a total memory of 131072 60-bit words or 1.31 MB; that machine cost \$10M in 1966 dollars.) It was difficult to fit a complex structure such as an airplane as one entity. Substructuring permitted the complete analysis to be carried out in stages through use of auxiliary storage such as tapes and disk.

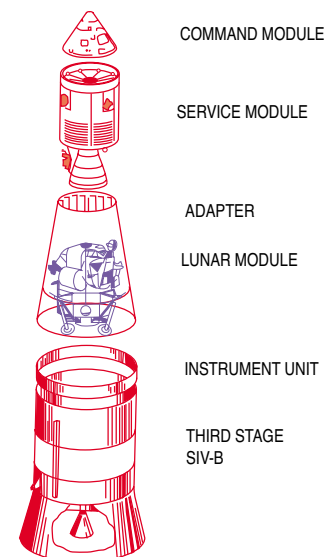


Figure 11.3. The Apollo short stack.

Of the three motivations, the first two still hold today. The third one has moved to a different plane: parallel processing, as noted in §11.1.2 below.

In the late 1960s the idea was picked up and developed extensively by the offshore and shipbuilding industries, the products of which tend to be modular and repetitive to reduce fabrication costs. As noted above, repetition favors the use of substructuring techniques.

At the other end of the superelement spectrum, the mesh units herein called macroelements appeared in the mid 1960s. They were motivated by user convenience. For example, in hand preparation of models, quadrilateral and bricks involve less human labor than triangles and tetrahedra, respectively. It was therefore natural to combine the latter to assemble the former. Going a step further one can assemble components such as “box elements” for applications such as box-girder bridges.

§11.1.2. Subdomains

Applied mathematicians working on solution procedures for parallel computation have developed the concept of *subdomains*. These are groupings of finite elements that are entirely motivated by computational considerations. They are subdivisions of the finite element model done more or less automatically by a program called *domain decomposer*.

Although the concepts of substructures and subdomains overlap in many respects, it is better to keep the two separate. The common underlying theme is divide and conquer but the motivation is different.

§11.1.3. *Mathematical Requirements

A superelement is said to be *rank-sufficient* if its only zero-energy modes are rigid-body modes. Equivalently, the superelement does not possess spurious kinematic mechanisms.

Verification of the rank-sufficient condition guarantees that the static condensation procedure described below will work properly.

§11.2. STATIC CONDENSATION

Degrees of freedom of a superelement are classified into two groups:

Internal Freedoms. Those that are not connected to the freedoms of another superelement. Nodes whose freedoms are internal are called *internal nodes*.

Boundary Freedoms. These are connected to at least another superelement. They usually reside at *boundary nodes* placed on the periphery of the superelement. See Figure 11.4.

The objective is to get rid of all displacement degrees of freedom associated with *internal freedoms*. This elimination process is called *static condensation*, or simply *condensation*.

Condensation may be presented in terms of explicit matrix operations, as shown in the next subsection. A more practical technique based on symmetric Gauss elimination is discussed later.

§11.2.1. Condensation by Explicit Matrix Operations

To carry out the condensation process, the assembled stiffness equations of the superelement are partitioned as follows:

$$\begin{bmatrix} \mathbf{K}_{bb} & \mathbf{K}_{bi} \\ \mathbf{K}_{ib} & \mathbf{K}_{ii} \end{bmatrix} \begin{bmatrix} \mathbf{u}_b \\ \mathbf{u}_i \end{bmatrix} = \begin{bmatrix} \mathbf{f}_b \\ \mathbf{f}_i \end{bmatrix}. \quad (11.1)$$

where subvectors \mathbf{u}_b and \mathbf{u}_i collect *boundary* and *interior* degrees of freedom, respectively. Take the second matrix equation:

$$\mathbf{K}_{ib}\mathbf{u}_b + \mathbf{K}_{ii}\mathbf{u}_i = \mathbf{f}_i, \quad (11.2)$$

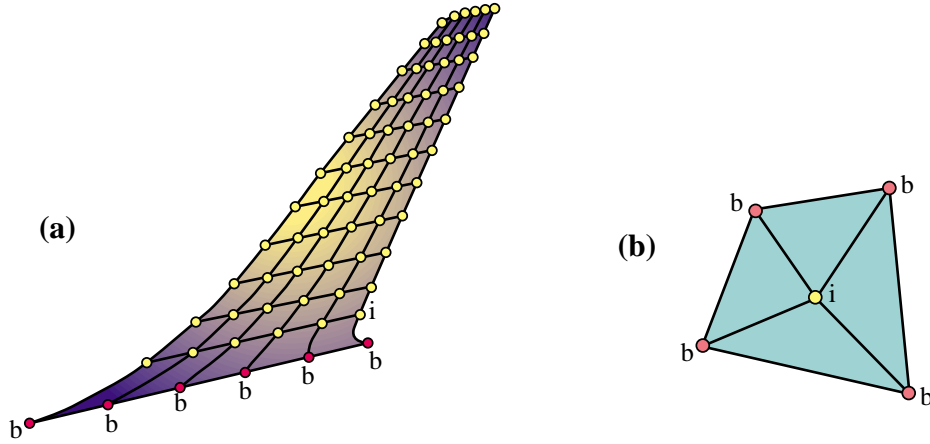


Figure 11.4. Classification of superelement freedoms into boundary and internal. (a) shows the vertical stabilizer substructure S_6 of Figure 11.2. (The FE mesh is depicted as two-dimensional for illustrative purposes; for an actual aircraft it will be three dimensional. Boundary freedoms are those associated to the boundary nodes labeled b (shown in red), which are connected to the fuselage substructure. (b) shows a quadrilateral macroelement mesh-unit fabricated with 4 triangles: it has one interior and four boundary nodes.

If \mathbf{K}_{ii} is nonsingular we can solve for the interior freedoms:

$$\mathbf{u}_i = \mathbf{K}_{ii}^{-1}(\mathbf{f}_i - \mathbf{K}_{ib}\mathbf{u}_b), \quad (11.3)$$

Replacing into the first matrix equation of (11.2) yields the *condensed stiffness equations*

$$\tilde{\mathbf{K}}_{bb}\mathbf{u}_b = \tilde{\mathbf{f}}_b. \quad (11.4)$$

In this equation,

$$\tilde{\mathbf{K}}_{bb} = \mathbf{K}_{bb} - \mathbf{K}_{bi}\mathbf{K}_{ii}^{-1}\mathbf{K}_{ib}, \quad \tilde{\mathbf{f}}_b = \mathbf{f}_b - \mathbf{K}_{bi}\mathbf{K}_{ii}^{-1}\mathbf{f}_i, \quad (11.5)$$

are called the *condensed* stiffness matrix and force vector, respectively, of the substructure.

From this point onward, the condensed superelement may be viewed, from the standpoint of further operations, as an *individual element* whose element stiffness matrix and nodal force vector are $\tilde{\mathbf{K}}_{bb}$ and $\tilde{\mathbf{f}}_b$, respectively.

Often each superelement has its own “local” coordinate system. A transformation of (11.5) to an overall global coordinate system is necessary upon condensation. In the case of multiple levels, the transformation is done with respect to the next-level superelement coordinate system. This coordinate transformation procedure automates the processing of repeated portions.

REMARK 11.1

The feasibility of the condensation process (11.5) hinges on the non-singularity of \mathbf{K}_{ii} . This matrix is nonsingular if the superelement is rank-sufficient in the sense stated in §11.1.3, and if fixing the boundary freedoms precludes all rigid body motions. If the former condition is verified but not the latter, the superelement is called *floating*. Processing floating superelements demands more advanced computational techniques, among which one may cite the use of projectors and generalized inverses [11.5].

```

CondenseLastFreedom[K_,f_]:=Module[{pivot,c,Kc,fc,
n=Length[K]}, If [n<=1,Return[{K,f}]];
Kc=Table[0,{n-1},{n-1}]; fc=Table[0,{n-1}];
pivot=K[[n,n]]; If [pivot==0, Print["CondenseLastFreedom:",
" Singular Matrix"]; Return[{K,f}]];
For [i=1,i<=n-1,i++, c=K[[i,n]]/pivot;
fc[[i]]=f[[i]]-c*f[[n]];
For [j=1,j<=i,j++,
Kc[[j,i]]=Kc[[i,j]]=K[[i,j]]-c*K[[n,j]]
];
];
Return[Simplify[{Kc,fc}]]
];

K={{6,-2,-1,-3},{-2,5,-2,-1},{-1,-2,7,-4},{-3,-1,-4,8}};
f={3,6,4,0};
Print["Before condensation:", " K=",K//MatrixForm," f=",f//MatrixForm];
{K,f}=CondenseLastFreedom[K,f];Print["Upon condensing DOF 4:",
" K=",K//MatrixForm," f=",f//MatrixForm];
{K,f}=CondenseLastFreedom[K,f];Print["Upon condensing DOF 3:",
" K=",K//MatrixForm," f=",f//MatrixForm];

```

Figure 11.5. *Mathematica* module to condense the last degree of freedom from a stiffness matrix and force vector. The test statements carry out the example (11.6)–(11.10).

§11.2.2. Condensation by Symmetric Gauss Elimination

In the computer implementation of the the static condensation process, calculations are not carried out as outlined above. There are two major differences. The equations of the substructure are not actually rearranged, and the explicit calculation of the inverse of \mathbf{K}_{ii} is avoided. The procedure may bein fact coded as a variant of symmetric Gauss elimination. To convey the flavor of this technique, consider the following stiffness equations of a superelement:

$$\begin{bmatrix} 6 & -2 & -1 & -3 \\ -2 & 5 & -2 & -1 \\ -1 & -2 & 7 & -4 \\ -3 & -1 & -4 & 8 \end{bmatrix} \begin{bmatrix} u_1 \\ u_2 \\ u_3 \\ u_4 \end{bmatrix} = \begin{bmatrix} 3 \\ 6 \\ 4 \\ 0 \end{bmatrix}. \quad (11.6)$$

Suppose that the last two displacement freedoms: u_3 and u_4 , are classified as interior and are to be statically condensed out. To eliminate u_4 , perform symmetric Gauss elimination of the fourth row and column:

$$\begin{bmatrix} 6 - \frac{(-3) \times (-3)}{8} & -2 - \frac{(-1) \times (-3)}{8} & -1 - \frac{(-4) \times (-3)}{8} \\ -2 - \frac{(-3) \times (-1)}{8} & 5 - \frac{(-1) \times (-1)}{8} & -2 - \frac{(-4) \times (-1)}{8} \\ -1 - \frac{(-3) \times (-4)}{8} & -2 - \frac{(-1) \times (-4)}{8} & 7 - \frac{(-4) \times (-4)}{8} \end{bmatrix} \begin{bmatrix} u_1 \\ u_2 \\ u_3 \end{bmatrix} = \begin{bmatrix} 3 - \frac{0 \times (-3)}{8} \\ 6 - \frac{0 \times (-1)}{8} \\ 4 - \frac{0 \times (-4)}{8} \end{bmatrix}, \quad (11.7)$$

or

$$\begin{bmatrix} \frac{39}{8} & -\frac{19}{8} & -\frac{5}{2} \\ -\frac{19}{8} & \frac{39}{8} & -\frac{5}{2} \\ -\frac{5}{2} & -\frac{5}{2} & 5 \end{bmatrix} \begin{bmatrix} u_1 \\ u_2 \\ u_3 \end{bmatrix} = \begin{bmatrix} 3 \\ 6 \\ 4 \end{bmatrix}. \quad (11.8)$$

Repeat the process for the third row and column to eliminate u_3 :

$$\begin{bmatrix} \frac{39}{8} - \frac{(-5/2) \times (-5/2)}{5} & -\frac{19}{8} - \frac{(-5/2) \times (-5/2)}{5} \\ -\frac{19}{8} - \frac{(-5/2) \times (-5/2)}{5} & \frac{39}{8} - \frac{(-5/2) \times (-5/2)}{5} \end{bmatrix} \begin{bmatrix} u_1 \\ u_2 \end{bmatrix} = \begin{bmatrix} 3 - \frac{4 \times (-5/2)}{5} \\ 6 - \frac{4 \times (-5/2)}{5} \end{bmatrix}, \quad (11.9)$$

or

$$\begin{bmatrix} \frac{29}{8} & -\frac{29}{8} \\ -\frac{29}{8} & \frac{29}{8} \end{bmatrix} \begin{bmatrix} u_1 \\ u_2 \end{bmatrix} = \begin{bmatrix} 5 \\ 8 \end{bmatrix}. \quad (11.10)$$

These are the condensed stiffness equations. Figure 11.5 shows a *Mathematica* program that carries out the foregoing steps. Module `CondenseLastFreedom` condenses the last freedom of a stiffness matrix K and a force vector f . It is invoked as $\{Kc, fc\} = \text{CondenseLastFreedom}[K, f]$. It returns the condensed stiffness Kc and force vector fc as new arrays. To do the example (11.6)–(11.10), the module is called twice, as illustrated in the test statements of Figure 11.5.

Obviously this procedure is much simpler than going through the explicit matrix inverse. Another important advantage of Gauss elimination is that equation rearrangement is not required even if the condensed degrees of freedom do not appear in any particular order. For example, suppose that the assembled substructure contains originally eight degrees of freedom and that the freedoms to be condensed out are numbered 1, 4, 5, 6 and 8. Then Gauss elimination is carried out over those equations only, and the condensed (3×3) stiffness and (3×1) force vector extracted from rows and columns 2, 3 and 7. An implementation of this process is considered in Exercise 11.2.

REMARK 11.2

The symmetric Gauss elimination procedure, as illustrated in steps (11.6)–(11.10), is primarily useful for macroelements and mesh units, since the number of stiffness equations for those typically does not exceed a few hundreds. This permits the use of full matrix storage. For substructures containing thousands or millions of degrees of freedom — such as in the airplane example — the elimination is carried out using more sophisticated sparse matrix algorithms; for example that described in [11.4].

REMARK 11.3

The static condensation process is a matrix operation called “partial inversion” or “partial elimination” that appears in many disciplines. Here is the general form. Suppose the linear system $\mathbf{Ax} = \mathbf{y}$, where \mathbf{A} is $n \times n$ square and \mathbf{x} and \mathbf{y} are n -vectors, is partitioned as

$$\begin{bmatrix} \mathbf{A}_{11} & \mathbf{A}_{12} \\ \mathbf{A}_{21} & \mathbf{A}_{22} \end{bmatrix} \begin{bmatrix} \mathbf{x}_1 \\ \mathbf{x}_2 \end{bmatrix} = \begin{bmatrix} \mathbf{y}_1 \\ \mathbf{y}_2 \end{bmatrix}. \quad (11.11)$$

Assuming the appropriate inverses to exist, then the following are easily verified matrix identities:

$$\begin{bmatrix} \mathbf{A}_{11}^{-1} & -\mathbf{A}_{11}^{-1}\mathbf{A}_{12} \\ \mathbf{A}_{21}\mathbf{A}_{11}^{-1} & \mathbf{A}_{22} - \mathbf{A}_{21}\mathbf{A}_{11}^{-1}\mathbf{A}_{12} \end{bmatrix} \begin{bmatrix} \mathbf{y}_1 \\ \mathbf{x}_2 \end{bmatrix} = \begin{bmatrix} \mathbf{x}_1 \\ \mathbf{y}_2 \end{bmatrix}, \quad \begin{bmatrix} \mathbf{A}_{11} - \mathbf{A}_{12}\mathbf{A}_{22}^{-1}\mathbf{A}_{21} & \mathbf{A}_{12}\mathbf{A}_{22}^{-1} \\ -\mathbf{A}_{22}^{-1}\mathbf{A}_{21} & \mathbf{A}_{22}^{-1} \end{bmatrix} \begin{bmatrix} \mathbf{x}_1 \\ \mathbf{y}_2 \end{bmatrix} = \begin{bmatrix} \mathbf{y}_1 \\ \mathbf{x}_2 \end{bmatrix}. \quad (11.12)$$

We say that \mathbf{x}_1 has been eliminated or “condensed out” in the left identity and \mathbf{x}_2 in the right one. In FEM applications, it is conventional to condense out the bottom vector \mathbf{x}_2 , so the right identity is relevant. If \mathbf{A} is symmetric, to retain symmetry in (11.12) it is necessary to change the sign of one of the subvectors.

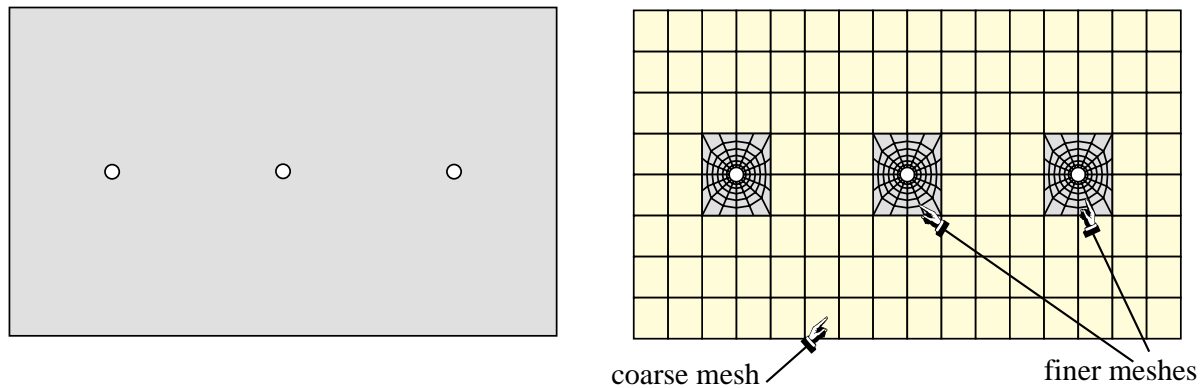


Figure 11.6. Left: example panel structure for global-local analysis.
Right: a FEM mesh for a one-shot analysis.

§11.3. GLOBAL-LOCAL ANALYSIS

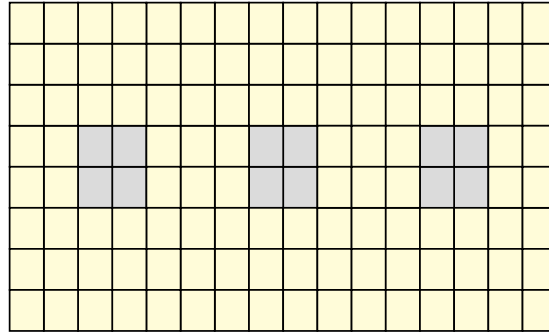
As discussed in the first Chapter, complex engineering systems are often modeled in a *multilevel* fashion following the divide and conquer approach. The superelement technique is a practical realization of that approach.

A related, but not identical, technique is *multiscale* analysis. The whole system is first analyzed as a global entity, discarding or passing over details deemed not to affect its overall behavior. Local details are then analyzed using the results of the global analysis as boundary conditions. The process can be continued into the analysis of further details of local models. And so on. When this procedure is restricted to two stages and applied in the context of finite element analysis, it is called *global-local* analysis in the FEM literature.

In the global stage the behavior of the entire structure is simulated with a finite element model that necessarily ignores details such as cutouts or joints. These details do not affect the overall behavior of the structure, but may have a bearing on safety. Such details are *a posteriori* incorporated in a series of local analyses.

The gist of the global-local approach is explained in the example illustrated in Figures 11.6 and 11.7. Although the structure is admittedly too simple to merit the application of global-local analysis, it does illustrate the basic ideas. Suppose one is faced with the analysis of the rectangular panel shown on the top of Figure 11.6, which contains three small holes. The bottom of that figure shows a standard (one-stage) FEM treatment using a largely regular mesh that is refined near the holes. Connecting the coarse and fine meshes usually involves using multifreedom constraints because the nodes at mesh boundaries do not match, as depicted in that figure.

Figure 11.7 illustrates the global-local analysis procedure. The global analysis is done with a coarse but regular FEM mesh which *ignores the effect of the holes*. This is followed by local analysis of the region near the holes using refined finite element meshes. The key ingredient for the local analyses is the application of boundary conditions (BCs) on the finer mesh boundaries. These BCs may be of displacement (essential) or of force (natural) type. If the former, the applied boundary displacements are interpolated from the global mesh solution. If the latter, the internal forces or stresses obtained from the global calculation are converted to nodal forces on the fine meshes



Global analysis with a coarse mesh, ignoring holes, followed bylocal analysis of the vicinity of the holes with finer meshes:

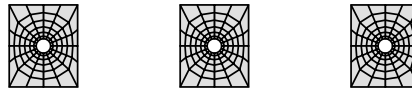


Figure 11.7. Global-local analysis of problem of Figure 11.6.

through a lumping process.

The BC choice noted above gives rise to two basic variations of the global-local approach. Experience accumulated over several decades³ has shown that the stress-BC approach generally gives more reliable answers.

The global-local technique can be extended to more than two levels, in which case it receives the more encompassing name *multiscale analysis*. Although this generalization is still largely in the realm of research, it is receiving increasing attention from various science and engineering communities for complex products such as the thermomechanical analysis of microelectronic components.

Notes and Bibliography

Substructuring was invented by aerospace engineers in the early 1960s. Przemieniecki's book [11.7] contains a fairly complete bibliography of early work. Most of this was in the form of largely inaccessible internal company or lab reports and so the actual history is difficult to trace. Macroelements appeared simultaneously in many of the early FEM codes. Quadrilateral macroelements fabricated with triangles are described in [11.3].

The generic term *superelement* was coined in the late 1960s by the SESAM group at DNV Veritas [11.2]. The matrix form of static condensation for a complete structure is presented in [11.1, p. 46], as a scheme to eliminate unloaded DOF in the displacement method. It is unclear when this idea was first applied to substructures or macroelements. The first application for reduced dynamical models is by Guyan [11.6].

The application of domain decomposition to parallel FEM solvers has produced an enormous and highly specialized literature. Procedures for handling floating superelements using generalized inverse methods are discussed in [11.5].

The global-local analysis procedure is also primarily used in industry and as such it is rarely mentioned in academic textbooks.

³ Particularly in the aerospace industry, in which the global-local technique has been used since the early 1960s.

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Homework Exercises for Chapter 11

Superelements and Global-Local Analysis

EXERCISE 11.1

[N:15] The free-free stiffness equations of a superelement are

$$\begin{bmatrix} 88 & -44 & -44 & 0 \\ -44 & 132 & -44 & -44 \\ -44 & -44 & 176 & -44 \\ 0 & -44 & -44 & 220 \end{bmatrix} \begin{bmatrix} u_1 \\ u_2 \\ u_3 \\ u_4 \end{bmatrix} = \begin{bmatrix} 5 \\ 10 \\ 15 \\ 20 \end{bmatrix}. \quad (\text{E11.1})$$

Eliminate u_2 and u_3 from (E11.1) by static condensation, and show (but do not solve) the condensed equation system. Use either the explicit matrix inverse formulas (11.5) or the symmetric Gauss elimination process explained in §11.2.2. Hint: regardless of method the result should be

$$\begin{bmatrix} 52 & -36 \\ -36 & 184 \end{bmatrix} \begin{bmatrix} u_1 \\ u_4 \end{bmatrix} = \begin{bmatrix} 15 \\ 30 \end{bmatrix}. \quad (\text{E11.2})$$

EXERCISE 11.2

[C+N:20] Generalize the program of Figure 11.5 to a module `CondenseFreedom[K,f,k]` that is able to condense the k th degree of freedom from \mathbf{K} and \mathbf{f} , which is not necessarily the last one. That is, k may range from 1 to n , where n is the number of freedoms in $\mathbf{K}\mathbf{u} = \mathbf{f}$. Apply that program to solve the previous Exercise.

Hint: here is a possible way of organizing the inner loop:

```
ii=0; For [i=1,i<=n,i++, If [i==k, Continue[]]; ii++;
      c=K[[i,k]]/pivot; fc[[ii]]=f[[i]]-c*f[[k]]; jj=0;
      For [j=1,j<=n,j++, If [j==k, Continue[]]; jj++;
        Kc[[ii,jj]]=K[[i,jj]]-c*K[[k,jj]]
      ];
    ];
Return[{Kc,fc}]
```

EXERCISE 11.3

[D:15] Explain the similarities and differences between superelement analysis and global-local FEM analysis.

EXERCISE 11.4

[A:20] If the superelement stiffness \mathbf{K} is symmetric, the static condensation process can be viewed as a special case of the master-slave transformation method discussed in Chapter 9. To prove this, take exterior freedoms \mathbf{u}_b as masters and interior freedoms \mathbf{u}_i as slaves. Assume the master-slave transformation relation

$$\mathbf{u} = \begin{bmatrix} \mathbf{u}_b \\ \mathbf{u}_i \end{bmatrix} = \begin{bmatrix} \mathbf{I} \\ -\mathbf{K}_{ii}^{-1} \mathbf{K}_{ib} \end{bmatrix} [\mathbf{u}_b] - \begin{bmatrix} \mathbf{0} \\ -\mathbf{K}_{ii}^{-1} \mathbf{f}_i \end{bmatrix} \stackrel{\text{def}}{=} \mathbf{T} \mathbf{u}_b - \mathbf{g}. \quad (\text{E11.3})$$

Work out $\hat{\mathbf{K}} = \mathbf{T}^T \mathbf{K} \mathbf{T}$ and $\hat{\mathbf{f}} = \mathbf{T}^T (\mathbf{f} - \mathbf{K} \mathbf{g})$, and show that $\hat{\mathbf{K}} \hat{\mathbf{u}} = \hat{\mathbf{f}}$ coalesces with the condensed stiffness equations (11.4)–(11.5) if $\mathbf{u}_b \equiv \hat{\mathbf{u}}$. (Take advantage of the symmetry properties $\mathbf{K}_{bi}^T = \mathbf{K}_{ib}$, $(\mathbf{K}_{ii}^{-1})^T = \mathbf{K}_{ii}^{-1}$.)

EXERCISE 11.5

[D:30] (Requires thinking) Explain the conceptual and operational differences between standard (one-stage) FEM analysis and global-local analysis of a problem such as that illustrated in Figures 11.6 and 11.7. Are the answers the same? What is gained, if any, by the global-local approach over the one-stage FEM analysis?

EXERCISE 11.6

[N:20] The widely used Guyan's scheme [11.6] for dynamic model reduction applies the static-condensation relation (E11.3) as master-slave transformation to both the stiffness and the mass matrix of the superelement. Use this procedure to eliminate the second and third DOF of the mass-stiffness system:

$$\mathbf{M} = \begin{bmatrix} 2 & 1 & 0 & 0 \\ 1 & 4 & 1 & 0 \\ 0 & 1 & 4 & 1 \\ 0 & 0 & 1 & 2 \end{bmatrix}, \quad \mathbf{K} = \begin{bmatrix} 1 & -1 & 0 & 0 \\ -1 & 2 & -1 & 0 \\ 0 & -1 & 2 & -1 \\ 0 & 0 & -1 & 1 \end{bmatrix}. \quad (\text{E11.4})$$

Compute and compare the vibration frequencies of the eigensystems $(\mathbf{M} + \omega_i^2 \mathbf{K})\mathbf{v}_i = \mathbf{0}$ and $(\hat{\mathbf{M}} + \hat{\omega}_i^2 \hat{\mathbf{K}})\hat{\mathbf{v}}_i = \mathbf{0}$ before and after reduction.

EXERCISE 11.7

[A:20] Two beam elements: 1–2 and 2–3, each of length L and rigidity EI are connected at node 2. Eliminate node 2 by condensation. Does the condensed stiffness equals the stiffness of a beam element of length $2L$ and rigidity EI ?

(For expressions of the beam stiffness matrices, see Chapter 13.)

12

Variational Formulation of Bar Element

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§12.1. A NEW BEGINNING

This Chapter begins Part II of the course. This Part focuses on the construction of structural and continuum finite elements using a *variational formulation* based on the Total Potential Energy. Why only elements? Because the other synthesis steps of the DSM: globalization, merge, BC application and solution, remain the same as in Part I. These operations are not element dependent.

Part II constructs *individual elements* beginning with the simplest ones and progressing to more complicated ones. The formulation of two-dimensional finite elements from a variational standpoint is discussed in Chapters 14 and following. Although the scope of that formulation is broad, exceeding structural mechanics, it is better understood by going through specific elements first.

From a geometrical standpoint the simplest finite elements are one-dimensional or *line elements*. This means that the *intrinsic dimensionality* is one, although these elements may be used in one, two or three space dimensions upon transformation to global coordinates as appropriate. The simplest one-dimensional structural element is the *two-node bar element*, which we have already encountered in Chapters 2, 3 and 6 as the truss member.

In this Chapter the bar stiffness equations are rederived using the variational formulation. For uniform properties the resulting equations are the same as those found previously using the physical or Mechanics of Materials approach. The variational method has the advantage of being readily extendible to more complicated situations, such as variable cross section or more than two nodes.

§12.2. DEFINITION OF BAR MEMBER

In structural mechanics a *bar* is a structural component characterized by two properties:

- (1) One bar dimension: the *longitudinal dimension* or *axial dimension* is much larger than the other two dimensions, which are collectively known as *transverse dimensions*. The intersection of a plane normal to the longitudinal dimension and the bar defines the *cross sections*. The longitudinal dimension defines the *longitudinal axis*. See Figure 12.1.
- (2) The bar resists an internal axial force along its longitudinal dimension.

In addition to trusses, bar elements are used to model cables, chains and ropes. They are also used as fictitious elements in penalty function methods, as discussed in Chapter 10.

We will consider here only *straight bars*, although their cross section may vary. The one-dimensional mathematical model assumes that the bar material is linearly elastic, obeying Hooke's law, and that displacements and strains are infinitesimal. Figure 12.2 pictures the relevant quantities for a fixed-free bar. Table 12.1 collects the necessary terminology for the governing equations.

Figure 12.3 displays the governing equations of the bar in a graphic format called a *Tonti diagram*. The formal similarity with the diagrams used in Chapter 6 to explain MoM elements should be noted, although the diagram of Figure 12.3 pertains to the continuum model rather than to the discrete one.

§12.3. VARIATIONAL FORMULATION

To illustrate the variational formulation, the finite element equations of the bar will be derived from the Minimum Potential Energy principle.

Table 12.1 Nomenclature for Mathematical Model of Axially Loaded Bar

<i>Quantity</i>	<i>Meaning</i>
x	Longitudinal bar axis*
$(.)'$	$d(.) / dx$
$u(x)$	Axial displacement
$q(x)$	Distributed axial force, given per unit of bar length
L	Total bar length
E	Elastic modulus
A	Cross section area; may vary with x
EA	Axial rigidity
$e = du/dx = u'$	Infinitesimal axial strain
$\sigma = Ee = Eu'$	Axial stress
$p = A\sigma = EAe = EAu'$	Internal axial force
P	Prescribed end load

* x is used in this Chapter instead of \bar{x} (as in Chapters 2–3) to simplify the notation.

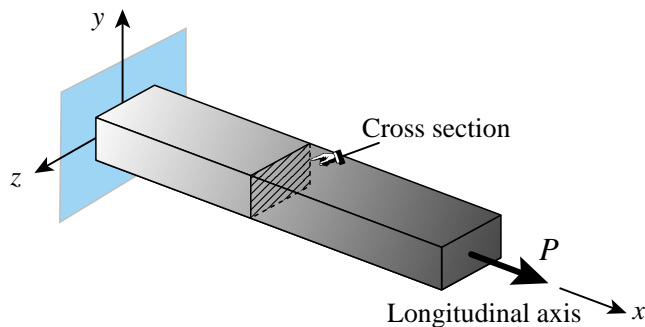


Figure 12.1. A fixed-free bar member.

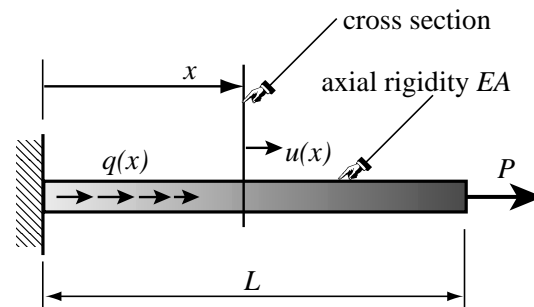


Figure 12.2. Quantities that appear in analysis of bar.

§12.3.1. The Total Potential Energy Functional

In Mechanics of Materials it is shown that the *internal energy density* at a point of a linear-elastic material subjected to a one-dimensional state of stress σ and strain e is $\mathcal{U} = \frac{1}{2}\sigma(x)e(x)$, where σ is to be regarded as linked to the displacement u through Hooke's law $\sigma = Ee$ and the strain-displacement relation $e = u' = du/dx$. This \mathcal{U} is also called the *strain energy density*. Integration over the volume of the bar gives the total internal energy

$$U = \frac{1}{2} \int_V \sigma e dV = \frac{1}{2} \int_0^L p e dx = \frac{1}{2} \int_0^L (EAu')u' dx = \frac{1}{2} \int_0^L u' EA u' dx, \quad (12.1)$$

in which all integrand quantities may depend on x .

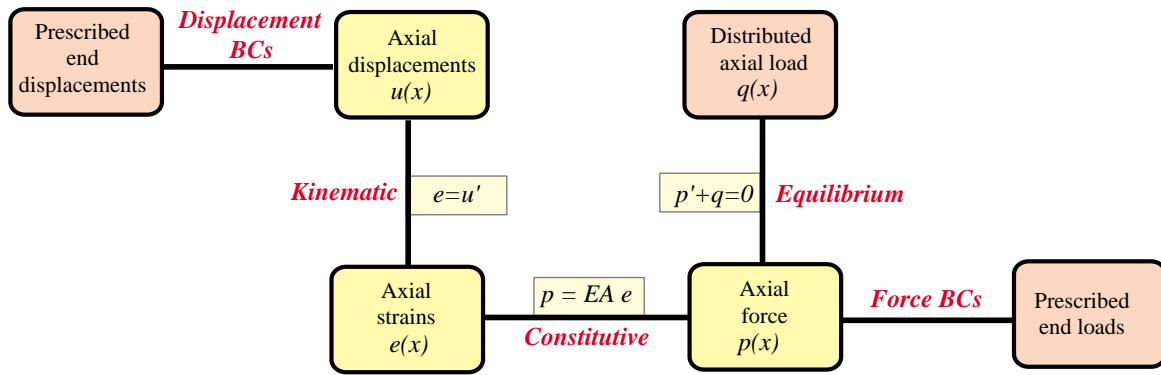


Figure 12.3. Tonti diagram for the mathematical model of a bar member. The diagram displays the field equations and boundary conditions as lines connecting the boxes. Yellow (brown) boxes contain unknown (given) quantities.

The *external energy* due to applied mechanical loads pools contributions from two sources:

1. The distributed load $q(x)$. This contributes a cross-section density of $q(x)u(x)$ because q is assumed to be already integrated over the section.
2. Any applied end load(s). For the fixed-free example of Figure 12.2 the end load P would contribute $P u(L)$.

The second source may be folded into the first by conventionally writing any point load P acting at a cross section $x = a$ as a contribution $P \delta(a)$ to $q(x)$, where $\delta(a)$ denotes the one-dimensional Dirac delta function at $x = a$. If this is done the external energy can be concisely expressed as

$$W = \int_0^L q u \, dx. \quad (12.2)$$

The total potential energy of the bar is given by

$$\boxed{\Pi = U - W} \quad (12.3)$$

Mathematically this is a functional, called the *Total Potential Energy* functional or TPE. It depends only on the axial displacement $u(x)$. In variational calculus this is called the *primary variable* of the functional. When the dependence of Π on u needs to be emphasized we shall write $\Pi[u] = U[u] - W[u]$, with brackets enclosing the primary variable. To display both primary and independent variables we write, for example, $\Pi[u(x)] = U[u(x)] - W[u(x)]$.

REMARK 12.1

According to the rules of Variational Calculus, the Euler-Lagrange equation for Π is

$$\frac{\partial \Pi}{\partial u} - \frac{d}{dx} \frac{\partial \Pi}{\partial u'} = -q - (EA u')' = 0 \quad (12.4)$$

This is the equation of equilibrium in terms of the axial displacement, usually written $(EA u')' + q = 0$, or $EA u'' + q = 0$ if EA is constant. This equation is not explicitly used in the FEM development. It is instead replaced by $\delta \Pi = 0$, with the variation restricted over the finite element interpolation functions.

§12.3.2. Variation of an Admissible Function

The concept of *admissible variation* is fundamental in both variational calculus and the variationally formulated FEM. *Only the primary variable(s) of a functional may be varied.* For the TPE functional (12.4) this is the axial displacement $u(x)$. Suppose that $u(x)$ is changed to $u(x) + \delta u(x)$.¹

This is illustrated in Figure 12.4, where for convenience $u(x)$ is plotted normal to x . The functional changes from Π to $\Pi + \delta\Pi$. The function $\delta u(x)$ and the scalar $\delta\Pi$ are called the *variations* of $u(x)$ and Π , respectively. The variation $\delta u(x)$ should not be confused with the ordinary differential $du(x) = u'(x) dx$ since on taking the variation the independent variable x is frozen; that is, $\delta x = 0$.

A displacement variation $\delta u(x)$ is said to be *admissible* when both $u(x)$ and $u(x) + \delta u(x)$ are *kinematically admissible* in the sense of the Principle of Virtual Work (PVW). This agrees with the conditions stated in the classic variational calculus.

A *kinematically admissible* axial displacement $u(x)$ obeys two conditions:

- (i) It is continuous over the bar length, that is, $u(x) \in \mathcal{C}_0$ in $x \in [0, L]$.
- (ii) It satisfies exactly any displacement boundary condition, such as the fixed-end specification $u(0) = 0$ of Figure 12.2.

The variation $\delta u(x)$ depicted in Figure 12.4 is kinematically admissible because both $u(x)$ and $u(x) + \delta u(x)$ satisfy the foregoing conditions. The physical meaning of (i)–(ii) is the subject of Exercise 12.1.

§12.3.3. The Minimum Potential Energy Principle

The Minimum Potential Energy (MPE) principle states that the actual displacement solution $u^*(x)$ that satisfies the governing equations is that which renders Π stationary:²

$$\delta\Pi = \delta U - \delta W = 0 \quad \text{iff} \quad u = u^* \quad (12.5)$$

with respect to *admissible* variations $u = u^* + \delta u$ of the exact displacement field $u^*(x)$.

REMARK 12.2

Using standard techniques of variational calculus³ it can be shown that if $EA > 0$ the solution $u^*(x)$ of (12.5) exists, is unique, and renders $\Pi[u]$ a minimum over the class of kinematically admissible displacements. The last attribute explains the “mininum” in the name of the principle.

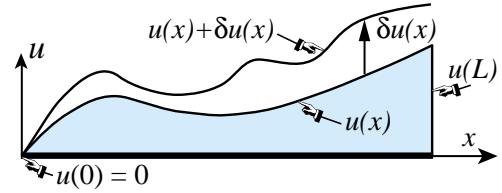


Figure 12.4. Concept of admissible variation of the axial displacement function $u(x)$. For convenience $u(x)$ is plotted normal to the longitudinal axis. Both depicted $u(x)$ and $u(x) + \delta u(x)$ are kinematically admissible, and so is the variation $\delta u(x)$.

¹ The symbol δ not immediately followed by a parenthesis is not a delta function but instead denotes variation with respect to the variable that follows.

² The symbol “iff” in (12.5) is an abbreviation for “if and only if”.

³ See references in **Notes and Bibliography** at the end of Chapter.

§12.3.4. TPE Discretization

To apply the TPE functional (12.2) to the derivation of finite element equations we replace the continuum mathematical model by a discrete one consisting of a union of bar elements. For example, Figure 12.5 illustrates the subdivision of a bar member into four two-node elements.

Functionals are scalars. Therefore, corresponding to a discretization such as that shown in Figure 12.5, the TPE functional (12.4) may be decomposed into a sum of contributions of individual elements:

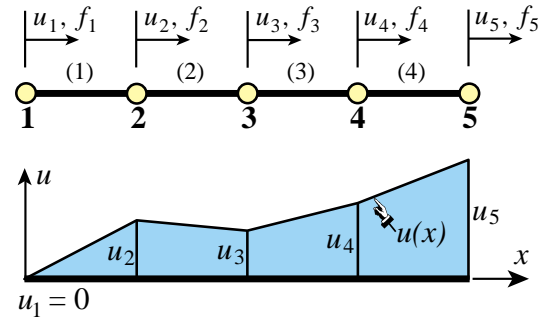


Figure 12.5. FEM discretization of bar member. A piecewise-linear admissible displacement trial function $u(x)$ is drawn underneath the mesh. It is assumed that the left end is fixed; thus $u_1 = 0$.

$$\Pi = \Pi^{(1)} + \Pi^{(2)} + \dots + \Pi^{(N_e)} \quad (12.6)$$

where N_e is the number of elements. The same decomposition applies to the internal and external energies, as well as to the stationarity condition (12.5):

$$\delta\Pi = \delta\Pi^{(1)} + \delta\Pi^{(2)} + \dots + \delta\Pi^{(N_e)} = 0. \quad (12.7)$$

Using the fundamental lemma of variational calculus,⁴ it can be shown that (12.7) implies that for a generic element e we may write

$$\delta\Pi^{(e)} = \delta U^{(e)} - \delta W^{(e)} = 0. \quad (12.8)$$

This *variational equation* is the basis for the derivation of element stiffness equations once the displacement field has been discretized over the element, as described next.

REMARK 12.3

In mathematics (12.8) is called a *weak form*. In mechanics it also states the Principle of Virtual Work for each element: $\delta U^{(e)} = \delta W^{(e)}$, which says that the virtual work of internal and external forces on admissible displacement variations is equal if the element is in equilibrium [12.8].

§12.3.5. Bar Element Discretization

Figure 12.6 depicts a generic bar element e . The element is referred to its local axis $\bar{x} = x - x_i$, which measures the distance from its left end. The two degrees of freedom are u_i and u_j . (Bars are not necessary on these values since the directions of \bar{x} and x are the same.)

The mathematical concept of bar finite elements is based on *approximation* of the axial displacement $u(x)$ over the element. The exact displacement u^* is replaced by an approximate displacement

$$u^*(x) \approx u(x) \quad (12.9)$$

⁴ See, e.g., Chapter II of Gelfand and Fomin [12.1].

over the finite element mesh. The value of $u(x)$ over element e is denoted by $u^{(e)}(x)$. This approximate displacement, $u(x)$, taken over all elements, is called the *finite element trial expansion* or simply *trial function*. See Figure 12.5.

This FE trial expansion must belong to the class of kinematically admissible displacements defined in §12.3.2. Consequently, it must be C_0 continuous over and between elements.

§12.3.6. Shape Functions

In a two-node bar element the only possible variation of the displacement $u^{(e)}$ that satisfies the interelement continuity requirement stated above is *linear*. It can be expressed by the interpolation formula

$$u^{(e)}(x) = N_i^{(e)} u_i^{(e)} + N_j^{(e)} u_j^{(e)} = [N_i^{(e)} \quad N_j^{(e)}] \begin{bmatrix} u_i^{(e)} \\ u_j^{(e)} \end{bmatrix} = \mathbf{N} \mathbf{u}^{(e)}. \quad (12.10)$$

The functions $N_i^{(e)}$ and $N_j^{(e)}$ that multiply the node displacements u_i and u_j are called *shape functions*. These functions *interpolate* the internal displacement $u^{(e)}$ directly from the node values.

See Figure 12.6. For the bar element, with $\bar{x} = x - x_i$ measuring the distance from the left node i , the shape functions are

$$N_i^{(e)} = 1 - \frac{\bar{x}}{\ell} = 1 - \zeta, \quad N_j^{(e)} = \frac{\bar{x}}{\ell} = \zeta. \quad (12.11)$$

Here $\ell = L^{(e)}$ denotes the element length whereas $\zeta = (x - x_i)/\ell = \bar{x}/\ell$ is a dimensionless coordinate, also known as a *natural coordinate*. Note that $dx = \ell d\zeta$ and $d\zeta = dx/\ell$. The shape function $N_i^{(e)}$ has the value 1 at node i and 0 at node j . Conversely, shape function $N_j^{(e)}$ has the value 0 at node i and 1 at node j . This is a general property. It follows from the fact that element displacement interpolations such as (12.10) are based on physical node values.

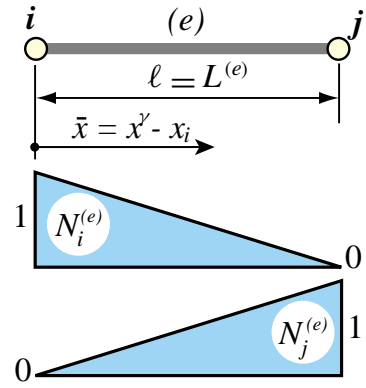


Figure 12.6. The shape functions of the generic bar element.

REMARK 12.4

In addition to continuity, shape functions must satisfy a *completeness* requirement with respect to the governing variational principle. This condition is stated and discussed in later Chapters. Suffices for now to say that the shape functions (12.11) do satisfy this requirement.

§12.3.7. The Strain-Displacement Equation

The axial strain over the element is

$$e = \frac{du^{(e)}}{dx} = (u^{(e)})' = \left[\frac{dN_i^{(e)}}{dx} \quad \frac{dN_j^{(e)}}{dx} \right] \begin{bmatrix} u_i^{(e)} \\ u_j^{(e)} \end{bmatrix} = \frac{1}{\ell} [-1 \quad 1] \begin{bmatrix} u_i^{(e)} \\ u_j^{(e)} \end{bmatrix} = \mathbf{B} \mathbf{u}^{(e)}, \quad (12.12)$$

where

$$\mathbf{B} = \frac{1}{\ell} [-1 \quad 1] \quad (12.13)$$

is called the *strain-displacement* matrix.

§12.3.8. *Trial Basis Functions

Shape functions are associated with elements. A *trial basis function*, or simply *basis function*, is associated with a node. Suppose node i of a bar discretization connects elements $(e1)$ and $(e2)$. The trial basis function N_i is defined as

$$N_i(x) = \begin{cases} N_i^{(e1)} & \text{if } x \in \text{element } (e1) \\ N_i^{(e2)} & \text{if } x \in \text{element } (e2) \\ 0 & \text{otherwise} \end{cases} \quad (12.14)$$

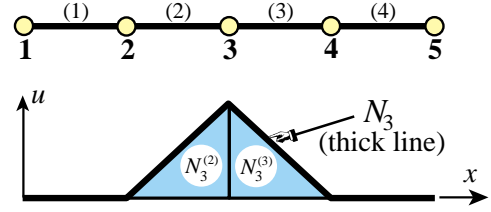


Figure 12.7. Trial basis function for node 3.

For a piecewise linear discretizations such as the two-node bar this function has the shape of a hat. Thus it is sometimes called a *hat function* or *chapeau function*. See Figure 12.7, in which $i = 3$, $e1 = 2$, $e2 = 3$. The concept is important in the variational interpretation of FEM as a Rayleigh-Ritz method.

§12.4. THE FINITE ELEMENT EQUATIONS

In linear FEM the discretization process for the TPE functional leads to the following algebraic form

$$\Pi^{(e)} = U^{(e)} - W^{(e)}, \quad U^{(e)} = \frac{1}{2}(\mathbf{u}^{(e)})^T \mathbf{K}^{(e)} \mathbf{u}^{(e)}, \quad W^{(e)} = (\mathbf{u}^{(e)})^T \mathbf{f}^{(e)}, \quad (12.15)$$

where $\mathbf{K}^{(e)}$ and $\mathbf{f}^{(e)}$ are called the *element stiffness matrix* and the *element consistent nodal force vector*, respectively. Note that in (12.15) the three energies are only function of the node displacements $\mathbf{u}^{(e)}$. $U^{(e)}$ and $W^{(e)}$ depend quadratically and linearly, respectively, on those displacements.

Taking the variation of the discretized TPE of (12.15) with respect to the node displacements gives⁵

$$\delta \Pi^{(e)} = (\delta \mathbf{u}^{(e)})^T \frac{\partial \Pi^{(e)}}{\partial \mathbf{u}^{(e)}} = (\delta \mathbf{u}^{(e)})^T [\mathbf{K}^{(e)} \mathbf{u}^{(e)} - \mathbf{f}^{(e)}] = 0. \quad (12.16)$$

Because the variations $\delta \mathbf{u}^{(e)}$ can be arbitrary, the bracketed quantity must vanish, which yields

$$\boxed{\mathbf{K}^{(e)} \mathbf{u}^{(e)} = \mathbf{f}^{(e)}} \quad (12.17)$$

These are the element stiffness equations. Hence the foregoing names given to $\mathbf{K}^{(e)}$ and $\mathbf{f}^{(e)}$ are justified *a posteriori*.

§12.4.1. The Stiffness Matrix

For the two-node bar element, the internal energy $U^{(e)}$ is

$$U^{(e)} = \frac{1}{2} \int_{x_i}^{x_j} e E A e dx = \frac{1}{2} \int_0^1 e E A e \ell d\zeta, \quad (12.18)$$

where the strain e is related to the nodal displacements through (12.12). This form is symmetrically expanded by inserting $e = \mathbf{B} \mathbf{u}^{(e)}$ into the second e and $e = e^T = (\mathbf{u}^{(e)})^T \mathbf{B}^T$ into the first e :

$$U^{(e)} = \frac{1}{2} \int_0^1 \begin{bmatrix} u_i^{(e)} & u_j^{(e)} \end{bmatrix} \frac{1}{\ell} \begin{bmatrix} -1 \\ 1 \end{bmatrix} E A \frac{1}{\ell} \begin{bmatrix} -1 & 1 \end{bmatrix} \begin{bmatrix} u_i^{(e)} \\ u_j^{(e)} \end{bmatrix} \ell d\zeta. \quad (12.19)$$

⁵ The $\frac{1}{2}$ factor disappears on taking the variation because $U^{(e)}$ is quadratic in the node displacements. For a review on the calculus of discrete quadratic forms, see Appendix D.

The nodal displacements can be moved out of the integral, giving

$$U^{(e)} = \frac{1}{2} [u_i^{(e)} \quad u_j^{(e)}] \int_0^1 \frac{EA}{\ell^2} \begin{bmatrix} 1 & -1 \\ -1 & 1 \end{bmatrix} \ell d\zeta \begin{bmatrix} u_i^{(e)} \\ u_j^{(e)} \end{bmatrix} = \frac{1}{2} (\mathbf{u}^{(e)})^T \mathbf{K}^{(e)} \mathbf{u}^{(e)}. \quad (12.20)$$

in which

$$\mathbf{K}^{(e)} = \int_0^1 EA \mathbf{B}^T \mathbf{B} \ell d\zeta = \int_0^1 \frac{EA}{\ell^2} \begin{bmatrix} 1 & -1 \\ -1 & 1 \end{bmatrix} \ell d\zeta. \quad (12.21)$$

is the element stiffness matrix. If the rigidity EA is constant over the element,

$$\mathbf{K}^{(e)} = EA \mathbf{B}^T \mathbf{B} \int_0^1 \ell d\zeta = \frac{EA}{\ell^2} \begin{bmatrix} 1 & -1 \\ -1 & 1 \end{bmatrix} \ell = \frac{EA}{\ell} \begin{bmatrix} 1 & -1 \\ -1 & 1 \end{bmatrix}. \quad (12.22)$$

This is the same element stiffness matrix of the prismatic truss member derived in Chapters 2 and 6 by a Mechanics of Materials approach, but now obtained through a variational argument.

§12.4.2. The Consistent Node Force Vector

The *consistent node force vector* $\mathbf{f}^{(e)}$ introduced in (12.15) comes from the element contribution to the external work potential W :

$$W^{(e)} = \int_{x_i}^{x_j} q u dx = \int_0^1 q \mathbf{N}^T \mathbf{u}^{(e)} \ell d\zeta = (\mathbf{u}^{(e)})^T \int_0^1 q \begin{bmatrix} 1 - \zeta \\ \zeta \end{bmatrix} \ell d\zeta = (\mathbf{u}^{(e)})^T \mathbf{f}^{(e)}, \quad (12.23)$$

in which $\zeta = (x - x_i)/\ell$. Consequently

$$\mathbf{f}^{(e)} = \int_{x_i}^{x_j} q \begin{bmatrix} 1 - \zeta \\ \zeta \end{bmatrix} dx = \int_0^1 q \begin{bmatrix} 1 - \zeta \\ \zeta \end{bmatrix} \ell d\zeta. \quad (12.24)$$

If the force q is constant over the element, one obtains the same results as with the EbE load-lumping method of Chapter 8. See Exercise 12.3.

§12.5. *ACCURACY ANALYSIS

Low order 1D elements may give surprisingly high accuracy. In particular the lowly two-node bar element can display infinite accuracy under some conditions. This phenomenon is studied in this advanced section as it provides an introduction to modified equation methods and Fourier analysis along the way.

§12.5.1. *Nodal Exactness and Superconvergence

Suppose that the following two conditions are satisfied:

1. The bar properties are constant along the length (prismatic member).
2. The distributed load $q(x)$ is zero between nodes. The only applied loads are point forces at the nodes.

If so, a linear axial displacement $u(x)$ as defined by (12.10) and (12.11) is the exact solution over each element since constant strain and stress satisfy, element by element, all of the governing equations listed in

Figure 12.3.⁶ It follows that if the foregoing conditions are verified the FEM solution is *exact*; that is, it agrees with the analytical solution of the mathematical model.⁷ Adding extra elements and nodes would not change the solution. That is the reason behind the truss discretizations used in Chapters 2–4: *one element per member is enough* if they are prismatic and loads are applied to joints. Such models are called *nodally exact*.

What happens if the foregoing assumptions are not met? Exactness is then generally lost, and several elements per member may be beneficial if spurious mechanisms are avoided.⁸ For a 1D lattice of equal-length, prismatic two-node bar elements, an interesting and more difficult result is: *the solution is nodally exact for any loading if consistent node forces are used*. This is proven in the subsection below. This result underlies the importance of computing node forces correctly.

If conditions such as equal-length are relaxed, the solution is no longer nodally exact but convergence at the nodes is extremely rapid (faster than could be expected by standard error analysis) as long as consistent node forces are used. This phenomenon is called *superconvergence* in the FEM literature.

§12.5.2. *Fourier Patch Analysis

The following analysis is based on the modified differential equation (MoDE) method of Warming and Hyett [12.16] combined with the Fourier patch analysis approach of Park and Flaggs [12.6,12.7].

Consider a lattice of two-node prismatic bar elements of constant rigidity EA and equal length ℓ , as illustrated in Figure 12.8. The total length of the lattice is L . The system is subject to an arbitrary axial load $q(x)$. The only requirement on $q(x)$ is that it has a convergent Fourier series in the space direction.

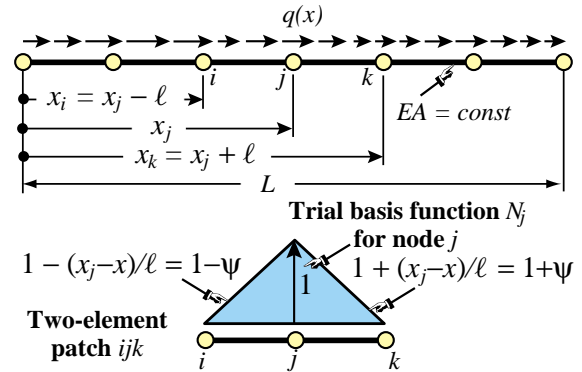


Figure 12.8. Superconvergence patch analysis.

From the lattice extract a patch⁹ of two elements connecting nodes x_i , x_j and x_k as shown in Figure 12.8. The FEM patch equations at node j are

$$\frac{EA}{\ell} \begin{bmatrix} -1 & 2 & -1 \end{bmatrix} \begin{bmatrix} u_i \\ u_j \\ u_k \end{bmatrix} = f_j, \quad (12.25)$$

in which the node force f_j is obtained by consistent lumping:

$$f_j = \int_{x_i}^{x_k} q(x) N_j(x) dx = \int_{-1}^0 q(x_j + \psi\ell)(1 + \psi) \ell d\psi + \int_0^1 q(x_j + \psi\ell)(1 - \psi) \ell d\psi. \quad (12.26)$$

Here $N_j(x)$ is the “hat” trial basis function for node j , depicted in Figure 12.8, and $\psi = (x - x_j)/\ell$ is a dimensionless coordinate that takes the values -1 , 0 and 1 at nodes i , j and k , respectively. If $q(x)$ is expanded in Fourier series

$$q(x) = \sum_{m=1}^M q_m e^{i\beta_m x}, \quad \beta_m = m\pi/L, \quad (12.27)$$

⁶ The internal equilibrium equation $p' + q = EA u'' + q = 0$ is trivially verified because $q = 0$ from the second assumption, and $u'' = 0$ because of shape function linearity.

⁷ In variational language: the Green function of the $u'' = 0$ problem is included in the FEM trial space.

⁸ These can happen when transforming such elements for 2D and 3D trusses. See Exercise E12.7.

⁹ A patch is the set of all elements connected to a node; in this case j .

(the term $m = 0$ requires special handling) the exact solution of the continuum equation $EA u'' + q = 0$ is

$$u^*(x) = \sum_{m=1}^M u_m^* e^{i\beta_m x}, \quad u_m^* = \frac{q_m e^{i\beta_m x}}{EA\beta_m^2}. \quad (12.28)$$

Evaluation of the consistent force using (12.26) gives

$$f_j = \sum_{m=1}^M f_{jm}, \quad f_{jm} = q_m \ell \frac{\sin^2(\frac{1}{2}\beta_m \ell)}{\frac{1}{4}\beta_m^2 \ell^2} e^{i\beta_m x_j}. \quad (12.29)$$

To construct a modified differential equation (MoDE), expand the displacement by Taylor series centered at node j . Evaluate at i and k : $u_i = u_j - \ell u'_j + \ell^2 u''_j/2! - \ell^3 u'''_j/3! + \ell^4 u^{iv}_j/4! + \dots$ and $u_k = u_j + \ell u'_j + \ell^2 u''_j/2 + \ell^3 u'''_j/3! + \ell^4 u^{iv}_j/4! + \dots$. Replace these series into (12.25) to get

$$-2EA\ell \left(\frac{1}{2!} u''_j + \frac{\ell^2}{4!} u^{iv}_j + \frac{\ell^4}{6!} u^{vi}_j + \dots \right) = f_j. \quad (12.30)$$

This is an ODE of infinite order. It can be reduced to an algebraic equation by assuming that the response of (12.30) to $q_m e^{i\beta_m x}$ is harmonic: $u_{jm} e^{i\beta_m x}$. If so $u''_{jm} = -\beta_m^2 u_{jm}$, $u^{iv}_{jm} = \beta_m^4 u_{jm}$, etc, and the MoDE becomes

$$2EA\ell\beta_m^2 \left(\frac{1}{2!} - \frac{\beta_m^2 \ell^2}{4!} + \frac{\beta_m^4 \ell^4}{6!} - \dots \right) u_{jm} = 4EA\ell \sin^2(\frac{1}{2}\beta_m \ell) u_{jm} = f_{jm} = q_m \ell \frac{\sin^2(\frac{1}{2}\beta_m \ell)}{\frac{1}{4}\beta_m^2 \ell^2} e^{i\beta_m x_j}. \quad (12.31)$$

Solving gives $u_{jm} = q_m e^{i\beta_m x_j} / (EA\beta_m^2)$, which compared with (12.28) shows that $u_{jm} = u_m^*$ for any $m > 0$. Consequently $u_j = u_j^*$. In other words, the MoDE (12.30) and the original ODE: $EAu'' + q = 0$ have the same value at $x = x_j$ for any load $q(x)$ developable as (12.27). This proves nodal exactness. In between nodes the two solutions will not agree.¹⁰

The case $m = 0$ has to be treated separately since the foregoing expressions become 0/0. The response to a uniform $q = q_0$ is a quadratic in x , and it is not difficult to prove nodal exactness.

Notes and Bibliography

The foregoing development pertains to the simplest structural finite element: the two-node bar element. For bars this may be generalized in various directions.

Refined bar elements. Adding internal nodes we can pass from linear to quadratic and cubic shape functions. These elements are rarely useful on their own right, but as accessories to 2D and 3D high order continuum elements (for example, to model edge reinforcements.) For that reason they are not considered here. The 3-node bar element is developed in exercises assigned in Chapter 16. *Two- and three-dimensional truss structures.* The only additional ingredients are the transformation matrices discussed in Chapters 3 and 6.

Curved bar elements. These can be derived using isoparametric mapping, a device introduced later.

Matrices for straight bar elements are available in any finite element book; for example Przemieniecki [12.9].

Tonti diagrams were introduced in the 1970s in papers now difficult to access, for example [12.13]. Scanned images are available, however, at <http://www.dic.units.it/perspage/discretephysics>

The fundamentals of Variational Calculus may be studied in the excellent textbook [12.1], which is now available in an inexpensive Dover edition. The proof of the MPE principle can be found in texts on variational

¹⁰ The FEM solution varies linearly between nodes whereas the exact one is generally trigonometric.

methods in mechanics. For example: Langhaar [12.5], which is the most readable “old fashioned” treatment of the energy principles of structural mechanics, with a beautiful treatment of virtual work. (Out of print but used copies may be found via the web engines cited in §1.5.2.) The elegant treatment by Lanczos [12.4] is recommended as reading material although it is more oriented to physics than structural mechanics.

The first accuracy study of FEM discretizations using modified equation methods is by Waltz et. al. [12.15]; however their procedures were faulty, which led to incorrect conclusions. The first correct derivation of modified equations appeared in [12.16]. The topic has recently attracted interest from applied mathematicians because modified equations provide a systematic tool for *backward error analysis* of differential equations: the discrete solution is the exact solution of the modified problem. This is particularly important for the study of long term behavior of discrete dynamical systems, whether deterministic or chaotic. Recommended references along these lines are [12.2,12.3,12.11].

Nodal exactness of bar models for point node loads is a particular case of a theorem by Tong [12.12]. For arbitrary loads it was proven by Park and Flaggs [12.6,12.7], who followed a variant of the scheme of §12.5.2. A different technique is used in Exercise 12.8. The budding concept of superconvergence, which emerged in the late 1960s, is outlined in the book of Strang and Fix [12.10]. There is a monograph [12.14] devoted to the subject; it covers only Poisson problems but provides a comprehensive reference list until 1995.

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Homework Exercises for Chapter 12

Variational Formulation of Bar Element

EXERCISE 12.1

[D:10] Explain the kinematic admissibility requirements stated in §12.3.2 in terms of physics, namely ruling out the possibility of gaps or interpenetration as the bar material deforms.

EXERCISE 12.2

[A/C:15] Using (12.21), derive the stiffness matrix for a *tapered* bar element in which the cross section area varies linearly along the element length:

$$A = A_i(1 - \zeta) + A_j \zeta, \quad (\text{E12.1})$$

where A_i and A_j are the areas at the end nodes, and $\zeta = x^{(e)}/\ell$ is the dimensionless coordinate defined in §12.3.6. Show that this yields the same answer as that of a stiffness of a constant-area bar with cross section $\frac{1}{2}(A_i + A_j)$. Note: the following *Mathematica* script may be used to solve this exercise:¹¹

```
ClearAll[Le,x,Em,A,Ai,Aj];
Be={{-1,1}}/Le; ζ=x/Le; A=Ai*(1-ζ)+Aj*ζ;
Ke=Integrate[Em*A*Transpose[Be].Be,{x,0,Le}];
Ke=Simplify[Ke];
Print["Ke for varying cross section bar: ",Ke//MatrixForm];
```

In this and following scripts Le stands for ℓ .

EXERCISE 12.3

[A:10] Using the area variation law (E12.1), find the consistent load vector $\mathbf{f}^{(e)}$ for a bar of constant area A subject to a uniform axial force $q = \rho g A$ per unit length along the element. Show that this vector is the same as that obtained with the element-by-element (EbE) “lumping” method of §8.4, which simply assigns half of the total load: $\frac{1}{2}\rho g A \ell$, to each node.

EXERCISE 12.4

[A/C:15] Repeat the previous calculation for the tapered bar element subject to a force $q = \rho g A$ per unit length, in which A varies according to (E12.1) whereas ρ and g are constant. Check that if $A_i = A_j$ one recovers $f_i = f_j = \frac{1}{2}\rho g A \ell$. Note: the following *Mathematica* script may be used to solve this exercise:¹²

```
ClearAll[q,A,Ai,Aj,ρ,g,Le,x];
ζ=x/Le; Ne={{1-ζ,ζ}}; A=Ai*(1-ζ)+Aj*ζ; q=ρ*g*A;
fe=Integrate[q*Ne,{x,0,Le}];
fe=Simplify[fe];
Print["fe for uniform load q: ",fe//MatrixForm];
ClearAll[A];
Print["fe check: ",Simplify[fe/.{Ai->A,Aj->A}]]//MatrixForm];
```

¹¹ The `ClearAll[...]` at the start of the script is recommended programming practice to initialize variables and avoid “cell crosstalk.” In a Module this is done by listing the local variables after the `Module` keyword.

¹² The `ClearAll[A]` before the last statement is essential; else A would retain the previous assignment.

EXERCISE 12.5

[A/C:20] A tapered bar element of length ℓ , end areas A_i and A_j with A interpolated as per (E12.1), and constant density ρ , rotates on a plane at uniform angular velocity ω (rad/sec) about node i . Taking axis x along the rotating bar with origin at node i , the centrifugal axial force is $q(x) = \rho A \omega^2 x$ along the length, in which $x \equiv x^{(e)}$. Find the consistent node forces as functions of ρ , A_i , A_j , ω and ℓ , and specialize the result to the prismatic bar $A = A_i = A_j$. Partial result check: $f_j = \frac{1}{3} \rho \omega^2 A \ell^2$ for $A = A_i = A_j$.

EXERCISE 12.6

[A:15] (Requires knowledge of Dirac's delta function properties.) Find the consistent load vector $\mathbf{f}^{(e)}$ if the bar is subjected to a concentrated axial force Q at a distance $x = a$ from its left end. Use Equation (12.31), with $q(x) = Q \delta(x - a)$, in which $\delta(x)$ is the one-dimensional Dirac's delta function at $x = a$. Note: the following script does it by *Mathematica*, but it is overkill:

```
ClearAll[Le,q,Q,a,x];
ξ=x/Le; Ne={{1-ξ,ξ}}; q=Q*DiracDelta[x-a];
fe=Simplify[ Integrate[q*Ne,{x,-Infinity,Infinity}] ];
Print["fe for point load Q at x=a: ",fe//MatrixForm];
```

EXERCISE 12.7

[C+D:20] In a learned paper, Dr. I. M. Clueless proposes “improving” the result for the example truss by putting three extra nodes, 4, 5 and 6, at the midpoint of members 1–2, 2–3 and 1–3, respectively. His “reasoning” is that more is better. Try Dr. C.’s suggestion using the *Mathematica* implementation of Chapter 5 and verify that the solution “blows up” because the modified master stiffness is singular. Explain physically what happens.

EXERCISE 12.8

[A:35, close to research paper level]. Prove nodal exactness of the two-node bar element for arbitrary but Taylor expandable loading without using the Fourier series approach. Hints: expand $q(x) = q(x_j) + (\ell\psi)q'(x_j) + (\ell\psi)^2 q''(x_j)/2! + \dots$, where $\ell\psi = x - x_j$ is the distance to node j , compute the consistent force $f_j(x)$ from (12.26), and differentiate the MoDE (12.30) repeatedly in x while truncating all derivatives to a maximum order $n \geq 2$. Show that the original ODE: $E A u'' + q = 0$, emerges as an identity regardless of how many derivatives are kept.

13

Variational Formulation of Plane Beam Element

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§13.1. INTRODUCTION

The previous Chapter introduced the TPE-based variational formulation of finite elements, which was illustrated for the bar element. This Chapter applies that technique to a more complicated one-dimensional element: the plane beam described by engineering beam theory.

Mathematically, the main difference of beams with respect to bars is the increased order of continuity required for the assumed transverse-displacement functions to be admissible. Not only must these functions be continuous but they must possess continuous x first derivatives. To meet this requirement both deflections *and* slopes are matched at nodal points. Slopes may be viewed as *rotational* degrees of freedom in the small-displacement assumptions used here.

§13.2. WHAT IS A BEAM?

Beams are the most common type of structural component, particularly in Civil and Mechanical Engineering. A *beam* is a bar-like structural member whose primary function is to support *transverse* loading and carry it to the supports. See Figure 13.1.

By “bar-like” it is meant that one of the dimensions is considerably larger than the other two. This dimension is called the *longitudinal dimension* or *beam axis*. The intersection of planes normal to the longitudinal dimension with the beam member are called *cross sections*. A *longitudinal plane* is one that passes through the beam axis.

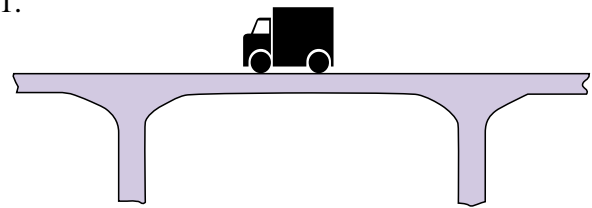


Figure 13.1. A beam is a structural member designed to resist transverse loads.

A beam resists transverse loads mainly through *bending action*. Bending produces compressive longitudinal stresses in one side of the beam and tensile stresses in the other.

The two regions are separated by a *neutral surface* of zero stress. The combination of tensile and compressive stresses produces an internal *bending moment*. This moment is the primary mechanism that transports loads to the supports. The mechanism is illustrated in Figure 13.2.

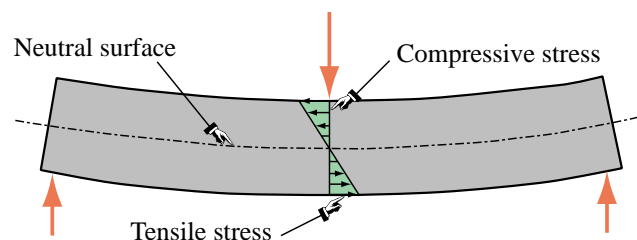


Figure 13.2. Beam transverse loads are primarily resisted by bending action.

§13.2.1. Terminology

A *general beam* is a bar-like member designed to resist a combination of loading actions such as biaxial bending, transverse shears, axial stretching or compression, and possibly torsion. If the internal axial force is compressive, the beam has also to be designed to resist buckling. If the beam is subject primarily to bending and axial forces, it is called a *beam-column*. If it is subjected primarily to bending forces, it is called simply a beam. A beam is *straight* if its longitudinal axis is straight. It is *prismatic* if its cross section is constant.

A *spatial beam* supports transverse loads that can act on arbitrary directions along the cross section. A *plane beam* resists primarily transverse loading on a preferred longitudinal plane. This Chapter considers only plane beams.

§13.2.2. Mathematical Models

One-dimensional mathematical models of structural beams are constructed on the basis of *beam theories*. Because beams are actually three-dimensional bodies, all models necessarily involve some form of approximation to the underlying physics. The simplest and best known models for straight, prismatic beams are based on the *Bernoulli-Euler* theory, also called *classical beam theory* or *engineering beam theory*, and the *Timoshenko beam theory*. The Bernoulli-Euler theory is that taught in introductory Mechanics of Materials, and is the one emphasized in this Chapter. The Timoshenko beam model is briefly outlined on §13.7.3, which contains advanced material.

Both models can be used to formulate beam finite elements. The classical beam theory used here leads to the so-called *Hermitian* beam elements.¹ These are also known as C^1 elements for the reason explained in §13.5.1. This model neglects transverse shear deformations. Elements based on Timoshenko beam theory, also known as C^0 elements, incorporate a first order correction for transverse shear effects. This model assumes additional importance in dynamics and vibration.

§13.2.3. Assumptions of Classical Beam Theory

The classical (Bernoulli-Euler) theory for *plane beams* rests on the following assumptions:

1. *Planar symmetry*. The longitudinal axis is straight, and the cross section of the beam has a longitudinal plane of symmetry. The resultant of the transverse loads acting on each section lies on this plane.
2. *Cross section variation*. The cross section is either constant or varies smoothly.
3. *Normality*. Plane sections originally normal to the longitudinal axis of the beam remain plane and normal to the deformed longitudinal axis upon bending.
4. *Strain energy*. The internal strain energy of the member accounts only for bending moment deformations. All other effects, notably transverse shear and axial force, are ignored.
5. *Linearization*. Transverse deflections, rotations and deformations are considered so small that the assumptions of infinitesimal deformations apply.
6. *Elastic behavior*. The material behavior is assumed to be elastic and isotropic. Heterogeneous beams fabricated with several materials, such as reinforced concrete, are not excluded.

§13.3. THE CLASSICAL BEAM THEORY

§13.3.1. Element Coordinate Systems

Under transverse loading one of the beam surfaces shortens while the other elongates; see Figure 13.2. Therefore a *neutral surface* exists between the top and the bottom that undergoes no elongation or contraction. The intersection of this surface with each cross section defines the *neutral axis* of that cross section.²

¹ The qualifier “Hermitian” relates to the use of a interpolation formula studied by the French mathematician Hermite. The term has nothing to do with the beam model used.

² If the beam is homogenous, the neutral axis passes through the centroid of the cross section. If the beam is fabricated of different materials — for example, a reinforced concrete beam — the neutral axes passes through the centroid of an “equivalent” cross section. This topic is covered in Mechanics of Materials textbooks; for example Popov [13.11].

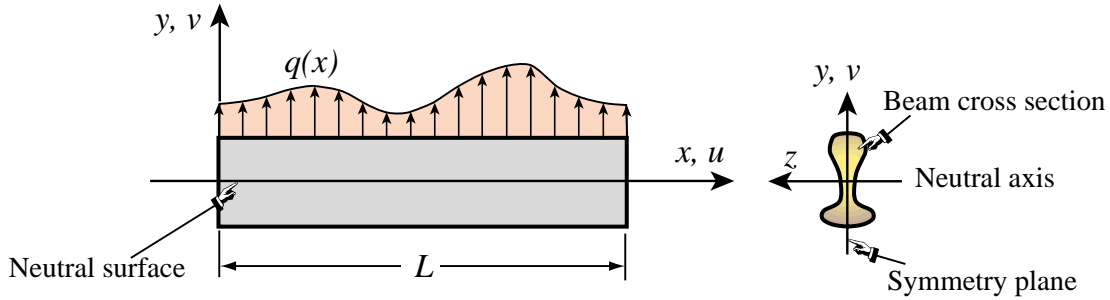


Figure 13.3. Terminology in Bernoulli-Euler model of plane beam.

The Cartesian axes for plane beam analysis are chosen as shown in Figure 13.3. Axis x lies along the longitudinal beam axis, at neutral axis height. Axis y lies in the symmetry plane and points upwards. Axis z is directed along the neutral axis, forming a RHS system with x and y . The origin is placed at the leftmost section. The total length (or span) of the beam member is called L .

§13.3.2. Kinematics

The *motion* under loading of a plane beam member in the x, y plane is described by the two dimensional displacement field

$$\begin{bmatrix} u(x, y) \\ v(x, y) \end{bmatrix}, \quad (13.1)$$

where u and v are the axial and transverse displacement components, respectively, of an arbitrary beam material point. The motion in the z direction, which is primarily due to Poisson's ratio effects, is of no interest. The normality assumption of the classical (Bernoulli-Euler) model can be represented mathematically as

$$u(x, y) = -y \frac{\partial v(x)}{\partial x} = -y v' = -y \theta, \quad v(x, y) = v(x). \quad (13.2)$$

Note that the slope $v' = \partial v / \partial x = dv / dx$ of the deflection curve has been identified with the *rotation* symbol θ . This is permissible because θ represents to first order, according to the kinematic assumptions of this model, the rotation of a cross section about z positive counterclockwise.

§13.3.3. Loading

The transverse force *per unit length* that acts on the beam in the $+y$ direction is denoted by $q(x)$, as illustrated in Figure 13.3. Concentrated loads and moments acting on isolated beam sections can be represented by the delta function and its derivative. For example, if a transverse point load F acts at $x = a$, it contributes $F\delta(a)$ to $q(x)$. If the concentrated moment C acts at $x = b$, positive counterclockwise, it contributes $C\delta'(b)$ to $q(x)$, where δ' denotes a doublet acting at $x = b$.

§13.3.4. Support Conditions

Support conditions for beams exhibit far more variety than for bar members. Two canonical cases often encountered in engineering practice: simple support and cantilever support. These are illustrated in Figures 13.4 and 13.5, respectively. Beams often appear as components of skeletal structures called frameworks, in which case the support conditions are of more complex type.

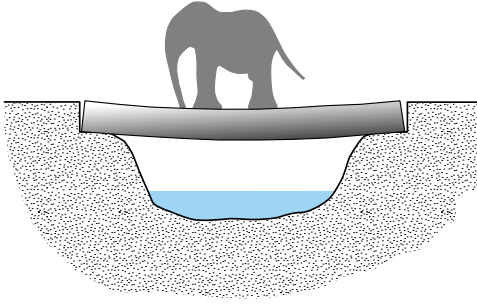


Figure 13.4. A simply supported beam has end supports that preclude transverse displacements but permit end rotations.

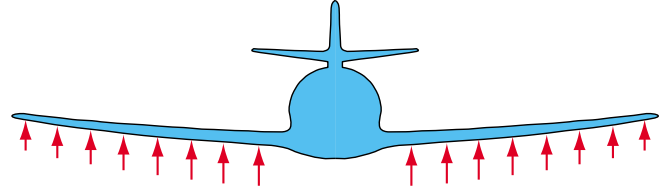


Figure 13.5. A cantilever beam is clamped at one end and free at the other. Airplane wings and stabilizers are examples of this configuration.

§13.3.5. Strains, Stresses and Bending Moments

The classical (Bernoulli-Euler) model assumes that the internal energy of beam member is entirely due to bending strains and stresses. Bending produces axial stresses σ_{xx} , which will be abbreviated to σ , and axial strains e_{xx} , which will be abbreviated to e . The strains can be linked to the displacements by differentiating the axial displacement $u(x)$ of (13.2):

$$e = \frac{\partial u}{\partial x} = -y \frac{\partial^2 v}{\partial x^2} = -y \frac{d^2 v}{dx^2} = -y\kappa. \quad (13.3)$$

Here κ denotes the deformed beam axis curvature, which to first order is $\partial^2 v / \partial x^2 \approx v''$. The bending stress $\sigma = \sigma_{xx}$ is linked to e through the one-dimensional Hooke's law

$$\sigma = Ee = -Ey \frac{d^2 v}{dx^2} = -Ey\kappa, \quad (13.4)$$

where E is the elastic modulus.

The most important stress resultant in classical beam theory is the *bending moment* M , which is defined as the cross section integral

$$M = \int_A -y\sigma \, dx = E \frac{d^2 v}{dx^2} \int_A y^2 \, dA = EI \kappa. \quad (13.5)$$

Here $I \equiv I_{zz}$ denotes the moment of inertia $\int_A y^2 \, dA$ of the cross section with respect to the z (neutral) axis. M is considered positive if it compresses the upper portion: $y > 0$, of the beam cross section, as illustrated in Figure 13.6. This explains the negative sign of y in the integral (13.5). The product EI is called the *bending rigidity* of the beam with respect to flexure about the z axis.

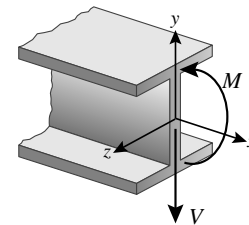


Figure 13.6. Positive sign convention for M and V .

The governing equations of the Bernoulli-Euler beam model are summarized in the Tonti diagram of Figure 13.7.

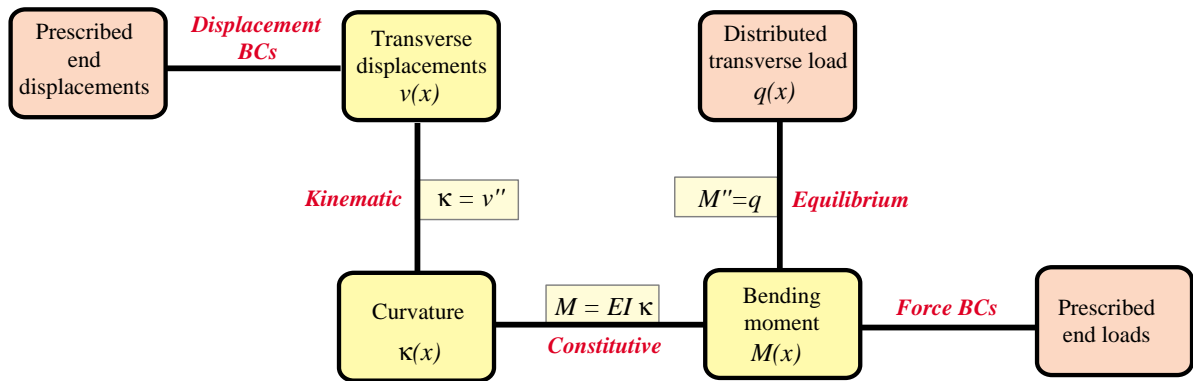


Figure 13.7. The Tonti diagram for the governing equations of the Bernoulli-Euler beam model.

§13.4. TOTAL POTENTIAL ENERGY FUNCTIONAL

The total potential energy of the beam is

$$\boxed{\Pi = U - W} \quad (13.6)$$

where as usual U and W denote the internal and external energies, respectively. As previously explained, in the Bernoulli-Euler model U includes only the bending energy:

$$\boxed{U = \frac{1}{2} \int_V \sigma e \, dV = \frac{1}{2} \int_0^L M \kappa \, dx = \frac{1}{2} \int_0^L EI \kappa^2 \, dx = \frac{1}{2} \int_0^L EI (v'')^2 \, dx = \frac{1}{2} \int_0^L v'' EI v'' \, dx.} \quad (13.7)$$

The external work W accounts for the applied transverse force:

$$\boxed{W = \int_0^L q v \, dx.} \quad (13.8)$$

The three functionals Π , U and W must be regarded as depending on the transverse displacement $v(x)$. When this dependence needs to be emphasized we write $\Pi[v]$, $U[v]$ and $W[v]$.

Note that $\Pi[v]$ includes up to second derivatives in v , because $v'' = \kappa$ appears in U . This number is called the *variational index*. Variational calculus tells us that since the index is 2, admissible displacements $v(x)$ must be continuous, have continuous first derivatives (slopes or rotations), and satisfy the displacement BCs exactly. This continuity requirement can be succinctly stated by saying that admissible displacements must be C^1 continuous. This condition guides the construction of beam finite elements described below.

REMARK 13.1

If there is an applied distributed moment $m(x)$ per unit of beam length, the external energy (13.8) must be augmented with a $\int_0^L m(x) \theta(x) \, dx$ term. This is further elaborated in Exercises 13.4 and 13.5. Such kind of distributed loading is uncommon in practice although in framework analysis occasionally the need arises for treating a concentrated moment between nodes.

§13.5. BEAM FINITE ELEMENTS

Beam finite elements are obtained by subdividing beam members longitudinally. The simplest Bernoulli-Euler plane beam element, depicted in Figure 13.8, has two end nodes, i and j , and four degrees of freedom collected in the node displacement vector

$$\mathbf{u}^{(e)} = [v_i^{(e)} \quad \theta_i^{(e)} \quad v_j^{(e)} \quad \theta_j^{(e)}]^T. \quad (13.9)$$

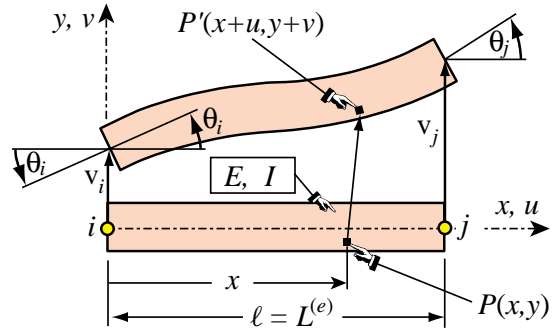


Figure 13.8. The two-node Bernoulli-Euler plane beam element with four DOFs.

§13.5.1. Finite Element Trial Functions

The freedoms (13.9) are used to define uniquely the variation of the transverse displacement $v^{(e)}(x)$ over the element. The C^1 continuity requirement says that both w and the slope $\theta = v'$ must be continuous over the entire beam member, and in particular between beam elements.

C^1 continuity can be trivially satisfied within each element by choosing polynomial interpolation functions as shown below. Matching the nodal displacements and rotations with adjacent beam elements enforces the necessary interelement continuity.

§13.5.2. Shape Functions

The simplest shape functions that meet the C^1 continuity requirement for the nodal freedom configuration (13.9) are called the *Hermitian cubic* shape functions. The interpolation formula based on these functions may be written as

$$v^{(e)} = [N_{v_i}^{(e)} \quad N_{\theta_i}^{(e)} \quad N_{v_j}^{(e)} \quad N_{\theta_j}^{(e)}] \begin{bmatrix} v_i^{(e)} \\ \theta_i^{(e)} \\ v_j^{(e)} \\ \theta_j^{(e)} \end{bmatrix} = \mathbf{N} \mathbf{u}^{(e)}. \quad (13.10)$$

These shape functions are conveniently written in terms of the dimensionless “natural” coordinate

$$\xi = \frac{2x}{\ell} - 1, \quad (13.11)$$

which varies from -1 at node i ($x = 0$) to $+1$ at node j ($x = \ell$); ℓ being the element length:

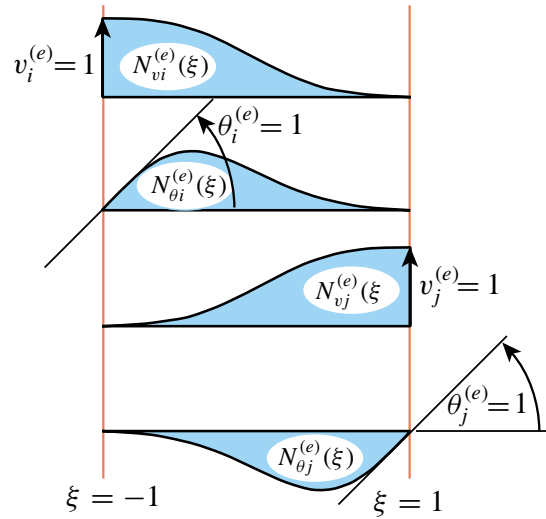


Figure 13.9. Cubic shape functions of plane beam element.

$$\begin{aligned} N_{v_i}^{(e)} &= \frac{1}{4}(1 - \xi)^2(2 + \xi), & N_{\theta_i}^{(e)} &= \frac{1}{8}\ell(1 - \xi)^2(1 + \xi), \\ N_{v_j}^{(e)} &= \frac{1}{4}(1 + \xi)^2(2 - \xi), & N_{\theta_j}^{(e)} &= -\frac{1}{8}\ell(1 + \xi)^2(1 - \xi). \end{aligned} \quad (13.12)$$

These four functions are depicted in Figure 13.9. The curvature κ that appears in U can be expressed in terms of the nodal displacements by differentiating twice with respect to x :

$$\kappa = \frac{d^2 v^{(e)}(x)}{dx^2} = \frac{4}{\ell^2} \frac{d^2 v^{(e)}(\xi)}{d\xi^2} = \frac{4}{\ell^2} \frac{d\mathbf{N}^{(e)}}{d\xi^2} \mathbf{u}^{(e)} = \mathbf{B}\mathbf{u}^{(e)} = \mathbf{N}''\mathbf{u}^{(e)}. \quad (13.13)$$

Here $\mathbf{B} = \mathbf{N}''$ is the 1×4 curvature-displacement matrix

$$\mathbf{B} = \frac{1}{\ell} \begin{bmatrix} 6\frac{\xi}{\ell} & 3\xi - 1 & -6\frac{\xi}{\ell} & 3\xi + 1 \end{bmatrix}. \quad (13.14)$$

REMARK 13.2

The $4/\ell^2$ factor in (13.13) comes from the differentiation chain rule. If $f(x)$ is a function of x , and $\xi = 2x/\ell - 1$,

$$\frac{df(x)}{dx} = \frac{df(\xi)}{d\xi} \frac{d\xi}{dx} = \frac{2}{\ell} \frac{df(\xi)}{d\xi}, \quad \frac{d^2 f(x)}{dx^2} = \frac{d(2/\ell)}{dx} \frac{df(\xi)}{d\xi} + \frac{2}{\ell} \frac{d}{dx} \left(\frac{df(\xi)}{d\xi} \right) = \frac{4}{\ell^2} \frac{d^2 f(\xi)}{d\xi^2}, \quad (13.15)$$

because $d(2/\ell)/dx = 0$.

§13.6. THE FINITE ELEMENT EQUATIONS

Insertion of (13.12) and (13.14) into the TPE functional specialized to this element, yields the quadratic form in the nodal displacements

$$\Pi^{(e)} = \frac{1}{2} \mathbf{u}^{(e)T} \mathbf{K}^{(e)} \mathbf{u}^{(e)} - \mathbf{u}^{(e)T} \mathbf{f}^{(e)}, \quad (13.16)$$

where

$$\mathbf{K}^{(e)} = \int_0^\ell EI \mathbf{B}^T \mathbf{B} dx = \int_{-1}^1 EI \mathbf{B}^T \mathbf{B} \frac{1}{2} \ell d\xi, \quad (13.17)$$

is the element stiffness matrix and

$$\mathbf{f}^{(e)} = \int_0^\ell \mathbf{N}^T q dx = \int_{-1}^1 \mathbf{N}^T q \frac{1}{2} \ell d\xi, \quad (13.18)$$

is the consistent element node force vector. The calculation of the entries of $\mathbf{K}^{(e)}$ and $\mathbf{f}^{(e)}$ for prismatic beams and uniform load q is studied next. More complex cases are treated in the Exercises.

§13.6.1. The Stiffness Matrix of a Prismatic Beam

If the bending rigidity EI is constant over the element it can be moved out of the ξ -integral in (13.17):

$$\mathbf{K}^{(e)} = \frac{1}{2} EI \ell \int_{-1}^1 \mathbf{B}^T \mathbf{B} d\xi = \frac{EI}{2\ell} \int_{-1}^1 \begin{bmatrix} \frac{6\xi}{\ell} \\ 3\xi - 1 \\ -\frac{6\xi}{\ell} \\ 3\xi + 1 \end{bmatrix} \begin{bmatrix} \frac{6\xi}{\ell} & 3\xi - 1 & -\frac{6\xi}{\ell} & 3\xi + 1 \end{bmatrix} d\xi. \quad (13.19)$$

Expanding and integrating over the element yields

$$\mathbf{K}^{(e)} = \frac{EI}{2\ell^3} \int_{-1}^1 \begin{bmatrix} 36\xi^2 & 6\xi(3\xi-1)\ell & -36\xi^2 & 6\xi(3\xi+1)\ell \\ (3\xi-1)^2\ell^2 & -6\xi(3\xi-1)\ell & 36\xi^2 & -6\xi(3\xi+1)\ell \\ \text{symm} & & & (3\xi+1)^2\ell^2 \end{bmatrix} d\xi = \frac{EI}{\ell^3} \begin{bmatrix} 12 & 6\ell & -12 & 6\ell \\ 4\ell^2 & -6\ell & 2\ell^2 & \\ & 12 & -6\ell & \\ \text{symm} & & & 4\ell^2 \end{bmatrix} \quad (13.20)$$

Although the foregoing integrals can be easily carried out by hand, it is equally expedient to use a CAS such as *Mathematica* or *Maple*. For example the *Mathematica* script listed in the top box of Figure 13.10 processes (13.20) using the `Integrate` function. The output, shown in the bottom box, corroborates the hand integration result.

```
ClearAll[EI,l,\xi];
B={{{6*\xi,(3*\xi-1)*l,-6*\xi,(3*\xi+1)*l}}/l^2;
Ke=(EI*l/2)*Integrate[Transpose[B].B,{\xi,-1,1}];
Ke=Simplify[Ke];Print["Ke for prismatic beam:"];
Print[Ke//MatrixForm];
```

Ke for prismatic beam:

$$\begin{pmatrix} \frac{12 EI}{l^3} & \frac{6 EI}{l^2} & -\frac{12 EI}{l^3} & \frac{6 EI}{l^2} \\ \frac{6 EI}{l^2} & \frac{4 EI}{l} & -\frac{6 EI}{l^2} & \frac{2 EI}{l} \\ -\frac{12 EI}{l^3} & -\frac{6 EI}{l^2} & \frac{12 EI}{l^3} & -\frac{6 EI}{l^2} \\ \frac{6 EI}{l^2} & \frac{2 EI}{l} & -\frac{6 EI}{l^2} & \frac{4 EI}{l} \end{pmatrix}$$

Figure 13.10. Using *Mathematica* to form $\mathbf{K}^{(e)}$ for a prismatic beam element.

§13.6.2. Consistent Nodal Force Vector for Uniform Load

If q does not depend on x it can be moved out of (13.18), giving

$$\mathbf{f}^{(e)} = \frac{1}{2}q\ell \int_{-1}^1 \mathbf{N}^T d\xi = \frac{1}{2}q\ell \int_{-1}^1 \begin{bmatrix} \frac{1}{4}(1-\xi)^2(2+\xi) \\ \frac{1}{8}\ell(1-\xi)^2(1+\xi) \\ \frac{1}{4}(1+\xi)^2(2-\xi) \\ -\frac{1}{8}\ell(1+\xi)^2(1-\xi) \end{bmatrix} d\xi = q\ell \begin{bmatrix} \frac{1}{2} \\ \frac{1}{12}\ell \\ \frac{1}{2} \\ -\frac{1}{12}\ell \end{bmatrix}. \quad (13.21)$$

This shows that a uniform load q over the beam element maps to two transverse node loads $q\ell/2$, as may be expected, plus two nodal moments $\pm q\ell^2/12$. The latter are called the *fixed-end moments* in the FEM literature.

The hand result (13.21) can be verified with the *Mathematica* script displayed in Figure 13.11, in which $\mathbf{f}^{(e)}$ is printed as a row vector to save space.

```
ClearAll[q,l,\xi]
Ne={{2*(1-\xi)^2*(2+\xi),(1-\xi)^2*(1+\xi)*l,
      2*(1+\xi)^2*(2-\xi),-(1+\xi)^2*(1-\xi)*l}}/8;
fe=(q*l/2)*Integrate[Ne,{\xi,-1,1}]; fe=Simplify[fe];
Print["fe^T for uniform load q:"];
Print[fe//MatrixForm];
```

fe^T for uniform load q:

$$\left(\frac{l q}{2} \quad \frac{l^2 q}{12} \quad \frac{l q}{2} \quad -\frac{l^2 q}{12} \right)$$

Figure 13.11. Using *Mathematica* to form $\mathbf{f}^{(e)}$ for uniform load q .

§13.7. *GENERALIZED CUBIC INTERPOLATION

For derivation of special and C^0 beam elements it is convenient to use a transverse-displacement cubic interpolation in which the nodal freedoms v_i, v_j, θ_i and θ_j are replaced by *generalized coordinates* q_1 to q_4 :

$$v(\xi) = N_{q1} q_1 + N_{q2} q_2 + N_{q3} q_3 + N_{q4} q_4 = \mathbf{N}_q \mathbf{q}. \quad (13.22)$$

The $N_{qi}(\xi)$ are shape functions that satisfy the completeness and continuity requirements discussed in Chapter 19. Note that \mathbf{N}_q is a 1×4 matrix whereas \mathbf{q} is a column 4-vector. This is called a *generalized interpolation*. It includes the physical interpolation (13.10) as an instance when $q_1 = v_i, q_2 = \theta_i, q_3 = v_j$ and $q_4 = \theta_j$.

§13.7.1. *Legendre Polynomials

An obvious generalized form is the polynomial $v(\xi) = q_1 + q_2 \xi + q_3 \xi^2 + q_4 \xi^3$, but this turns out not to be particularly useful. A more seminal form is

$$v(\xi) = L_1 q_1 + L_2 q_2 + L_3 q_3 + L_4 q_4 = \mathbf{L} \mathbf{q}, \quad (13.23)$$

where the L_i are the first four Legendre polynomials³

$$L_1(\xi) = 1, \quad L_2(\xi) = \xi, \quad L_3(\xi) = \frac{1}{2}(3\xi^2 - 1), \quad L_4(\xi) = \frac{1}{2}(5\xi^3 - 3\xi). \quad (13.24)$$

In this case q_1 through q_4 have dimension of length. The functions (13.24) and their first two ξ -derivatives are plotted in Figure 13.12.

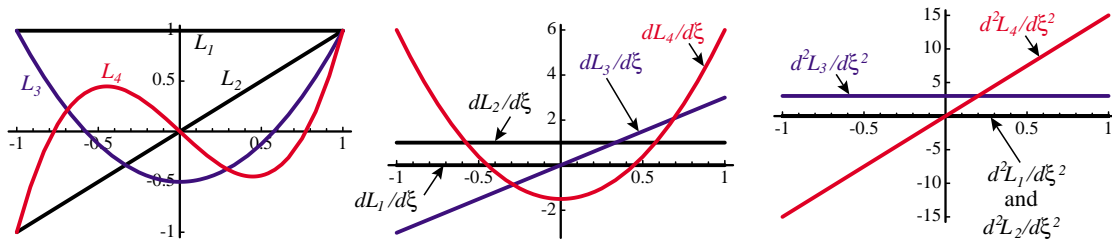


Figure 13.12. The Legendre polynomials and their first two ξ -derivatives plotted over $\xi \in [-1, 1]$. Rigid body modes (L_1 and L_2) in black. Deformational modes (L_3 and L_4) in color.

Unlike the shape functions (13.12), the L_i have a clear physical meaning as can be gathered from the picture:

- L_1 Translational rigid body mode.
- L_2 Rotational rigid body mode.
- L_3 Constant-curvature symmetric deformation mode (symmetric with respect to $\xi = 0$)
- L_4 Linear-curvature antisymmetric deformation mode (antisymmetric with respect to $\xi = 0$)

These properties are also shared by the standard polynomial interpolation $q_1 + q_2 \xi + q_3 \xi^2 + q_4 \xi^3$. What distinguishes the set (13.24) are the orthogonality properties

$$\mathbf{Q} = \int_0^\ell \mathbf{L}^T \mathbf{L} dx = \ell \begin{bmatrix} 1 & 0 & 0 & 0 \\ 0 & 1/3 & 0 & 0 \\ 0 & 0 & 1/5 & 0 \\ 0 & 0 & 0 & 1/7 \end{bmatrix}, \quad \mathbf{S} = \int_0^\ell (\mathbf{L}'')^T \mathbf{L}'' dx = \frac{48}{\ell^3} \begin{bmatrix} 0 & 0 & 0 & 0 \\ 0 & 0 & 0 & 0 \\ 0 & 0 & 3 & 0 \\ 0 & 0 & 0 & 25 \end{bmatrix}. \quad (13.25)$$

³ This is a FEM oriented nomenclature; L_1 through L_4 are called P_0 through P_3 in mathematical handbooks; e.g. Chapter 22 of Abramowitz and Stegun [13.1]. They are normalized as per the usual conventions: $L_i(1) = 1, L_i(-1) = (-1)^{i-1}$. One important application in numerical analysis is the construction of one-dimensional Gauss integration rules: the abscissas of the p -point rule are the zeros of $L_{p+1}(\xi)$.

The stiffness matrix in terms of the q_i is called the generalized stiffness. Denoting the bending rigidity by R :

$$\mathbf{K}_q = \int_0^\ell R (\mathbf{L}'')^T \mathbf{L}'' dx. \quad (13.26)$$

For a prismatic Bernoulli-Euler plane beam element $R = EI$ is constant and $\mathbf{K}_q = EI\mathbf{S}$, where \mathbf{S} is the second diagonal matrix in (13.25). With view to future applications it is convenient to differentiate between symmetric and antisymmetric bending rigidities R_s and R_a , which are associated with the responses to deformation modes L_3 and L_4 , respectively. Assuming R_s and R_a to be uniform along the element we get

$$\mathbf{K}_q = \mathbf{K}_{qs} + \mathbf{K}_{qa}, \quad \mathbf{K}_{qs} = \frac{144R_s}{\ell^3} \begin{bmatrix} 0 & 0 & 0 & 0 \\ 0 & 0 & 0 & 0 \\ 0 & 0 & 1 & 0 \\ 0 & 0 & 0 & 0 \end{bmatrix}, \quad \mathbf{K}_{qa} = \frac{1200R_a}{\ell^3} \begin{bmatrix} 0 & 0 & 0 & 0 \\ 0 & 0 & 0 & 0 \\ 0 & 0 & 0 & 0 \\ 0 & 0 & 0 & 1 \end{bmatrix}. \quad (13.27)$$

For a Bernoulli-Euler model, the generalized coordinates q_i of (13.23) can be connected to the physical DOFs by the transformations

$$\begin{bmatrix} v_i \\ \theta_i \\ v_j \\ \theta_j \end{bmatrix} = \begin{bmatrix} 1 & -1 & 1 & -1 \\ 0 & 2/\ell & -6/\ell & 12/\ell \\ 1 & 1 & 1 & 1 \\ 0 & 2/\ell & 6/\ell & 12/\ell \end{bmatrix} \begin{bmatrix} q_1 \\ q_2 \\ q_3 \\ q_4 \end{bmatrix} = \mathbf{G}\mathbf{q}, \quad \begin{bmatrix} q_1 \\ q_2 \\ q_3 \\ q_4 \end{bmatrix} = \frac{1}{60} \begin{bmatrix} 30 & 5\ell & 30 & -5\ell \\ -36 & -3\ell & 36 & -3\ell \\ 0 & -5\ell & 0 & -5\ell \\ 6 & 3\ell & -6 & 3\ell \end{bmatrix} \begin{bmatrix} v_i \\ \theta_i \\ v_j \\ \theta_j \end{bmatrix} = \mathbf{H}\mathbf{u}^{(e)}. \quad (13.28)$$

in which $\mathbf{H} = \mathbf{G}^{-1}$. The physical stiffness is

$$\mathbf{K}^{(e)} = \mathbf{H}^T \mathbf{K}_q \mathbf{H} = \frac{1}{\ell^3} \begin{bmatrix} 12R_a & 6R_a\ell & -12R_a & 6R_a\ell \\ 6R_a\ell & (3R_a + R_s)\ell^2 & -6R_a\ell & (3R_a - R_s)\ell^2 \\ -12R_a & -6R_a\ell & 12R_a & -6R_a\ell \\ 6R_a\ell & (3R_a - R_s)\ell^2 & -6R_a\ell & (3R_a + R_s)\ell^2 \end{bmatrix} \quad (13.29)$$

If $R_s = R_a = EI$ the well known stiffness matrix (13.20) is recovered, as can be expected. The additional flexibility conferred by (13.29) is exhibited below in two advanced applications.

§13.7.2. *Hinged Plane Beam Element

The two-node prismatic plane beam element depicted in Figure 13.13 has a mechanical hinge at midspan ($\xi = 0$). The cross sections on both sides of the hinge can rotate respect to each other. The figure also sketches a fabrication method sometimes used in short-span pedestrian bridges. Gaps on either side of the section cuts are filled with bituminous material that permits slow rotations.

It follows that both the curvature κ and the bending moment M must vanish at midspan. But in a element described by a cubic interpolation of $v(x)$ the curvature must vary linearly in both ξ and x . Consequently the mean curvature, which is controlled by Legendre function L_2 (see blue lines in Figure 13.12) must be zero.

The kinematic constraint of zero mean curvature can be enforced by setting the symmetric rigidity $R_s = 0$ whereas the antisymmetric rigidity is the normal one: $R_a = EI$. Inserting these into (13.29) immediately

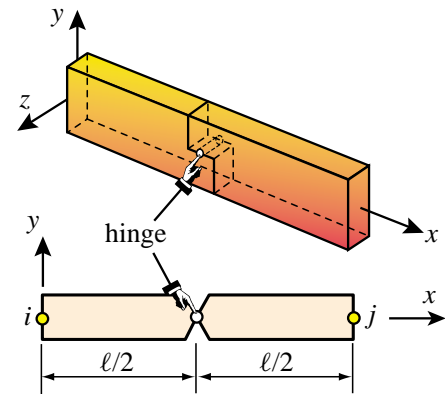


Figure 13.13. Beam element with hinge located at midspan. The upper figure sketches a hinge fabrication method.

gives

$$\mathbf{K}^{(e)} = \frac{EI}{\ell^3} \begin{bmatrix} 12 & 6\ell & -12 & 6 \\ 6\ell & 3\ell^2 & -6 & 3\ell^2 \\ -12 & -6\ell & 12 & -6\ell \\ 6\ell & 3\ell^2 & -6\ell & 3\ell^2 \end{bmatrix} \quad (13.30)$$

The eigenvalues of this matrix are

$$\left[0 \quad 0 \quad 0 \quad \frac{6EI(4 + \ell^2)}{\ell^3} \right] \quad (13.31)$$

It has rank one, as it should be. The eigenvector associated with the only nonzero eigenvalue pertains to the antisymmetric deformational mode L_3 . Matrix (13.30) can be derived by more sophisticated methods, but the present technique is the most expedient one.

§13.7.3. *Timoshenko Plane Beam Element

As noted in §13.2, the Timoshenko model includes a first order correction for transverse shear. The kinematic assumption of classical beam theory is changed to “plane sections remain plane but not necessarily normal to the deformed neutral surface.” This is depicted in Figure 13.14 for a plane beam element. The cross section rotation is still called θ , positive CCW. Ignoring axial forces the displacement field is analogous to (13.2):

$$u(x, y) = -y\theta, \quad v(x, y) = v(x) \quad (13.32)$$

But now

$$\theta = \frac{\partial v}{\partial x} - \gamma = v' - \gamma, \quad \gamma = \frac{V}{GA_s}. \quad (13.33)$$

Here V is the transverse shear force, $\gamma = \gamma(x)$ the “shear rotation” averaged over the cross section, G the shear modulus and A_s the effective shear area.⁴

The negative sign of γ in $\theta = v' - \gamma$ comes from the sign convention for V commonly used in Mechanics of Materials and employed here: a positive V produces a CW shear rotation. See Figure 13.14. For convenience introduce here the dimensionless factor $\Phi = 12EI/(GA_s\ell^2)$. This characterizes the “shear slenderness” of the beam element. As $\Phi \rightarrow 0$ the Timoshenko model reduces to the Bernoulli-Euler one.

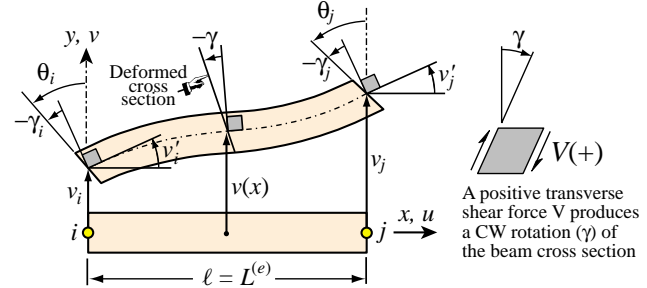


Figure 13.14. A two-node Timoshenko beam element.

From equilibrium: $V = M' = EI v'''$ and $\gamma = V/(GA_s) = EI v'''/(GA_s) = v''' \ell^2/(12\Phi)$. From the Legendre interpolation (13.23), $v''' = 120q_4/\ell^3$ and $\gamma = (10\Phi/\ell) q_4$, which is constant over the element. The element internal energy is now

$$U^{(e)} = \frac{1}{2} \int_0^\ell (EI \kappa^2 + GA_s \gamma^2) dx. \quad (13.34)$$

The remaining derivations involve lengthy algebra and are relegated to Exercises 13.11ff. The final result is that the element stiffness fits the form (13.29) in which $R_s = EI$ and $R_a = EI/(1 + \Phi)$. This gives

$$\mathbf{K}^{(e)} = \frac{EI}{\ell^3(1 + \Phi)} \begin{bmatrix} 12 & 6\ell & -12 & 6\ell \\ 6\ell & \ell^2(4 + \Phi) & -6\ell & \ell^2(2 - \Phi) \\ -12 & -6\ell & 12 & -6\ell \\ 6\ell & \ell^2(2 - \Phi) & -6\ell & \ell^2(4 + \Phi) \end{bmatrix} \quad (13.35)$$

If $\Phi = 0$ this reduces to the Bernoulli-Euler beam stiffness, as can be expected.

⁴ A concept defined in Mechanics of Materials; see e.g. Chapter 10 of Popov [13.11].

§13.8. *ACCURACY ANALYSIS

This section presents the accuracy analysis of a repeating lattice of beam elements, analogous to that done for the bar element in §12.5. The analysis uses the method of modified differential equations (MoDE) mentioned in the **Notes and Bibliography** of Chapter 12. It is performed using a repeated differentiations scheme similar to that used in solving Exercise 12.8. Only the case of the Bernoulli-Euler model is worked out in detail.

§13.8.1. *Accuracy of Bernoulli-Euler Beam Element

Consider a lattice of repeating two-node Bernoulli-Euler prismatic plane beam elements of rigidity EI and length ℓ , as illustrated in Figure 13.15. The system is subject to an arbitrary lateral load $q(x)$, which is assumed infinitely differentiable in x . From the lattice extract a patch of two elements: (1) and (2), connecting nodes i – j and j – k , respectively, as shown in Figure 13.15.

The FEM patch equations at node j are

$$\frac{EI}{\ell^3} \begin{bmatrix} -12 & -6\ell & 24 & 0 & -12 & 6\ell \\ 6\ell & 2\ell^2 & 0 & 8\ell^2 & -6\ell & 2\ell^2 \end{bmatrix} \begin{bmatrix} v_i \\ \theta_i \\ v_j \\ \theta_j \\ v_k \\ \theta_k \end{bmatrix} = \begin{bmatrix} f_j \\ m_j \end{bmatrix} \quad (13.36)$$

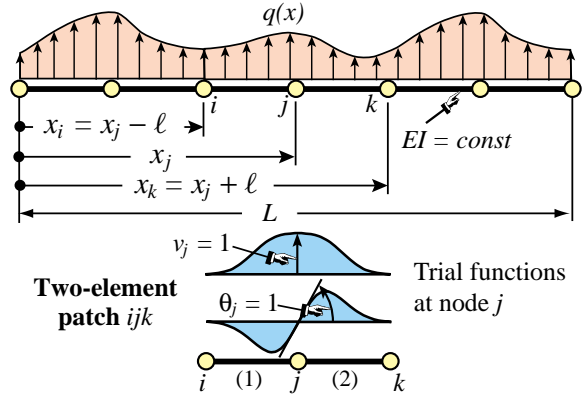


Figure 13.15. Repeating beam lattice for accuracy analysis.

Expand $v(x)$, $\theta(x)$ and $q(x)$ in Taylor series about $x = x_j$, truncating at $n+1$ terms for v and θ , and $m+1$ terms for q . Using $\psi = (x - x_j)/\ell$, the series are $v(x) = v_j + \psi\ell v'_j + (\psi^2\ell^2/2!)v''_j + \dots + (\psi^n\ell^n/n!)v_j^{[n]}$, $\theta(x) = \theta_j + \psi\ell\theta'_j + (\psi^2\ell^2/2!)\theta''_j + \dots + (\psi^n\ell^n/n!)\theta_j^{[n]}$, and $q(x) = q_j + \psi\ell q'_j + (\psi^2\ell^2/2!)q''_j + \dots + (\psi^m\ell^m/m!)q_j^{[m]}$. Here $v_j^{[n]}$, etc., is an abbreviation for $d^n v(x_j)/dx^n$.⁵ Evaluate the $v(x)$ and $\theta(x)$ series at i and k by setting $\psi = \pm 1$, and insert in (13.36). Use the $q(x)$ series evaluated over elements (1) and (2), to compute the consistent forces f_j and m_j as

$$f_j = \frac{\ell}{2} \left(\int_{-1}^1 N_3^{(1)} q^{(1)} d\xi^{(1)} + \int_{-1}^1 N_1^{(2)} q^{(2)} d\xi^{(2)} \right), \quad m_j = \frac{\ell}{2} \left(\int_{-1}^1 N_4^{(1)} q^{(1)} d\xi^{(1)} + \int_{-1}^1 N_2^{(2)} q^{(2)} d\xi^{(2)} \right). \quad (13.37)$$

Here $N_3^{(1)} = \frac{1}{4}(1 + \xi^{(1)})^2(2 - \xi^{(1)})$, $N_4^{(1)} = -\frac{\ell}{8}(1 + \xi^{(1)})^2(1 - \xi^{(1)})$, $N_1^{(2)} = \frac{1}{4}(1 - \xi^{(2)})^2(2 + \xi^{(2)})$, and $N_2^{(2)} = \frac{\ell}{8}(1 - \xi^{(2)})^2(1 + \xi^{(2)})$ are the Hermitian shape functions components of the j node trial function, whereas $q^{(1)} = q(\psi^{(1)})$, $\psi^{(1)} = -\frac{1}{2}(1 - \xi^{(1)})$ and $q^{(2)} = q(\psi^{(2)})$, $\psi^{(2)} = \frac{1}{2}(1 + \xi^{(2)})$ denote the lateral loads. To show the resulting system in compact matrix form it is convenient to collect the derivatives at node j into vectors:

$$\mathbf{v}_j = [v_j \ v'_j \ v''_j \ \dots \ v_j^{[n]}]^T, \quad \boldsymbol{\theta}_j = [\theta_j \ \theta'_j \ \theta''_j \ \dots \ \theta_j^{[n]}]^T, \quad \mathbf{q}_j = [q_j \ q'_j \ q''_j \ \dots \ q_j^{[n]}]^T. \quad (13.38)$$

The resulting differential system can be compactly written

$$\begin{bmatrix} \mathbf{S}_{vv} & \mathbf{S}_{v\theta} \\ \mathbf{S}_{\theta v} & \mathbf{S}_{\theta\theta} \end{bmatrix} \begin{bmatrix} \mathbf{v}_j \\ \boldsymbol{\theta}_j \end{bmatrix} = \begin{bmatrix} \mathbf{P}_v \\ \mathbf{P}_\theta \end{bmatrix} \mathbf{q}_j \quad (13.39)$$

⁵ Brackets are used instead of parentheses to avoid confusion with element superscripts. If derivatives are indexed by primes or roman numerals the brackets are omitted.

here \mathbf{S}_{vv} , $\mathbf{S}_{v\theta}$, $\mathbf{S}_{\theta v}$ and $\mathbf{S}_{\theta\theta}$ are triangular Toeplitz matrices of order $(n+1) \times (n+1)$ whereas \mathbf{P}_v and \mathbf{P}_θ are generally rectangular matrices of order $(n+1) \times (m+1)$. Here is the expression of these matrices for $n = 8$, $m = 4$:

$$\begin{aligned} \mathbf{S}_{vv} = EI & \begin{bmatrix} 0 & 0 & \frac{-12}{\ell} & 0 & -\ell & 0 & \frac{-\ell^3}{30} & 0 & \frac{-\ell^5}{1680} \\ 0 & 0 & 0 & \frac{-12}{\ell} & 0 & -\ell & 0 & \frac{-\ell^3}{30} & 0 \\ 0 & 0 & 0 & 0 & \frac{-12}{\ell} & 0 & -\ell & 0 & \frac{-\ell^3}{30} \\ 0 & 0 & 0 & 0 & 0 & \frac{-12}{\ell} & 0 & -\ell & 0 \\ 0 & 0 & 0 & 0 & 0 & 0 & \frac{-12}{\ell} & 0 & -\ell \\ 0 & 0 & 0 & 0 & 0 & 0 & 0 & \frac{-12}{\ell} & 0 \\ 0 & 0 & 0 & 0 & 0 & 0 & 0 & 0 & \frac{-12}{\ell} \\ 0 & 0 & 0 & 0 & 0 & 0 & 0 & 0 & 0 \\ 0 & 0 & 0 & 0 & 0 & 0 & 0 & 0 & 0 \end{bmatrix}, & \mathbf{S}_{v\theta} = EI & \begin{bmatrix} 0 & \frac{12}{\ell} & 0 & 2\ell & 0 & \frac{\ell^3}{10} & 0 & \frac{\ell^5}{420} & 0 \\ 0 & 0 & \frac{12}{\ell} & 0 & 2\ell & 0 & \frac{\ell^3}{10} & 0 & \frac{\ell^5}{420} \\ 0 & 0 & 0 & \frac{12}{\ell} & 0 & 2\ell & 0 & \frac{\ell^3}{10} & 0 \\ 0 & 0 & 0 & 0 & \frac{12}{\ell} & 0 & 2\ell & 0 & \frac{\ell^3}{10} \\ 0 & 0 & 0 & 0 & 0 & \frac{12}{\ell} & 0 & 2\ell & 0 \\ 0 & 0 & 0 & 0 & 0 & 0 & \frac{12}{\ell} & 0 & 2\ell \\ 0 & 0 & 0 & 0 & 0 & 0 & 0 & \frac{12}{\ell} & 0 \\ 0 & 0 & 0 & 0 & 0 & 0 & 0 & 0 & \frac{12}{\ell} \\ 0 & 0 & 0 & 0 & 0 & 0 & 0 & 0 & 0 \end{bmatrix} \\ \mathbf{S}_{\theta v} = EI & \begin{bmatrix} 0 & \frac{-12}{\ell} & 0 & -2\ell & 0 & \frac{-\ell^3}{10} & 0 & \frac{-\ell^5}{420} & 0 \\ 0 & 0 & \frac{-12}{\ell} & 0 & -2\ell & 0 & \frac{-\ell^3}{10} & 0 & \frac{-\ell^5}{420} \\ 0 & 0 & 0 & \frac{-12}{\ell} & 0 & -2\ell & 0 & \frac{-\ell^3}{10} & 0 \\ 0 & 0 & 0 & 0 & \frac{-12}{\ell} & 0 & -2\ell & 0 & \frac{-\ell^3}{10} \\ 0 & 0 & 0 & 0 & 0 & \frac{-12}{\ell} & 0 & -2\ell & 0 \\ 0 & 0 & 0 & 0 & 0 & 0 & \frac{-12}{\ell} & 0 & -2\ell \\ 0 & 0 & 0 & 0 & 0 & 0 & 0 & \frac{-12}{\ell} & 0 \\ 0 & 0 & 0 & 0 & 0 & 0 & 0 & 0 & \frac{-12}{\ell} \\ 0 & 0 & 0 & 0 & 0 & 0 & 0 & 0 & 0 \end{bmatrix}, & \mathbf{S}_{\theta\theta} = EI & \begin{bmatrix} \frac{12}{\ell} & 0 & 2\ell & 0 & \frac{\ell^3}{6} & 0 & \frac{\ell^5}{180} & 0 & \frac{\ell^7}{10080} \\ 0 & \frac{12}{\ell} & 0 & 2\ell & 0 & \frac{\ell^3}{6} & 0 & \frac{\ell^5}{180} & 0 \\ 0 & 0 & \frac{12}{\ell} & 0 & 2\ell & 0 & \frac{\ell^3}{6} & 0 & \frac{\ell^5}{180} \\ 0 & 0 & 0 & \frac{12}{\ell} & 0 & 2\ell & 0 & \frac{\ell^3}{6} & 0 \\ 0 & 0 & 0 & 0 & \frac{12}{\ell} & 0 & 2\ell & 0 & \frac{\ell^3}{6} \\ 0 & 0 & 0 & 0 & 0 & \frac{12}{\ell} & 0 & 2\ell & 0 \\ 0 & 0 & 0 & 0 & 0 & 0 & \frac{12}{\ell} & 0 & 2\ell \\ 0 & 0 & 0 & 0 & 0 & 0 & 0 & \frac{12}{\ell} & 0 \\ 0 & 0 & 0 & 0 & 0 & 0 & 0 & 0 & \frac{12}{\ell} \end{bmatrix} \end{aligned} \quad (13.40)$$

$$\mathbf{P}_v = \begin{bmatrix} \ell & 0 & \frac{\ell^3}{15} & 0 & \frac{\ell^5}{560} \\ 0 & \ell & 0 & \frac{\ell^3}{15} & 0 \\ 0 & 0 & \ell & 0 & \frac{\ell^3}{15} \\ 0 & 0 & 0 & \ell & 0 \\ 0 & 0 & 0 & 0 & \ell \\ 0 & 0 & 0 & 0 & 0 \\ 0 & 0 & 0 & 0 & 0 \\ 0 & 0 & 0 & 0 & 0 \\ 0 & 0 & 0 & 0 & 0 \end{bmatrix}, \quad \mathbf{P}_\theta = \begin{bmatrix} 0 & \frac{\ell^3}{15} & 0 & \frac{\ell^5}{315} & 0 \\ 0 & 0 & \frac{\ell^3}{15} & 0 & \frac{\ell^5}{315} \\ 0 & 0 & 0 & \frac{\ell^3}{15} & 0 \\ 0 & 0 & 0 & 0 & \frac{\ell^3}{15} \\ 0 & 0 & 0 & 0 & 0 \\ 0 & 0 & 0 & 0 & 0 \\ 0 & 0 & 0 & 0 & 0 \\ 0 & 0 & 0 & 0 & 0 \\ 0 & 0 & 0 & 0 & 0 \end{bmatrix} \quad (13.41)$$

Elimination of θ_j gives the condensed system

$$\mathbf{S}\mathbf{v}_j = \mathbf{P}\mathbf{q}_j, \quad \text{with} \quad \mathbf{S} = \mathbf{S}_{vv} - \mathbf{S}_{v\theta}\mathbf{S}_{\theta\theta}^{-1}\mathbf{S}_{\theta v}, \quad \mathbf{P} = \mathbf{P}_v - \mathbf{S}_{v\theta}\mathbf{S}_{\theta\theta}^{-1}\mathbf{P}_\theta. \quad (13.42)$$

$$\mathbf{S} = \mathbf{S}_{vv} - \mathbf{S}_{v\theta}\mathbf{S}_{\theta\theta}^{-1}\mathbf{S}_{\theta v} = EI \begin{bmatrix} 0 & 0 & 0 & 0 & \ell & 0 & 0 & 0 & \frac{-\ell^5}{720} \\ 0 & 0 & 0 & 0 & 0 & \ell & 0 & 0 & 0 \\ 0 & 0 & 0 & 0 & 0 & 0 & \ell & 0 & 0 \\ 0 & 0 & 0 & 0 & 0 & 0 & 0 & \ell & 0 \\ 0 & 0 & 0 & 0 & 0 & 0 & 0 & 0 & \ell \\ 0 & 0 & 0 & 0 & 0 & 0 & 0 & 0 & 0 \\ 0 & 0 & 0 & 0 & 0 & 0 & 0 & 0 & 0 \\ 0 & 0 & 0 & 0 & 0 & 0 & 0 & 0 & 0 \\ 0 & 0 & 0 & 0 & 0 & 0 & 0 & 0 & 0 \end{bmatrix}, \quad \mathbf{P} = \mathbf{P}_v - \mathbf{S}_{v\theta}\mathbf{S}_{\theta\theta}^{-1}\mathbf{P}_\theta = \begin{bmatrix} \ell & 0 & 0 & 0 & \frac{-\ell^5}{720} \\ 0 & \ell & 0 & 0 & 0 \\ 0 & 0 & \ell & 0 & 0 \\ 0 & 0 & 0 & \ell & 0 \\ 0 & 0 & 0 & 0 & \ell \\ 0 & 0 & 0 & 0 & 0 \\ 0 & 0 & 0 & 0 & 0 \\ 0 & 0 & 0 & 0 & 0 \\ 0 & 0 & 0 & 0 & 0 \end{bmatrix} \quad (13.43)$$

The nontrivial differential relations⁶ given by $\mathbf{S}\mathbf{v}_j = \mathbf{P}\mathbf{q}_j$ are $EI(v_j^{iv} - \ell^4 v_j^{viii}/720) = q_j - \ell^4 q_j^{iv}/720$, $EI v_j^v = q_j'$, $EI v_j^{vi} = q_j''$, $EI v_j^{vii} = q_j'''$, and $EI v_j^{viii} = q_j^{iv}$. The first one is a truncation of the infinite order MoDE. Elimination of all v_j derivatives but v_j^{iv} yields

$$EI v_j^{iv} = q_j, \quad (13.44)$$

exactly. That this is not a fluke can be confirmed by increasing n and while taking $m = n - 4$. The first 4 columns and last 4 rows of \mathbf{S} , which are always zero, are removed to get $\hat{\mathbf{S}}$. The last 4 rows of \mathbf{P} , which are also zero, are removed to get $\hat{\mathbf{P}}$. With $m = n-4$ both matrices are of order $n-3 \times n-3$. Then $\hat{\mathbf{S}} = \hat{\mathbf{P}}$ for any n , which leads immediately to (13.44). This was confirmed with *Mathematica* running n up to 24.

The foregoing analysis shows that the BE cubic element is *nodally exact for any smooth load* over a repeating lattice if consistent load computation is used. Exercise 13.18 verifies that the property is lost if elements are not identical. Numerical experiments confirm these conclusions. The Laplace transform method only works part way: it gives a different infinite order MoDE but recovers (13.44) as $n \rightarrow \infty$.

These accuracy properties are not widely known. If all beam elements are prismatic and subjected only to point loads at nodes, overall exactness follows from a theorem by Tong [13.15], which is not surprising since the exact solution is contained in the FEM approximation. For general distributed loads the widespread belief is that the cubic element incurs $O(\ell^4)$ errors. The first study of this nature by Waltz et. al. [13.16] gave the modified differential equation (MoDE) for a uniform load q as

$$v^{iv} - \frac{\ell^4}{720} v^{viii} + \dots = \frac{q}{EI}. \quad (13.45)$$

The terms shown are correct. In fact a more complete expression, obtained in this study, is

$$EI \left(v^{iv} - \frac{\ell^4}{720} v^{viii} + \frac{\ell^6}{3024} v^{x} - \frac{7\ell^8}{259200} v^{xii} + \dots \right) = q - \frac{\ell^4}{720} q^{iv} + \frac{\ell^6}{3024} q^{vi} - \frac{7\ell^8}{259200} q^{viii} + \dots \quad (13.46)$$

But the conclusion that “the principal error term” is of order ℓ^4 [13.16, p. 1009] is incorrect. The misinterpretation is due to (13.46) being an ODE of infinite order. Truncation is fine *if followed by elimination of higher derivatives*. If this is done, the finite order MoDE (13.44) emerges regardless of where (13.46) is truncated; an obvious clue being the repetition of coefficients in both sides. The moral is that conclusions based on infinite order ODEs should be viewed with caution, unless corroborated by independent means.

§13.8.2. *Accuracy of Timoshenko Beam Element

Following the same procedure it can be shown that the infinite order MoDE for a Timoshenko beam element repeating lattice with the stiffness (13.35) and consistent node forces is

$$\begin{aligned} EI \left(v_j^{iv} + \frac{\ell^2 \Phi}{12} v_j^{vi} - \frac{\ell^4 (1 + 5\Phi - 5\Phi^2)}{720} v_j^{viii} + \frac{\ell^6 (20 + 7\Phi - 70\Phi^2 + 35\Phi^3)}{60480} v_j^x + \dots \right) \\ = q_j - \frac{\ell^4 (1 + 5\Phi)}{720} q_j^{iv} + \frac{\ell^6 (20 + 14\Phi - 35\Phi^2)}{60480} q_j^{vi} + \dots, \quad \text{in which } \Phi = 12EI/(GA_s \ell^2). \end{aligned} \quad (13.47)$$

This reduces to (13.46) if $\Phi = 0$. Elimination of higher order derivatives gives the finite order MoDE (aka FOMoDE)

$$EI v_j^{iv} = q_j - \frac{\ell^2 \Phi}{12} q_j'' = q_j - \frac{EI}{GA_s} q_j''. \quad (13.48)$$

⁶ Derivatives of order 4 and higher are indicated by Roman numeral superscripts instead of primes.

which repeats for any $n > 8$. This happens to be the exact governing differential equation for a statically loaded Timoshenko beam [13.7, p. 23]. Consequently the Timoshenko beam is *nodally exact* under the same conditions previously stated for the Bernoulli-Euler model.

Notes and Bibliography

A comprehensive source of stiffness and mass matrices of plane and spatial beams is the book by Przemieniecki [13.12]. The derivation of stiffness matrices there is carried out using differential equations rather than energy methods. This was in fact the common practice before 1963 [13.10]. Results for prismatic elements, however, are identical. Energy derivations were popularized by Archer [13.2,13.3] and Martin [13.9].

The Legendre interpolation (13.23) to construct optimal mass-stiffness combinations for beam elements is further studied in [13.5,13.6] in the context of finite element templates [13.4]. The diagonal covariance matrix \mathbf{Q} given in (13.25) plays a key role in the customization of the mass matrix. The hinged element stiffness (13.30) is rederived in the Advanced FEM Lecture Notes using a mixed variational principle.

The Timoshenko beam model was originally proposed in [13.14]. It has become important as a model for transient response and control simulations because its dynamic form is strictly hyperbolic.⁷ The equilibrium based form of the Timoshenko beam element (13.35) is derived in Section 5.6 of [13.12] using differential equations. This form is optimal in that it solves nodally-loaded discretizations exactly. This beam model belongs to the class of “ C^0 elements”, which have been extensively studied in the FEM literature after 1968. The book of Hughes [13.8] provides a comprehensive treatment for beams and plates.

As the study in §13.8 illustrates, modified equation methods in boundary value problems⁸ are delicate and should be used with care. Their intricacies baffled Strang and Fix who, upon doing Fourier analysis of a cubic beam element, incorrectly stated [13.13, p. 171] that only one of the discrete equations — that for $v(x)$ — is consistent “and the others are completely inconsistent.” The alternative is the variational approach to error analysis. Although more robust and forgiving, predictions are often so conservative as to be of little value.

References

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⁷ The Bernoulli-Euler beam dynamic model is parabolic and thus exhibits an infinite transverse wave speed. Such a model is unsuitable for wave propagation problems.

⁸ They are more forgiving in initial value problems.

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Homework Exercises for Chapter 13

Variational Formulation of Plane Beam Element

EXERCISE 13.1

[A/C:20] Use (13.17) to derive the element stiffness matrix $\mathbf{K}^{(e)}$ of a Hermitian beam element of variable bending rigidity given by the inertia law

$$I(x) = I_i \left(1 - \frac{x}{\ell}\right) + I_j \frac{x}{\ell} = I_i \frac{1}{2}(1 - \xi) + I_j \frac{1}{2}(1 + \xi). \quad (\text{E13.1})$$

Use of *Mathematica* or similar CAS tool is recommended since the integrals are time consuming. *Mathematica* hint: write

$$\text{EI} = \text{EIi}*(1-\xi)/2 + \text{EIj}*(1+\xi)/2; \quad (\text{E13.2})$$

and keep EI inside the argument of `Integrate`. Check whether you get back (13.20) if $\text{EI}=\text{EIi}=\text{EIj}$. If you use *Mathematica*, this check can be simply done after you got and printed the tapered beam Ke , by writing `ClearAll[EI]; Ke=Simplify[Ke/.{EIi->EI,EIj->EI}];` and printing this matrix.⁹

EXERCISE 13.2

[A/C:20] Use (13.18) to derive the consistent node force vector $\mathbf{f}^{(e)}$ for a Hermitian beam element under linearly varying transverse load q defined by

$$q(x) = q_i \left(1 - \frac{x}{\ell}\right) + q_j \frac{x}{\ell} = q_i \frac{1}{2}(1 - \xi) + q_j \frac{1}{2}(1 + \xi). \quad (\text{E13.3})$$

Again use of a CAS is recommended, particularly since the polynomials to be integrated are quartic in ξ , and hand computations are error prone. *Mathematica* hint: write

$$q = qi*(1-\xi)/2 + qj*(1+\xi)/2; \quad (\text{E13.4})$$

and keep q inside the argument of `Integrate`. Check whether you get back (13.21) if $q_i = q_j = q$ (See previous Exercise for *Mathematica* procedure).

EXERCISE 13.3

[A:20] Obtain the consistent node force vector $\mathbf{f}^{(e)}$ of a Hermitian beam element subject to a transverse point load P at abscissa $x = a$ where $0 \leq a \leq \ell$. Use the Dirac's delta function expression $q(x) = P \delta(a)$ and the fact that for any continuous function $f(x)$, $\int_0^\ell f(x) \delta(a) dx = f(a)$ if $0 \leq a \leq \ell$.

EXERCISE 13.4

[A:25] Derive the consistent node force vector $\mathbf{f}^{(e)}$ of a Hermitian beam element subject to a uniformly distributed z -moment m per unit length. Use the fact that the external work per unit length is $m(x)\theta(x) = m(x) v'(x) = (\mathbf{u}^{(e)})^T (d\mathbf{N}/dx)^T m(x)$. For arbitrary $m(x)$ show that this gives

$$\mathbf{f}^{(e)} = \int_0^\ell \frac{\partial \mathbf{N}^T}{\partial x} m dx = \int_{-1}^1 \frac{\partial \mathbf{N}^T}{\partial \xi} \frac{2}{\ell} m \frac{1}{2} \ell d\xi = \int_{-1}^1 \mathbf{N}_\xi^T m d\xi, \quad (\text{E13.5})$$

where \mathbf{N}_ξ^T denote the column vectors of beam shape function derivatives with respect to ξ . Can you see a shortcut that avoids the integral for constant m ?

⁹ `ClearAll[EI]` discards the previous definition (E13.2) of EI; the same effect can be achieved by writing `EI=. (dot)`.

EXERCISE 13.5

[A:20] Obtain the consistent node force vector $\mathbf{f}^{(e)}$ of a Hermitian beam element subject to a concentrated moment C applied at $x = a$. Use the expression (E13.5) in which $m(x) = C \delta(a)$, where $\delta(a)$ denotes the Dirac's delta function at $x = a$.

EXERCISE 13.6

[A/C:25] Consider the one-dimensional Gauss integration rules.¹⁰

$$\text{One point : } \int_{-1}^1 f(\xi) d\xi \doteq 2f(0) \quad (\text{E13.6})$$

$$\text{Two points: } \int_{-1}^1 f(\xi) d\xi \doteq f(-1/\sqrt{3}) + f(1/\sqrt{3}) \quad (\text{E13.7})$$

$$\text{Three points: } \int_{-1}^1 f(\xi) d\xi \doteq \frac{5}{9}f(-\sqrt{3/5}) + \frac{8}{9}f(0) + \frac{5}{9}f(\sqrt{3/5}) \quad (\text{E13.8})$$

Try each rule on the monomial integrals

$$\int_{-1}^1 d\xi, \quad \int_{-1}^1 \xi d\xi, \quad \int_{-1}^1 \xi^2 d\xi, \quad \dots \quad (\text{E13.9})$$

until the rule fails. In this way verify that the rules (E13.6), (E13.7) and (E13.8) are exact for polynomials of degree up to 1, 3 and 5, respectively. (*Labor-saving hint*: for odd monomial degree no computations need to be done; why?).

EXERCISE 13.7

[A/C:25] Repeat the derivation of Exercise 13.1 using the two-point Gauss rule (E13.7) to evaluate integrals in ξ . A CAS is recommended. If using *Mathematica* you may use a function definition to save typing; for example to evaluate $\int_{-1}^1 f(\xi) d\xi$ in which $f(\xi) = 6\xi^4 - 3\xi^2 + 7$, by the 3-point Gauss rule (E13.8), say

```
f[ξ_]:=6ξ^4-3ξ^2+7; int=Simplify[(5/9)*(f[-Sqrt[3/5]]+f[Sqrt[3/5]])+(8/9)*f[0]];
```

and print int. To form an element by Gauss integration define matrix functions in terms of ξ , for example $\text{Be}[\xi_]$, or use the substitution operator $/.$, whatever you prefer. Check whether one obtains the same answers as with analytical integration, and explain why there is agreement or disagreement. Hint for the explanation: consider the order of the ξ polynomials you are integrating over the element.

EXERCISE 13.8

[A/C:25] As above but for Exercise 13.2.

EXERCISE 13.9

[A/C:25=15+10] Consider a cantilever beam of constant rigidity EI and length L , which is modeled with one element. The beam is end loaded by a transverse force P .

- Find the displacement v_j and the rotation θ_j in terms of P , E , I and L . Compare with the analytical values $PL^3/(3EI)$ and $PL^2/(2EI)$.
- Why does the finite element model provides the exact answer with one element?

¹⁰ Gauss integration is studied further in Chapter 17.

EXERCISE 13.10

[A/C:30] Derive the classical beam stiffness matrix (13.20) using the method of differential equations. To do this integrate the homogeneous differential equation $EI v'''' = 0$ four times over a cantilever beam clamped at node i over $x \in [0, \ell]$ to get $v(x)$. The process yields four constants of integration C_1 through C_4 , which are determined by matching the two zero-displacement BCs at node i and the two force BCs at node j . This provides a 2×2 flexibility matrix relating forces and displacements at node j . Invert to get a deformational stiffness, and expand to 4×4 by letting node i translate and rotate.

EXERCISE 13.11

[A:15] Evaluate the strain and stress fields associated to the Timoshenko beam displacement field (13.32)-(13.33).

EXERCISE 13.12

[A/C:30] (Tricky) Carry out the complete derivation of the Timoshenko beam element stiffness (13.35).

EXERCISE 13.13

[A/C:20] Find the consistent node forces for a Timoshenko beam element under a uniform transverse load q . (Solution of the previous exercise is useful.)

EXERCISE 13.14

[A:25] (Requires math ability). Discuss what happens in (13.35) if $\Phi \rightarrow \infty$. Is the result useful for a shear-only “spar” element? Hint: eliminate θ_i and θ_j by a master-slave MFC.

EXERCISE 13.15

[C:30] Write *Mathematica* code to verify the nodal exactness conclusion of §13.8.1 using the repeated differentiation approach.

EXERCISE 13.16

[C:30] As above, but using the Laplace transform. Show that this only does half the job.

EXERCISE 13.17

[A/C:35] Find the general symbolic expression of the terms in the infinite order MoDE (13.46).

EXERCISE 13.18

[A/C:40] Analyze nodal accuracy if the length of the beam elements in the lattice of Figure 13.15 alternates between $(1 \pm \alpha)\ell$, where $0 \leq \alpha \leq \frac{1}{2}$.

EXERCISE 13.19

[A/C:40] Using *Mathematica*, verify the results (13.47) and (13.48) in §13.8.2.

14

The Plane Stress Problem

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§14.1. INTRODUCTION

We now pass to the variational formulation of two-dimensional *continuum* finite elements. The problem of plane stress will serve as the vehicle for illustrating such formulations. As narrated in §1.7.1, continuum finite elements were invented in the aircraft industry (at Boeing, early 1950s) to solve this kind of problem when it arose in the design and analysis of delta wing panels.

The problem is presented here within the framework of the linear theory of elasticity.

§14.1.1. Plate in Plane Stress

In structural mechanics, a flat thin sheet of material is called a *plate*.¹ The distance between the plate faces is called the *thickness* and denoted by h . The *midplane* lies halfway between the two faces.

The direction normal to the midplane is called the *transverse* direction. Directions parallel to the midplane are called *in-plane* directions. The global axis z will be oriented along the transverse direction. Axes x and y are placed in the midplane, forming a right-handed Rectangular Cartesian Coordinate (RCC) system. Thus the midplane equation is $z = 0$. See Figure 14.1.

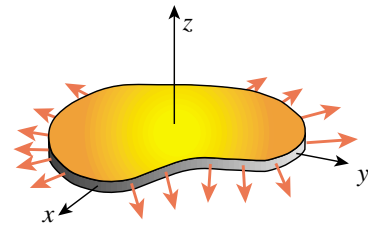


Figure 14.1. A plate structure in plane stress.

A plate loaded in its midplane is said to be in a state of *plane stress*, or a *membrane state*, if the following assumptions hold:

1. All loads applied to the plate act in the midplane direction, and are symmetric with respect to the midplane.
2. All support conditions are symmetric about the midplane.
3. In-plane displacements, strains and stresses can be taken to be uniform through the thickness.
4. The normal and shear stress components in the z direction are zero or negligible.

The last two assumptions are not necessarily consequences of the first two. For the latter to hold, the thickness h should be small, typically 10% or less, than the shortest in-plane dimension. If the plate thickness varies it should do so gradually. Finally, the plate fabrication must exhibit symmetry with respect to the midplane.

To these four assumptions we add the following restriction:

5. The plate is fabricated of the same material through the thickness. Such plates are called *transversely homogeneous* or *monocoque* plates.

The last assumption excludes wall constructions of importance in aerospace, in particular composite and honeycomb sandwich plates. The development of mathematical models for such configurations requires a more complicated integration over the thickness as well as the ability to handle coupled bending and stretching effects, and will not be considered here.

¹ If it is relatively thick, as in concrete pavements or Argentinian beefsteaks, the term *slab* is also used but not for plane stress conditions.

REMARK 14.1

Selective relaxation from assumption 4 lead to the so-called *generalized plane stress state*, in which z stresses are accepted. The *plane strain state* is obtained if strains in the z direction are precluded. Although the construction of finite element models for those states has many common points with plane stress, we shall not consider those models here. For isotropic materials the plane stress and plane strain problems can be mapped into each other through a fictitious-property technique; see Exercise 14.1.

REMARK 14.2

Transverse loading on a plate produces *plate bending*, which is associated with a more complex configuration of internal forces and deformations. This subject is studied in more advanced courses.

§14.1.2. Mathematical Model

The mathematical model of the plate in plane stress is a two-dimensional boundary value problem (BVP). This problem is posed over a plane domain Ω with a boundary Γ , as illustrated in Figure 14.2.

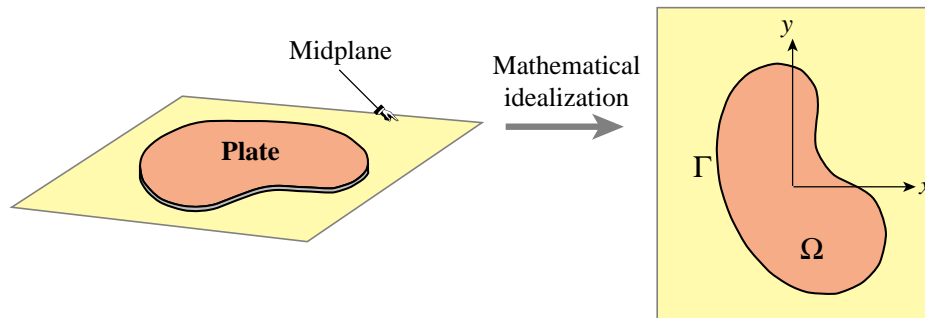


Figure 14.2. Mathematical model of plate in plane stress.

In this idealization the third dimension is represented as functions of x and y that are *integrated through the plate thickness*. Engineers often work with internal plate forces, which result from integrating the in-plane stresses through the thickness. See Figure 14.3.

§14.2. PLANE STRESS PROBLEM DESCRIPTION**§14.2.1. Problem Data**

The given data includes:

Domain geometry. This is defined by the boundary Γ illustrated in Figure 14.2.

Thickness. Most plates used as structural components have constant thickness. If the thickness does vary, in which case $h = h(x, y)$, it should do so gradually to maintain the plane stress state.

Material data. This is defined by the constitutive equations. Here we shall assume that the plate material is linearly elastic but not necessarily isotropic.

Specified Interior Forces. These are known forces that act in the interior Ω of the plate. There are of two types. *Body forces* or *volume forces* are forces specified per unit of plate volume; for example the plate weight. *Face forces* act tangentially to the plate faces and are transported to the

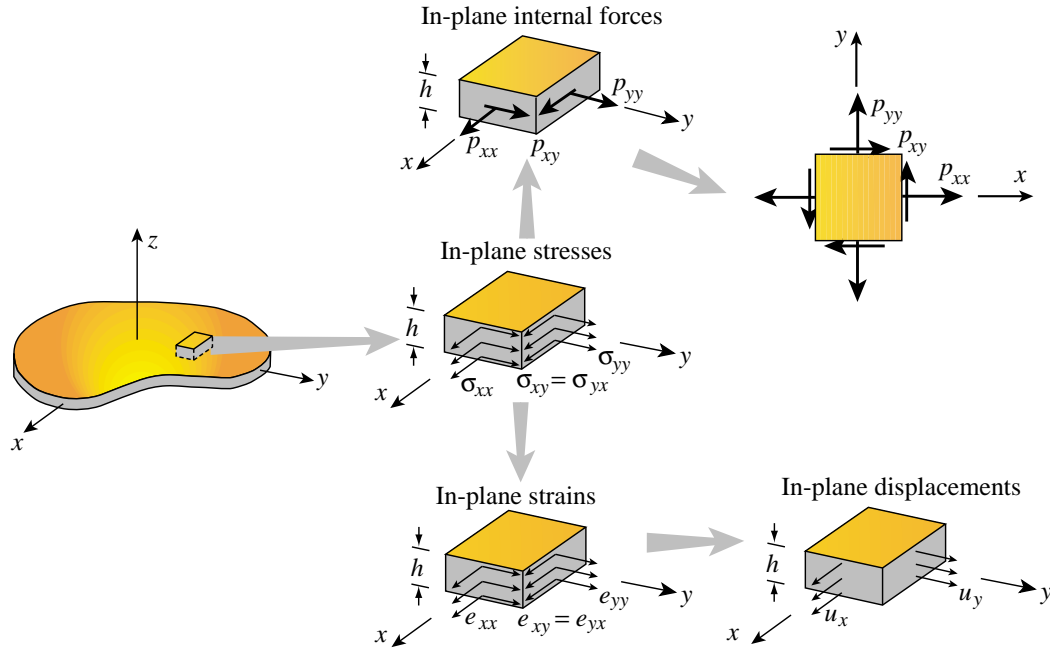


Figure 14.3. Notation for in-plane stresses, strains, displacements and internal forces for a plate in plane stress.

midplane. For example, the friction or drag force on an airplane skin is of this type if the skin is modeled to be in plane stress.

Specified Surface Forces. These are known forces that act on the boundary Γ of the plate. In elasticity these are called *surface tractions*. In actual applications it is important to know whether these forces are specified per unit of surface area or per unit length.

Displacement Boundary Conditions. These specify how the plate is supported. Points on the plate boundary may be fixed, allowed to move in one direction, or subject to multipoint constraints. In addition symmetry and antisymmetry lines may be identified as discussed in Chapter 8.

If no displacement boundary conditions are imposed, the plate structure is said to be *free-free*.

§14.2.2. Problem Unknowns

The three unknown fields are displacements, strains and stresses. Because of the assumed wall fabrication homogeneity the in-plane components are assumed to be *uniform through the plate thickness*. Consequently the dependence on z disappears and all such components become functions of x and y only.

Displacements. The in-plane displacement field is defined by two components:

$$\mathbf{u}(x, y) = \begin{bmatrix} u_x(x, y) \\ u_y(x, y) \end{bmatrix} \quad (14.1)$$

The transverse displacement component $u_z(x, y, z)$ component is generally nonzero because of Poisson's ratio effects, and depends on z . However, this displacement does not appear in the governing equations.

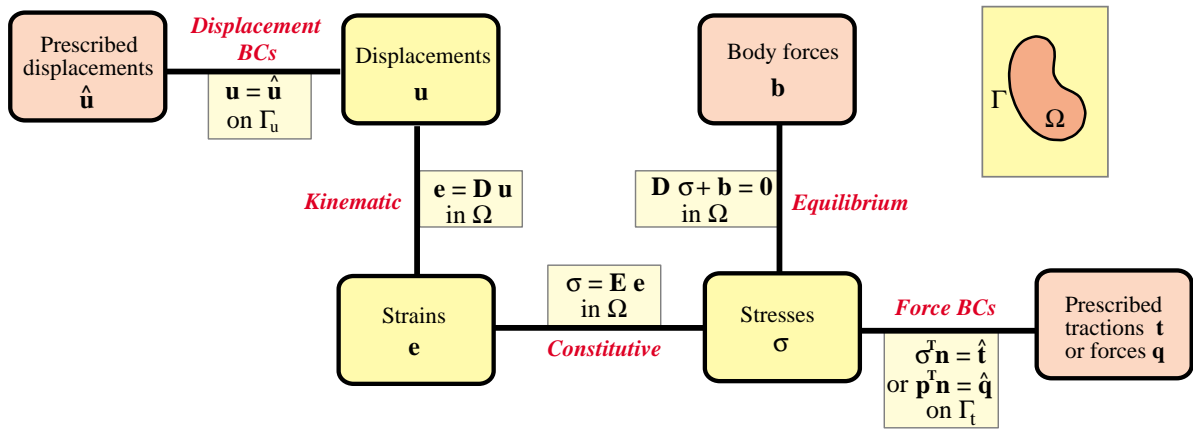


Figure 14.4. The Strong Form of the plane stress equations of linear elastostatics displayed as a Tonti diagram. Yellow boxes identify prescribed fields whereas orange boxes denote unknown fields. The distinction between Strong and Weak Forms is explained in §14.3.3.

Strains. The in-plane strain field forms a tensor defined by three independent components: e_{xx} , e_{yy} and e_{xy} . To allow stating the FE equations in matrix form, these components are conventionally arranged to form a 3-component “strain vector”

$$\mathbf{e}(x, y) = \begin{bmatrix} e_{xx}(x, y) \\ e_{yy}(x, y) \\ 2e_{xy}(x, y) \end{bmatrix} \quad (14.2)$$

The factor of 2 in e_{xy} shortens strain energy expressions. The shear strain components e_{xz} and e_{yz} vanish. The transverse normal strain e_{zz} is generally nonzero because of Poisson effects. This strain does not enter the governing equations as unknown because the associated stress σ_{zz} is zero, which eliminates the contribution of $\sigma_{zz}e_{zz}$ to the internal energy.

Stresses. The in-plane stress field forms a tensor defined by three independent components: σ_{xx} , σ_{yy} and σ_{xy} . As in the case of strains, to allow stating the FE equations in matrix form, these components are conventionally arranged to form a 3-component “stress vector”

$$\boldsymbol{\sigma}(x, y) = \begin{bmatrix} \sigma_{xx}(x, y) \\ \sigma_{yy}(x, y) \\ \sigma_{xy}(x, y) \end{bmatrix} \quad (14.3)$$

The remaining three stress components: σ_{zz} , σ_{xz} and σ_{yz} , are assumed to vanish.

The *plate internal forces* are obtained on integrating the stresses through the thickness. Under the assumption of uniform stress distribution,

$$p_{xx} = \sigma_{xx}h, \quad p_{yy} = \sigma_{yy}h, \quad p_{xy} = \sigma_{xy}h. \quad (14.4)$$

These p ’s also form a tensor. They are called *membrane forces* in the literature. See Figure 14.3.

§14.3. LINEAR ELASTICITY EQUATIONS

We shall develop plane stress finite elements in the framework of classical linear elasticity. The necessary governing equations are presented below. They are graphically represented in the Strong Form Tonti diagram of Figure 14.4.

§14.3.1. Governing Equations

The three internal fields: displacements, strains and stresses (14.2)-(14.4) are connected by three field equations: kinematic, constitutive and internal-equilibrium equations. If initial strain effects are ignored, these equations read

$$\begin{aligned} \begin{bmatrix} e_{xx} \\ e_{yy} \\ 2e_{xy} \end{bmatrix} &= \begin{bmatrix} \partial/\partial x & 0 \\ 0 & \partial/\partial y \\ \partial/\partial y & \partial/\partial x \end{bmatrix} \begin{bmatrix} u_x \\ u_y \end{bmatrix}, \\ \begin{bmatrix} \sigma_{xx} \\ \sigma_{yy} \\ \sigma_{xy} \end{bmatrix} &= \begin{bmatrix} E_{11} & E_{12} & E_{13} \\ E_{12} & E_{22} & E_{23} \\ E_{13} & E_{23} & E_{33} \end{bmatrix} \begin{bmatrix} e_{xx} \\ e_{yy} \\ 2e_{xy} \end{bmatrix}, \\ \begin{bmatrix} \partial/\partial x & 0 & \partial/\partial y \\ 0 & \partial/\partial y & \partial/\partial x \end{bmatrix} \begin{bmatrix} \sigma_{xx} \\ \sigma_{yy} \\ \sigma_{xy} \end{bmatrix} + \begin{bmatrix} b_x \\ b_y \end{bmatrix} &= \begin{bmatrix} 0 \\ 0 \end{bmatrix}. \end{aligned} \quad (14.5)$$

In these equations, b_x and b_y are the components of the interior body force \mathbf{b} , \mathbf{E} is the 3×3 stress-strain matrix of plane stress elastic moduli, \mathbf{D} is the 3×2 symmetric-gradient operator and its transpose the 2×3 tensor-divergence operator. The dependence on (x, y) has been omitted to reduce clutter.

The compact matrix version of (14.5) is

$$\boxed{\mathbf{e} = \mathbf{D}\mathbf{u}, \quad \boldsymbol{\sigma} = \mathbf{E}\mathbf{e}, \quad \mathbf{D}^T \boldsymbol{\sigma} + \mathbf{b} = \mathbf{0},} \quad (14.6)$$

in which $\mathbf{E} = \mathbf{E}^T$.

If the plate material is isotropic with elastic modulus E and Poisson's ratio ν , the moduli in the constitutive matrix \mathbf{E} reduce to $E_{11} = E_{22} = E/(1 - \nu^2)$, $E_{33} = \frac{1}{2}E/(1 + \nu) = G$, $E_{12} = \nu E_{11}$ and $E_{13} = E_{23} = 0$. See also Exercise 14.1.

§14.3.2. Boundary Conditions

Boundary conditions prescribed on Γ may be of two types: displacement BC or force BC (also called stress BC or traction BC). To write down those conditions it is conceptually convenient to break up Γ into two subsets: Γ_u and Γ_t , over which displacements and force or stresses, respectively, are specified. See Figure 14.5.

Displacement boundary conditions are prescribed on Γ_u in the form

$$\boxed{\mathbf{u} = \hat{\mathbf{u}}.} \quad (14.7)$$

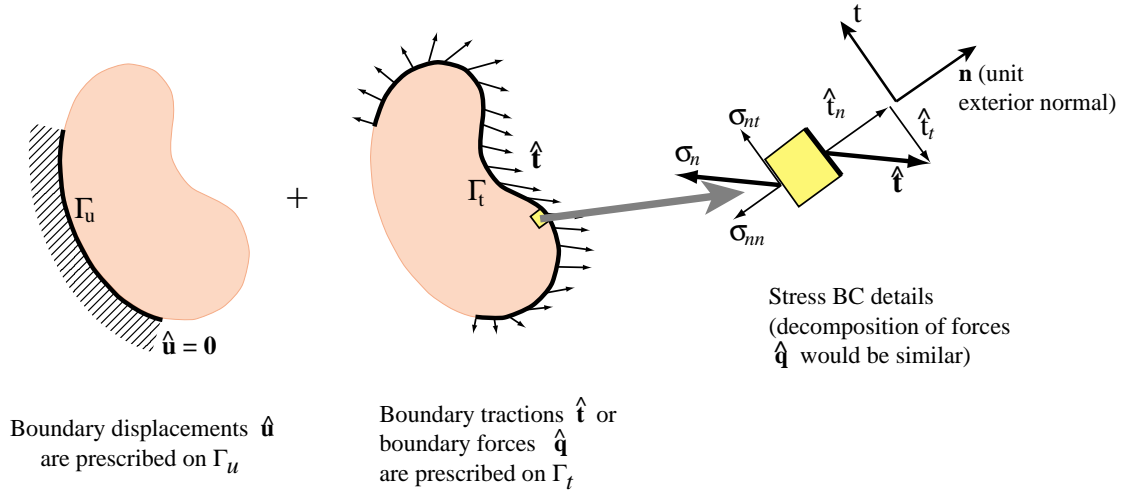


Figure 14.5. Displacement and force (stress, traction) boundary conditions for the plane stress problem.

Here $\hat{\mathbf{u}}$ are prescribed displacements. Often $\hat{\mathbf{u}} = \mathbf{0}$. This happens in fixed portions of the boundary, as the ones illustrated in Figure 14.5.

Force boundary conditions (also called stress BCs and traction BCs in the literature) are specified on Γ_t . They take the form

$$\sigma_n = \hat{\mathbf{t}}. \quad (14.8)$$

Here $\hat{\mathbf{t}}$ are prescribed surface tractions specified as a force per unit area (that is, not integrated through the thickness), and σ_n is the stress vector shown in Figure 14.5.

An alternative form of (14.8) uses the internal plate forces:

$$\mathbf{p}_n = \hat{\mathbf{q}}. \quad (14.9)$$

Here $\mathbf{p}_n = \sigma_n h$ and $\hat{\mathbf{q}} = \hat{\mathbf{t}} h$. This form is used more often than (14.8) in structural design, particularly when the plate wall construction is nonhomogeneous.

The components of σ_n in Cartesian coordinates follow from Cauchy's stress transformation formula

$$\sigma_n = \begin{bmatrix} \sigma_{xx}n_x + \sigma_{xy}n_y \\ \sigma_{xy}n_x + \sigma_{yy}n_y \end{bmatrix} = \begin{bmatrix} n_x & 0 & n_y \\ 0 & n_y & n_x \end{bmatrix} \begin{bmatrix} \sigma_{xx} \\ \sigma_{yy} \\ \sigma_{xy} \end{bmatrix}, \quad (14.10)$$

in which n_x and n_y denote the Cartesian components of the unit normal vector $\mathbf{n}^{(e)}$ (also called the direction cosines of the normal). Thus (14.8) splits into two scalar conditions: $\hat{t}_x = \sigma_{nx}$ and $\hat{t}_y = \sigma_{ny}$.

It is sometimes convenient to write the condition (14.8) in terms of normal n and tangential t directions:

$$\sigma_{nn} = \hat{t}_n, \quad \sigma_{nt} = \hat{t}_t \quad (14.11)$$

in which $\sigma_{nn} = \sigma_{nx}n_x + \sigma_{ny}n_y$ and $\sigma_{nt} = -\sigma_{nx}n_y + \sigma_{ny}n_x$.

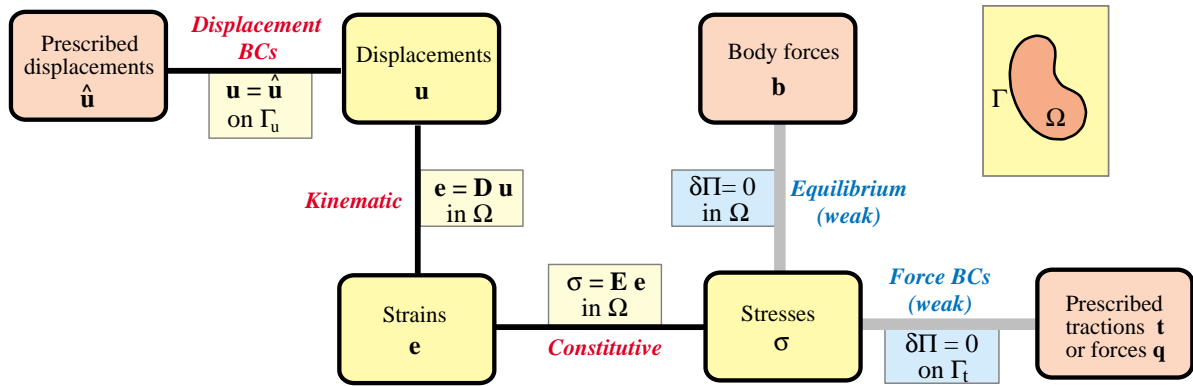


Figure 14.6. The TPE-based Weak Form of the plane stress equations of linear elastostatics. Weak links are marked with grey lines.

REMARK 14.3

The separation of Γ into Γ_u and Γ_t is useful for conciseness in the mathematical formulation, such as the energy integrals presented below. It does not exhaust, however, all BC possibilities. Frequently at points of Γ one specifies a displacement in one direction and a force (or stress) in the other. An example of these are roller and sliding conditions as well as lines of symmetry and antisymmetry. To cover these situations one needs either a generalization of the split, in which Γ_u and Γ_t are permitted to overlap, or to define another portion Γ_m for “mixed” conditions. Such generalizations will not be presented here, as they become unimportant once the FE discretization is done.

§14.3.3. Weak Forms versus Strong Form

We introduce now some further terminology from variational calculus. The Tonti diagram of Figure 14.4 is said to display the Strong Form of the governing equations because all relations are verified point by point. These relations, called *strong links*, are shown in the diagram with black lines.

A Weak Form is obtained by *relaxing* one or more strong links. Those are replaced by *weak links*, which enforce relations in an average or integral sense rather than point by point. The weak links are then provided by the variational formulation chosen for the problem. Because in general many variational forms of the same problem are possible, there are many possible Weak Forms. On the other hand the Strong Form is unique.

The Weak Form associated with the Total Potential Energy (TPE) variational form is illustrated in Figure 14.6. The internal equilibrium equations and stress BC become weak links, which are indicated by gray lines. These equations are given by the variational statement $\delta\Pi = 0$, where the TPE functional Π is given in the next subsection. The FEM displacement formulation discussed below is based on this particular Weak Form.

§14.3.4. Total Potential Energy

As usual the Total Potential Energy functional for the plane stress problem is given by

$$\Pi = U - W. \quad (14.12)$$

The internal energy is the elastic strain energy:

$$U = \frac{1}{2} \int_{\Omega} h \boldsymbol{\sigma}^T \mathbf{e} d\Omega = \frac{1}{2} \int_{\Omega} h \mathbf{e}^T \mathbf{E} \mathbf{e} d\Omega. \quad (14.13)$$

The derivation details are relegated to Exercise E14.5. The external energy is the sum of contributions from known interior and boundary forces:

$$W = \int_{\Omega} h \mathbf{u}^T \mathbf{b} d\Omega + \int_{\Gamma_t} h \mathbf{u}^T \hat{\mathbf{t}} d\Gamma. \quad (14.14)$$

Note that the boundary integral over Γ is taken only over Γ_t . That is, the portion of the boundary over which tractions or forces are specified.

§14.4. FINITE ELEMENT EQUATIONS

The necessary equations to apply the finite element method are collected here and expressed in matrix form. The domain of Figure 14.7(a) is discretized by a finite element mesh as illustrated in Figure 14.7(b). From this mesh we extract a generic element labeled (e) with $n \geq 3$ node points. In the subsequent derivation the number n is kept *arbitrary*. Therefore, the formulation is applicable to arbitrary two-dimensional elements, for example those sketched in Figure 14.8.

Departing from previous practice in 1D elements, the element node points will be labelled 1 through n . These are called *local node numbers*. The element domain and boundary are denoted by $\Omega^{(e)}$ and $\Gamma^{(e)}$, respectively.

The element has $2n$ degrees of freedom. These are collected in the element node displacement vector

$$\mathbf{u}^{(e)} = [u_{x1} \quad u_{y1} \quad u_{x2} \quad \dots \quad u_{xn} \quad u_{yn}]^T. \quad (14.15)$$

§14.4.1. Displacement Interpolation

The displacement field $\mathbf{u}^{(e)}(x, y)$ over the element is interpolated from the node displacements. We shall assume that the same interpolation functions are used for both displacement components.² Thus

$$u_x(x, y) = \sum_{i=1}^n N_i^{(e)}(x, y) u_{xi}, \quad u_y(x, y) = \sum_{i=1}^n N_i^{(e)}(x, y) u_{yi}, \quad (14.16)$$

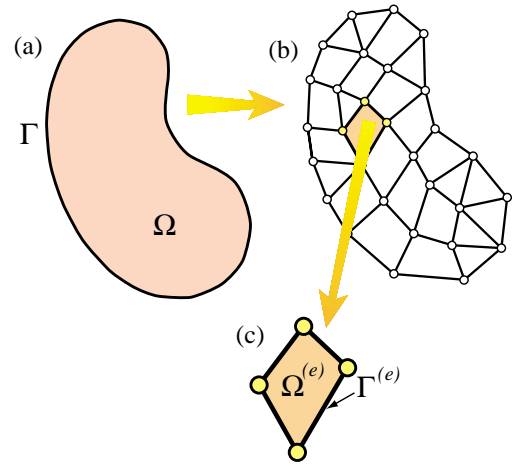


Figure 14.7. Finite element discretization and extraction of generic element.

² This is the so called *element isotropy* condition, which is studied and justified in advanced FEM courses.

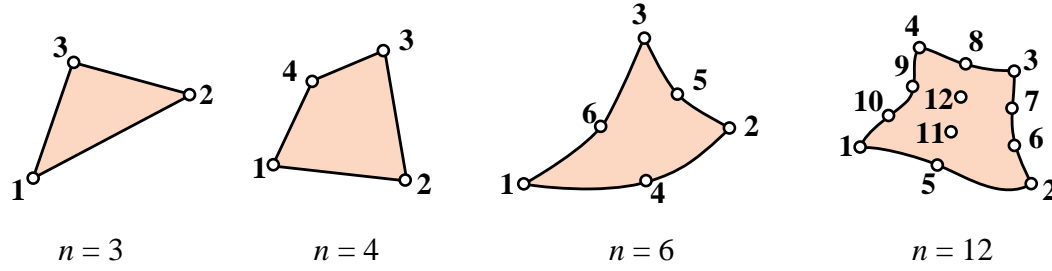


Figure 14.8. Example two-dimensional finite elements, characterized by their number of nodes n .

where $N_i^{(e)}(x, y)$ are the element shape functions. In matrix form:

$$\mathbf{u}(x, y) = \begin{bmatrix} u_x(x, y) \\ u_y(x, y) \end{bmatrix} = \begin{bmatrix} N_1^{(e)} & 0 & N_2^{(e)} & 0 & \dots & N_n^{(e)} & 0 \\ 0 & N_1^{(e)} & 0 & N_2^{(e)} & \dots & 0 & N_n^{(e)} \end{bmatrix} \mathbf{u}^{(e)} = \mathbf{N}^{(e)} \mathbf{u}^{(e)}. \quad (14.17)$$

The minimum conditions on $N_i^{(e)}(x, y)$ is that it must take the value one at the i^{th} node and zero at all others, so that the interpolation (14.17) is correct at the nodes. Additional requirements on the shape functions are stated in later Chapters.

Differentiating the finite element displacement field yields the strain-displacement relations:

$$\mathbf{e}(x, y) = \begin{bmatrix} \frac{\partial N_1^{(e)}}{\partial x} & 0 & \frac{\partial N_2^{(e)}}{\partial x} & 0 & \dots & \frac{\partial N_n^{(e)}}{\partial x} & 0 \\ 0 & \frac{\partial N_1^{(e)}}{\partial y} & 0 & \frac{\partial N_2^{(e)}}{\partial y} & \dots & 0 & \frac{\partial N_n^{(e)}}{\partial y} \\ \frac{\partial N_1^{(e)}}{\partial y} & \frac{\partial N_1^{(e)}}{\partial x} & \frac{\partial N_2^{(e)}}{\partial y} & \frac{\partial N_2^{(e)}}{\partial x} & \dots & \frac{\partial N_n^{(e)}}{\partial y} & \frac{\partial N_n^{(e)}}{\partial x} \end{bmatrix} \mathbf{u}^{(e)} = \mathbf{B} \mathbf{u}^{(e)}. \quad (14.18)$$

The strain-displacement matrix $\mathbf{B} = \mathbf{D}\mathbf{N}^{(e)}$ is $3 \times 2n$. The superscript (e) is omitted from it to reduce clutter. The stresses are given in terms of strains and displacements by $\boldsymbol{\sigma} = \mathbf{E} \mathbf{e} = \mathbf{E}\mathbf{B}\mathbf{u}^{(e)}$, which is assumed to hold at all points of the element.

§14.4.2. Element Energy

To obtain finite element stiffness equations, the variation of the TPE functional is decomposed into contributions from individual elements:

$$\delta \Pi^{(e)} = \delta U^{(e)} - \delta W^{(e)} = 0. \quad (14.19)$$

where

$$U^{(e)} = \frac{1}{2} \int_{\Omega^{(e)}} h \boldsymbol{\sigma}^T \mathbf{e} d\Omega^{(e)} = \frac{1}{2} \int_{\Omega^{(e)}} h \mathbf{e}^T \mathbf{E} \mathbf{e} d\Omega^{(e)} \quad (14.20)$$

and

$$W^{(e)} = \int_{\Omega^{(e)}} h \mathbf{u}^T \mathbf{b} d\Omega^{(e)} + \int_{\Gamma^{(e)}} h \mathbf{u}^T \hat{\mathbf{t}} d\Gamma^{(e)} \quad (14.21)$$

Note that in (14.21) $\Gamma_i^{(e)}$ has been taken equal to the complete boundary $\Gamma^{(e)}$ of the element. This is a consequence of the fact that displacement boundary conditions are applied *after* assembly, to a free-free structure. Consequently it does not harm to assume that all boundary conditions are of stress type insofar as forming the element equations.

§14.4.3. Element Stiffness Equations

Inserting the relations $\mathbf{u} = \mathbf{N}\mathbf{u}^{(e)}$, $\mathbf{e} = \mathbf{B}\mathbf{u}^{(e)}$ and $\boldsymbol{\sigma} = \mathbf{E}\mathbf{e}$ into $\Pi^{(e)}$ yields the quadratic form in the nodal displacements

$$\Pi^{(e)} = \frac{1}{2} \mathbf{u}^{(e)T} \mathbf{K}^{(e)} \mathbf{u}^{(e)} - \mathbf{u}^{(e)T} \mathbf{f}^{(e)}, \quad (14.22)$$

where the element stiffness matrix is

$$\mathbf{K}^{(e)} = \int_{\Omega^{(e)}} h \mathbf{B}^T \mathbf{E} \mathbf{B} d\Omega^{(e)}, \quad (14.23)$$

and the consistent element nodal force vector is

$$\mathbf{f}^{(e)} = \int_{\Omega^{(e)}} h \mathbf{N}^T \mathbf{b} d\Omega^{(e)} + \int_{\Gamma^{(e)}} h \mathbf{N}^T \hat{\mathbf{t}} d\Gamma^{(e)}. \quad (14.24)$$

In the second integral the matrix \mathbf{N} is evaluated on the element boundary only.

The calculation of the entries of $\mathbf{K}^{(e)}$ and $\mathbf{f}^{(e)}$ for several elements of historical or practical interest is described in subsequent Chapters.

Notes and Bibliography

The plane stress problem is well suited for introducing continuum finite elements, from both historical and technical standpoints. Some books use the Poisson equation for this purpose, but problems such as heat conduction cannot illustrate features such as vector-mixed boundary conditions and shear effects.

The first continuum structural finite elements were developed at Boeing in the early 1950s to model delta-wing skin panels [14.2,14.4]. A plane stress model was naturally chosen for the panels. The paper that gave the method its name [14.1] used the plane stress problem as application driver.

The technical aspects of plane stress can be found in any book on elasticity. A particularly readable one is Fung [14.3], which is unfortunately out of print.

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Homework Exercises for Chapter 14

The Plane Stress Problem

EXERCISE 14.1

[A:25] Suppose that the structural material is isotropic, with elastic modulus E and Poisson's ratio ν . The in-plane stress-strain relations for plane stress ($\sigma_{zz} = \sigma_{xz} = \sigma_{yz} = 0$) and plane strain ($e_{zz} = e_{xz} = e_{yz} = 0$) as given in any textbook on elasticity, are

$$\begin{aligned} \text{plane stress: } \begin{bmatrix} \sigma_{xx} \\ \sigma_{yy} \\ \sigma_{xy} \end{bmatrix} &= \frac{E}{1-\nu^2} \begin{bmatrix} 1 & \nu & 0 \\ \nu & 1 & 0 \\ 0 & 0 & \frac{1-\nu}{2} \end{bmatrix} \begin{bmatrix} e_{xx} \\ e_{yy} \\ 2e_{xy} \end{bmatrix}, \\ \text{plane strain: } \begin{bmatrix} \sigma_{xx} \\ \sigma_{yy} \\ \sigma_{xy} \end{bmatrix} &= \frac{E(1-\nu)}{(1+\nu)(1-2\nu)} \begin{bmatrix} 1 & \frac{\nu}{1-\nu} & 0 \\ \frac{\nu}{1-\nu} & 1 & 0 \\ 0 & 0 & \frac{1-2\nu}{2(1-\nu)} \end{bmatrix} \begin{bmatrix} e_{xx} \\ e_{yy} \\ 2e_{xy} \end{bmatrix}. \end{aligned} \quad (\text{E14.1})$$

Show that the constitutive matrix of plane strain can be formally obtained by replacing E by a fictitious modulus E^* and ν by a fictitious Poisson's ratio ν^* in the plane stress constitutive matrix and suppressing the stars. Find the expression of E^* and ν^* in terms of E and ν . This device permits “reusing” a plane stress FEM program to do plane strain, as long as the material is isotropic.

EXERCISE 14.2

[A:25] In the finite element formulation of near incompressible isotropic materials (as well as plasticity and viscoelasticity) it is convenient to use the so-called *Lamé constants* λ and μ instead of E and ν in the constitutive equations. Both λ and μ have the physical dimension of stress and are related to E and ν by

$$\lambda = \frac{\nu E}{(1+\nu)(1-2\nu)}, \quad \mu = G = \frac{E}{2(1+\nu)}. \quad (\text{E14.2})$$

Conversely

$$E = \frac{\mu(3\lambda + 2\mu)}{\lambda + \mu}, \quad \nu = \frac{\lambda}{2(\lambda + \mu)}. \quad (\text{E14.3})$$

Substitute (E14.3) into (E14.1) to express the two stress-strain matrices in terms of λ and μ . Then split the stress-strain matrix \mathbf{E} of plane strain as

$$\mathbf{E} = \mathbf{E}_\mu + \mathbf{E}_\lambda \quad (\text{E14.4})$$

in which \mathbf{E}_μ and \mathbf{E}_λ contain only μ and λ , respectively, with \mathbf{E}_μ diagonal and $E_{\lambda 33} = 0$. This is the Lamé or $\{\lambda, \mu\}$ splitting of the plane strain constitutive equations, which leads to the so-called **B**-bar formulation of near-incompressible finite elements.³ Express \mathbf{E}_μ and \mathbf{E}_λ also in terms of E and ν .

For the plane stress case perform a similar splitting in which where \mathbf{E}_λ contains only $\bar{\lambda} = 2\lambda\mu/(\lambda + 2\mu)$ with $E_{\lambda 33} = 0$, and \mathbf{E}_μ is a diagonal matrix function of μ and $\bar{\lambda}$.⁴ Express \mathbf{E}_μ and \mathbf{E}_λ also in terms of E and ν .

³ Equation (E14.4) is sometimes referred to as the deviatoric+volumetric splitting of the stress-strain law, on account of its physical meaning in plane strain. That meaning is lost, however, for plane stress.

⁴ For the physical significance of $\bar{\lambda}$ see I. Sokolnikoff, *The Mathematical Theory of Elasticity*, McGraw-Hill, 2nd ed., 1956, p. 254ff.

EXERCISE 14.3

[A:15] Derive the Cauchy stress-to-traction equations (14.10) using force equilibrium along x and y and the geometric relations shown in Figure E14.1. (This is the “wedge method” in Mechanics of Materials.) Hint: $t_x ds = \sigma_{xx} dy + \sigma_{xy} dx$, etc.

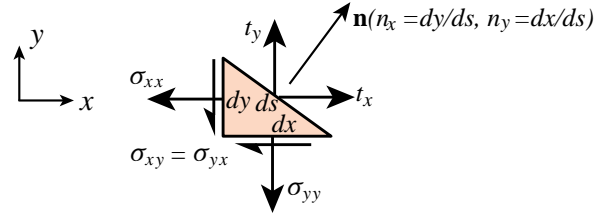


Figure E14.1. Geometry for deriving (14.10).

EXERCISE 14.4

[A:20] Include thermoelastic effects in the plane stress field equations, assuming a thermally isotropic material with coefficient of linear expansion α . Hint: start from the inverse of the first of (E14.1), add thermal strains $e_{xx}T = e_{yy}T = \alpha T$, and solve for stresses.

EXERCISE 14.5

[A:25=5+5+15] A plate is in linearly elastic plane stress. It is shown in courses in elasticity that the internal strain energy density stored per unit volume is

$$\mathcal{U} = \frac{1}{2}(\sigma_{xx}e_{xx} + \sigma_{yy}e_{yy} + \sigma_{xy}e_{xy} + \sigma_{yx}e_{yx}) = \frac{1}{2}(\sigma_{xx}e_{xx} + \sigma_{yy}e_{yy} + 2\sigma_{xy}e_{xy}). \quad (\text{E14.5})$$

- (a) Show that (E14.5) can be written in terms of strains only as

$$\mathcal{U} = \frac{1}{2} \mathbf{e}^T \mathbf{E} \mathbf{e}, \quad (\text{E14.6})$$

and hence justify (14.13).

- (b) Show that (E14.5) can be written in terms of stresses only as

$$\mathcal{U} = \frac{1}{2} \boldsymbol{\sigma}^T \mathbf{C} \boldsymbol{\sigma}, \quad (\text{E14.7})$$

where $\mathbf{C} = \mathbf{E}^{-1}$ is the elastic compliance (strain-stress) matrix.

- (c) Suppose you want to write (E14.5) in terms of the extensional strains $\{e_{xx}, e_{yy}\}$ and of the shear stress $\sigma_{xy} = \sigma_{yx}$. This is known as a mixed representation. Show that

$$\mathcal{U} = \frac{1}{2} \begin{bmatrix} e_{xx} \\ e_{yy} \\ \sigma_{xy} \end{bmatrix}^T \begin{bmatrix} A_{11} & A_{12} & A_{13} \\ A_{12} & A_{22} & A_{23} \\ A_{13} & A_{23} & A_{33} \end{bmatrix} \begin{bmatrix} e_{xx} \\ e_{yy} \\ \sigma_{xy} \end{bmatrix}, \quad (\text{E14.8})$$

and explain how the entries A_{ij} can be calculated⁵ in terms of the elastic moduli E_{ij} .

⁵ The process of computing \mathbf{A} is an instance of “partial inversion” of matrix \mathbf{E} . See Remark 11.3.

15

The Linear Plane Stress Triangle

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§15.1. INTRODUCTION

This Chapter presents the element equations of a three-node triangle with assumed linear displacements for the plane stress problem formulated in Chapter 14. This element is called the *linear triangle*. This particular element is distinguished in several respects:

- (1) It belongs to both the isoparametric and superparametric element families, which are covered in the next Chapter.
- (2) It allows closed form derivations for its stiffness and consistent forces without the need for numerical integration.
- (3) It cannot be improved by the addition of internal degrees of freedom.

In addition the linear triangle has historical importance.¹ Although it is not a good performer for structural stress analysis, it is still used in problems that do not require high accuracy, as well as in non-structural applications. One reason is that triangular meshes are easily generated over arbitrary domains using techniques such as Delaunay triangulations.

§15.1.1. Parametric Representation of Functions

The concept of *parametric representation* of functions is crucial in modern FEM presentations. Together with numerical integration, it has become a key tool for the systematic development of elements in two and three space dimensions. Without these two tools the element developer would become lost in an algebraic maze as element geometric shapes get more complicated.

The essentials of the idea of parametric representation can be illustrated through a simple example. Consider the following alternative representations of the unit-circle function, $x^2 + y^2 = 1$:

$$y = \sqrt{1 - x^2} \quad (15.1)$$

$$x = \cos \theta, \quad y = \sin \theta \quad (15.2)$$

The direct representation (15.1) fits the conventional function notation, i.e., $y = f(x)$. Given a value of x , it returns one or more y . On the other hand, the representation (15.2) is parametric: both x and y are given in terms of one parameter, the angle θ . Elimination of θ through the trigonometric identity $\cos^2 \theta + \sin^2 \theta = 1$ recovers $x^2 + y^2 = 1$. But there are many situations in which working with the parametric form throughout the development is more convenient. Continuum finite elements provide a striking illustration of this point.

§15.2. TRIANGLE GEOMETRY AND COORDINATE SYSTEMS

The geometry of the 3-node triangle shown in Figure 15.1(a) is specified by the location of its three corner nodes on the $\{x, y\}$ plane. The nodes are labelled 1, 2, 3 while traversing the sides in *counterclockwise* fashion. The location of the corners is defined by their Cartesian coordinates: $\{x_i, y_i\}$ for $i = 1, 2, 3$.

¹ This was one of the two continuum elements presented by Martin, Turner, Clough and Topp in their landmark 1956 paper. Its publication is widely regarded as the start of the present FEM. See Appendix H.

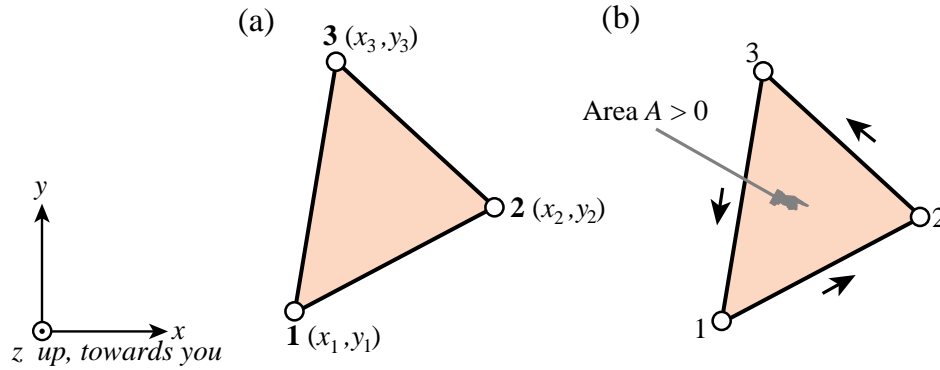


Figure 15.1. The three-node, linear-displacement plane stress triangular element: (a) geometry; (b) area and positive boundary traversal.

The element has six degrees of freedom, defined by the six nodal displacement components $\{u_{xi}, u_{yi}\}$, for $i = 1, 2, 3$. The interpolation of the internal displacements $\{u_x, u_y\}$ from these six values is studied in §15.3, after triangular coordinates are introduced.

The area of the triangle is denoted by A and is given by

$$2A = \det \begin{bmatrix} 1 & 1 & 1 \\ x_1 & x_2 & x_3 \\ y_1 & y_2 & y_3 \end{bmatrix} = (x_2y_3 - x_3y_2) + (x_3y_1 - x_1y_3) + (x_1y_2 - x_2y_1). \quad (15.3)$$

The area given by (15.3) is a *signed* quantity. It is positive if the corners are numbered in cyclic counterclockwise order as shown in Figure 15.1(b). This convention is followed in the sequel.

§15.2.1. Triangular Coordinates

Points of the triangle may also be located in terms of a *parametric* coordinate system:

$$\zeta_1, \zeta_2, \zeta_3. \quad (15.4)$$

In the literature these three parameters receive an astonishing number of names, as the list given in Table 15.1 shows. In the sequel the name *triangular coordinates* will be used to emphasize the close association with this particular geometry.

Equations

$$\zeta_i = \text{constant} \quad (15.5)$$

represent a set of straight lines parallel to the side opposite to the i^{th} corner, as depicted in Figure 15.2. The equation of sides 1–2, 2–3 and 3–1 are $\zeta_1 = 0$, $\zeta_2 = 0$ and $\zeta_3 = 0$, respectively. The three corners have coordinates $(1,0,0)$, $(0,1,0)$ and $(0,0,1)$. The three midpoints of the sides have coordinates $(\frac{1}{2}, \frac{1}{2}, 0)$, $(0, \frac{1}{2}, \frac{1}{2})$ and $(\frac{1}{2}, 0, \frac{1}{2})$, the centroid has coordinates $(\frac{1}{3}, \frac{1}{3}, \frac{1}{3})$, and so on. The coordinates are not independent because their sum is unity:

$$\zeta_1 + \zeta_2 + \zeta_3 = 1. \quad (15.6)$$

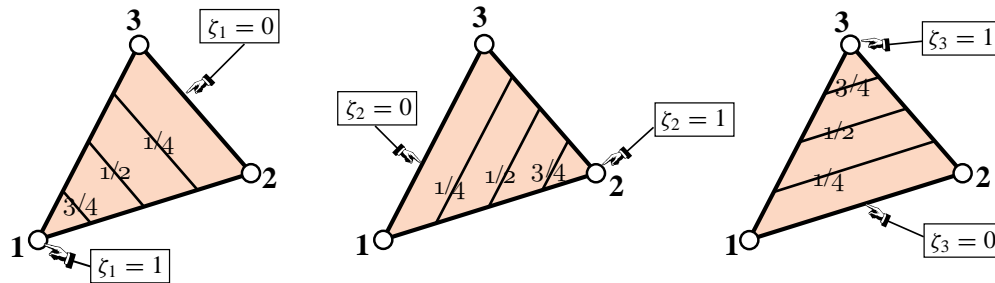


Figure 15.2. Triangular coordinates.

REMARK 15.1

In older (pre-1970) FEM publications triangular coordinates are often called *area coordinates*. This name comes from the following interpretation: $\zeta_i = A_{jk}/A$, where A_{jk} is the area subtended by the triangle formed by the point P and corners j and k , in which j and k are cyclic permutations of i . Historically this was the way the coordinates were defined in 1960s papers. Unfortunately the interpretation does not carry over to general isoparametric triangles with curved sides and thus it is not used here.

Table 15.1 Names of element parametric coordinates in the FEM literature

<i>Name</i>	<i>Applicable to</i>
natural coordinates	all elements
isoparametric coordinates	isoparametric elements
shape function coordinates	isoparametric elements
barycentric coordinates	triangles, tetrahedra
Möbius coordinates	triangles
triangular coordinates	all triangles
area coordinates	straight-sided triangles

§15.2.2. Linear Interpolation

Consider a function $f(x, y)$ that varies *linearly* over the triangle domain. In terms of Cartesian coordinates it may be expressed as

$$f(x, y) = a_0 + a_1x + a_2y, \quad (15.7)$$

where a_0, a_1 and a_2 are coefficients to be determined from three conditions. In finite element work such conditions are often the *nodal values* taken by f at the corners:

$$f_1, f_2, f_3 \quad (15.8)$$

The expression in triangular coordinates makes direct use of those three values:

$$f(\zeta_1, \zeta_2, \zeta_3) = f_1\zeta_1 + f_2\zeta_2 + f_3\zeta_3 = [f_1 \ f_2 \ f_3] \begin{bmatrix} \zeta_1 \\ \zeta_2 \\ \zeta_3 \end{bmatrix} = [\zeta_1 \ \zeta_2 \ \zeta_3] \begin{bmatrix} f_1 \\ f_2 \\ f_3 \end{bmatrix}. \quad (15.9)$$

Expression (15.9) is called a *linear interpolant* for f .

§15.2.3. Coordinate Transformations

Quantities that are closely linked with the element geometry are naturally expressed in triangular coordinates. On the other hand, quantities such as displacements, strains and stresses are often expressed in the Cartesian system x, y . We therefore need transformation equations through which we can pass from one coordinate system to the other.

Cartesian and triangular coordinates are linked by the relation

$$\begin{bmatrix} 1 \\ x \\ y \end{bmatrix} = \begin{bmatrix} 1 & 1 & 1 \\ x_1 & x_2 & x_3 \\ y_1 & y_2 & y_3 \end{bmatrix} \begin{bmatrix} \zeta_1 \\ \zeta_2 \\ \zeta_3 \end{bmatrix}. \quad (15.10)$$

The first equation says that the sum of the three coordinates is one. The next two express x and y linearly as homogeneous forms in the triangular coordinates. These apply the interpolant formula (15.9) to the Cartesian coordinates: $x = x_1\zeta_1 + x_2\zeta_2 + x_3\zeta_3$ and $y = y_1\zeta_1 + y_2\zeta_2 + y_3\zeta_3$.

Inversion of (15.10) yields

$$\begin{bmatrix} \zeta_1 \\ \zeta_2 \\ \zeta_3 \end{bmatrix} = \frac{1}{2A} \begin{bmatrix} x_2y_3 - x_3y_2 & y_2 - y_3 & x_3 - x_2 \\ x_3y_1 - x_1y_3 & y_3 - y_1 & x_1 - x_3 \\ x_1y_2 - x_2y_1 & y_1 - y_2 & x_2 - x_1 \end{bmatrix} \begin{bmatrix} 1 \\ x \\ y \end{bmatrix} = \frac{1}{2A} \begin{bmatrix} 2A_{23} & y_{23} & x_{32} \\ 2A_{31} & y_{31} & x_{13} \\ 2A_{12} & y_{12} & x_{21} \end{bmatrix} \begin{bmatrix} 1 \\ x \\ y \end{bmatrix}. \quad (15.11)$$

Here $x_{jk} = x_j - x_k$, $y_{jk} = y_j - y_k$, A is the triangle area given by (15.3) and A_{jk} denotes the area subtended by corners j, k and the origin of the x - y system. If this origin is taken at the centroid of the triangle, $A_{23} = A_{31} = A_{12} = A/3$.

§15.2.4. Partial Derivatives

From equations (15.10) and (15.11) we immediately obtain the following relations between partial derivatives:

$$\frac{\partial x}{\partial \zeta_i} = x_i, \quad \frac{\partial y}{\partial \zeta_i} = y_i, \quad (15.12)$$

$$2A \frac{\partial \zeta_i}{\partial x} = y_{jk}, \quad 2A \frac{\partial \zeta_i}{\partial y} = x_{kj}. \quad (15.13)$$

In (15.13) j and k denote the *cyclic permutations* of i . For example, if $i = 2$, then $j = 3$ and $k = 1$.

The derivatives of a function $f(\zeta_1, \zeta_2, \zeta_3)$ with respect to x or y follow immediately from (15.13) and application of the chain rule:

$$\begin{aligned} \frac{\partial f}{\partial x} &= \frac{1}{2A} \left(\frac{\partial f}{\partial \zeta_1} y_{23} + \frac{\partial f}{\partial \zeta_2} y_{31} + \frac{\partial f}{\partial \zeta_3} y_{12} \right) \\ \frac{\partial f}{\partial y} &= \frac{1}{2A} \left(\frac{\partial f}{\partial \zeta_1} x_{32} + \frac{\partial f}{\partial \zeta_2} x_{13} + \frac{\partial f}{\partial \zeta_3} x_{21} \right) \end{aligned} \quad (15.14)$$

which in matrix form is

$$\begin{bmatrix} \frac{\partial f}{\partial x} \\ \frac{\partial f}{\partial y} \end{bmatrix} = \frac{1}{2A} \begin{bmatrix} y_{23} & y_{31} & y_{12} \\ x_{32} & x_{13} & x_{21} \end{bmatrix} \begin{bmatrix} \frac{\partial f}{\partial \zeta_1} \\ \frac{\partial f}{\partial \zeta_2} \\ \frac{\partial f}{\partial \zeta_3} \end{bmatrix}. \quad (15.15)$$

With these mathematical ingredients in place we are now in a position to handle the derivation of straight-sided triangular elements, and in particular the linear triangle.

§15.3. ELEMENT DERIVATION

The simplest triangular element for plane stress (and in general, for 2D problems of variational index $m = 1$) is the three-node triangle with *linear shape functions*. The shape functions are simply the triangular coordinates. That is, $N_i^{(e)} = \zeta_i$ for $i = 1, 2, 3$.

§15.3.1. Displacement Interpolation

For the plane stress problem we will select the linear interpolation (15.9) for the displacement components u_x and u_y at an arbitrary point $P(\zeta_1, \zeta_2, \zeta_3)$:

$$\begin{aligned} u_x &= u_{x1}\zeta_1 + u_{x2}\zeta_2 + u_{x3}\zeta_3, \\ u_y &= u_{y1}\zeta_1 + u_{y2}\zeta_2 + u_{y3}\zeta_3, \end{aligned} \quad (15.16)$$

as illustrated in Figure 15.3.

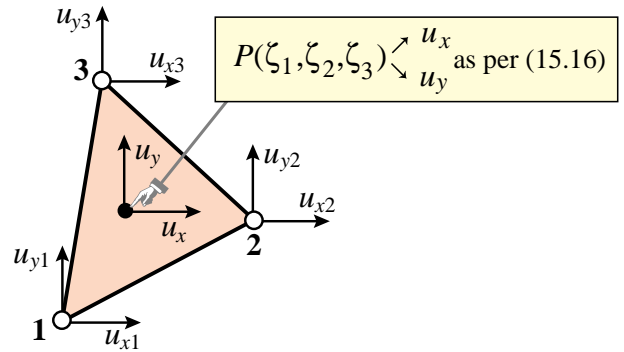


Figure 15.3. Displacement interpolation over triangle.

These relations can be combined in a matrix form that befits the general expression (14.17) for an arbitrary plane stress element:

$$\begin{bmatrix} u_x \\ u_y \end{bmatrix} = \begin{bmatrix} \zeta_1 & 0 & \zeta_2 & 0 & \zeta_3 & 0 \\ 0 & \zeta_1 & 0 & \zeta_2 & 0 & \zeta_3 \end{bmatrix} \begin{bmatrix} u_{x1} \\ u_{y1} \\ u_{x2} \\ u_{y2} \\ u_{x3} \\ u_{y3} \end{bmatrix} = \mathbf{N}^{(e)} \mathbf{u}^{(e)}. \quad (15.17)$$

§15.3.2. Strain-Displacement Equations

The strains within the elements are obtained by differentiating the shape functions with respect to x and y . Using (15.14) and the general form (14.18) we get

$$\mathbf{e} = \mathbf{D}\mathbf{N}^{(e)}\mathbf{u}^{(e)} = \frac{1}{2A} \begin{bmatrix} y_{23} & 0 & y_{31} & 0 & y_{12} & 0 \\ 0 & x_{32} & 0 & x_{13} & 0 & x_{21} \\ x_{32} & y_{23} & x_{13} & y_{31} & x_{21} & y_{12} \end{bmatrix} \begin{bmatrix} u_{x1} \\ u_{y1} \\ u_{x2} \\ u_{y2} \\ u_{x3} \\ u_{y3} \end{bmatrix} = \mathbf{B}\mathbf{u}^{(e)}, \quad (15.18)$$

in which \mathbf{D} denotes the symbolic strain-to-displacement differentiation operator given in (14.6).

Note that the strains are *constant* over the element. This is the origin of the name *constant strain triangle* (CST) given it in many finite element publications.

§15.3.3. Stress-Strain Equations

The stress field $\boldsymbol{\sigma}$ is related to the strain field by the elastic constitutive equation in (14.5), which is repeated here for convenience:

$$\boldsymbol{\sigma} = \begin{bmatrix} \sigma_{xx} \\ \sigma_{yy} \\ \sigma_{xy} \end{bmatrix} = \begin{bmatrix} E_{11} & E_{12} & E_{13} \\ E_{12} & E_{22} & E_{23} \\ E_{13} & E_{23} & E_{33} \end{bmatrix} \begin{bmatrix} e_{xx} \\ e_{yy} \\ 2e_{xy} \end{bmatrix} = \mathbf{E}\mathbf{e}, \quad (15.19)$$

where E_{ij} are plane stress elastic moduli. The constitutive matrix \mathbf{E} will be assumed to be constant over the element. Because the strains are constant, so are the stresses.

§15.3.4. The Stiffness Matrix

The element stiffness matrix is given by the general formula (14.23), which is repeated here for convenience:

$$\mathbf{K}^{(e)} = \int_{\Omega^{(e)}} h \mathbf{B}^T \mathbf{E} \mathbf{B} d\Omega^{(e)}, \quad (15.20)$$

where $\Omega^{(e)}$ is the triangle domain, and h is the plate thickness that appears in the plane stress problem. Since \mathbf{B} and \mathbf{E} are constant, they can be taken out of the integral:

$$\begin{aligned} \mathbf{K}^{(e)} &= \mathbf{B}^T \mathbf{E} \mathbf{B} \int_{\Omega^{(e)}} h d\Omega^{(e)} \\ &= \frac{1}{4A^2} \begin{bmatrix} y_{23} & 0 & x_{32} \\ 0 & x_{32} & y_{23} \\ y_{31} & 0 & x_{13} \\ 0 & x_{13} & y_{31} \\ y_{12} & 0 & x_{21} \\ 0 & x_{21} & y_{12} \end{bmatrix} \begin{bmatrix} E_{11} & E_{12} & E_{13} \\ E_{12} & E_{22} & E_{23} \\ E_{13} & E_{23} & E_{33} \end{bmatrix} \begin{bmatrix} y_{23} & 0 & y_{31} & 0 & y_{12} & 0 \\ 0 & x_{32} & 0 & x_{13} & 0 & x_{21} \\ x_{32} & y_{23} & x_{13} & y_{31} & x_{21} & y_{12} \end{bmatrix} \int_{\Omega^{(e)}} h d\Omega^{(e)}. \end{aligned} \quad (15.21)$$

If the thickness h is uniform over the element the integral in (15.21) is simply hA , and we obtain the closed form

$$\mathbf{K}^{(e)} = Ah \mathbf{B}^T \mathbf{E} \mathbf{B} = \frac{h}{4A} \begin{bmatrix} y_{23} & 0 & x_{32} \\ 0 & x_{32} & y_{23} \\ y_{31} & 0 & x_{13} \\ 0 & x_{13} & y_{31} \\ y_{12} & 0 & x_{21} \\ 0 & x_{21} & y_{12} \end{bmatrix} \begin{bmatrix} E_{11} & E_{12} & E_{13} \\ E_{12} & E_{22} & E_{23} \\ E_{13} & E_{23} & E_{33} \end{bmatrix} \begin{bmatrix} y_{23} & 0 & y_{31} & 0 & y_{12} & 0 \\ 0 & x_{32} & 0 & x_{13} & 0 & x_{21} \\ x_{32} & y_{23} & x_{13} & y_{31} & x_{21} & y_{12} \end{bmatrix}. \quad (15.22)$$

Exercise 15.1 deals with the case of a linearly varying thickness.

§15.3.5. The Consistent Nodal Force Vector

For simplicity we consider here only internal body forces² defined by the vector field

$$\mathbf{b} = \begin{bmatrix} b_x \\ b_y \end{bmatrix} \quad (15.23)$$

which is specified per unit of volume. The consistent nodal force vector $\mathbf{f}^{(e)}$ is given by the general formula (14.23) of the previous Chapter:

$$\mathbf{f}^{(e)} = \int_{\Omega^{(e)}} h (\mathbf{N}^{(e)})^T \mathbf{b} d\Omega^{(e)} = \int_{\Omega^{(e)}} h \begin{bmatrix} \zeta_1 & 0 \\ 0 & \zeta_1 \\ \zeta_2 & 0 \\ 0 & \zeta_2 \\ \zeta_3 & 0 \\ 0 & \zeta_3 \end{bmatrix} \mathbf{b} d\Omega^{(e)}. \quad (15.24)$$

The simplest case is when the body force components (15.23) as well as the thickness h are constant over the element. Then we need the integrals

$$\int_{\Omega^{(e)}} \zeta_1 d\Omega^{(e)} = \int_{\Omega^{(e)}} \zeta_2 d\Omega^{(e)} = \int_{\Omega^{(e)}} \zeta_3 d\Omega^{(e)} = \frac{1}{3}A \quad (15.25)$$

which replaced into (15.24) gives

$$\mathbf{f}^{(e)} = \frac{Ah}{3} \begin{bmatrix} b_x \\ b_y \\ b_x \\ b_y \\ b_x \\ b_y \end{bmatrix}. \quad (15.26)$$

This agrees with the simple force-lumping procedure, which assigns one third of the total force along the $\{x, y\}$ directions: Ahb_x and Ahb_y , to each corner.

² For consistent force computations corresponding to distributed boundary loads over a side, see Exercise 15.4.

REMARK 15.2

The integral (15.25) is a particular case of the general integration formula of monomials in triangular coordinates:

$$\frac{1}{2A} \int_{\Omega^{(e)}} \zeta_1^i \zeta_2^j \zeta_3^k d\Omega^{(e)} = \frac{i! j! k!}{(i + j + k + 2)!}, \quad i \geq 0, j \geq 0, k \geq 0. \quad (15.27)$$

which can be proven by recursive integration by parts. This formula *only holds for triangles with straight sides*, and consequently is useless for higher order curved elements. The result (15.25) is obtained by setting $i = 1, j = k = 0$ in (15.27).

§15.4. *CONSISTENCY VERIFICATION

It remains to check whether the piecewise linear expansion (15.16) for the element displacements meets the completeness and continuity criteria studied in more detail in Chapter 19 for finite element trial functions. Such *consistency* conditions are sufficient to insure convergence towards the exact solution of the mathematical model as the mesh is refined.

The variational index for the plane stress problem is $m = 1$. Consequently the trial functions should be 1-complete, C^0 continuous, and C^1 piecewise differentiable.

§15.4.1. *Checking Continuity

Along any triangle side, the variation of u_x and u_y is *linear and uniquely determined by the value at the nodes on that side*. For example, over the side 1–2 of an individual triangle:

$$\begin{aligned} u_x &= u_{x1}\zeta_1 + u_{x2}\zeta_2 + u_{x3}\zeta_3 = u_{x1}\zeta_1 + u_{x2}\zeta_2, \\ u_y &= u_{y1}\zeta_1 + u_{y2}\zeta_2 + u_{y3}\zeta_3 = u_{y1}\zeta_1 + u_{y2}\zeta_2, \end{aligned} \quad (15.28)$$

because $\zeta_3 = 0$ along that side.

An identical argument holds for that side when it belongs to an adjacent triangle, such as elements (e1) and (e2) shown in Figure 15.4. Since the node values on all elements that meet at a node are the same, u_x and u_y match along the side, and the trial function is C^0 continuous across elements. Because the functions are continuous inside the elements, it follows that the conformity requirement is met.

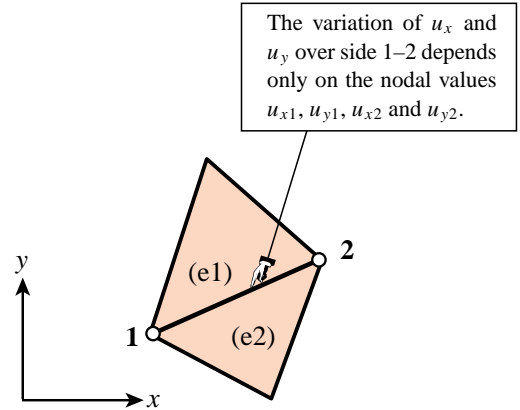


Figure 15.4. Interelement continuity check.

§15.4.2. *Checking Completeness

The completeness condition for variational order $m = 1$ require that the shape functions $N_i = \zeta_i$ be able to represent exactly any linear displacement field:

$$u_x = \alpha_0 + \alpha_1 x + \alpha_2 y, \quad u_y = \beta_0 + \beta_1 x + \beta_2 y. \quad (15.29)$$

To check this we obtain the nodal values associated with the motion (15.29)

$$\left. \begin{aligned} u_{xi} &= \alpha_0 + \alpha_1 x_i + \alpha_2 y_i \\ u_{yi} &= \beta_0 + \beta_1 x_i + \beta_2 y_i \end{aligned} \right\} \quad i = 1, 2, 3, \quad (15.30)$$

replace them into (15.17) and see if we recover (15.29). Here are the detailed calculations for component u_x :

$$\begin{aligned} u_x &= \sum_i u_{xi} \zeta_i = \sum_i (\alpha_0 + \alpha_1 x_i + \alpha_2 y_i) \zeta_i = \sum_i (\alpha_0 \zeta_i + \alpha_1 x_i \zeta_i + \alpha_2 y_i \zeta_i) \\ &= \alpha_0 \sum_i \zeta_i + \alpha_1 \sum_i (x_i \zeta_i) + \alpha_2 \sum_i (y_i \zeta_i) = \alpha_0 + \alpha_1 x + \alpha_2 y. \end{aligned} \quad (15.31)$$

Component u_y can be similarly checked. Consequently (15.17) satisfies the completeness requirement for the plane stress problem (and in general, for any problem of variational index 1).

Finally, a piecewise linear trial function is obviously C^1 piecewise differentiable and consequently has finite energy. Thus the two continuity requirements are satisfied.

§15.5. *THE TONTI DIAGRAM OF THE LINEAR TRIANGLE

For further developments covered in advanced FEM course, it is convenient to split the governing equations of the element. In the case of the linear triangle they are, omitting element superscripts:

$$\epsilon = \mathbf{B}\mathbf{u}, \quad \sigma = \mathbf{E}\epsilon, \quad \mathbf{f} = \mathbf{A}^T \sigma = V \mathbf{B}^T \sigma. \quad (15.32)$$

in which $V = h_m A$ is the volume of the element, h_m being the mean thickness.

The equations (15.32) may be represented with the matrix diagram shown in Figure 15.5.

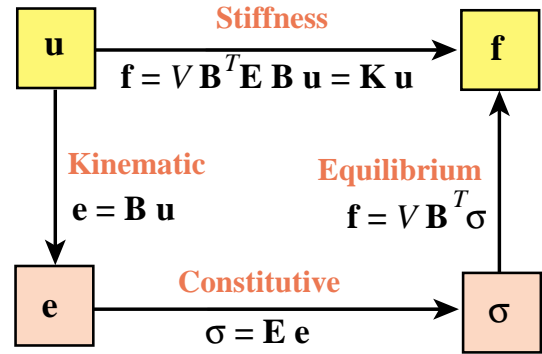


Figure 15.5. Tonti matrix diagram for linear triangle.

§15.6. *DERIVATION USING NATURAL STRAINS AND STRESSES

The element derivation in §15.3 uses Cartesian strains and stresses, as well as $x - y$ displacements. The only intrinsic quantities are the triangle coordinates. This subsection examines the derivation of the element stiffness matrix through natural strains, natural stresses and covariant displacements. Although the procedure does not offer obvious shortcuts over the previous derivation, it becomes important in the construction of more complicated high performance elements. It also helps reading recent literature.

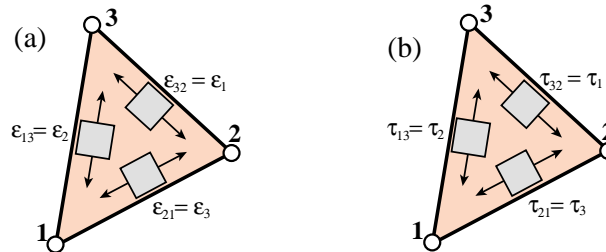


Figure 15.6. Geometry-intrinsic strain and stress fields for the 3-node linear triangle: (a) natural strains ϵ_i , (b) natural stresses τ_i .

§15.6.1. *Natural Strains and Stresses

Natural strains are extensional strains directed parallel to the triangle sides, as shown in Figure 15.6(a). They are denoted by $\epsilon_{21} \equiv \epsilon_3$, $\epsilon_{32} \equiv \epsilon_1$, and $\epsilon_{13} \equiv \epsilon_2$. Because they are constant over the triangle, no node value association is needed.

Natural stresses are normal stresses directed parallel to the triangle sides, as shown in Figure 15.6(b). They are denoted by $\tau_{21} \equiv \tau_3$, $\tau_{32} \equiv \tau_1$, and $\tau_{13} \equiv \tau_2$. Because they are constant over the triangle, no node value association is needed.

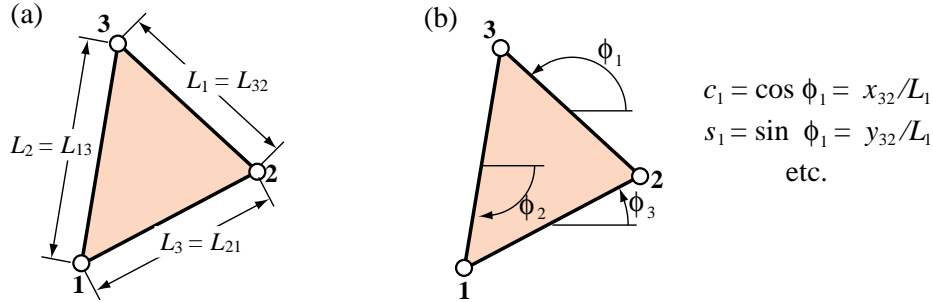


Figure 15.7. Quantities appearing in natural strain and stress calculations: (a) side lengths, (b) side directions.

The natural strains can be related to Cartesian strains by the tensor transformation³

$$\boldsymbol{\epsilon} = \begin{bmatrix} \epsilon_1 \\ \epsilon_2 \\ \epsilon_3 \end{bmatrix} = \begin{bmatrix} c_1^2 & s_1^2 & s_1 c_1 \\ c_2^2 & s_2^2 & s_2 c_2 \\ c_3^2 & s_3^2 & s_3 c_3 \end{bmatrix} \begin{bmatrix} e_{xx} \\ e_{yy} \\ 2e_{xy} \end{bmatrix} = \mathbf{T}_e^{-1} \mathbf{e}. \quad (15.33)$$

Here $c_1 = x_{32}/L_1$, $s_1 = y_{32}/L_1$, $c_2 = x_{13}/L_2$, $s_2 = y_{13}/L_2$, $c_3 = x_{21}/L_3$, and $s_3 = y_{21}/L_3$, are sine/cosines of the side directions with respect to $\{x, y\}$, as illustrated in Figure 15.7. The inverse of this relation is

$$\mathbf{e} = \begin{bmatrix} e_{xx} \\ e_{yy} \\ 2e_{xy} \end{bmatrix} = \frac{1}{4A^2} \begin{bmatrix} y_{31}y_{21}L_1^2 & y_{12}y_{32}L_2^2 & y_{23}y_{13}L_3^2 \\ x_{31}x_{21}L_1^2 & x_{12}x_{32}L_2^2 & x_{23}x_{13}L_3^2 \\ (y_{31}x_{12} + x_{13}y_{21})L_1^2 & (y_{12}x_{23} + x_{21}y_{32})L_2^2 & (y_{23}x_{31} + x_{32}y_{13})L_3^2 \end{bmatrix} \begin{bmatrix} \epsilon_1 \\ \epsilon_2 \\ \epsilon_3 \end{bmatrix} = \mathbf{T}_e \boldsymbol{\epsilon}. \quad (15.34)$$

Note that \mathbf{T}_e is constant over the triangle.

From the invariance of the strain energy density $\boldsymbol{\sigma}^T \mathbf{e} = \boldsymbol{\tau}^T \boldsymbol{\epsilon}$ it follows that the stresses transform as $\boldsymbol{\tau} = \mathbf{T}_e \boldsymbol{\sigma}$ and $\boldsymbol{\sigma} = \mathbf{T}_e^{-1} \boldsymbol{\tau}$. That strain energy density may be expressed as

$$\mathcal{U} = \frac{1}{2} \mathbf{e}^T \mathbf{E} \mathbf{e} = \frac{1}{2} \boldsymbol{\epsilon}^T \mathbf{E}_n \boldsymbol{\epsilon}, \quad \mathbf{E}_n = \mathbf{T}_e^T \mathbf{E} \mathbf{T}_e. \quad (15.35)$$

Here \mathbf{E}_n is a stress-strain matrix that relates natural stresses to natural strains as $\boldsymbol{\tau} = \mathbf{E}_n \boldsymbol{\epsilon}$. It may be therefore called the natural constitutive matrix.

³ This is the “strainage rosette” transformation studied in Mechanics of Materials books.

§15.6.2. *Covariant Node Displacements

Covariant node displacements d_i are directed along the side directions, as shown in Figure 15.8, which defines the notation used for them. They are related to the Cartesian node displacements by

$$\mathbf{d} = \begin{bmatrix} d_1 \\ d_2 \\ d_3 \\ d_4 \\ d_5 \\ d_6 \end{bmatrix} = \begin{bmatrix} c_3 & s_3 & 0 & 0 & 0 & 0 \\ c_2 & s_2 & 0 & 0 & 0 & 0 \\ 0 & 0 & c_1 & s_1 & 0 & 0 \\ 0 & 0 & c_3 & s_3 & 0 & 0 \\ 0 & 0 & 0 & 0 & c_2 & s_2 \\ 0 & 0 & 0 & 0 & c_1 & s_1 \end{bmatrix} \begin{bmatrix} u_{x1} \\ u_{y1} \\ u_{x2} \\ u_{y2} \\ u_{x3} \\ u_{y3} \end{bmatrix} = \mathbf{T}_d \mathbf{u}. \quad (15.36)$$

The inverse relation is

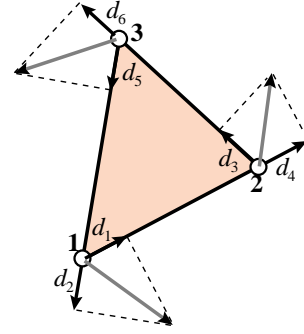


Figure 15.8. Covariant node displacements d_i .

$$\mathbf{u} = \begin{bmatrix} u_{x1} \\ u_{y1} \\ u_{x2} \\ u_{y2} \\ u_{x3} \\ u_{y3} \end{bmatrix} = \frac{1}{2A} \begin{bmatrix} L_3 y_{31} & L_2 y_{21} & 0 & 0 & 0 & 0 \\ L_3 x_{13} & L_2 x_{12} & 0 & 0 & 0 & 0 \\ 0 & 0 & L_1 y_{12} & L_3 y_{32} & 0 & 0 \\ 0 & 0 & L_1 x_{21} & L_3 x_{23} & 0 & 0 \\ 0 & 0 & 0 & 0 & L_2 y_{23} & L_1 y_{13} \\ 0 & 0 & 0 & 0 & L_2 x_{32} & L_1 x_{31} \end{bmatrix} \begin{bmatrix} d_1 \\ d_2 \\ d_3 \\ d_4 \\ d_5 \\ d_6 \end{bmatrix} = \mathbf{T}_d^{-1} \mathbf{d}. \quad (15.37)$$

The natural strains are evidently given by the relations $\epsilon_1 = (d_6 - d_3)/L_1$, $\epsilon_2 = (d_2 - d_5)/L_2$ and $\epsilon_3 = (d_4 - d_1)/L_3$. Collecting these in matrix form:

$$\boldsymbol{\epsilon} = \begin{bmatrix} \epsilon_1 \\ \epsilon_2 \\ \epsilon_3 \end{bmatrix} = \begin{bmatrix} 0 & 0 & -1/L_1 & 0 & 0 & 1/L_1 \\ 0 & 1/L_2 & 0 & 0 & -1/L_2 & 0 \\ -1/L_3 & 0 & 0 & 1/L_3 & 0 & 0 \end{bmatrix} \begin{bmatrix} d_1 \\ d_2 \\ d_3 \\ d_4 \\ d_5 \\ d_6 \end{bmatrix} = \mathbf{B}_\epsilon \mathbf{d}. \quad (15.38)$$

§15.6.3. *The Natural Stiffness Matrix

The natural stiffness matrix for constant h is

$$\mathbf{K}_n = (Ah) \mathbf{B}_\epsilon^T \mathbf{E}_n \mathbf{B}_\epsilon, \quad \mathbf{E}_n = \mathbf{T}_e^T \mathbf{E} \mathbf{T}_e. \quad (15.39)$$

The Cartesian stiffness matrix is

$$\mathbf{K} = \mathbf{T}_d^T \mathbf{K}_n \mathbf{T}_d. \quad (15.40)$$

Comparing with $\mathbf{K} = (Ah) \mathbf{B}^T \mathbf{E} \mathbf{B}$ we see that

$$\mathbf{B} = \mathbf{T}_e \mathbf{B}_\epsilon \mathbf{T}_d, \quad \mathbf{B}_\epsilon = \mathbf{T}_e^{-1} \mathbf{B} \mathbf{T}_d^{-1}. \quad (15.41)$$

Notes and Bibliography

The linear triangle, as a structural element, was first developed in the 1956 paper by Turner, Clough, Martin and Topp [15.8]. The envisioned application was modeling of skin panels of delta wings. Because of its geometric flexibility, the element was soon adopted in aircraft structural analysis codes in the late 1950's. It moved to Civil Engineering applications through the research and teaching of Ray Clough [15.3] at Berkeley.

The method of [15.8] would look unfamiliar to present FEM practitioners used to the displacement method. It was based on assumed stress modes. More precisely: the element, referred to a local Cartesian system $\{x, y\}$, is put under three constant stress states: σ_{xx} , σ_{yy} and σ_{xy} collected in array $\boldsymbol{\sigma}$. Lumping the stress field to the nodes gives the node forces: $\mathbf{f} = \mathbf{L}\boldsymbol{\sigma}$. The strain field computed from stresses is $\mathbf{e} = \mathbf{E}^{-1}\boldsymbol{\sigma}$. This is integrated to get a deformation displacement field, to which 3 rigid body modes are added as integration constants. Evaluating at the nodes produces $\mathbf{e} = \mathbf{A}\mathbf{u}$, and the stiffness matrix follows on eliminating $\boldsymbol{\sigma}$ and \mathbf{e} : $\mathbf{K} = \mathbf{L}\mathbf{E}\mathbf{A}$. It happens that $\mathbf{L} = \mathbf{V}\mathbf{A}^T$ and so $\mathbf{K} = \mathbf{V}\mathbf{A}^T\mathbf{E}\mathbf{A}$ happily turned out to be symmetric. (This \mathbf{A} is the \mathbf{B} of (15.13) times $2A$.)

The derivation from assumed displacements evolved slowly, and it is not clear who presented it first. The equivalence, through energy principles, had been noted by Gallagher [15.6]. Early displacement derivations typically started by starting from linear polynomials in the Cartesian coordinates. For example Przemieniecki [15.7] begins with

$$u_x = c_1x + c_2y + c_3, \quad u_y = c_4x + c_5y + c_6. \quad (15.42)$$

Here the c_i play the role of generalized coordinates, which have to be eventually eliminated in favor of node displacements. The same approach is used by Clough in a widely disseminated 1965 article [15.4]. Even for this simple element the approach is unnecessarily complicated and leads to long computations. The elegant derivation in natural coordinates was popularized by Argyris [15.2].

The idea of using linear interpolation over a triangular net precedes [15.8] by 13 years. It appears in the Appendix of an article by Courant [15.5], where it is applied to a Poisson's equation modeling St. Venant's torsion. The idea did not influence early work in FEM, however, since as noted the original derivation in [15.8] was not based on displacements.

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Homework Exercises for Chapter 15

The Linear Plane Stress Triangle

EXERCISE 15.1

[A:15] Assume that the 3-node plane stress triangle has *variable* thickness defined over the element by the linear interpolation formula

$$h(\zeta_1, \zeta_2, \zeta_3) = h_1\zeta_1 + h_2\zeta_2 + h_3\zeta_3, \quad (\text{E15.1})$$

where h_1 , h_2 and h_3 are the thicknesses at the corner nodes. Show that the element stiffness matrix is still given by (15.23) but with h replaced by the mean thickness $h_m = (h_1 + h_2 + h_3)/3$. *Hint:* use (15.22) and (15.28).

EXERCISE 15.2

[A:20] The exact integrals of triangle-coordinate monomials over a straight-sided triangle are given by the formula

$$\frac{1}{2A} \int_A \zeta_1^i \zeta_2^j \zeta_3^k dA = \frac{i! j! k!}{(i + j + k + 2)!} \quad (\text{E15.2})$$

where A denotes the area of the triangle, and i , j and k are nonnegative integers. Tabulate the right-hand side for combinations of exponents i , j and k such that $i + j + k \leq 3$, beginning with $i = j = k = 0$. Remember that $0! = 1$. (*Labor-saving hint:* don't bother repeating exponent permutations; for example $i = 2, j = 1, k = 0$ and $i = 1, j = 2, k = 0$ are permutations of the same thing. Hence one needs to tabulate only cases in which $i \geq j \geq k$).

EXERCISE 15.3

[A/C:20] Compute the consistent node force vector $\mathbf{f}^{(e)}$ for body loads over a linear triangle, if the element thickness varies as per (E15.1), $b_x = 0$, and $b_y = b_{y1}\zeta_1 + b_{y2}\zeta_2 + b_{y3}\zeta_3$. Check that for $h_1 = h_2 = h_3 = h$ and $b_{y1} = b_{y2} = b_{y3} = b_y$ you recover (15.26). For the integrals over the triangle area use the formula (E15.2).

Partial result: $f_{y1} = (A/60)[b_{y1}(6h_1 + 2h_2 + 2h_3) + b_{y2}(2h_1 + 2h_2 + h_3) + b_{y3}(2h_1 + h_2 + 2h_3)]$.

EXERCISE 15.4

[A/C:20] Derive the formula for the consistent force vector $\mathbf{f}^{(e)}$ of a linear triangle of constant thickness h , if side 1-2 ($\zeta_3 = 0$, $\zeta_2 = 1 - \zeta_1$), is subject to a linearly varying boundary force $\mathbf{q} = h\hat{\mathbf{t}}$ such that

$$q_x = q_{x1}\zeta_1 + q_{x2}\zeta_2 = q_{x1}(1 - \zeta_2) + q_{x2}\zeta_2, \quad q_y = q_{y1}\zeta_1 + q_{y2}\zeta_2 = q_{y1}(1 - \zeta_2) + q_{y2}\zeta_2. \quad (\text{E15.3})$$

This “line force” \mathbf{q} has dimension of force per unit of side length.

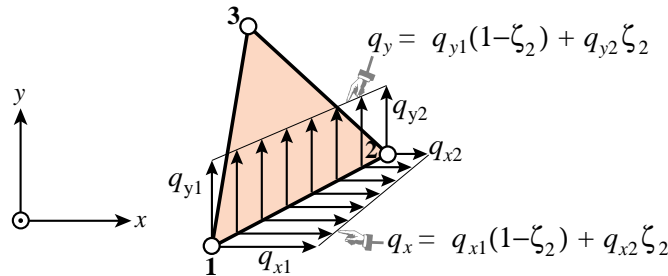


Figure E15.1. Line forces on triangle side 1-2 for Exercise 15.4.

Procedure. Use the last term of the line integral (14.21), in which $\hat{\mathbf{t}}$ is replaced by \mathbf{q}/h , and show that since the contribution of sides 2-3 and 3-1 to the line integral vanish,

$$W^{(e)} = (\mathbf{u}^{(e)})^T \mathbf{f}^{(e)} = \int_{\Gamma^{(e)}} \mathbf{u}^T \mathbf{q} d\Gamma^{(e)} = \int_0^1 \mathbf{u}^T \mathbf{q} L_{21} d\zeta_2, \quad (\text{E15.4})$$

where L_{21} is the length of side 1-2. Replace $u_x(\zeta_2) = u_{x1}(1 - \zeta_2) + u_{x2}\zeta_2$; likewise for u_y , q_x and q_y , integrate and identify with the inner product shown as the second term in (E15.4). Partial result: $f_{x1} = L_{21}(2q_{x1} + q_{x2})/6$, $f_{x3} = f_{y3} = 0$. *Note.* The following *Mathematica* script solves this Exercise. If you decide to use it, explain the logic.

```
ClearAll[ux1,uy1,ux2,uy2,ux3,uy3,z2,L12];
ux=ux1*(1-z2)+ux2*z2; uy=uy1*(1-z2)+uy2*z2;
qx=qx1*(1-z2)+qx2*z2; qy=qy1*(1-z2)+qy2*z2;
We=Simplify[L12*Integrate[qx*ux+qy*uy,{z2,0,1}]];
fe=Table[Coefficient[We,{ux1,uy1,ux2,uy2,ux3,uy3}[[i]]],{i,1,6}];
fe=Simplify[fe]; Print["fe=",fe];
```

EXERCISE 15.5

[C+N:15] Compute the entries of $\mathbf{K}^{(e)}$ for the following plane stress triangle:

$$x_1 = 0, y_1 = 0, x_2 = 3, y_2 = 1, x_3 = 2, y_3 = 2, \quad \mathbf{E} = \begin{bmatrix} 100 & 25 & 0 \\ 25 & 100 & 0 \\ 0 & 0 & 50 \end{bmatrix}, \quad h = 1. \quad (\text{E15.5})$$

This may be done by hand (it is a good exercise in matrix multiplication) or (more quickly) using the following *Mathematica* script:

```
Stiffness3NodePlaneStressTriangle[{{x1_,y1_},{x2_,y2_},{x3_,y3_}},
  Emat_,{h_}]:=Module[{x21,x13,x32,y12,y31,y23,A,Be,Ke},
  A=Simplify[(x2*y3-x3*y2+(x3*y1-x1*y3)+(x1*y2-x2*y1))/2];
  {x21,x13,x32}={x2-x1,x1-x3,x3-x2};
  {y12,y31,y23}={y1-y2,y3-y1,y2-y3};
  Be={{y23,0,y31,0,y12,0},{0,x32,0,x13,0,x21},
    {x32,y23,x13,y31,x21,y12}}/(2*A);
  Ke=A*h*Transpose[Be].Emat.Be;Return[Ke];

Ke=Stiffness3NodePlaneStressTriangle[{{0,0},{3,1},{2,2}},
  {{100,25,0},{25,100,0},{0,0,50}},{1}];
Print["Ke=",Ke//MatrixForm];
Print["eigs of Ke=",Chop[Eigenvalues[N[Ke]]]];
Show[Graphics[Line[{{0,0},{3,1},{2,2},{0,0}}]],Axes->True];
```

Check it out: $K_{11} = 18.75$, $K_{66} = 118.75$. The last statement draws the triangle.

EXERCISE 15.6

[A+C:15] Show that the sum of the rows (and columns) 1, 3 and 5 of $\mathbf{K}^{(e)}$ as well as the sum of rows (and columns) 2, 4 and 6 must vanish. Check it with the foregoing script.

EXERCISE 15.7

[A/C:30] Let $p(\zeta_1, \zeta_2, \zeta_3)$ represent a *polynomial* expression in the natural coordinates. The integral

$$\int_{\Omega^{(e)}} p(\zeta_1, \zeta_2, \zeta_3) d\Omega \quad (\text{E15.6})$$

over a straight-sided triangle can be computed symbolically by the following *Mathematica* module:

```
IntegrateOverTriangle[expr_, tcoord_, A_, max_] := Module[{p, i, j, k, z1, z2, z3, c, s = 0},
  p = Expand[expr]; {z1, z2, z3} = tcoord;
  For[i = 0, i <= max, i++, For[j = 0, j <= max, j++, For[k = 0, k <= max, k++,
    c = Coefficient[Coefficient[Coefficient[p, z1, i], z2, j], z3, k];
    s += 2*c*(i!*j!*k!)/((i+j+k+2)!);
  ]]];
  Return[Simplify[A*s]]];
```

This is referenced as `int=IntegrateOverTriangle[p,{z1,z2,z3},A,max]`. Here p is the polynomial to be integrated, $z1$, $z2$ and $z3$ denote the symbols used for the triangular coordinates, A is the triangle area and max the highest exponent appearing in a triangular coordinate. The module name returns the integral. For example, if $p=16+5*b*z2^2+z1^3+z2*z3*(z2+z3)$ the call `int=IntegrateOverTriangle[p,{z1,z2,z3},A,3]` returns `int=A*(97+5*b)/6`. Explain how the module works.

EXERCISE 15.8

[C+D:25] Access the file `Trig3PlaneStress.nb` from the course Web site by clicking on the appropriate link in Chapter 15 Index. This is a *Mathematica* 4.1 Notebook that does plane stress FEM analysis using the 3-node linear triangle.

Download the Notebook into your directory. Load into *Mathematica*. Execute the top 7 input cells (which are actually initialization cells) so the necessary modules are compiled. Each cell is preceded by a short comment cell which outlines the purpose of the modules it holds. Notes: (1) the plot-module cell may take a while to run through its tests; be patient; (2) to get rid of unsightly messages and silly beeps about similar names, initialize each cell twice.

After you are satisfied everything works fine, run the cantilever beam problem, which is defined in the last input cell.

After you get a feel of how this code operate, study the source. Prepare a hierarchical diagram of the modules,⁴ beginning with the main program of the last cell. Note which calls what, and briefly explain the purpose of each module. Return this diagram as answer to the homework. You do not need to talk about the actual run and results; those will be discussed in Part III.

Hint: a hierarchical diagram for `Trig3PlaneStress.nb` begins like

⁴ A hierarchical diagram is a list of modules and their purposes, with indentation to show dependence, similar to the table of contents of a book. For example, if module AAAA calls BBBB and CCCC, and BBBB calls DDDD, the hierarchical diagram may look like:

```
AAAA - purpose of AAAA
  BBBB - purpose of BBBB
    DDDD - purpose of DDDD
  CCCC - purpose of CCCC
```

```
Main program in Cell 8 - drives the FEM analysis
  GenerateNodes - generates node coordinates of regular mesh
  GenerateTriangles - generate element node lists of regular mesh
  .....
```

16

The Isoparametric Representation

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§16.1. INTRODUCTION

The technique used in Chapter 15 for the linear triangle can be formally extended to quadrilateral elements as well as higher order triangles. But it quickly runs into two technical difficulties:

1. The construction of shape functions that satisfy consistency requirements for higher order elements with curved boundaries becomes increasingly complicated.
2. Integrals that appear in the expressions of the element stiffness matrix and consistent nodal force vector can no longer be carried out in closed form.

These difficulties can be overcome through the concepts of *isoparametric elements* and *numerical quadrature*, respectively. The combination of these two ideas transformed the field of finite element methods in the late 1960s. Together they support a good portion of what is presently used in production finite element programs.

In the present Chapter the concept of isoparametric representation is introduced for two dimensional elements. This representation is illustrated on specific elements. In the next Chapter these techniques, combined with numerical integration, are applied to quadrilateral elements.

§16.2. ISOPARAMETRIC REPRESENTATION

§16.2.1. Motivation

The linear triangle presented in Chapter 15 is an isoparametric element although was not originally derived as such. The two key equations are (15.10), which defines the triangle geometry, and (15.16), which defines the primary variable, in this case the displacement field. These equations are reproduced here for convenience:

$$\begin{bmatrix} 1 \\ x \\ y \end{bmatrix} = \begin{bmatrix} 1 & 1 & 1 \\ x_1 & x_2 & x_3 \\ y_1 & y_2 & y_3 \end{bmatrix} \begin{bmatrix} \zeta_1 \\ \zeta_2 \\ \zeta_3 \end{bmatrix}, \quad (16.1)$$

$$\begin{aligned} u_x &= u_{x1}N_1^{(e)} + u_{x2}N_2^{(e)} + u_{x3}N_3^{(e)} = u_{x1}\zeta_1 + u_{x2}\zeta_2 + u_{x3}\zeta_3 \\ u_y &= u_{y1}N_1^{(e)} + u_{y2}N_2^{(e)} + u_{y3}N_3^{(e)} = u_{y1}\zeta_1 + u_{y2}\zeta_2 + u_{y3}\zeta_3 \end{aligned} \quad (16.2)$$

The interpretation of these equations is as follows. The triangular coordinates define the element geometry via (16.1). The displacement expansion (16.2) is defined by the shape functions, which are in turn expressed in terms of the triangular coordinates. For the linear triangle, shape functions and triangular coordinates coalesce

These relations are diagrammed in Figure 16.1. Evidently the element geometry and element displacements are not treated equally. If we proceed to higher order triangles with straight sides, only the displacement expansion is refined whereas the geometry definition remains the same.

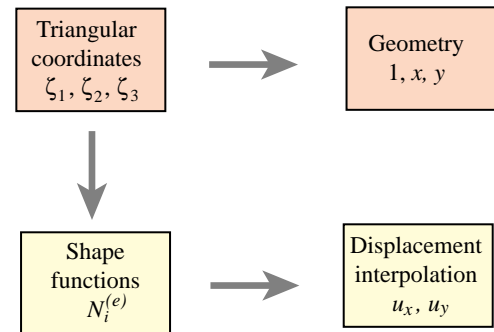


Figure 16.1. Superparametric representation of triangular element.

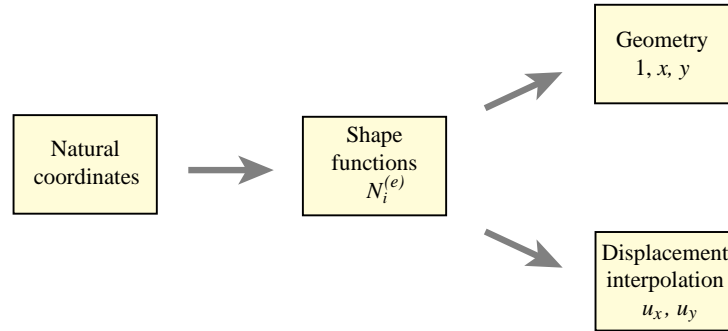


Figure 16.3. Isoparametric representation of arbitrary 2D elements: triangles or quadrilaterals. For 3D elements, expand the geometry list to $\{1, x, y, z\}$ and the displacements to $\{u_x, u_y, u_z\}$.

Elements built according to the foregoing prescription are called *superparametric*, a term that emphasizes that unequal treatment.

§16.2.2. Equalizing Geometry and Displacements

On first inspection (16.1) and (16.2) do not look alike. Their inherent similarity can be displayed, however, if the second one is rewritten and adjoined to (16.1) to look as follows:

$$\begin{bmatrix} 1 \\ x \\ y \\ u_x \\ u_y \end{bmatrix} = \begin{bmatrix} 1 & 1 & 1 \\ x_1 & x_2 & x_3 \\ y_1 & y_2 & y_3 \\ u_{x1} & u_{x2} & u_{y3} \\ u_{y1} & u_{y2} & u_{y3} \end{bmatrix} \begin{bmatrix} \zeta_1 \\ \zeta_2 \\ \zeta_3 \end{bmatrix} = \begin{bmatrix} 1 & 1 & 1 \\ x_1 & x_2 & x_3 \\ y_1 & y_2 & y_3 \\ u_{x1} & u_{x2} & u_{y3} \\ u_{y1} & u_{y2} & u_{y3} \end{bmatrix} \begin{bmatrix} N_1^{(e)} \\ N_2^{(e)} \\ N_3^{(e)} \end{bmatrix}. \quad (16.3)$$

This form emphasizes that geometry and element displacements are given by the *same* parametric representation, as illustrated in Figure 16.2.

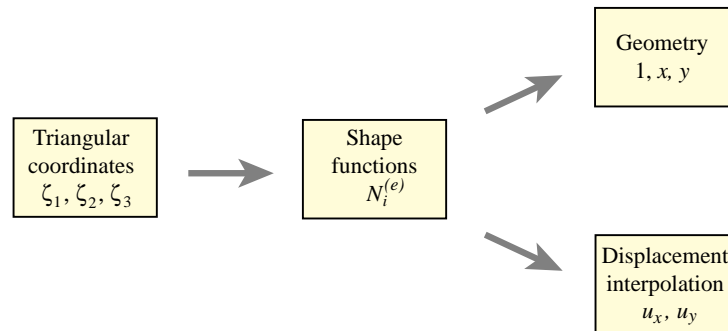


Figure 16.2. Isoparametric representation of triangular elements.

The key idea is to use the shape functions to represent *both the element geometry and the problem unknowns*, which in structural mechanics are displacements. Hence the name *isoparametric element* (“iso” means equal), often abbreviated to *iso-P element*. This property may be generalized to arbitrary elements by replacing the term “triangular coordinates” by the more general one “natural coordinates” as illustrated in Figure 16.3.

Under this generalization, natural coordinates (triangular coordinates for triangles, quadrilateral coordinates for quadrilaterals) appear as *parameters* that define the shape functions. The shape functions connect the geometry with the displacements.

REMARK 16.1

The terms *isoparametric* and *superparametric* were introduced by Irons and coworkers at Swansea in 1966. See **Notes and Bibliography** at the end of this Chapter. There are also *subparametric* elements whose geometry is more refined than the displacement expansion.

§16.3. GENERAL ISOPARAMETRIC FORMULATION

The generalization of (16.3) to an arbitrary two-dimensional element with n nodes is straightforward. Two set of relations, one for the element geometry and the other for the element displacements, are required. Both sets exhibit the same interpolation in terms of the shape functions.

Geometric relations:

$$1 = \sum_{i=1}^n N_i^{(e)}, \quad x = \sum_{i=1}^n x_i N_i^{(e)}, \quad y = \sum_{i=1}^n y_i N_i^{(e)}. \quad (16.4)$$

Displacement interpolation:

$$u_x = \sum_{i=1}^n u_{xi} N_i^{(e)}, \quad u_y = \sum_{i=1}^n u_{yi} N_i^{(e)}. \quad (16.5)$$

These two sets of equations may be combined in matrix form as

$$\begin{bmatrix} 1 \\ x \\ y \\ u_x \\ u_y \end{bmatrix} = \begin{bmatrix} 1 & 1 & \dots & 1 \\ x_1 & x_2 & \dots & x_n \\ y_1 & y_2 & \dots & y_n \\ u_{x1} & u_{x2} & \dots & u_{xn} \\ u_{y1} & u_{y2} & \dots & u_{yn} \end{bmatrix} \begin{bmatrix} N_1^{(e)} \\ N_2^{(e)} \\ \vdots \\ N_n^{(e)} \end{bmatrix}. \quad (16.6)$$

The first three equations express the geometry definition, and the last two the displacement expansion. Note that additional rows may be added to this matrix expression if additional variables are interpolated by the same shape functions. For example, suppose that the thickness h and a temperature field T is interpolated from the n node values:

$$\begin{bmatrix} 1 \\ x \\ y \\ u_x \\ u_y \\ h \\ T \end{bmatrix} = \begin{bmatrix} 1 & 1 & \dots & 1 \\ x_1 & x_2 & \dots & x_n \\ y_1 & y_2 & \dots & y_n \\ u_{x1} & u_{x2} & \dots & u_{xn} \\ u_{y1} & u_{y2} & \dots & u_{yn} \\ h_1 & h_2 & \dots & h_n \\ T_1 & T_2 & \dots & T_n \end{bmatrix} \begin{bmatrix} N_1^{(e)} \\ N_2^{(e)} \\ \vdots \\ N_n^{(e)} \end{bmatrix}. \quad (16.7)$$

Note that the column of shape functions is not touched.

To illustrate the use of the isoparametric concept, we take a look at specific 2D isoparametric elements that are commonly used in structural and non-structural applications. These are separated into triangles and quadrilaterals because they use different natural coordinates.

§16.4. TRIANGULAR ELEMENTS

§16.4.1. The Linear Triangle

The three-noded linear triangle studied in Chapter 15 and reproduced in Figure 16.4, may be presented as an isoparametric element:

$$\begin{bmatrix} 1 \\ x \\ y \\ u_x \\ u_y \end{bmatrix} = \begin{bmatrix} 1 & 1 & 1 \\ x_1 & x_2 & x_3 \\ y_1 & y_2 & y_3 \\ u_{x1} & u_{x2} & u_{x3} \\ u_{y1} & u_{y2} & u_{y3} \end{bmatrix} \begin{bmatrix} N_1^{(e)} \\ N_2^{(e)} \\ N_3^{(e)} \end{bmatrix}. \quad (16.8)$$

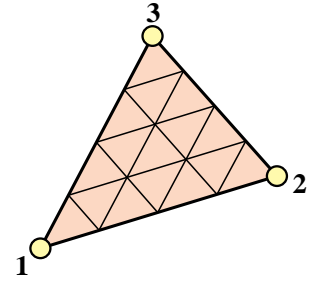


Figure 16.4. The 3-node linear triangle.

The shape functions are simply the triangular coordinates:

$$N_1^{(e)} = \zeta_1, \quad N_2^{(e)} = \zeta_2, \quad N_3^{(e)} = \zeta_3. \quad (16.9)$$

The linear triangle is the only triangular element that is both superparametric and isoparametric.

§16.4.2. The Quadratic Triangle

The six node triangle shown in Figure 16.5 is the next complete-polynomial member of the isoparametric triangle family. The isoparametric definition is

$$\begin{bmatrix} 1 \\ x \\ y \\ u_x \\ u_y \end{bmatrix} = \begin{bmatrix} 1 & 1 & 1 & 1 & 1 & 1 \\ x_1 & x_2 & x_3 & x_4 & x_5 & x_6 \\ y_1 & y_2 & y_3 & y_4 & y_5 & y_6 \\ u_{x1} & u_{x2} & u_{x3} & u_{x4} & u_{x5} & u_{x6} \\ u_{y1} & u_{y2} & u_{y3} & u_{y4} & u_{y5} & u_{y6} \end{bmatrix} \begin{bmatrix} N_1^{(e)} \\ N_2^{(e)} \\ N_3^{(e)} \\ N_4^{(e)} \\ N_5^{(e)} \\ N_6^{(e)} \end{bmatrix} \quad (16.10)$$

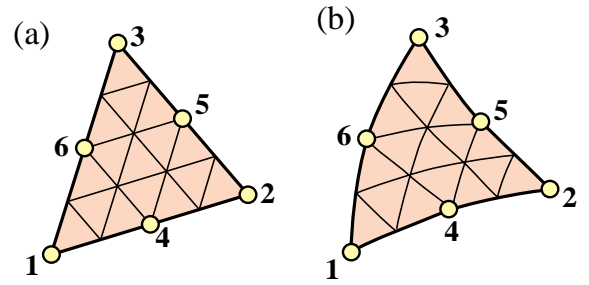


Figure 16.5. The 6-node quadratic triangle: (a) the superparametric version, with straight sides and midside nodes at midpoints; (b) the isoparametric version.

The shape functions are

$$\begin{aligned} N_1^{(e)} &= \zeta_1(2\zeta_1 - 1), & N_2^{(e)} &= \zeta_2(2\zeta_2 - 1), & N_3^{(e)} &= \zeta_3(2\zeta_3 - 1), \\ N_4^{(e)} &= 4\zeta_1\zeta_2, & N_5^{(e)} &= 4\zeta_2\zeta_3, & N_6^{(e)} &= 4\zeta_3\zeta_1. \end{aligned} \quad (16.11)$$

The element may have parabolically curved sides defined by the location of the midnodes 4, 5 and 6. The triangular coordinates for a curved triangle are no longer straight lines, but form a curvilinear system as can be observed in Figure 16.5.

§16.4.3. *The Cubic Triangle

The cubic triangle has ten nodes. This shape functions of this element are the subject of an Exercise in Chapter 18.

§16.5. QUADRILATERAL ELEMENTS

§16.5.1. Quadrilateral Coordinates and Iso-P Mappings

Before presenting examples of quadrilateral elements, we must introduce the appropriate *natural coordinate system* for that geometry. The natural coordinates for a triangular element are the triangular coordinates ζ_1 , ζ_2 and ζ_3 . The natural coordinates for a quadrilateral element are ξ and η , which are illustrated in Figure 16.6 for straight sided and curved side quadrilaterals. These are called *quadrilateral coordinates*.

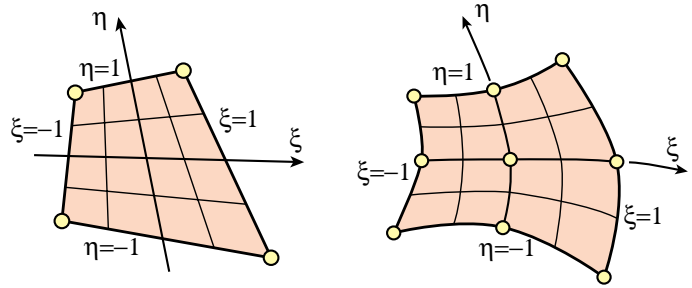


Figure 16.6. Quadrilateral coordinates.

These coordinates vary from -1 on one side to $+1$ at the other, taking the value zero on the medians. This particular variation range (instead of, say, 0 to 1) was chosen by Irons and coworkers to facilitate the use of the standard Gauss integration formulas. Those formulas are discussed in the following Chapter.

In some FEM derivations it is useful to visualize the quadrilateral coordinates plotted as Cartesian coordinates in the $\{\xi, \eta\}$ plane. This is called the *reference plane*. All quadrilateral elements in the reference plane become a square of side 2, called the *reference element*, which extends over $\xi \in [-1, 1]$, $\eta \in [-1, 1]$. The transformation between $\{\xi, \eta\}$ and $\{x, y\}$ dictated by the second and third equations of (16.4) is called the *isoparametric mapping*. A similar version exists for triangles. An important application of this mapping is discussed in §16.6; see Figure 16.9 there.

§16.5.2. The Bilinear Quadrilateral

The four-node quadrilateral shown in Figure 16.7 is the simplest member of the quadrilateral family. It is defined by

$$\begin{bmatrix} 1 \\ x \\ y \\ u_x \\ u_y \end{bmatrix} = \begin{bmatrix} 1 & 1 & 1 & 1 \\ x_1 & x_2 & x_3 & x_4 \\ y_1 & y_2 & y_3 & y_4 \\ u_{x1} & u_{x2} & u_{x3} & u_{x4} \\ u_{y1} & u_{y2} & u_{y3} & u_{y4} \end{bmatrix} \begin{bmatrix} N_1^{(e)} \\ N_2^{(e)} \\ N_3^{(e)} \\ N_4^{(e)} \end{bmatrix}. \quad (16.12)$$

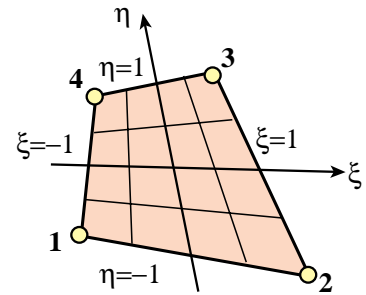


Figure 16.7. The 4-node bilinear quadrilateral.

The shape functions are

$$\begin{aligned} N_1^{(e)} &= \frac{1}{4}(1 - \xi)(1 - \eta), & N_2^{(e)} &= \frac{1}{4}(1 + \xi)(1 - \eta), \\ N_3^{(e)} &= \frac{1}{4}(1 + \xi)(1 + \eta), & N_4^{(e)} &= \frac{1}{4}(1 - \xi)(1 + \eta). \end{aligned} \quad (16.13)$$

These functions vary *linearly* on quadrilateral coordinate lines $\xi = \text{const}$ and $\eta = \text{const}$, but are not linear polynomials as in the case of the three-node triangle.

§16.5.3. The Biquadratic Quadrilateral

The nine-node quadrilateral shown in Figure 16.8(a) is the next *bicomplete* member of the quadrilateral family. It has eight external nodes and one internal node. It is defined by

$$\begin{bmatrix} 1 \\ x \\ y \\ u_x \\ u_y \end{bmatrix} = \begin{bmatrix} 1 & 1 & 1 & 1 & 1 & 1 & 1 & 1 & 1 \\ x_1 & x_2 & x_3 & x_4 & x_5 & x_6 & x_7 & x_8 & x_9 \\ y_1 & y_2 & y_3 & y_4 & y_5 & y_6 & y_7 & y_8 & y_9 \\ u_{x1} & u_{x2} & u_{x3} & u_{x4} & u_{x5} & u_{x6} & u_{x7} & u_{x8} & u_{x9} \\ u_{y1} & u_{y2} & u_{y3} & u_{y4} & u_{y5} & u_{y6} & u_{y7} & u_{y8} & u_{y9} \end{bmatrix} \begin{bmatrix} N_1^{(e)} \\ N_2^{(e)} \\ \vdots \\ N_9^{(e)} \end{bmatrix}. \quad (16.14)$$

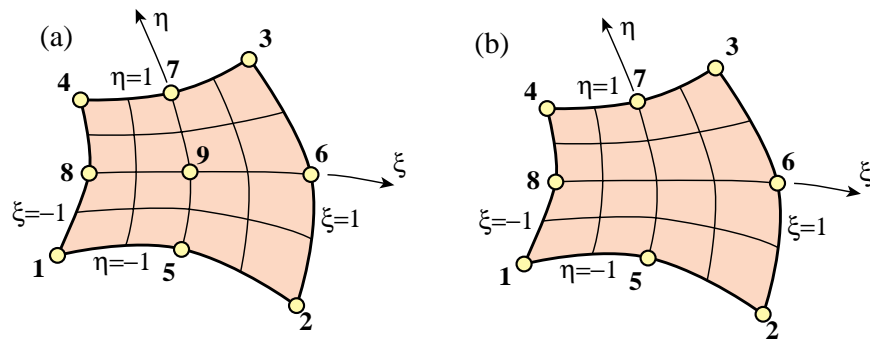


Figure 16.8. Two widely used higher order quadrilaterals: (a) the nine-node biquadratic quadrilateral; (b) the eight-node “serendipity” quadrilateral.

The shape functions are

$$\begin{aligned} N_1^{(e)} &= \frac{1}{4}(1 - \xi)(1 - \eta)\xi\eta & N_5^{(e)} &= -\frac{1}{2}(1 - \xi^2)(1 - \eta)\eta \\ N_2^{(e)} &= -\frac{1}{4}(1 + \xi)(1 - \eta)\xi\eta & N_6^{(e)} &= \frac{1}{2}(1 + \xi)(1 - \eta^2)\xi & N_9^{(e)} &= (1 - \xi^2)(1 - \eta^2) \\ &\dots & & \dots & & \end{aligned} \quad (16.15)$$

These functions vary *quadratically* along the coordinate lines $\xi = \text{const}$ and $\eta = \text{const}$. The shape function associated with the internal node 9 is called a *bubble function* in the FEM literature.

Figure 16.8(b) depicts a widely used eight-node variant called the “serendipity” quadrilateral.¹ The internal node is eliminated by kinematic constraints as worked out in Exercise 18.11.

¹ A name that originated from circumstances surrounding the element discovery.

§16.6. *COMPLETENESS PROPERTIES OF ISO-P ELEMENTS

Some general conclusions as regards the range of applications of isoparametric elements can be obtained from a *completeness analysis*. More specifically, whether the general prescription (16.6) that combines (16.4) and (16.5) satisfies the *completeness* criterion of finite element trial expansions. This is one of the conditions for convergence to the analytical solution. The requirement is treated generally in Chapter 19, and is stated here in recipe form.

§16.6.1. *Completeness Analysis

The plane stress problem has variational index $m = 1$. A set of shape functions is complete for this problem if they can represent exactly any *linear* displacement motions such as

$$u_x = \alpha_0 + \alpha_1 x + \alpha_2 y, \quad u_y = \beta_0 + \beta_1 x + \beta_2 y. \quad (16.16)$$

To carry out the check, evaluate (16.16) at the nodes

$$u_{xi} = \alpha_0 + \alpha_1 x_i + \alpha_2 y_i \quad u_{yi} = \beta_0 + \beta_1 x_i + \beta_2 y_i, \quad i = 1, \dots, n. \quad (16.17)$$

Insert this into the displacement expansion (16.5) to see whether the linear displacement field (16.16) is recovered. Here are the computations for the displacement component u_x :

$$u_x = \sum_{i=1}^n (\alpha_0 + \alpha_1 x_i + \alpha_2 y_i) N_i^{(e)} = \alpha_0 \sum_i N_i^{(e)} + \alpha_1 \sum_i x_i N_i^{(e)} + \alpha_2 \sum_i y_i N_i^{(e)} = \alpha_0 + \alpha_1 x + \alpha_2 y. \quad (16.18)$$

For the last step we have used the geometry definition relations (16.4), reproduced here for convenience:

$$\boxed{1 = \sum_{i=1}^n N_i^{(e)}, \quad x = \sum_{i=1}^n x_i N_i^{(e)}, \quad y = \sum_{i=1}^n y_i N_i^{(e)}}. \quad (16.19)$$

A similar calculation may be made for u_y . It appears that the isoparametric displacement expansion represents (16.18) for *any* element, and consequently meets the completeness requirement for variational order $m = 1$. The derivation carries without essential change to three dimensions.²

Can you detect a flaw in this conclusion? The fly in the ointment is the last replacement step of (16.18), which assumes that the geometry relations (16.19) *are identically satisfied*. Indeed they are for all the example elements presented in the previous sections. But if the new shape functions are constructed directly by the methods of Chapter 18, *a posteriori* checks of those identities are necessary.

§16.6.2. *Completeness Checks

The first check in (16.19) is easy: *the sum of shape functions must be unity*. This is also called the *unit sum condition*. It can be easily verified by hand for simple elements. Here are two examples.

Check for the linear triangle: directly from the definition of triangular coordinates,

$$N_1^{(e)} + N_2^{(e)} + N_3^{(e)} = \zeta_1 + \zeta_2 + \zeta_3 = 1. \quad (16.20)$$

² This derivation is due to B. M. Irons, see textbook cited in footnote 2, p. 75. The property was known since the mid 1960s and contributed substantially to the rapid acceptance of iso-P elements.

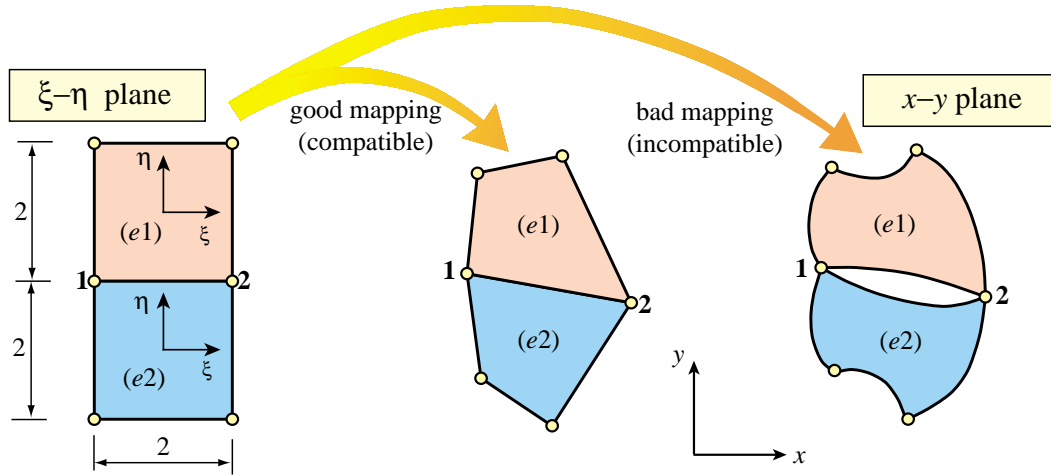


Figure 16.9. Good and bad isoparametric mappings of 4-node quadrilateral from the $\{\xi, \eta\}$ reference plane onto the $\{x, y\}$ physical plane.

Check for the 4-node bilinear quadrilateral:

$$N_1^{(e)} + N_2^{(e)} + N_3^{(e)} + N_4^{(e)} = \frac{1}{4}(1 - \xi - \eta + \xi\eta) + \frac{1}{4}(1 + \xi - \eta - \xi\eta) + \frac{1}{4}(1 + \xi + \eta + \xi\eta) + \frac{1}{4}(1 - \xi + \eta - \xi\eta) = 1 \quad (16.21)$$

For more complicated elements see Exercises 16.2 and 16.3.

The other two checks are less obvious. For specificity consider the 4-node bilinear quadrilateral. The geometry definition equations are

$$x = \sum_{i=1}^4 x_i N_i^{(e)}(\xi, \eta), \quad y = \sum_{i=1}^4 y_i N_i^{(e)}(\xi, \eta). \quad (16.22)$$

Given the corner coordinates, $\{x_i, y_i\}$ and a point $P(x, y)$ one can try to solve for $\{\xi, \eta\}$. This solution requires nontrivial work because it involves two coupled quadratics, but can be done. Reinserting into (16.22) simply gives back x and y , and nothing is gained.³

The correct question to pose is: is the correct geometry of the quadrilateral preserved by the mapping from $\{\xi, \eta\}$ to $\{x, y\}$? In particular, are the sides straight lines? Figure 16.9 illustrate these questions. Two side-two squares: $(e1)$ and $(e2)$, contiguous in the $\{\xi, \eta\}$ reference plane, are mapped to quadrilaterals $(e1)$ and $(e2)$ in the $\{x, y\}$ physical plane through (16.22). The common side 1-2 must remain a straight line to preclude interelement gaps or interpenetration.

We are therefore lead to consider *geometric compatibility* upon mapping. But this is equivalent to the question of *interelement displacement compatibility*, which is posed as item (C) in §18.1. The statement “the displacement along a side must be uniquely determined by nodal displacements on that side” translates to “the coordinates of a side must be uniquely determined by nodal coordinates on that side.” Summarizing:

Unit-sum condition + interelement compatibility \rightarrow completeness.

(16.23)

This subdivision of work significantly reduces the labor involved in element testing.

³ This tautology is actually a blessing, since finding explicit expressions for the natural coordinates in terms of x and y rapidly becomes impossible for higher order elements. See, for example, the complications that already arise for bilinear elements in §23.3.

§16.6.3. *Completeness for Higher Variational Index

The completeness conditions for variational index 2 are far more demanding because they involve quadratic motions. No simple isoparametric configurations satisfy those conditions. Consequently isoparametric formulations have limited importance in the finite element analysis of plate and shell bending.

§16.7. ISO-P ELEMENTS IN ONE AND THREE DIMENSIONS

The reader should not think that the concept of isoparametric representation is confined to two-dimensional elements. It applies without conceptual changes to one and three dimensions *as long as the variational index remains one*.⁴ Three-dimensional solid elements are covered in an advanced course. The use of the isoparametric formulation to construct a 3-node bar element is the subject of Exercises 16.4 through 16.6.

Notes and Bibliography

A detailed presentation of the isoparametric concept, with references to the original 1960 papers may be found in the textbook [16.2].

This matrix representation for isoparametric elements used here was introduced in [16.1].

References

- [16.1] Felippa, C. A. and Clough R. W., The finite element method in solid mechanics, in *Numerical Solution of Field Problems in Continuum Physics*, ed. by G. Birkhoff and R. S. Varga, SIAM–AMS Proceedings II, American Mathematical Society, Providence, R.I., 210–252, 1969.
- [16.2] Irons, B. M. and Ahmad, S., *Techniques of Finite Elements*, Ellis Horwood Ltd, Chichester, UK, 1980.

⁴ A limitation explained in §16.6.3.

Homework Exercises for Chapter 16

The Isoparametric Representation

EXERCISE 16.1

[D:10] What is the physical interpretation of the shape-function unit-sum condition discussed in §16.6? Hint: the element must respond exactly in terms of displacements to rigid-body translations in the x and y directions.

EXERCISE 16.2

[A:15] Check by algebra that the sum of the shape functions for the six-node quadratic triangle (16.11) is exactly one regardless of natural coordinates values. Hint: show that the sum is expressible as $2S_1^2 - S_1$, where $S_1 = \zeta_1 + \zeta_2 + \zeta_3$.

EXERCISE 16.3

[A/C:15] Complete the table of shape functions (16.23) of the nine-node biquadratic quadrilateral. Verify that their sum is exactly one.

EXERCISE 16.4

[A:20] Consider a three-node bar element referred to the natural coordinate ξ . The two end nodes and the midnode are identified as 1, 2 and 3, respectively. The natural coordinates of nodes 1, 2 and 3 are $\xi = -1$, $\xi = 1$ and $\xi = 0$, respectively. The variation of the shape functions $N_1(\xi)$, $N_2(\xi)$ and $N_3(\xi)$ is sketched in Figure E16.1. These functions must be quadratic polynomials in ξ :

$$N_1^{(e)}(\xi) = a_0 + a_1\xi + a_2\xi^2, \quad N_2^{(e)}(\xi) = b_0 + b_1\xi + b_2\xi^2, \quad N_3^{(e)}(\xi) = c_0 + c_1\xi + c_2\xi^2. \quad (\text{E16.1})$$

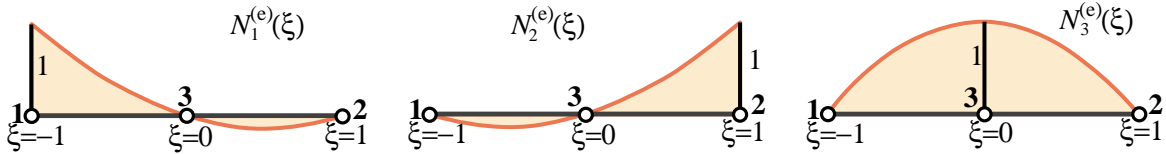


Figure E16.1. Isoparametric shape functions for 3-node bar element (sketch).
Node 3 has been drawn at the 1–2 midpoint but it may be moved away from it, as in Exercises E16.5 and E16.6.

Determine the coefficients a_0 , through c_2 using the node value conditions depicted in Figure E16.1; for example $N_1^{(e)} = 1, 0$ and 0 for $\xi = -1, 0$ and 1 at nodes 1, 3 and 2, respectively. Proceeding this way show that

$$N_1^{(e)}(\xi) = -\frac{1}{2}\xi(1 - \xi), \quad N_2^{(e)}(\xi) = \frac{1}{2}\xi(1 + \xi), \quad N_3^{(e)}(\xi) = 1 - \xi^2. \quad (\text{E16.2})$$

Verify that their sum is identically one.

EXERCISE 16.5

[A/C:10+10+15+5+10] A 3-node straight bar element is defined by 3 nodes: 1, 2 and 3, with axial coordinates x_1 , x_2 and x_3 , respectively, as illustrated in Figure E16.2. The element has axial rigidity EA and length $\ell = x_2 - x_1$. The axial displacement is $u(x)$. The 3 degrees of freedom are the axial node displacements u_1 , u_2 and u_3 . The isoparametric definition of the element is

$$\begin{bmatrix} 1 \\ x \\ u \end{bmatrix} = \begin{bmatrix} 1 & 1 & 1 \\ x_1 & x_2 & x_3 \\ u_1 & u_2 & u_3 \end{bmatrix} \begin{bmatrix} N_1^{(e)} \\ N_2^{(e)} \\ N_3^{(e)} \end{bmatrix}, \quad (\text{E16.3})$$

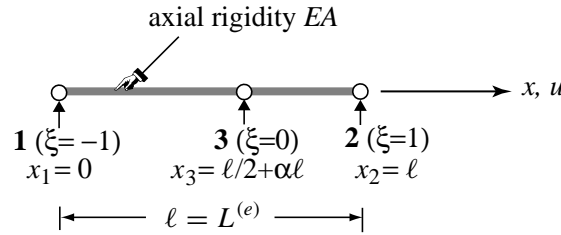


Figure E16.2. The 3-node bar element in its local system.

in which $N_i^{(e)}(\xi)$ are the shape functions (E16.2) of the previous Exercise. Node 3 lies between 1 and 2 but is not necessarily at the midpoint $x = \frac{1}{2}\ell$. For convenience define

$$x_1 = 0, \quad x_2 = \ell, \quad x_3 = \left(\frac{1}{2} + \alpha\right)\ell, \quad (\text{E16.4})$$

where $-\frac{1}{2} < \alpha < \frac{1}{2}$ characterizes the location of node 3 with respect to the element center. If $\alpha = 0$ node 3 is located at the midpoint between 1 and 2. See Figure E16.2.

- From (E16.4) and the second equation of (E16.3) get the Jacobian $J = dx/d\xi$ in terms of ℓ , α and ξ . Show that: (i) if $-\frac{1}{4} < \alpha < \frac{1}{4}$ then $J > 0$ over the whole element $-1 \leq \xi \leq 1$; (ii) if $\alpha = 0$, $J = \ell/2$ is constant over the element.
- Obtain the 1×3 strain-displacement matrix \mathbf{B} relating $e = du/dx = \mathbf{B}\mathbf{u}^{(e)}$, where $\mathbf{u}^{(e)}$ is the column 3-vector of node displacements u_1 , u_2 and u_3 . The entries of \mathbf{B} are functions of ℓ , α and ξ . Hint: $\mathbf{B} = d\mathbf{N}/dx = J^{-1}d\mathbf{N}/d\xi$, where $\mathbf{N} = [N_1 \ N_2 \ N_3]$ and J comes from item (a).
- Show that the element stiffness matrix is given by

$$\mathbf{K}^{(e)} = \int_0^\ell EA \mathbf{B}^T \mathbf{B} dx = \int_{-1}^1 EA \mathbf{B}^T \mathbf{B} J d\xi. \quad (\text{E16.5})$$

Evaluate the rightmost integral for arbitrary α but constant EA using the 2-point Gauss quadrature rule (E13.7). Specialize the result to $\alpha = 0$, for which you should get $K_{11} = K_{22} = 7EA/(3\ell)$, $K_{33} = 16EA/(3\ell)$, $K_{12} = EA/(3\ell)$ and $K_{13} = K_{23} = -8EA/(3\ell)$, with eigenvalues $\{8EA/\ell, 2EA/\ell, 0\}$. Note: use of a CAS is recommended for this item to save time.

- What is the minimum number of Gauss points needed to integrate $\mathbf{K}^{(e)}$ exactly if $\alpha = 0$?
- This item addresses the question of why $\mathbf{K}^{(e)}$ was computed by numerical integration in (c). Why not use exact integration? The answer is that the exact stiffness for arbitrary α is numerically useless. To see why, try the following script in *Mathematica*:

```

ClearAll[EA,alpha,xi]; (* Define J and B={{B1,B2,B3}} here *)
Ke=Simplify[Integrate[EA*Transpose[B].B*J,{xi,-1,1}]];
Print["exact Ke=",Ke//MatrixForm];
Print["exact Ke for alpha=0",Simplify[Ke/.alpha->0]//MatrixForm];
Keseries=Normal[Series[Ke,{alpha,0,2}]];
Print["Ke series about alpha=0:",Keseries//MatrixForm];
Print["Ke for alpha=0",Simplify[Keseries/.alpha->0]//MatrixForm];

```

At the start of this script define J and B with the results of items (a) and (b), respectively. Then run the script, the fourth line of which will trigger error messages. Comment on why the exact stiffness cannot

be evaluated directly at $\alpha = 0$, and why it contains complex numbers. A Taylor series expansion about $\alpha = 0$ removes both problems but the 2-point Gauss integration rule gives the same answer without the gyrations.

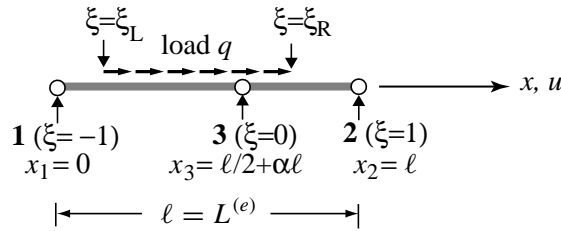


Figure E16.3. The 3-node bar element under a “box” axial load q .

EXERCISE 16.6

[A/C:20] Construct the consistent force vector for the 3-node bar element of the foregoing exercise, if the bar is loaded by a uniform axial force q (given per unit of x length) that extends from $\xi = \xi_L$ through $\xi = \xi_R$, and is zero otherwise. Here $-1 \leq \xi_L < \xi_R \leq 1$. See Figure E16.3. Use

$$\mathbf{f}^{(e)} = \int_{-\xi_L}^{\xi_R} q \mathbf{N}^T J d\xi, \quad (\text{E16.6})$$

with the $J = dx/d\xi$ found in Exercise 16.5(a) and analytical integration. The answer is quite complicated and nearly hopeless by hand. Specialize the result to $\alpha = 0$, $\xi_L = -1$ and $\xi_R = 1$.

17

Isoparametric Quadrilaterals

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§17.1. INTRODUCTION

In this Chapter the isoparametric representation of element geometry and shape functions discussed in the previous Chapter is used to construct *quadrilateral* elements for the plane stress problem. The formulas given in Chapter 14 for the stiffness matrix and consistent load vector of general plane stress elements are of course applicable to these elements. For a practical implementation, however, we must go through the following steps:

1. Construction of shape functions.
2. Computations of shape function derivatives to evaluate the strain-displacement matrix.
3. Numerical integration over the element by Gauss quadrature rules.

The first topic was dealt in the previous Chapter in recipe form, and is systematically covered in the next one. Assuming the shape functions have been constructed (or readily found in the FEM literature) the second and third items are combined in an algorithm suitable for programming any isoparametric quadrilateral. The implementation of the algorithm in the form of element modules is partly explained in the Exercises of this Chapter, and more systematically in Chapter 23.

We shall not cover isoparametric triangles here to keep the exposition focused. Triangular coordinates, being linked by a constraint, require “special handling” techniques that would complicate and confuse the exposition. Chapter 24 discusses isoparametric triangular elements in detail.

§17.2. PARTIAL DERIVATIVE COMPUTATION

Partial derivatives of shape functions with respect to the Cartesian coordinates x and y are required for the strain and stress calculations. Since the shape functions are not directly functions of x and y but of the natural coordinates ξ and η , the determination of Cartesian partial derivatives is not trivial. The derivative calculation procedure is presented below for the case of an arbitrary isoparametric quadrilateral element with n nodes.

§17.2.1. The Jacobian

In element derivations we will need the Jacobian of two-dimensional transformations that connect the differentials of $\{x, y\}$ to those of $\{\xi, \eta\}$ and vice-versa:

$$\begin{bmatrix} dx \\ dy \end{bmatrix} = \begin{bmatrix} \frac{\partial x}{\partial \xi} & \frac{\partial x}{\partial \eta} \\ \frac{\partial y}{\partial \xi} & \frac{\partial y}{\partial \eta} \end{bmatrix} \begin{bmatrix} d\xi \\ d\eta \end{bmatrix} = \mathbf{J}^T \begin{bmatrix} d\xi \\ d\eta \end{bmatrix}, \quad \begin{bmatrix} d\xi \\ d\eta \end{bmatrix} = \begin{bmatrix} \frac{\partial \xi}{\partial x} & \frac{\partial \xi}{\partial y} \\ \frac{\partial \eta}{\partial x} & \frac{\partial \eta}{\partial y} \end{bmatrix} \begin{bmatrix} dx \\ dy \end{bmatrix} = \mathbf{J}^{-T} \begin{bmatrix} dx \\ dy \end{bmatrix}, \quad (17.1)$$

Here \mathbf{J} denotes the Jacobian matrix of (x, y) with respect to (ξ, η) , whereas \mathbf{J}^{-1} is the Jacobian matrix of (ξ, η) with respect to (x, y) :

$$\mathbf{J} = \frac{\partial(x, y)}{\partial(\xi, \eta)} = \begin{bmatrix} \frac{\partial x}{\partial \xi} & \frac{\partial y}{\partial \xi} \\ \frac{\partial x}{\partial \eta} & \frac{\partial y}{\partial \eta} \end{bmatrix} = \begin{bmatrix} J_{11} & J_{12} \\ J_{21} & J_{22} \end{bmatrix}, \quad \mathbf{J}^{-1} = \frac{\partial(\xi, \eta)}{\partial(x, y)} = \begin{bmatrix} \frac{\partial \xi}{\partial x} & \frac{\partial \eta}{\partial x} \\ \frac{\partial \xi}{\partial y} & \frac{\partial \eta}{\partial y} \end{bmatrix} \quad (17.2)$$

In finite element work \mathbf{J} and \mathbf{J}^{-1} are often called the *Jacobian* and *inverse Jacobian*, respectively; the fact that it is a matrix being understood. The scalar symbol J means the determinant of

J: $J = |\mathbf{J}| = \det \mathbf{J}$. In one dimension \mathbf{J} and J coalesce. Jacobians play a crucial role in differential geometry. For the general definition of Jacobian matrix of a differential transformation, see Appendix D.

REMARK 17.1

Note that the matrices relating the differentials in (17.1) are the transposes of \mathbf{J} and \mathbf{J}^{-1} . The reason is that differentials transform as covariant quantities as per the chain rule: $dx = (\partial x / \partial \xi) d\xi + (\partial x / \partial \eta) d\eta$, etc. But Jacobians have traditionally been arranged as in (17.2) because of earlier use in contravariant transformations: $\partial \phi / \partial \xi = (\partial \phi / \partial x)(\partial x / \partial \xi) + (\partial \phi / \partial y)(\partial y / \partial \xi)$, as in (17.5) below.

REMARK 17.2

To show that \mathbf{J} and \mathbf{J}^{-1} are in fact inverses of each other we form their product:

$$\mathbf{J}^{-1} \mathbf{J} = \begin{bmatrix} \frac{\partial x}{\partial \xi} \frac{\partial \xi}{\partial x} + \frac{\partial x}{\partial \eta} \frac{\partial \eta}{\partial x} & \frac{\partial y}{\partial \xi} \frac{\partial \xi}{\partial x} + \frac{\partial y}{\partial \eta} \frac{\partial \eta}{\partial x} \\ \frac{\partial x}{\partial \xi} \frac{\partial \xi}{\partial y} + \frac{\partial x}{\partial \eta} \frac{\partial \eta}{\partial y} & \frac{\partial y}{\partial \xi} \frac{\partial \xi}{\partial y} + \frac{\partial y}{\partial \eta} \frac{\partial \eta}{\partial y} \end{bmatrix} = \begin{bmatrix} \frac{\partial x}{\partial x} & \frac{\partial y}{\partial x} \\ \frac{\partial x}{\partial y} & \frac{\partial y}{\partial y} \end{bmatrix} = \begin{bmatrix} 1 & 0 \\ 0 & 1 \end{bmatrix}, \quad (17.3)$$

where we have taken into account that $x = x(\xi, \eta)$, $y = y(\xi, \eta)$ and the fact that x and y are independent coordinates. This proof would collapse, however, if instead of $\{\xi, \eta\}$ we had the triangular coordinates $\{\zeta_1, \zeta_2, \zeta_3\}$ because rectangular matrices have no conventional inverses. This case requires special handling and is covered in Chapter 24.

§17.2.2. Shape Function Derivatives

The shape functions of a quadrilateral element are expressed in terms of the quadrilateral coordinates ξ and η introduced in §16.7. The derivatives with respect to x and y are given by the chain rule:

$$\begin{aligned} \frac{\partial N_i^{(e)}}{\partial x} &= \frac{\partial N_i^{(e)}}{\partial \xi} \frac{\partial \xi}{\partial x} + \frac{\partial N_i^{(e)}}{\partial \eta} \frac{\partial \eta}{\partial x}, \\ \frac{\partial N_i^{(e)}}{\partial y} &= \frac{\partial N_i^{(e)}}{\partial \xi} \frac{\partial \xi}{\partial y} + \frac{\partial N_i^{(e)}}{\partial \eta} \frac{\partial \eta}{\partial y}. \end{aligned} \quad (17.4)$$

In matrix form:

$$\begin{bmatrix} \frac{\partial N_i^{(e)}}{\partial x} \\ \frac{\partial N_i^{(e)}}{\partial y} \end{bmatrix} = \begin{bmatrix} \frac{\partial \xi}{\partial x} & \frac{\partial \eta}{\partial x} \\ \frac{\partial \xi}{\partial y} & \frac{\partial \eta}{\partial y} \end{bmatrix} \begin{bmatrix} \frac{\partial N_i^{(e)}}{\partial \xi} \\ \frac{\partial N_i^{(e)}}{\partial \eta} \end{bmatrix} = \frac{\partial(\xi, \eta)}{\partial(x, y)} \begin{bmatrix} \frac{\partial N_i^{(e)}}{\partial \xi} \\ \frac{\partial N_i^{(e)}}{\partial \eta} \end{bmatrix} = \mathbf{J}^{-1} \begin{bmatrix} \frac{\partial N_i^{(e)}}{\partial \xi} \\ \frac{\partial N_i^{(e)}}{\partial \eta} \end{bmatrix}. \quad (17.5)$$

where \mathbf{J}^{-1} is defined in (17.2). The computation of \mathbf{J} is addressed in the next subsection.

§17.2.3. Computing the Jacobian Matrix

To compute the entries of \mathbf{J} at any quadrilateral location we make use of the last two geometric relations in (16.4), which are repeated here for convenience:

$$x = \sum_{i=1}^n x_i N_i^{(e)}, \quad y = \sum_{i=1}^n y_i N_i^{(e)}. \quad (17.6)$$

Differentiating with respect to the quadrilateral coordinates,

$$\begin{aligned}\frac{\partial x}{\partial \xi} &= \sum_{i=1}^n x_i \frac{\partial N_i^{(e)}}{\partial \xi}, & \frac{\partial y}{\partial \xi} &= \sum_{i=1}^n y_i \frac{\partial N_i^{(e)}}{\partial \xi}, \\ \frac{\partial x}{\partial \eta} &= \sum_{i=1}^n x_i \frac{\partial N_i^{(e)}}{\partial \eta}, & \frac{\partial y}{\partial \eta} &= \sum_{i=1}^n y_i \frac{\partial N_i^{(e)}}{\partial \eta}.\end{aligned}\quad (17.7)$$

because the x_i and y_i do not depend on ξ and η . In matrix form:

$$\mathbf{J} = \mathbf{P}\mathbf{X} = \begin{bmatrix} \frac{\partial N_1^{(e)}}{\partial \xi} & \frac{\partial N_2^{(e)}}{\partial \xi} & \cdots & \frac{\partial N_n^{(e)}}{\partial \xi} \\ \frac{\partial N_1^{(e)}}{\partial \eta} & \frac{\partial N_2^{(e)}}{\partial \eta} & \cdots & \frac{\partial N_n^{(e)}}{\partial \eta} \end{bmatrix} \begin{bmatrix} x_1 & y_1 \\ x_2 & y_2 \\ \vdots & \vdots \\ x_n & y_n \end{bmatrix}. \quad (17.8)$$

Given a quadrilateral point of coordinates ξ, η we can calculate the entries of \mathbf{J} using (17.8). The inverse Jacobian \mathbf{J}^{-1} may be obtained by numerically inverting this 2×2 matrix.

REMARK 17.3

The symbolic inversion of \mathbf{J} for arbitrary ξ, η in general leads to extremely complicated expressions unless the element has a particularly simple geometry, (for example rectangles as in Exercises 17.1–17.3). This was one of the factors that motivated the use of Gaussian numerical quadrature, as discussed below.

§17.2.4. The Strain-Displacement Matrix

The strain-displacement matrix \mathbf{B} that appears in the computation of the element stiffness matrix is given by the general expression (14.18), reproduced here for convenience:

$$\mathbf{e} = \begin{bmatrix} e_{xx} \\ e_{yy} \\ 2e_{xy} \end{bmatrix} = \begin{bmatrix} \frac{\partial N_1^{(e)}}{\partial x} & 0 & \frac{\partial N_2^{(e)}}{\partial x} & 0 & \cdots & \frac{\partial N_n^{(e)}}{\partial x} & 0 \\ 0 & \frac{\partial N_1^{(e)}}{\partial y} & 0 & \frac{\partial N_2^{(e)}}{\partial y} & \cdots & 0 & \frac{\partial N_n^{(e)}}{\partial y} \\ \frac{\partial N_1^{(e)}}{\partial y} & \frac{\partial N_1^{(e)}}{\partial x} & \frac{\partial N_2^{(e)}}{\partial y} & \frac{\partial N_2^{(e)}}{\partial x} & \cdots & \frac{\partial N_n^{(e)}}{\partial y} & \frac{\partial N_n^{(e)}}{\partial x} \end{bmatrix} \mathbf{u}^{(e)} = \mathbf{B}\mathbf{u}^{(e)}. \quad (17.9)$$

The entries of the $3 \times 3n$ matrix \mathbf{B} are partials of the shape functions with respect to x and y . The calculation of these entries is done via (17.8) and (17.5).

§17.3. NUMERICAL INTEGRATION BY GAUSS RULES

The use of numerical integration is essential for evaluating integrals over isoparametric elements. The standard practice has been to use *Gauss integration*¹ because such rules use a *minimal number of sample points to achieve a desired level of accuracy*. This property is important for efficient element calculations, as we shall see that at each sample point a matrix product is evaluated. The fact that the location of the sample points in the Gauss rules is usually given by non-rational numbers is of no concern in digital computation.

¹ See **Notes and Bibliography** at the end of the Chapter.

§17.3.1. One Dimensional Rules

The standard Gauss integration rules in one dimension are defined by

$$\int_{-1}^1 F(\xi) d\xi \approx \sum_{i=1}^p w_i F(\xi_i). \quad (17.10)$$

Here $p \geq 1$ is the number of Gauss integration points, w_i are the integration weights, and ξ_i are sample-point abscissae in the interval $[-1, 1]$. The use of the canonical interval $[-1, 1]$ is no restriction, because an integral over another range, say from a to b , can be transformed to $[-1, +1]$ via a simple linear transformation of the independent variable, as shown in the Remark below.

The first five one-dimensional Gauss rules, illustrated in Figure 17.1, are:

$$\begin{aligned} \text{One point: } & \int_{-1}^1 F(\xi) d\xi \approx 2F(0), \\ \text{Two points: } & \int_{-1}^1 F(\xi) d\xi \approx F(-1/\sqrt{3}) + F(1/\sqrt{3}), \\ \text{Three points: } & \int_{-1}^1 F(\xi) d\xi \approx \frac{5}{9}F(-\sqrt{3/5}) + \frac{8}{9}F(0) + \frac{5}{9}F(\sqrt{3/5}), \\ \text{Four points: } & \int_{-1}^1 F(\xi) d\xi \approx w_{14}F(\xi_{14}) + w_{24}F(\xi_{24}) + w_{34}F(\xi_{34}) + w_{44}F(\xi_{44}), \\ \text{Five points: } & \int_{-1}^1 F(\xi) d\xi \approx w_{15}F(\xi_{15}) + w_{25}F(\xi_{25}) + w_{35}F(\xi_{35}) + w_{45}F(\xi_{45}) + w_{55}F(\xi_{55}). \end{aligned} \quad (17.11)$$

For the 4-point rule, $\xi_{34} = -\xi_{24} = \sqrt{(3 - 2\sqrt{6/5})/7}$, $\xi_{44} = -\xi_{14} = \sqrt{(3 + 2\sqrt{6/5})/7}$, $w_{14} = w_{44} = \frac{1}{2} - \frac{1}{6}\sqrt{5/6}$, and $w_{24} = w_{34} = \frac{1}{2} + \frac{1}{6}\sqrt{5/6}$. For the five point rule $\xi_{55} = -\xi_{15} = \frac{1}{3}\sqrt{5 + 2\sqrt{10/7}}$, $\xi_{45} = -\xi_{35} = \frac{1}{3}\sqrt{5 - 2\sqrt{10/7}}$, $\xi_{35} = 0$, $w_{15} = w_{55} = (322 - 13\sqrt{70})/900$, $w_{25} = w_{45} = (322 + 13\sqrt{70})/900$ and $w_{35} = 512/900$.

The rules (17.11) integrate exactly polynomials in ξ of orders up to 1, 3, 5, 7 and 9, respectively. In general a one-dimensional Gauss rule with p points integrates exactly polynomials of order up to $2p - 1$. This is called the *degree* of the formula.

REMARK 17.4

A more general integral, such as $F(x)$ over $[a, b]$ in which $\ell = b - a > 0$, is transformed to the canonical interval $[-1, 1]$ through the the mapping $x = \frac{1}{2}a(1 - \xi) + \frac{1}{2}b(1 + \xi) = \frac{1}{2}(a + b) + \frac{1}{2}\ell\xi$, or $\xi = (2/\ell)(x - \frac{1}{2}(a + b))$. The Jacobian of this mapping is $J = dx/d\xi = \frac{1}{2}\ell$. Thus

$$\int_a^b F(x) dx = \int_{-1}^1 F(\xi) J d\xi = \int_{-1}^1 F(\xi) \frac{1}{2}\ell d\xi. \quad (17.12)$$

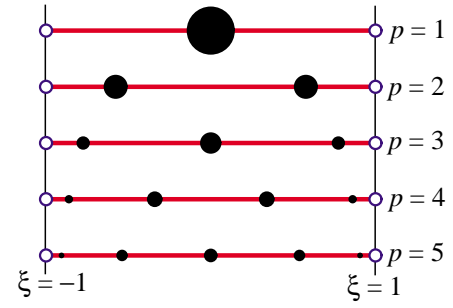


Figure 17.1. The first five one-dimensional Gauss rules $p = 1, 2, 3, 4$ depicted over the line segment $\xi \in [-1, +1]$. Sample point locations are marked with black circles. The radii of these circles are proportional to the integration weights.

REMARK 17.5

Higher order Gauss rules are tabulated in standard manuals for numerical computation. For example, the widely used Handbook of Mathematical Functions [17.1] tabulates (in Table 25.4) rules with up to 96 points. For $p > 6$ the abscissas and weights of sample points are not expressible as rational numbers or radicals, and can only be given as floating-point numbers.

§17.3.2. Mathematica Implementation of 1D Rules

The following *Mathematica* module returns exact or floating-point information for the first five unidimensional Gauss rules:

```
LineGaussRuleInfo[{rule_,numer_},point_]:= Module[
  {g2={-1,1}/Sqrt[3],w3={5/9,8/9,5/9},
   g3={-Sqrt[3/5],0,Sqrt[3/5]},
   w4={(1/2)-Sqrt[5/6]/6, (1/2)+Sqrt[5/6]/6,
        (1/2)+Sqrt[5/6]/6, (1/2)-Sqrt[5/6]/6},
   g4={-Sqrt[(3+2*Sqrt[6/5])/7],-Sqrt[(3-2*Sqrt[6/5])/7],
        Sqrt[(3-2*Sqrt[6/5])/7], Sqrt[(3+2*Sqrt[6/5])/7]},
   g5={-Sqrt[5+2*Sqrt[10/7]],-Sqrt[5-2*Sqrt[10/7]],0,
        Sqrt[5-2*Sqrt[10/7]], Sqrt[5+2*Sqrt[10/7]]}/3,
   w5={322-13*Sqrt[70],322+13*Sqrt[70],512,
        322+13*Sqrt[70],322-13*Sqrt[70]}/900,
   i=point,p=rule,info={Null,0}},
  If [p==1, info={0,2}];
  If [p==2, info={g2[[i]],1}];
  If [p==3, info={g3[[i]],w3[[i]]}];
  If [p==4, info={g4[[i]],w4[[i]]}];
  If [p==5, info={g5[[i]],w5[[i]]}];
  If [numer, Return[N[info]], Return[Simplify[info]]];
];
```

To get information for the i^{th} point of the p^{th} rule, in which $1 \leq i \leq p$ and $p = 1, 2, 3, 4, 5$, call the module as $\{xi, wi\} = \text{LineGaussRuleInfo}[\{p, numer\}, i]$. Logical flag *numer* is *True* to get numerical (floating-point) information, or *False* to get exact information. The module returns the sample point abscissa ξ_i in *xi* and the weight w_i in *wi*. Example: $\text{LineGaussRuleInfo}[\{3, \text{False}\}, 2]$ returns $\{0, 8/9\}$. If *p* is not in range 1 through 5, the module returns $\{\text{Null}, 0\}$.

§17.3.3. Two Dimensional Rules

The simplest two-dimensional Gauss rules are called *product rules*. They are obtained by applying the one-dimensional rules to each independent variable in turn. To apply these rules we must first reduce the integrand to the canonical form:

$$\int_{-1}^1 \int_{-1}^1 F(\xi, \eta) d\xi d\eta = \int_{-1}^1 d\eta \int_{-1}^1 F(\xi, \eta) d\xi. \quad (17.13)$$

Once this is done we can process numerically each integral in turn:

$$\int_{-1}^1 \int_{-1}^1 F(\xi, \eta) d\xi d\eta = \int_{-1}^1 d\eta \int_{-1}^1 F(\xi, \eta) d\xi \approx \sum_{i=1}^{p_1} \sum_{j=1}^{p_2} w_i w_j F(\xi_i, \eta_j). \quad (17.14)$$

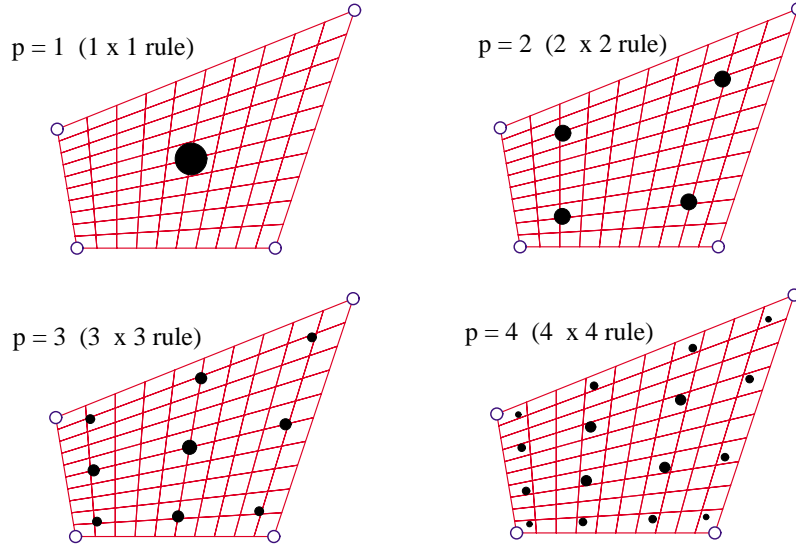


Figure 17.2. The first four two-dimensional Gauss product rules $p = 1, 2, 3, 4$ depicted over a straight-sided quadrilateral region. Sample points are marked with black circles. The areas of these circles are proportional to the integration weights.

where p_1 and p_2 are the number of Gauss points in the ξ and η directions, respectively. Usually the same number $p = p_1 = p_2$ is chosen if the shape functions are taken to be the same in the ξ and η directions. This is in fact the case for all quadrilateral elements presented here. The first four two-dimensional Gauss product rules with $p = p_1 = p_2$ are illustrated in Figure 17.2.

§17.3.4. Mathematica Implementation of 2D Gauss Rules

The following *Mathematica* module implements the two-dimensional product Gauss rules with 1 through 5 points in each direction. The number of points in each direction may be the same or different.

```
QuadGaussRuleInfo[{rule_, numer_}, point_] := Module[
  {xi, eta, p1, p2, i1, i2, w1, w2, k, info = {{Null, Null}, 0},
  If [Length[rule] == 2, {p1, p2} = rule, p1 = p2 = rule];
  If [Length[point] == 2, {i1, i2} = point,
    k = point; i2 = Floor[(k-1)/p1] + 1; i1 = k - p1*(i2-1) ];
  {xi, w1} = LineGaussRuleInfo[{p1, numer}, i1];
  {eta, w2} = LineGaussRuleInfo[{p2, numer}, i2];
  info = {{xi, eta}, w1*w2};
  If [numer, Return[N[info]], Return[Simplify[info]]];
];
```

If the rule has the same number of points p in both directions the module is called in either of two ways:

$$\begin{aligned} \{\{x_{ii}, \eta_{aj}\}, w_{ij}\} &= \text{QuadGaussRuleInfo}[\{p, \text{numer}\}, \{i, j\}] \\ \{\{x_{ii}, \eta_{aj}\}, w_{ij}\} &= \text{QuadGaussRuleInfo}[\{p, \text{numer}\}, k] \end{aligned} \quad (17.15)$$

The first form is used to get information for point $\{i, j\}$ of the $p \times p$ rule, in which $1 \leq i \leq p$ and

$1 \leq j \leq p$. The second form specifies that point by a “visiting counter” k that runs from 1 through p^2 ; if so $\{i, j\}$ are internally extracted² as $j = \text{Floor}[(k-1)/p] + 1$; $i = k - p*(j-1)$.

If the integration rule has p_1 points in the ξ direction and p_2 points in the η direction, the module may be called also in two ways:

$$\begin{aligned} \{\{xii, etaj\}, wij\} &= \text{QuadGaussRuleInfo}[\{\{p1, p2\}, \text{numer}\}, \{i, j\}] \\ \{\{xii, etaj\}, wij\} &= \text{QuadGaussRuleInfo}[\{\{p1, p2\}, \text{numer}\}, k] \end{aligned} \quad (17.16)$$

The meaning of the second argument is as follows. In the first form i runs from 1 to p_1 and j from 1 to p_2 . In the second form k runs from 1 to $p_1 p_2$; if so i and j are extracted by $j = \text{Floor}[(k-1)/p1] + 1$; $i = k - p1*(j-1)$.

In all four forms, logical flag `numer` is set to `True` if numerical information is desired and to `False` if exact information is desired. The module returns ξ_i and η_j in `xii` and `etaj`, respectively, and the weight product $w_i w_j$ in `wij`. This code is used in the Exercises at the end of the chapter. If the inputs are not in range, the module returns $\{\{Null, Null\}, 0\}$.

§17.4. THE STIFFNESS MATRIX

The stiffness matrix of a general plane stress element is given by the expression (14.23), which is reproduced here:

$$\mathbf{K}^{(e)} = \int_{\Omega^{(e)}} h \mathbf{B}^T \mathbf{E} \mathbf{B} d\Omega^{(e)} \quad (17.17)$$

Of the terms that appear in (17.17) the strain-displacement matrix \mathbf{B} has been discussed previously. The thickness h , if variable, may be interpolated via the shape functions. The stress-strain matrix \mathbf{E} is usually constant in elastic problems, but we could in principle interpolate it as appropriate should it vary over the element.

To apply a Gauss product rule for the numerical integration of this equation we have to reduce it to the canonical form (17.14), *i.e.*

$$\mathbf{K}^{(e)} = \int_{-1}^1 \int_{-1}^1 \mathbf{F}(\xi, \eta) d\xi d\eta. \quad (17.18)$$

If ξ and η are the quadrilateral coordinates, everything in (17.18) fits this form, except the element of area $d\Omega^{(e)}$. To complete the reduction we need to express $d\Omega^{(e)}$ in terms of the differentials $d\xi$ and $d\eta$. The desired relation is (see Remark below)

$$d\Omega^{(e)} = dx dy = \det \mathbf{J} d\xi d\eta. \quad (17.19)$$

We therefore have

$$\mathbf{F}(\xi, \eta) = h \mathbf{B}^T \mathbf{E} \mathbf{B} \det \mathbf{J}. \quad (17.20)$$

This matrix function can be numerically integrated over the domain $-1 \leq \xi \leq +1$, $-1 \leq \eta \leq +1$ by an appropriate Gauss product rule.

² Indices i and j are denoted by `i1` and `i2`, respectively, inside the module.

REMARK 17.6

To geometrically justify the area transformation formula, consider the element of area OACB depicted in Figure 17.3. Then

$$\begin{aligned} dA &= \vec{OB} \times \vec{OA} = \frac{\partial x}{\partial \xi} d\xi \frac{\partial y}{\partial \eta} d\eta - \frac{\partial x}{\partial \eta} d\eta \frac{\partial y}{\partial \xi} d\xi \\ &= \begin{vmatrix} \frac{\partial x}{\partial \xi} & \frac{\partial x}{\partial \eta} \\ \frac{\partial y}{\partial \xi} & \frac{\partial y}{\partial \eta} \end{vmatrix} d\xi d\eta = |\mathbf{J}| d\xi d\eta = \det \mathbf{J} d\xi d\eta. \end{aligned} \quad (17.21)$$

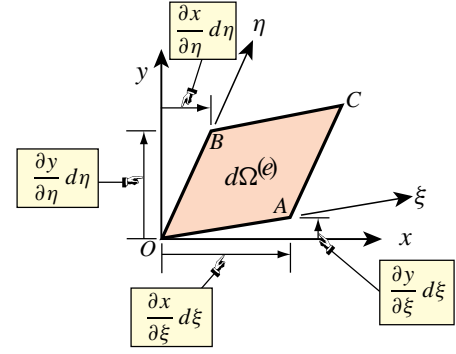


Figure 17.3. Geometric interpretation of the Jacobian-determinant formula.

§17.5. *EFFICIENT COMPUTATION OF ELEMENT STIFFNESS

This section is relevant only if the stiffness is coded in a low-level language such as Fortran or C. A Mathematica implementation for the 4-node quadrilateral is introduced in the Exercises and then elaborated in Chapter 23 for more complex elements.

Efficiency considerations in a low-level programming dictate that we take a close look at the matrix products that appear in (17.20). It is evident from a glance at (17.9) that the strain-displacement matrix \mathbf{B} contains many zero entries. It is therefore of interest to form the matrix product $\mathbf{B}^T \mathbf{E} \mathbf{B}$ at each of the Gaussian points while avoiding multiplications by zero. This goal can be achieved as follows. To develop a simple expression in matrix form, suppose for the moment that the node displacement vector $\mathbf{u}^{(e)}$ is arranged as

$$[u_{x1} \ u_{x2} \ \dots \ u_{xn} \ u_{y1} \ u_{y2} \ \dots \ u_{yn}]^T \quad (17.22)$$

If so \mathbf{B} can be expressed in partitioned-matrix form

$$\mathbf{B} = \begin{bmatrix} \mathbf{N}_x & \mathbf{0} \\ \mathbf{0} & \mathbf{N}_y \\ \mathbf{N}_y & \mathbf{N}_x \end{bmatrix} \quad (17.23)$$

where \mathbf{N}_x and \mathbf{N}_y are row vectors of length n that contain the partial derivatives of the shape functions with respect to x and y , respectively. Then

$$\mathbf{E} \mathbf{B} = \begin{bmatrix} E_{11} \mathbf{N}_x + E_{13} \mathbf{N}_y & E_{12} \mathbf{N}_y + E_{13} \mathbf{N}_x \\ E_{12} \mathbf{N}_x + E_{23} \mathbf{N}_y & E_{22} \mathbf{N}_y + E_{23} \mathbf{N}_x \\ E_{13} \mathbf{N}_x + E_{33} \mathbf{N}_y & E_{23} \mathbf{N}_y + E_{33} \mathbf{N}_x \end{bmatrix}, \quad \mathbf{B}^T \mathbf{E} \mathbf{B} = \begin{bmatrix} \mathbf{S}_{xx} & \mathbf{S}_{xy} \\ \mathbf{S}_{xy}^T & \mathbf{S}_{yy} \end{bmatrix}, \quad (17.24)$$

in which

$$\begin{aligned} \mathbf{S}_{xx} &= E_{11} \mathbf{N}_x^T \mathbf{N}_x + E_{13} (\mathbf{N}_x^T \mathbf{N}_y + \mathbf{N}_y^T \mathbf{N}_x) + E_{33} \mathbf{N}_y^T \mathbf{N}_y \\ \mathbf{S}_{xy} &= E_{13} \mathbf{N}_x^T \mathbf{N}_x + E_{12} \mathbf{N}_x^T \mathbf{N}_y + E_{33} \mathbf{N}_y^T \mathbf{N}_x + E_{23} \mathbf{N}_y^T \mathbf{N}_y \\ \mathbf{S}_{yy} &= E_{33} \mathbf{N}_x^T \mathbf{N}_x + E_{23} (\mathbf{N}_x^T \mathbf{N}_y + \mathbf{N}_y^T \mathbf{N}_x) + E_{22} \mathbf{N}_y^T \mathbf{N}_y \end{aligned} \quad (17.25)$$

In actual implementations the node displacement arrangement (17.22) is not used, but rather the x and y components are grouped node by node:

$$[u_{x1} \ u_{y1} \ u_{x2} \ u_{y2} \ \dots \ u_{xn} \ u_{yn}]^T \quad (17.26)$$

This change is easily implemented through appropriate array indexing.

As noted above, these considerations are important when programming in a low-level language such as C or Fortran. It should be avoided when programming in *Mathematica* or *Matlab* because in such high-level languages explicit indexing of arrays and lists is costly; that overhead in fact overwhelms any advantage gained from skipping zero operations.

§17.6. *INTEGRATION VARIANTS

Several deviations from the standard integration schemes described in the foregoing sections are found in the FEM literature. Two variations are described below and supplemented with motivation Exercises.

§17.6.1. *Weighted Integration

It is sometimes useful to form the element stiffness as a linear combination of stiffnesses produced by two different integration rules. Such schemes are known as *weighted integration* methods. They are distinguished from the selective-integration schemes described in the next subsection in that the constitutive properties are not modified.

For the 4-node bilinear element weighted integration is done by combining the stiffnesses $\mathbf{K}_{1 \times 1}^{(e)}$ and $\mathbf{K}_{2 \times 2}^{(e)}$ produced by 1×1 and 2×2 Gauss product rules, respectively:

$$\mathbf{K}_\beta^{(e)} = (1 - \beta)\mathbf{K}_{1 \times 1}^{(e)} + \beta\mathbf{K}_{2 \times 2}^{(e)}. \quad (17.27)$$

Here β is a scalar in the range $[0, 1]$. If $\beta = 0$ or $\beta = 1$ one recovers the element integrated by the 1×1 or 2×2 rule, respectively.³

The idea behind (17.27) is that $\mathbf{K}_{1 \times 1}^{(e)}$ is rank-deficient and too soft whereas $\mathbf{K}_{2 \times 2}^{(e)}$ is rank-sufficient but too stiff. A combination of too-soft and too-stiff hopefully “balances” the stiffness. An application of this idea to the mitigation of *shear locking* for modeling in-plane bending is the subject of Exercise E17.4.

§17.6.2. *Selective Integration

In the FEM literature the term *selective integration* is used to describe a scheme for forming $\mathbf{K}^{(e)}$ as the sum of two or more matrices computed with different integration rules *and* different constitutive properties.⁴ We consider here the case of a two-way decomposition. Split the plane stress constitutive matrix \mathbf{E} into two:

$$\mathbf{E} = \mathbf{E}_I + \mathbf{E}_{II} \quad (17.28)$$

This is called a *stress-strain splitting*. Inserting (17.28) into (17.17) the expression of the stiffness matrix becomes

$$\mathbf{K}^{(e)} = \int_{\Omega^{(e)}} h \mathbf{B}^T \mathbf{E}_I \mathbf{B} d\Omega^{(e)} + \int_{\Omega^{(e)}} h \mathbf{B}^T \mathbf{E}_{II} \mathbf{B} d\Omega^{(e)} = \mathbf{K}_I^{(e)} + \mathbf{K}_{II}^{(e)}. \quad (17.29)$$

If these two integrals were done through the same integration rule, the stiffness would be identical to that obtained by integrating $h \mathbf{B}^T \mathbf{E} \mathbf{B} d\Omega^{(e)}$. The trick is to use two different rules: rule (I) for the first integral and rule (II) for the second.

In practice selective integration is mostly useful for the 4-node bilinear quadrilateral. For this element rules (I) and (II) are the 1×1 and 2×2 Gauss product rules, respectively. Exercises E17.5–7 investigate stress-strain splittings (17.28) that improve the in-plane bending performance of rectangular elements.

³ For programming the combination (17.27) may be regarded as a 5-point integration rule with weights $w_1 = 4(1-\beta)$ at the sample point at $\xi = \eta = 0$ and $w_i = \beta$ ($i = 2, 3, 4, 5$) at the four sample points at $\xi = \pm 1/\sqrt{3}$, $\eta = \pm 1/\sqrt{3}$.

⁴ This technique is also called “selective reduced integration” to reflect the fact that one of the rules (the “reduced rule”) underintegrates the element.

Notes and Bibliography

This 4-node quadrilateral has a checkered history. It was first derived as a rectangular panel with edge reinforcements (not included here) by Argyris in his 1954 *Aircraft Engineering* series [17.2, p. 49 in the Butterworths reprint]. Argyris used bilinear displacement interpolation in Cartesian coordinates.

After much flailing, a conforming generalization to arbitrary geometry was published in 1964 by Taig and Kerr [17.9] using quadrilateral-fitted coordinates called $\{\xi, \eta\}$ but running from 0 to 1. (Reference [17.9] cites an 1961 English Electric Aircraft internal report as original source but [17.8, p. 520] remarks that the work goes back to 1957.) Bruce Irons, who was aware of Taig's work while at Rolls Royce, created the seminal isoparametric family as a far-reaching extension upon moving to Swansea [17.3,17.4,17.6–17.8].

Gauss integration is also called Gauss-Legendre quadrature. Gauss presented these rules, derived from first principles, in 1814. See Sec 4.11 of [17.5]. The name of Legendre is often adjoined because the abscissas of the one-dimensional sample points turned later to be the zeros of Legendre polynomials.

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Homework Exercises for Chapter 17

Isoparametric Quadrilaterals

The *Mathematica* module `Quad4IsoPMembraneStiffness` in Figure E17.1 computes the element stiffness matrix of the 4-node bilinear quadrilateral. This module is useful as a tool for the Exercises that follow.

```

Quad4IsoPMembraneStiffness[ncoor_,mprop_,fprop_,options_]:=
Module[{i,k,p=2,numer=False,Emat,th=1,h,qcoor,c,w,Nf,
  dNx,dNy,Jdet,B,Ke=Table[0,{8},{8}]}, Emat=mprop[[1]];
If [Length[options]==2, {numer,p}=options, {numer}=options];
If [Length[fprop]>0, th=fprop[[1]]];
If [p<1|p>4, Print["p out of range"];Return[Null]];
For [k=1, k<=p*p, k++,
  {qcoor,w}= QuadGaussRuleInfo[{p,numer},k];
  {Nf,dNx,dNy,Jdet}=Quad4IsoPShapeFunDer[ncoor,qcoor];
  If [Length[th]==0, h=th, h=th.Nf]; c=w*Jdet*h;
  B={ Flatten[Table[{dNx[[i]], 0},{i,4}]],
      Flatten[Table[{0, dNy[[i]]},{i,4}]],
      Flatten[Table[{dNy[[i]],dNx[[i]]},{i,4}]]];
  Ke+=Simplify[c*Transpose[B].(Emat.B)];
]; Return[Ke]
];

Quad4IsoPShapeFunDer[ncoor_,qcoor_]:= Module[
{Nf,dNx,dNy,dNξ,dNη,i,J11,J12,J21,J22,Jdet,ξ,η,x1,x2,x3,x4,
 y1,y2,y3,y4,x,y},
{ξ,η}=qcoor; {{x1,y1},{x2,y2},{x3,y3},{x4,y4}}=ncoor;
Nf={{(1-ξ)*(1-η),(1+ξ)*(1-η),(1+ξ)*(1+η),(1-ξ)*(1+η)}/4;
dNξ={-(1-η),(1-η),(1+η),-(1+η)}/4;
dNη={-(1-ξ),-(1+ξ),(1+ξ),(1-ξ)}/4;
x={x1,x2,x3,x4}; y={y1,y2,y3,y4};
J11=dNξ.x; J12=dNξ.y; J21=dNη.x; J22=dNη.y;
Jdet=Simplify[J11*J22-J12*J21];
dNx=(J22*dNξ-J12*dNη)/Jdet; dNx=Simplify[dNx];
dNy=(-J21*dNξ+J11*dNη)/Jdet; dNy=Simplify[dNy];
Return[{Nf,dNx,dNy,Jdet}]
];

```

Figure E17.1. Mathematica modules for Exercises 17.1–3.

The module makes use also of the Gauss integration modules `QuadGaussRuleInfo` and `LineGaussRuleInfo`. These are not shown in Figure E17.1 since they were listed in §17.3.2 and §17.3.4, but are included in the web-posted Notebook `Quad4Stiffness.nb`.⁵ The arguments of the module are:

- `ncoor` Quadrilateral node coordinates arranged in two-dimensional list form:
 $\{\{x_1, y_1\}, \{x_2, y_2\}, \{x_3, y_3\}, \{x_4, y_4\}\}$.
- `mprop` Material properties supplied as the list $\{\text{Emat}, \rho, \alpha\}$. `Emat` is a two-dimensional list storing the 3×3 plane stress matrix of elastic moduli:

$$\mathbf{E} = \begin{bmatrix} E_{11} & E_{12} & E_{13} \\ E_{12} & E_{22} & E_{23} \\ E_{13} & E_{23} & E_{33} \end{bmatrix} \quad (\text{E17.1})$$

⁵ This Notebook does not include scripts for doing the Exercises below, although it has some text statements at the bottom of the cell. You will need to enter the exercise scripts yourself.

If the material is isotropic with elastic modulus E and Poisson's ratio ν , this matrix becomes

$$\mathbf{E} = \frac{E}{1 - \nu^2} \begin{bmatrix} 1 & \nu & 0 \\ \nu & 1 & 0 \\ 0 & 0 & \frac{1}{2}(1 - \nu) \end{bmatrix} \quad (\text{E17.2})$$

The other two items in `mprop` are not used in this module so zeros may be inserted as placeholders.

fprop Fabrication properties. The plate thickness specified as a four-entry list: $\{h_1, h_2, h_3, h_4\}$, a one-entry list: $\{h\}$, or an empty list: $\{\}$.

The first form is used to specify an element of variable thickness, in which case the entries are the four corner thicknesses and h is interpolated bilinearly. The second form specifies uniform thickness h . If an empty list appears the module assumes a uniform unit thickness.

options Processing options. This list may contain two items: $\{\text{numer}, p\}$ or one: $\{\text{numer}\}$.

numer is a logical flag with value `True` or `False`. If `True`, the computations are forced to proceed in floating point arithmetic. For symbolic or exact arithmetic work set **numer** to `False`.⁶

p specifies the Gauss product rule to have p points in each direction. p may be 1 through 4. For rank sufficiency, p must be 2 or higher. If p is 1 the element will be rank deficient by two.⁷ If omitted $p = 2$ is assumed.

The module returns \mathbf{K}_e as an 8×8 symmetric matrix pertaining to the following arrangement of nodal displacements:

$$\mathbf{u}^{(e)} = [u_{x1} \ u_{y1} \ u_{x2} \ u_{y2} \ u_{x3} \ u_{y3} \ u_{x4} \ u_{y4}]^T. \quad (\text{E17.3})$$

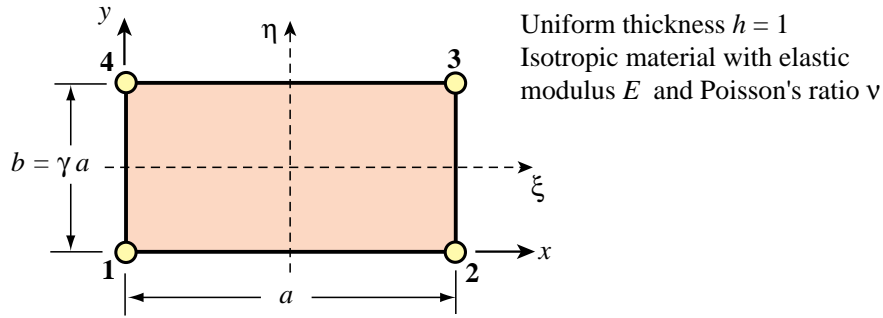


Figure 17.2. Element for Exercises 17.1 to 17.3.

For the following three exercises we consider the specialization of the general 4-node bilinear quadrilateral (16.12)-(16.13) to a *rectangular* element dimensioned a and b in the x and y directions, respectively, as depicted in Figure E17.2. The element has uniform unit thickness h . The material is isotropic with elastic modulus E and Poisson's ratio ν , and consequently \mathbf{E} reduces to (E17.2). The stiffness matrix of this element can be expressed in closed form.⁸ For convenience define $\gamma = b/a$, $\psi_1 = (1 + \nu)\gamma$, $\psi_2 = (1 - 3\nu)\gamma$,

⁶ The reason for this option is speed. A symbolic or exact computation can take orders of magnitude more time than a floating-point evaluation. This becomes more pronounced as elements get more complex.

⁷ The rank of an element stiffness is discussed in Chapter 19.

⁸ This closed form can be obtained by either exact integration, or numerical integration with a 2×2 or higher Gauss rule.

$\psi_3 = 1 - \nu + 2\gamma^2$, $\psi_4 = 2 + (1 - \nu)\gamma^2$, $\psi_5 = 1 - \nu - 4\gamma^2$, $\psi_6 = 1 - \nu - \gamma^2$, $\psi_7 = 4 - (1 - \nu)\gamma^2$ and $\psi_8 = 1 - (1 - \nu)\gamma^2$. Then

$$\mathbf{K}^{(e)} = \frac{Eh}{24\gamma(1 - \nu^2)} \begin{bmatrix} 4\psi_3 & -3\psi_1 & 2\psi_5 & 3\psi_2 & -4\psi_6 & -3\psi_2 & -2\psi_3 & 3\psi_1 \\ & 4\psi_4 & -3\psi_2 & 4\psi_8 & 3\psi_2 & -2\psi_7 & 3\psi_1 & -2\psi_4 \\ & & 4\psi_3 & 3\psi_1 & -2\psi_3 & -3\psi_1 & -4\psi_6 & 3\psi_2 \\ & & & 4\psi_4 & -3\psi_1 & -2\psi_4 & -3\psi_2 & -2\psi_7 \\ & & & & 4\psi_3 & 3\psi_1 & 2\psi_5 & -3\psi_2 \\ & & & & & 4\psi_4 & 3\psi_2 & 4\psi_8 \\ & & & & & & 4\psi_3 & -3\psi_1 \\ \text{symm} & & & & & & & 4\psi_4 \end{bmatrix}. \quad (\text{E17.4})$$

EXERCISE 17.1

[C:15] Exercise the *Mathematica* module of Figure E17.1 with the following script:

```
ClearAll[Em,nu,a,b,h]; Em=48; h=1; a=4; b=2; nu=0;
ncoor={{0,0},{a,0},{a,b},{0,b}};
Emat=Em/(1-nu^2)*{{1,nu,0},{nu,1,0},{0,0,(1-nu)/2}};
For [p=1, p<=4, p++,
  Ke= Quad4IsoPMembraneStiffness[ncoor,{Emat,0,0},{h},{True,p}];
  Print["Gauss integration rule: ",p," x ",p];
  Print["Ke=",Chop[Ke]//MatrixForm];
  Print["Eigenvalues of Ke=",Chop[Eigenvalues[N[Ke]]]]
];
```

Verify that for integration rules $p = 2, 3, 4$ the stiffness matrix does not change and has three zero eigenvalues, which correspond to the three two-dimensional rigid body modes. On the other hand, for $p = 1$ the stiffness matrix is different and displays five zero eigenvalues, which is physically incorrect. (This phenomenon is discussed in detail in Chapter 19.) Question: why does the stiffness matrix stays exactly the same for $p \geq 2$? Hint: take a look at the entries of the integrand $h\mathbf{B}^T \mathbf{E} \mathbf{B} J$ — for a *rectangular geometry* are those polynomials in ξ and η , or rational functions?

EXERCISE 17.2

[C:20] Check the rectangular element stiffness closed form given in (E17.4). This may be done by hand (takes a few days) or running the following script that calls the *Mathematica* module of Figure E17.1:

```
ClearAll[Em,nu,a,b,h]; b=γ*a;
ncoor={{0,0},{a,0},{a,b},{0,b}};
Emat=Em/(1-ν^2)*{{1,ν,0},{ν,1,0},{0,0,(1-ν)/2}};
Ke= Quad4IsoPMembraneStiffness[ncoor,{Emat,0,0},{h},{False,2}];
scaledKe=Simplify[Ke*(24*γ*(1-ν^2)/(Em*h))];
Print["Ke=",Em*h/(24*γ*(1-ν^2)), "\n",scaledKe//MatrixForm];
```

Figure E17.3. Script suggested for Exercise E17.2.

The scaling introduced in the last two lines is for matrix visualization convenience. Verify (E17.4) by printout inspection and report any typos to instructor.

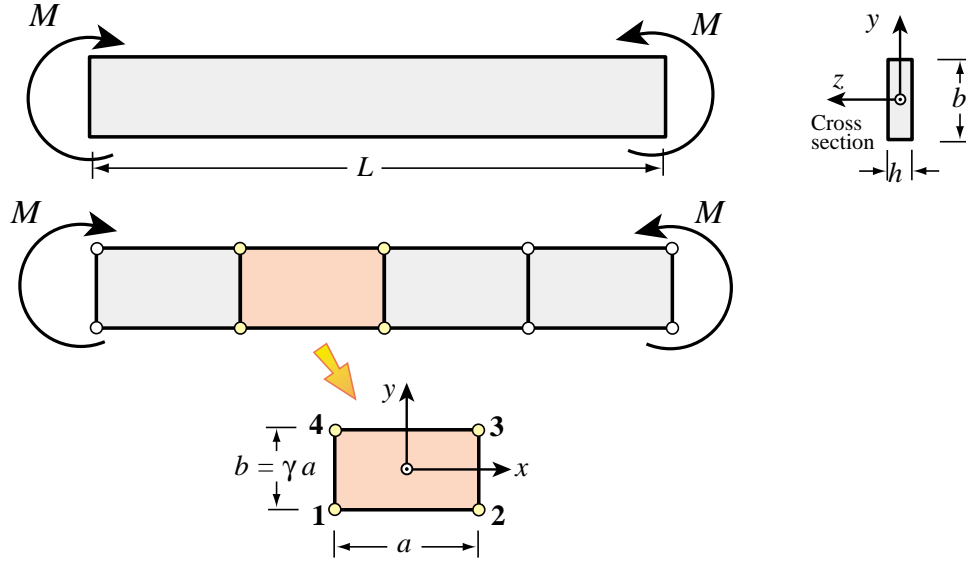


Figure E17.4. Pure bending of Bernoulli-Euler plane beam of thin rectangular cross section, for Exercises 17.3–7. The beam is modeled by one layer of 4-node iso-P bilinear quadrilaterals through its height.

EXERCISE 17.3

[A/C:25=5+10+10] A Bernoulli-Euler plane beam of thin rectangular cross-section with span L , height b and thickness h (normal to the plane of the figure) is bent under end moments M as illustrated in Figure E17.4. The beam is fabricated of isotropic material with elastic modulus E and Poisson's ratio ν . The *exact* solution of the beam problem (from both the theory-of-elasticity and beam-theory standpoints) is a constant bending moment M along the span. Consequently the beam deforms with uniform curvature $\kappa = M/(EI_z)$, in which $I_z = \frac{1}{12}hb^3$ is the cross-section second moment of inertia about z .

The beam is modeled with *one layer* of identical 4-node iso-P bilinear quadrilaterals through its height. These are rectangles with horizontal dimension a ; in the Figure $a = L/4$. The aspect ratio b/a is denoted by γ . By analogy with the exact solution, all rectangles in the finite element model will undergo the same deformation. We can therefore isolate a typical element as illustrated in Figure E17.4.

The exact displacement field for the beam segment referred to the $\{x, y\}$ axes placed at the element center as shown in the bottom of Figure E17.4, are

$$u_x = -\kappa xy, \quad u_y = \frac{1}{2}\kappa(x^2 + \nu y^2), \quad (\text{E17.5})$$

where κ is the deformed beam curvature M/EI . The stiffness equations of the typical rectangular element are given by the close form expression (E17.4).

The purpose of this Exercise is to compare the in-plane bending response of the 4-node iso-P bilinear rectangle to that of a Bernoulli-Euler beam element (which would be exact for this configuration). The quadrilateral element will be called *x-bending exact* if it reproduces the beam solution for all $\{\gamma, \nu\}$. This comparison is distributed into three items.

- Check that (E17.5), as a plane stress 2D elasticity solution, is in full agreement with Bernoulli-Euler beam theory. This can be done by computing the strains $e_{xx} = \partial u_x / \partial x$, $e_{yy} = \partial u_y / \partial y$ and $2e_{xy} = \partial u_y / \partial x + \partial u_x / \partial y$. Then get the stresses σ_{xx} , σ_{yy} and σ_{xy} through the plane stress constitutive matrix (E17.2) of an isotropic material. Verify that both σ_{yy} and σ_{xy} vanish for any ν , and that $\sigma_{xx} = -E\kappa y = -My/I_z$, which agrees with equation (13.4) in Chapter 13.

- (b) Compute the strain energy $U_{\text{quad}} = \frac{1}{2}(\mathbf{u}_{\text{beam}})^T \mathbf{K}^{(e)} \mathbf{u}_{\text{beam}}$ absorbed by the 4-node element under nodal displacements \mathbf{u}_{beam} constructed by evaluating (E17.5) at the nodes 1,2,3,4. To simplify this calculation, it is convenient to decompose that vector as follows:

$$\mathbf{u}_{\text{beam}} = \mathbf{u}_{\text{beam}}^x + \mathbf{u}_{\text{beam}}^y = \frac{1}{4}\kappa ab [-1 \ 0 \ 1 \ 0 \ -1 \ 0 \ 1 \ 0]^T + \frac{1}{8}\kappa(a^2 + vb^2)[0 \ 1 \ 0 \ 1 \ 0 \ 1 \ 0 \ 1]^T \quad (\text{E17.6})$$

Explain why $\mathbf{K}^{(e)} \mathbf{u}_{\text{beam}}^y$ must vanish and consequently

$$U_{\text{quad}} = \frac{1}{2}(\mathbf{u}_{\text{beam}}^x)^T \mathbf{K}^{(e)} \mathbf{u}_{\text{beam}}^x. \quad (\text{E17.7})$$

This energy can be easily computed by *Mathematica* by using the first 4 lines of the script of the previous Exercise, except that here `ncoord = {{-a, -b}, {a, -b}, {a, b}, {-a, b}}/2`. If vector $\mathbf{u}_{\text{beam}}^x$ is formed in `u` as a one-dimensional list, `Uquad = Simplify[u.Ke.u/2]`. This should come out as a function of M , E , ν , h , a and γ because $\kappa = M/(EI_z) = 12M/(Eha^3\gamma^3)$.

- (c) From Mechanics of Materials, or equation (13.7) of Chapter 13, the strain energy absorbed by the beam segment of length a under a constant bending moment M is $U_{\text{beam}} = \frac{1}{2}M\kappa a = M^2a/(2EI_z) = 6M^2/(Eha^2\gamma^3)$. Form the *energy ratio* $r = U_{\text{quad}}/U_{\text{beam}}$ and show that it is a function of the rectangle aspect ratio $\gamma = b/a$ and of Poisson's ratio ν only:

$$r = r(\gamma, \nu) = \frac{2\gamma^2(1 - \nu^2)}{1 + 2\gamma^2 - \nu}. \quad (\text{E17.8})$$

This happens to be the ratio of the 2D model solution to the exact (beam) solution. Hence $r = 1$ means that we get the exact answer, that is the 2D model is *x-bending exact*. If $r < 1$ the 2D model is overstiff, and if $r > 1$ the 2D model is overflexible. Evidently $r < 1$ for all γ if $0 \leq \nu \leq \frac{1}{2}$. Moreover if $b \ll a$, $r \ll 1$; for example if $a = 10b$ and $\nu = 0$, $r \approx 1/50$ and the 2D model gives only about 2% of the correct solution.⁹ Draw conclusions as to the adequacy or inadequacy of the 2D model to capture inplane bending effects, and comment on how you might improve results by modifying the discretization of Figure E17.4.¹⁰

EXERCISE 17.4

[A+C:20] A naive remedy to shear locking can be attempted with the weighted integration methodology outlined in §17.6.1. Let $\mathbf{K}_{1 \times 1}^{(e)}$ and $\mathbf{K}_{2 \times 2}^{(e)}$ denote the element stiffnesses produced by 1×1 and 2×2 Gauss product rules, respectively. Take

$$\mathbf{K}_{\beta}^{(e)} = (1 - \beta)\mathbf{K}_{1 \times 1}^{(e)} + \beta\mathbf{K}_{2 \times 2}^{(e)} \quad (\text{E17.9})$$

where β is adjusted so that shear locking is reduced or eliminated. It is not difficult to find β if the element is rectangular and isotropic. For the definition of *x-bending exact* please read the previous Exercise. Inserting $\mathbf{K}_{\beta}^{(e)}$ into the test introduced there verify that

$$r = \frac{2\gamma^2(1 - \nu^2)}{\beta(1 + 2\gamma^2 - \nu)}. \quad (\text{E17.10})$$

⁹ This phenomenon is referred to in the FEM literature as *shear locking*, because overstiffness is due to the bending motion triggering spurious shear energy in the element. Remedies to shear locking at the element level are studied in advanced FEM courses.

¹⁰ Note that even if we make $a \rightarrow 0$ and $\gamma = b/a \rightarrow \infty$ by taking an infinite number of rectangular elements along x , the energy ratio r remains less than one if $\nu > 0$ since $r \rightarrow 1 - \nu^2$. Thus the 2D model would not generally converge to the correct solution if we keep one layer through the height.

Whence show that if

$$\beta = \frac{2\gamma^2(1 - \nu^2)}{1 + 2\gamma^2 - \nu}, \quad (\text{E17.11})$$

then $r \equiv 1$ for all $\{\gamma, \nu\}$ and the element is x -bending exact. A deficiency of this idea is that it does not make it y -bending exact because $r(\gamma) \neq r(1/\gamma)$ if $\gamma \neq 1$. Moreover the device is not easily extended to non-rectangular geometries or non-isotropic material.

EXERCISE 17.5

[A+C:35] (Advanced) To understand this Exercise please begin by reading Exercise 17.3, and the concept of shear locking. The material is again assumed isotropic with elastic modules E and Poisson's ratio ν . The 4-node rectangular element will be said to be *bending exact* if $r = 1$ for any $\{\gamma, \nu\}$ if the bending test described in Exercise 17.3 is done in both x and y directions. A bending-exact element is completely shear-lock-free.

The selective integration scheme outlined in §17.6.2 is more effective than weighted integration (covered in the previous exercise) to fully eliminate shear locking. Let the integration rules (I) and (II) be the 1×1 and 2×2 product rules, respectively. However the latter is generalized so the sample points are located at $\{-\chi, \chi\}$, $\{\chi, -\chi\}$, $\{\chi, \chi\}$ and $\{-\chi, \chi\}$, with weight 1.¹¹ Consider the stress-strain splitting

$$\mathbf{E} = \frac{E}{1-\nu^2} \begin{bmatrix} 1 & \nu & 0 \\ \nu & 1 & 0 \\ 0 & 0 & \frac{1-\nu}{2} \end{bmatrix} = \frac{E}{1-\nu^2} \begin{bmatrix} \alpha & \beta & 0 \\ \beta & \alpha & 0 \\ 0 & 0 & \frac{1-\nu}{2} \end{bmatrix} + \frac{E}{1-\nu^2} \begin{bmatrix} 1-\alpha & \nu-\beta & 0 \\ \nu-\beta & 1-\alpha & 0 \\ 0 & 0 & 0 \end{bmatrix} = \mathbf{E}_I + \mathbf{E}_{II}, \quad (\text{E17.12})$$

where α and β are scalars. Show that if

$$\chi = \sqrt{\frac{1 - \nu^2}{3(1 - \alpha)}} \quad (\text{E17.13})$$

the resulting element stiffness $\mathbf{K}_I^{(e)} + \mathbf{K}_{II}^{(e)}$ is bending exact for any $\{\alpha, \beta\}$. As a corollary show that that if $\alpha = \nu^2$, which corresponds to the splitting

$$\mathbf{E} = \frac{E}{1-\nu^2} \begin{bmatrix} 1 & \nu & 0 \\ \nu & 1 & 0 \\ 0 & 0 & \frac{1-\nu}{2} \end{bmatrix} = \frac{E}{1-\nu^2} \begin{bmatrix} \nu^2 & \beta & 0 \\ \beta & \nu^2 & 0 \\ 0 & 0 & \frac{1-\nu}{2} \end{bmatrix} + \frac{E}{1-\nu^2} \begin{bmatrix} 1-\nu^2 & \nu-\beta & 0 \\ \nu-\beta & 1-\nu^2 & 0 \\ 0 & 0 & 0 \end{bmatrix} = \mathbf{E}_I + \mathbf{E}_{II}, \quad (\text{E17.14})$$

then $\chi = 1/\sqrt{3}$ and rule (II) becomes the standard 2×2 Gauss product rule. What are two computationally convenient settings for β ?

EXERCISE 17.6

[A+C:35] (Advanced) A variation on the previous exercise on selective integration to make the isotropic rectangular 4-node element bending exact. Integration rule (I) is not changed. However rule (II) has four sample points located at $\{0, -\chi\}$, $\{\chi, 0\}$, $\{0, \chi\}$ and $\{-\chi, 0\}$ each with weight 1.¹² Show that if one selects the stress-strain splitting (E17.12) and

$$\chi = \sqrt{\frac{2(1 - \nu^2)}{3(1 - \alpha)}} \quad (\text{E17.15})$$

the resulting element stiffness $\mathbf{K}_I^{(e)} + \mathbf{K}_{II}^{(e)}$ is bending exact for any $\{\alpha, \beta\}$. Discuss which choices of α reduce χ to $1/\sqrt{3}$ and $\sqrt{2/3}$, respectively.

¹¹ For a rectangular geometry these sample points lie on the diagonals. In the case of the standard 2-point Gauss product rule $\chi = 1/\sqrt{3}$.

¹² This is called a 4-point median rule, since the four points are located on the quadrilateral medians.

EXERCISE 17.7

[A+C:40] (Advanced, research paper level, requires a CAS to be tractable) Extend Exercise 17.5 to consider the case of general anisotropic material:

$$\mathbf{E} = \begin{bmatrix} E_{11} & E_{12} & E_{13} \\ E_{12} & E_{22} & E_{23} \\ E_{13} & E_{23} & E_{33} \end{bmatrix} \quad (\text{E17.16})$$

The rules for the selective integration scheme are as described in Exercise 17.5. The appropriate stress-strain splitting is

$$\mathbf{E} = \mathbf{E}_I + \mathbf{E}_{II} = \begin{bmatrix} E_{11}\alpha_1 & E_{12}\beta & E_{13} \\ E_{12}\beta & E_{22}\alpha_2 & E_{23} \\ E_{13} & E_{23} & E_{33} \end{bmatrix} + \begin{bmatrix} E_{11}(1-\alpha_1) & E_{12}(1-\beta) & 0 \\ E_{12}(1-\beta) & E_{22}(1-\alpha_2) & 0 \\ 0 & 0 & 0 \end{bmatrix} \quad (\text{E17.17})$$

in which β is arbitrary and

$$\begin{aligned} 1 - \alpha_1 &= \frac{|\mathbf{E}|}{3\chi^2 E_{11}(E_{22}E_{33} - E_{23}^2)} = \frac{1}{3\chi^2 C_{11}}, & 1 - \alpha_2 &= \frac{|\mathbf{E}|}{3\chi^2 E_{22}(E_{11}E_{33} - E_{13}^2)} = \frac{1}{3\chi^2 C_{22}}, \\ |\mathbf{E}| &= \det(\mathbf{E}) = E_{11}E_{22}E_{33} + 2E_{12}E_{13}E_{23} - E_{11}E_{23}^2 - E_{22}E_{13}^2 - E_{33}E_{12}^2, \\ C_{11} &= E_{11}(E_{22}E_{33} - E_{23}^2)/|\mathbf{E}|, & C_{22} &= E_{22}(E_{11}E_{33} - E_{13}^2)/|\mathbf{E}|. \end{aligned} \quad (\text{E17.18})$$

Show that the resulting rectangular element is bending exact for any \mathbf{E} and $\chi \neq 0$. (In practice one would select $\chi = 1/\sqrt{3}$.)

18

Shape Function Magic

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§18.1. REQUIREMENTS

This Chapter explains, through a series of examples, how isoparametric shape functions can be directly constructed by geometric considerations. For a problem of variational index 1, the isoparametric shape function $N_i^{(e)}$ associated with node i of element e must satisfy the following conditions:

- (A) *Interpolation condition.* Takes a unit value at node i , and is zero at all other nodes.
- (B) *Local support condition.* Vanishes along any element boundary (a side in 2D, a face in 3D) that does not include node i .
- (C) *Interelement compatibility condition.* Satisfies C^0 continuity between adjacent elements on any element boundary that includes node i .
- (D) *Completeness condition.* The interpolation is able to represent exactly any displacement field which is a linear polynomial in x and y ; in particular, a constant value.

Requirement (A) follows directly from interpolation from node values. Conditions (B), (C) and (D) are consequences of the *convergence* requirements discussed further in the next Chapter.¹ For the moment these three conditions may be viewed as recipes.

One can readily verify that the isoparametric shape functions listed in Chapter 16 satisfy the first two conditions from construction. Direct verification of condition (C) is also straightforward for those examples. A statement equivalent to (C) is that the value of the shape function along a side common to two elements must uniquely depend only on its nodal values on that side.

Completeness is a property of *all* element isoparametric shape functions taken together, rather than of an individual one. If the element satisfies (B) and (C), in view of the discussion in §16.6 it is sufficient to check that the *sum of shape functions is identically one*.

§18.2. DIRECT FABRICATION OF SHAPE FUNCTIONS

Contrary to the what the title of this Chapter implies, the isoparametric shape functions listed in Chapter 16 did not come out of a magician's hat. They can be derived systematically by a judicious inspection process. By “inspection” it is meant that the *geometric* visualization of shape functions plays a crucial role.

The method is based on the following observation. In all examples given so far the isoparametric shape functions are given as *products* of fairly simple polynomial expressions in the natural coordinates. This is no accident but a direct consequence of the definition of natural coordinates. A shape function can be generally expressed as the product of m factors:

$$N_i^{(e)} = c_i L_1 L_2 \dots L_m, \quad (18.1)$$

where

$$L_j = 0, \quad j = 1, \dots, m. \quad (18.2)$$

are the homogeneous equation of lines or curves expressed as *linear* functions in the natural coordinates, and c_i is a normalization coefficient.

¹ Convergence means that the discrete FEM solution approaches the exact analytical solution as the mesh is refined.

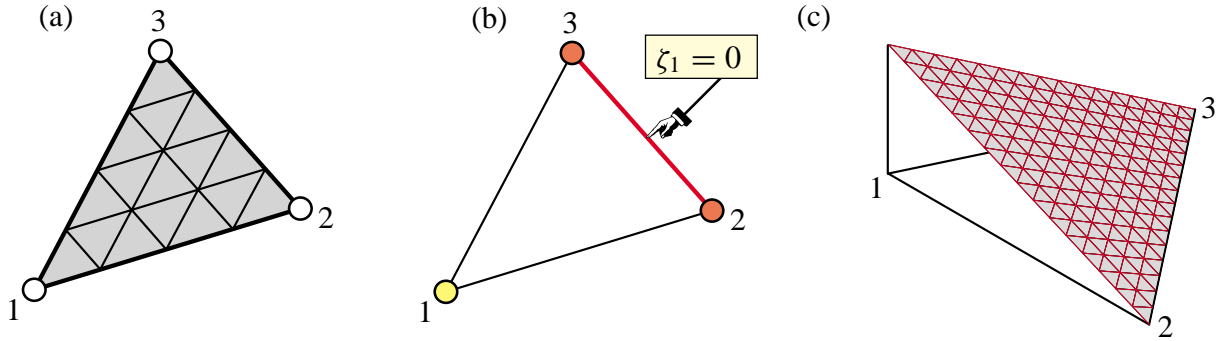


Figure 18.1. The three-node linear triangle: (a) element geometry; (b) equation of side opposite corner 1; (c) perspective view of the shape function $N_1 = \zeta_1$.

For two-dimensional isoparametric elements, the ingredients in (18.1) are chosen according to the following rules.

1. Select the L_j as the minimal number of lines or curves linear in the natural coordinates that cross all nodes except the i^{th} node. Primary choices are the element sides and medians. The examples below illustrate how this is done.
2. Set coefficient c_i so that $N_i^{(e)}$ has the value 1 at the i^{th} node.
3. Check the polynomial order variation over each interelement boundary that contains node i . If this order is n , there must be exactly $n + 1$ nodes on the boundary for the compatibility condition to hold.
4. If compatibility is satisfied, check that the sum of shape functions is identically one.

Specific two-dimensional examples in the following subsections show these rules in action. Essentially the same technique is applicable to one- and three-dimensional elements.

§18.3. TRIANGULAR ELEMENT SHAPE FUNCTIONS

This section illustrates the use of (18.1) in the construction of shape functions for the linear and the quadratic triangle. The cubic triangle is dealt with in Exercise 18.1.

§18.3.1. The Three-Node Linear Triangle

Figure 18.1 shows the three-node linear triangle, which was studied in detail in Chapter 15. The three shape functions are simply the triangular coordinates: $N_i = \zeta_i$, for $i = 1, 2, 3$. Although this result follows directly from the linear interpolation formula of §15.2.4, it can be also quickly derived from the present methodology as follows.

The equation of the triangle side opposite to node i is $L_{j-k} = \zeta_i = 0$, where j and k are the cyclic permutations of i . Here symbol L_{j-k} denotes the left hand side of the homogeneous equation of the natural coordinate line that passes through node points j and k . See Figure 18.1(b) for $i = 1$, $j = 2$ and $k = 3$. Hence the obvious guess is

$$N_i^{(e)} \stackrel{\text{guess}}{=} c_i L_i. \quad (18.3)$$

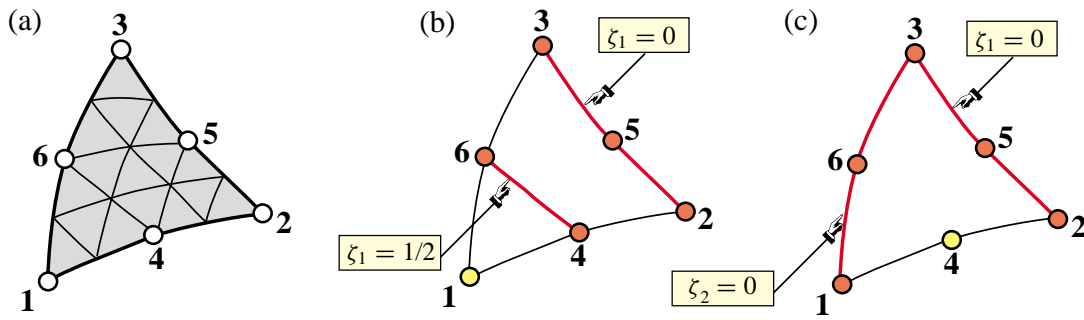


Figure 18.2. The six-node quadratic triangle: (a) element geometry; (b) sides whose product yields $N_1^{(e)}$; (c) Sides whose product yields $N_4^{(e)}$.

This satisfies conditions (A) and (B) except the unit value at node i ; this follows if $c_i = 1$. The compatibility condition (C) follows from the fact that the variation of ζ_i along the sides meeting at node i is linear and that there are two nodes per side; cf. §15.4.2. Completeness is automatic since $\zeta_1 + \zeta_2 + \zeta_3 = 1$. Figure 18.1(c) displays the shape function $N_1 = \zeta_1$ drawn normal to the element in perspective view.

§18.3.2. The Six-Node Quadratic Triangle

The geometry of the six-node quadratic triangle is shown in Figure 18.2(a). Inspection reveals two types of nodes: corners (1, 2 and 3) and midside nodes (4, 5 and 6). Consequently we can expect two types of associated shape functions. We select nodes 1 and 4 as representative cases.

For both cases we try the product of *two* linear functions in the triangular coordinates because we expect the shape functions to be quadratic. These functions are illustrated in Figures 18.2(b,c) for corner node 1 and midside node 4, respectively.

For corner node 1, inspection of Figure 18.2(b) suggests trying

$$N_1^{(e)} \stackrel{\text{guess}}{=} c_1 L_{2-3} L_{4-6}, \quad (18.4)$$

Why is (18.4) expected to work? Clearly $N_1^{(e)}$ will vanish along lines 2-5-3 and 4-6. This will make the function zero at nodes 2, 3, 4, 5 and 6, as is obvious upon inspection of Figure 18.2(b), while being nonzero at node 1. This value can be adjusted to be unity if c_1 is appropriately chosen. Furthermore, $N_1^{(e)}$ will vanish along the element boundary 2-5-3 that does not contain node 1, thus satisfying rule (B) of §18.1. The equations of the lines that appear in (18.4) are

$$L_{2-3}: \quad \zeta_1 = 0, \quad L_{4-6}: \quad \zeta_1 - \frac{1}{2} = 0. \quad (18.5)$$

Replacing into (18.3) we get

$$N_1^{(e)} = c_1 \zeta_1 (\zeta_1 - \frac{1}{2}), \quad (18.6)$$

To find c_1 , evaluate $N_1^{(e)}(\zeta_1, \zeta_2, \zeta_3)$ at node 1. The triangular coordinates of this node are $\zeta_1 = 1$, $\zeta_2 = \zeta_3 = 0$. We require that it takes a unit value there: $N_1^{(e)}(1, 0, 0) = c_1 \times 1 \times \frac{1}{2} = 1$, from

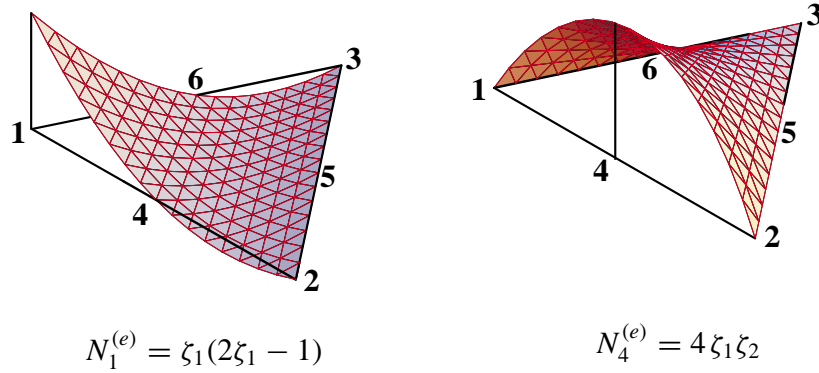


Figure 18.3. Perspective view of shape functions N_1 and N_4 for the quadratic triangle. The plot is done over a straight side triangle for programming simplicity.

which $c_1 = 2$ and finally

$$N_1^{(e)} = 2\zeta_1(\zeta_1 - \frac{1}{2}) = \zeta_1(2\zeta_1 - 1), \quad (18.7)$$

as listed in §16.5.2. Figure 18.3 shows a perspective view. The other two corner shape functions follow by cyclic permutations of the corner index.

For midside node 4, inspection of Figure 18.2(c) suggests trying

$$N_4^{(e)} \stackrel{\text{guess}}{=} c_4 L_{2-3} L_{1-3} \quad (18.8)$$

Evidently (18.8) satisfies requirements (A) and (B) if c_4 is appropriately normalized. The equation of sides L_{2-3} and L_{1-3} are $\zeta_1 = 0$ and $\zeta_2 = 0$, respectively. Therefore $N_4^{(e)}(\zeta_1, \zeta_2, \zeta_3) = c_4 \zeta_1 \zeta_2$. To find c_4 , evaluate this function at node 4, the triangular coordinates of which are $\zeta_1 = \zeta_2 = \frac{1}{2}$, $\zeta_3 = 0$. We require that it takes a unit value there: $N_4^{(e)}(\frac{1}{2}, \frac{1}{2}, 0) = c_4 \times \frac{1}{2} \times \frac{1}{2} = 1$. Hence $c_4 = 4$, which gives

$$N_4^{(e)} = 4\zeta_1\zeta_2 \quad (18.9)$$

as listed in §16.5.2. Figure 18.3 shows a perspective view of this shape function. The other two midside shape functions follow by cyclic permutations of the node indices.

It remains to carry out the interelement continuity check. Consider node 1. The boundaries containing node 1 and common to adjacent elements are 1–2 and 1–3. Over each one the variation of $N_1^{(e)}$ is quadratic in ζ_1 . Therefore the polynomial order over each side is 2. Because there are three nodes on each boundary, the compatibility condition (C) of §18.1 is verified. A similar check can be carried out for midside node shape functions. Exercise 16.1 verified that the sum of the N_i is unity; therefore the element is complete.

§18.4. QUADRILATERAL ELEMENT SHAPE FUNCTIONS

Three quadrilateral elements, with 4, 9 and 8 nodes, respectively, which are commonly used in computational mechanics serve as examples to illustrate the construction of shape functions. Elements with more nodes, such as the bicubic quadrilateral, are not treated as they are rarely used.

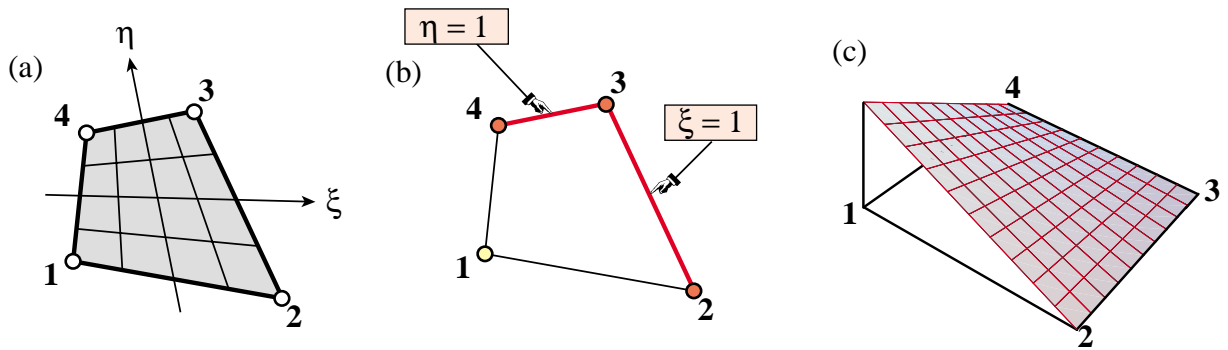


Figure 18.4. The four-node bilinear quadrilateral: (a) element geometry; (b) equations of sides opposite corner 1; (c) perspective view of the shape function $N_1^{(e)}$.

§18.4.1. The Four-Node Bilinear Quadrilateral

The element geometry and natural coordinates are shown in Figure 18.4(a). Only one type of node (corner) and associated shape function is present. Consider node 1 as typical. Inspection of Figure 18.4(b) suggests trying

$$N_1^{(e)} \stackrel{\text{guess}}{=} c_1 L_{2-3} L_{3-4} \quad (18.10)$$

This clearly vanishes over nodes 2, 3 and 4, and can be normalized to unity at node 1 by adjusting c_1 . The equation of side 2-3 is $\xi = 1$, or $\xi - 1 = 0$. The equation of side 3-4 is $\eta = 1$, or $\eta - 1 = 0$. Replacing in (18.10): yields

$$N_1^{(e)}(\xi, \eta) = c_1(\xi - 1)(\eta - 1) = c_1(1 - \xi)(1 - \eta). \quad (18.11)$$

To find coefficient c_1 , evaluate this function at node 1, the natural coordinates of which are $\xi = \eta = -1$:

$$N_1^{(e)}(-1, -1) = c_1 \times 2 \times 2 = 4c_1 = 1. \quad (18.12)$$

Hence $c_1 = \frac{1}{4}$ and the shape function is

$$N_1^{(e)} = \frac{1}{4}(1 - \xi)(1 - \eta), \quad (18.13)$$

as listed in §16.6.2. Figure 18.4(c) shows a perspective view.

For the other three nodes the procedure is the same, traversing the element cyclically. It can be verified that the general expression of the shape functions for this element is

$$N_i^{(e)} = \frac{1}{4}(1 + \xi_i \xi)(1 + \eta_i \eta). \quad (18.14)$$

The continuity check proceeds as follows, using node 1 as example. Node 1 belongs to interelement boundaries 1-2 and 1-3. On side 1-2, $\eta = -1$ is constant and $N_1^{(e)}$ is a *linear* function of ξ ; to see this, replace $\eta = -1$ in (18.13). On side 1-3, $\xi = -1$ is constant and $N_1^{(e)}$ is a *linear* function of η . Consequently the polynomial variation order is 1 over both sides. Because there are two nodes on each side the compatibility condition is satisfied. The sum of the shape functions is one, as shown in (16.21); thus the element is complete.

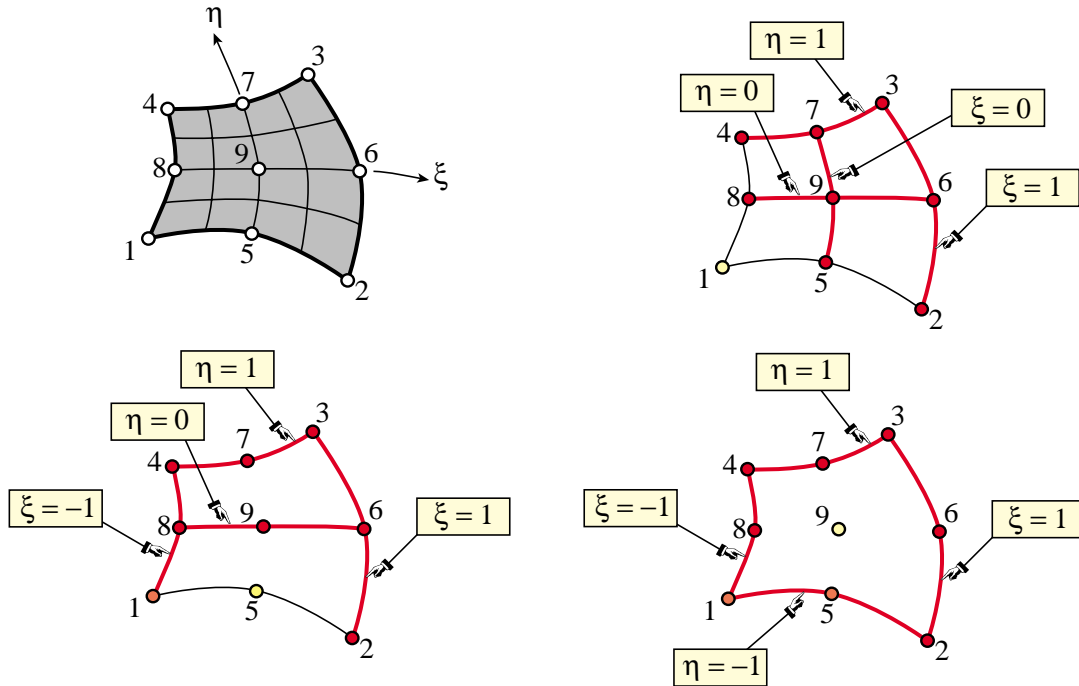


Figure 18.5. The nine-node biquadratic quadrilateral: (a) element geometry; (b,c,d) lines whose product make up the shape functions $N_1^{(e)}$, $N_5^{(e)}$ and $N_9^{(e)}$, respectively.

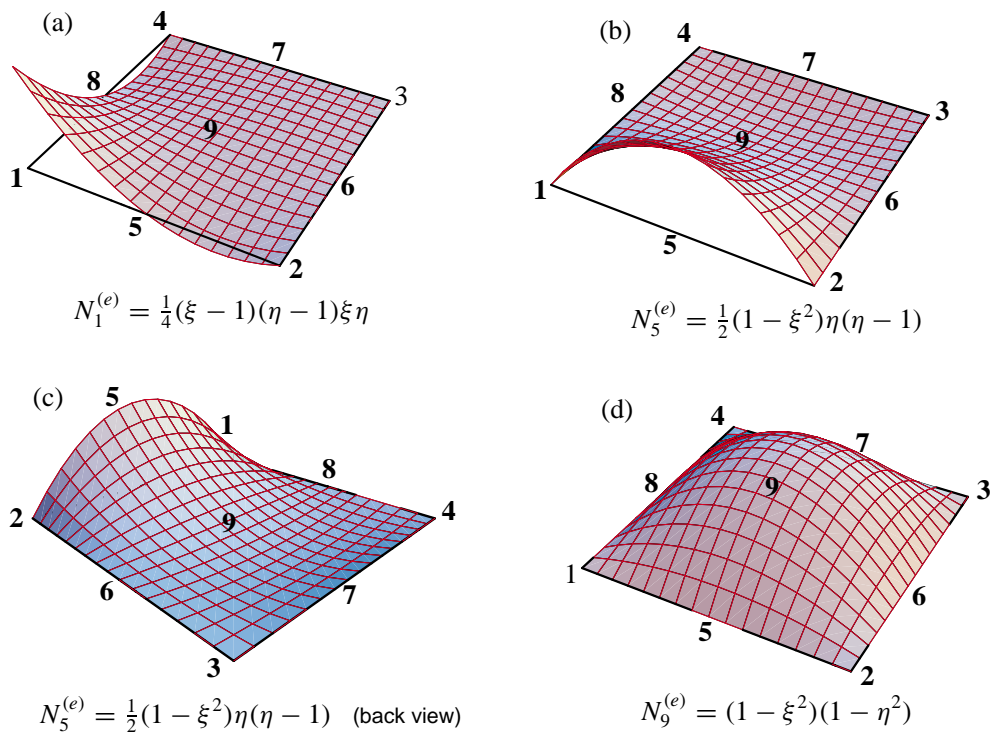


Figure 18.6. Perspective view of the shape functions for nodes 1, 5 and 9 of the nine-node biquadratic quadrilateral.

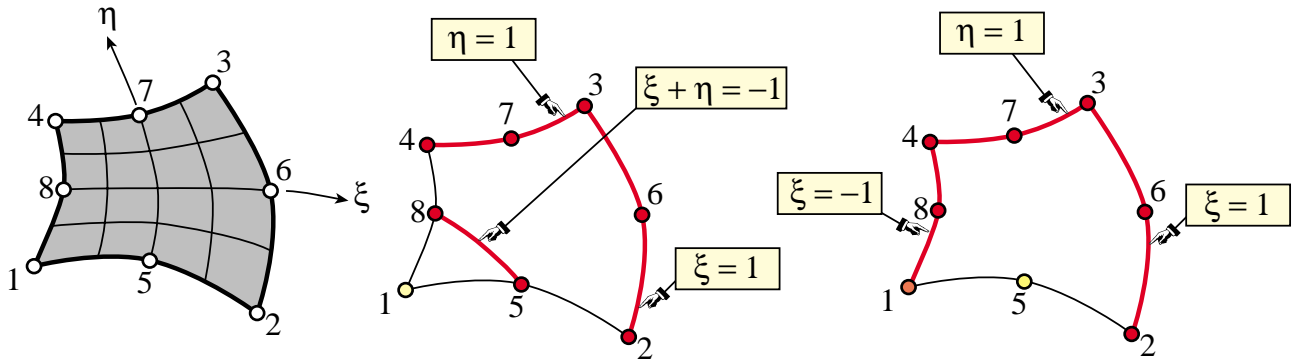


Figure 18.7. The eight-node serendipity quadrilateral: (a) element geometry; (b,c) lines whose product make up the shape functions $N_1^{(e)}$ and $N_5^{(e)}$, respectively.

§18.4.2. The Nine-Node Biquadratic Quadrilateral

The element geometry is shown in Figure 18.5(a). This element has three types of shape functions, which are associated with corner nodes, midside nodes and center node, respectively.

The lines whose product is used to construct three types of shape functions are illustrated in Figure 18.5(b,c,d) for nodes 1, 5 and 9, respectively. The technique has been sufficiently illustrated in previous examples. Here we summarize the calculations for nodes 1, 5 and 9, which are taken as representatives of the three types:

$$N_1^{(e)} = c_1 L_{2-3} L_{3-4} L_{5-7} L_{6-8} = c_1 (\xi - 1)(\eta - 1)\xi\eta. \quad (18.15)$$

$$N_5^{(e)} = c_5 L_{2-3} L_{1-4} L_{6-8} L_{3-4} = c_5 (\xi - 1)(\xi + 1)\eta(\eta - 1) = c_5 (1 - \xi^2)\eta(1 - \eta). \quad (18.16)$$

$$N_9^{(e)} = c_9 L_{1-2} L_{2-3} L_{3-4} L_{4-1} = c_9 (\xi - 1)(\eta - 1)(\xi + 1)(\eta + 1) = c_9 (1 - \xi^2)(1 - \eta^2) \quad (18.17)$$

Imposing the normalization conditions we find

$$c_1 = \frac{1}{4}, \quad c_5 = -\frac{1}{2}, \quad c_9 = 1, \quad (18.18)$$

and we obtain the shape functions listed in §16.6.3. Perspective views are shown in Figure 18.6. The remaining N_i 's are constructed through a similar procedure.

Verification of the interelement continuity condition is immediate: the polynomial variation order over any interelement boundary is two and there are three nodes on each boundary. Exercise 16.2 checks that the sum of shape function is unity; thus the element is complete.

§18.4.3. The Eight-Node “Serendipity” Quadrilateral

This is an eight-node quadrilateral element that results when the center node 9 of the biquadratic quadrilateral is eliminated by kinematic constraints. The geometry and node configuration is shown in Figure 18.7(a). This element was widely used in commercial codes during the 70s and 80s but it is gradually being phased out in favor of the 9-node quadrilateral.

The 8-node quadrilateral has two types of shape functions, which are associated with corner nodes and midside nodes. Lines whose products yields the shape functions for nodes 1 and 5 are shown in Figure 18.7(b,c).

Here are the calculations for shape functions of nodes 1 and 5, which are taken again as representative cases.

$$N_1^{(e)} = c_1 L_{2-3} L_{3-4} L_{5-8} = c_1 (\xi - 1)(\eta - 1)(1 + \xi + \eta) = c_1 (1 - \xi)(1 - \eta)(1 + \xi + \eta), \quad (18.19)$$

$$N_5^{(e)} = c_5 L_{2-3} L_{3-4} L_{4-1} = c_5 (\xi - 1)(\xi + 1)(\eta - 1) = c_5 (1 - \xi^2)(1 - \eta). \quad (18.20)$$

Imposing the normalization conditions we find

$$c_1 = -\frac{1}{4}, \quad c_5 = \frac{1}{2} \quad (18.21)$$

The other shape functions follow by appropriate permutation of nodal indices.

The interelement continuity and completeness verification is similar to that carried out for the nine-node element, and is left as an exercise.

Cell 18.1 *Mathematica* Module to Draw a Function over a Triangle Region

```

PlotTriangleShapeFunction[xytrig_,f_,Nsub_,aspect_]:=Module[
  {Ni,line3D={},poly3D={},zc1,zc2,zc3,xyf1,xyf2,xyf3,
   xc,yc, x1,x2,x3,y1,y2,y3,z1,z2,z3,iz1,iz2,iz3,d},
  {{x1,y1,z1},{x2,y2,z2},{x3,y3,z3}}=Take[xytrig,3];
  xc={x1,x2,x3}; yc={y1,y2,y3}; Ni=Nsub*3;
  Do [ Do [iz3=Ni-iz1-iz2; If [iz3<=0, Continue[]]; d=0;
    If [Mod[iz1+2,3]==0&&Mod[iz2-1,3]==0, d= 1];
    If [Mod[iz1-2,3]==0&&Mod[iz2+1,3]==0, d=-1];
    If [d==0, Continue[]];
    zc1=N[{iz1+d,d,iz2-d,iz3-d}/Ni];
    zc2=N[{iz1-d,iz2+d,d,iz3-d}/Ni];
    zc3=N[{iz1-d,iz2-d,iz3+d,d}/Ni];
    xyf1={xc.zc1,yc.zc1,f[zc1[[1]],zc1[[2]],zc1[[3]]]};
    xyf2={xc.zc2,yc.zc2,f[zc2[[1]],zc2[[2]],zc2[[3]]]};
    xyf3={xc.zc3,yc.zc3,f[zc3[[1]],zc3[[2]],zc3[[3]]]};
    AppendTo[poly3D,Polygon[{xyf1,xyf2,xyf3}]];
    AppendTo[line3D,Line[{xyf1,xyf2,xyf3,xyf1}],
    {iz2,1,Ni-iz1}],{iz1,1,Ni}];
  Show[ Graphics3D[RGBColor[1,0,0],Graphics3D[poly3D],
    Graphics3D[Thickness[.002]],Graphics3D[line3D],
    Graphics3D[RGBColor[0,0,0],Graphics3D[Thickness[.005]],
    Graphics3D[Line[xytrig]],PlotRange->All,
    BoxRatios->{1,1,aspect},Boxed->False]
];
ClearAll[f1,f4];
xyc1={0,0,0}; xyc2={3,0,0}; xyc3={Sqrt[3],3/2,0};
xytrig=N[{xyc1,xyc2,xyc3,xyc1}]; Nsub=16;
f1[zeta1_,zeta2_,zeta3_]:=zeta1*(2*zeta1-1);
f4[zeta1_,zeta2_,zeta3_]:=4*zeta1*zeta2;
PlotTriangleShapeFunction[xytrig,f1,Nsub,1/2];
PlotTriangleShapeFunction[xytrig,f4,Nsub,1/2.5];

```

§18.5. *MATHEMATICA MODULES TO PLOT SHAPE FUNCTIONS

A *Mathematica* module called `PlotTriangleShape Functions`, listed in Cell 18.1, has been developed to draw perspective plots of shape functions $N_i(\zeta_1, \zeta_2, \zeta_3)$ over a triangular region. The region is assumed to have straight sides to simplify the logic. The test statements that follow the module produce the shape function plots shown in Figure 18.3 for the 6-node quadratic triangle. Argument `Nsub` controls the plot resolution while `aspect` controls the *xyz* box aspect ratio. The remaining arguments are self explanatory.

Another *Mathematica* module called `PlotQuadrilateralShape Functions`, listed in Cell 18.2, has been developed to produce perspective plots of shape functions $N_i(\xi, \eta)$ over a quadrilateral region. The region is assumed to have straight sides to simplify the logic. The test statements that follow the module produce

Cell 18.2 *Mathematica* Module to Draw a Function over a Quadrilateral Region

```

PlotQuadrilateralShapeFunction[xyquad_,f_,Nsub_,aspect_] :=Module[
  {Ne,Nev,line3D={},poly3D={},xyf1,xyf2,xyf3,i,j,n,ixi,ieta,
  xi,eta,x1,x2,x3,x4,y1,y2,y3,y4,z1,z2,z3,z4,xc,yc},
  {{x1,y1,z1},{x2,y2,z2},{x3,y3,z3},{x4,y4,z4}}=Take[xyquad,4];
  xc={x1,x2,x3,x4}; yc={y1,y2,y3,y4};
  Ne[xi_,eta_] :=N[{(1-xi)*(1-eta),(1+xi)*(1-eta),
    (1+xi)*(1+eta),(1-xi)*(1+eta)}/4]; n=Nsub;
  Do [ Do [ ixi=(2*i-n-1)/n; ieta=(2*j-n-1)/n;
    {xi,eta}=N[{ixi-1/n,ieta-1/n}]; Nev=Ne[xi,eta];
    xyf1={xc.Nev,yc.Nev,f[xi,eta]};
    {xi,eta}=N[{ixi+1/n,ieta-1/n}]; Nev=Ne[xi,eta];
    xyf2={xc.Nev,yc.Nev,f[xi,eta]};
    {xi,eta}=N[{ixi+1/n,ieta+1/n}]; Nev=Ne[xi,eta];
    xyf3={xc.Nev,yc.Nev,f[xi,eta]};
    {xi,eta}=N[{ixi-1/n,ieta+1/n}]; Nev=Ne[xi,eta];
    xyf4={xc.Nev,yc.Nev,f[xi,eta]};
    AppendTo[poly3D,Polygon[{xyf1,xyf2,xyf3,xyf4}]];
    AppendTo[line3D,Line[{xyf1,xyf2,xyf3,xyf4,xyf1}]],
  {i,1,Nsub}],{j,1,Nsub}];
  Show[ Graphics3D[RGBColor[1,0,0]],Graphics3D[poly3D],
    Graphics3D[Thickness[.002]],Graphics3D[line3D],
    Graphics3D[RGBColor[0,0,0]],Graphics3D[Thickness[.005]],
    Graphics3D[Line[xyquad]], PlotRange->All,
    BoxRatios->{1,1,aspect},Boxed->False]
];
ClearAll[f1,f5,f9];
xyc1={0,0,0}; xyc2={3,0,0}; xyc3={3,3,0}; xyc4={0,3,0};
xyquad=N[{xyc1,xyc2,xyc3,xyc4,xyc1}]; Nsub=16;
f1[xi_,eta_] :=(1/2)*(xi-1)*(eta-1)*xi*eta;
f5[xi_,eta_] :=(1/2)*(1-xi^2)*eta*(eta-1);
f9[xi_,eta_] :=(1-xi^2)*(1-eta^2);
PlotQuadrilateralShapeFunction[xyquad,f1,Nsub,1/2];
PlotQuadrilateralShapeFunction[xyquad,f5,Nsub,1/2.5];
PlotQuadrilateralShapeFunction[xyquad,f9,Nsub,1/3];

```

the shape function plots shown in Figure 18.6(a,b,d) for the 9-node biquadratic quadrilateral. Argument *Nsub* controls the plot resolution while *aspect* controls the *xyz* box aspect ratio. The remaining arguments are self explanatory.

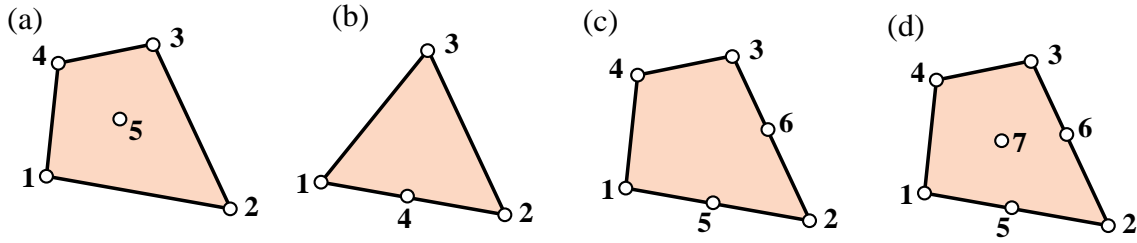


Figure 18.8. Node configurations for which the magic wand (18.1-18.2) does not work.

§18.6. DOES THE MAGIC WAND ALWAYS WORK?

The “cross the dots” recipe (18.1)-(18.2) is not foolproof. It fails for certain node configurations although it is a reasonable way to start.

It runs into difficulties, for instance, in the problem posed in Exercise 18.6, which deals with the 5-node quadrilateral depicted in Figure 18.8(a). If for node 1 one tries the product of side 2–3, side 3–4, and the diagonal 2–5–4, the shape function is easily worked out to be $N_1 = -\frac{1}{8}(1-\xi)(1-\eta)(\xi+\eta)$. This satisfies conditions (A) and (B). However, it violates (C) along sides 1–2 and 4–1, because it is quadratic over them with only two nodes to define its variation.

§18.6.1. Hierarchical Corrections

A more general technique relies on a *correction approach*, which employs a combination of terms such as (18.1). For example, a combination of two patterns, one with m factors and one with n factors, is

$$N_i^{(e)} = c_i L_1^c L_2^c \dots L_m^c + d_i L_1^d L_2^d \dots L_n^d, \quad (18.22)$$

Here two normalization coefficients: c_i and d_i , appear. In practice trying forms such as (18.22) from scratch becomes cumbersome. The development is best done *hierarchically*. The first term is taken to be that of a lower order element, called the *parent element*, for which the one-shot approach works. The second term is then a corrective shape function that vanishes at the nodes of the parent element. If this is insufficient one more corrective term is added, and so on.

The technique is best explained through examples. Exercise 18.6 illustrates the procedure for the element of Figure 18.8(a). The next subsection works out the element of Figure 18.8(b).

§18.6.2. Transition Element Example

The hierarchical correction technique is useful for *transition elements*, which have corner nodes but midnodes only over certain sides. Three examples are provided in Figure 18.8(b,c,d). Shape functions that work can be derived with one, two and three hierarchical corrections, respectively.

As an example, let us construct the shape function $N_1^{(e)}$ for the 4-node transition triangle shown in Figure 18.8(d). Candidate lines for the recipe (18.1) are obviously the side 2–3: $\zeta_1 = 0$, and the median 3–4: $\zeta_1 = \zeta_2$. Thus we try

$$N_1^{(e)} \stackrel{\text{guess}}{=} c_1 \zeta_1 (\zeta_1 - \zeta_2), \quad N_1(1, 0, 0) = 1 = c_1. \quad (18.23)$$

This satisfies conditions (A) and (B) but fails compatibility: over side 1–3 of equation $\zeta_2 = 0$, $N_1^{(e)} = \zeta_1^2$ varies quadratically but there are only 2 nodes. Thus (18.23) is no good. To proceed hierarchically we start from the shape function for the 3-node linear triangle: $N_1^{(e)} = \zeta_1$. This will not vanish at node 4, so apply a correction that vanishes at all nodes but 4. From knowledge of the quadratic triangle midpoint functions, that is obviously $\zeta_1\zeta_2$ times a coefficient to be determined. The new guess is

$$N_1^{(e)} \stackrel{\text{guess}}{=} \zeta_1 + c_1\zeta_1\zeta_2. \quad (18.24)$$

Coefficient c_1 is determined by requiring that $N_1^{(e)}$ vanish at 4: $N_1^{(e)}(\frac{1}{2}, \frac{1}{2}, 0) = \frac{1}{2} + c_1\frac{1}{4} = 0$, whence $c_1 = -2$ and the shape function is

$$N_1^{(e)} = \zeta_1 - 2\zeta_1\zeta_2. \quad (18.25)$$

This is easily checked to satisfy compatibility on all sides. The verification of completeness is left to Exercise 18.8.

Note that since $N_1^{(e)} = \zeta_1(1 - 2\zeta_2)$, (18.25) can be constructed as the normalized product of lines $\zeta_1 = 0$ and $\zeta_2 = \frac{1}{2}$. The latter passes through 4 and is parallel to 1–3. As part of the opening moves in the shape function game this would be a lucky guess indeed. If one goes to a more complicated element no obvious factorization is possible.

Notes and Bibliography

The name “shape functions” for interpolation functions directly expressed in terms of physical coordinates (the node displacements for isoparametric elements) was coined by Irons. The earliest published reference seems to be the paper [18.1]. This was presented in 1965 at the first Wright-Patterson conference, which strongly influenced the early development of FEM. The key connection to numerical integration was presented in [18.3]. A comprehensive exposition is given in the textbook by Irons and Ahmad [18.4].

The quick way of developing shape functions presented here was used in the writer’s 1966 thesis [18.2] for triangles. The qualifier “magic” arose from the timing for covering Chapter 18 in a Fall Semester course: that lecture falls near Halloween.

References

- [18.1] Bazeley, G. P., Cheung, Y. K., Irons, B. M. and Zienkiewicz, O. C., Triangular elements in plate bending – conforming and nonconforming solutions, in *Proc. 1st Conf. Matrix Meth. Struc. Mech.*, ed. by J. Przemieniecki et. al., AFFDL-TR-66-80, Air Force Institute of Technology, Dayton, Ohio, 1966, 547–576.
- [18.2] Felippa, C. A., Refined finite element analysis of linear and nonlinear two-dimensional structures, *Ph.D. Dissertation*, Dept of Civil Engineering, University of California at Berkeley, CA, 1966.
- [18.3] Irons, B. M., Engineering application of numerical integration in stiffness methods, *AIAA J.*, **4**, pp. 2035–2037, 1966.
- [18.4] Irons, B. M. and Ahmad, S., *Techniques of Finite Elements*, Ellis Horwood Ltd, Chichester, UK, 1980.

Homework Exercises for Chapter 18

Shape Function Magic

EXERCISE 18.1

[A/C:15+10] The complete cubic triangle for plane stress has 10 nodes located as shown in Figure E18.1, with their triangular coordinates listed in parentheses.

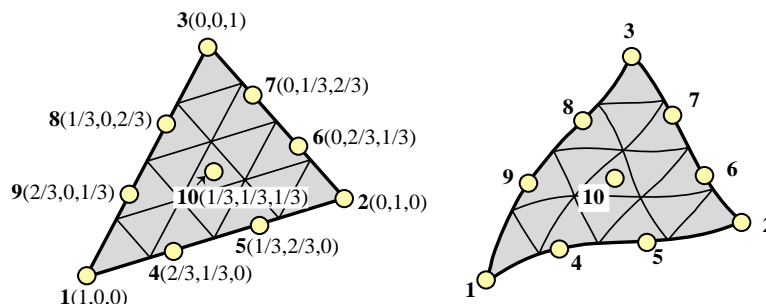


Figure E18.1. Ten-node cubic triangle for Exercise 18.1. The left picture displays the superparametric element whereas the right picture shows the general isoparametric version with curved sides.

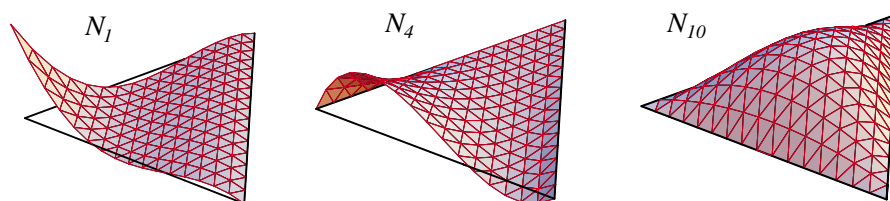


Figure E18.2. Perspective plots of the shape functions N_1 , N_4 and N_{10} for the 10-node cubic triangle.

- Construct the cubic shape functions $N_1^{(e)}$, $N_4^{(e)}$ and $N_{10}^{(e)}$ for nodes 1, 4, and 10 using the line-product technique. [Hint: each shape function is the product of 3 and only 3 lines.] Perspective plots of those 3 functions are shown in Figure E18.2.
- Construct the missing 7 shape functions by appropriate node number permutations, and verify that the sum of the 10 functions is identically one. For the unit sum check use the fact that $\zeta_1 + \zeta_2 + \zeta_3 = 1$.

EXERCISE 18.2

[A:15] Find an alternative shape function $N_1^{(e)}$ for corner node 1 of the 9-node quadrilateral of Figure 18.5(a) by using the diagonal lines 5–8 and 2–9–4 in addition to the sides 2–3 and 3–4. Show that the resulting shape function violates the compatibility condition (C) stated in §18.1.

EXERCISE 18.3

[A/C:15] Complete the above exercise for all nine nodes. Add the shape functions (use a CAS and simplify) and verify whether their sum is unity.

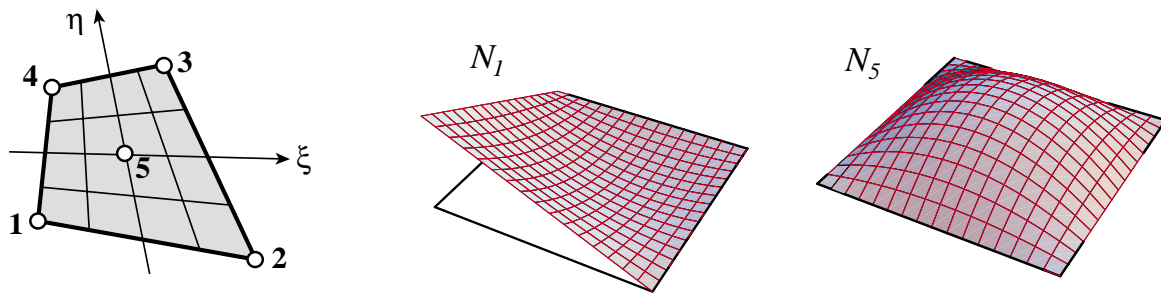


Figure E18.3. Five node quadrilateral element for Exercise 18.6.

EXERCISE 18.4

[A/C:20] Verify that the shape functions $N_1^{(e)}$ and $N_5^{(e)}$ of the eight-node serendipity quadrilateral discussed in §18.4.3 satisfy the interelement compatibility condition (C) stated in §18.1. Obtain all 8 shape functions and verify that their sum is unity.

EXERCISE 18.5

[C:15] Plot the shape functions $N_1^{(e)}$ and $N_5^{(e)}$ of the eight-node serendipity quadrilateral studied in §18.4.3 using the module `PlotQuadrilateralShapeFunction` listed in Cell 18.2.

EXERCISE 18.6

[A:15] A five node quadrilateral element has the nodal configuration shown in Figure E18.3. Perspective views of $N_1^{(e)}$ and $N_5^{(e)}$ are shown in that Figure.² Find five shape functions $N_i^{(e)}$, $i = 1, 2, 3, 4, 5$ that satisfy compatibility, and also verify that their sum is unity.

Hint: develop $N_5(\xi, \eta)$ first for the 5-node quad using the line-product method; then the corner shape functions $\bar{N}_i(\xi, \eta)$ ($i = 1, 2, 3, 4$) for the 4-node quad (already given in the Notes); finally combine $N_i = \bar{N}_i + \alpha N_5$, determining α so that all N_i vanish at node 5. Check that $N_1 + N_2 + N_3 + N_4 + N_5 = 1$ identically.

EXERCISE 18.7

[A:15] An eight-node “brick” finite element for three dimensional analysis has three isoparametric natural coordinates called ξ , η and μ . These coordinates vary from -1 at one face to $+1$ at the opposite face, as sketched in Figure E18.4.

Construct the (trilinear) shape function for node 1 (follow the node numbering of the figure). The equations of the brick faces are:

1485 : $\xi = -1$	2376 : $\xi = +1$
1265 : $\eta = -1$	4378 : $\eta = +1$
1234 : $\mu = -1$	5678 : $\mu = +1$

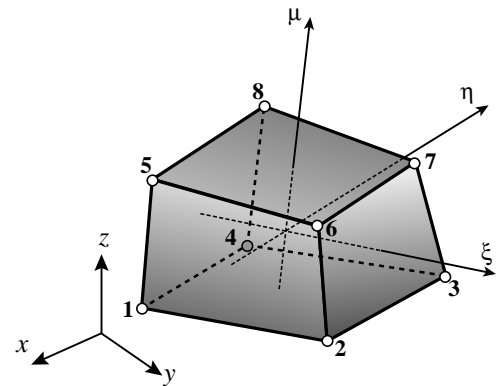


Figure E18.4. Eight-node isoparametric “brick” element for Exercise 18.7.

² Although this $N_1^{(e)}$ resembles the $N_1^{(e)}$ of the 4-node quadrilateral depicted in Figure 18.4, they are not the same. That shown in Figure E18.3 must vanish at node 5, that is, at $\xi = \eta = 0$. On the other hand, the $N_1^{(e)}$ of Figure 18.4 takes the value $\frac{1}{4}$ there.

EXERCISE 18.8

[A:15] Consider the 4-node transition triangular element of Figure 18.8(b). The shape function for node 1, $N_1 = \zeta_1 - 2\zeta_1\zeta_2$ was derived in §18.6.2. Show that the others are $N_2 = \zeta_2 - 2\zeta_1\zeta_2$, $N_3 = \zeta_3$ and $N_4 = 4\zeta_1\zeta_2$. Check that compatibility and completeness are verified.

EXERCISE 18.9

[A:20] Construct the six shape functions for the 6-node transition quadrilateral element of Figure 18.8(c). Hint: for the corner nodes, use two corrections to the shape functions of the 4-node bilinear quadrilateral. Check compatibility and completeness. Partial result: $N_1 = \frac{1}{4}(1 - \xi)(1 - \eta) - \frac{1}{4}(1 - \xi^2)(1 - \eta)$.

EXERCISE 18.10

[A:20] Consider a 5-node transition triangle in which midnode 6 on side 1–3 is missing. Show that $N_1^{(e)} = \zeta_1 - 2\zeta_1\zeta_2 - 2\zeta_2\zeta_3$. Can this be expressed as a line product like (18.1)?

EXERCISE 18.11

[A:30] The three-node linear triangle is known to be a poor performer for stress analysis. In an effort to improve it, Dr. I. M. Clueless proposes two sets of quadratic shape functions:

$$\text{CL1:} \quad N_1 = \zeta_1^2, \quad N_2 = \zeta_2^2, \quad N_3 = \zeta_3^2. \quad (\text{E18.1})$$

$$\text{CL2:} \quad N_1 = \zeta_1^2 + 2\zeta_2\zeta_3, \quad N_2 = \zeta_2^2 + 2\zeta_3\zeta_1, \quad N_3 = \zeta_3^2 + 2\zeta_1\zeta_2. \quad (\text{E18.2})$$

Dr. C. writes a learned paper claiming that both sets satisfy the interpolation condition, that set CL1 will work because it is conforming and that set CL2 will work because $N_1 + N_2 + N_3 = 1$. He provides no numerical examples. You get the paper for review. Show that the claims are false, and both sets are worthless. Hint: study §16.6 and Figure E18.5.

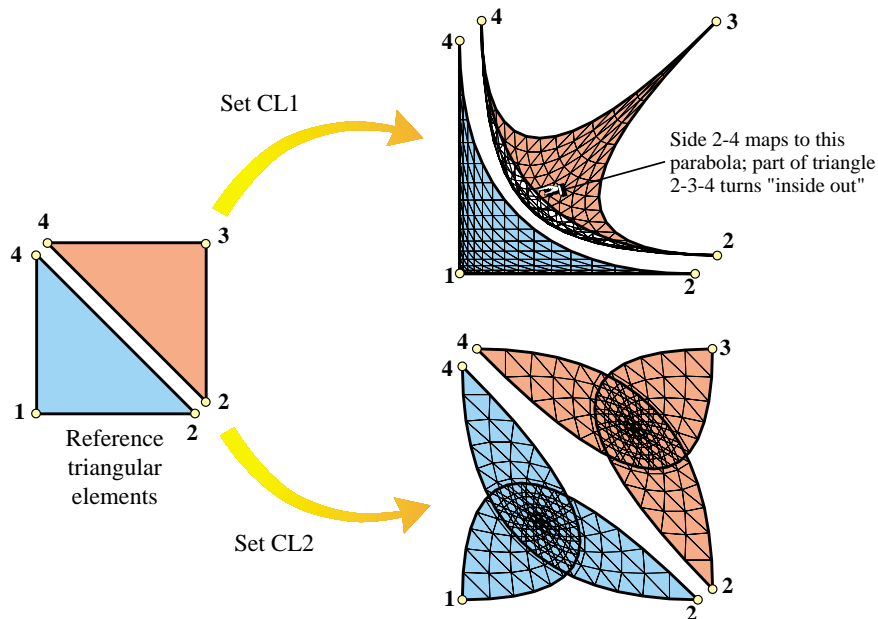


Figure E18.5. Mapping of reference triangles under sets (E18.3) and (E18.4). Triangles are slightly separated at the diagonal 2-4 for visualization convenience.

EXERCISE 18.12

[A:25] Another way of constructing shape functions for “incomplete” elements is through kinematic multi-freedom constraints (MFCs) applied to a “parent” element that contains the one to be derived. Suppose that the 9-node biquadratic quadrilateral is chosen as parent, with shape functions called N_i^P , $i = 1, \dots, 9$ given in §18.4.2. To construct the shape functions of the 8-node serendipity quadrilateral, the motions of node 9 are expressed in terms of the motions of the corner and midside nodes by the interpolation formulas

$$\begin{aligned} u_{x9} &= \alpha(u_{x1} + u_{x2} + u_{x3} + u_{x4}) + \beta(u_{x5} + u_{x6} + u_{x7} + u_{x8}), \\ u_{y9} &= \alpha(u_{y1} + u_{y2} + u_{y3} + u_{y4}) + \beta(u_{y5} + u_{y6} + u_{y7} + u_{y8}), \end{aligned} \quad (\text{E18.3})$$

where α and β are scalars to be determined. (In the terminology of Chapter 9, u_{x9} and u_{y9} are slaves.) Show that the shape functions of the 8-node quadrilateral are then $N_i = N_i^P + \alpha N_9^P$ for $i = 1, \dots, 4$ and $N_i = N_i^P + \beta N_9^P$ for $i = 5, \dots, 8$. Furthermore, show that α and β can be determined by two conditions:

1. The unit sum condition: $\sum_{i=1}^8 N_i = 1$, leads to $4\alpha + 4\beta = 1$.
2. Exactness of displacement interpolation for ξ^2 and η^2 leads to $2\alpha + \beta = 0$.

Solve these two equations for α and β , and verify that the serendipity shape functions given in §18.4.3 result.

EXERCISE 18.13

[A:25] Construct the 16 shape functions of the bicubic quadrilateral.

19

FEM Convergence Requirements

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§19.1. OVERVIEW

Chapters 12 through 18 have listed, in piecemeal fashion, various requirements for shape functions of isoparametric elements. These requirements are motivated by *convergence*: as the finite element mesh is refined, the solution should approach the analytical solution of the mathematical model.¹ This attribute is obviously necessary to instill confidence in FEM results from the standpoint of mathematics.

This Chapter provides unified information on convergence requirements. These requirements can be grouped into three:

Completeness. The elements must have enough *approximation power* to capture the analytical solution in the limit of a mesh refinement process. This intuitive statement is rendered more precise below.

Compatibility. The shape functions must provide *displacement continuity* between elements. Physically these insure that no material gaps appear as the elements deform. As the mesh is refined, such gaps would multiply and may absorb or release spurious energy.

Stability. The system of finite element equations must satisfy certain *well posedness* conditions that preclude nonphysical zero-energy modes in elements, as well as the absence of excessive element distortion.

Completeness and compatibility are two aspects of the so-called **consistency** condition between the discrete and mathematical models. A finite element model that passes both completeness and continuity requirements is called *consistent*. The famous Lax-Wendroff theorem² says that consistency and stability imply convergence.

A deeper mathematical analysis done in more advanced courses shows that completeness is *necessary* for convergence whereas failure of the other requirements does not necessarily precludes it. Nonetheless, the satisfaction of the three criteria guarantees convergence and may therefore be regarded as a safe choice.

§19.2. THE VARIATIONAL INDEX

For the mathematical statement of the completeness and continuity conditions, the variational index alluded to in previous sections plays a fundamental role.

The FEM is based on the direct discretization of an energy functional $\Pi[u]$, where u (displacements for the elements considered in this book) is the primary variable, or (equivalently) the function to be varied. Let m be the highest spatial derivative order of u that appears in Π . This m is called the *variational index*.

EXAMPLE 19.1

In the bar problem discussed in Chapter 12,

$$\Pi[u] = \int_0^L \left(\frac{1}{2} u' E A u' - q u \right) dx. \quad (19.1)$$

¹ Of course FEM convergence does not guarantee the correctness of the mathematical model in capturing the physics. As discussed in Chapter 1, that is a different and far more difficult problem.

² Proven originally for classical finite difference discretizations.

The highest derivative of the displacement $u(x)$ is $u' = du/dx$, which is first order in the space coordinate x . Consequently $m = 1$. This is also the case on the plane stress problem studied in Chapter 14, because the strains are expressed in terms of first order derivatives of the displacements.

EXAMPLE 19.2

In the plane beam problem discussed in Chapter 13,

$$\Pi[v] = \int_0^L \left(\frac{1}{2} v'' EI v'' - qv \right) dx. \quad (19.2)$$

The highest derivative of the transverse displacement is the curvature $\kappa = v'' = d^2v/dx^2$, which is of second order in the space coordinate x . Consequently $m = 2$.

§19.3. CONSISTENCY REQUIREMENTS

Using the foregoing definition of variational index, we can proceed to state the two key requirements for finite element shape functions.

§19.3.1. Completeness

The element shape functions must represent exactly all polynomial terms of order $\leq m$ in the Cartesian coordinates. A set of shape functions that satisfies this condition is called m -complete.

Note that this requirement applies *at the element level* and involves *all* shape functions of the element.

EXAMPLE 19.3

Suppose a displacement-based element is for a plane stress problem, in which $m = 1$. Then 1-completeness requires that the linear displacement field

$$u_x = \alpha_0 + \alpha_1 x + \alpha_2 y, \quad u_y = \alpha_0 + \alpha_1 x + \alpha_2 y \quad (19.3)$$

be exactly represented for any value of the α coefficients. This is done by evaluating (19.3) at the nodes to form a displacement vector $\mathbf{u}^{(e)}$ and then checking that $\mathbf{u} = \mathbf{N}^{(e)} \mathbf{u}^{(e)}$ recovers exactly (19.3). Section 16.6 presents the details of this calculation for an arbitrary isoparametric plane stress element. The analysis shows that completeness is satisfied if the *sum of the shape functions is unity* and the *element is compatible*.

EXAMPLE 19.4

For the plane beam problem, in which $m = 2$, the quadratic transverse displacement

$$v = \alpha_0 + \alpha_1 x + \alpha_2 x^2 \quad (19.4)$$

must be exactly represented over the element. This is easily verified in for the 2-node beam element developed in Chapter 13, because the assumed transverse displacement is a complete cubic polynomial in x . A complete cubic contains the quadratic (19.4) as special case.

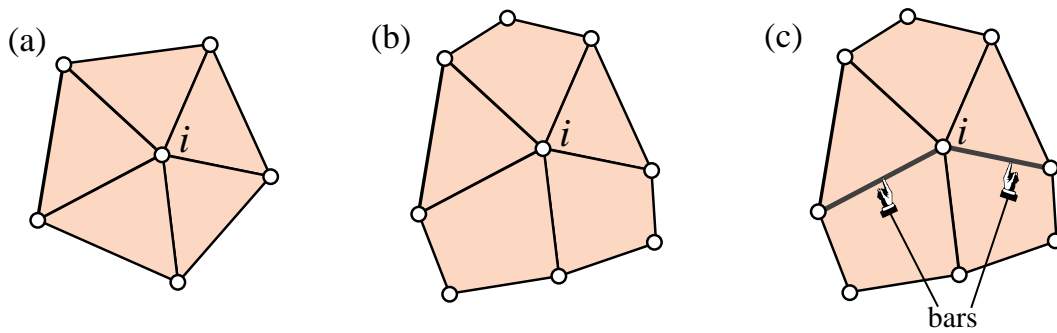


Figure 19.1. An element patch is the set of all elements attached to a patch node, herein labeled i . (a) illustrates a patch of triangles; (b) a mixture of triangles and quadrilaterals; (c) a mixture of triangles, quadrilaterals, and bars.

§19.3.2. Compatibility

To state this requirement succinctly, it is convenient to introduce the concept of *element patch*, or simply *patch*. This is the set of all elements attached to a given node. This node is called the *patch node*. The definition is illustrated in Figure 19.1, which shows three different kind of patches attached to patch node i in a plane stress problem. The patch of Figure 19.1(a) contains only one type of element: 3-node linear triangles. The patch of Figure 19.1(b) mixes two plane stress element types: 3-node linear triangles and 4-node bilinear quadrilaterals. The patch of Figure 19.1(c) combines three element types: 3-node linear triangles, 4-node bilinear quadrilaterals, and 2-node bars.

We define a finite element *patch trial function* as the union of shape functions activated by setting a degree of freedom at the patch node to unity, while all other freedoms are zero.

A patch trial function “propagates” only over the patch, and is zero beyond it. This property follows from the local-support requirement stated in §18.1.

With the benefit of these definitions we can enunciate the compatibility requirement as follows.

Patch trial functions must be $C^{(m-1)}$ continuous between interconnected elements, and C^m piecewise differentiable inside each element.

In the common case $m = 1$, the patch trial functions must be C^0 continuous between elements, and C^1 inside elements.

A set of shape functions that satisfies the first requirement is called *conforming*. A conforming expansion that satisfies the second requirement is said to be of *finite energy*. Note that this condition applies at two levels: individual element, and element patch. An element endowed with conforming shape functions is said to be *conforming*. A conforming element that satisfies the finite energy requirement is said to be *compatible*.³

³ The FEM literature is fuzzy as regards these terms. It seems better to leave the qualifier “conforming” to denote

Figures 19.1(b,c) illustrates the fact that one needs to check the possible connection of *matching elements* of different types and possibly different dimensionality.

As stated, compatibility refers to the *complete finite element mesh* because mesh trial functions are a combination of patch trial functions, which in turn are the union of element shape functions. This generality poses some logistical difficulties because the condition is necessarily mesh dependent. Compatibility can be checked at the *element level* by restricting attention to *matching meshes*. A matching mesh is one in which adjacent elements share sides, nodes and degrees of freedom, as in the patches shown in Figure 19.1.

For a matching mesh it is sufficient to restrict consideration first to a pair of adjacent elements, and then to the side shared by these elements. Suppose that the variation of a shape function *along that side* is controlled by k nodal values. Then a polynomial variation of order up to $k - 1$ in the natural coordinate(s) can be specified uniquely over the side. This is sufficient to verify interelement compatibility for $m = 1$, implying C^0 continuity, if the shape functions are polynomials.

This simplified criterion is the one used in previous Chapters. Specific 2D examples were given in Chapters 15 through 18.

REMARK 19.1

If the variational index is $m = 2$ and the problem is multidimensional, as in the case of plates and shells, the check is far more involved because continuity of *normal derivatives along a side* is involved. This practically important scenario is examined in advanced FEM treatments. The case of non-polynomial shape functions is, on the other hand, of little practical interest.

§19.4. STABILITY

Stability may be informally characterized as ensuring that the finite element model enjoys the same solution uniqueness properties of the analytical solution of the mathematical model. For example, if the only motions that produce zero internal energy in the mathematical model are rigid body motions, the finite element model must inherit that property. Since FEM can be arbitrary assemblies of elements, including individual elements, this property is required to hold at the element level.

In the present outline we are concerned with stability at the element level. Stability is not a property of shape functions *per se* but of the implementation of the element as well as its geometrical definition. It involves two subordinate requirements: rank sufficiency, and Jacobian positiveness.

§19.4.1. Rank Sufficiency

The element stiffness matrix must not possess any zero-energy kinematic mode other than rigid body modes.

This can be mathematically expressed as follows. Let n_F be the number of element degrees of freedom, and n_R be the number of independent rigid body modes. Let r denote the rank of $\mathbf{K}^{(e)}$. The element is called *rank sufficient* if $r = n_F - n_R$ and *rank deficient* if $r < n_F - n_R$. In the latter case,

$$d = (n_F - n_R) - r \quad (19.5)$$

interelement compatibility; informally “an element that gets along with its neighbors.” The qualifier “compatible” is used in the stricter sense of conforming while possessing sufficient internal smoothness.

Table 19.1 Rank-sufficient Gauss Rules for Some Plane Stress Elements

Element	n	n_F	$n_F - 3$	Min n_G	Recommended rule
3-node triangle	3	6	3	1	centroid*
6-node triangle	6	12	9	3	3-point rules*
10-node triangle	10	20	17	6	6-point rule*
4-node quadrilateral	4	8	5	2	2 x 2
8-node quadrilateral	8	16	13	5	3 x 3
9-node quadrilateral	9	18	15	5	3 x 3
16-node quadrilateral	16	32	29	10	4 x 4

* These triangle integration rules are introduced in §24.2.

is called the rank deficiency.

If an isoparametric element is numerically integrated, let n_G be the number of Gauss points, while n_E denotes the order of the stress-strain matrix \mathbf{E} . Two additional assumptions are made:

- (i) The element shape functions satisfy completeness in the sense that the rigid body modes are exactly captured by them.
- (ii) Matrix \mathbf{E} is of full rank.

Then each Gauss point adds n_E to the rank of $\mathbf{K}^{(e)}$, up to a maximum of $n_F - n_R$. Hence the rank of $\mathbf{K}^{(e)}$ will be

$$r = \min(n_F - n_R, n_E n_G) \quad (19.6)$$

To attain rank sufficiency, $n_E n_G$ must equal or exceed $n_F - n_R$:

$$n_E n_G \geq n_F - n_R \quad (19.7)$$

from which the appropriate Gauss integration rule can be selected.

In the plane stress problem, $n_E = 3$ because \mathbf{E} is a 3×3 matrix of elastic moduli; see Chapter 14. Also $n_R = 3$. Consequently $r = \min(n_F - 3, 3n_G)$ and $3n_G \geq n_F - 3$.

EXAMPLE 19.5

Consider a plane stress 6-node quadratic triangle. Then $n_F = 2 \times 6 = 12$. To attain the proper rank of $12 - n_R = 12 - 3 = 9$, $n_G \geq 3$. A 3-point Gauss rule, such as the midpoint rule defined in §24.2, makes the element rank sufficient.

EXAMPLE 19.6

Consider a plane stress 9-node biquadratic quadrilateral. Then $n_F = 2 \times 9 = 18$. To attain the proper rank of $18 - n_R = 18 - 3 = 15$, $n_G \geq 5$. The 2×2 product Gauss rule is insufficient because $n_G = 4$. Hence a 3×3 rule, which yields $n_G = 9$, is required to attain rank sufficiency.

Table 19.1 collects rank-sufficient Gauss integration rules for some widely used plane stress elements with n nodes and $n_F = 2n$ freedoms.

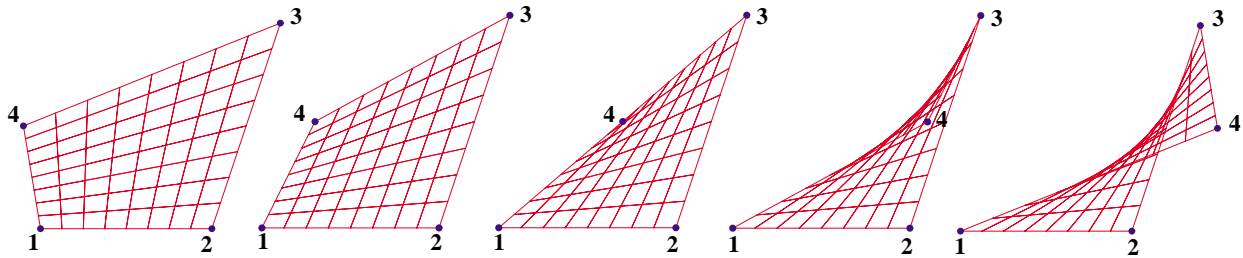


Figure 19.2. Effect of displacing node 4 of the four-node bilinear quadrilateral shown on the leftmost picture, to the right.

§19.4.2. Jacobian Positiveness

The geometry of the element must be such that the determinant $J = \det \mathbf{J}$ of the Jacobian matrix defined⁴ in §17.2, is positive everywhere. As illustrated in Equation (17.20), J characterizes the local metric of the element natural coordinates.

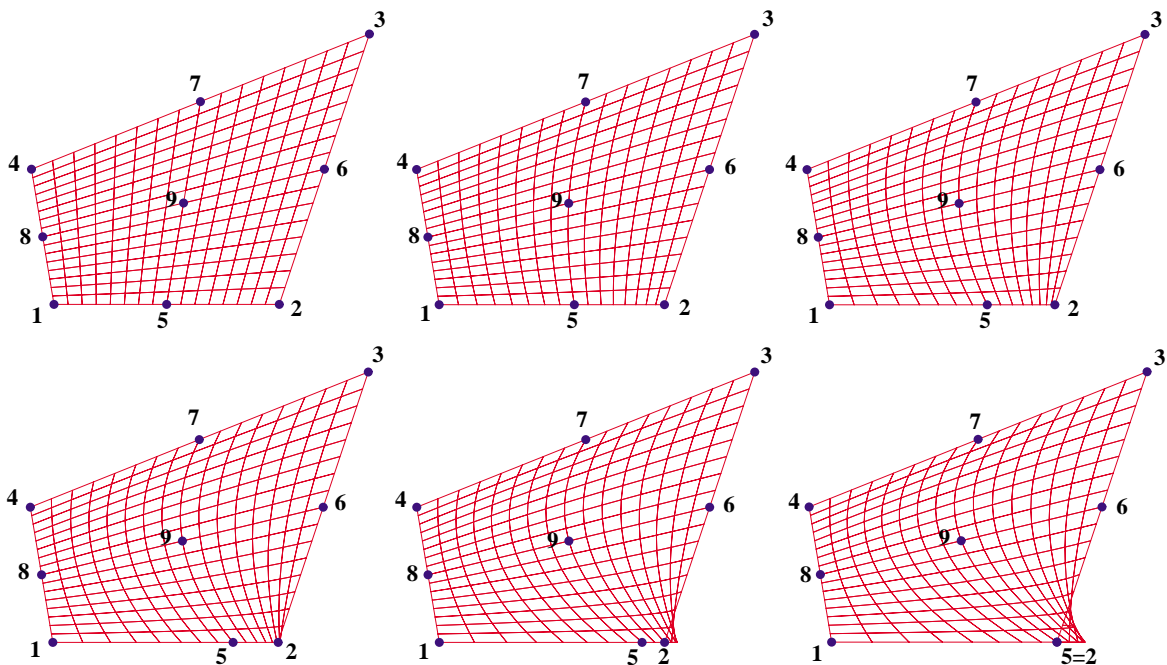


Figure 19.3. Effect of moving midpoint 5 of a 9-node biquadratic quadrilateral tangentially toward corner 2.

⁴ This definition applies to quadrilateral elements. The Jacobian determinant of an arbitrary triangular element is defined in §24.2.

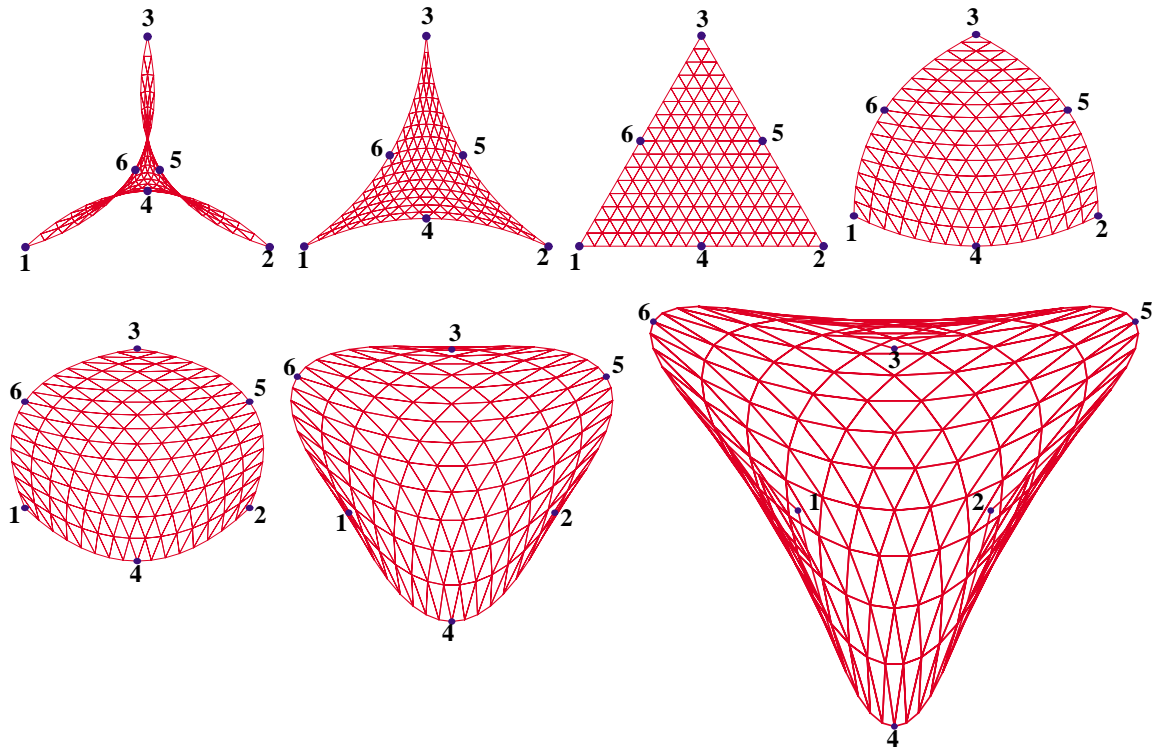


Figure 19.4. Effect of displacing midpoints 4, 5 and 6 of an equilateral 6-node triangle along the midpoint normals. Motion is inwards in first two top frames, outwards in the last four. In the lower leftmost picture nodes 1 through 6 lie on a circle.

For a three-node triangle J is constant and in fact equal to $2A$. The requirement $J > 0$ is equivalent to saying that corner nodes must be positioned and numbered so that a positive area $A > 0$ results. This is called a *convexity condition*. It is easily checked by a finite element program.

But for 2D elements with more than 3 nodes distortions may render *portions* of the element metric negative. This is illustrated in Figure 19.2 for a 4-node quadrilateral in which node 4 is gradually moved to the right. The quadrilateral morphs from a convex figure into a nonconvex one. The center figure is a triangle; note that the metric near node 4 is badly distorted (in fact $J = 0$ there) rendering the element unacceptable. This contradicts the (erroneous) advice of some FE books, which state that quadrilaterals can be reduced to triangles as special cases, thereby rendering triangular elements unnecessary.

For higher order elements proper location of corner nodes is not enough. The non-corner nodes (midside, interior, etc.) must be placed sufficiently close to their natural locations (midpoints, centroids, etc.) to avoid violent local distortions. The effect of midpoint motions in quadratic elements is illustrated in Figures 19.3 and 19.4.

Figure 19.3 depicts the effect of moving midside node 5 tangentially in a 9-node quadrilateral element while keeping all other 8 nodes fixed. When the location of 5 reaches the quarter-point of side 1-2, the metric at corner 2 becomes singular in the sense that $J = 0$ there. Although this is disastrous in ordinary FE work, it has applications in the construction of special “crack” elements

for linear fracture mechanics.

Displacing midside nodes normally to the sides is comparatively more forgiving, as illustrated in Figure 19.4. This depicts a 6-node equilateral triangle in which midside nodes 4, 5 and 6 are moved inwards and outwards along the normals to the midpoint location. As shown in the lower left picture, the element may be even morphed into a “parabolic circle” without the metric breaking down.

Notes and Bibliography

The literature on the mathematics of finite element methods has grown exponentially since the monograph of Strang and Fix [19.1]. This is very readable but out of print. A more up-to-date exposition is the textbook by Szabo and Babuska [19.2]. The subjects collected in this Chapter tend to be dispersed in recent monographs.

References

- [19.1] Strang, G., Fix, G., *An Analysis of the Finite Element Method*. Prentice-Hall, 1973.
- [19.2] Szabo, B., Babuska, I., *Finite Element Analysis*, Wiley, New York, 1991.

Homework Exercises for Chapter 19

FEM Convergence Requirements

EXERCISE 19.1

[D:15] Draw a picture of a 2D non-matching mesh in which element nodes on two sides of a boundary do not share the same locations. Discuss why enforcing compatibility becomes difficult.

EXERCISE 19.2

[A:25] The isoparametric definition of the straight 3-node bar element in its local system \bar{x} is

$$\begin{bmatrix} 1 \\ \bar{x} \\ \bar{v} \end{bmatrix} = \begin{bmatrix} 1 & 1 & 1 \\ \bar{x}_1 & \bar{x}_2 & \bar{x}_3 \\ \bar{u}_1 & \bar{u}_2 & \bar{u}_3 \end{bmatrix} \begin{bmatrix} N_1^{(e)}(\xi) \\ N_2^{(e)}(\xi) \\ N_3^{(e)}(\xi) \end{bmatrix} \quad (\text{E19.1})$$

where ξ is the isoparametric coordinate that takes the values -1 , 1 and 0 at nodes 1, 2 and 3, respectively, while $N_1^{(e)}$, $N_2^{(e)}$ and $N_3^{(e)}$ are the shape functions found in Exercise 16.3.

For simplicity, take $\bar{x}_1 = 0$, $\bar{x}_2 = L$, $\bar{x}_3 = \frac{1}{2}L + \alpha L$. Here L is the bar length and α a parameter that characterizes how far node 3 is away from the midpoint location $\bar{x} = \frac{1}{2}L$. Show that the minimum α 's (minimal in absolute value sense) for which $J = d\bar{x}/d\xi$ vanishes at a point in the element are $\pm 1/4$ (the quarter-points). Interpret this result as a singularity by showing that the axial strain becomes infinite at an end point.

EXERCISE 19.3

[A:15] Consider one dimensional bar-like elements with n nodes and 1 degree of freedom per node so $n_F = n$. The correct number of rigid body modes is 1. Each Gauss integration point adds 1 to the rank; that is $N_E = 1$. By applying (19.7), find the minimal rank-preserving Gauss integration rules with p points in the longitudinal direction if the number of node points is $n = 2, 3$ or 4 .

EXERCISE 19.4

[A:20] Consider three dimensional solid “brick” elements with n nodes and 3 degrees of freedom per node so $n_F = 3n$. The correct number of rigid body modes is 6. Each Gauss integration point adds 6 to the rank; that is, $N_E = 6$. By applying (19.7), find the minimal rank-preserving Gauss integration rules with p points in each direction (that is, $1 \times 1 \times 1$, $2 \times 2 \times 2$, etc) if the number of node points is $n = 8, 20, 27$, or 64 .

EXERCISE 19.5

[A/C:35] (Requires use of a CAS help to be tractable). Repeat Exercise 19.2 for a 9-node plane stress element. The element is initially a perfect square, nodes 5,6,7,8 are at the midpoint of the sides, and 9 at the center of the square. Displace 5 tangentially towards 4 until the jacobian determinant vanishes. This result is important in the construction of “singular elements” for fracture mechanics.

EXERCISE 19.6

[A/C:35] Repeat Exercise 19.5 but moving node 5 along the normal to the side. Discuss the range of motion for which $\det \mathbf{J} > 0$ within the element.

EXERCISE 19.7

[A:20] A plane stress triangular element has 3 nodes located at the midpoints of the sides. Develop the 3 shape functions and study whether the element satisfies compatibility and completeness.

21

Implementation of One-Dimensional Elements

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This Chapter begins Part III of the course, which deals with the computer implementation of the Finite Element Method. It is organized in “bottom up” fashion. It begins with simple topics, such as programming of bar and beam elements, and gradually builds up toward more complex models and calculations.

Specific examples of this Chapter illustrate the programming of one-dimensional elements: beams, using *Mathematica* as implementation language. The programming of both stiffness and mass matrices are illustrated although only the stiffness matrix is used in this course.

§21.1. THE PLANE BAR ELEMENT

The two-node, prismatic, two-dimensional bar element was studied in Chapters 2-3 for modeling plane trusses. It is reproduced in Figure 21.1 for convenience. It has two nodes and four degrees of freedom. The element node displacements and conjugate forces are

$$\mathbf{u}^{(e)} = \begin{bmatrix} u_{x1} \\ u_{y1} \\ u_{x2} \\ u_{y2} \end{bmatrix}, \quad \mathbf{f}^{(e)} = \begin{bmatrix} f_{x1} \\ f_{y1} \\ f_{x2} \\ f_{y2} \end{bmatrix}. \quad (21.1)$$

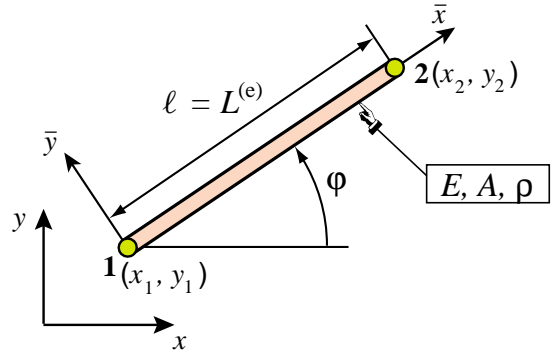


Figure 21.1. Plane bar element.

The element geometry is described by the coordinates $\{x_i, y_i\}$, $i = 1, 2$ of the two end nodes. The two material properties involved in the stiffness and mass computations are the modulus of elasticity E and the mass density per unit volume ρ . The only fabrication property required is the cross section area A . All of these properties are taken to be constant over the element.

§21.1.1. Element Formulation

The element stiffness matrix in global $\{x, y\}$ coordinates is given by the explicit expression derived in §3.1:

$$\mathbf{K}^{(e)} = \frac{EA}{\ell} \begin{bmatrix} c^2 & sc & -c^2 & -sc \\ sc & s^2 & -sc & -s^2 \\ -c^2 & -sc & c^2 & sc \\ -sc & -s^2 & sc & s^2 \end{bmatrix} = \frac{EA}{\ell^3} \begin{bmatrix} x_{21}x_{21} & x_{21}y_{21} & -x_{21}x_{21} & -x_{21}y_{21} \\ x_{21}y_{21} & y_{21}y_{21} & -x_{21}y_{21} & -y_{21}y_{21} \\ -x_{21}x_{21} & -x_{21}y_{21} & x_{21}x_{21} & x_{21}y_{21} \\ -x_{21}y_{21} & -y_{21}y_{21} & x_{21}y_{21} & y_{21}y_{21} \end{bmatrix}. \quad (21.2)$$

Here $c = \cos \varphi = x_{21}/\ell$, $s = \sin \varphi = y_{21}/\ell$, in which $x_{21} = x_2 - x_1$, $y_{21} = y_2 - y_1$, $\ell = \sqrt{x_{21}^2 + y_{21}^2}$, and φ is the angle formed by \bar{x} and x , measured from x positive counterclockwise (see Figure 21.1). The second matrix expression in (21.2) is useful in symbolic work, because it enhances simplification possibilities.

The consistent and lumped mass matrix are given by

$$\mathbf{M}_C^{(e)} = \frac{\rho A \ell}{6} \begin{bmatrix} 2 & 0 & 1 & 0 \\ 0 & 2 & 0 & 1 \\ 1 & 0 & 2 & 0 \\ 0 & 1 & 0 & 2 \end{bmatrix}, \quad \mathbf{M}_L^{(e)} = \frac{\rho A \ell}{2} \begin{bmatrix} 1 & 0 & 0 & 0 \\ 0 & 1 & 0 & 0 \\ 0 & 0 & 1 & 0 \\ 0 & 0 & 0 & 1 \end{bmatrix}. \quad (21.3)$$

```

PlaneBar2Stiffness[ncoor_,mprop_,fprop_,opt_]:= Module[
{
  {x1,x2,y1,y2,x21,y21,Em,Gm,ρ,α,A,numer,L,LL,LLL,Ke},
  {{x1,y1},{x2,y2}}=ncoor; {x21,y21}={x2-x1,y2-y1};
  {Em,Gm,ρ,α}=mprop; {A}=fprop; {numer}=opt;
  If [numer,{x21,y21,Em,A}=N[{x21,y21,Em,A}]];
  LL=x21^2+y21^2; L=PowerExpand[Sqrt[LL]]; LLL=Simplify[LL*L];
  Ke=(Em*A/LLL)*{
    { x21*x21, x21*y21,-x21*x21,-x21*y21},
    { y21*x21, y21*y21,-y21*x21,-y21*y21},
    { -x21*x21,-x21*y21, x21*x21, x21*y21},
    { -y21*x21,-y21*y21, y21*x21, y21*y21}};

  Return[Ke]
];

PlaneBar2ConsMass[ncoor_,mprop_,fprop_,opt_]:= Module[
{
  {x1,x2,y1,y2,x21,y21,Em,Gm,ρ,α,A,numer,L,MeC},
  {{x1,y1},{x2,y2}}=ncoor; {x21,y21}={x2-x1,y2-y1};
  {Em,Gm,ρ,α}=mprop; {A}=fprop; {numer}=opt;
  If [numer,{x21,y21,ρ,A}=N[{x21,y21,ρ,A}]];
  L=Simplify[PowerExpand[Sqrt[x21^2+y21^2]]];
  MeC=(ρ*A*L/6)*{
    {2,0,1,0},{0,2,0,1},{1,0,2,0},{0,1,0,2}};
  Return[MeC]
];

PlaneBar2LumpMass[ncoor_,mprop_,fprop_,opt_]:= Module[
{
  {x1,x2,y1,y2,x21,y21,Em,Gm,ρ,α,A,numer,L,MeL},
  {{x1,y1},{x2,y2}}=ncoor; {x21,y21}={x2-x1,y2-y1};
  {Em,Gm,ρ,α}=mprop; {A}=fprop; {numer}=opt;
  If [numer,{x21,y21,ρ,A}=N[{x21,y21,ρ,A}]];
  L=Simplify[PowerExpand[Sqrt[x21^2+y21^2]]];
  MeL=(ρ*A*L/2)*IdentityMatrix[4];
  Return[MeL]
];

```

Figure 21.2. *Mathematica* modules to form stiffness and mass matrices of a 2-node, prismatic plane bar element.

Both mass matrices are independent of the orientation of the element, that is, do not depend on the angle φ .¹

§21.1.2. Element Formation Modules

Figure 21.2 lists three *Mathematica* modules:

PlaneBar2Stiffness returns the element stiffness matrix $\mathbf{K}^{(e)}$ of a 2-node, prismatic plane bar element, given by (21.2). PlaneBar2ConsMass returns the consistent mass matrix $\mathbf{M}_C^{(e)}$ of a 2-node, prismatic plane bar element, given by (21.3). PlaneBar2LumpMass returns the lumped mass matrix $\mathbf{M}_L^{(e)}$ of a 2-node, prismatic plane bar element, given by (21.3).

The argument sequence of the three modules is exactly the same:

[ncoor, mprop, fprop, opt]

These four arguments are actually *lists* that collect the following data:

node coordinates, material properties, fabrication properties, options

¹ For the derivation of the consistent mass matrix, see references in **Notes and Bibliography** at the end of this Chapter.

The internal structure of these lists is

$$\begin{aligned}
 \text{ncoor} & \quad \{ \{ x1, y1 \}, \{ x2, y2 \} \} \\
 \text{mprop} & \quad \{ E, G, \rho, \alpha \} \\
 \text{fprop} & \quad \{ A \} \\
 \text{opt} & \quad \{ \text{numer} \}
 \end{aligned} \tag{21.4}$$

Here $x1, y1$ and $x2, y2$ are the coordinates of the end nodes, and E, G, ρ, α and A stand for E, G, ρ, α and A , respectively. For the stiffness matrix only E and A are used. In the mass matrix modules only ρ and A are used. Entries G and α are included to take care of other 1D elements.

The only option is `numer`, which is a logical flag with the value `True` or `False`. If `True` the computations are carried out in numerical floating-point arithmetic.

§21.1.3. Testing the Plane Bar Element Modules

The modules are tested by the statements listed in Figures 21.3 and 21.5.

The script of Figure 21.3 tests a numerically defined element with end nodes located at $(0, 0)$ and $(30, 40)$, with $E = 1000$, $A = 5$, $\rho = 6/10$, and `numer` set to `True`. Executing the statements in Figure 21.3 produces the results listed in Figure 21.4. The tests consist of the following operations:

Testing $\mathbf{K}^{(e)}$. The stiffness matrix returned in `Ke` is printed. Its four eigenvalues are computed and printed. As expected three eigenvalues, which correspond to the three independent rigid body motions of the element, are zero. The remaining eigenvalue is positive and equal to EA/ℓ . The symmetry of `Ke` is also checked but the output is not shown to save space.

Testing $\mathbf{M}_C^{(e)}$. The consistent mass matrix returned in `MeC` is printed. Its four eigenvalues are computed and printed. As expected they are all positive and form two pairs. The symmetry is also checked but the output is now shown.

Testing $\mathbf{M}_L^{(e)}$. The lumped mass matrix returned in `MeL` is printed. This is a diagonal matrix and consequently its eigenvalues are the same as the diagonal entries.

The script of Figure 21.5 tests a symbolically defined element with end nodes located at $(0, 0)$ and $(L, 0)$, which is aligned with the x axis. The element properties E, ρ and A are kept symbolic. Executing the statements in Figure 21.4 produces the results shown in Figure 21.5.

The sequence of tests on the symbolic element is essentially the same carried out before, but a frequency test has been added. This consists of solving the one-element vibration eigenproblem

$$\mathbf{K}^{(e)} \mathbf{v}_i = \omega_i^2 \mathbf{M}_C^{(e)} \mathbf{v}_i, \quad \mathbf{K}^{(e)} \mathbf{v}_i = \omega_i^2 \mathbf{M}_L^{(e)} \mathbf{v}_i, \tag{21.5}$$

in which ω_i are circular frequencies and \mathbf{v}_i the associated eigenvectors or vibration mode shapes. Because the 3 rigid body modes are solution of (21.5), three zero frequencies are expected, which is borne out by the tests.

The single positive nonzero frequency $\omega_a > 0$ corresponds to the vibration mode of axial extension and contraction. The consistent mass yields $\omega_a^2 = 12E/(\rho L^2)$ whereas the lumped mass yields $\omega_a^2 = 4E/(\rho L^2)$. The exact continuum solution for this axial mode is $\omega_a^2 = \pi^2 E/(\rho L^2) \approx$

```

ncoor={{0,0},{30,40}}; mprop={1000,0,6/10,0};
fprop={5}; opt={True};

Ke= PlaneBar2Stiffness[ncoor,mprop,fprop,opt];
Print["Numerical Elem Stiff Matrix: "];
Print[Ke//MatrixForm];
Print["Eigenvalues of Ke=",Chop[Eigenvalues[N[Ke]]]];
Print["Symmetry check=",Simplify[Transpose[Ke]-Ke]];

MeC= PlaneBar2ConsMass[ncoor,mprop,fprop,opt];
Print["Numerical Consistent Mass Matrix: "];
Print[MeC//MatrixForm];
Print["Eigenvalues of MeC=",Eigenvalues[N[MeC]]];
Print["Symmetry check=",Simplify[Transpose[MeC]-MeC]];

MeL= PlaneBar2LumpMass[ncoor,mprop,fprop,opt];
Print["Numerical Lumped Mass Matrix: "];
Print[MeL//MatrixForm];
Print["Eigenvalues of MeL=",Eigenvalues[N[MeL]]];

```

Figure 21.3. Test of plane bar element modules with numerical inputs.

```

Numerical Elem Stiff Matrix:
( 36.  48. -36. -48. )
( 48.  64. -48. -64. )
( -36. -48.  36.  48. )
( -48. -64.  48.  64. )
Eigenvalues of Ke={200., 0, 0, 0}
Numerical Consistent Mass Matrix:
( 50.  0  25.  0 )
( 0  50.  0  25. )
( 25.  0  50.  0 )
( 0  25.  0  50. )
Eigenvalues of MeC={75., 75., 25., 25.}
Numerical Lumped Mass Matrix:
( 75.  0  0  0 )
( 0  75.  0  0 )
( 0  0  75.  0 )
( 0  0  0  75. )
Eigenvalues of MeL={75., 75., 75., 75.}

```

Figure 21.4. Output from test statements of Figure 21.3 (output from symmetry checks not shown).

$9.86E/(\rho L^2)$. Hence the consistent mass overestimates the true frequency whereas the lumped mass underestimates it, which is a well known property.

Running the script of Figure 21.5 produces the output shown in Figure 21.6. One thing to be noticed is the use of stiffness and mass matrix scaling factors, called *kfac* and *mfac*, respectively, in Figure 21.5. These embody symbolic quantities that can be taken out as matrix factors, for example EA/L


```

ClearAll[A,Em,ρ,L,opt];
ncoor={{0,0},{L,0}}; mprop={Em,0,ρ,0}; fprop={A}; opt={False};
Ke= PlaneBar2Stiffness[ncoor,mprop,fprop,opt];
kfac=Em*A/L; Ke=Simplify[Ke/kfac];
Print["Symbolic Elem Stiff Matrix: "];
Print[kfac," ",Ke//MatrixForm];
Print["Eigenvalues of Ke=",kfac,"*",Eigenvalues[Ke]];
MeC= PlaneBar2ConsMass[ncoor,mprop,fprop,opt];
mfac=ρ*L*A/6; MeC= Simplify[MeC/mfac];
Print["Symbolic Consistent Mass Matrix: "];
Print[mfac," ",MeC//MatrixForm];
Print["Eigenvalues of MeC=",mfac,"*",Eigenvalues[MeC]];
Print["Squared frequencies=",Simplify[kfac/mfac],"*",
      Simplify[Eigenvalues[Inverse[MeC].Ke]]];
MeL= PlaneBar2LumpMass[ncoor,mprop,fprop,opt];
mfac=ρ*L*A/2; MeL= Simplify[MeL/mfac];
Print["Symbolic Lumped Mass Matrix: "];
Print[mfac," ",MeL//MatrixForm];
Print["Eigenvalues of MeL=",mfac,"*",Eigenvalues[MeL]];
Print["Squared frequencies=",Simplify[kfac/mfac],"*",
      Simplify[Eigenvalues[Inverse[MeC].Ke]]];

```

Figure 21.5. Test of plane bar element modules with symbolic inputs.

```

Symbolic Elem Stiff Matrix:

$$\frac{A E m}{L} \begin{pmatrix} 1 & 0 & -1 & 0 \\ 0 & 0 & 0 & 0 \\ -1 & 0 & 1 & 0 \\ 0 & 0 & 0 & 0 \end{pmatrix}$$

Eigenvalues of Ke= $\frac{A E m}{L} * \{0, 0, 0, 2\}$ 
Symbolic Consistent Mass Matrix:

$$\frac{A L \rho}{6} \begin{pmatrix} 2 & 0 & 1 & 0 \\ 0 & 2 & 0 & 1 \\ 1 & 0 & 2 & 0 \\ 0 & 1 & 0 & 2 \end{pmatrix}$$

Eigenvalues of MeC= $\frac{A L \rho}{6} * \{1, 1, 3, 3\}$ 
Squared frequencies= $\frac{6 E m}{L^2 \rho} * \{0, 0, 0, 2\}$ 
Symbolic Lumped Mass Matrix:

$$\frac{A L \rho}{2} \begin{pmatrix} 1 & 0 & 0 & 0 \\ 0 & 1 & 0 & 0 \\ 0 & 0 & 1 & 0 \\ 0 & 0 & 0 & 1 \end{pmatrix}$$

Eigenvalues of MeL= $\frac{A L \rho}{2} * \{1, 1, 1, 1\}$ 
Squared frequencies= $\frac{2 E m}{L^2 \rho} * \{0, 0, 0, 2\}$ 

```

Figure 21.6. Output from test statements of Figure 21.5.

in $\mathbf{K}^{(e)}$. The effect is to clean up matrix and vector output, as can be observed in Figure 21.6.

§21.2. THE PLANE BEAM-COLUMN ELEMENT

Beam-column elements model structural members that resist both axial and bending actions. This is the case in skeletal structures such as frameworks which are common in buildings. A plane beam-column element is a combination of a plane bar (such as that considered in §21.1), and a plane beam.

We consider such an element in its local system (\bar{x}, \bar{y}) as shown in Figure 21.7, and then in the global system (x, y) as shown in Figure 21.8. The six degrees of freedom and conjugate node forces of the elements are:

$$\bar{\mathbf{u}}^{(e)} = \begin{bmatrix} \bar{u}_{x1} \\ \bar{u}_{y1} \\ \theta_{z1} \\ \bar{u}_{x2} \\ \bar{u}_{y2} \\ \theta_{z2} \end{bmatrix}, \quad \bar{\mathbf{f}}^{(e)} = \begin{bmatrix} \bar{f}_{x1} \\ \bar{f}_{y1} \\ \bar{m}_{z1} \\ \bar{f}_{x2} \\ \bar{f}_{y2} \\ \bar{m}_{z2} \end{bmatrix}, \quad \mathbf{u}^{(e)} = \begin{bmatrix} u_{x1} \\ u_{y1} \\ \theta_{z1} \\ u_{x2} \\ u_{y2} \\ \theta_{z2} \end{bmatrix}, \quad \mathbf{f}^{(e)} = \begin{bmatrix} f_{x1} \\ f_{y1} \\ m_{z1} \\ f_{x2} \\ f_{y2} \\ m_{z2} \end{bmatrix}. \quad (21.6)$$

The rotation angles θ and the nodal moments m are the same in the local and the global systems because they are about the z axis, which does not change.

The element geometry is described by the coordinates $\{x_i, y_i\}$, $i = 1, 2$ of the two end nodes. The element length is ℓ . The two material properties involved in the stiffness and mass computations are the modulus of elasticity E and the mass density per unit volume ρ . The fabrication properties required are the cross section area A and the bending moment of inertia $I = I_{zz}$ about the neutral axis, which is taken as defining the position of the z axis. All of these properties are taken to be constant over the element.

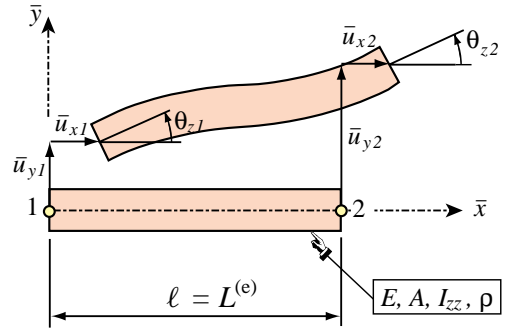


Figure 21.7. Plane beam-column element in its local system.

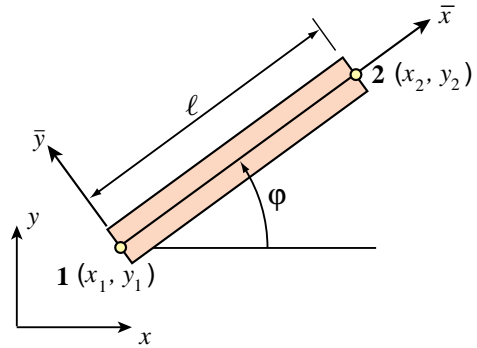


Figure 21.8. Plane beam-column element transformation to global system.

§21.2.1. Element Formulation

For the plane bar and plane beam components we use the elements derived in Chapters 12 and 13, respectively. (For beam bending this is the Euler-Bernoulli mathematical model.) Combining these two we obtain the stiffness matrix of prismatic beam-column element in the local system \bar{x}, \bar{y} as

$$\bar{\mathbf{K}}^{(e)} = \frac{EA}{\ell} \begin{bmatrix} 1 & 0 & 0 & -1 & 0 & 0 \\ 0 & 0 & 0 & 0 & 0 & 0 \\ 0 & 0 & 0 & 0 & 0 & 0 \\ 0 & 0 & 0 & 1 & 0 & 0 \\ 0 & 0 & 0 & 0 & 0 & 0 \\ \text{symm} & & & & & 0 \end{bmatrix} + \frac{EI}{\ell^3} \begin{bmatrix} 0 & 0 & 0 & 0 & 0 & 0 \\ 0 & 12 & 6\ell & 0 & -12 & 6\ell \\ 0 & 6\ell & 4\ell^2 & 0 & -6\ell & 2\ell^2 \\ 0 & 0 & 0 & 0 & 0 & 0 \\ 0 & -12 & 6\ell & 0 & 12 & -6\ell \\ 0 & 6\ell & 2\ell^2 & 0 & -6\ell & 4\ell^2 \end{bmatrix} \quad (21.7)$$

```

PlaneBeamColumn2Stiffness[ncoor_,mprop_,fprop_,opt_]:= Module[
{xl,x2,y1,y2,x21,y21,Em,Gm,ρ,α,A,Izz,numer,c,s,L,LL,
  LLL,ra,rb,T,Kebar,Ke},
{{x1,y1},{x2,y2}}=ncoor; {x21,y21}={x2-x1,y2-y1};
{Em,Gm,ρ,α}=mprop; {A,Izz}=fprop; {numer}=opt;
LL=Simplify[x21^2+y21^2]; L=PowerExpand[Sqrt[LL]]; LLL=L*LL;
c=x21/L; s=y21/L; ra=Em*A/L; rb= 2*Em*Izz/LLL;
Kebar= ra*{
{ 1,0,0,-1,0,0},{0,0,0,0,0,0},{0,0,0,0,0,0},
{-1,0,0, 1,0,0},{0,0,0,0,0,0},{0,0,0,0,0,0}} +
rb*{
{ 0,0,0,0,0,0},{0, 6, 3*L,0,-6, 3*L},{0,3*L,2*LL,0,-3*L, LL},
{ 0,0,0,0,0,0},{0,-6,-3*L,0, 6,-3*L},{0,3*L, LL,0,-3*L,2*LL}};
T={ {c,s,0,0,0,0},{-s,c,0,0,0,0},{0,0,1,0,0,0},
{0,0,0,c,s,0},{0,0,0,-s,c,0},{0,0,0,0,0,1}};
Ke=Transpose[T].Kebar.T; If [numer,Ke=N[Ke]];
Return[Ke]
];

```

Figure 21.9. 9. *Mathematica* module to form the stiffness matrix of a 2-node, prismatic plane beam-column element.

The first matrix on the right is the contribution from the bar stiffness, but in which rows and columns have been rearranged in accordance with the nodal freedoms (21.6). The second matrix is the contribution from the Bernoulli-Euler bending stiffness; again freedoms have been rearranged as appropriate.

The consistent mass matrix in the local system \bar{x}, \bar{y} is

$$\bar{\mathbf{M}}_c^{(e)} = \frac{\rho A \ell}{6} \begin{bmatrix} 2 & 0 & 0 & 1 & 0 & 0 \\ & 0 & 0 & 0 & 0 & 0 \\ & & 0 & 0 & 0 & 0 \\ & & & 2 & 0 & 0 \\ & & & & 0 & 0 \\ \text{symm} & & & & & 0 \end{bmatrix} + \frac{\rho A \ell}{420} \begin{bmatrix} 0 & 0 & 0 & 0 & 0 & 0 \\ & 156 & 21\ell & 0 & 54 & -13\ell \\ & & 4\ell^2 & 0 & 13\ell & -3\ell^2 \\ & & & 0 & 0 & 0 \\ & & & & 156 & -21\ell \\ \text{symm} & & & & & 4\ell^2 \end{bmatrix} \quad (21.8)$$

The first matrix on the right is the contribution from the axial (bar) inertia, whereas the second one comes from the bending (beam) inertia. The expression for the latter neglect the shear and rotatory inertia. Their derivation may be followed in the book of Przemieniecki cited in footnote 1.

The displacement transformation matrix between local and global systems is

$$\bar{\mathbf{u}}^{(e)} = \begin{bmatrix} \bar{u}_{x1} \\ \bar{u}_{y1} \\ \theta_{z1} \\ u_{x2} \\ \bar{u}_{y2} \\ \theta_{z2} \end{bmatrix} = \begin{bmatrix} c & s & 0 & 0 & 0 & 0 \\ -s & c & 0 & 0 & 0 & 0 \\ 0 & 0 & 1 & 0 & 0 & 0 \\ 0 & 0 & 0 & c & s & 0 \\ 0 & 0 & 0 & -s & c & 0 \\ 0 & 0 & 0 & 0 & 0 & 1 \end{bmatrix} \begin{bmatrix} u_{x1} \\ u_{y1} \\ \theta_{z1} \\ u_{x2} \\ u_{y2} \\ \theta_{z2} \end{bmatrix} = \mathbf{T} \mathbf{u}^{(e)} \quad (21.9)$$

where $c = \cos \varphi$, $s = \sin \varphi$, and φ is the angle between \bar{x} and x , measured positive-counterclockwise from x ; see Figure 21.3. The stiffness and consistent mass matrix in the global system are obtained

```

PlaneBeamColumn2ConsMass[ncoor_,mprop_,fprop_,opt_]:= Module[
{
x1,x2,y1,y2,x21,y21,Em,Gm,ρ,α,A,Izz,numer,c,s,L,LL,m,
T,MeClocal,MeC},
{
{x1,y1},{x2,y2}}=ncoor; {x21,y21}={x2-x1,y2-y1};
{Em,Gm,ρ,α}=mprop; {A,Izz}=fprop; {numer}=opt;
LL=x21^2+y21^2; L=PowerExpand[Sqrt[LL]];
c=x21/L; s=y21/L; m=ρ*L*A;
MeClocal= (m/6)*{
{2,0,0,1,0,0},{0,0,0,0,0,0},{0,0,0,0,0,0},
{1,0,0,2,0,0},{0,0,0,0,0,0},{0,0,0,0,0,0}}+
(m/420)*{
{0,0,0,0,0,0},{0,156,22*L,0,54,-13*L},
{0,22*L,4*LL,0,13*L,-3*LL},
{0,0,0,0,0,0},{0,54,13*L,0,156,-22*L},
{0,-13*L,-3*LL,0,-22*L,4*LL}}};
T={{c,s,0,0,0,0},{-s,c,0,0,0,0},{0,0,1,0,0,0},
{0,0,0,c,s,0},{0,0,0,-s,c,0},{0,0,0,0,0,1}};
MeC=Transpose[T].MeClocal.T; If [numer,MeC=N[MeC]];
Return[MeC]
};

PlaneBeamColumn2LumpMass[ncoor_,mprop_,fprop_,opt_]:= Module[
{
x1,x2,y1,y2,x21,y21,Em,Gm,ρ,α,A,Izz,numer,L,LL,MeL},
{
{x1,y1},{x2,y2}}=ncoor; {x21,y21}={x2-x1,y2-y1};
{Em,Gm,ρ,α}=mprop; {A,Izz}=fprop; {numer}=opt;
LL=Simplify[x21^2+y21^2]; L=PowerExpand[Sqrt[LL]];
(* HRZ lumping scheme for rotational masses *)
MeL=(ρ*A*L/2)*
{{1,0,0,0,0,0},{0,1,0,0,0,0},{0,0,LL/39,0,0,0},
{0,0,0,1,0,0},{0,0,0,0,1,0},{0,0,0,0,0,LL/39}};
If [numer,MeL=N[MeL]];
Return[MeL]
};

```

Figure 21.10. 10. *Mathematica* modules to form mass matrices of a 2-node, prismatic plane beam-column element.

through the congruential transformation $\mathbf{K}^{(e)} = \mathbf{T}^T \bar{\mathbf{K}}^{(e)} \mathbf{T}$, $\mathbf{M}_C^{(e)} = \mathbf{T}^T \bar{\mathbf{M}}^{(e)} \mathbf{T}$. The lumped mass matrix is diagonal and is the same in the local and global systems:

$$\mathbf{M}_L^{(e)} = \bar{\mathbf{M}}_L^{(e)} = \frac{\rho \ell}{2} \begin{bmatrix} 1 & 0 & 0 & 0 & 0 & 0 \\ 0 & 1 & 0 & 0 & 0 & 0 \\ 0 & 0 & m_\theta & 0 & 0 & 0 \\ 0 & 0 & 0 & 1 & 0 & 0 \\ 0 & 0 & 0 & 0 & 1 & 0 \\ 0 & 0 & 0 & 0 & 0 & m_\theta \end{bmatrix}. \quad (21.10)$$

Here the rotational lumped mass is shown as m_θ . There are several ways to compute m_θ , including the simplest one of leaving it zero. In the implementation shown in Figure 21.6, $m_\theta = \rho A \ell^3 / 78$. This is called the *HRZ lumping scheme* in the FEM literature.

§21.2.2. Element Formation Modules

Figures 21.9 and 21.10 list three *Mathematica* modules:

```

ncoor={{0,0},{3,4}}; mprop={100,0,84/5,0}; fprop={125,250};
opt={True};

Ke= PlaneBeamColumn2Stiffness[ncoor,mprop,fprop,opt];
Print["Numerical Elem Stiff Matrix: "];
Print[Ke//MatrixForm];
Print["Eigenvalues of Ke=",Chop[Eigenvalues[Ke]]];

MeC= PlaneBeamColumn2ConsMass[ncoor,mprop,fprop,opt];
Print["Numerical Consistent Mass Matrix: "];
Print[MeC//MatrixForm];
Print["Eigenvalues of MeC=",Eigenvalues[MeC]];
Print["Squared frequencies (consistent)=",
      Chop[Eigenvalues[Inverse[MeC].Ke],10^(-7)]];

MeL= PlaneBeamColumn2LumpMass[ncoor,mprop,fprop,opt];
Print["Numerical Lumped Mass Matrix: "];
Print[MeL//MatrixForm];
Print["Eigenvalues of MeL=",Eigenvalues[MeL]];
Print["Squared frequencies (lumped)=",
      Chop[Eigenvalues[Inverse[MeL].Ke],10^(-7)]];

```

Figure 21.11. Test of 2-node plane beam-column element with numeric inputs.

```

Numerical Elem Stiff Matrix:
( 2436.   48.  -4800.  -2436.  -48.  -4800. )
( 48.   2464.  3600.   -48.  -2464.  3600. )
( -4800. 3600. 20000.  4800. -3600. 10000. )
( -2436. -48.  4800.  2436.   48.   4800. )
( -48.  -2464. -3600.   48.   2464. -3600. )
( -4800. 3600. 10000.  4800. -3600. 20000. )
Eigenvalues of Ke={34800., 10000., 5000., 0, 0, 0}
Numerical Consistent Mass Matrix:
( 3756. -192. -2200. 1494.  192. 1300. )
( -192. 3644. 1650.  192. 1606. -975. )
( -2200. 1650. 2500. -1300. 975. -1875. )
( 1494.  192. -1300. 3756. -192. 2200. )
(  192. 1606.  975. -192. 3644. -1650. )
( 1300. -975. -1875. 2200. -1650. 2500. )
Eigenvalues of MeC={9209.32, 5250., 3068.05, 1750., 415.679, 106.949}
Squared frequencies (consistent)={160., 13.7143, 2.85714, 0, 0, 0}
Numerical Lumped Mass Matrix:
( 5250.  0.  0.  0.  0.  0. )
(  0.  5250.  0.  0.  0.  0. )
(  0.  0. 3365.38  0.  0.  0. )
(  0.  0.  0.  5250.  0.  0. )
(  0.  0.  0.  0.  5250.  0. )
(  0.  0.  0.  0.  0. 3365.38 )
Eigenvalues of MeL={5250., 5250., 5250., 5250., 3365.38, 3365.38}
Squared frequencies (lumped)={9.82857, 2.97143, 0.952381, 0, 0, 0}

```

Figure 21.12. Output from test statements of Figure 21.11.

`PlaneBeamColumn2Stiffness` returns the element stiffness matrix $\mathbf{K}^{(e)}$ of a 2-node, prismatic plane beam-column element, given by (21.7). `PlaneBeamColumn2ConsMass` returns the consistent mass matrix $\mathbf{M}_C^{(e)}$ of a 2-node, prismatic plane beam-column element, given by (21.8). `PlaneBeamColumn2LumpMass` returns the lumped mass matrix $\mathbf{M}_L^{(e)}$ of a 2-node, prismatic plane beam-column element, given by (21.10).

As in the case of the plane bar, the calling argument sequence of the three modules is exactly the same:

[ncoor, mprop, fprop, opt]

These four arguments are *lists* that collect the following data:

node coordinates, material properties, fabrication properties, options

The internal structure of these lists is

ncoor	{ { x1, y1 }, { x2, y2 } }	
mprop	{ Em, Gm, rho, alpha }	
fprop	{ A, Izz }	
opt	{ num }	(21.11)

Here x_1, y_1 and x_2, y_2 are the coordinates of the end nodes, while E, G, ρ, α, A and I_{zz} stand for E, G, ρ, α, A and I_{zz} , respectively. For the stiffness matrix only E, A and I_{zz} are used. In the mass matrix modules only ρ and A are used. Entries Gm and α are included to take care of other 1D elements.

The only option is `numer`, which is a logical flag with the value `True` or `False`. If `numer=True` the computations are carried out in numerical floating-point arithmetic.

§21.2.3. Testing the Beam-Column Element Modules

The modules are tested by the statements collected in the scripts of Figures 21.11 and 21.13.

The script of Figure 21.7 tests a numerically defined element of length $\ell = 5$ with end nodes located at $(0, 0)$ and $(3, 4)$ respectively, with $E = 100, \rho = 84/5, A = 125$ and $I_{zz} = 250$. The output is shown in Figure 21.12. The tests consist of the following operations:

Testing $\mathbf{K}^{(e)}$. The stiffness matrix returned in `Ke` is printed. Its six eigenvalues are computed and printed. As expected three eigenvalues, which correspond to the three independent rigid body motions of the element, are zero. The remaining three eigenvalues are positive.

Testing $\mathbf{M}_C^{(e)}$. The consistent mass matrix returned in `MeC` is printed. Its symmetry is checked. The six eigenvalues are computed and printed. As expected they are all positive. A frequency test is also carried out.

Testing $\mathbf{M}_L^{(e)}$. The lumped mass matrix returned in `MeL` is printed. This is a diagonal matrix and thus its eigenvalues are the same as the diagonal entries. A frequency test is also carried out.

The script of Figure 21.13 tests a symbolically defined element with end nodes located at $(0, 0)$ and $(L, 0)$, which is aligned with the x axis. The element properties E, ρ, A and I_{zz} are kept in symbolic form. The output is shown in Figure 21.14, except that the output of the eigenvalues of the mass matrices has been deleted to save space.

```

ClearAll[L,Em,ρ,A,Izz,opt];
ncoor={{0,0},{L,0}}; mprop={Em,0,ρ,0}; fprop={A,Izz};
opt={False};
Ke= PlaneBeamColumn2Stiffness[ncoor,mprop,fprop,opt];
Print["Symbolic Elem Stiff Matrix:"]; kfac=Em*A/L;
Ke=Simplify[Ke/kfac]; Print[kfac," ",Ke//MatrixForm];
Print["Eigenvalues of Ke=",kfac,"*",Eigenvalues[Ke]];
MeC= PlaneBeamColumn2ConsMass[ncoor,mprop,fprop,opt];
Print["Symbolic Consistent Mass Matrix:"]; mfac=ρ*A*L;
MeC= Simplify[MeC/mfac]; Print[mfac," ",MeC//MatrixForm];
Print["Eigenvalues of MeC=",mfac,"*",Eigenvalues[MeC]];
Print["Squared frequencies=",Simplify[kfac/mfac],"*",
      Simplify[Eigenvalues[Inverse[MeC].Ke]]];
MeL= PlaneBeamColumn2LumpMass[ncoor,mprop,fprop,opt];
Print["Symbolic Lumped Mass Matrix:"]; mfac=ρ*A*L;
MeL= Simplify[MeL/mfac]; Print[mfac," ",MeL//MatrixForm];
Print["Eigenvalues of MeL=",Eigenvalues[MeL]];
Print["Squared frequencies=",Simplify[kfac/mfac],"*",
      Simplify[Eigenvalues[Inverse[MeL].Ke]]];

```

Figure 21.13. Test of 2-node plane beam-column element with symbolic inputs.

Symbolic Elem Stiff Matrix:

$$\frac{A E m}{L} \begin{pmatrix} 1 & 0 & 0 & -1 & 0 & 0 \\ 0 & \frac{12 I_{zz}}{A L^2} & \frac{6 I_{zz}}{A L} & 0 & -\frac{12 I_{zz}}{A L^2} & \frac{6 I_{zz}}{A L} \\ 0 & \frac{6 I_{zz}}{A L} & \frac{4 I_{zz}}{A} & 0 & -\frac{6 I_{zz}}{A L} & \frac{2 I_{zz}}{A} \\ -1 & 0 & 0 & 1 & 0 & 0 \\ 0 & -\frac{12 I_{zz}}{A L^2} & -\frac{6 I_{zz}}{A L} & 0 & \frac{12 I_{zz}}{A L^2} & -\frac{6 I_{zz}}{A L} \\ 0 & \frac{6 I_{zz}}{A L} & \frac{2 I_{zz}}{A} & 0 & -\frac{6 I_{zz}}{A L} & \frac{4 I_{zz}}{A} \end{pmatrix}$$

Eigenvalues of Ke = $\frac{A E m}{L} * \{0, 0, 0, 2, \frac{2 I_{zz}}{A}, \frac{6 (4 I_{zz} + I_{zz} L^2)}{A L^2}\}$

Symbolic Consistent Mass Matrix:

$$A L \rho \begin{pmatrix} \frac{1}{3} & 0 & 0 & \frac{1}{6} & 0 & 0 \\ 0 & \frac{13}{35} & \frac{11 L}{210} & 0 & \frac{9}{70} & -\frac{13 L}{420} \\ 0 & \frac{11 L}{210} & \frac{L^2}{105} & 0 & \frac{13 L}{420} & -\frac{L^2}{140} \\ \frac{1}{6} & 0 & 0 & \frac{1}{3} & 0 & 0 \\ 0 & \frac{9}{70} & \frac{13 L}{420} & 0 & \frac{13}{35} & -\frac{11 L}{210} \\ 0 & -\frac{13 L}{420} & -\frac{L^2}{140} & 0 & -\frac{11 L}{210} & \frac{L^2}{105} \end{pmatrix}$$

Squared frequencies = $\frac{E m}{L^2 \rho} * \{0, 0, 0, 12, \frac{720 I_{zz}}{A L^2}, \frac{8400 I_{zz}}{A L^2}\}$

Symbolic Lumped Mass Matrix:

$$A L \rho \begin{pmatrix} \frac{1}{2} & 0 & 0 & 0 & 0 & 0 \\ 0 & \frac{1}{2} & 0 & 0 & 0 & 0 \\ 0 & 0 & \frac{L^2}{78} & 0 & 0 & 0 \\ 0 & 0 & 0 & \frac{1}{2} & 0 & 0 \\ 0 & 0 & 0 & 0 & \frac{1}{2} & 0 \\ 0 & 0 & 0 & 0 & 0 & \frac{L^2}{78} \end{pmatrix}$$

Squared frequencies = $\frac{E m}{L^2 \rho} * \{0, 0, 0, 4, \frac{156 I_{zz}}{A L^2}, \frac{516 I_{zz}}{A L^2}\}$

Figure 21.14. Results from test statements of Figure 21.13. Output of eigenvalues of mass matrices deleted to save space.

The sequence of tests on the symbolic element is similar to that performed for the bar element in Figure 21.4. The vibration eigenproblems are

$$\mathbf{K}^{(e)} \mathbf{v}_i = \omega_i^2 \mathbf{M}_C^{(e)} \mathbf{v}_i, \quad \mathbf{K}^{(e)} \mathbf{v}_i = \omega_i^2 \mathbf{M}_L^{(e)} \mathbf{v}_i, \quad (21.12)$$

where ω_i are circular frequencies and \mathbf{v}_i the associated eigenvectors or vibration mode shapes. Because the three rigid body modes are solution of (21.12), three zero frequencies are expected for the consistent mass eigenproblem, which is borne out by the tests. The three positive nonzero frequencies corresponds to one free-free axial and two free-free bending modes. The consistent mass gives for the latter $\omega^2 = 720EI/(\rho AL\ell^4)$ and $\omega^2 = 8400EI/(\rho AL\ell^4)$. These are upper bounds to the exact continuum solution for the first two free-free bending frequencies, which are $502EI/(\rho AL\ell^4)$ and $1382EI/(\rho AL\ell^4)$.

The lumped mass vibration eigenproblem, using the HRZ rotational-mass lumping scheme, yields three zero frequencies: one finite positive axial vibration frequency, which is the same as that provided by the bar element, and two bending frequencies $\omega^2 = 156EI/(\rho AL\ell^4)$ and $\omega^2 = 516EI/(\rho AL\ell^4)$. These are lower bounds to the exact free-free continuum frequencies given above.

§21.3. *THE SPACE BAR ELEMENT

To provide a taste of the world of space structures, this and the next section describe the extension of the bar and beam column elements, respectively, to three dimensions.

The two-node, prismatic, three-dimensional bar element is shown in Figure 21.15. It has two nodes and six degrees of freedom. The element node displacements and conjugate forces are

$$\mathbf{u}^{(e)} = \begin{bmatrix} u_{x1} \\ u_{y1} \\ u_{z1} \\ u_{x2} \\ u_{y2} \\ u_{z2} \end{bmatrix}, \quad \mathbf{f}^{(e)} = \begin{bmatrix} f_{x1} \\ f_{y1} \\ f_{z1} \\ f_{x2} \\ f_{y2} \\ f_{z2} \end{bmatrix}. \quad (21.13)$$

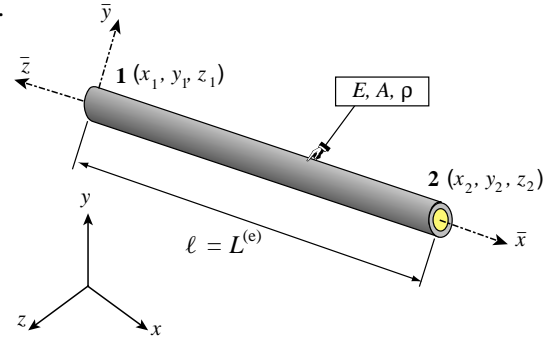


Figure 21.15. The space (3D) bar element.

The element geometry is described by the coordinates $\{x_i, y_i, z_i\}$, $i = 1, 2$ of the two end nodes. The two material properties involved in the stiffness and mass computations are the modulus of elasticity E and the mass density per unit volume ρ . The only fabrication property required is the cross section area A . All of these properties are assumed to be constant over the element.

§21.3.1. Element Formulation

For the 3D bar element, introduce the notation $x_{21} = x_2 - x_1$, $y_{21} = y_2 - y_1$, $z_{21} = z_2 - z_1$, $\ell = \sqrt{x_{21}^2 + y_{21}^2 + z_{21}^2}$. It can be shown² that the element stiffness matrix in global coordinates is given by

$$\mathbf{K}^{(e)} = \frac{E^{(e)} A^{(e)}}{\ell^3} \begin{bmatrix} x_{21}x_{21} & x_{21}y_{21} & x_{21}z_{21} & -x_{21}x_{21} & -x_{21}y_{21} & -x_{21}z_{21} \\ x_{21}y_{21} & y_{21}y_{21} & y_{21}z_{21} & -x_{21}y_{21} & -y_{21}y_{21} & -y_{21}z_{21} \\ x_{21}z_{21} & y_{21}z_{21} & z_{21}z_{21} & -x_{21}z_{21} & -y_{21}z_{21} & -z_{21}z_{21} \\ -x_{21}x_{21} & -x_{21}y_{21} & -x_{21}z_{21} & x_{21}x_{21} & x_{21}y_{21} & x_{21}z_{21} \\ -x_{21}y_{21} & -y_{21}y_{21} & -y_{21}z_{21} & x_{21}y_{21} & y_{21}y_{21} & y_{21}z_{21} \\ -x_{21}z_{21} & -y_{21}z_{21} & -z_{21}z_{21} & x_{21}z_{21} & y_{21}z_{21} & z_{21}z_{21} \end{bmatrix}. \quad (21.14)$$

² The derivation was the subject of Exercise 6.10.


```

SpaceBar2Stiffness[ncoor_,mprop_,fprop_,opt_]:=Module[
  {x1,x2,y1,y2,z1,z2,Em,Gm,ρ,α,A,numer,x21,y21,z21,L,LL,Ke},
  {{x1,y1,z1},{x2,y2,z2}}=ncoor;
  {x21,y21,z21}={x2-x1,y2-y1,z2-z1};
  {Em,Gm,ρ,α}=mprop;{A}=fprop; {numer}=opt;
  If [numer,{x21,y21,z21,Em,A}=N[{x21,y21,z21,Em,A}]];
  LL=Simplify[x21^2+y21^2+z21^2]; L=PowerExpand[Sqrt[LL]];
  Ke=(Em*A/(LL*L))*
    {{ x21*x21, x21*y21, x21*z21,-x21*x21,-x21*y21,-x21*z21},
     { y21*x21, y21*y21, y21*z21,-y21*x21,-y21*y21,-y21*z21},
     { z21*x21, z21*y21, z21*z21,-z21*x21,-z21*y21,-z21*z21},
     {-x21*x21,-x21*y21,-x21*z21, x21*x21, x21*y21, x21*z21},
     {-y21*x21,-y21*y21,-y21*z21, y21*x21, y21*y21, y21*z21},
     {-z21*x21,-z21*y21,-z21*z21, z21*x21, z21*y21, z21*z21}};
  Return[Ke];
];

SpaceBar2ConsMass[ncoor_,mprop_,fprop_,opt_]:= Module[
  {x1,x2,y1,y2,z1,z2,Em,Gm,ρ,α,A,numer,x21,y21,z21,L,LL,Ke},
  {{x1,y1,z1},{x2,y2,z2}}=ncoor;
  {x21,y21,z21}={x2-x1,y2-y1,z2-z1};
  {Em,Gm,ρ,α}=mprop;{A}=fprop; {numer}=opt;
  If [numer,{x21,y21,z21,ρ,A}=N[{x21,y21,z21,ρ,A}]];
  LL=Simplify[x21^2+y21^2+z21^2]; L=PowerExpand[Sqrt[LL]];
  MeC=(ρ*A*L/6)*{
    {2,0,0,1,0,0},{0,2,0,0,1,0},{0,0,2,0,0,1},
    {1,0,0,2,0,0},{0,1,0,0,2,0},{0,0,1,0,0,2}};
  Return[MeC];
];

SpaceBar2LumpMass[ncoor_,mprop_,fprop_,opt_]:= Module[
  {x1,x2,y1,y2,z1,z2,Em,Gm,ρ,α,A,numer,x21,y21,z21,L,LL,Ke},
  {{x1,y1,z1},{x2,y2,z2}}=ncoor;
  {x21,y21,z21}={x2-x1,y2-y1,z2-z1};
  {Em,Gm,ρ,α}=mprop; {A}=fprop; {numer}=opt;
  If [numer,{x21,y21,z21,ρ,A}=N[{x21,y21,z21,ρ,A}]];
  LL=Simplify[x21^2+y21^2+z21^2]; L=PowerExpand[Sqrt[LL]];
  MeL=(ρ*A*L/2)*IdentityMatrix[6];
  Return[MeL];
];

```

Figure 21.16. Module to form the stiffness and mass of the space (3D) bar element.

This matrix expression in terms of coordinate differences is useful in symbolic work, because it enhances simplification possibilities.

The consistent and lumped mass matrix are given by

$$\mathbf{M}_C^{(e)} = \frac{\rho A \ell}{6} \begin{bmatrix} 2 & 0 & 0 & 1 & 0 & 0 \\ 0 & 2 & 0 & 0 & 1 & 0 \\ 0 & 0 & 2 & 0 & 0 & 1 \\ 1 & 0 & 0 & 2 & 0 & 0 \\ 0 & 1 & 0 & 0 & 2 & 0 \\ 0 & 0 & 1 & 0 & 0 & 2 \end{bmatrix}, \quad \mathbf{M}_L^{(e)} = \frac{\rho A \ell}{2} \begin{bmatrix} 1 & 0 & 0 & 0 & 0 & 0 \\ 0 & 1 & 0 & 0 & 0 & 0 \\ 0 & 0 & 1 & 0 & 0 & 0 \\ 0 & 0 & 0 & 1 & 0 & 0 \\ 0 & 0 & 0 & 0 & 1 & 0 \\ 0 & 0 & 0 & 0 & 0 & 1 \end{bmatrix}. \quad (21.15)$$

Both mass matrices (21.15) are independent of the orientation of the element. For the derivation see, e.g., the book by Przemieniecki cited in footnote 1.

```

ncoor={{0,0,0},{2,3,6}}; mprop={343,0,6/10,0};
fprop={10}; opt={True};

Ke= SpaceBar2Stiffness[ncoor,mprop,fprop,opt];
Print["Numerical Elem Stiff Matrix: "];
Print[Ke//MatrixForm];
Print["Eigenvalues of Ke=",Chop[Eigenvalues[Ke]]];

MeC= SpaceBar2ConsMass[ncoor,mprop,fprop,opt];
Print["Numerical Consistent Mass Matrix: "];
Print[MeC//MatrixForm];
Print["Eigenvalues of MeC=",Eigenvalues[N[MeC]]];

MeL= SpaceBar2LumpMass[ncoor,mprop,fprop,opt];
Print["Numerical Lumped Mass Matrix: "];
Print[MeL//MatrixForm];
Print["Eigenvalues of MeL=",Eigenvalues[N[MeL]]];

```

Figure 21.17. Testing the space bar element with numerical inputs.

```

Numerical Elem Stiff Matrix:
( 40.   60.   120.  -40.  -60.  -120. )
( 60.   90.   180.  -60.  -90.  -180. )
( 120.  180.   360. -120. -180. -360. )
( -40.  -60.  -120.  40.   60.   120. )
( -60.  -90.  -180.  60.   90.   180. )
( -120. -180. -360. 120.  180.   360. )

Eigenvalues of Ke={980., 0, 0, 0, 0, 0}
Numerical Consistent Mass Matrix:
( 14.   0   0   7.   0   0 )
( 0  14.   0   0   7.   0 )
( 0   0  14.   0   0   7. )
( 7.   0   0  14.   0   0 )
( 0   7.   0   0  14.   0 )
( 0   0   7.   0   0  14. )

Eigenvalues of MeC={21., 21., 21., 7., 7., 7.}
Numerical Lumped Mass Matrix:
( 21.   0   0   0   0   0 )
( 0  21.   0   0   0   0 )
( 0   0  21.   0   0   0 )
( 0   0   0  21.   0   0 )
( 0   0   0   0  21.   0 )
( 0   0   0   0   0  21. )

```

Figure 21.18. Results from running the test script of Figure 21.17.

§21.3.2. Element Formation Modules

Figure 21.16 lists three *Mathematica* modules:

```

ClearAll[A,Em,ρ,L,opt];
ncoor={{0,0,0},{L,2*L,2*L}/3}; mprop={Em,0,ρ,0};
fprop={A}; opt={False};

Ke= SpaceBar2Stiffness[ncoor,mprop,fprop,opt];
kfac=A*Em/L; Ke=Simplify[Ke/kfac];
Print["Symbolic Elem Stiff Matrix: "];
Print[kfac," ",Ke//MatrixForm];
Print["Eigenvalues of Ke=",kfac,"*",Eigenvalues[Ke]];

MeC= SpaceBar2ConsMass[ncoor,mprop,fprop,opt];
mfac=A*L*ρ/6; MeC= Simplify[MeC/mfac];
Print["Symbolic Consistent Mass Matrix: "];
Print[mfac," ",MeC//MatrixForm];
Print["Eigenvalues of MeC=",mfac,"*",Eigenvalues[MeC]];
Print["Squared frequencies=",Simplify[kfac/mfac],"*",
      Simplify[Eigenvalues[Inverse[MeC].Ke]]];

MeL= SpaceBar2LumpMass[ncoor,mprop,fprop,opt];
mfac=A*L*ρ/2; MeL= Simplify[MeL/mfac];
Print["Symbolic Lumped Mass Matrix: "];
Print[mfac," ",MeL//MatrixForm];
Print["Eigenvalues of MeL=",mfac,"*",Eigenvalues[MeL]];
Print["Squared frequencies=",Simplify[kfac/mfac],"*",
      Simplify[Eigenvalues[Inverse[MeC].Ke]]];

```

Figure 21.19. Testing the space bar element with symbolic inputs.

SpaceBar2Stiffness returns the element stiffness matrix $\mathbf{K}^{(e)}$ of a 2-node, 3-dimensional, prismatic bar element, given by (21.14). SpaceBar2ConsMass returns the consistent mass matrix $\mathbf{M}_C^{(e)}$ of a 2-node, 3-dimensional, prismatic bar element, given by (21.15). SpaceBar2LumpMass returns the lumped mass matrix $\mathbf{M}_L^{(e)}$ of a 2-node, 3-dimensional, prismatic bar element, given by (21.15).

As in the case of the plane bar and beam, the argument sequence of the three modules is:

[ncoor, mprop, fprop, opt]

These four arguments are *lists* that collect the following data:

node coordinates, material properties, fabrication properties, options

The internal structure of these lists is

ncoor	{{ x1,y1,z1 }, { x2,y2,z2 }}	
mprop	{ Em,Gm,rho,alpha }	
fprop	{ A }	
opt	{ num }	(21.16)

Here x_1, y_1, z_1 and x_2, y_2, z_2 are the coordinates of the end nodes with respect to an $\{x, y, z\}$ global coordinate system, while E, G, ρ, α, A and I_{zz} stand for E, G, ρ, α, A and I_{zz} , respectively. For the stiffness matrix only E and A are used. In the mass matrix modules only ρ and A are used. Entries G and α are included to take care of other 1D elements.

The only option is `numer`, which is a logical flag with the value `True` or `False`. If `True` the computations are carried out in numerical floating-point arithmetic.

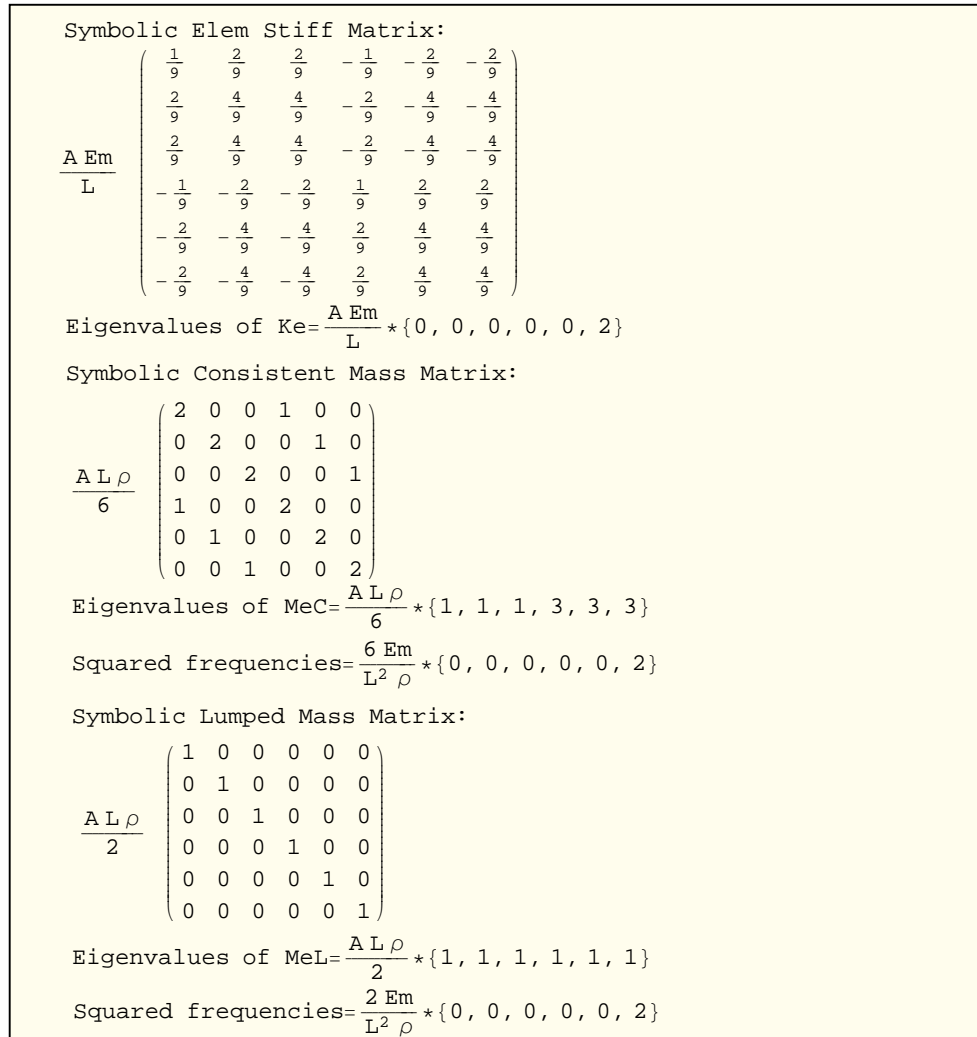


Figure 21.20. Results from running the test script of Figure 21.19.

§21.3.3. Testing the Space Bar Element Modules

The modules are tested by the statements listed in Figures 21.17 and 21.19. As these are similar to previous tests on the plane bar they need not be described in detail.

The script of Figure 21.17 tests a numerically defined element with end nodes located at (0, 0, 0) and (30, 40, 0), with $E = 1000$, $A = 5$, $\rho = 6/10$, and `numer` set to `True`. Executing the statements in Figure 21.17 produces the results listed in Figure 21.18.

The script of Figure 21.5 tests a symbolically defined element with end nodes located at (0, 0) and (L , 0, 0), which is aligned with the x axis. The element properties E , ρ and A are kept symbolic. Executing the statements in Figure 21.19 produces the results shown in Figure 21.20.

§21.4. *THE SPACE BEAM ELEMENT

A second example in 3D is the general beam element shown in Figure 21.21. The element is prismatic and has two end nodes: 1 and 2, placed at the centroid of the end cross sections.

These define the local \bar{x} axis as directed from 1 to 2. For simplicity the cross section will be assumed to be doubly symmetric, as is the case in commercial I and double-T profiles. The principal moments of inertia are defined by these symmetries. The local \bar{y} and \bar{z} axes are aligned with the symmetry lines of the cross section forming a RH system with \bar{x} . Consequently the principal moments of inertia are I_{yy} and I_{zz} , the bars being omitted for convenience.

The global coordinate system is $\{x, y, z\}$. To define the orientation of $\{\bar{y}, \bar{z}\}$ with respect to the global system, a third orientation node 3, which must not be collinear with 1–2, is introduced. See Figure 21.15. Axis \bar{y} lies in the 1–2–3 plane and \bar{z} is normal to 1–2–3 forming a RH system.

Six global degrees of freedom are defined at each node i : the three translations u_{xi}, u_{yi}, u_{zi} and the three rotations $\theta_{xi}, \theta_{yi}, \theta_{zi}$.

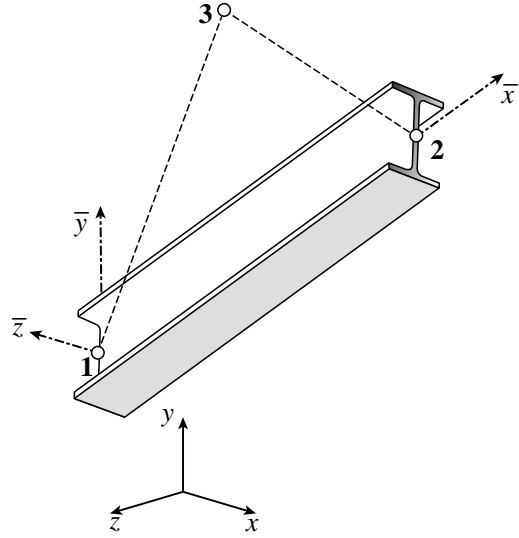


Figure 21.21. The space (3D) beam element.

The element global node displacements and conjugate forces are arranged as

$$\begin{aligned} \mathbf{u}^{(e)} &= [u_{x1} \quad u_{y1} \quad u_{z1} \quad \theta_{x1} \quad \theta_{y1} \quad \theta_{z1} \quad u_{x2} \quad u_{y2} \quad u_{z2} \quad \theta_{x2} \quad \theta_{y2} \quad \theta_{z2}]^T, \\ \mathbf{f}^{(e)} &= [f_{x1} \quad f_{y1} \quad f_{z1} \quad m_{x1} \quad m_{y1} \quad m_{z1} \quad f_{x2} \quad f_{y2} \quad f_{z2} \quad m_{x2} \quad m_{y2} \quad m_{z2}]^T. \end{aligned} \quad (21.17)$$

The beam material is characterized by the elastic modulus E and the shear modulus G (the latter appears in the torsional stiffness). Four cross section properties are needed: the cross section area A , the moment of inertia J that characterizes torsional rigidity,³ and the two principal moments of inertia I_{yy} and I_{zz} taken with respect to \bar{y} and \bar{z} , respectively. The length of the element is denoted by L . The Bernoulli-Euler model is used; thus the effect of transverse shear on the beam stiffness is neglected.

To simplify the following expressions, define the following “rigidity” combinations by symbols: $R^a = EA/L$, $R^t = GJ/L$, $R_{y3}^b = EI_{yy}/L^3$, $R_{y2}^b = EI_{yy}/L^2$, $R_y^b = EI_{yy}/L$, $R_{z3}^b = EI_{zz}/L^3$, $R_{z2}^b = EI_{zz}/L^2$, $R_z^b = EI_{zz}/L$. Note that R^a is the axial rigidity, R^t the torsional rigidity, while the R^b 's are bending rigidities scaled by the length in various ways. Then the 12×12 local stiffness matrix can be written as⁴

³ For circular and annular cross sections, J is the polar moment of inertia of the cross section with respect to the centroidal axis \bar{x} . For other cross sections J has dimensions of a moment of inertia but must be calculated according to St. Venant's theory of torsion, or approximate theories.

⁴ Cf. Pzremieniecki [21.2] page 79. The presentation in this book includes transverse shear effects as per Timoshenko's beam theory. The form (21.18) results from neglecting those effects.

$$\bar{\mathbf{K}}^{\square} = \begin{bmatrix} R^a & 0 & 0 & 0 & 0 & 0 & -R^a & 0 & 0 & 0 & 0 & 0 \\ 0 & 12R_{z3}^b & 0 & 0 & 0 & 6R_{z2}^b & 0 & -12R_{z3}^b & 0 & 0 & 0 & 6R_{z2}^b \\ 0 & 0 & 12R_{y3}^b & 0 & -6R_{y2}^b & 0 & 0 & 0 & -12R_{y3}^b & 0 & -6R_{y2}^b & 0 \\ 0 & 0 & 0 & R^t & 0 & 0 & 0 & 0 & 0 & -R^t & 0 & 0 \\ 0 & 0 & -6R_{y2}^b & 0 & 4R_y^b & 0 & 0 & 0 & 6R_{y2}^b & 0 & 2R_y^b & 0 \\ 0 & 6R_{z2}^b & 0 & 0 & 0 & 4R_z^b & 0 & -6R_{z2}^b & 0 & 0 & 0 & 2R_z^b \\ -R^a & 0 & 0 & 0 & 0 & 0 & R^a & 0 & 0 & 0 & 0 & 0 \\ 0 & -12R_{z3}^b & 0 & 0 & 0 & -6R_{z2}^b & 0 & 12R_{z3}^b & 0 & 0 & 0 & -6R_{z2}^b \\ 0 & 0 & -12R_{y3}^b & 0 & 6R_{y2}^b & 0 & 0 & 0 & 12R_{y3}^b & 0 & 6R_{y2}^b & 0 \\ 0 & 0 & 0 & -R^t & 0 & 0 & 0 & 0 & 0 & R^t & 0 & 0 \\ 0 & 0 & -6R_{y2}^b & 0 & 2R_y^b & 0 & 0 & 0 & 6R_{y2}^b & 0 & 4R_y^b & 0 \\ 0 & 6R_{z2}^b & 0 & 0 & 0 & 2R_z^b & 0 & -6R_{z2}^b & 0 & 0 & 0 & 4R_z^b \end{bmatrix} \quad (21.18)$$

The formation and testing of this element is the subject of Exercise 21.8.

Notes and Bibliography

The four elements treated here are derived in most books dealing with matrix structural analysis. Przemieniecki [21.2] has been recommended on account of being inexpensive. The implementation and testing procedures are normally not covered.

For the HRZ mass lumping scheme, see [21.1].

References

- [21.1] R. D. Cook, D. S. Malkus and M. E. Plesha, *Concepts and Applications of Finite Element Analysis*, 3rd ed., Wiley, 1989.
- [21.2] Przemieniecki, J. S., *Theory of Matrix Structural Analysis*, McGraw-Hill, New York, 1968; Dover edition 1986.

Homework Exercises for Chapter 21

Implementation of One-Dimensional Elements

EXERCISE 21.1

[C:15] Download the modules of Figure 21.2 and their testers in Figures 21.3 and 21.5, and verify the test results reported in the output Figures 21.3 and 21.3. Comment on whether the stiffness matrix \mathbf{K}_e has the correct rank of 1.

EXERCISE 21.2

[C:15] Download the modules of Figure 21.9 and their testers, and verify the results reported in the output figures.

EXERCISE 21.3

[C:15] Download the modules of Figure 21.16 and their testers, and verify the results reported in the output figures.

EXERCISE 21.4

[A+C:30] Implement the 2D spar element and test it.

EXERCISE 21.5

[C:25] Implement the plane bar, plane beam and space bar stiffness element module in a lower level programming language and check them by writing a short test driver. [Do not bother about the mass modules.] Your choices are C, Fortran 77 or Fortran 90. (C++ is overkill for this kind of software).

EXERCISE 21.6

[A:25] Explain why the eigenvalues of $\mathbf{K}^{(e)}$ of any the elements given here do not change if the $\{x, y, z\}$ global axes change.

EXERCISE 21.7

[A+C:30] (Advanced) Implement a 3-node space bar element. *Hint:* use the results of Exercise 16.5 and transform the local stiffness to global coordinates via a 3×9 transformation matrix. Test the element and verify that it has two nonzero eigenvalues.

EXERCISE 21.8

[A/C:30] Figure E21.1 shows a *Mathematica* implementation of the Bernoulli-Euler 2-node general beam element in 3D described in §21.4. biaxial bending and torsion. Explain the logic of the module.⁵

EXERCISE 21.9

[C:25] Test the space beam element listed in Figure E21.1 using the scripts given in Figures E21.2 and E21.3, and report results. Comment on whether the element has the correct rank of 6.

⁵ The expression for the local stiffness \mathbf{K}_{bar} is taken from §5.6 of Przemieniecki, *Matrix Structural Analysis*, Dover, 1968, with the shear contribution ignored.

```

SpaceBeamColumn2Stiffness[ncoor_,mprop_,fprop_,opt_]:= Module[
{
x1,x2,y1,y2,z1,z2,x21,y21,z21,xm,ym,zm,x0,y0,z0,dx,dy,dz,
Em,Gm,ρ,α,A,Iz,numer,ra,ry,ry2,ry3,rz,rz2,rz3,rx,
L,LL,LLL,yL,txx,txy,txz,tyx,tyy,tyz,tzx,tzy,tzz,T,Kebar,Ke},
{
x1,y1,z1}=ncoor[[1]]; {x2,y2,z2}=ncoor[[2]];
{x0,y0,z0}={xm,ym,zm}={x1+x2,y1+y2,z1+z2}/2;
If [Length[ncoor]==2,{x0,y0,z0}+={0,0,1}];
If [Length[ncoor]==3,{x0,y0,z0}=ncoor[[3]] ];
{x21,y21,z21}={x2-x1,y2-y1,z2-z1};
{dx,dy,dz}={x0-xm,ym-y0,zm-z0};
{Em,Gm,ρ,α}=mprop; {A,Izz,Iyy,Jxx}=fprop; {numer}=opt;
LL=Simplify[x21^2+y21^2+z21^2]; L=PowerExpand[Sqrt[LL]];
LLL=LL*L; ra=Em*A/L; rx=Gm*Jxx/L;
ry=2*Em*Iyy/L; ry2=6*Em*Iyy/LL; ry3=12*Em*Iyy/LLL;
rz=2*Em*Izz/L; rz2=6*Em*Izz/LL; rz3=12*Em*Izz/LLL;
Kebar={
{
ra, 0, 0, 0, 0, 0, -ra, 0, 0, 0, 0, 0},
{
0, rz3, 0, 0, 0, rz2, 0, -rz3, 0, 0, 0, rz2},
{
0, 0, ry3, 0, -ry2, 0, 0, 0, -ry3, 0, -ry2, 0},
{
0, 0, 0, rx, 0, 0, 0, 0, 0, -rx, 0, 0},
{
0, 0, -ry2, 0, 2*ry, 0, 0, 0, ry2, 0, ry, 0},
{
0, rz2, 0, 0, 0, 2*rz, 0, -rz2, 0, 0, 0, rz},
{
-ra, 0, 0, 0, 0, 0, ra, 0, 0, 0, 0, 0},
{
0, -rz3, 0, 0, 0, -rz2, 0, rz3, 0, 0, 0, -rz2},
{
0, 0, -ry3, 0, ry2, 0, 0, 0, ry3, 0, ry2, 0},
{
0, 0, 0, -rx, 0, 0, 0, 0, 0, rx, 0, 0},
{
0, 0, -ry2, 0, ry, 0, 0, 0, ry2, 0, 2*ry, 0},
{
0, rz2, 0, 0, 0, rz, 0, -rz2, 0, 0, 0, 2*rz}}};
{txx,txy,txz}={x21,y21,z21}/L; {dx,dy,dz}={x0-xm,ym-y0,zm-z0};
tyx=dy*z21-dz*y21; tyx=dy*z21-dz*y21; tyx=dy*z21-dz*y21;
tyy=dz*x21-dx*z21; tyx=dy*z21-dz*y21; tyx=dy*z21-dz*y21;
tyz=dx*y21-dy*x21; tyx=dy*z21-dz*y21; tyx=dy*z21-dz*y21;
yL=Sqrt[tyx^2+tyy^2+tyz^2]; yL=Simplify[PowerExpand[yL]];
{tyx,tyy,tyz}={tyx,tyy,tyz}/yL;
txx=tyx*txy-tyy*txz; txz=tyx*txy-tyy*txz; txz=tyx*txy-tyy*txz;
txz=tyx*txy-tyy*txz; txz=tyx*txy-tyy*txz; txz=tyx*txy-tyy*txz;
T={
{txx,txy,txz, 0, 0, 0, 0, 0, 0, 0, 0, 0},
{tyx,tyy,tyz, 0, 0, 0, 0, 0, 0, 0, 0, 0},
{txx,txy,txz, 0, 0, 0, 0, 0, 0, 0, 0, 0},
{0, 0, 0, txx,txy,txz, 0, 0, 0, 0, 0, 0},
{0, 0, 0, tyx,tyy,tyz, 0, 0, 0, 0, 0, 0},
{0, 0, 0, txx,txy,txz, 0, 0, 0, 0, 0, 0},
{0, 0, 0, 0, 0, 0, txx,txy,txz, 0, 0, 0},
{0, 0, 0, 0, 0, 0, tyx,tyy,tyz, 0, 0, 0},
{0, 0, 0, 0, 0, 0, txx,txy,txz, 0, 0, 0},
{0, 0, 0, 0, 0, 0, 0, 0, 0, txx,txy,txz},
{0, 0, 0, 0, 0, 0, 0, 0, 0, tyx,tyy,tyz},
{0, 0, 0, 0, 0, 0, 0, 0, 0, txx,txy,txz}}};
Ke=Transpose[T].Kebar.T; If [numer,Ke=N[Ke]];
Return[Ke]
];

```

Figure E21.1. Module to form stiffness of space (3D) beam.

```

ncoor={{0,0,0},{1,8,4}}; mprop={540,30,84/5,0};
fprop={18,36,72,27}; opt={True};

Ke= SpatialBeamColumn2Stiffness[ncoor,mprop,fprop,opt];
Print["Numerical Elem Stiff Matrix: "];
Print[SetPrecision[Ke,4]/MatrixForm];
Print["Eigenvalues of Ke=",Chop[Eigenvalues[Ke]]];

```

Figure E21.2. Script for numeric testing of the space beam module of Figure E21.1.


```

ClearAll[L,Em,Gm, $\rho$ , $\alpha$ ,A,Izz,Iyy,Jxx,opt];
ncoor={{0,0,0},{2*L,2*L,L}/3}; mprop={Em,Gm, $\rho$ , $\alpha$ };
fprop={A,Izz,Iyy,Jxx}; opt={False};

Ke= SpatialBeamColumn2Stiffness[ncoor,mprop,fprop,opt];
kfac=Em*A/L; Ke=Simplify[Ke/kfac];
Print["Numerical Elem Stiff Matrix: "];
Print[kfac," ",Ke//MatrixForm];
Print["Eigenvalues of Ke=",kfac,"*",Eigenvalues[Ke]];

```

Figure E21.3. Script for symbolic testing of the space beam module of Figure E21.1.

22

FEM Programs for Plane Trusses and Frames

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§22.1. INTRODUCTION

This Chapter presents a complete FEM program for analysis of plane trusses, programmed in *Mathematica*. The program includes some simple minded graphics, including animation. Exercises 22.2-22.4 discuss a complete FEM program for frame analysis for homework assignments.

The description is done in “bottom up” fashion. That means the basic modules are presented, then the driver program. Graphic modules are provided in posted Notebooks but not explained.

§22.2. ANALYSIS STAGES

As in all FEM programs, the analysis of a structure by the Direct Stiffness Method involves three major stages: (I) preprocessing or model definition, (II) processing, and (III) postprocessing.

The *preprocessing* portion of the plane truss analysis is done by the driver program, which directly sets the data structures.

- I.1 Model definition by direct setting of the data structures.
- I.2 Plot of the FEM mesh, including nodes and element labels.

The *processing* stage involves three steps:

- II.1 Assembly of the master stiffness matrix, with a subordinate element stiffness module.
- II.2 Modification of master stiffness matrix and node force vector for displacement boundary conditions.
- II.3 Solution of the modified equations for displacements. For the programs presented here the built in *Mathematica* function `LinearSolve` is used.

Upon executing these three processing steps, the displacements are available. The following *post-processing* steps may follow:

- III.1 Recovery of forces including reactions, done through a **Ku** matrix multiplication.
- III.2 Computation of internal (axial) forces in truss members.
- III.3 Plotting deflected shapes and member stress levels.

These steps will be demonstrated in class from a laptop computer.

§22.3. ANALYSIS SUPPORT MODULES

We begin by listing here the modules that support various analysis steps, and which are put into separate cells for testing convenience.

§22.3.1. Assembling the Master Stiffness

The function `PlaneTrussMasterStiffness`, listed in Figure 22.1, assembles the master stiffness matrix of a plane truss. It uses the element stiffness formation module `PlaneBar2Stiffness` described in the previous Chapter. The statements at the end of the cell test this module by forming and printing **K** of the example truss.

The arguments of `PlaneTrussMasterStiffness` are

- `nodcoor` Nodal coordinates arranged as a two-dimensional list:
 $\{\{x_1, y_1\}, \{x_2, y_2\}, \dots, \{x_n, y_n\}\}$,
 where n is the total number of nodes.

```

PlaneTrussMasterStiffness[nodcoor_,elenod_,
  elemat_,elefab_,eleopt_]:=Module[
  {numele=Length[elenod],numnod=Length[nodcoor],
  e,eNL,eftab,ni,nj,i,j,ncoor,mprop,fprop,opt,Ke,K},
  K=Table[0,{2*numnod},{2*numnod}];
  For [e=1, e<=numele, e++,
    eNL=elenod[[e]]; {ni,nj}=eNL;
    eftab={2*ni-1,2*ni,2*nj-1,2*nj};
    ncoor={nodcoor[[ni]],nodcoor[[nj]]};
    mprop=elemat[[e]]; fprop=elefab[[e]]; opt=eleopt;
    Ke=PlaneBar2Stiffness[ncoor,mprop,fprop,opt];
    neldof=Length[Ke];
    For [i=1, i<=neldof, i++, ii=eftab[[i]];
      For [j=i, j<=neldof, j++, jj=eftab[[j]];
        K[[jj,ii]]=K[[ii,jj]]+Ke[[i,j]]
      ];
    ];
  ]; Return[K];
];

PlaneBar2Stiffness[ncoor_,mprop_,fprop_,opt_]:= Module[
  {x1,x2,y1,y2,x21,y21,Em,Gm,rho,alpha,A,numer,L,LL,LLL,Ke},
  {{x1,y1},{x2,y2}}=ncoor; {x21,y21}={x2-x1,y2-y1};
  {Em,Gm,rho,alpha}=mprop; {A}=fprop; {numer}=opt;
  If [numer,{x21,y21,Em,A}=N[{x21,y21,Em,A}]];
  LL=x21^2+y21^2; L=PowerExpand[Sqrt[LL]]; LLL=Simplify[LL*L];
  Ke=(Em*A/LLL)*{{ x21*x21, x21*y21,-x21*x21,-x21*y21},
    { y21*x21, y21*y21,-y21*x21,-y21*y21},
    {-x21*x21,-x21*y21, x21*x21, x21*y21},
    {-y21*x21,-y21*y21, y21*x21, y21*y21}};

  Return[Ke]
];

nodcoor={{0,0},{10,0},{10,10}};
elenod= {{1,2},{2,3},{1,3}};
elemat= Table[{100,0,0,0},{3}];
elefab= {{1},{1/2},{2*Sqrt[2]}};
eleopt= {True};
K=PlaneTrussMasterStiffness[nodcoor,elenod,
  elemat,elefab,eleopt];
Print["Master Stiffness of Example Truss:"];
Print[K//MatrixForm];

```

Figure 22.1. Master stiffness assembly module, with test statements in red.

- elenod** Element end nodes arranged as a two-dimensional list:
 {{i1,j1},{i2,j2},...{ie,je}},
 where e is the total number of elements.
- elemat** Element material properties arranged as a two-dimensional list:
 {{Em1,Gm1,rho1,alpha1},...{Eme,Gme,rhoe,alphae}},
 where e is the total number of elements. Only the elastic modulus Em is used in this program. For the other properties zeros may be stored as placeholders.
- elefab** Element fabrication properties arranged as a two-dimensional list:
 {{A1},...{Ae}},
 where e is the total number of elements, and A the cross section area.

```

ModifiedMasterStiffness[pdof_,K_] := Module[
  {i,j,k,n=Length[K],np=Length[pdof],Kmod}, Kmod=K;
  For [k=1,k<=np,k++, i=pdof[[k]]];
    For [j=1,j<=n,j++, Kmod[[i,j]]=Kmod[[j,i]]=0];
    Kmod[[i,i]]=1
  ];
  Return[Kmod]
];
ModifiedNodeForces[pdof_,f_] := Module[
  {i,k,np=Length[pdof],fmod}, fmod=f;
  For [k=1,k<=np,k++, i=pdof[[k]]; fmod[[i]]=0];
  Return[fmod]
];
K=Array[Kij,{6,6}];
Print["Assembled Master Stiffness:"];Print[K//MatrixForm];
K=ModifiedMasterStiffness[{1,2,4},K];
Print["Master Stiffness Modified For Displacement B.C.:"];
Print[K//MatrixForm];
f=Array[fi,{6}];
Print["Node Force Vector:"]; Print[f];
f=ModifiedNodeForces[{1,2,4},f];
Print["Node Force Vector Modified For Displacement B.C.:"];
Print[f];

```

Figure 22.2. Modifying the master stiffness and node force vector for displacement boundary conditions, with test statements in red.

eleopt Element processing option: set to { True } to tell PlaneBar2Stiffness to carry out element stiffness computations in floating-point arithmetic. Else set to { False } to keep computations in exact arithmetic or symbolic form.

The assembler uses the freedom-pointer-table technique described in §3.4 for merging the element stiffness matrix into the master stiffness. The module returns the master stiffness matrix **K** in list K, which is stored as a full matrix.

The statements at the end of Figure 22.1 test this module by forming and printing the master stiffness matrix of the example truss. These statements are executed when the cell is initialized.

§22.3.2. Modifying the Master Stiffness Equations

Following the assembly process the master stiffness equations $\mathbf{Ku} = \mathbf{f}$ must be modified to account for displacement boundary conditions. This is done through the computer-oriented equation modification process described in §3.4.2. Modules that perform this operation are listed in Figure 22.2, along with test statements.

Module ModifiedMasterStiffness carries out this process for the master stiffness matrix **K**, whereas ModifiedNodalForces does this for the nodal force vector **f**. The logic of ModifiedNodalForces is considerably simplified by assuming that *all prescribed displacements are zero*, that is, the BCs are homogeneous. This is the case in the implementation shown here.

Module ModifiedMasterStiffness receives two arguments:

pdof A list of the prescribed degrees of freedom identified by their global number. For the example truss of Chapters 2-3 this list would have three entries: {1, 2, 4}, which

```

PlaneTrussIntForces[nodcoor_,elenod_,elemat_,elefab_,
  eleopt_,u_]:= Module[{numele=Length[elenod],
  numnod=Length[nodcoor],e,eNL,eftab,ni,nj,i,
  ncoor,mprop,fprop,opt,ue,p},
  p=Table[0,{numele}]; ue=Table[0,{4}];
  For [e=1, e<=numele, e++,
    eNL=elenod[[e]]; {ni,nj}=eNL;
    eftab={2*ni-1,2*ni,2*nj-1,2*nj};
    ncoor={nodcoor[[ni]],nodcoor[[nj]]};
    mprop=elemat[[e]]; fprop=elefab[[e]]; opt=eleopt;
    For [i=1,i<=4,i++, ii=eftab[[i]]; ue[[ii]]=u[[ii]]];
    p[[e]]=PlaneBar2IntForce[ncoor,mprop,fprop,opt,ue]
  ];
  Return[p]
];
PlaneBar2IntForce[ncoor_,mprop_,fprop_,opt_,ue_]:= Module[
  {x1,x2,y1,y2,x21,y21,Em,Gm,rho,alpha,A,numer,LL,pe},
  {{x1,y1},{x2,y2}}=ncoor; {x21,y21}={x2-x1,y2-y1};
  {Em,Gm,rho,alpha}=mprop; {A}=fprop; {numer}=opt;
  (*If [numer,{x21,y21,Em,A}]=N[{x21,y21,Em,A}];*)
  LL=x21^2+y21^2;
  pe=Em*A*(x21*(ue[[3]]-ue[[1]])+y21*(ue[[4]]-ue[[2]]))/LL;
  Return[pe]
];
nodcoor={{0,0},{10,0},{10,10}}; elenod= {{1,2},{2,3},{1,3}};
elemat= Table[{100,0,0,0},{3}]; elefab= {{1},{1/2},{2*Sqrt[2]}};
eleopt= {True}; u={0,0,0,0,0.4,-0.2};
p=PlaneTrussIntForces[nodcoor,elenod,elemat,elefab,eleopt,u];
Print["Int Forces of Example Truss:"];
Print[p];

```

Figure 22.3. Calculation of truss internal forces, with test statements in red.

are the freedom numbers of u_{x1} , u_{y1} and u_{y2} in \mathbf{u} .

- K The master stiffness matrix \mathbf{K} produced by the assembler module described in the previous subsection.

The module clears appropriate rows and columns of \mathbf{K} , places ones on the diagonal, and returns the thus modified \mathbf{K} as function value. Note the use of the *Mathematica* function `Length` to control loops: `np=Length[pdof]` sets `np` to the number of prescribed freedoms. Similarly `n=Length[K]` sets `n` to the order of the master stiffness matrix \mathbf{K} , which is used to bound the row and column clearing loop. These statements may be placed in the list that declares local variables.

Module `ModifiedNodalForces` has a very similar structure and logic and need not be described in detail. It is important to note, however, that for homogeneous BCs the two module are independent of each other and may be called in any order. On the other hand, if there were nonzero prescribed displacements the force modification must be done *before* the stiffness modification. This is because stiffness coefficients that are cleared in the latter are needed for modifying the force vector.

The test statements at the bottom of Figure 22.2 are chosen to illustrate another feature of *Mathematica*: the use of the `Array` function to generate subscripted symbolic arrays of one and two dimensions. The test output should be self explanatory and is not shown here. Both the force vector and its modified form are printed as row vectors to save space.

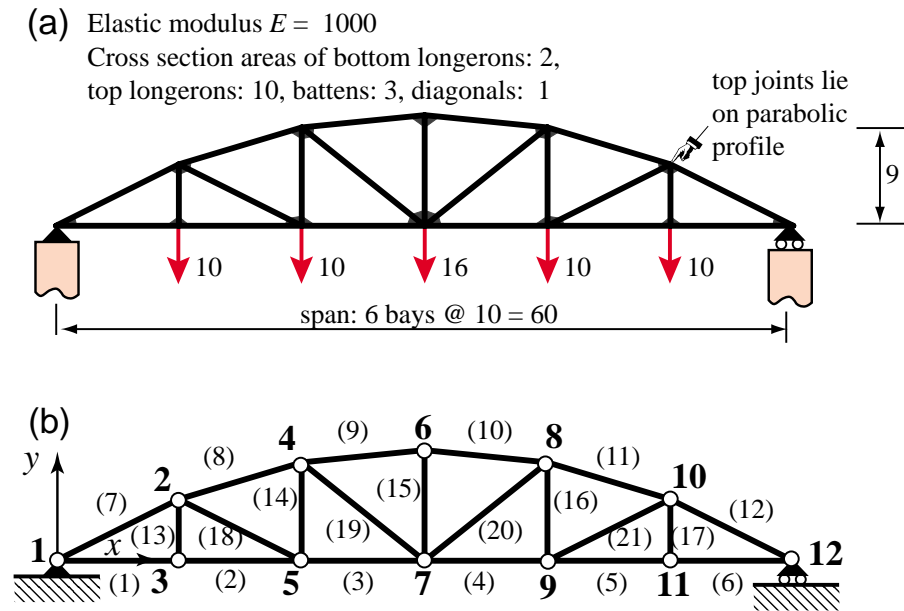


Figure 22.4. Six-bay bridge truss used as example problem: (a) truss structure showing supports and applied loads; (b) finite element idealization as pin-jointed truss.

§22.3.3. Internal Force Recovery

Module `PlaneTrussIntForces` listed in Figure 22.3 computes the internal forces (axial forces) in all truss members. The first five arguments are the same as for the assembler routine described previously. The last argument, u , contains the computed node displacements arranged as a flat, one dimensional list:

$$\{ux1, uy1 \dots uxn, uyn\} \quad (22.1)$$

`PlaneTrussIntForces` makes use of `PlaneBar2IntForce`, which computes the internal force in an *individual member*. `PlaneBar2IntForce` is similar in argument sequence and logic to `PlaneBar2Stiffness` of Figure 22.1. The first four arguments are identical. The last argument, ue , contains the list of the four element node displacements in the global system. The logic of the recovery module is straightforward and follows the method outlined in §3.2.

The statements at the bottom of Figure 22.3 test the internal force recovery for the example truss of Chapters 2-3, a and should return forces of 0, -1 and $2\sqrt{2}$ for members 1, 2 and 3, respectively.

§22.3.4. Graphic Modules

Graphic modules that support preprocessing are placed in Cells 4A, 4B and 4C of the *Mathematica* notebook. These plot unlabeled elements, elements and nodes with labels, and boundary conditions.

Graphic modules that support postprocessing are placed in Cells 5A and 5B of the Notebook. These plot deformed shapes and axial stress levels in color.

These modules are not listed since they are still undergoing modifications at the time of this writing. One unresolved problem is to find a way for absolute placement of supported nodes for correct deflected-shape animations.

§22.4. A BRIDGE TRUSS EXAMPLE

The driver program in Cell 6 defines and runs the analysis of the 6-bay bridge truss problem defined in Figure 22.4. This truss has 12 nodes and 17 elements. It is fixed at node 1 and on rollers at node 12.

```
ClearAll[];
NodeCoordinates={ {0,0},{10,5},{10,0},{20,8},{20,0},{30,9},
  {30,0},{40,8},{40,0},{50,5},{50,0},{60,0}};
ElemNodeLists= { {1,3},{3,5},{5,7},{7,9},{9,11},{11,12},
  {1,2},{2,4},{4,6},{6,8},{8,10},{10,12},
  {2,3},{4,5},{6,7},{8,9},{10,11},
  {2,5},{4,7},{7,8},{9,10}};
numnod=Length[NodeCoordinates];
numele=Length[ElemNodeLists]; numdof=2*numnod;
ElemMaterial= Table[{1000,0,0,0},{numele}];
Abot=2; Atop=10; Abat=3; Adia=1;
ElemFabrication=Join[Table[{Abot},{6}]1,Table[{Atop},{6}]1,
  Table[{Abat},{5}]1,Table[{Adia},{4}]1];
ProcessOptions= {True}; aspect=0;
PlotLineElements[NodeCoordinates,ElemNodeLists,aspect,
  "test mesh"];
PlotLineElementsAndNodes[NodeCoordinates,ElemNodeLists,aspect,
  "test mesh with elem & node labels",{True,0.12},{True,0.05}];

FreedomTag=FreedomValue=Table[{0,0},{numnod}];
FreedomValue[[3]]={0,-10}; FreedomValue[[5]]={0,-10};
FreedomValue[[7]]={0,-16};
FreedomValue[[9]]={0,-10}; FreedomValue[[11]]={0,-10};
Print["Applied node forces="]; Print[FreedomValue];
FreedomTag[[1]]={1,1}; (* fixed node 1 *)
FreedomTag[[numnod]]={0,1}; (* hroller @ node 12 *)
```

Figure 22.5. Driver for analysis of the bridge truss: preprocessing.

The driver is listed in Figures 22.5 and 22.6. It begins by defining the problem through specification of the following data structures:

NodeCoordinates Same configuration as nodcoor

ElemNodeLists Same configuration as elenod

ElemMaterials Same configuration as elemat

ElemFabrication Same configuration as elefab. This list is built up from four repeating cross sectional areas: Abot, Atop, Abat and Adia, for the areas of bottom longerons, top longerons, battens and diagonals, respectively.

ProcessOptions Same configuration as eleopt

FreedomValue. This is a two-dimensional list that specifies freedom values, node by node. It is initialized to zero on creation, then the value of nonzero applied loads is set at nodes 3, 5, 7, 9 and 11. For example $\text{FreedomValue}[[7]] = \{0, -16\}$ specifies $f_{x7} = 0$ and $f_{y7} = -16$.

FreedomTag. This is a two-dimensional list that specifies, for each nodal freedom, whether the value of the force (tag 0) or displacement (tag 1) is prescribed. It is initialized to zero on creation, then the

```

f=Flatten[FreedomValue];
K=PlaneTrussMasterStiffness[NodeCoordinates,
  ElemNodeLists,ElemMaterial,ElemFabrication,ProcessOptions];
pdof={}; For[n=1,n<=numnod,n++, For[j=1,j<=2,j++,
  If[FreedomTag[[n,j]]>0, AppendTo[pdof,2*(n-1)+j]]];
Kmod=ModifiedMasterStiffness[pdof,K];
fmod=ModifiedNodeForces [pdof,f];
u=LinearSolve[Kmod,fmod]; u=Chop[u];
Print["Computed Nodal Displacements:"]; Print[u];
f=Simplify[K.u]; f=Chop[f];
Print["External Node Forces Including Reactions:"]; Print[f];
p=PlaneTrussIntForces[NodeCoordinates,ElemNodeLists,
  ElemMaterial,ElemFabrication,eleopt,u]; p=Chop[p];
sigma=Table[p[[i]]/ElemFabrication[[i,1]],{i,1,numele}];
Print["Internal Member Forces:"]; Print[p];
PlotTrussDeformedShape[NodeCoordinates,ElemNodeLists,u,
  1.0,aspect,"Deformed shape"];
PlotAxialStressLevel[NodeCoordinates,ElemNodeLists,sigma,
  1.0,aspect,"Axial stress level"];

```

Figure 22.6. Driver for analysis of the bridge truss: processing and postprocessing.

support tags at nodes 1 (fixed) and 12 (horizontal roller) are set. For example, `FreedomTag[[1]] = {1,1}` says that both node displacement components u_{x1} and u_{y1} of node 1 are specified.

Processing commands are listed in Figure 22.6. The master stiffness matrix is assembled by the module listed in Figure 22.1. The stiffness matrix is placed in `K`. The applied force vector stored as a one-dimensional list, is placed in `f`, which results from the application of built-in function `Flatten` to `Freedom Value`.

The prescribed degree-of-freedom array `pdof` is constructed by scanning `FreedomTag` for nonzero values. For this problem `pdof={1,2,23}`. The displacement boundary conditions are applied by the modules of Figure 22.2, which return the modified master stiffness `Kmod` and the modified node force vector `fmod`. Note that the modified stiffness matrix is stored into `Kmod` rather than `K` to save the original form of the master stiffness for the recovery of reaction forces later.

The complete displacement vector is obtained by the matrix calculation

$$u = \text{LinearSolve}[Kmod, fmod] \quad (22.2)$$

which takes advantage of the built-in linear solver provided by *Mathematica*.

The remaining calculations recover the node vector including reactions by the matrix-vector multiply $f = K \cdot u$ (recall that `K` contains the unmodified master stiffness matrix). The member internal forces `p` are obtained through the module listed in Figure 22.3. The program prints `u`, `f` and `p` as row vectors to conserve space.

Running the program of Figures 22.5–6 produces the output shown in Figure 22.7. Output plot results are collected in Figure 22.8.

Applied node forces:

```
{0, 0}, {0, 0}, {0, -10}, {0, 0}, {0, -10}, {0, 0},
{0, -16}, {0, 0}, {0, -10}, {0, 0}, {0, -10}, {0, 0}}
```

Computed Nodal Displacements:

```
{0, 0, 0.809536, -1.7756, 0.28, -1.79226, 0.899001,
-2.29193, 0.56, -2.3166, 0.8475, -2.38594,
0.8475, -2.42194, 0.795999, -2.29193, 1.135, -2.3166,
0.885464, -1.7756, 1.415, -1.79226, 1.695, 0}
```

External Node Forces Including Reactions:

```
{0, 28., 0, 0, 0, -10., 0, 0, 0, -10., 0, 0, 0,
-16., 0, 0, 0, -10., 0, 0, 0, -10., 0, 28.}
```

Internal Member Forces:

```
{56., 56., 57.5, 57.5, 56., 56., -62.6099,
-60.0318, -60.2993, -60.2993,
-60.0318, -62.6099, 10., 9.25, 12., 9.25,
10., 1.67705, 3.20156, 3.20156, 1.67705}
```

Figure 22.7. Printed output from the bridge truss analysis.

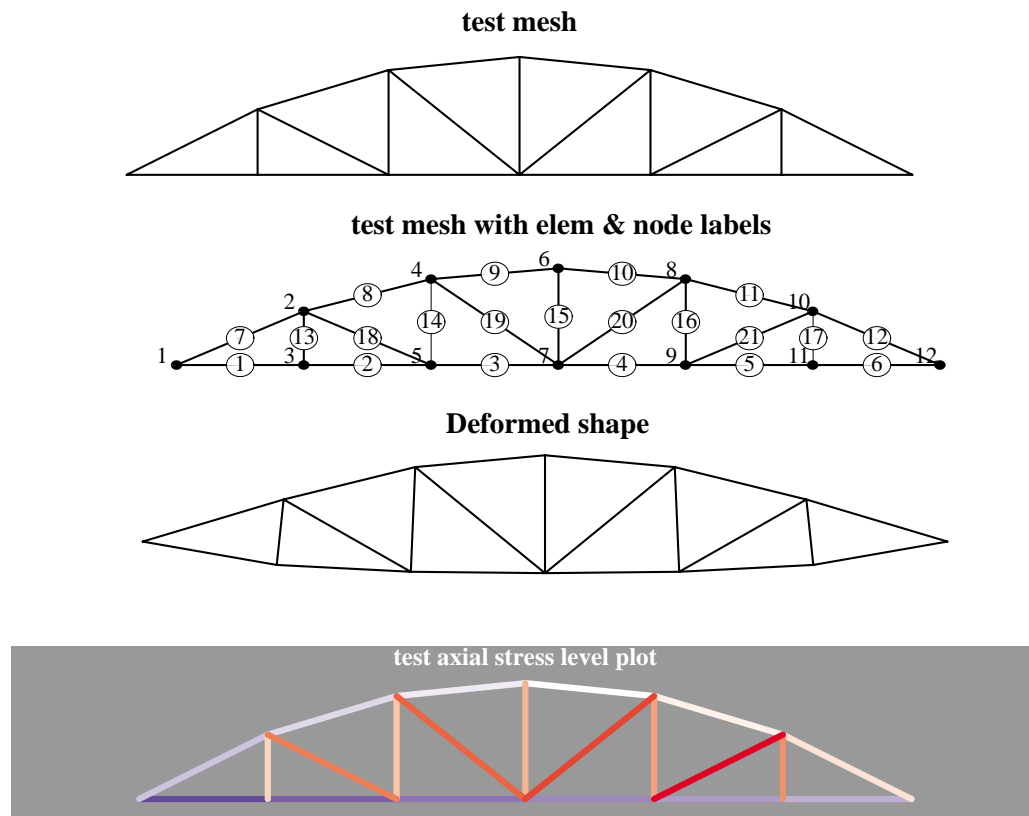


Figure 22.8. Graphics output from the bridge truss analysis.

Homework Exercises for Chapter 22

FEM Programs for Plane Trusses and Frames

EXERCISE 22.1

Placeholder.

EXERCISE 22.2

[C:25] Using the `PlaneTruss.nb` Notebook posted on the web site as guide, complete the `PlaneFrame.nb` Notebook that implements the analysis of an arbitrary plane frame using the plane beam stiffness presented in Chapter 21. To begin this homework, download the `PlaneFrame.nb` Notebook file from the web site.

The modification involves selected Cells of the Notebook, as described below. For additional details refer to the Notebook; some of the modifications involve only partially filled Modules.

Cell 1: Assembler. The element stiffness routine is the beam-column stiffness presented in Chapter 21. This module, called `PlaneBeamColumn2Stiffness`, is included in Cell 1 of the `PlaneFrame.nb` notebook linked from the Chapter 22 index.¹ In modifying the assembler module, remember that there are three degrees of freedom per node: u_{xi} , u_{yi} and θ_i , not just two.

The incomplete assembler is shown in Figure E22.1. The test statements are shown in red after the module. The lower cell shows the test output produced by a correctly completed module.

Cell 2: BC Applicator. No modifications are necessary. The two modules should work correctly for this problem since they don't assume anything about freedoms per nodes. The same test statements can be kept.

Cell 3: Internal Force Recovery. Both modules require modifications because the beam has 3 internal forces: (i) the axial force (which is recovered exactly as for bars), (ii) the bending moment $m_i = EI\kappa_i$ at end node i , and (iii) the bending moment $m_j = EI\kappa_j$ at end node j .² Furthermore the element displacement vector has six degrees of freedom, not just four. Test the modules on a simple beam problem.

The incomplete internal force module is shown in Figure E22.2. The test statements are shown in red after the module. The lower cell shows the test output produced by a correctly completed module.

Cell 4-5: Graphic Modules. These are now provided ready to use, and should not be touched.

Cell 6: Driver Program: use this for Exercise 22.3. Do not touch for this one.

As solution to this Exercise return a listing of the completed Cells 1 and 3, along with the output of the test statements for those cells.

EXERCISE 22.3

[C:25] Use the program developed in the previous Exercise to analyze the plane frame shown in Figure E22.3. The frame is fixed³ at A , B and C . It is loaded by a downward point load P at G and by a horizontal point

¹ If you want to extract an individual cell from a Notebook to work on a separate file (a good idea for working on groups) select the cell, then pick `SaveSelectionAs -> Plain Text` from the Edit menu, give it a file name in the pop-up dialogue box, and save it.

² Two end moments are required because the beam element used here has linearly varying curvatures and hence moments. To recover the end moments refer to the pertinent equations in Chapter 13. The steps are as follows. For element e recover the transverse displacements \bar{u}_{yi} and \bar{u}_{yj} in the local element system $\bar{x}^{(e)}$, $\bar{y}^{(e)}$. Complete these with rotations θ_i and θ_j (which do not need to be transformed) to form the 4×1 beam local node displacement vector. Then apply the curvature-displacement relation (13.13) at $\xi = -1$ and $\xi = 1$ to get the two end curvatures $\kappa_i^{(e)}$ and $\kappa_j^{(e)}$, respectively. Finally multiply those curvatures by $E^{(e)}I_{zz}^{(e)}$ to get the bending moments.

³ For a plane frame, a fixed condition, also known as clamped condition, means that the x , y displacements and the rotation about z are zero.

```
(* PlaneBeamColumn2Stiffness module goes here *)
PlaneFrameMasterStiffness[nodcoor_,elenod_,
  elemat_,elefab_,eleopt_]:=Module[
  {numele=Length[elenod],numnod=Length[nodcoor],
  e,eNL,eftab,ni,nj,i,j,ncoor,mprop,fprop,opt,Ke,K},
  K=Table[0,{3*numnod},{3*numnod}];
  For [e=1, e<=numele, e++,
    (* missing statements *)
    Ke=PlaneBeamColumn2Stiffness[ncoor,mprop,fprop,opt];
    (* missing statements *)
  ];
  Return[K]
];
nodcoor={{0,0},{10,0},{10,10}};
elenod= {{1,2},{2,3},{1,3}}; elemat= Table[{100,0,0,0},{3}];
elefab= {{1,10},{1/2,10},{2*Sqrt[2],10}}; eleopt= {True};
K=PlaneFrameMasterStiffness[nodcoor,elenod,elemat,elefab,eleopt];
Print["Master Stiffness of Example Frame:"];
Print[K//MatrixForm]; Print[Chop[Eigenvalues[K]]];
```

Master Stiffness of Example Frame:

22.1213	7.87868	-21.2132	-10.	0.	0.	-12.1213	-7.87868	-21.2132
7.87868	24.1213	81.2132	0.	-12.	60.	-7.87868	-12.1213	21.2132
-21.2132	81.2132	682.843	0.	-60.	200.	21.2132	-21.2132	141.421
-10.	0.	0.	22.	0.	-60.	-12.	0.	-60.
0.	-12.	-60.	0.	17.	-60.	0.	-5.	0.
0.	60.	200.	-60.	-60.	800.	60.	0.	200.
-12.1213	-7.87868	21.2132	-12.	0.	60.	24.1213	7.87868	81.2132
-7.87868	-12.1213	-21.2132	0.	-5.	0.	7.87868	17.1213	-21.2132
-21.2132	21.2132	141.421	-60.	0.	200.	81.2132	-21.2132	682.843

{1121.7, 555.338, 531.652, 46.6129, 22.4971, 14.366, 0, 0, 0}

Figure E22.1. The incomplete assembler module for Exercise 22.2, with test statements in red. Module PlaneBeamColumn2Stiffness, which goes in the upper portion of the cell is omitted to save space; that module was listed in Figure 21.9. Output cell results are produced by a correct module.

load $\frac{1}{2}P$ at D . All members have the same cross section $a \times a$ for simplicity, and the material is the same.

The SI physical units to be used are: mm for lengths, N for forces, and MPa=N/mm² for elastic moduli. For the calculations use the following numerical data: $L = 10,000$ mm (10 m), $H = 6,000$ (6 m), $a = 500$ mm (0.5 m), $P = 4,800$ N, $E = 35,000$ MPa (high strength concrete). The member cross section area is $A = a^2$, and the flexural moment of inertia about the neutral axis is $I_{zz} = a^4/12$.

The recommended finite element discretization of two elements per member is shown in Figure E22.4.

As solution to this exercise list the driver program you used and the displacement and internal force outputs. The results of primary interest are:

1. The horizontal displacement at D , and the vertical displacement at G (mm).
2. The axial forces (N) in columns AD , BE and CF , with appropriate sign⁴.
3. The maximum bending moment (N.mm) over the floor members DE and EF , and the maximum bending

⁴ Plus for tension, minus for compression

```

PlaneFrameIntForces[nodcoor_,elenod_,elemat_,elefab_,
  eleopt_,u_]:= Module[{numele=Length[elenod],
    numnod=Length[nodcoor],e,eNL,eftab,ni,nj,i,
    ncoor,mprop,fprop,opt,ue,p},
  p=Table[0,{numele}]; ue=Table[0,{6}];
  For [e=1, e<=numele, e++,
    eNL=elenod[[e]]; {ni,nj}=eNL;
    ncoor={nodcoor[[ni]],nodcoor[[nj]]};
    mprop=elemat[[e]]; fprop=elefab[[e]]; opt=eleopt;
    eftab={3*ni-2,3*ni-1,3*ni,3*nj-2,3*nj-1,3*nj};
    For [i=1,i<=6,i++, ii=eftab[[i]]; ue[[i]]=u[[ii]]];
    p[[e]]=PlaneBeamColumn2IntForces[ncoor,mprop,fprop,opt,ue]
  ];
  Return[p]
];

PlaneBeamColumn2IntForces[ncoor_,mprop_,fprop_,opt_,ue_]:=
Module[{x1,x2,y1,y2,x21,y21,Em,Gm,rho,alpha,A,Izz,num,LL,L,
  dvy,cv1,cv2,pe=Table[0,{3}]],
  {{x1,y1},{x2,y2}}=ncoor; {x21,y21}={x2-x1,y2-y1};
  {Em,Gm,rho,alpha}=mprop; {A,Izz}=fprop; {num}=opt;
  If [num,{x21,y21,Em,A,Izz}=N[{x21,y21,Em,A,Izz}]];
  (* missing statements for Exercise 22.2 *)
  Return[pe]
];

ClearAll[L,Em,A1,A2,Izz1,Izz2,Izz3,uz1,uz2,uz3];
nodcoor={{0,0},{L,0},{L,L}}; elenod= {{1,2},{2,3},{1,3}};
elemat= Table[{Em,0,0,0},{3}];
elefab= {{A1,Izz1},{A2,Izz2},{A3,Izz3}}; eleopt= {False};
u={0,uy1,0, 0,uy2,0, 0,uy3,0};
p=PlaneFrameIntForces[nodcoor,elenod,elemat,elefab,eleopt,u];
Print["Int Forces of Example Frame:"]; Print[p];

```

Int Forces of Example Frame:

$$\left\{ \left\{ 0, \frac{6 \text{ Em Izz1 } (-uy1 + uy2)}{L^2}, -\frac{6 \text{ Em Izz1 } (-uy1 + uy2)}{L^2} \right\}, \left\{ \frac{A2 \text{ Em } (-uy2 + uy3)}{L}, 0, 0 \right\}, \right. \\ \left. \left\{ \frac{A3 \text{ Em } (-uy1 + uy3)}{2 L}, \frac{3 \text{ Em Izz3 } (-uy1 + uy3)}{\sqrt{2} L^2}, -\frac{3 \text{ Em Izz3 } (-uy1 + uy3)}{\sqrt{2} L^2} \right\} \right\}$$

Figure E22.2. Incomplete element internal force recovery module for Exercise 22.2. Test statements in red. Results in output cell are those produced by a correct module.

moment in the columns *AD*, *BE* and *CF*.⁵

Provide deformed shape plot (but not the animation) and the frame stress level plots as part of the homework results. Note: the Notebook on the web contains some of the actual plots produced by the complete Notebook, to be used as targets.

⁵ The bending moment varies linearly over each element. It should be continuous at all joints except *E*.

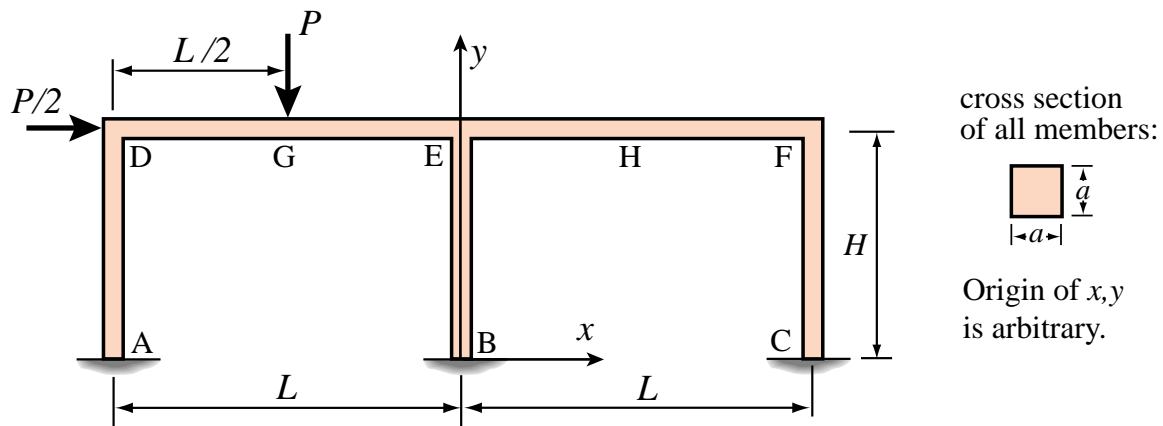


Figure E22.3. Plane frame structure for Exercise 22.3.

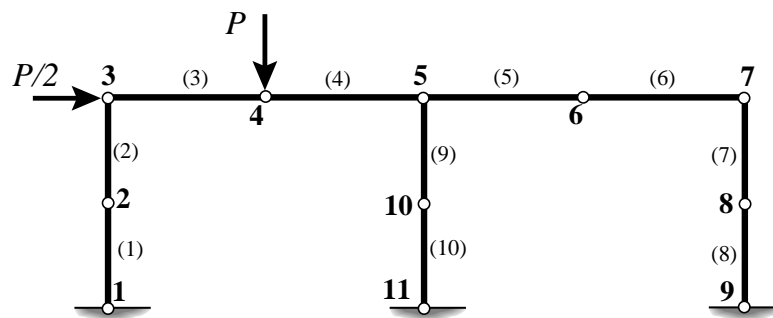


Figure E22.4. Recommended FEM discretization for the plane frame of the previous figure.

Partial answers for this exercise:

Largest vertical displacement: -0.232 mm (\downarrow) at node 4

Largest negative bending moment: $-6.8 \times 10^6 \text{ N}\cdot\text{mm}$, at node 5 of element (4)

Axial forces: -2651 N over (3) and (4)

EXERCISE 22.4

[D:20] Explain why the solution given by the FEM model of Figure E22.3 is exact for the Bernoulli-Euler bending model; that is, cannot be improved by subdividing each element into more elements.

23

Implementation of Iso-P Quadrilateral Elements

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§23.1. INTRODUCTION

This Chapter illustrates, through a specific example, the computer implementation of isoparametric *quadrilateral* elements for the plane stress problem. Triangles, which present some programming quirks, are covered in the next Chapter.

The programming example is that of the four-node bilinear quadrilateral. It covers the computation of the element stiffness matrix and consistent node force vector for a body force field. The organization of the computations is typical of isoparametric-element modules in any number of space dimensions.

§23.2. IMPLEMENTATION OF THE BILINEAR QUADRILATERAL

We consider the implementation of the 4-node bilinear quadrilateral for plane stress, depicted in Figure 23.1.

The element stiffness matrix of simple one-dimensional elements, such as the ones discussed in Chapters 21 and 22, can be easily packaged in a simple module. For 2D and 3D elements, however, it is convenient to break up the implementation into *application dependent* and *application independent* modules, as sketched in Figure 23.2. The application independent modules can be “reused” in other FEM applications, for example to form thermal, fluid or electromagnetic elements.

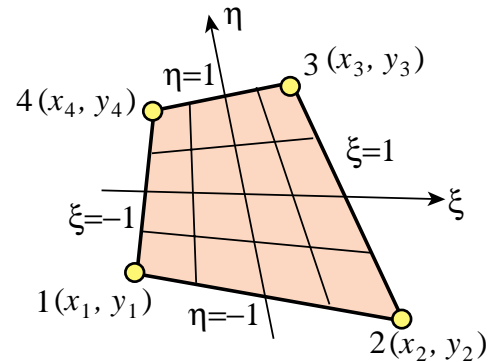


Figure 23.1. The 4-node bilinear quadrilateral element.

For the 4-node quadrilateral studied here, the subdivision of Figure 23.2 is done through the following modules:

```
Stiffness4NodePlaneStressQuad - forms Ke of 4-node bilinear quad in plane stress
QuadGaussRuleInfo - returns Gauss quadrature product rules of order 1-4
IsoQuad4ShapeFunctions - evaluates shape functions and their x/y derivatives
```

These modules are described in further detail in the following subsections, in a “bottom up” fashion.

§23.2.1. Gauss Quadrature Rule Information

Recall from §17.3 that Gauss quadrature rules for isoparametric quadrilateral elements have the canonical form

$$\int_{-1}^1 \int_{-1}^1 \mathbf{F}(\xi, \eta) d\xi d\eta = \int_{-1}^1 d\eta \int_{-1}^1 \mathbf{F}(\xi, \eta) d\xi \doteq \sum_{i=1}^{p_1} \sum_{j=1}^{p_2} w_i w_j \mathbf{F}(\xi_i, \eta_j). \quad (23.1)$$

Here $\mathbf{F} = h\mathbf{B}^T \mathbf{E} \mathbf{B} J$ is the matrix to be integrated, and p_1 and p_2 are the number of Gauss points in the ξ and η directions, respectively. Often, but not always, the same number $p = p_1 = p_2$ is chosen in both directions. A formula with $p_1 = p_2$ is called an *isotropic integration rule* because directions ξ and η are treated alike.

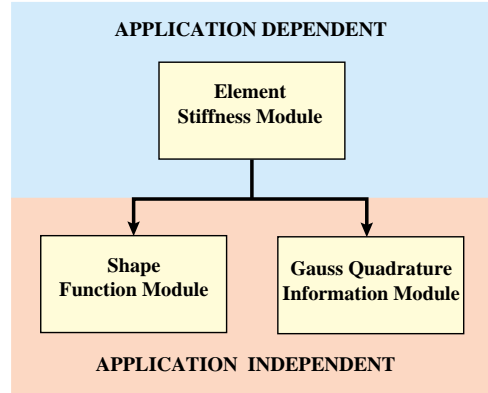


Figure 23.2. Organization of element stiffness modules.

```

QuadGaussRuleInfo[{rule_,number_},point_]:= Module[
{ξ,η,p1,p2,i1,i2,w1,w2,k,info=NULL},
If [Length[rule]==2, {p1,p2}=rule,p1=p2=rule];
If [Length[point]==2,{i1,i2}=point,
k=point; i2=Floor[(k-1)/p1]+1; i1=k-p1*(i2-1) ];
{ξ,w1}= LineGaussRuleInfo[{p1,number},i1];
{η,w2}= LineGaussRuleInfo[{p2,number},i2];
info={{ξ,η},w1*w2};
If [number,Return[N[info]],Return[Simplify[info]]];
];
LineGaussRuleInfo[{rule_,number_},point_]:= Module[
{g2={-1,1}/Sqrt[3],w3={5/9,8/9,5/9},
g3={-Sqrt[3/5],0,Sqrt[3/5]},
w4={(1/2)-Sqrt[5/6]/6,(1/2)+Sqrt[5/6]/6,
(1/2)+Sqrt[5/6]/6,(1/2)-Sqrt[5/6]/6},
g4={-Sqrt[(3+2*Sqrt[6/5])/7],-Sqrt[(3-2*Sqrt[6/5])/7],
Sqrt[(3-2*Sqrt[6/5])/7],Sqrt[(3+2*Sqrt[6/5])/7]},
i=point,info=NULL},
If [rule==1,info={0,2}];
If [rule==2,info={g2[[i]],1}];
If [rule==3,info={g3[[i]],w3[[i]]}];
If [rule==4,info={g4[[i]],w4[[i]]}];
If [number,Return[N[info]],Return[Simplify[info]]];
];

```

Figure 23.3. Module to get Gauss-product quadrature information for a quadrilateral.

QuadGaussRuleInfo is an application independent module QuadGaussRuleInfo that implements the two-dimensional product Gauss rules with 1 through 4 points in each direction. The number of points in each direction may be the same or different. Use of this module was described in detail in §17.3.4. For the readers convenience it is listed, along with its subordinate module LineGaussRuleInfo, in Figure 23.3.

§23.2.2. Shape Function Evaluation

Quad4IsoPShapeFunDer is an application independent module that computes the shape functions $N_i^{(e)}$, $i = 1, 2, 3, 4$ and its x - y partial derivatives at the sample integration points. The logic, listed in Figure 23.4, is straightforward and follows closely the description of Chapter 17.

```

Quad4IsoPShapeFunDer[ncoor_,qcoor_]:= Module[
  {Nf,dNx,dNy,dNξ,dNη,i,J11,J12,J21,J22,Jdet,ξ,η,x,y},
  {ξ,η}=qcoor;
  Nf={ (1-ξ)*(1-η), (1+ξ)*(1-η), (1+ξ)*(1+η), (1-ξ)*(1+η) }/4;
  dNξ = { -(1-η), (1-η), (1+η), -(1+η) }/4;
  dNη= { -(1-ξ), -(1+ξ), (1+ξ), (1-ξ) }/4;
  x=Table[ncoor[[i,1]],{i,4}]; y=Table[ncoor[[i,2]],{i,4}];
  J11=dNξ.x; J21=dNξ.y; J12=dNη.x; J22=dNη.y;
  Jdet=Simplify[J11*J22-J12*J21];
  dNx= ( J22*dNξ-J21*dNη)/Jdet; dNx=Simplify[dNx];
  dNy= (-J12*dNξ+J11*dNη)/Jdet; dNy=Simplify[dNy];
  Return[{Nf,dNx,dNy,Jdet}]
];

```

Figure 23.4. Shape function module for 4-node bilinear quadrilateral.

The arguments of the module are the $\{x, y\}$ quadrilateral corner coordinates, which are passed in `ncoor`, and the two quadrilateral coordinates $\{\xi, \eta\}$, which are passed in `qcoor`. The former have the same configuration as described for the element stiffness module below.

The quadrilateral coordinates define the element location at which the shape functions and their derivatives are to be evaluated. For the stiffness formation these are Gauss points, but for strain and stress computations these may be other points, such as corner nodes.

`Quad4IsoPShapeFunDer` returns the two-level list $\{Nf, Nx, Ny, Jdet\}$, in which the first three are 4-entry lists. List `Nf` collects the shape function values, `Nx` the shape function x -derivatives, `Ny` the shape function y -derivatives, and `Jdet` is the Jacobian determinant called J in Chapters 17.

§23.2.3. Element Stiffness

Module `Quad4IsoPMembraneStiffness` computes the stiffness matrix of a four-noded isoparametric quadrilateral element in plane stress. The module configuration is typical of isoparametric elements in any number of dimensions. It follows closely the procedure outlined in Chapter 17. The module logic is listed in Figure 23.5. The statements at the bottom of the module box (not shown in Figure 23.5) test it for a specific configuration.

The arguments of the module are:

- `ncoor` Quadrilateral node coordinates arranged in two-dimensional list form:
 $\{\{x_1, y_1\}, \{x_2, y_2\}, \{x_3, y_3\}, \{x_4, y_4\}\}$.
- `mprop` Material properties supplied as the list $\{Emat, \rho, \alpha\}$. `Emat` is a two-dimensional list storing the 3×3 plane stress matrix of elastic moduli:

$$\mathbf{E} = \begin{bmatrix} E_{11} & E_{12} & E_{13} \\ E_{12} & E_{22} & E_{23} \\ E_{13} & E_{23} & E_{33} \end{bmatrix} \quad (23.2)$$

If the material is isotropic with elastic modulus E and Poisson's ratio ν , this matrix becomes

$$\mathbf{E} = \frac{E}{1-\nu^2} \begin{bmatrix} 1 & \nu & 0 \\ \nu & 1 & 0 \\ 0 & 0 & \frac{1}{2}(1-\nu) \end{bmatrix} \quad (23.3)$$

```

Quad4IsoPMembraneStiffness[ncoor_,mprop_,fprop_,options_]:=
Module[{i,k,p=2,numer=False,Emat,th=1,h,qcoor,c,w,Nf,
  dNx,dNy,Jdet,B,Ke=Table[0,{8},{8}]}, Emat=mprop[[1]];
If [Length[options]==2, {numer,p}=options, {numer}=options];
If [Length[fprop]>0, th=fprop[[1]]];
If [p<1||p>4, Print["p out of range"];Return[Null]];
For [k=1, k<=p*p, k++,
  {qcoor,w}= QuadGaussRuleInfo[{p,numer},k];
  {Nf,dNx,dNy,Jdet}=Quad4IsoPShapeFunDer[ncoor,qcoor];
  If [Length[th]==0, h=th, h=th.Nf]; c=w*Jdet*h;
  B={ Flatten[Table[{dNx[[i]], 0},{i,4}]],
    Flatten[Table[{0, dNy[[i]]},{i,4}]],
    Flatten[Table[{dNy[[i]],dNx[[i]]},{i,4}]]};
  Ke+=Simplify[c*Transpose[B].(Emat.B)];
]; Return[Ke]
];

```

Figure 23.5. Element stiffness formation module for 4-node bilinear quadrilateral.

The other two items in `mprop` are not used in this module so zeros may be inserted as placeholders.

fprop Fabrication properties. The plate thickness specified as a four-entry list: { h_1, h_2, h_3, h_4 }, a one-entry list: { h }, or an empty list: { }.

The first form is used to specify an element of variable thickness, in which case the entries are the four corner thicknesses and h is interpolated bilinearly. The second form specifies uniform thickness h . If an empty list appears the module assumes a uniform unit thickness.

options Processing options. This list may contain two items: { `numer,p` } or one: { `numer` }.

`numer` is a logical flag with value True or False. If True, the computations are forced to proceed in floating point arithmetic. For symbolic or exact arithmetic work set `numer` to False.

`p` specifies the Gauss product rule to have p points in each direction. p may be 1 through 4. For rank sufficiency, p must be 2 or higher. If p is 1 the element will be rank deficient by two. If omitted $p = 2$ is assumed.

The module returns K_e as an 8×8 symmetric matrix pertaining to the following arrangement of nodal displacements:

$$\mathbf{u}^{(e)} = [u_{x1} \quad u_{y1} \quad u_{x2} \quad u_{y2} \quad u_{x3} \quad u_{y3} \quad u_{x4} \quad u_{y4}]^T. \quad (23.4)$$

§23.3. TEST OF BILINEAR QUADRILATERAL

The stiffness module is tested on the two quadrilateral geometries shown in Figure 23.6. Both elements have unit thickness and isotropic material. The left one is a rectangle of base $2a$ and height a . The right one is a right trapezoid with base $2a$, top width a and height a . Both geometries will be used to illustrate the effect of the numerical integration rule.

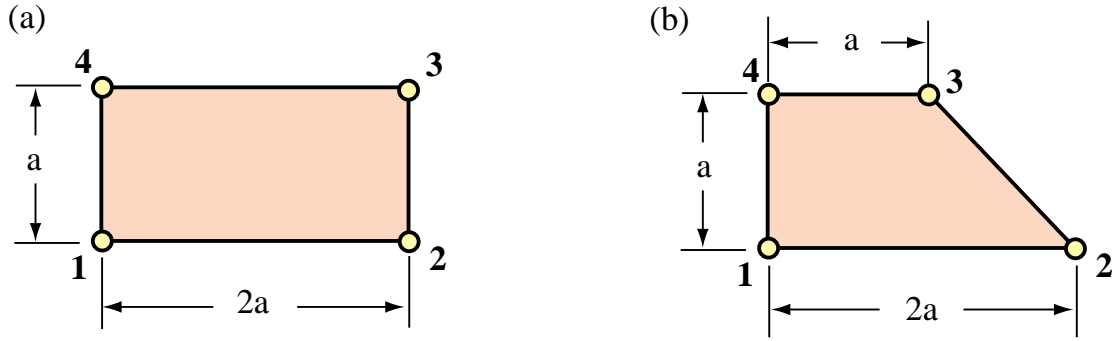


Figure 23.6. Test quadrilateral element geometries.

§23.3.1. Test of Rectangular Geometry

The test statements of Figure 23.7 compute and print the stiffness of the rectangular element shown in Figure 23.6(a). This is a rectangle of base $2a$ and height a . The element has unit thickness and isotropic material with $E = 96$ and $\nu = 1/3$, giving the stress-strain constitutive matrix

$$\mathbf{E} = \begin{bmatrix} 108 & 36 & 0 \\ 36 & 108 & 0 \\ 0 & 0 & 36 \end{bmatrix} \quad (23.5)$$

Using a 2×2 Gauss integration rule returns the stiffness matrix

$$\mathbf{K}^{(e)} = \begin{bmatrix} 42 & 18 & -6 & 0 & -21 & -18 & -15 & 0 \\ 18 & 78 & 0 & 30 & -18 & -39 & 0 & -69 \\ -6 & 0 & 42 & -18 & -15 & 0 & -21 & 18 \\ 0 & 30 & -18 & 78 & 0 & -69 & 18 & -39 \\ -21 & -18 & -15 & 0 & 42 & 18 & -6 & 0 \\ -18 & -39 & 0 & -69 & 18 & 78 & 0 & 30 \\ -15 & 0 & -21 & 18 & -6 & 0 & 42 & -18 \\ 0 & -69 & 18 & -39 & 0 & 30 & -18 & 78 \end{bmatrix} \quad (23.6)$$

Note that the rectangle dimension a does not appear in (23.6). This is a general property: *the stiffness matrix of plane stress elements is independent of inplane dimension scalings*. This follows from the fact that entries of the strain-displacement matrix \mathbf{B} have dimensions $1/L$, where L denotes a characteristic inplane length. Consequently entries of $\mathbf{B}^T \mathbf{B}$ have dimension $1/L^2$. Integration over the element area cancels out L^2 .

Using a higher order Gauss integration rule, such as 3×3 and 4×4 , reproduces exactly (23.6). This is a property characteristic of the rectangular geometry, since in that case the entries of \mathbf{B} vary linearly in ξ and η , and J is constant. Therefore the integrand $h \mathbf{B}^T \mathbf{E} \mathbf{B} J$ is at most quadratic in ξ and η , and 2 Gauss points in each direction suffice to compute the integral exactly. Using a 1×1 rule yields a rank-deficiency matrix, a result illustrated in detail in §23.2.2.

The stiffness matrix (23.6) has the eigenvalues

$$[223.64 \quad 90 \quad 78 \quad 46.3603 \quad 42 \quad 0 \quad 0 \quad 0] \quad (23.7)$$

which verifies that $\mathbf{K}^{(e)}$ has the correct rank of five (8 freedoms minus 3 rigid body modes).

```

ClearAll[Em,nu,a,b,e,h,p,num]; h=1;
Em=96; nu=1/3; (* isotropic material *)
Emat=Em/(1-nu^2)*{{1,nu,0},{nu,1,0},{0,0,(1-nu)/2}};
Print["Emat=",Emat//MatrixForm];
ncoor={{0,0},{2*a,0},{2*a,a},{0,a}}; (* 2:1 rectangular geometry *)
p=2; (* 2 x 2 Gauss rule *) num=False; (* exact symbolic/arithmetic *)
Ke=Quad4IsoPMembraneStiffness[ncoor,{Emat,0,0},{h},{num,p}];
Ke=Simplify[Chop[Ke]]; Print["Ke=",Ke//MatrixForm];
Print["Eigenvalues of Ke=",Chop[Eigenvalues[N[Ke]],.0000001]];

```

Figure 23.7. Driver for stiffness calculation of rectangular element of Figure 23.6(a).

```

ClearAll[Em,nu,h,a,p]; h=1;
Em=48*63*13*107; nu=1/3;
Emat=Em/(1-nu^2)*{{1,nu,0},{nu,1,0},{0,0,(1-nu)/2}};
ncoor={{0,0},{2*a,0},{a,a},{0,a}};
For [p=1,p<=4,p++,
  Ke=Quad4IsoPMembraneStiffness[ncoor,{Emat,0,0},{h},{True,p}];
  Ke=Rationalize[Ke,0.0000001]; Print["Ke=",Ke//MatrixForm];
  Print["Eigenvalues of Ke=",Chop[Eigenvalues[N[Ke]],.0000001]]];

```

Figure 23.8. Driver for stiffness calculation of trapezoidal element of Figure 23.6(b) for four Gauss integration rules.

§23.3.2. Test of Trapezoidal Geometry

The trapezoidal element geometry of Figure 23.6(b) is used to illustrate the effect of changing the $p \times p$ Gauss integration rule. Unlike the rectangular case, the element stiffness keeps changing as p is varied from 1 to 4. The element is rank sufficient, however, for $p \geq 2$ in agreement with the analysis of Chapter 19.

The computations are driven with the script shown in Figure 23.8. The value of p is changed in a loop. The flag number is set to True to use floating-point computation for speed (see Remark 23.1). The computed entries of $\mathbf{K}^{(e)}$ are transformed to the nearest rational number (exact integers in this case) using the built-in function Rationalize. The strange value of $E = 48 \times 63 \times 13 \times 107 = 4206384$, in conjunction with $\nu = 1/3$, makes all entries of $\mathbf{K}^{(e)}$ exact integers when computed with the first 4 Gauss rules. This device facilitates visual comparison between the computed stiffness matrices:

$$\mathbf{K}_{1 \times 1}^{(e)} = \begin{bmatrix} 1840293 & 1051596 & -262899 & -262899 & -1840293 & -1051596 & 262899 & 262899 \\ 1051596 & 3417687 & -262899 & 1314495 & -1051596 & -3417687 & 262899 & -1314495 \\ -262899 & -262899 & 1051596 & -525798 & 262899 & 262899 & -1051596 & 525798 \\ -262899 & 1314495 & -525798 & 1051596 & 262899 & -1314495 & 525798 & -1051596 \\ -1840293 & -1051596 & 262899 & 262899 & 1840293 & 1051596 & -262899 & -262899 \\ -1051596 & -3417687 & 262899 & -1314495 & 1051596 & 3417687 & -262899 & 1314495 \\ 262899 & 262899 & -1051596 & 525798 & -262899 & -262899 & 1051596 & -525798 \\ 262899 & -1314495 & 525798 & -1051596 & -262899 & 1314495 & -525798 & 1051596 \end{bmatrix} \quad (23.8)$$

$$\mathbf{K}_{2 \times 2}^{(e)} = \begin{bmatrix} 2062746 & 1092042 & -485352 & -303345 & -1395387 & -970704 & -182007 & 182007 \\ 1092042 & 3761478 & -303345 & 970704 & -970704 & -2730105 & 182007 & -2002077 \\ -485352 & -303345 & 1274049 & -485352 & -182007 & 182007 & -606690 & 606690 \\ -303345 & 970704 & -485352 & 1395387 & 182007 & -2002077 & 606690 & -364014 \\ -1395387 & -970704 & -182007 & 182007 & 2730105 & 1213380 & -1152711 & -424683 \\ -970704 & -2730105 & 182007 & -2002077 & 1213380 & 4792851 & -424683 & -60669 \\ -182007 & 182007 & -606690 & 606690 & -1152711 & -424683 & 1941408 & -364014 \\ 182007 & -2002077 & 606690 & -364014 & -424683 & -60669 & -364014 & 2426760 \end{bmatrix} \quad (23.9)$$

$$\mathbf{K}_{3 \times 3}^{(e)} = \begin{bmatrix} 2067026 & 1093326 & -489632 & -304629 & -1386827 & -968136 & -190567 & 179439 \\ 1093326 & 3764046 & -304629 & 968136 & -968136 & -2724969 & 179439 & -2007213 \\ -489632 & -304629 & 1278329 & -484068 & -190567 & 179439 & -598130 & 609258 \\ -304629 & 968136 & -484068 & 1397955 & 179439 & -2007213 & 609258 & -358878 \\ -1386827 & -968136 & -190567 & 179439 & 2747225 & 1218516 & -1169831 & -429819 \\ -968136 & -2724969 & 179439 & -2007213 & 1218516 & 4803123 & -429819 & -70941 \\ -190567 & 179439 & -598130 & 609258 & -1169831 & -429819 & 1958528 & -358878 \\ 179439 & -2007213 & 609258 & -358878 & -429819 & -70941 & -358878 & 2437032 \end{bmatrix} \quad (23.10)$$

$$\mathbf{K}_{4 \times 4}^{(e)} = \begin{bmatrix} 2067156 & 1093365 & -489762 & -304668 & -1386567 & -968058 & -190827 & 179361 \\ 1093365 & 3764124 & -304668 & 968058 & -968058 & -2724813 & 179361 & -2007369 \\ -489762 & -304668 & 1278459 & -484029 & -190827 & 179361 & -597870 & 609336 \\ -304668 & 968058 & -484029 & 1398033 & 179361 & -2007369 & 609336 & -358722 \\ -1386567 & -968058 & -190827 & 179361 & 2747745 & 1218672 & -1170351 & -429975 \\ -968058 & -2724813 & 179361 & -2007369 & 1218672 & 4803435 & -429975 & -71253 \\ -190827 & 179361 & -597870 & 609336 & -1170351 & -429975 & 1959048 & -358722 \\ 179361 & -2007369 & 609336 & -358722 & -429975 & -71253 & -358722 & 2437344 \end{bmatrix} \quad (23.11)$$

As can be seen entries change substantially in going from $p = 1$ to $p = 2$, then more slowly. The eigenvalues of these matrices are:

Rule	Eigenvalues (scaled by 10^{-6}) of $\mathbf{K}^{(e)}$							
1×1	8.77276	3.68059	2.26900	0	0	0	0	0
2×2	8.90944	4.09769	3.18565	2.64521	1.54678	0	0	0
3×3	8.91237	4.11571	3.19925	2.66438	1.56155	0	0	0
4×4	8.91246	4.11627	3.19966	2.66496	1.56199	0	0	0

(23.12)

The stiffness matrix computed by the one-point rule is rank deficient by two. The eigenvalues do not change appreciably after $p = 2$. Because the nonzero eigenvalues measure the internal energy taken up by the element in deformation eigenmodes, it can be seen that raising the order of the integration stiffens the element.

REMARK 23.1

The formation of the trapezoidal element stiffness using floating-point computation by setting `numer=True` took 0.017, 0.083, 0.15 and 0.25 seconds for $p = 1, 2, 3, 4$, respectively, on a Mac G4/867. Changing `numer=False` to do exact computation increases the formation time to 0.033, 1.7, 4.4 and 44.6 seconds, respectively. (The unusually high value for $p = 4$ is due to the time spent in the simplification of the highly complex exact expressions produced by the Gauss quadrature rule.) This underscores the speed advantage of using floating-point arithmetic when exact symbolic and algebraic calculations are not required.


```

Quad4IsoPMembraneBodyForces[ncoor_,mprop_,fprop_,options_,bfor_]:=
Module[{i,k,p=2,numer=False,Emat,th=1,h,
  bx,by,bx1,by1,bx2,by2,bx3,by3,bx4,by4,bxc,byc,qcoor,
  c,w,Nf,dNx,dNy,Jdet,B,qctab,fe=Table[0,{8}]},
If [Length[options]==2, {numer,p}=options, {numer}=options];
If [Length[fprop]>0, th=fprop[[1]]];
If [Length[bfor]==2, {bx,by}=bfor;bx1=bx2=bx3=bx4=bx;by1=by2=by3=by4=by];
If [Length[bfor]==4, {{bx1,by1},{bx2,by2},{bx3,by3},{bx4,by4}}=bfor];
If [p<1||p>4, Print["p out of range"]; Return[Null]];
bxc={bx1,bx2,bx3,bx4}; byc={by1,by2,by3,by4};
For [k=1, k<=p*p, k++,
  {qcoor,w}= QuadGaussRuleInfo[{p,numer},k];
  {Nf,dNx,dNy,Jdet}=Quad4IsoPShapeFunDer[ncoor,qcoor];

  bk=Flatten[Table[{Nf[[i]]*bx,Nf[[i]]*by},{i,4}]];
  fe+=c*bk;
]; Return[fe]
];

```

Figure 23.9. Module for computation of consistent node forces from a given body force field.

§23.4. CONSISTENT NODE FORCES FOR BODY FORCE FIELD

The module `Quad4IsoPMembraneBodyForces` listed in Figure 23.8 computes the consistent force associated with a body force field $\vec{b} = \{b_x, b_y\}$ given over a four-node iso-P quadrilateral in plane stress. The field is specified per unit of volume in componentwise form. For example if the element is subjected to a gravity acceleration field (self-weight) in the $-y$ direction, $b_x = 0$ and $b_y = -\rho g$, where ρ is the mass density.

The arguments of the module are exactly the same as for `Quad4IsoPMembraneStiffness` except for the following differences.

- | | |
|--------------------|---|
| <code>mprop</code> | Not used; retained as placeholder. |
| <code>bfor</code> | Body forces per unit volume. Specified as a two-item one-dimensional list: $\{b_x, b_y\}$, or as a four-entry two-dimensional list: $\{b_{x1}, b_{y1}\}, \{b_{x2}, b_{y2}\}, \{b_{x3}, b_{y3}\}, \{b_{x4}, b_{y4}\}$. In the first form the body force field is taken to be constant over the element. The second form assumes body forces to vary over the element and specified by values at the four corners, from which the field is interpolated bilinearly. |

The module returns `fe` as an 8×1 one dimensional array arranged $\{f_{x1}, f_{y1}, f_{x2}, f_{y2}, f_{x3}, f_{y3}, f_{x4}, f_{y4}\}$ to represent the vector

$$\mathbf{f}^{(e)} = [f_{x1} \ f_{y1} \ f_{x2} \ f_{y2} \ f_{x3} \ f_{y3} \ f_{x4} \ f_{y4}]^T. \quad (23.13)$$

```

Quad4IsoPMembraneStresses[ncoor_,mprop_,fprop_,options_,udis_]:=
Module[{i,k,numer=False,Emat,th=1,h,qcoor,Nf,
  dNx,dNy,Jdet,B,qctab,ue=udis,sige=Table[0,{4},{3}1]},
qctab={{-1,-1},{1,-1},{1,1},{-1,1}};
Emat=mprop[[1]]; numer=options[[1]];
If [Length[udis]==4, ue=Flatten[udis]];
For [k=1, k<=Length[sige], k++,
  qcoor=qctab[[k]]; If [numer, qcoor=N[qcoor]];
  {Nf,dNx,dNy,Jdet}=Quad4IsoPShapeFunDer[ncoor,qcoor];
  B={ Flatten[Table[{dNx[[i]], 0},{i,4}1],
    Flatten[Table[{0, dNy[[i]]},{i,4}1],
    Flatten[Table[{dNy[[i]],dNx[[i]]},{i,4}1]]};
  sige[[k]]=Emat.(B.ue);
]; Return[sige]
];

```

Figure 23.10. Module for calculation of corner stresses.

§23.5. RECOVERY OF CORNER STRESSES

Although the subject of stress recovery is treated in further detail in a later chapter, for completeness a stress computation module for the 4-node quad is shown in Figure 23.10.

The arguments of the module are exactly the same as for Quad4IsoPMembraneStiffness except for the following differences.

fprop	Not used; retained as placeholder.
udis	The 8 corner displacements components. these may be specified as a 8-entry one-dimensional list form: {ux1,uy1, ux2,uy2, ux3,uy3, ux4,uy4}, or as a 4-entry two-dimensional list: {ux1,uy1},{ux2,uy2},{ux3,uy3},{ux4,uy4}.

The module returns the corner stresses stored in a 4-entry, two-dimensional list:

{{sigxx1,sigyy1,sigxy1},{sigxx2,sigyy2,sigxy2}, {sigxx3,sigyy3,sigxy3},
{sigxx4,sigyy4,sigxy4}} to represent the stress array

$$\sigma^{(e)} = \begin{bmatrix} \sigma_{xx1} & \sigma_{xx2} & \sigma_{xx3} & \sigma_{xx4} \\ \sigma_{yy1} & \sigma_{yy2} & \sigma_{yy3} & \sigma_{yy4} \\ \sigma_{xy1} & \sigma_{xy2} & \sigma_{xy3} & \sigma_{xy4} \end{bmatrix} \quad (23.14)$$

The stresses are directly evaluated at the corner points without invoking any smoothing procedure. A more elaborated recovery scheme is presented in a later Chapter.

§23.6. *QUADRILATERAL COORDINATES OF GIVEN POINT

The following inverse problem arises in some applications. Given a 4-node quadrilateral, defined by the Cartesian coordinates $\{x_i, y_i\}$ of its corners, and an arbitrary point $P(x_P, y_P)$, find the quadrilateral coordinates ξ_P, η_P of P . In answering this question it is understood that the quadrilateral coordinates can be extended outside the element, as illustrated in Figure 23.11.

The governing equations are $x_P = x_1 N_1 + x_2 N_2 + x_3 N_3 + x_4 N_4$ and $y_P = y_1 N_1 + y_2 N_2 + y_3 N_3 + y_4 N_4$, where $N_1 = \frac{1}{4}(1 - \xi_P)(1 - \eta_P)$, etc. These bilinear equations are to be solved for $\{\xi_P, \eta_P\}$. Elimination of say, ξ_P , leads to a quadratic equation in η_P : $a\eta_P^2 + b\eta_P + c = 0$. It can be shown that $b^2 \geq 4ac$ so there are two real roots: η_1 and η_2 . These can be back substituted to get ξ_1 and ξ_2 . Of the two solutions: $\{\xi_1, \eta_1\}$ and $\{\xi_2, \eta_2\}$ the one closest to $\xi = 0, \eta = 0$ is to be taken.

Although seemingly straightforward, the process is prone to numerical instabilities. For example, if the quadrilateral becomes a rectangle or parallelogram, the quadratic equations degenerate to linear, and one of the roots takes off to ∞ . In floating point arithmetic severe cancellation can occur in the other root. A robust numerical algorithm, which works stably for any geometry, is obtained by eliminating ξ and η in turn, getting the minimum-modulus root of $a\eta_P^2 + b\eta_P + c = 0$ with the stable formula.¹ $\eta_P^{min} = b/(b + \sqrt{b^2 - 4ac})$, forming the other quadratic equation, and computing its minimum-modulus root the same way. In addition, x_P and y_P are referred to the quadrilateral center as coordinate origin. The resulting algorithm can be presented as follows. Given $\{x_1, y_1, \dots, x_4, y_4\}$ and $\{x_P, y_P\}$, compute

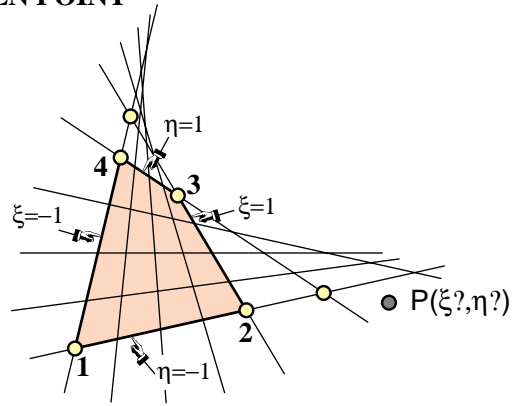


Figure 23.11. Quadrilateral coordinates can be extended outside the element to answer the problem posed in §23.6. The six yellow-filled circles identify the four corners plus the intersections of the opposite sides. This six-point set defines the so-called complete quadrilateral, which is important in projective geometry. The evolute of the coordinate lines is a parabola.

$$\begin{aligned}
 x_b &= x_1 - x_2 + x_3 - x_4, & y_b &= y_1 - y_2 + y_3 - y_4, & x_{cx} &= x_1 + x_2 - x_3 - x_4, & y_{cx} &= y_1 + y_2 - y_3 - y_4, \\
 x_{ce} &= x_1 - x_2 - x_3 + x_4, & y_{ce} &= y_1 - y_2 - y_3 + y_4, & A &= \frac{1}{2}((x_3 - x_1)(y_4 - y_2) - (x_4 - x_2)(y_3 - y_1)), \\
 J_1 &= (x_3 - x_4)(y_1 - y_2) - (x_1 - x_2)(y_3 - y_4), & J_2 &= (x_2 - x_3)(y_1 - y_4) - (x_1 - x_4)(y_2 - y_3), \\
 x_0 &= \frac{1}{4}(x_1 + x_2 + x_3 + x_4), & y_0 &= \frac{1}{4}(y_1 + y_2 + y_3 + y_4), & x_{P0} &= x_P - x_0, & y_{P0} &= y_P - y_0, \\
 b_\xi &= A - x_{P0} y_b + y_{P0} x_b, & b_\eta &= -A - x_{P0} y_b + y_{P0} x_b, & c_\xi &= x_{P0} y_{cx} - y_{P0} x_{cx}, \\
 c_\eta &= x_{P0} y_{ce} - y_{P0} x_{ce}, & \xi_P &= \frac{2c_\xi}{-\sqrt{b_\xi^2 - 2J_1 c_\xi} - b_\xi}, & \eta_P &= \frac{2c_\eta}{\sqrt{b_\eta^2 + 2J_2 c_\eta} - b_\eta}.
 \end{aligned}$$

(23.15)

One common application is to find whether P is inside the quadrilateral: if both ξ_P and η_P are in the range $[-1, 1]$ the point is inside, else outside. This occurs, for example, in relating experimental data from given sensor locations² to an existing FEM mesh.

A *Mathematica* module that implements (23.15) is listed in Figure 23.12.

¹ See Section 5.6 of [23.1].

² While at Boeing in 1969 the writer had to solve a collocation problem of this nature, although in three dimensions. Pressure data measured at a wind tunnel had to be transported to an independently constructed FEM quadrilateral mesh modeling the wing skin.

```

QuadCoordinatesOfPoint[{{x1_,y1_},{x2_,y2_},{x3_,y3_},
{x4_,y4_}},{x_,y_}]:= Module[{A,J0,J1,J2,
xb=x1-x2+x3-x4,yb=y1-y2+y3-y4,xcξ=x1+x2-x3-x4,ycξ=y1+y2-y3-y4,
xcη=x1-x2-x3+x4,ycη=y1-y2-y3+y4,bξ,bη,cξ,cη,
x0=(x1+x2+x3+x4)/4,y0=(y1+y2+y3+y4)/4,dx,dy,ξ,η},
J0=(x3-x1)*(y4-y2)-(x4-x2)*(y3-y1); A=J0/2;
J1=(x3-x4)*(y1-y2)-(x1-x2)*(y3-y4);
J2=(x2-x3)*(y1-y4)-(x1-x4)*(y2-y3);
dx=x-x0; dy=y-y0;
bξ=A-dx*yb+dy*xb; bη=-A-dx*yb+dy*xb;
cξ= dx*ycξ-dy*xcξ; cη=dx*ycη-dy*xcη;
ξ=2*cξ/(-Sqrt[bξ^2-2*J1*cξ]-bξ);
η=2*cη/( Sqrt[bη^2+2*J2*cη]-bη);
Return[{ξ,η}]];

```

Figure 23.11. A *Mathematica* module implementing the algorithm (23.15).

Notes and Bibliography

For an outline of the history of the 4-node quadrilateral, see **Notes and Bibliography** in Chapter 17. The element is called the Taig quadrilateral in the early FEM literature, recognizing his developer [23.4]. This paper actually uses the exactly integrated stiffness matrix. Gauss numerical integration was advocated by Irons [23.2–23.3], who changed the range of the quadrilateral coordinates to $[-1, +1]$ to fit tabulated rules.

References

- [23.1] Press, W. J. et al., *Numerical Recipes: The Art of Scientific Computing*, 2nd ed., Cambridge Univ. Press, 1992.
- [23.2] Irons, B. M., Engineering application of numerical integration in stiffness methods, *AIAA J.*, **4**, pp. 2035–2037, 1966.
- [23.3] Irons, B. M. and Ahmad, S., *Techniques of Finite Elements*, Ellis Horwood Ltd, Chichester, UK, 1980.
- [23.4] Taig, I. C. and Kerr, R. I., Some problems in the discrete element representation of aircraft structures, in *Matrix Methods of Structural Analysis*, ed. by B. M. Fraeijs de Veubeke, Pergamon Press, London, 1964.

Homework Exercises for Chapter 23

Implementation of Iso-P Quadrilateral Elements

EXERCISE 23.1

[C:15] Figures E23.1–2 show the *Mathematica* implementation of the stiffness modules for the 5-node, “bilinear+bubble” iso-P quadrilateral of Figure E18.3. Module Quad5IsoPMembraneStiffness returns the 10×10 stiffness matrix whereas module Quad5IsoPShapeFunDer returns shape function values and Cartesian derivatives. (The Gauss quadrature module is reused.) Both modules follow the style of the 4-node quadrilateral implementation listed in Figures 23.4–5. The only differences in argument lists is that ncoor has five node coordinates: $\{\{x_1, y_1\}, \{x_2, y_2\}, \{x_3, y_3\}, \{x_4, y_4\}, \{x_5, y_5\}\}$, and that a variable plate thickness in fprop (one of the 3 possible formats) is specified as $\{h_1, h_2, h_3, h_4, h_5\}$.

```
Quad5IsoPMembraneStiffness[ncoor_,mprop_,fprop_,options_]:=
Module[{i,j,k,p=2,numer=False,Emat,th=1,h,qcoor,c,w,Nf,
  dNx,dNy,Jdet,B,Ke=Table[0,{10},{10}]},
  Emat=mprop[[1]];
  If [Length[options]==2, {numer,p}=options, {numer}=options];
  If [Length[fprop]>0, th=fprop[[1]]];
  If [p<1||p>4, Print["p out of range"]; Return[Null]];
  For [k=1, k<=p*p, k++,
    {qcoor,w}= QuadGaussRuleInfo[{p,numer},k];
    {Nf,dNx,dNy,Jdet}=Quad5IsoPShapeFunDer[ncoor,qcoor];
    If [Length[th]==0, h=th, h=th.Nf]; c=w*Jdet*h;
    B={ Flatten[Table[{dNx[[i]], 0},{i,5}]],
        Flatten[Table[{0, dNy[[i]]},{i,5}]],
        Flatten[Table[{dNy[[i]],dNx[[i]]},{i,5}]]];
    Ke+=Simplify[c*Transpose[B].(Emat.B)];
  ]; Return[Ke];
];
```

Figure E23.1. Stiffness module for the 5-node “bilinear+bubble” iso-P quadrilateral.

```
Quad5IsoPShapeFunDer[ncoor_,qcoor_]:= Module[
  {Nf,dNx,dNy,dNξ,dNη,Nb,dNbξ,dNbη,J11,J12,J21,J22,Jdet,ξ,η,x,y},
  {ξ,η}=qcoor; Nb=(1-ξ^2)*(1-η^2); (* Nb: node-5 "bubble" function *)
  dNbξ=2*ξ*(η^2-1); dNbη=2*η*(ξ^2-1);
  Nf= { ((1-ξ)*(1-η)-Nb)/4, ((1+ξ)*(1-η)-Nb)/4,
        ((1+ξ)*(1+η)-Nb)/4, ((1-ξ)*(1+η)-Nb)/4, Nb};
  dNξ={ -(1-η+dNbξ)/4, (1-η-dNbξ)/4,
        (1+η-dNbξ)/4, -(1+η+dNbξ)/4, dNbξ};
  dNη={ -(1-ξ+dNbη)/4, -(1+ξ+dNbη)/4,
        (1+ξ-dNbη)/4, (1-ξ-dNbη)/4, dNbη};
  x=Table[ncoor[[i,1]],{i,5}]; y=Table[ncoor[[i,2]],{i,5}];
  J11=dNξ.x; J21=dNξ.y; J12=dNη.x; J22=dNη.y;
  Jdet=Simplify[J11*J22-J12*J21];
  dNx= ( J22*dNξ-J21*dNη)/Jdet; dNx=Simplify[dNx];
  dNy= (-J12*dNξ+J11*dNη)/Jdet; dNy=Simplify[dNy];
  Return[{Nf,dNx,dNy,Jdet}]
];
```

Figure E23.2. The shape function module for the 5-node “bilinear+bubble” iso-P quadrilateral.

```

Quad5IsoPMembraneCondStiffness[Ke5_] :=
Module[{i,j,k,n,c,Ke=Ke5,Kc=Table[0,{8},{8}]},
  For [n=10,n>=9,n--,
    For [i=1,i<=n-1,i++, c=Ke[[i,n]]/Ke[[n,n]];
      For [j=1,j<=i,j++, Ke[[j,i]]=Ke[[i,j]]=Ke[[i,j]]-c*Ke[[n,j]];
    ]];
  For [i=1,i<=8,i++, For [j=1,j<=8,j++, Kc[[i,j]]=Ke[[i,j]]];
  Return[Kc]
];

```

Figure E23.3. A mystery module for Exercise 23.2.

Test `Quad5IsoPMembraneStiffness` for the 2:1 rectangular element studied in §23.3.1, with node 5 placed at the element center. Use Gauss rules 1×1 , 2×2 and 3×3 . Take $E = 96 \times 30 = 2880$ in lieu of $E = 96$ to get exact integer entries in $\mathbf{K}^{(e)}$ for all Gauss rules while keeping $\nu = 1/3$ and $h = 1$. Report on which rules give rank sufficiency. Partial result: $K_{22} = 3380$ and 3588 for the 2×2 and 3×3 rules, respectively.

EXERCISE 23.2

[D:10] Module `Quad5IsoPMembraneCondStiffness` in Figure E23.3 is designed to receive, as only argument, the 10×10 stiffness \mathbf{Ke} computed by `Quad5IsoPMembraneStiffness`, and returns a smaller (8×8) stiffness matrix. State what the function of the module is but do not describe programming details.

EXERCISE 23.3

[C:20] Repeat Exercise 17.3 for the problem illustrated in Figure E17.4, but with the 5-node “bilinear+bubble” iso-P quadrilateral as the 2D element that models the plane beam. Skip item (a). Use the modules of Figures E23.1–3 to do the following. Form the 10×10 stiffness matrix $\mathbf{Ke5}$ using `Quad5IsoPMembraneStiffness` with $p = 2$ and `num=False`. Insert this $\mathbf{Ke5}$ into `Quad5IsoPMembraneCondStiffness`, which returns a 8×8 stiffness \mathbf{Ke} . Stick this \mathbf{Ke} into equations (E17.6) and (E17.7) to get U_{quad} . Show that the energy ratio is

$$r = \frac{U_{quad}}{U_{beam}} = \frac{\gamma^2(1+\nu)(2+\gamma^2(1-\nu))}{(1+\gamma^2)^2}. \quad (\text{E23.1})$$

Compare this to the energy ratio (E17.8) for $\gamma = 1/10$ and $\nu = 0$ to conclude that shear locking has not been eliminated, or even mitigated, by the injection of the bubble shape functions associated with the interior node.³

EXERCISE 23.4

[C:25] Implement the 9-node biquadratic element for plane stress to get its 18×18 stiffness matrix. Follow the style of Figures 23.3–4 or E23.1–2. (The Gauss quadrature module may be reused without change.) Test it for the 2:1 rectangular element studied in §23.3.1, with nodes 5–8 placed at the side midpoints, and node 9 at the element center. For the elastic modulus take $E = 96 \times 39 \times 11 \times 55 \times 7 = 15855840$ instead of $E = 96$, along with $\nu = 1/3$ and $h = 1$, so as to get exact integer entries in $\mathbf{K}^{(e)}$. Use both 2×2 and 3×3 Gauss integration rules and show that the 2×2 rule produces a rank deficiency of 3 in the stiffness. (If the computation with `num=False` takes too long on a slow PC, set `num=True` and `Rationalize` entries as in Figure 23.8.) Partial result: $K_{11} = 5395390$ and 6474468 for the 2×2 and 3×3 rules, respectively.

³ Even the addition of an infinite number of bubble functions to the 4-node iso-P quadrilateral will not cure shear locking. This “bubble futility” has been known since the late 1960s. But memories are short. Bubbles have been recently revived by some FEM authors for other application contexts, such as multiscale modeling.

EXERCISE 23.5

[C:25] An element is said to be *distortion insensitive* when the discrete solution does not appreciably change when the mesh is geometrically distorted while keeping the same number of elements, nodes and degrees of freedom. It is *distortion sensitive* otherwise. A distortion sensitivity test often found in the FEM literature for plane stress quadrilateral elements is pictured in Figure E23.4.

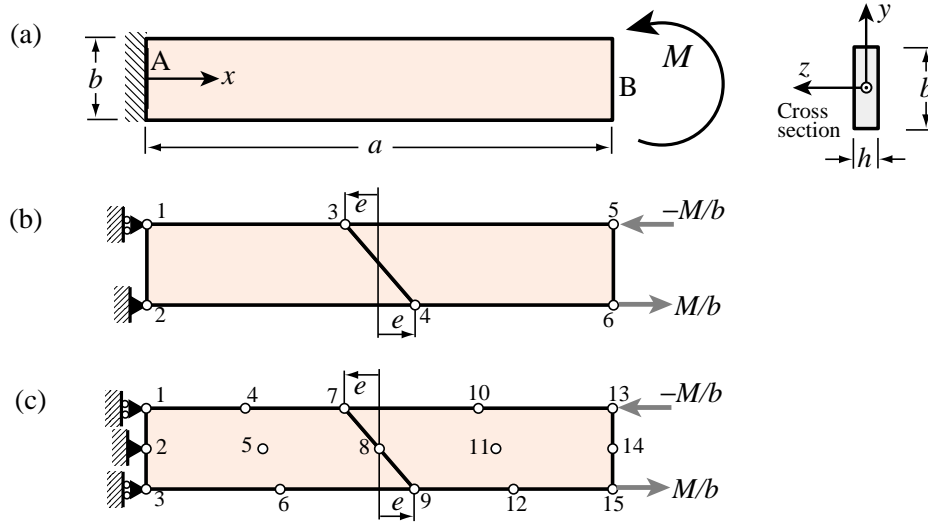


Figure E23.4. Two-element cantilever model for assessing distortion sensitivity, for Exercise E23.5.

The cantilever beam of span a , height b and narrow rectangular cross section of thickness h depicted in Figure E23.4 is under an applied end moment M . If the material is isotropic with elastic modulus E and Poisson ratio $\nu = 0$ the end vertical deflection given by the Bernoulli-Euler beam theory is $v_{beam} = Ma^2/(2EI_{zz})$, where $I_{zz} = hb^3/12$. This is also the exact solution given by elasticity theory if the surface tractions over the free end section match the beam stress distribution.⁴

The problem is discretized with two 4-node isoP bilinear quadrilaterals as illustrated in Figure E23.4(b), which show appropriate displacement and force boundary conditions. The mesh distortion is parametrized by the distance e , which defines the slope of interface 3-4. If $e = 0$ there is no distortion.

Obtain the finite element solution for $a = 10$, $b = 2$, $h = E = M = 1$ and $e = 0, 1, 2, 3, 5$ and record the end deflection v_{quad} as the average of the vertical displacements u_{y5} and u_{y6} . Define the ratio $r(e) = v_{quad}/v_{beam}$ and plot $g(e) = r(e)/r(0)$ as function of e . This function $g(e)$ characterizes the distortion sensitivity. Show that $g(e) < 1$ as e increases and thus the mesh stiffness further as a result of the distortion. Conclude that the 4-node bilinear element is distortion sensitive.

EXERCISE 23.6

[C:25] Repeat the steps of Exercise 23.5 for the mesh depicted in Figure E23.4(c). The cantilever beam is modeled with two 9-node biquadratic quadrilaterals integrated by the 3×3 Gauss product rule. It is important to keep the side nodes at the midpoints of the sides, and the center node at the crossing of the medians (that is, the 9-node elements are superparametric). Show that $r = 1$ for any $e < \frac{1}{2}a$. Thus not only this element is exact for the regular mesh, but it is also distortion insensitive.

⁴ This statement would not be true if $\nu \neq 0$ since the fixed-displacement BC at the cantilever root would preclude the lateral expansion or contraction of the cross section. However, the support condition shown in the models (b) and (c) allow such changes so the results are extendible to nonzero ν .

24

Implementation of Iso-P Triangular Elements

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§24.1. INTRODUCTION

This Chapter continues with the subject of the computer implementation of two-dimensional finite elements. It covers the programming of isoparametric *triangular* elements for the plane stress problem. Triangular elements bring two *sui generis* implementation quirks with respect to quadrilateral elements:

- (1) The numerical integration rules for triangles are not product of one-dimensional Gauss rules, as in the case of quadrilaterals. They are instead specialized to the triangle geometry.
- (2) The computation of x - y partial derivatives and the element-of-area scaling by the Jacobian determinant must account for the fact that the triangular coordinates ζ_1 , ζ_2 and ζ_3 do not form an independent set.

We deal with each of these two differences in turn.

§24.2. GAUSS QUADRATURE FOR TRIANGLES

The numerical integration schemes for quadrilaterals introduced in §17.3 and implemented in §23.2 are built as “tensor products” of two one-dimensional Gauss formulas. On the other hand, Gauss rules for triangles are *not* derivable from one-dimensional rules, and must be constructed especially for the triangular geometry.

§24.2.1. Requirements for Gauss Rules

Gauss quadrature rules for triangles must possess *triangular symmetry* in the following sense:

If the sample point $(\zeta_1, \zeta_2, \zeta_3)$ is present in a Gauss integration rule with weight w , then all other points obtainable by permuting the three triangular coordinates arbitrarily must appear in that rule, and have the same weight.

(24.1)

This rule guarantees that the result of the quadrature process will not depend on element node numbering.¹ If ζ_1 , ζ_2 , and ζ_3 are different, condition (24.1) forces six equal-weight sample points to be present in the rule, because $3! = 6$. If two triangular coordinates are equal, the six points coalesce to three, and the condition forces three equal-weight sample points to be present. Finally, if the three coordinates are equal (which can only happen for the centroid $\zeta_1 = \zeta_2 = \zeta_3 = 1/3$), the six points coalesce to one.²

Additional requirements for a Gauss rule to be numerically acceptable are:

All sample points must be inside the triangle (or on the triangle boundary) and all weights must be positive.

(24.2)

These are called *numerical stability* conditions.

¹ It would be disconcerting to users, to say the least, to have the FEM solution depend on how nodes are numbered.

² It follows that the number of sample points in triangle Gauss quadrature rules must be of the form $6i + 3j + k$, where i and j are nonnegative integers and k is 0 or 1. Consequently there are no rules with 2, 5 or 8 points.

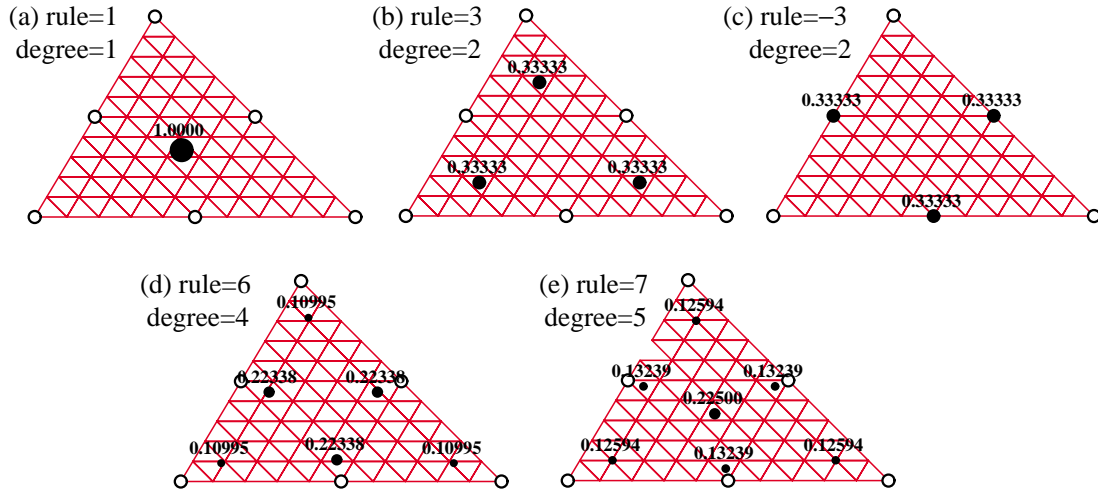


Figure 24.1. Location of sample points (dark circles) of five Gauss quadrature rules for straight sided (superparametric) 6-node triangles. Weight written to 5 places near each sample point; sample-point circle areas are proportional to weight.

REMARK 24.1

The reasons for the stability conditions (24.2) cannot be covered in an elementary course. They are automatically satisfied by all Gauss product rules for quadrilaterals, and so it was not necessary to call attention to them. On the other hand, for triangles there are Gauss rules with as few as 4 and 6 points that violate those conditions.

A rule is said to be of degree n if it integrates exactly all polynomials in the triangular coordinates of order n or less when the Jacobian determinant is constant, and there is at least one polynomial of order $n + 1$ that is not exactly integrated by the rule.

§24.2.2. Superparametric Triangles

We first consider superparametric straight-sided triangles geometry defined by the three corner nodes. Over such triangles the Jacobian determinant defined in §24.3 is constant. The five simplest Gauss rules that satisfy the requirements (24.1) and (24.2) have 1, 3, 3, 6 and 7 points, respectively. The two rules with 3 points differ in the location of the sample points. The five rules are depicted in Figure 24.1 over 6-node quadratic triangles; for such triangles to be superparametric the side nodes must be located at the midpoint of the sides.

One point rule. The simplest Gauss rule for a triangle has *one* sample point located at the centroid. For a straight sided triangle,

$$\frac{1}{A} \int_{\Omega^{(e)}} F(\zeta_1, \zeta_2, \zeta_3) d\Omega^{(e)} \approx F\left(\frac{1}{3}, \frac{1}{3}, \frac{1}{3}\right), \quad (24.3)$$

where A is the triangle area:

$$A = \int_{\Omega^{(e)}} d\Omega^{(e)} = \frac{1}{2} \det \begin{bmatrix} 1 & 1 & 1 \\ x_1 & x_2 & x_3 \\ y_1 & y_2 & y_3 \end{bmatrix} = \frac{1}{2} [(x_2 y_3 - x_3 y_2) + (x_3 y_1 - x_1 y_3) + (x_1 y_2 - x_2 y_1)]. \quad (24.4)$$

This rule is depicted in Figure 24.1(a). It has degree 1, meaning that it integrates exactly up to linear polynomials in triangular coordinates. For example, $F = 4 - \zeta_1 + 2\zeta_2 - \zeta_3$ is exactly integrated by (24.3).

Three Point Rules. The next two rules in order of simplicity contain *three* sample points:

$$\frac{1}{A} \int_{\Omega^{(e)}} F(\zeta_1, \zeta_2, \zeta_3) d\Omega^{(e)} \approx \frac{1}{3} F\left(\frac{2}{3}, \frac{1}{6}, \frac{1}{6}\right) + \frac{1}{3} F\left(\frac{1}{6}, \frac{2}{3}, \frac{1}{6}\right) + \frac{1}{3} F\left(\frac{1}{6}, \frac{1}{6}, \frac{2}{3}\right). \quad (24.5)$$

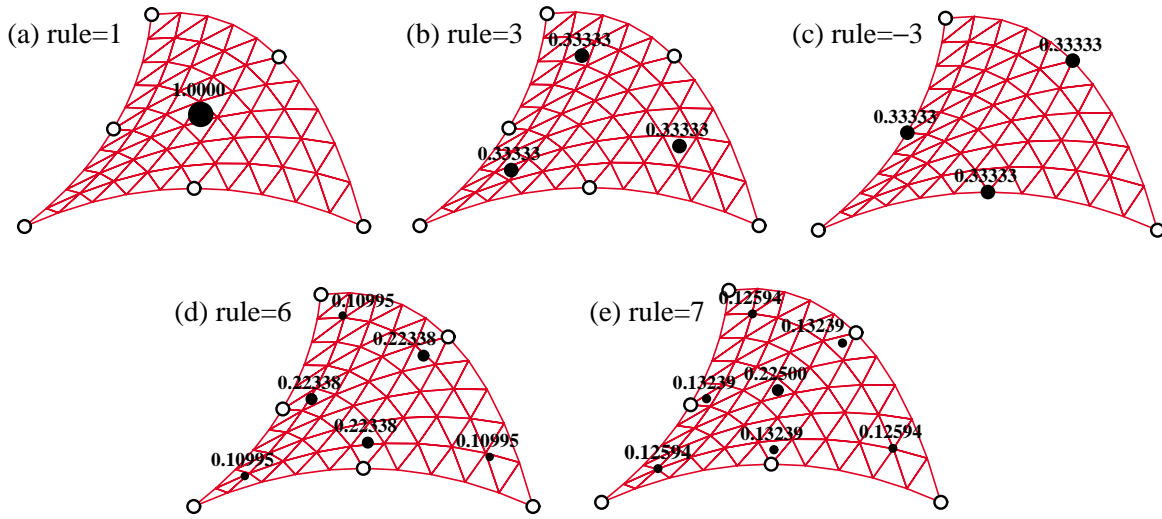


Figure 24.2. Location of sample points (dark circles) of five Gauss quadrature rules for curve sided 6-node triangles. Weight written to 5 places near each sample point; sample-point circle areas are proportional to weight.

$$\frac{1}{A} \int_{\Omega^{(e)}} F(\zeta_1, \zeta_2, \zeta_3) d\Omega^{(e)} \approx \frac{1}{3} F\left(\frac{1}{2}, \frac{1}{2}, 0\right) + \frac{1}{3} F\left(0, \frac{1}{2}, \frac{1}{2}\right) + \frac{1}{3} F\left(\frac{1}{2}, 0, \frac{1}{2}\right). \quad (24.6)$$

These are depicted in Figures 24.1(b) and (c), respectively. Both rules are of degree 2; that is, exact up to quadratic polynomials in the triangular coordinates. For example, the function $F = 6 + \zeta_1 + 3\zeta_3 + \zeta_2^2 - \zeta_3^2 + 3\zeta_1\zeta_3$ is integrated exactly by either rule. Formula (24.6) is called the *midpoint rule*.

REMARK 24.2

In certain applications, such as the axisymmetric solid structures considered in advanced FEM, the internal rule is preferred to avoid 0/0 singularities if an element edge falls on the axis of revolution.

Six and Seven Point Rules. There is a 4-point rule of degree 3 but it has a negative weight, which violates the stability condition (24.2). There are no symmetric rules with 5 points. The next useful rules have six and seven points. There is a 6-point rule of degree 4 and a 7-point rule of degree 5, which integrate exactly up to quartic and quintic polynomials, respectively. The 7-point rule includes the centroid as sample point. The abscissas and weights are expressible as rational combinations of square roots of integers and fractions. The expressions are listed in the *Mathematica* implementation shown in Figure 24.3. The rule configurations are depicted in Figures 24.1(d) and (e).

§24.2.3. Arbitrary Iso-P Triangles

If the triangle has variable metric, as in the curved-side 6-node triangle geometries pictured in Figure 24.2, the foregoing formulas need adjustment because the element of area $d\Omega^{(e)}$ becomes a function of position. Consider the more general case of an isoparametric element with n nodes and shape functions N_i . In §24.3 it is shown that the differential area element is given by

$$d\Omega^{(e)} = J d\zeta_1 d\zeta_2 d\zeta_3, \quad J = \frac{1}{2} \det \begin{bmatrix} \sum_{i=1}^n x_i \frac{\partial N_i}{\partial \zeta_1} & \sum_{i=1}^n x_i \frac{\partial N_i}{\partial \zeta_2} & \sum_{i=1}^n x_i \frac{\partial N_i}{\partial \zeta_3} \\ \sum_{i=1}^n y_i \frac{\partial N_i}{\partial \zeta_1} & \sum_{i=1}^n y_i \frac{\partial N_i}{\partial \zeta_2} & \sum_{i=1}^n y_i \frac{\partial N_i}{\partial \zeta_3} \end{bmatrix} \quad (24.7)$$

```

TrigGaussRuleInfo[{rule_,number_},point_]:= Module[
{zeta,p=rule,i=point,g1,g2,info=NULL},
If [p== 1, info={{1/3,1/3,1/3},1}];
If [p== -3, zeta={1/2,1/2,1/2}; zeta[[i]]=0; info={zeta,1/3}];
If [p== 3, zeta={1/6,1/6,1/6}; zeta[[i]]=2/3; info={zeta,1/3}];
If [p== 6,
If [i<=3, g1=(8-Sqrt[10]+Sqrt[38-44*Sqrt[2/5]])/18;
zeta={g1,g1,g1}; zeta[[i]]=1-2*g1;
info={zeta,(620+Sqrt[213125-53320*Sqrt[10]])/3720}];
If [i>3, g2=(8-Sqrt[10]-Sqrt[38-44*Sqrt[2/5]])/18;
zeta={g2,g2,g2}; zeta[[i-3]]=1-2*g2;
info={zeta,(620-Sqrt[213125-53320*Sqrt[10]])/3720}];
If [p== 7,
If [i==1,info={{1/3,1/3,1/3},9/40} ];
If [i>1&&i<=4,zeta=Table[(6-Sqrt[15])/21,{3}];
zeta[[i-1]]=(9+2*Sqrt[15])/21;
info={zeta,(155-Sqrt[15])/1200}];
If [i>4, zeta=Table[(6+Sqrt[15])/21,{3}];
zeta[[i-4]]=(9-2*Sqrt[15])/21;
info={zeta,(155+Sqrt[15])/1200}];
If [number, Return[N[info]], Return[Simplify[info]]];
];

```

Figure 24.3. Module to get triangle Gauss quadrature rule information.

Here J is the Jacobian determinant, which plays the same role as J in the isoparametric quadrilaterals. If the metric is simply defined by the 3 corners, as in Figure 24.1, the geometry shape functions are $N_1 = \zeta_1$, $N_2 = \zeta_2$ and $N_3 = \zeta_3$. Then the foregoing determinant reduces to that of (24.4), and $J = A$ everywhere. But for general geometries $J = J(\zeta_1, \zeta_2, \zeta_3)$, and the triangle area A cannot be factored out of the integration rules. For example the one point rule becomes

$$\int_{\Omega^{(e)}} F(\zeta_1, \zeta_2, \zeta_3) d\Omega^{(e)} \approx J\left(\frac{1}{3}, \frac{1}{3}, \frac{1}{3}\right) F\left(\frac{1}{3}, \frac{1}{3}, \frac{1}{3}\right). \quad (24.8)$$

whereas the midpoint rule becomes

$$\int_{\Omega^{(e)}} F(\zeta_1, \zeta_2, \zeta_3) d\Omega^{(e)} \approx \frac{1}{3} J\left(\frac{1}{2}, \frac{1}{2}, 0\right) F\left(\frac{1}{2}, \frac{1}{2}, 0\right) + \frac{1}{3} J\left(0, \frac{1}{2}, \frac{1}{2}\right) F\left(0, \frac{1}{2}, \frac{1}{2}\right) + \frac{1}{3} J\left(\frac{1}{2}, 0, \frac{1}{2}\right) F\left(\frac{1}{2}, 0, \frac{1}{2}\right). \quad (24.9)$$

These can be rewritten more compactly by saying that the Gauss integration rule is applied to JF .

§24.2.4. Implementation in Mathematica

The five rules of Figures 24.1–2 are implemented in a *Mathematica* module called TrigGaussRuleInfo, which is listed in Figure 24.3. It is called as

$$\{\{zeta1, zeta2, zeta3\}, w\} = \text{TrigGaussRuleInfo}[\{\text{rule}, \text{numer}\}, \text{point}] \quad (24.10)$$

The module has 3 arguments: rule, numer and i. The first two are grouped in a two-item list. Argument rule, which can be 1, 3, -3, 6 or 7, defines the integration formula as follows: Abs[rule] is the number of sample points. Of the two 3-point choices, if rule is -3 the midpoint rule is picked, else if +3 the 3-interior point rule is chosen. Logical flag numer is set to True or False to request floating-point or exact information, respectively

Argument point is the index of the sample point, which may range from 1 through Abs[rule].

The module returns the two-level list $\{\{\zeta_1, \zeta_2, \zeta_3\}, w\}$, where $\zeta_1, \zeta_2, \zeta_3$ are the triangular coordinates of the sample point, and w is the integration weight. For example, the call `TrigGaussRuleInfo[{3,False},1]` returns $\{\{2/3, 1/6, 1/6\}, 1/3\}$. If rule is not 1, 3, -3, 6 or 7, the module returns Null.

§24.3. PARTIAL DERIVATIVE COMPUTATION

The calculation of Cartesian partial derivatives will be illustrated in this section for the 6-node quadratic isoparametric triangle shown in Figure 24.4. However the results are applicable to arbitrary iso-P triangles with any number of nodes.

The element geometry is defined by the corner coordinates $\{x_i, y_i\}$, with $i = 1, 2, \dots, 6$. Corners are numbered 1,2,3 in counterclockwise sense. Side nodes are numbered 4,5,6 opposite to corners 3,1,2, respectively. Side nodes may be arbitrarily located within positive Jacobian constraints as discussed in §19.4.2. The triangular coordinates are as usual denoted by ζ_1, ζ_2 and ζ_3 , which satisfy $\zeta_1 + \zeta_2 + \zeta_3 = 1$. The quadratic displacement field $\{u_x(\zeta_1, \zeta_2, \zeta_3), u_y(\zeta_1, \zeta_2, \zeta_3)\}$ is defined by the 12 node displacements $\{u_{xi}, u_{yi}\}$, $i = 1, 2, \dots, 6$, as per the iso-P quadratic interpolation formula (16.10–11). That formula is repeated here for convenience:

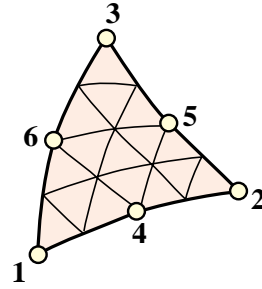


Figure 24.4. The 6-node quadratic iso-P triangle.

$$\begin{bmatrix} 1 \\ x \\ y \\ u_x \\ u_y \end{bmatrix} = \begin{bmatrix} 1 & 1 & 1 & 1 & 1 & 1 \\ x_1 & x_2 & x_3 & x_4 & x_5 & x_6 \\ y_1 & y_2 & y_3 & y_4 & y_5 & y_6 \\ u_{x1} & u_{x2} & u_{x3} & u_{x4} & u_{x5} & u_{x6} \\ u_{y1} & u_{y2} & u_{y3} & u_{y4} & u_{y5} & u_{y6} \end{bmatrix} \mathbf{N}, \quad (24.11)$$

in which the shape functions and their natural derivatives are

$$\mathbf{N} = \begin{bmatrix} N_1^{(e)} \\ N_2^{(e)} \\ N_3^{(e)} \\ N_4^{(e)} \\ N_5^{(e)} \\ N_6^{(e)} \end{bmatrix} = \begin{bmatrix} \zeta_1(2\zeta_1 - 1) \\ \zeta_2(2\zeta_2 - 1) \\ \zeta_3(2\zeta_3 - 1) \\ 4\zeta_1\zeta_2 \\ 4\zeta_2\zeta_3 \\ 4\zeta_3\zeta_1 \end{bmatrix}, \quad \frac{\partial \mathbf{N}}{\partial \zeta_1} = \begin{bmatrix} 4\zeta_1 - 1 \\ 0 \\ 0 \\ 4\zeta_2 \\ 0 \\ 4\zeta_3 \end{bmatrix}, \quad \frac{\partial \mathbf{N}}{\partial \zeta_2} = \begin{bmatrix} 0 \\ 4\zeta_2 - 1 \\ 0 \\ 4\zeta_1 \\ 4\zeta_3 \\ 0 \end{bmatrix}, \quad \frac{\partial \mathbf{N}}{\partial \zeta_3} = \begin{bmatrix} 0 \\ 0 \\ 4\zeta_3 - 1 \\ 0 \\ 4\zeta_2 \\ 4\zeta_1 \end{bmatrix}. \quad (24.12)$$

§24.3.1. Triangle Coordinate Partial

The bulk of the shape function subroutine is concerned with the computation of the partial derivatives of the shape functions (24.12) with respect to x and y at any point in the element. For this purpose consider a generic scalar function: $w(\zeta_1, \zeta_2, \zeta_3)$, which is quadratically interpolated over the triangle by

$$w = [w_1 \quad w_2 \quad w_3 \quad w_4 \quad w_5 \quad w_6] \begin{bmatrix} \zeta_1(2\zeta_1 - 1) \\ \zeta_2(2\zeta_2 - 1) \\ \zeta_3(2\zeta_3 - 1) \\ 4\zeta_1\zeta_2 \\ 4\zeta_2\zeta_3 \\ 4\zeta_3\zeta_1 \end{bmatrix}. \quad (24.13)$$

Symbol w may represent 1, x , y , u_x or u_y , which are interpolated in the iso-P representation (24.11), or other element-varying quantities such as thickness, temperature, etc. Taking partials with respect to x and y and applying the chain rule twice yields

$$\begin{aligned}\frac{\partial w}{\partial x} &= \sum w_i \frac{\partial N_i}{\partial x} = \sum w_i \left(\frac{\partial N_i}{\partial \zeta_1} \frac{\partial \zeta_1}{\partial x} + \frac{\partial N_i}{\partial \zeta_2} \frac{\partial \zeta_2}{\partial x} + \frac{\partial N_i}{\partial \zeta_3} \frac{\partial \zeta_3}{\partial x} \right), \\ \frac{\partial w}{\partial y} &= \sum w_i \frac{\partial N_i}{\partial y} = \sum w_i \left(\frac{\partial N_i}{\partial \zeta_1} \frac{\partial \zeta_1}{\partial y} + \frac{\partial N_i}{\partial \zeta_2} \frac{\partial \zeta_2}{\partial y} + \frac{\partial N_i}{\partial \zeta_3} \frac{\partial \zeta_3}{\partial y} \right),\end{aligned}\quad (24.14)$$

where all sums are understood to run from $i = 1$ through 6. In matrix form:

$$\begin{bmatrix} \frac{\partial w}{\partial x} \\ \frac{\partial w}{\partial y} \end{bmatrix} = \begin{bmatrix} \frac{\partial \zeta_1}{\partial x} & \frac{\partial \zeta_2}{\partial x} & \frac{\partial \zeta_3}{\partial x} \\ \frac{\partial \zeta_1}{\partial y} & \frac{\partial \zeta_2}{\partial y} & \frac{\partial \zeta_3}{\partial y} \end{bmatrix} \begin{bmatrix} \sum w_i \frac{\partial N_i}{\partial \zeta_1} \\ \sum w_i \frac{\partial N_i}{\partial \zeta_2} \\ \sum w_i \frac{\partial N_i}{\partial \zeta_3} \end{bmatrix}. \quad (24.15)$$

Transposing both sides of (24.15) while exchanging left and right sides:

$$\begin{bmatrix} \sum w_i \frac{\partial N_i}{\partial \zeta_1} & \sum w_i \frac{\partial N_i}{\partial \zeta_2} & \sum w_i \frac{\partial N_i}{\partial \zeta_3} \end{bmatrix} \begin{bmatrix} \frac{\partial \zeta_1}{\partial x} & \frac{\partial \zeta_1}{\partial y} \\ \frac{\partial \zeta_2}{\partial x} & \frac{\partial \zeta_2}{\partial y} \\ \frac{\partial \zeta_3}{\partial x} & \frac{\partial \zeta_3}{\partial y} \end{bmatrix} = \begin{bmatrix} \frac{\partial w}{\partial x} & \frac{\partial w}{\partial y} \end{bmatrix}. \quad (24.16)$$

Now make $w \equiv 1, x, y$ and stack the results row-wise:

$$\begin{bmatrix} \sum \frac{\partial N_i}{\partial \zeta_1} & \sum \frac{\partial N_i}{\partial \zeta_2} & \sum \frac{\partial N_i}{\partial \zeta_3} \\ \sum x_i \frac{\partial N_i}{\partial \zeta_1} & \sum x_i \frac{\partial N_i}{\partial \zeta_2} & \sum x_i \frac{\partial N_i}{\partial \zeta_3} \\ \sum y_i \frac{\partial N_i}{\partial \zeta_1} & \sum y_i \frac{\partial N_i}{\partial \zeta_2} & \sum y_i \frac{\partial N_i}{\partial \zeta_3} \end{bmatrix} \begin{bmatrix} \frac{\partial \zeta_1}{\partial x} & \frac{\partial \zeta_1}{\partial y} \\ \frac{\partial \zeta_2}{\partial x} & \frac{\partial \zeta_2}{\partial y} \\ \frac{\partial \zeta_3}{\partial x} & \frac{\partial \zeta_3}{\partial y} \end{bmatrix} = \begin{bmatrix} \frac{\partial 1}{\partial x} & \frac{\partial 1}{\partial y} \\ \frac{\partial x}{\partial x} & \frac{\partial x}{\partial y} \\ \frac{\partial y}{\partial x} & \frac{\partial y}{\partial y} \end{bmatrix}. \quad (24.17)$$

But $\partial x/\partial x = \partial y/\partial y = 1$ and $\partial 1/\partial x = \partial 1/\partial y = \partial x/\partial y = \partial y/\partial x = 0$ because x and y are independent coordinates. It is shown in Remark 24.2 below that, if $\sum N_i = 1$, the entries of the first row of the coefficient matrix are equal to a constant C . These can be scaled to unity because the first row of the right-hand side is null. Consequently we arrive at a system of linear equations of order 3 with two right-hand sides:

$$\begin{bmatrix} 1 & 1 & 1 \\ \sum x_i \frac{\partial N_i}{\partial \zeta_1} & \sum x_i \frac{\partial N_i}{\partial \zeta_2} & \sum x_i \frac{\partial N_i}{\partial \zeta_3} \\ \sum y_i \frac{\partial N_i}{\partial \zeta_1} & \sum y_i \frac{\partial N_i}{\partial \zeta_2} & \sum y_i \frac{\partial N_i}{\partial \zeta_3} \end{bmatrix} \begin{bmatrix} \frac{\partial \zeta_1}{\partial x} & \frac{\partial \zeta_1}{\partial y} \\ \frac{\partial \zeta_2}{\partial x} & \frac{\partial \zeta_2}{\partial y} \\ \frac{\partial \zeta_3}{\partial x} & \frac{\partial \zeta_3}{\partial y} \end{bmatrix} = \begin{bmatrix} 0 & 0 \\ 1 & 0 \\ 0 & 1 \end{bmatrix}. \quad (24.18)$$

§24.3.2. Solving the Jacobian System

By analogy with the quadrilateral isoparametric elements, the coefficient matrix of system (24.18) is called the *Jacobian matrix* and is denoted by \mathbf{J} . Its determinant scaled by one half is equal to the Jacobian $J = \frac{1}{2} \det \mathbf{J}$ used in the expression of the area element introduced in §24.2.3. For compactness (24.18) is rewritten

$$\mathbf{J}\mathbf{P} = \begin{bmatrix} 1 & 1 & 1 \\ J_{x1} & J_{x2} & J_{x3} \\ J_{y1} & J_{y2} & J_{y3} \end{bmatrix} \begin{bmatrix} \frac{\partial \zeta_1}{\partial x} & \frac{\partial \zeta_1}{\partial y} \\ \frac{\partial \zeta_2}{\partial x} & \frac{\partial \zeta_2}{\partial y} \\ \frac{\partial \zeta_3}{\partial x} & \frac{\partial \zeta_3}{\partial y} \end{bmatrix} = \begin{bmatrix} 0 & 0 \\ 1 & 0 \\ 0 & 1 \end{bmatrix}. \quad (24.19)$$

Solving this system gives

$$\begin{bmatrix} \frac{\partial \zeta_1}{\partial x} & \frac{\partial \zeta_1}{\partial y} \\ \frac{\partial \zeta_2}{\partial x} & \frac{\partial \zeta_2}{\partial y} \\ \frac{\partial \zeta_3}{\partial x} & \frac{\partial \zeta_3}{\partial y} \end{bmatrix} = \frac{1}{2J} \begin{bmatrix} J_{y23} & J_{x32} \\ J_{y31} & J_{x13} \\ J_{y12} & J_{x21} \end{bmatrix} = \mathbf{P}, \quad (24.20)$$

in which $J_{xji} = J_{xj} - J_{xi}$, $J_{yji} = J_{yj} - J_{yi}$ and $J = \frac{1}{2} \det \mathbf{J} = \frac{1}{2} (J_{x21} J_{y31} - J_{y12} J_{x13})$. Substituting into (24.14) we arrive at

$$\begin{aligned} \frac{\partial w}{\partial x} &= \sum w_i \frac{\partial N_i}{\partial x} = \sum \frac{w_i}{2J} \left(\frac{\partial N_i}{\partial \zeta_1} J_{y23} + \frac{\partial N_i}{\partial \zeta_2} J_{y31} + \frac{\partial N_i}{\partial \zeta_3} J_{y12} \right), \\ \frac{\partial w}{\partial y} &= \sum w_i \frac{\partial N_i}{\partial y} = \sum \frac{w_i}{2J} \left(\frac{\partial N_i}{\partial \zeta_1} J_{x32} + \frac{\partial N_i}{\partial \zeta_2} J_{x13} + \frac{\partial N_i}{\partial \zeta_3} J_{x21} \right). \end{aligned} \quad (24.21)$$

In particular, the partials of the shape functions are

$$\begin{aligned} \frac{\partial N_i}{\partial x} &= \frac{1}{2J} \left(\frac{\partial N_i}{\partial \zeta_1} J_{y23} + \frac{\partial N_i}{\partial \zeta_2} J_{y31} + \frac{\partial N_i}{\partial \zeta_3} J_{y12} \right), \\ \frac{\partial N_i}{\partial y} &= \frac{1}{2J} \left(\frac{\partial N_i}{\partial \zeta_1} J_{x32} + \frac{\partial N_i}{\partial \zeta_2} J_{x13} + \frac{\partial N_i}{\partial \zeta_3} J_{x21} \right). \end{aligned} \quad (24.22)$$

In matrix form:

$$\begin{bmatrix} \frac{\partial N_i}{\partial x} \\ \frac{\partial N_i}{\partial y} \end{bmatrix} = \mathbf{P}^T \begin{bmatrix} \frac{\partial N_i}{\partial \zeta_1} & \frac{\partial N_i}{\partial \zeta_2} & \frac{\partial N_i}{\partial \zeta_3} \end{bmatrix}^T, \quad (24.23)$$

where \mathbf{P} is the 3×2 matrix of triangular coordinate partials defined in (24.20).

REMARK 24.3

Here is the proof that each of the entries of the first row of (24.17) is a constant, say C . Suppose the shape functions are polynomials of order n in the triangular coordinates, and let $Z = \zeta_1 + \zeta_2 + \zeta_3$. The completeness identity is

$$S = \sum N_i = 1 = c_1 Z + c_2 Z^2 + \dots c_n Z^n, \quad c_1 + c_2 + \dots + c_n = 1. \quad (24.24)$$

where the c_i are element dependent scalar coefficients. Differentiating S with respect to the ζ_i 's and setting $Z = 1$ yields

$$\begin{aligned} C = \sum \frac{\partial N_i}{\partial \zeta_1} &= \sum \frac{\partial N_i}{\partial \zeta_2} = \sum \frac{\partial N_i}{\partial \zeta_3} = c_1 + 2c_2 Z + c_3 Z^3 + \dots (n-1) Z^{n-1} = c_1 + 2c_2 + 3c_3 + \dots + c_{n-1} \\ &= 1 + c_2 + 2c_3 + \dots + (n-2)c_n, \end{aligned} \quad (24.25)$$

which proves the assertion. For the 3-node linear triangle, $S = Z$ and $C = 1$. For the 6-node quadratic triangle, $S = 2Z^2 - Z$ and $C = 3$. For the 10-node cubic triangle, $S = 9Z^3/2 - 9Z^2/2 + Z$ and $C = 11/2$. Because the first equation in (24.18) is homogeneous, the C 's can be scaled to unity.


```

Trig6IsoPShapeFunDer[ncoor_,tcoor_]:= Module[
{ξ1,ξ2,ξ3,x1,x2,x3,x4,x5,x6,y1,y2,y3,y4,y5,y6,
dx4,dx5,dx6,dy4,dy5,dy6,Jx21,Jx32,Jx13,Jy12,Jy23,Jy31,
Nf,dNx,dNy,Jdet}, {ξ1,ξ2,ξ3}=tcoor;
{{x1,y1},{x2,y2},{x3,y3},{x4,y4},{x5,y5},{x6,y6}}=ncoor;
dx4=x4-(x1+x2)/2; dx5=x5-(x2+x3)/2; dx6=x6-(x3+x1)/2;
dy4=y4-(y1+y2)/2; dy5=y5-(y2+y3)/2; dy6=y6-(y3+y1)/2;
Nf={ξ1*(2*ξ1-1),ξ2*(2*ξ2-1),ξ3*(2*ξ3-1),4*ξ1*ξ2,4*ξ2*ξ3,4*ξ3*ξ1};
Jx21= x2-x1+4*(dx4*(ξ1-ξ2)+(dx5-dx6)*ξ3);
Jx32= x3-x2+4*(dx5*(ξ2-ξ3)+(dx6-dx4)*ξ1);
Jx13= x1-x3+4*(dx6*(ξ3-ξ1)+(dx4-dx5)*ξ2);
Jy12= y1-y2+4*(dy4*(ξ2-ξ1)+(dy6-dy5)*ξ3);
Jy23= y2-y3+4*(dy5*(ξ3-ξ2)+(dy4-dy6)*ξ1);
Jy31= y3-y1+4*(dy6*(ξ1-ξ3)+(dy5-dy4)*ξ2);
Jdet = Jx21*Jy31-Jy12*Jx13;
dNx= {(4*ξ1-1)*Jy23,(4*ξ2-1)*Jy31,(4*ξ3-1)*Jy12,4*(ξ2*Jy23+ξ1*Jy31),
4*(ξ3*Jy31+ξ2*Jy12),4*(ξ1*Jy12+ξ3*Jy23)}/Jdet;
dNy= {(4*ξ1-1)*Jx32,(4*ξ2-1)*Jx13,(4*ξ3-1)*Jx21,4*(ξ2*Jx32+ξ1*Jx13),
4*(ξ3*Jx13+ξ2*Jx21),4*(ξ1*Jx21+ξ3*Jx32)}/Jdet;
Return[Simplify[{Nf,dNx,dNy,Jdet}]]
];

```

Figure 24.5. Shape function module for 6-node quadratic triangle.

§24.4. THE QUADRATIC TRIANGLE

§24.4.1. Shape Function Module

We specialize now the results of §24.3 to work out the stiffness matrix of the 6-node quadratic triangle in plane stress. Taking the dot products of the natural-coordinate partials (24.12) with the node coordinates we obtain the entries of the Jacobian matrix (24.19):

$$\begin{aligned}
J_{x1} &= x_1(4\xi_1 - 1) + 4(x_4\xi_2 + x_6\xi_3), & J_{x2} &= x_2(4\xi_2 - 1) + 4(x_5\xi_3 + x_4\xi_1), & J_{x3} &= x_3(4\xi_3 - 1) + 4(x_6\xi_1 + x_5\xi_2), \\
J_{y1} &= y_1(4\xi_1 - 1) + 4(y_4\xi_2 + y_6\xi_3), & J_{y2} &= y_2(4\xi_2 - 1) + 4(y_5\xi_3 + y_4\xi_1), & J_{y3} &= y_3(4\xi_3 - 1) + 4(y_6\xi_1 + y_5\xi_2),
\end{aligned} \tag{24.26}$$

from which the matrix \mathbf{J} can be computed, and $\partial N_i / \partial x$ and $\partial N_i / \partial y$ obtained from (24.22). Somewhat simpler expressions, however, can be obtained by using the following “hierarchical” side node coordinates:

$$\begin{aligned}
\Delta x_4 &= x_4 - \frac{1}{2}(x_1 + x_2), & \Delta x_5 &= x_5 - \frac{1}{2}(x_2 + x_3), & \Delta x_6 &= x_6 - \frac{1}{2}(x_3 + x_1), \\
\Delta y_4 &= y_4 - \frac{1}{2}(y_1 + y_2), & \Delta y_5 &= y_5 - \frac{1}{2}(y_2 + y_3), & \Delta y_6 &= y_6 - \frac{1}{2}(y_3 + y_1).
\end{aligned} \tag{24.27}$$

Geometrically these represent the deviations from the midpoint positions; thus for a superparametric element $\Delta x_4 = \Delta x_5 = \Delta x_6 = \Delta y_4 = \Delta y_5 = \Delta y_6 = 0$. The Jacobian coefficients become

$$\begin{aligned}
J_{x21} &= x_{21} + 4(\Delta x_4(\xi_1 - \xi_2) + (\Delta x_5 - \Delta x_6)\xi_3), & J_{x32} &= x_{32} + 4(\Delta x_5(\xi_2 - \xi_3) + (\Delta x_6 - \Delta x_4)\xi_1), \\
J_{x13} &= x_{13} + 4(\Delta x_6(\xi_3 - \xi_1) + (\Delta x_4 - \Delta x_5)\xi_2), & J_{y12} &= y_{12} + 4(\Delta y_4(\xi_2 - \xi_1) + (\Delta y_6 - \Delta y_5)\xi_3), \\
J_{y23} &= y_{23} + 4(\Delta y_5(\xi_3 - \xi_2) + (\Delta y_4 - \Delta y_6)\xi_1), & J_{y31} &= y_{31} + 4(\Delta y_6(\xi_1 - \xi_3) + (\Delta y_5 - \Delta y_4)\xi_2).
\end{aligned} \tag{24.28}$$

From this one gets $J = \frac{1}{2} \det \mathbf{J} = \frac{1}{2} (J_{x21} J_{y31} - J_{y12} J_{x13})$ and

$$\mathbf{P}^T = \frac{1}{2J} \begin{bmatrix} y_{23} + 4(\Delta y_5(\xi_3 - \xi_2) + (\Delta y_4 - \Delta y_6)\xi_1) & x_{32} + 4(\Delta x_5(\xi_2 - \xi_3) + (\Delta x_6 - \Delta x_4)\xi_1) \\ y_{31} + 4(\Delta y_6(\xi_1 - \xi_3) + (\Delta y_5 - \Delta y_4)\xi_2) & x_{13} + 4(\Delta x_6(\xi_3 - \xi_1) + (\Delta x_4 - \Delta x_5)\xi_2) \\ y_{12} + 4(\Delta y_4(\xi_2 - \xi_1) + (\Delta y_6 - \Delta y_5)\xi_3) & x_{21} + 4(\Delta x_4(\xi_1 - \xi_2) + (\Delta x_5 - \Delta x_6)\xi_3) \end{bmatrix}. \tag{24.29}$$

```

Trig6IsoPMembraneStiffness[ncoor_,mprop_,fprop_,opt_]:=
Module[{i,k,l,p=3,number=False,Emat,th={fprop},h,tcoor,w,c,
  Nf,dNx,dNy,Jdet,B,Ke=Table[0,{12},{12}]},
  Emat=mprop[[1]]; If [Length[fprop]>0, th=fprop[[1]]];
  If [Length[opt]>0, number=opt[[1]]];
  If [Length[opt]>1, p= opt[[2]]];
  If [p!=-3&&p!=1&&p!=3&&p!=7, Print["Illegal p"];Return[Null]];
  For [k=1, k<=Abs[p], k++,
    {tcoor,w}= TrigGaussRuleInfo[{p,number},k];
    {Nf,dNx,dNy,Jdet}= Trig6IsoPShapeFunDer[ncoor,tcoor];
    If [Length[th]==0, h=th, h=th.Nf]; c=w*Jdet*h/2;
    B= {Flatten[Table[{dNx[[i]],0},{i,6}]],
        Flatten[Table[{0,dNy[[i]]},{i,6}]],
        Flatten[Table[{dNy[[i]],dNx[[i]]},{i,6}]]};
    Ke+=c*Transpose[B].(Emat.B);
  ]; If[!number,Ke=Simplify[Ke]]; Return[Ke]
];

```

Figure 24.6. Stiffness matrix module for 6-node plane stress triangle.

Specifically, the Cartesian derivatives of the shape functions are given by

$$\frac{\partial \mathbf{N}}{\partial x} = \frac{1}{2J} \begin{bmatrix} (4\zeta_1 - 1)J_{y23} \\ (4\zeta_2 - 1)J_{y31} \\ (4\zeta_3 - 1)J_{y12} \\ 4(\zeta_2 J_{y23} + \zeta_1 J_{y31}) \\ 4(\zeta_3 J_{y31} + \zeta_2 J_{y12}) \\ 4(\zeta_1 J_{y12} + \zeta_3 J_{y23}) \end{bmatrix}, \quad \frac{\partial \mathbf{N}}{\partial y} = \frac{1}{2J} \begin{bmatrix} (4\zeta_1 - 1)J_{x32} \\ (4\zeta_2 - 1)J_{x13} \\ (4\zeta_3 - 1)J_{x21} \\ 4(\zeta_2 J_{x32} + \zeta_1 J_{x13}) \\ 4(\zeta_3 J_{x13} + \zeta_2 J_{x21}) \\ 4(\zeta_1 J_{x21} + \zeta_3 J_{x32}) \end{bmatrix}. \quad (24.30)$$

Using these expressions, a *Mathematica* implementation of the shape functions is programmed as module Trig6IsoPShapeFunDer, which is shown in Figure 24.5. This module receives two arguments: ncoor and tcoor. The first one is the list of $\{x_i, y_i\}$ coordinates of the six nodes: $\{\{x_1, y_1\}, \{x_2, y_2\}, \dots, \{x_6, y_6\}\}$. The second is the list of three triangular coordinates $\{\zeta_1, \zeta_2, \zeta_3\}$ of the location at which the shape functions and their Cartesian derivatives are to be computed.

The module returns $\{Nf, dNx, dNy, Jdet\}$ as module value. Here Nf collects the shape function values, dNx the shape function x derivative values, dNy the shape function y derivative values, and Jdet is the determinant of matrix \mathbf{J} , equal to $2J$ in the notation used here.

§24.4.2. Stiffness Module

The integration formula for the triangle stiffness may be stated as

$$\mathbf{K}^{(e)} = \int_{\Omega^{(e)}} h \mathbf{B}^T \mathbf{E} \mathbf{B} d\Omega^{(e)} \approx \sum_{i=1}^p w_i \mathbf{F}(\zeta_{1i}, \zeta_{2i}, \zeta_{3i}), \quad \text{where} \quad \mathbf{F}(\zeta_1, \zeta_2, \zeta_3) = h \mathbf{B}^T \mathbf{E} \mathbf{B} J. \quad (24.31)$$

Here p denotes the number of sample points of the Gauss rule being used, w_i is the integration weight for the i^{th} sample point, $\zeta_{1i}, \zeta_{2i}, \zeta_{3i}$ are the sample point triangular coordinates and $J = \frac{1}{2} \det \mathbf{J}$. The last four numbers are returned by TrigGaussRuleInfo as explained in the previous section.

Module Trig6IsoPMembraneStiffness, listed in Figure 24.6, implements the computation of the element stiffness matrix of a quadratic plane stress triangle. The arguments of the module are

ncoor Node coordinates arranged in two-dimensional list form:
 $\{\{x_1, y_1\}, \{x_2, y_2\}, \{x_3, y_3\}, \{x_4, y_4\}, \{x_5, y_5\}, \{x_6, y_6\}\}.$

mprop Material properties supplied as the list $\{\text{Emat}, \rho, \alpha\}$. Emat is a two-dimensional list storing the 3×3 plane stress matrix of elastic moduli:

$$\mathbf{E} = \begin{bmatrix} E_{11} & E_{12} & E_{13} \\ E_{12} & E_{22} & E_{23} \\ E_{13} & E_{23} & E_{33} \end{bmatrix} \quad (24.32)$$

The other two items in **mprop** are not used in this module so zeros may be inserted as placeholders.

fprop Fabrication properties. The plate thickness specified as a one-entry list: $\{h\}$, as a six-entry list: $\{h_1, h_2, h_3, h_4, h_5, h_6\}$, or as an empty list: $\{\}$.

The one-entry form specifies uniform thickness h . The six-entry form is used to specify an element of variable thickness, in which case the entries are the six node thicknesses and h is interpolated quadratically. If an empty list appears the module assumes a uniform unit thickness.

options Processing options. This list may contain two items: $\{\text{number}, \text{rule}\}$ or one: $\{\text{number}\}$.

number is a flag with value **True** or **False**. If **True**, the computations are forced to go in floating point arithmetic. For symbolic or exact arithmetic work set **number** to **False**.

rule specifies the triangle Gauss rule as described in §24.2.4. **rule** may be 1, 3, -3, 6 or 7. For the 6-node element the three point rules are sufficient to get the correct rank. If omitted **rule** = 3 is assumed.

The module returns \mathbf{K}_e as an 12×12 symmetric matrix pertaining to the following arrangement of nodal displacements:

$$\mathbf{u}^{(e)} = [u_{x1} \ u_{y1} \ u_{x2} \ u_{y2} \ u_{x3} \ u_{y3} \ u_{x4} \ u_{y4} \ u_{x5} \ u_{y5} \ u_{x6} \ u_{y6}]^T. \quad (24.33)$$

§24.4.3. Test Elements

The stiffness module is tested on the two triangle geometries shown in Figure 24.7. Both elements have unit thickness and isotropic material. The left one has the corner nodes placed at $(0, 0)$, $(4, 2)$ and $(6, 4)$ with side nodes 4,5,6 at the midpoints of the sides. The right one has the 3 corners forming an equilateral triangle: $-1/2, 0, 1/2, 0, 0, \sqrt{3}/2$ whereas the side nodes are placed at $0, -1/(2\sqrt{3}), 1/2, 1/\sqrt{3}, -1/2, 1/\sqrt{3}$ so that the six nodes lie on a circle of radius $1/\sqrt{3}$. These geometries will be used to illustrate the effect of the numerical integration rule.

The following test statements form $\mathbf{K}^{(e)}$ for the left triangle of Figure 24.7 with $E = 288$, $\nu = 1/3$ and $h = 1$:

```
ClearAll[Em,nu,h]; h=1; Em=288; nu=1/3;
ncoor={{0,0},{6,2},{4,4},{3,1},{5,3},{2,2}};
Emat=Em/(1-nu^2)*{{1,nu,0},{nu,1,0},{0,0,(1-nu)/2}};
Ke=Trig6IsoPMembraneStiffness[ncoor,{Emat,0,0},{h},{False,-3}];
Ke=Simplify[Ke]; Print[Chop[Ke]//MatrixForm];
Print["eigs of Ke=",Chop[Eigenvalues[N[Ke]]]];
```

The returned stiffness matrix for integration **rule**=3, **rule**=-3, and **rule**=7 is the same since those are exact for this element if the side nodes are at the midpoints, which is the case here. That stiffness is

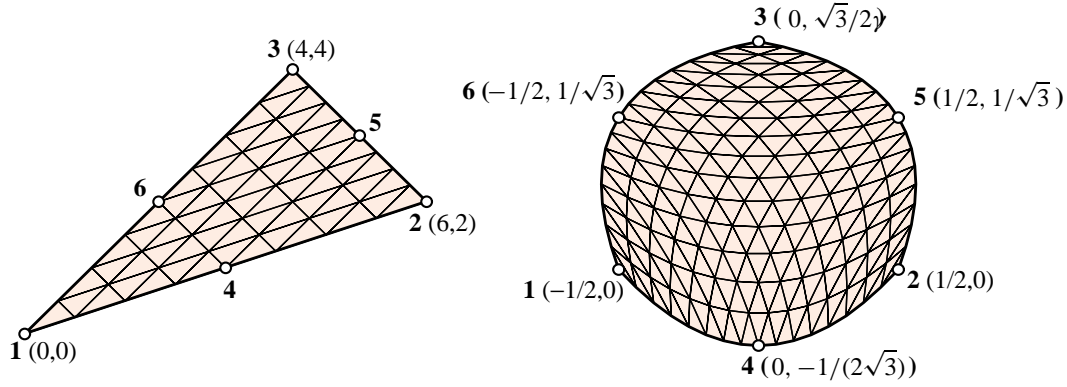


Figure 24.7. Test 6-node triangle element geometries.

$$\begin{bmatrix}
 54 & 27 & 18 & 0 & 0 & 9 & -72 & 0 & 0 & 0 & 0 & -36 \\
 27 & 54 & 0 & -18 & 9 & 36 & 0 & 72 & 0 & 0 & -36 & -144 \\
 18 & 0 & 216 & -108 & 54 & -36 & -72 & 0 & -216 & 144 & 0 & 0 \\
 0 & -18 & -108 & 216 & -36 & 90 & 0 & 72 & 144 & -360 & 0 & 0 \\
 0 & 9 & 54 & -36 & 162 & -81 & 0 & 0 & -216 & 144 & 0 & -36 \\
 9 & 36 & -36 & 90 & -81 & 378 & 0 & 0 & 144 & -360 & -36 & -144 \\
 -72 & 0 & -72 & 0 & 0 & 0 & 576 & -216 & 0 & -72 & -432 & 288 \\
 0 & 72 & 0 & 72 & 0 & 0 & -216 & 864 & -72 & -288 & 288 & -720 \\
 0 & 0 & -216 & 144 & -216 & 144 & 0 & -72 & 576 & -216 & -144 & 0 \\
 0 & 0 & 144 & -360 & 144 & -360 & -72 & -288 & -216 & 864 & 0 & 144 \\
 0 & -36 & 0 & 0 & 0 & -36 & -432 & 288 & -144 & 0 & 576 & -216 \\
 -36 & -144 & 0 & 0 & -36 & -144 & 288 & -720 & 0 & 144 & -216 & 864
 \end{bmatrix} \quad (24.34)$$

The eigenvalues are:

$$[1971.66 \ 1416.75 \ 694.82 \ 545.72 \ 367.7 \ 175.23 \ 157.68 \ 57.54 \ 12.899 \ 0 \ 0 \ 0] \quad (24.35)$$

The 3 zero eigenvalues pertain to the three independent rigid-body modes. The 9 other ones are positive. Consequently the computed $\mathbf{K}^{(e)}$ has the correct rank of 9.

For the “circle” geometry the results for $E = 504$, $\nu = 0$, $h = 1$ and the rank-sufficient rules 3, -3, 6, 7 are obtained through the following script:

```

ClearAll[Em,nu,h]; Em=7*72; nu=0; h=1;
{x1,y1}={-1,0}/2; {x2,y2}={1,0}/2; {x3,y3}={0,Sqrt[3]}/2;
{x4,y4}={0,-1/Sqrt[3]}/2; {x5,y5}={1/2,1/Sqrt[3]}; {x6,y6}={-1/2,1/Sqrt[3]};
ncoor= {{x1,y1},{x2,y2},{x3,y3},{x4,y4},{x5,y5},{x6,y6}};
Emat=Em/(1-nu^2)*{{1,nu,0},{nu,1,0},{0,0,(1-nu)/2}};
For [i=2,i<=5,i++, p={1,-3,3,6,7}[[i]];
  Ke=Trig6IsoPMembraneStiffness[ncoor,{Emat,0,0},{h},{True,p}];
  Ke=Chop[Simplify[Ke]];
  Print["Ke=",SetPrecision[Ke,4]//MatrixForm];
  Print["Eigenvalues of Ke=",Chop[Eigenvalues[N[Ke]],.0000001]]];
];

```

This is straightforward except perhaps for the `SetPrecision[Ke,4]` statement in the `Print`. This specifies that the stiffness matrix entries be printed to only 4 places to save space (the default is otherwise 6 places).

For rule=3:

$$\begin{bmatrix} 344.7 & 75.00 & -91.80 & 21.00 & -86.60 & -24.00 & -124.7 & -72.00 & -20.78 & -36.00 & -20.78 & 36.00 \\ 75.00 & 258.1 & -21.00 & -84.87 & 18.00 & -90.07 & 96.00 & 0 & -36.00 & 20.78 & -132.0 & -103.9 \\ -91.80 & -21.00 & 344.7 & -75.00 & -86.60 & 24.00 & -124.7 & 72.00 & -20.78 & -36.00 & -20.78 & 36.00 \\ 21.00 & -84.87 & -75.00 & 258.1 & -18.00 & -90.07 & -96.00 & 0 & 132.0 & -103.9 & 36.00 & 20.78 \\ -86.60 & 18.00 & -86.60 & -18.00 & 214.8 & 0 & 41.57 & 0 & -41.57 & 144.0 & -41.57 & -144.0 \\ -24.00 & -90.07 & 24.00 & -90.07 & 0 & 388.0 & 0 & -41.57 & -24.00 & -83.14 & 24.00 & -83.14 \\ -124.7 & 96.00 & -124.7 & -96.00 & 41.57 & 0 & 374.1 & 0 & -83.14 & -72.00 & -83.14 & 72.00 \\ -72.00 & 0 & 72.00 & 0 & 0 & -41.57 & 0 & 374.1 & -72.00 & -166.3 & 72.00 & -166.3 \\ -20.78 & -36.00 & -20.78 & 132.0 & -41.57 & -24.00 & -83.14 & -72.00 & 374.1 & 0 & -207.8 & 0 \\ -36.00 & 20.78 & -36.00 & -103.9 & 144.0 & -83.14 & -72.00 & -166.3 & 0 & 374.1 & 0 & -41.57 \\ -20.78 & -132.0 & -20.78 & 36.00 & -41.57 & 24.00 & -83.14 & 72.00 & -207.8 & 0 & 374.1 & 0 \\ 36.00 & -103.9 & 36.00 & 20.78 & -144.0 & -83.14 & 72.00 & -166.3 & 0 & -41.57 & 0 & 374.1 \end{bmatrix} \quad (24.36)$$

For rule=-3:

$$\begin{bmatrix} 566.4 & 139.0 & 129.9 & 21.00 & 79.67 & 8.000 & -364.9 & -104.0 & -205.5 & -36.00 & -205.5 & -28.00 \\ 139.0 & 405.9 & -21.00 & 62.93 & 50.00 & 113.2 & 64.00 & -129.3 & -36.00 & -164.0 & -196.0 & -288.7 \\ 129.9 & -21.00 & 566.4 & -139.0 & 79.67 & -8.000 & -364.9 & 104.0 & -205.5 & 28.00 & -205.5 & 36.00 \\ 21.00 & 62.93 & -139.0 & 405.9 & -50.00 & 113.2 & -64.00 & -129.3 & 196.0 & -288.7 & 36.00 & -164.0 \\ 79.67 & 50.00 & 79.67 & -50.00 & 325.6 & 0 & -143.2 & 0 & -170.9 & 176.0 & -170.9 & -176.0 \\ 8.000 & 113.2 & -8.000 & 113.2 & 0 & 646.6 & 0 & -226.3 & 8.000 & -323.3 & -8.000 & -323.3 \\ -364.9 & 64.00 & -364.9 & -64.00 & -143.2 & 0 & 632.8 & 0 & 120.1 & -104.0 & 120.1 & 104.0 \\ -104.0 & -129.3 & 104.0 & -129.3 & 0 & -226.3 & 0 & 485.0 & -104.0 & 0 & 104.0 & 0 \\ -205.5 & -36.00 & -205.5 & 196.0 & -170.9 & 8.000 & 120.1 & -104.0 & 521.9 & -64.00 & -60.04 & 0 \\ -36.00 & -164.0 & 28.00 & -288.7 & 176.0 & -323.3 & -104.0 & 0 & -64.00 & 595.8 & 0 & 180.1 \\ -205.5 & -196.0 & -205.5 & 36.00 & -170.9 & -8.000 & 120.1 & 104.0 & -60.04 & 0 & 521.9 & 64.00 \\ -28.00 & -288.7 & 36.00 & -164.0 & -176.0 & -323.3 & 104.0 & 0 & 0 & 180.1 & 64.00 & 595.8 \end{bmatrix} \quad (24.37)$$

The stiffness for rule=6 is omitted to save space. For rule=7:

$$\begin{bmatrix} 661.9 & 158.5 & 141.7 & 21.00 & 92.53 & 7.407 & -432.1 & -117.2 & -190.2 & -29.10 & -273.8 & -40.61 \\ 158.5 & 478.8 & -21.00 & 76.13 & 49.41 & 125.3 & 50.79 & -182.7 & -29.10 & -156.6 & -208.6 & -341.0 \\ 141.7 & -21.00 & 661.9 & -158.5 & 92.53 & -7.407 & -432.1 & 117.2 & -273.8 & 40.61 & -190.2 & 29.10 \\ 21.00 & 76.13 & -158.5 & 478.8 & -49.41 & 125.3 & -50.79 & -182.7 & 208.6 & -341.0 & 29.10 & -156.6 \\ 92.53 & 49.41 & 92.53 & -49.41 & 387.3 & 0 & -139.8 & 0 & -216.3 & 175.4 & -216.3 & -175.4 \\ 7.407 & 125.3 & -7.407 & 125.3 & 0 & 753.4 & 0 & -207.0 & 7.407 & -398.5 & -7.407 & -398.5 \\ -432.1 & 50.79 & -432.1 & -50.79 & -139.8 & 0 & 723.6 & 0 & 140.2 & -117.2 & 140.2 & 117.2 \\ -117.2 & -182.7 & 117.2 & -182.7 & 0 & -207.0 & 0 & 562.6 & -117.2 & 4.884 & 117.2 & 4.884 \\ -190.2 & -29.10 & -273.8 & 208.6 & -216.3 & 7.407 & 140.2 & -117.2 & 602.8 & -69.71 & -62.78 & 0 \\ -29.10 & -156.6 & 40.61 & -341.0 & 175.4 & -398.5 & -117.2 & 4.884 & -69.71 & 683.3 & 0 & 207.9 \\ -273.8 & -208.6 & -190.2 & 29.10 & -216.3 & -7.407 & 140.2 & 117.2 & -62.78 & 0 & 602.8 & 69.71 \\ -40.61 & -341.0 & 29.10 & -156.6 & -175.4 & -398.5 & 117.2 & 4.884 & 0 & 207.9 & 69.71 & 683.3 \end{bmatrix} \quad (24.38)$$

The eigenvalues of these matrices are:

Rule	Eigenvalues of $\mathbf{K}^{(e)}$										
3	702.83	665.11	553.472	553.472	481.89	429.721	429.721	118.391	118.391	0	0
-3	1489.80	1489.80	702.833	665.108	523.866	523.866	481.890	196.429	196.429	0	0
6	1775.53	1775.53	896.833	768.948	533.970	533.970	495.570	321.181	321.181	0	0
7	1727.11	1727.11	880.958	760.719	532.750	532.750	494.987	312.123	312.123	0	0

(24.39)

Since the metric of this element is very distorted near its boundary, the stiffness matrix entries and eigenvalues

change substantially as the integration formulas are advanced from 3 to 6 and 7 points. However as can be seen the matrix remains rank-sufficient.

§24.5. *THE CUBIC TRIANGLE

The 10-node cubic triangle, depicted in Figure 24.8, is rarely used in practice as such because of the difficulty of combining it with other elements. Nevertheless the derivation of the element modules is instructive as this triangle has other and more productive uses as a generator of more practical elements with “drilling” rotational degrees of freedom at corners for modeling shells. Such transformations are studied in advanced FEM courses.

The geometry of the triangle is defined by the coordinates of the ten nodes. One change in node labeling should be noted: the interior node is relabeled as 0 instead of 10 (cf. Figure 24.8) to avoid confusion with notation such as $x_{12} = x_1 - x_2$ for coordinate differences.

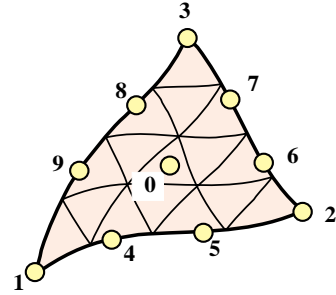


Figure 24.8. The 10-node cubic triangle.

§24.5.1. *Shape Function Module

The shape functions were obtained in Chapter 18. They are reproduced here along with their natural derivatives:

$$\mathbf{N} = \begin{bmatrix} \frac{1}{2}\zeta_1(3\zeta_1-1)(3\zeta_1-2) \\ \frac{1}{2}\zeta_2(3\zeta_2-1)(3\zeta_2-2) \\ \frac{1}{2}\zeta_3(3\zeta_3-1)(3\zeta_3-2) \\ \frac{9}{2}\zeta_1\zeta_2(3\zeta_1-1) \\ \frac{9}{2}\zeta_1\zeta_2(3\zeta_2-1) \\ \frac{9}{2}\zeta_2\zeta_3(3\zeta_2-1) \\ \frac{9}{2}\zeta_2\zeta_3(3\zeta_3-1) \\ \frac{9}{2}\zeta_3\zeta_1(3\zeta_3-1) \\ \frac{9}{2}\zeta_3\zeta_1(3\zeta_1-1) \\ 27\zeta_1\zeta_2\zeta_3 \end{bmatrix}, \quad \frac{\partial \mathbf{N}}{\partial \zeta_1} = \frac{9}{2} \begin{bmatrix} \frac{2}{9}-2\zeta_1+3\zeta_1^2 \\ 0 \\ 0 \\ \zeta_2(6\zeta_1-1) \\ \zeta_2(3\zeta_2-1) \\ 0 \\ 0 \\ \zeta_3(3\zeta_3-1) \\ \zeta_3(6\zeta_1-1) \\ 6\zeta_2\zeta_3 \end{bmatrix}, \quad \frac{\partial \mathbf{N}}{\partial \zeta_2} = \frac{9}{2} \begin{bmatrix} 0 \\ \frac{2}{9}-2\zeta_2+3\zeta_2^2 \\ 0 \\ \zeta_1(3\zeta_1-1) \\ \zeta_1(6\zeta_2-1) \\ \zeta_3(6\zeta_2-1) \\ \zeta_3(3\zeta_3-1) \\ 0 \\ 0 \\ 6\zeta_1\zeta_3 \end{bmatrix}, \quad \frac{\partial \mathbf{N}}{\partial \zeta_3} = \frac{9}{2} \begin{bmatrix} 0 \\ 0 \\ \frac{2}{9}-2\zeta_3+3\zeta_3^2 \\ 0 \\ 0 \\ \zeta_2(3\zeta_2-1) \\ \zeta_2(6\zeta_3-1) \\ \zeta_1(6\zeta_3-1) \\ \zeta_1(3\zeta_1-1) \\ 6\zeta_1\zeta_2 \end{bmatrix}. \quad (24.40)$$

As in the case of the 6-node triangle it is convenient to introduce the deviations from thirdpoints and centroid: $\Delta x_4 = x_4 - \frac{1}{3}(2x_1+x_2)$, $\Delta x_5 = x_5 - \frac{1}{3}(x_1+2x_2)$, $\Delta x_6 = x_6 - \frac{1}{3}(2x_2+x_3)$, $\Delta x_7 = x_7 - \frac{1}{3}(x_2+2x_3)$, $\Delta x_8 = x_8 - \frac{1}{3}(2x_3+x_1)$, $\Delta x_9 = x_9 - \frac{1}{3}(x_3+2x_1)$, $\Delta x_0 = x_{10} - \frac{1}{3}(x_1+x_2+x_3)$, $\Delta y_4 = y_4 - \frac{1}{3}(2y_1+y_2)$, $\Delta y_5 = y_5 - \frac{1}{3}(y_1+2y_2)$, $\Delta y_6 = y_6 - \frac{1}{3}(2y_2+y_3)$, $\Delta y_7 = y_7 - \frac{1}{3}(y_2+2y_3)$, $\Delta y_8 = y_8 - \frac{1}{3}(2y_3+y_1)$, $\Delta y_9 = y_9 - \frac{1}{3}(y_3+2y_1)$, and $\Delta y_0 = y_{10} - \frac{1}{3}(y_1+y_2+y_3)$. Using *Mathematica* the expressions of the Jacobian coefficients are found to be

$$\begin{aligned} J_{x21} &= x_{21} + \frac{9}{2} [\Delta x_4 (\zeta_1(3\zeta_1-6\zeta_2-1)+\zeta_2) + \Delta x_5 (\zeta_2(1-3\zeta_2+6\zeta_1)-\zeta_1) + \Delta x_6 \zeta_3(6\zeta_2-1) \\ &\quad + \Delta x_7 \zeta_3(3\zeta_3-1) + \Delta x_8 \zeta_3(1-3\zeta_3) + \Delta x_9 \zeta_3(1-6\zeta_1) + 6\Delta x_0 \zeta_3(\zeta_1-\zeta_2)], \\ J_{x32} &= x_{32} + \frac{9}{2} [\Delta x_4 \zeta_1(1-3\zeta_1) + \Delta x_5 \zeta_1(1-6\zeta_2) + \Delta x_6 (\zeta_2(3\zeta_2-6\zeta_3-1)+\zeta_3) \\ &\quad + \Delta x_7 (\zeta_3(1-3\zeta_3+6\zeta_2)-\zeta_2) + \Delta x_8 \zeta_1(6\zeta_3-1) + \Delta x_9 \zeta_1(3\zeta_1-1) + 6\Delta x_0 \zeta_1(\zeta_2-\zeta_3)], \end{aligned} \quad (24.41)$$

$$\begin{aligned} J_{x13} &= x_{13} + \frac{9}{2} [\Delta x_4 (6\zeta_1-1)\zeta_2 + \Delta x_5 \zeta_2(3\zeta_2-1) + \Delta x_6 \zeta_2(1-3\zeta_2) + \Delta x_7 \zeta_2(1-6\zeta_3) \\ &\quad + \Delta x_8 (\zeta_3(3\zeta_3-6\zeta_1-1)+\zeta_1) + \Delta x_9 (\zeta_1(1-3\zeta_1+6\zeta_3)-\zeta_3) + 6\Delta x_0 \zeta_2(\zeta_3-\zeta_1)], \end{aligned}$$

$$\begin{aligned} J_{y12} &= y_{12} - \frac{9}{2} [\Delta y_4 (\zeta_1(3\zeta_1-6\zeta_2-1)+\zeta_2) + \Delta y_5 (\zeta_2(1-3\zeta_2+6\zeta_1)-\zeta_1) + \Delta y_6 \zeta_3(6\zeta_2-1) \\ &\quad + \Delta y_7 \zeta_3(3\zeta_3-1) + \Delta y_8 \zeta_3(1-3\zeta_3) + \Delta y_9 \zeta_3(1-6\zeta_1) + 6\Delta y_0 \zeta_3(\zeta_1-\zeta_2)], \end{aligned}$$

$$\begin{aligned} J_{y23} &= y_{23} - \frac{9}{2} [\Delta y_4 \zeta_1(1-3\zeta_1) + \Delta y_5 \zeta_1(1-6\zeta_2) + \Delta y_6 (\zeta_2(3\zeta_2-6\zeta_3-1)+\zeta_3) \\ &\quad + \Delta y_7 (\zeta_3(1-3\zeta_3+6\zeta_2)-\zeta_2) + \Delta y_8 \zeta_1(6\zeta_3-1) + \Delta y_9 \zeta_1(3\zeta_1-1) + 6\Delta y_0 \zeta_1(\zeta_2-\zeta_3)], \end{aligned} \quad (24.42)$$

$$\begin{aligned} J_{y31} &= y_{31} - \frac{9}{2} [\Delta y_4 (6\zeta_1-1)\zeta_2 + \Delta y_5 \zeta_2(3\zeta_2-1) + \Delta y_6 \zeta_2(1-3\zeta_2) + \Delta y_7 \zeta_2(1-6\zeta_3) \\ &\quad + \Delta y_8 (\zeta_3(3\zeta_3-6\zeta_1-1)+\zeta_1) + \Delta y_9 (\zeta_1(1-3\zeta_1+6\zeta_3)-\zeta_3) + 6\Delta y_0 \zeta_2(\zeta_3-\zeta_1)]. \end{aligned}$$

```

Trig10IsoPShapeFunDer[ncoor_,tcoor_]:= Module[
{ξ1,ξ2,ξ3,x1,x2,x3,x4,x5,x6,x7,x8,x9,x0,y1,y2,y3,y4,y5,y6,y7,y8,y9,y0,
dx4,dx5,dx6,dx7,dx8,dx9,dx0,dy4,dy5,dy6,dy7,dy8,dy9,dy0,
Jx21,Jx32,Jx13,Jy12,Jy23,Jy31,Nf,dNx,dNy,Jdet},
{{x1,y1},{x2,y2},{x3,y3},{x4,y4},{x5,y5},{x6,y6},{x7,y7},
{x8,y8},{x9,y9},{x0,y0}}=ncoor; {ξ1,ξ2,ξ3}=tcoor;
dx4=x4-(2*x1+x2)/3; dx5=x5-(x1+2*x2)/3; dx6=x6-(2*x2+x3)/3;
dx7=x7-(x2+2*x3)/3; dx8=x8-(2*x3+x1)/3; dx9=x9-(x3+2*x1)/3;
dy4=y4-(2*y1+y2)/3; dy5=y5-(y1+2*y2)/3; dy6=y6-(2*y2+y3)/3;
dy7=y7-(y2+2*y3)/3; dy8=y8-(2*y3+y1)/3; dy9=y9-(y3+2*y1)/3;
dx0=x0-(x1+x2+x3)/3; dy0=y0-(y1+y2+y3)/3;
Nf={ξ1*(3*ξ1-1)*(3*ξ1-2),ξ2*(3*ξ2-1)*(3*ξ2-2),ξ3*(3*ξ3-1)*(3*ξ3-2),
9*ξ1*ξ2*(3*ξ1-1),9*ξ1*ξ2*(3*ξ2-1),9*ξ2*ξ3*(3*ξ2-1),
9*ξ2*ξ3*(3*ξ3-1),9*ξ3*ξ1*(3*ξ3-1),9*ξ3*ξ1*(3*ξ1-1),54*ξ1*ξ2*ξ3}/2;
Jx21=x2-x1+(9/2)*(dx4*(ξ1*(3*ξ1-6*ξ2-1)+ξ2)+
dx5*(ξ2*(1-3*ξ2+6*ξ1)-ξ1)+dx6*ξ3*(6*ξ2-1)+dx7*ξ3*(3*ξ3-1)+
dx8*ξ3*(1-3*ξ3)+dx9*ξ3*(1-6*ξ1)+6*dx0*ξ3*(ξ1-ξ2));
Jx32=x3-x2+(9/2)*(dx4*ξ1*(1-3*ξ1)+dx5*ξ1*(1-6*ξ2)+
dx6*(ξ2*(3*ξ2-6*ξ3-1)+ξ3)+dx7*(ξ3*(1-3*ξ3+6*ξ2)-ξ2)+
dx8*ξ1*(6*ξ3-1)+dx9*ξ1*(3*ξ1-1)+6*dx0*ξ1*(ξ2-ξ3));
Jx13=x1-x3+(9/2)*(dx4*(6*ξ1-1)*ξ2+dx5*ξ2*(3*ξ2-1)+
dx6*ξ2*(1-3*ξ2)+dx7*ξ2*(1-6*ξ3)+dx8*(ξ3*(3*ξ3-6*ξ1-1)+ξ1)+
dx9*(ξ1*(1-3*ξ1+6*ξ3)-ξ3)+6*dx0*ξ2*(ξ3-ξ1));
Jy12=y1-y2-(9/2)*(dy4*(ξ1*(3*ξ1-6*ξ2-1)+ξ2)+
dy5*(ξ2*(1-3*ξ2+6*ξ1)-ξ1)+dy6*ξ3*(6*ξ2-1)+dy7*ξ3*(3*ξ3-1)+
dy8*ξ3*(1-3*ξ3)+dy9*ξ3*(1-6*ξ1)+6*dy0*ξ3*(ξ1-ξ2));
Jy23=y2-y3-(9/2)*(dy4*ξ1*(1-3*ξ1)+dy5*ξ1*(1-6*ξ2)+
dy6*(ξ2*(3*ξ2-6*ξ3-1)+ξ3)+dy7*(ξ3*(1-3*ξ3+6*ξ2)-ξ2)+
dy8*ξ1*(6*ξ3-1)+dy9*ξ1*(3*ξ1-1)+6*dy0*ξ1*(ξ2-ξ3));
Jy31=y3-y1-(9/2)*(dy4*(6*ξ1-1)*ξ2+dy5*ξ2*(3*ξ2-1)+
dy6*ξ2*(1-3*ξ2)+dy7*ξ2*(1-6*ξ3)+dy8*(ξ3*(3*ξ3-6*ξ1-1)+ξ1)+
dy9*(ξ1*(1-3*ξ1+6*ξ3)-ξ3)+6*dy0*ξ2*(ξ3-ξ1));
Jdet= Jx21*Jy31-Jy12*Jx13;
dNx={Jy23*(2/9-2*ξ1+3*ξ1^2),Jy31*(2/9-2*ξ2+3*ξ2^2),Jy12*(2/9-2*ξ3+3*ξ3^2),
Jy31*ξ1*(3*ξ1-1)+Jy23*ξ2*(6*ξ1-1),Jy23*ξ2*(3*ξ2-1)+Jy31*ξ1*(6*ξ2-1),
Jy12*ξ2*(3*ξ2-1)+Jy31*ξ3*(6*ξ2-1),Jy31*ξ3*(3*ξ3-1)+Jy12*ξ2*(6*ξ3-1),
Jy23*ξ3*(3*ξ3-1)+Jy12*ξ1*(6*ξ3-1),Jy12*ξ1*(3*ξ1-1)+Jy23*ξ3*(6*ξ1-1),
6*(Jy12*ξ1*ξ2+Jy31*ξ1*ξ3+Jy23*ξ2*ξ3)}/(2*Jdet/9);
dNy={Jx32*(2/9-2*ξ1+3*ξ1^2),Jx13*(2/9-2*ξ2+3*ξ2^2),Jx21*(2/9-2*ξ3+3*ξ3^2),
Jx13*ξ1*(3*ξ1-1)+Jx32*ξ2*(6*ξ1-1),Jx32*ξ2*(3*ξ2-1)+Jx13*ξ1*(6*ξ2-1),
Jx21*ξ2*(3*ξ2-1)+Jx13*ξ3*(6*ξ2-1),Jx13*ξ3*(3*ξ3-1)+Jx21*ξ2*(6*ξ3-1),
Jx32*ξ3*(3*ξ3-1)+Jx21*ξ1*(6*ξ3-1),Jx21*ξ1*(3*ξ1-1)+Jx32*ξ3*(6*ξ1-1),
6*(Jx21*ξ1*ξ2+Jx13*ξ1*ξ3+Jx32*ξ2*ξ3)}/(2*Jdet/9);
Return[Simplify[{Nf,dNx,dNy,Jdet}]]
];

```

Figure 24.9. Shape function module for 10-node cubic triangle.

```

Trig10IsoPMembraneStiffness[ncoor_,mprop_,fprop_,opt_]:=
Module[{i,k,l,p=6,numer=False,Emat,th={fprop},h,tcoor,w,c,
Nf,dNx,dNy,Jdet,B,Ke=Table[0,{20},{20}]},
Emat=mprop[[1]]; If [Length[fprop]>0, th=fprop[[1]]];
If [Length[opt]>0, numer=opt[[1]]];
If [Length[opt]>1, p= opt[[2]]];
If [p!=1&&p!=-3&&p!=3&&p!=6&&p!=7, Print["Illegal p"];Return[Null]];
For [k=1, k<=Abs[p], k++,
{tcoor,w}= TrigGaussRuleInfo[{p,numer},k];
{Nf,dNx,dNy,Jdet}= Trig10IsoPShapeFunDer[ncoor,tcoor];
If [Length[th]==0, h=th, h=th.Nf]; c=w*Jdet*h/2;
B= {Flatten[Table[{dNx[[i]],0},{i,10}]],
Flatten[Table[{0,dNy[[i]]},{i,10}]],
Flatten[Table[{dNy[[i]],dNx[[i]]},{i,10}]]};
Ke+=c*Transpose[B].(Emat.B);
]; If [!numer,Ke=Simplify[Ke]]; Return[Ke]
];

```

Figure 24.10. Stiffness module for 10-node cubic triangle.

The Jacobian determinant is $J = \frac{1}{2}(J_{x21}J_{y31} - J_{y12}J_{x13})$. Using (24.23) the shape function partial derivatives can be expressed as

$$\frac{\partial \mathbf{N}}{\partial x} = \frac{9}{4J} \begin{bmatrix} J_{y23}(\frac{2}{9} - 2\zeta_1 + 3\zeta_1^2/2) \\ J_{y31}(\frac{2}{9} - 2\zeta_2 + 3\zeta_2^2/2) \\ J_{y12}(\frac{2}{9} - 2\zeta_3 + 3\zeta_3^2/2) \\ J_{y31}\zeta_1(3\zeta_1 - 1) + J_{y23}\zeta_2(6\zeta_1 - 1) \\ J_{y23}\zeta_2(3\zeta_2 - 1) + J_{y31}\zeta_1(6\zeta_2 - 1) \\ J_{y12}\zeta_2(3\zeta_2 - 1) + J_{y31}\zeta_3(6\zeta_2 - 1) \\ J_{y31}\zeta_3(3\zeta_3 - 1) + J_{y12}\zeta_2(6\zeta_3 - 1) \\ J_{y23}\zeta_3(3\zeta_3 - 1) + J_{y12}\zeta_1(6\zeta_3 - 1) \\ J_{y12}\zeta_1(3\zeta_1 - 1) + J_{y23}\zeta_3(6\zeta_1 - 1) \\ 6(J_{y12}\zeta_1\zeta_2 + J_{y31}\zeta_1\zeta_3 + J_{y23}\zeta_2\zeta_3) \end{bmatrix}, \quad \frac{\partial \mathbf{N}}{\partial y} = \frac{9}{4J} \begin{bmatrix} J_{x32}(\frac{2}{9} - 2\zeta_1 + 3\zeta_1^2/2) \\ J_{x13}(\frac{2}{9} - 2\zeta_2 + 3\zeta_2^2/2) \\ J_{x21}(\frac{2}{9} - 2\zeta_3 + 3\zeta_3^2/2) \\ J_{x13}\zeta_1(3\zeta_1 - 1) + J_{x32}\zeta_2(6\zeta_1 - 1) \\ J_{x32}\zeta_2(3\zeta_2 - 1) + J_{x13}\zeta_1(6\zeta_2 - 1) \\ J_{x21}\zeta_2(3\zeta_2 - 1) + J_{x13}\zeta_3(6\zeta_2 - 1) \\ J_{x13}\zeta_3(3\zeta_3 - 1) + J_{x21}\zeta_2(6\zeta_3 - 1) \\ J_{x32}\zeta_3(3\zeta_3 - 1) + J_{x21}\zeta_1(6\zeta_3 - 1) \\ J_{x21}\zeta_1(3\zeta_1 - 1) + J_{x32}\zeta_3(6\zeta_1 - 1) \\ 6(J_{x21}\zeta_1\zeta_2 + J_{x13}\zeta_1\zeta_3 + J_{x32}\zeta_2\zeta_3) \end{bmatrix}. \quad (24.43)$$

The shape function module `Trig10IsoPShapeFunDer` that implements these expressions is listed in Figure 24.9. This has the same arguments as `Trig6IsoPShapeFunDer`. As there `Jdet` denotes $2J$.

§24.5.2. *Stiffness Module

Module `Trig10IsoPMembraneStiffness`, listed in Figure 24.10, implements the computation of the element stiffness matrix of the 10-node cubic plane stress triangle. The arguments of the module are

- `ncoor` Node coordinates arranged in two-dimensional list form:
 $\{\{x_1, y_1\}, \{x_2, y_2\}, \{x_3, y_3\}, \dots, \{x_9, y_9\}, \{x_0, y_0\}\}$.
- `mprop` Same as for `Trig6IsoPMembraneStiffness`.
- `fprop` Fabrication properties. The plate thickness specified as a one-entry list: $\{h\}$, as a ten-entry list: $\{h_1, h_2, h_3, h_4, \dots, h_9, h_0\}$, or as an empty list: $\{\}$.
 The one-entry form specifies uniform thickness h . The ten-entry form is used to specify an element of variable thickness, in which case the entries are the ten nodal thicknesses and h is interpolated cubically. If an empty list appears the module assumes a uniform unit thickness.
- `options` Processing options. This list may contain two items: $\{\text{numer}, \text{rule}\}$ or one: $\{\text{numer}\}$.
 numer is a flag with value `True` or `False`. If `True`, the computations are forced to go in floating point arithmetic. For symbolic or exact arithmetic work set numer to `False`.
 rule specifies the triangle Gauss rule as described in §24.2.4. rule may be 1, 3, -3, 6 or 7. For this element both the 6-point and 7-point rules are sufficient to produce the correct rank. If omitted $\text{rule} = 6$ is assumed. See Remark below.

The module returns \mathbf{K}_e as an 20×20 symmetric matrix pertaining to the following arrangement of nodal displacements:

$$\mathbf{u}^{(e)} = [u_{x1} \ u_{y1} \ u_{x2} \ u_{y2} \ u_{x3} \ u_{y3} \ u_{x4} \ u_{y4} \ \dots \ u_{x9} \ u_{y9} \ u_{x0} \ u_{y0}]^T. \quad (24.44)$$

REMARK 24.4

For symbolic work with this element the 7-point rule should be preferred because the exact expressions of the abscissas and weights at sample points are much simpler than for the 6-point rule, as can be observed in Figure 24.3. This greatly speeds up algebraic simplification. For numerical work the 6-point rule is slightly faster.

§24.5.3. *Test Element

The stiffness module is tested on the superparametric triangle geometry shown in Figure 24.11, already used for the quadratic triangle. The corner nodes are at $(0, 0)$, $(4, 2)$ and $(6, 4)$, with the side nodes at the thirdpoints of the sides and the interior node at the centroid. The following test statements form $\mathbf{K}^{(e)}$ for isotropic material with $E = 1920$, $\nu = 0$ and $h = 1$, using exact arithmetic:

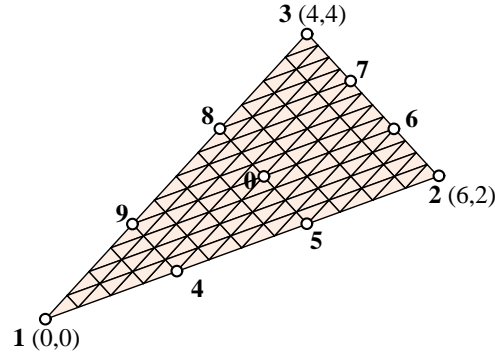


Figure 24.11. Test 10-node triangle element geometry.

```

ClearAll[Em,nu,a,b,e,h]; h=1; Em=1920; nu=0;
ncoor={{0,0},{6,2},{4,4}};
x4=(2*x1+x2)/3; x5=(x1+2*x2)/3; y4=(2*y1+y2)/3; y5=(y1+2*y2)/3;
x6=(2*x2+x3)/3; x7=(x2+2*x3)/3; y6=(2*y2+y3)/3; y7=(y2+2*y3)/3;
x8=(2*x3+x1)/3; x9=(x3+2*x1)/3; y8=(2*y3+y1)/3; y9=(y3+2*y1)/3;
x0=(x1+x2+x3)/3; y0=(y1+y2+y3)/3;
ncoor= {{x1,y1},{x2,y2},{x3,y3},{x4,y4},{x5,y5},{x6,y6},{x7,y7},
        {x8,y8},{x9,y9},{x0,y0}};
Emat=Em/(1-nu^2)*{{1,nu,0},{nu,1,0},{0,0,(1-nu)/2}};
Ke=Trig10IsoPMembraneStiffness[ncoor,{Emat,0,0},{h},{False,7}];
Ke=Simplify[Ke]; Print[Ke//MatrixForm]; ev=Chop[Eigenvalues[N[Ke]]];
Print["eigs of Ke=",SetPrecision[ev,5]]

```

The returned stiffness matrix using either 6- or 7-point integration is the same, since for a superparametric element the integrand is quartic in the triangular coordinates. For the given combination of inputs the entries are exact integers:

$$\mathbf{K}^{(e)} = \begin{bmatrix}
 204 & 102 & -42 & 0 & 0 & -21 & -324 & 9 & 162 & 9 \\
 102 & 204 & 0 & 42 & -21 & -84 & 9 & 360 & 9 & -126 \\
 -42 & 0 & 816 & -408 & -126 & 84 & 216 & -36 & -270 & -36 \\
 0 & 42 & -408 & 816 & 84 & -210 & -36 & -72 & -36 & 414 \\
 0 & -21 & -126 & 84 & 612 & -306 & -54 & 27 & -54 & 27 \\
 -21 & -84 & 84 & -210 & -306 & 1428 & 27 & -126 & 27 & -126 \\
 -324 & 9 & 216 & -36 & -54 & 27 & 3240 & -1215 & -1134 & 243 \\
 9 & 360 & -36 & -72 & 27 & -126 & -1215 & 4860 & 243 & -486 \\
 162 & 9 & -270 & -36 & -54 & 27 & -1134 & 243 & 3240 & -1215 \\
 9 & -126 & -36 & 414 & 27 & -126 & 243 & -486 & -1215 & 4860 \\
 -18 & -9 & -954 & 648 & 486 & -315 & 0 & 81 & 0 & -405 \\
 -9 & -18 & 648 & -1638 & -315 & 846 & 81 & 324 & -405 & -1620 \\
 -18 & -9 & 504 & -324 & -972 & 657 & 0 & 81 & 0 & 81 \\
 -9 & -18 & -324 & 792 & 657 & -1584 & 81 & 324 & 81 & 324 \\
 18 & 81 & -72 & 36 & 54 & -198 & 486 & -324 & 486 & -324 \\
 81 & 306 & 36 & -72 & -198 & -558 & -324 & 810 & -324 & 810 \\
 18 & -162 & -72 & 36 & 54 & 45 & -2430 & 1620 & 486 & -324 \\
 -162 & -666 & 36 & -72 & 45 & 414 & 1620 & -4050 & -324 & 810 \\
 0 & 0 & 0 & 0 & 0 & 0 & 0 & -486 & -2916 & 1944 \\
 0 & 0 & 0 & 0 & 0 & 0 & -486 & -1944 & 1944 & -4860
 \end{bmatrix}$$

$$\begin{bmatrix}
 -18 & -9 & -18 & -9 & 18 & 81 & 18 & -162 & 0 & 0 \\
 -9 & -18 & -9 & -18 & 81 & 306 & -162 & -666 & 0 & 0 \\
 -954 & 648 & 504 & -324 & -72 & 36 & -72 & 36 & 0 & 0 \\
 648 & -1638 & -324 & 792 & 36 & -72 & 36 & -72 & 0 & 0 \\
 486 & -315 & -972 & 657 & 54 & -198 & 54 & 45 & 0 & 0 \\
 -315 & 846 & 657 & -1584 & -198 & -558 & 45 & 414 & 0 & 0 \\
 0 & 81 & 0 & 81 & 486 & -324 & -2430 & 1620 & 0 & -486 \\
 81 & 324 & 81 & 324 & -324 & 810 & 1620 & -4050 & -486 & -1944 \\
 0 & -405 & 0 & 81 & 486 & -324 & 486 & -324 & -2916 & 1944 \\
 -405 & -1620 & 81 & 324 & -324 & 810 & -324 & 810 & 1944 & -4860 \\
 3240 & -1215 & -2106 & 1215 & 162 & 0 & 162 & 0 & -972 & 0 \\
 -1215 & 4860 & 1215 & -3402 & 0 & -162 & 0 & -162 & 0 & 972 \\
 -2106 & 1215 & 3240 & -1215 & -810 & 0 & 162 & 0 & 0 & -486 \\
 1215 & -3402 & -1215 & 4860 & 0 & 810 & 0 & -162 & -486 & -1944 \\
 162 & 0 & -810 & 0 & 3240 & -1215 & -648 & 0 & -2916 & 1944 \\
 0 & -162 & 0 & 810 & -1215 & 4860 & 0 & -1944 & 1944 & -4860 \\
 162 & 0 & 162 & 0 & -648 & 0 & 3240 & -1215 & -972 & 0 \\
 0 & -162 & 0 & -162 & 0 & -1944 & -1215 & 4860 & 0 & 972 \\
 -972 & 0 & 0 & -486 & -2916 & 1944 & -972 & 0 & 7776 & -2916 \\
 0 & 972 & -486 & -1944 & 1944 & -4860 & 0 & 972 & -2916 & 11664
 \end{bmatrix} \quad (24.45)$$

The eigenvalues are

$$\begin{aligned}
 & [26397. \ 16597. \ 14937. \ 12285. \ 8900.7 \ 7626.2 \ 5417.8 \ 4088.8 \ 3466.8 \ 3046.4 \\
 & \ 1751.3 \ 1721.4 \ 797.70 \ 551.82 \ 313.22 \ 254.00 \ 28.019 \ 0 \ 0 \ 0] \quad (24.46)
 \end{aligned}$$

The 3 zero eigenvalues pertain to the three independent rigid-body modes. The 17 other ones are positive. Consequently the computed $\mathbf{K}^{(e)}$ has the correct rank of 17.

Notes and Bibliography

The 3-node, 6-node and 10-node plane stress triangular elements are generated by complete polynomials in two-dimensions. In order of historical appearance:

1. The three-node linear triangle, also known as *Constant Strain Triangle* (CST) and Turner triangle, was developed as plane stress element by Jon Turner, Ray Clough and Harold Martin in 1952–53 [24.5]. It was published in 1956 [24.12].
2. The six-node quadratic triangle, also known as *Linear Strain Triangle* (LST) and Veubeke triangle, was developed by B. M. Fraeijs de Veubeke in 1962–63 [24.15]; published 1965 [24.7].
3. The ten-node cubic triangle, also known as *Quadratic Strain Triangle* (QST), was developed by the writer in 1965; published 1966 [24.6]. Shape functions for the cubic triangle were presented there but used for plate bending instead of plane stress.

A version of the cubic triangle with freedoms migrated to corners and recombined to produce a “drilling rotation” at corners, was used in static and dynamic shell analysis in Carr’s thesis under Ray Clough [24.3,24.4]. The drilling-freedom idea was independently exploited for rectangular and quadrilateral elements, respectively, in the theses of Abu-Ghazaleh [24.2] and Willam [24.13], both under Alex Scordelis. A variant of the Willam quadrilateral, developed by Bo Almroth at Lockheed, has survived in the nonlinear shell analysis code STAGS as element 410 [24.10].

Numerical integration came into FEM by the mid sixties. Five triangle integration rules were tabulated in the writer’s thesis [24.6, pp. 38–39]. These were gathered from three sources: two papers by Hammer and Stroud [24.8,24.9] and the 1964 Handbook of Mathematical Functions [24.1, §25.4]. They were adapted to FEM by converting Cartesian abscissas to triangle natural coordinates. The table has been reproduced in Zienkiewicz’ book since the second edition [24.14, Table 8.2] and, with corrections and additions, in the monograph of Strang and Fix [24.11, p. 184].

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Homework Exercises for Chapter 24

Implementation of Iso-P Triangular Elements

EXERCISE 24.1

[C:20] Write an element stiffness module and a shape function module for the 4-node “transition” iso-P triangular element with only one side node: 4, which is located between 1 and 2. See Figure E24.1. The shape functions are $N_1 = \zeta_1 - 2\zeta_1\zeta_2$, $N_2 = \zeta_2 - 2\zeta_1\zeta_2$, $N_3 = \zeta_3$ and $N_4 = 4\zeta_1\zeta_2$. Use 3 interior point integration rule=3. Test the element for the geometry of the triangle depicted on the left of Figure 24.7, removing nodes 5 and 6, and with $E = 2880$, $\nu = 1/3$ and $h = 1$. Report results for the stiffness matrix $\mathbf{K}^{(e)}$ and its 8 eigenvalues. Note: Notebook Trig6Stiffness.nb for the 6-node triangle, posted on the index of Chapter 24, may be used as “template” in support of this Exercise.

Partial results: $K_{11} = 1980$, $K_{18} = 1440$.

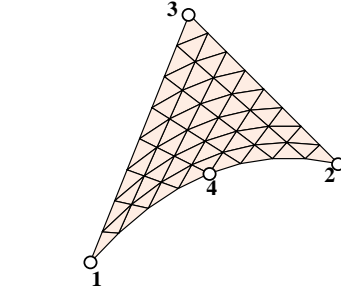


Figure E24.1. The 4-node transition triangle for Exercise 24.1.

EXERCISE 24.2

[A+C:20] Consider the superparametric straight-sided 6-node triangle where side nodes are located at the midpoints. By setting $x_4 = \frac{1}{2}(x_1 + x_2)$, $x_5 = \frac{1}{2}(x_2 + x_3)$, $x_6 = \frac{1}{2}(x_3 + x_1)$, $y_4 = \frac{1}{2}(y_1 + y_2)$, $y_5 = \frac{1}{2}(y_2 + y_3)$, $y_6 = \frac{1}{2}(y_3 + y_1)$ in (24.35), deduce that

$$\mathbf{J} = \begin{bmatrix} 1 & 1 & 1 \\ 3x_1 & 2x_1 + x_2 & 2x_1 + x_3 \\ 3y_1 & 2y_1 + y_2 & 2y_1 + y_3 \end{bmatrix} \zeta_1 + \begin{bmatrix} 1 & 1 & 1 \\ x_1 + 2x_2 & 3x_2 & 2x_2 + x_3 \\ y_1 + 2y_2 & 3y_2 & 2y_2 + y_3 \end{bmatrix} \zeta_2 + \begin{bmatrix} 1 & 1 & 1 \\ x_1 + 2x_3 & x_2 + 2x_3 & 3x_3 \\ y_1 + 2y_3 & y_2 + 2y_3 & 3y_3 \end{bmatrix} \zeta_3. \quad (\text{E24.1})$$

This contradicts publications that, by mistakenly assuming that the results for the linear triangle (Chapter 15) can be extended by analogy, take

$$\mathbf{J} = \begin{bmatrix} 1 & 1 & 1 \\ x_1 & x_2 & x_3 \\ y_1 & y_2 & y_3 \end{bmatrix} \quad (\text{E24.2})$$

Show, however, that

$$2J = 2A = \det \mathbf{J} = x_3 y_{12} + x_1 y_{23} + x_2 y_{31}, \quad \mathbf{P} = \frac{1}{2J} \begin{bmatrix} y_{23} & y_{32} \\ y_{31} & x_{13} \\ y_{12} & x_{21} \end{bmatrix} \quad (\text{E24.3})$$

are the same for both (E24.1) and (E24.2). Thus the mistake has no effect on the computation of derivatives.

EXERCISE 24.3

[A/C:15+15] Consider the superparametric straight-sided 6-node triangle where side nodes are at the midpoints of the sides and the thickness h is constant. Using the 3-midpoint quadrature rule show that the element stiffness can be expressed in a closed form obtained in 1966 [24.6]:

$$\mathbf{K}^{(e)} = \frac{1}{3} Ah (\mathbf{B}_1^T \mathbf{E} \mathbf{B}_1 + \mathbf{B}_2^T \mathbf{E} \mathbf{B}_2 + \mathbf{B}_3^T \mathbf{E} \mathbf{B}_3) \quad (\text{E24.4})$$

in which A is the triangle area and

$$\begin{aligned} \mathbf{B}_1 &= \frac{1}{2A} \begin{bmatrix} y_{32} & 0 & y_{31} & 0 & y_{12} & 0 & 2y_{23} & 0 & 2y_{32} & 0 & 2y_{23} & 0 \\ 0 & x_{23} & 0 & x_{13} & 0 & x_{21} & 0 & 2x_{32} & 0 & 2x_{23} & 0 & 2x_{32} \\ x_{23} & y_{32} & x_{13} & y_{31} & x_{21} & y_{12} & 2x_{32} & 2y_{23} & 2x_{23} & 2y_{32} & 2x_{32} & 2y_{23} \end{bmatrix} \\ \mathbf{B}_2 &= \frac{1}{2A} \begin{bmatrix} y_{23} & 0 & y_{13} & 0 & y_{12} & 0 & 2y_{31} & 0 & 2y_{31} & 0 & 2y_{13} & 0 \\ 0 & x_{32} & 0 & x_{31} & 0 & x_{21} & 0 & 2x_{13} & 0 & 2x_{13} & 0 & 2x_{31} \\ x_{32} & y_{23} & x_{31} & y_{13} & x_{21} & y_{12} & 2x_{13} & 2y_{31} & 2x_{13} & 2y_{31} & 2x_{31} & 2y_{13} \end{bmatrix} \\ \mathbf{B}_3 &= \frac{1}{2A} \begin{bmatrix} y_{23} & 0 & y_{31} & 0 & y_{21} & 0 & 2y_{21} & 0 & 2y_{12} & 0 & 2y_{12} & 0 \\ 0 & x_{32} & 0 & x_{13} & 0 & x_{12} & 0 & 2x_{12} & 0 & 2x_{21} & 0 & 2x_{21} \\ x_{32} & y_{23} & x_{13} & y_{31} & x_{12} & y_{21} & 2x_{12} & 2y_{21} & 2x_{21} & 2y_{12} & 2x_{21} & 2y_{12} \end{bmatrix} \end{aligned} \quad (\text{E24.5})$$

With this form $\mathbf{K}^{(e)}$ can be computed in approximately 1000 floating-point operations.

Using next the 3-interior-point quadrature rule, show that the element stiffness can be expressed again as (E24.4) but with

$$\begin{aligned}
 \mathbf{B}_1 &= \frac{1}{6A} \begin{bmatrix} 5y_{23} & 0 & y_{13} & 0 & y_{21} & 0 & 2y_{21} + 6y_{31} & 0 & 2y_{32} & 0 & 6y_{12} + 2y_{13} & 0 \\ 0 & 5x_{32} & 0 & x_{31} & 0 & x_{12} & 0 & 2x_{12} + 6x_{13} & 0 & 2x_{23} & 0 & 6x_{21} + 2x_{31} \\ 5x_{32} & 5y_{23} & x_{31} & y_{13} & x_{12} & y_{21} & 2x_{12} + 6x_{13} & 2y_{21} + 6y_{31} & 2x_{23} & 2y_{32} & 6x_{21} + 2x_{31} & 6y_{12} + 2y_{13} \end{bmatrix} \\
 \mathbf{B}_2 &= \frac{1}{6A} \begin{bmatrix} y_{32} & 0 & 5y_{31} & 0 & y_{21} & 0 & 2y_{21} + 6y_{23} & 0 & 6y_{12} + 2y_{32} & 0 & 2y_{13} & 0 \\ 0 & x_{23} & 0 & 5x_{13} & 0 & x_{12} & 0 & 2x_{12} + 6x_{32} & 0 & 6x_{21} + 2x_{23} & 0 & 2x_{31} \\ x_{23} & y_{32} & 5x_{13} & 5y_{31} & x_{12} & y_{21} & 2x_{12} + 6x_{32} & 2y_{21} + 6y_{23} & 6x_{21} + 2x_{23} & 6y_{12} + 2y_{32} & 2x_{31} & 2y_{13} \end{bmatrix} \\
 \mathbf{B}_3 &= \frac{1}{6A} \begin{bmatrix} y_{32} & 0 & y_{13} & 0 & 5y_{12} & 0 & 2y_{21} & 0 & 6y_{31} + 2y_{32} & 0 & 2y_{13} + 6y_{23} & 0 \\ 0 & x_{23} & 0 & x_{31} & 0 & 5x_{21} & 0 & 2x_{12} & 0 & 6x_{13} + 2x_{23} & 0 & 2x_{31} + 6x_{32} \\ x_{23} & y_{32} & x_{31} & y_{13} & 5x_{21} & 5y_{12} & 2x_{12} & 2y_{21} & 6x_{13} + 2x_{23} & 6y_{31} + 2y_{32} & 2x_{31} + 6x_{32} & 2y_{13} + 6y_{23} \end{bmatrix}
 \end{aligned} \tag{E24.6}$$

The fact that two very different expressions yield the same $\mathbf{K}^{(e)}$ explain why sometimes authors rediscover the same element derived with different methods.

Yet another set that produces the correct stiffness is

$$\begin{aligned}
 \mathbf{B}_1 &= \frac{1}{6A} \begin{bmatrix} y_{23} & 0 & y_{31} & 0 & y_{21} & 0 & 2y_{21} & 0 & 2y_{12} & 0 & 2y_{12} & 0 \\ 0 & x_{32} & 0 & x_{13} & 0 & x_{12} & 0 & 2x_{12} & 0 & 2x_{21} & 0 & 2x_{21} \\ x_{32} & y_{23} & x_{13} & y_{31} & x_{12} & y_{21} & 2x_{12} & 2y_{21} & 2x_{21} & 2y_{12} & 2x_{21} & 2y_{12} \end{bmatrix} \\
 \mathbf{B}_2 &= \frac{1}{6A} \begin{bmatrix} y_{32} & 0 & y_{31} & 0 & y_{12} & 0 & 2y_{23} & 0 & 2y_{32} & 0 & 2y_{23} & 0 \\ 0 & x_{23} & 0 & x_{13} & 0 & x_{21} & 0 & 2x_{32} & 0 & 2x_{23} & 0 & 2x_{32} \\ x_{23} & y_{32} & x_{13} & y_{31} & x_{21} & y_{12} & 2x_{32} & 2y_{23} & 2x_{23} & 2y_{32} & 2x_{32} & 2y_{23} \end{bmatrix} \\
 \mathbf{B}_3 &= \frac{1}{6A} \begin{bmatrix} y_{23} & 0 & y_{13} & 0 & y_{12} & 0 & 2y_{31} & 0 & 2y_{31} & 0 & 2y_{13} & 0 \\ 0 & x_{32} & 0 & x_{31} & 0 & x_{21} & 0 & 2x_{13} & 0 & 2x_{13} & 0 & 2x_{31} \\ x_{32} & y_{23} & x_{31} & y_{13} & x_{21} & y_{12} & 2x_{13} & 2y_{31} & 2x_{13} & 2y_{31} & 2x_{31} & 2y_{13} \end{bmatrix}
 \end{aligned} \tag{E24.7}$$

EXERCISE 24.4

[A/C:40] (Research paper level) Characterize the most general form of the \mathbf{B}_i ($i = 1, 2, 3$) matrices that produce the same stiffness matrix $\mathbf{K}^{(e)}$ in (E24.4). (Entails solving an algebraic Riccati equation.)

25

The Assembly Process

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§25.1. INTRODUCTION

Previous Chapters have explained operations for forming element level entities such as stiffness matrices. Sandwiched between element processing and solution there is the *assembly* process, in which the master stiffness equations are constructed.

The position of the assembler in a DSM-based code for static analysis is sketched in Figure 25.1. In practice the assembler is implemented as a *driver* of the element library. That is, instead of forming all elements first and then assembling, the assembler constructs one element at a time, and immediately merges it into the master equations.

Assembly is the most complex stage of a production finite element program in terms of data flow. The complexity is not due to the mathematical operations, which merely involve matrix addition, but to the combined effect of formation of individual elements and merge. Forming an element requires access to node, element, constitutive and fabrication data. Merge requires access to the freedom and connection data, as well as knowledge of the sparse matrix format in which \mathbf{K} is stored. As illustrated in Figure 25.1, the assembler sits at the crossroads of this process.

The coverage of the assembly process for a general FEM implementation is beyond the scope of an introductory course. Instead this Chapter uses assumptions that lead to drastic simplifications, which in turn allows the basic aspects to be covered with a few examples.

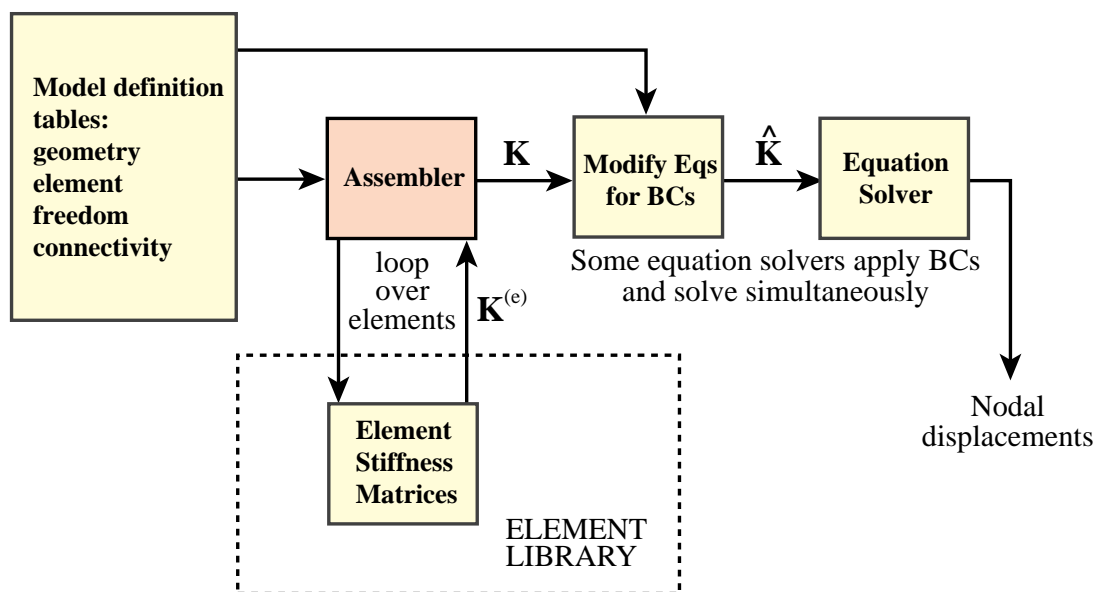


Figure 25.1. Role of assembler in FEM program.

§25.2. SIMPLIFIED ASSEMBLY PROCESS

The assembly process is considerably simplified if the implementation has the following properties:

1. All elements are of the same type. For example: all elements are plane stress elements.
2. The number and configuration of degrees of freedom at each node is the same.
3. There are no “gaps” in the node numbering sequence.

4. There are no multifreedom constraints done by master-slave or Lagrange multiplier methods.
5. The master stiffness matrix is stored as a full symmetric matrix.

If the first four conditions are met the implementation is simpler because the element freedom table described below can be constructed “on the fly” from the element node numbers. The last condition simplifies the merge process.

§25.2.1. A Plane Truss Example Structure

The plane truss structure shown in Figure 25.2 will be used to illustrate the details of the assembly process. The structure has 5 elements, 4 nodes and 8 degrees of freedom.

Begin by clearing all entries of the 8×8 matrix \mathbf{K} to zero, so that we effectively start with the null matrix:

$$\mathbf{K} = \begin{bmatrix} 0 & 0 & 0 & 0 & 0 & 0 & 0 & 0 \\ 0 & 0 & 0 & 0 & 0 & 0 & 0 & 0 \\ 0 & 0 & 0 & 0 & 0 & 0 & 0 & 0 \\ 0 & 0 & 0 & 0 & 0 & 0 & 0 & 0 \\ 0 & 0 & 0 & 0 & 0 & 0 & 0 & 0 \\ 0 & 0 & 0 & 0 & 0 & 0 & 0 & 0 \\ 0 & 0 & 0 & 0 & 0 & 0 & 0 & 0 \\ 0 & 0 & 0 & 0 & 0 & 0 & 0 & 0 \end{bmatrix} \begin{matrix} 1 \\ 2 \\ 3 \\ 4 \\ 5 \\ 6 \\ 7 \\ 8 \end{matrix} \quad (25.1)$$

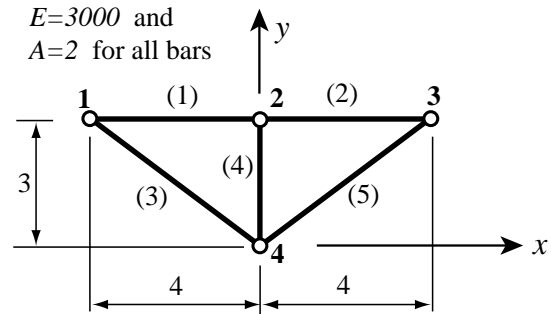


Figure 25.2. Sample structure to illustrate simplified assembly process.

The numbers written after each row of \mathbf{K} are the *global freedom numbers*.

Element (1) goes from node 1 to 2. The degrees of freedom of these nodes have global numbers $2 \times 1 - 1 = 1$, $2 \times 1 = 2$, $2 \times 2 - 1 = 3$ and $2 \times 2 = 4$. These 4 numbers are collected into an array called the *element freedom table*, or EFT for short. The element stiffness matrix is listed below with the EFT entries listed after the matrix rows:

$$\mathbf{K}^{(1)} = \begin{bmatrix} 1500 & 0 & -1500 & 0 \\ 0 & 0 & 0 & 0 \\ -1500 & 0 & 1500 & 0 \\ 0 & 0 & 0 & 0 \end{bmatrix} \begin{matrix} 1 \\ 2 \\ 3 \\ 4 \end{matrix} \quad (25.2)$$

Merging the element into (25.1) gives

$$\mathbf{K} = \begin{bmatrix} 1500 & 0 & -1500 & 0 & 0 & 0 & 0 & 0 \\ 0 & 0 & 0 & 0 & 0 & 0 & 0 & 0 \\ -1500 & 0 & 1500 & 0 & 0 & 0 & 0 & 0 \\ 0 & 0 & 0 & 0 & 0 & 0 & 0 & 0 \\ 0 & 0 & 0 & 0 & 0 & 0 & 0 & 0 \\ 0 & 0 & 0 & 0 & 0 & 0 & 0 & 0 \\ 0 & 0 & 0 & 0 & 0 & 0 & 0 & 0 \\ 0 & 0 & 0 & 0 & 0 & 0 & 0 & 0 \end{bmatrix} \begin{matrix} 1 \\ 2 \\ 3 \\ 4 \\ 5 \\ 6 \\ 7 \\ 8 \end{matrix} \quad (25.3)$$

See Figure 25.3.

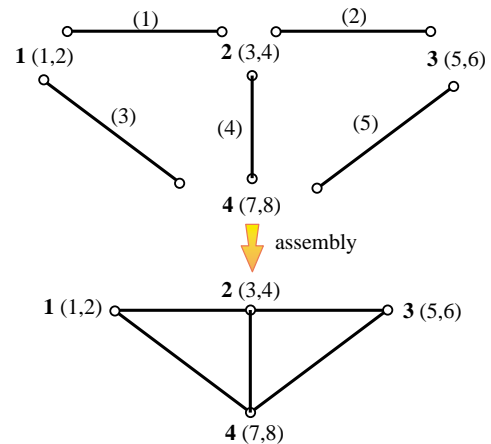


Figure 25.3. Illustration of disassembly/assembly process for plane truss example structure. Numbers in parentheses written after node numbers are the global freedom numbers.

Element (2) goes from node 2 to 3. The degrees of freedom at these nodes have global numbers $2 \times 2 - 1 = 3$, $2 \times 2 = 4$, $2 \times 3 - 1 = 5$ and $2 \times 3 = 6$. Hence the EFT is 3,4,5,6, and

$$\mathbf{K}^{(2)} = \begin{bmatrix} 1500 & 0 & -1500 & 0 \\ 0 & 0 & 0 & 0 \\ -1500 & 0 & 1500 & 0 \\ 0 & 0 & 0 & 0 \end{bmatrix} \begin{matrix} 3 \\ 4 \\ 5 \\ 6 \end{matrix} \quad (25.4)$$

Merging into (25.3) yields:

$$\mathbf{K} = \begin{bmatrix} 1500 & 0 & -1500 & 0 & 0 & 0 & 0 & 0 \\ 0 & 0 & 0 & 0 & 0 & 0 & 0 & 0 \\ -1500 & 0 & 3000 & 0 & -1500 & 0 & 0 & 0 \\ 0 & 0 & 0 & 0 & 0 & 0 & 0 & 0 \\ 0 & 0 & -1500 & 0 & 1500 & 0 & 0 & 0 \\ 0 & 0 & 0 & 0 & 0 & 0 & 0 & 0 \\ 0 & 0 & 0 & 0 & 0 & 0 & 0 & 0 \\ 0 & 0 & 0 & 0 & 0 & 0 & 0 & 0 \end{bmatrix} \begin{matrix} 3 \\ 4 \\ 5 \\ 6 \end{matrix} \quad (25.5)$$

Element (3) goes from node 1 to 4. Its EFT is 1,2,7,8, and its stiffness is

$$\mathbf{K}^{(3)} = \begin{bmatrix} 768 & -576 & -768 & 576 \\ -576 & 432 & 576 & -432 \\ -768 & 576 & 768 & -576 \\ 576 & -432 & -576 & 432 \end{bmatrix} \begin{matrix} 1 \\ 2 \\ 7 \\ 8 \end{matrix} \quad (25.6)$$

Merging this into (25.5) yields:

$$\mathbf{K} = \begin{bmatrix} 2268 & -576 & -1500 & 0 & 0 & 0 & -768 & 576 \\ -576 & 432 & 0 & 0 & 0 & 0 & 576 & -432 \\ -1500 & 0 & 3000 & 0 & -1500 & 0 & 0 & 0 \\ 0 & 0 & 0 & 0 & 0 & 0 & 0 & 0 \\ 0 & 0 & -1500 & 0 & 1500 & 0 & 0 & 0 \\ 0 & 0 & 0 & 0 & 0 & 0 & 0 & 0 \\ -768 & 576 & 0 & 0 & 0 & 0 & 768 & -576 \\ 576 & -432 & 0 & 0 & 0 & 0 & -576 & 432 \end{bmatrix} \begin{matrix} 1 \\ 2 \\ 3 \\ 4 \\ 5 \\ 6 \\ 7 \\ 8 \end{matrix} \quad (25.7)$$

Element (4) goes from node 2 to 4. Its EFT is 3,4,7,8, and its stiffness is

$$\mathbf{K}^{(4)} = \begin{bmatrix} 0 & 0 & 0 & 0 \\ 0 & 2000 & 0 & -2000 \\ 0 & 0 & 0 & 0 \\ 0 & -2000 & 0 & 2000 \end{bmatrix} \begin{matrix} 3 \\ 4 \\ 7 \\ 8 \end{matrix} \quad (25.8)$$

Merging this into (25.7) yields

$$\mathbf{K} = \begin{bmatrix} 2268 & -576 & -1500 & 0 & 0 & 0 & -768 & 576 \\ -576 & 432 & 0 & 0 & 0 & 0 & 576 & -432 \\ -1500 & 0 & 3000 & 0 & -1500 & 0 & 0 & 0 \\ 0 & 0 & 0 & 2000 & 0 & 0 & 0 & -2000 \\ 0 & 0 & -1500 & 0 & 1500 & 0 & 0 & 0 \\ 0 & 0 & 0 & 0 & 0 & 0 & 0 & 0 \\ -768 & 576 & 0 & 0 & 0 & 0 & 768 & -576 \\ 576 & -432 & 0 & -2000 & 0 & 0 & -576 & 2432 \end{bmatrix} \begin{matrix} 3 \\ 4 \\ 5 \\ 6 \\ 7 \\ 8 \end{matrix} \quad (25.9)$$

Finally, element (5) goes from node 3 to 4. Its EFT is 5,6,7,8, and its element stiffness is

$$\mathbf{K}^{(5)} = \begin{bmatrix} 768 & 576 & -768 & -576 \\ 576 & 432 & -576 & -432 \\ -768 & -576 & 768 & 576 \\ -576 & -432 & 576 & 432 \end{bmatrix} \begin{matrix} 5 \\ 6 \\ 7 \\ 8 \end{matrix} \quad (25.10)$$

Merging this into (25.9) yields

$$\mathbf{K} = \begin{bmatrix} 2268 & -576 & -1500 & 0 & 0 & 0 & -768 & 576 \\ -576 & 432 & 0 & 0 & 0 & 0 & 576 & -432 \\ -1500 & 0 & 3000 & 0 & -1500 & 0 & 0 & 0 \\ 0 & 0 & 0 & 2000 & 0 & 0 & 0 & -2000 \\ 0 & 0 & -1500 & 0 & 2268 & 576 & -768 & -576 \\ 0 & 0 & 0 & 0 & 576 & 432 & -576 & -432 \\ -768 & 576 & 0 & 0 & -768 & -576 & 1536 & 0 \\ 576 & -432 & 0 & -2000 & -576 & -432 & 0 & 2864 \end{bmatrix} \begin{matrix} 5 \\ 6 \\ 7 \\ 8 \\ 5 \\ 6 \\ 7 \\ 8 \end{matrix} \quad (25.11)$$

Since all elements have been processed, (25.11) is the complete free-free master stiffness.

§25.2.2. A Mathematica Implementation

See Cell 25.1 for a *Mathematica* implementation of the assembly process just described. This is a reproduction of the assembler module used in Chapter 22 for a plane truss structure. The listing of the stiffness formation module `Stiffness2DBar`, which is the same as that used in Chapter 22, is omitted to save space.

Running the test statements shown at the end of the cell produces the output shown in Cell 25.2. This reproduces the master stiffness (25.11). The eigenvalue analysis verifies that the assembled stiffness has the correct rank of $8 - 3 = 5$.

§25.3. COMPLICATIONS

We now proceed to follow the assembly of a more complicated structure in which two of the simplifications listed previously do not apply:

1. The structure contains different element types: bars and beam-columns.
2. The nodal freedom configuration varies.

Under these conditions it is not possible to compute the global freedom number directly from the node number. It is necessary to build a Node-to-Freedom Map Table, or NFMT, first. The code to generate an NFMT is the topic of Exercise 25.3.

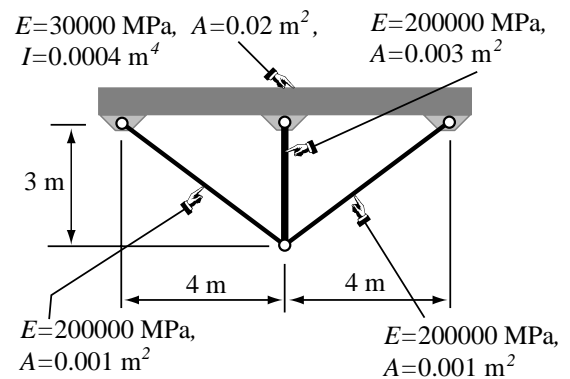


Figure 25.4. A frame-truss structure that illustrates assembly with multiple element types (beams and bars) and different nodal freedom configurations.

Cell 25.1 - A Mathematica Assembler for the Truss Example of Figure 25.2

```

AssembleMasterStiffnessPlaneTruss[nodcoor_,elenod_,
  elemat_,elefab_,eleopt_]:=Module[
  {numele=Length[elenod],numnod=Length[nodcoor],
  e,eNL,eftab,ni,nj,i,j,ncoor,mprop,fprop,opt,Ke,K},
  K=Table[0,{2*numnod},{2*numnod}];
  For [e=1, e<=numele, e++,
    eNL=elenod[[e]]; {ni,nj}=eNL;
    eftab={2*ni-1,2*ni,2*nj-1,2*nj};
    ncoor={nodcoor[[ni]],nodcoor[[nj]]};
    mprop=elemat[[e]]; fprop=elefab[[e]]; opt=eleopt;
    Ke=Stiffness2DBar[ncoor,mprop,fprop,opt];
    neldof=Length[Ke];
    For [i=1, i<=neldof, i++, ii=eftab[[i]];
      For [j=i, j<=neldof, j++, jj=eftab[[j]];
        K[[jj,ii]]=K[[ii,jj]]+=Ke[[i,j]]
      ];
    ];
  ];
  Return[K]];

(* Stiffness2DBar listing omitted to save space *)

nodcoor={{-4,3},{0,3},{4,3},{0,0}};
elenod= {{1,2},{2,3},{1,4},{2,4},{3,4}};
elemat= Table[{3000,0,0,0},{5}];
elefab= Table[{2},{5}];
eleopt= {True};
K=AssembleMasterStiffnessPlaneTruss[nodcoor,elenod,elemat,elefab,eleopt];
Print["Master Stiffness of Example Truss:"];
Print[K//MatrixForm];
Print["Eigs of K=",Chop[Eigenvalues[N[K]]]];

```

Cell 25.2 - Output from the Program of Cell 25.1

```

Master Stiffness of Example Truss:
 2268.   -576.  -1500.    0    0    0   -768.   576.
-576.    432.    0    0    0    0    576.  -432.
-1500.    0   3000.    0  -1500.    0    0    0
 0    0    0   2000.    0    0    0  -2000.
 0    0  -1500.    0   2268.   576.  -768.  -576.
 0    0    0    0   576.   432.  -576.  -432.
-768.   576.    0    0  -768.  -576.  1536.    0.
 576.  -432.    0  -2000.  -576.  -432.    0.  2864.
Eigs of K={5007.22, 4743.46, 2356.84, 2228.78, 463.703, 0, 0, 0}

```

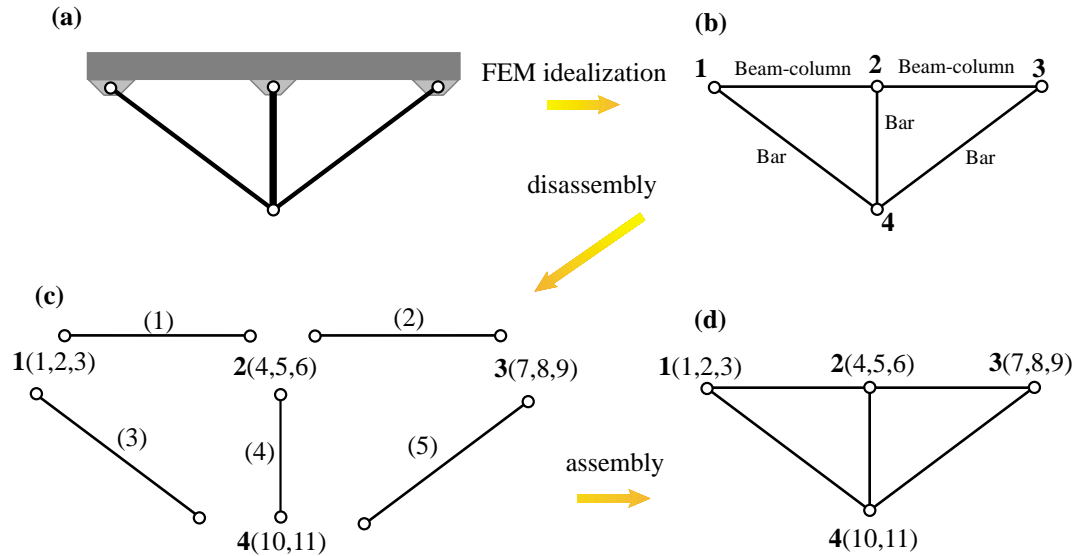


Figure 25.5. Illustration of disassembly/assembly process for frame-truss example structure. Numbers written in parentheses after the node numbers are the global freedom numbers.

§25.3.1. Frame-Truss Example Structure

We illustrate the assembly process with the structure shown in Figure 25.4. The finite element discretization, disassembly into elements and assembly are illustrated in Figure 25.5.

The structure is a frame-truss, with two element types: bars and beams (more precisely, beam-columns). It has 11 degrees of freedom, 3 at nodes 1, 2 and 3, and 2 at node 3. They are ordered as follows:

GDOF #:	1	2	3	4	5	6	7	8	9	10	11
DOF:	u_{x1}	u_{y1}	θ_{z1}	u_{x2}	u_{y2}	θ_{z2}	u_{x3}	u_{y3}	θ_{z3}	u_{x4}	u_{y4}

(25.12)

The position of each freedom, as identified by the numbers in the first row, is called the *global freedom number*.

We now proceed to go over the assembly process by hand. The master stiffness \mathbf{K} is displayed as a fully stored matrix for visualization convenience. The process begins by initialization of $\mathbf{K} = \mathbf{0}$.

Element (1): This is a plane beam-column element with freedoms

1	2	3	4	5	6
u_{x1}	u_{y1}	θ_{z1}	u_{x2}	u_{y2}	θ_{z2}

(25.13)

The mapping between the element and global freedoms is specified by the EFT

element:	1	2	3	4	5	6
assembly:	1	2	3	4	5	6

(25.14)

The element stiffness is

$$\mathbf{K}^{(1)} = \begin{bmatrix} 150. & 0. & 0. & -150. & 0. & 0. \\ 0. & 22.5 & 45. & 0. & -22.5 & 45. \\ 0. & 45. & 120. & 0. & -45. & 60. \\ -150. & 0. & 0. & 150. & 0. & 0. \\ 0. & -22.5 & -45. & 0. & 22.5 & -45. \\ 0. & 45. & 60. & 0. & -45. & 120. \end{bmatrix} \begin{matrix} 1 \\ 2 \\ 3 \\ 4 \\ 5 \\ 6 \end{matrix} \quad (25.15)$$

Upon merging this element the master stiffness matrix becomes

$$\mathbf{K} = \begin{bmatrix} 150. & 0. & 0. & -150. & 0. & 0. & 0 & 0 & 0 & 0 & 0 \\ 0. & 22.5 & 45. & 0. & -22.5 & 45. & 0 & 0 & 0 & 0 & 0 \\ 0. & 45. & 120. & 0. & -45. & 60. & 0 & 0 & 0 & 0 & 0 \\ -150. & 0. & 0. & 150. & 0. & 0. & 0 & 0 & 0 & 0 & 0 \\ 0. & -22.5 & -45. & 0. & 22.5 & -45. & 0 & 0 & 0 & 0 & 0 \\ 0. & 45. & 60. & 0. & -45. & 120. & 0 & 0 & 0 & 0 & 0 \\ 0 & 0 & 0 & 0 & 0 & 0 & 0 & 0 & 0 & 0 & 0 \\ 0 & 0 & 0 & 0 & 0 & 0 & 0 & 0 & 0 & 0 & 0 \\ 0 & 0 & 0 & 0 & 0 & 0 & 0 & 0 & 0 & 0 & 0 \\ 0 & 0 & 0 & 0 & 0 & 0 & 0 & 0 & 0 & 0 & 0 \\ 0 & 0 & 0 & 0 & 0 & 0 & 0 & 0 & 0 & 0 & 0 \end{bmatrix} \begin{matrix} 1 \\ 2 \\ 3 \\ 4 \\ 5 \\ 6 \\ 7 \\ 8 \\ 9 \\ 10 \\ 11 \end{matrix} \quad (25.16)$$

Element (2): This is a plane beam-column element with freedoms

1	2	3	4	5	6
u_{x2}	u_{y2}	θ_{z2}	u_{x3}	u_{y3}	θ_{z3}

(25.17)

The mapping between the element and global freedoms is

element:	1	2	3	4	5	6
assembly:	4	5	6	7	8	9

(25.18)

The element stiffness matrix $\mathbf{K}^{(2)}$ is identical to $\mathbf{K}^{(1)}$:

$$\mathbf{K}^{(2)} = \begin{bmatrix} 150. & 0. & 0. & -150. & 0. & 0. \\ 0. & 22.5 & 45. & 0. & -22.5 & 45. \\ 0. & 45. & 120. & 0. & -45. & 60. \\ -150. & 0. & 0. & 150. & 0. & 0. \\ 0. & -22.5 & -45. & 0. & 22.5 & -45. \\ 0. & 45. & 60. & 0. & -45. & 120. \end{bmatrix} \begin{matrix} 4 \\ 5 \\ 6 \\ 7 \\ 8 \\ 9 \end{matrix} \quad (25.19)$$

Upon merge the master stiffness matrix becomes

$$\mathbf{K} = \begin{bmatrix} 150. & 0. & 0. & -150. & 0. & 0. & 0 & 0 & 0 & 0 & 0 \\ 0. & 22.5 & 45. & 0. & -22.5 & 45. & 0 & 0 & 0 & 0 & 0 \\ 0. & 45. & 120. & 0. & -45. & 60. & 0 & 0 & 0 & 0 & 0 \\ -150. & 0. & 0. & 300. & 0. & 0. & -150. & 0. & 0. & 0 & 0 \\ 0. & -22.5 & -45. & 0. & 45. & 0. & 0. & -22.5 & 45. & 0 & 0 \\ 0. & 45. & 60. & 0. & 0. & 240. & 0. & -45. & 60. & 0 & 0 \\ 0 & 0 & 0 & -150. & 0. & 0. & 150. & 0. & 0. & 0 & 0 \\ 0 & 0 & 0 & 0. & -22.5 & -45. & 0. & 22.5 & -45. & 0 & 0 \\ 0 & 0 & 0 & 0. & 45. & 60. & 0. & -45. & 120. & 0 & 0 \\ 0 & 0 & 0 & 0 & 0 & 0 & 0 & 0 & 0 & 0 & 0 \\ 0 & 0 & 0 & 0 & 0 & 0 & 0 & 0 & 0 & 0 & 0 \end{bmatrix} \begin{matrix} 4 \\ 5 \\ 6 \\ 7 \\ 8 \\ 9 \end{matrix} \quad (25.20)$$

Element (3): This is a plane bar element with freedoms

1	2	3	4
u_{x1}	u_{y1}	u_{x4}	u_{y4}

(25.21)

The mapping between the element and global freedoms is

element:	1	2	3	4
assembly:	1	2	10	11

(25.22)

The element stiffness matrix is

$$\mathbf{K}^{(3)} = \begin{bmatrix} 25.6 & -19.2 & -25.6 & 19.2 \\ -19.2 & 14.4 & 19.2 & -14.4 \\ -25.6 & 19.2 & 25.6 & -19.2 \\ 19.2 & -14.4 & -19.2 & 14.4 \end{bmatrix} \begin{matrix} 1 \\ 2 \\ 10 \\ 11 \end{matrix} \quad (25.23)$$

Upon merge the master stiffness matrix becomes

$$\mathbf{K} = \begin{bmatrix} 175.6 & -19.2 & 0. & -150. & 0. & 0. & 0 & 0 & 0 & -25.6 & 19.2 \\ -19.2 & 36.9 & 45. & 0. & -22.5 & 45. & 0 & 0 & 0 & 19.2 & -14.4 \\ 0. & 45. & 120. & 0. & -45. & 60. & 0 & 0 & 0 & 0 & 0 \\ -150. & 0. & 0. & 300. & 0. & 0. & -150. & 0. & 0. & 0 & 0 \\ 0. & -22.5 & -45. & 0. & 45. & 0. & 0. & -22.5 & 45. & 0 & 0 \\ 0. & 45. & 60. & 0. & 0. & 240. & 0. & -45. & 60. & 0 & 0 \\ 0 & 0 & 0 & -150. & 0. & 0. & 150. & 0. & 0 & 0 & 0 \\ 0 & 0 & 0 & 0. & -22.5 & -45. & 0. & 22.5 & -45. & 0 & 0 \\ 0 & 0 & 0 & 0. & 45. & 60. & 0. & -45. & 120. & 0 & 0 \\ -25.6 & 19.2 & 0 & 0 & 0 & 0 & 0 & 0 & 0 & 25.6 & -19.2 \\ 19.2 & -14.4 & 0 & 0 & 0 & 0 & 0 & 0 & 0 & -19.2 & 14.4 \end{bmatrix} \begin{matrix} 1 \\ 2 \\ \\ 10 \\ 11 \end{matrix} \quad (25.24)$$

Element (4): This is a plane bar element with freedoms

1	2	3	4
u_{x2}	u_{y2}	u_{x4}	u_{y4}

(25.25)

The mapping between the element and global freedoms is

element:	1	2	3	4
assembly:	4	5	10	11

(25.26)

The element stiffness matrix is

$$\mathbf{K}^{(4)} = \begin{bmatrix} 0 & 0 & 0 & 0 \\ 0 & 200. & 0 & -200. \\ 0 & 0 & 0 & 0 \\ 0 & -200. & 0 & 200. \end{bmatrix} \begin{matrix} 4 \\ 5 \\ 10 \\ 11 \end{matrix} \quad (25.27)$$

Upon merge the master stiffness matrix becomes

$$\mathbf{K} = \begin{bmatrix} 175.6 & -19.2 & 0. & -150. & 0. & 0. & 0 & 0 & 0 & -25.6 & 19.2 \\ -19.2 & 36.9 & 45. & 0. & -22.5 & 45. & 0 & 0 & 0 & 19.2 & -14.4 \\ 0. & 45. & 120. & 0. & -45. & 60. & 0 & 0 & 0 & 0 & 0 \\ -150. & 0. & 0. & 300. & 0. & 0. & -150. & 0. & 0. & 0 & 0 \\ 0. & -22.5 & -45. & 0. & 245. & 0. & 0. & -22.5 & 45. & 0 & -200. \\ 0. & 45. & 60. & 0. & 0. & 240. & 0. & -45. & 60. & 0 & 0 \\ 0 & 0 & 0 & -150. & 0. & 0. & 150. & 0. & 0 & 0 & 0 \\ 0 & 0 & 0 & 0. & -22.5 & -45. & 0. & 22.5 & -45. & 0 & 0 \\ 0 & 0 & 0 & 0. & 45. & 60. & 0. & -45. & 120. & 0 & 0 \\ -25.6 & 19.2 & 0 & 0 & 0 & 0 & 0 & 0 & 0 & 25.6 & -19.2 \\ 19.2 & -14.4 & 0 & 0 & -200. & 0 & 0 & 0 & 0 & -19.2 & 214.4 \end{bmatrix} \begin{matrix} \\ \\ \\ 4 \\ 5 \\ \\ \\ \\ 10 \\ 11 \end{matrix} \quad (25.28)$$

Element (5): This is a plane bar element with freedoms

1	2	3	4
u_{x2}	u_{y2}	u_{x4}	u_{y4}

(25.29)

The mapping between the element and global freedoms is

element:	1	2	3	4
assembly:	7	8	10	11

(25.30)

The element stiffness matrix is

$$\mathbf{K}^{(5)} = \begin{bmatrix} 25.6 & 19.2 & -25.6 & -19.2 \\ 19.2 & 14.4 & -19.2 & -14.4 \\ -25.6 & -19.2 & 25.6 & 19.2 \\ -19.2 & -14.4 & 19.2 & 14.4 \end{bmatrix} \begin{matrix} 7 \\ 8 \\ 10 \\ 11 \end{matrix} \quad (25.31)$$

Upon merge the master stiffness matrix becomes

$$\mathbf{K} = \begin{bmatrix} 175.6 & -19.2 & 0. & -150. & 0. & 0. & 0 & 0 & 0 & -25.6 & 19.2 \\ -19.2 & 36.9 & 45. & 0. & -22.5 & 45. & 0 & 0 & 0 & 19.2 & -14.4 \\ 0. & 45. & 120. & 0. & -45. & 60. & 0 & 0 & 0 & 0 & 0 \\ -150. & 0. & 0. & 300. & 0. & 0. & -150. & 0. & 0. & 0 & 0 \\ 0. & -22.5 & -45. & 0. & 245. & 0. & 0. & -22.5 & 45. & 0 & -200. \\ 0. & 45. & 60. & 0. & 0. & 240. & 0. & -45. & 60. & 0 & 0 \\ 0 & 0 & 0 & -150. & 0. & 0. & 175.6 & 19.2 & 0 & -25.6 & -19.2 \\ 0 & 0 & 0 & 0. & -22.5 & -45. & 19.2 & 36.9 & -45. & -19.2 & -14.4 \\ 0 & 0 & 0 & 0. & 45. & 60. & 0. & -45. & 120. & 0 & 0 \\ -25.6 & 19.2 & 0 & 0 & 0 & 0 & -25.6 & -19.2 & 0 & 51.2 & 0 \\ 19.2 & -14.4 & 0 & 0 & -200. & 0 & -19.2 & -14.4 & 0 & 0 & 228.8 \end{bmatrix} \begin{matrix} \\ \\ \\ \\ \\ \\ 7 \\ 8 \\ \\ 10 \\ 11 \end{matrix} \quad (25.32)$$

Since all elements have been processed, (25.32) is the master stiffness matrix of the example structure of Figure 25.4. That it has the proper rank of $11 - 3 = 8$ may be verified by an eigenvalue analysis.

REMARK 25.1

For storage as a skyline matrix the connection between nodes 1 and 3 would be out of the skyline envelope. Omitting the lower triangle the skyline template for (25.32) looks like

$$\mathbf{K} = \begin{bmatrix} 175.6 & -19.2 & 0. & -150. & 0. & 0. & & & & -25.6 & 19.2 \\ & 36.9 & 45. & 0. & -22.5 & 45. & & & & 19.2 & -14.4 \\ & & 120. & 0. & -45. & 60. & & & & 0 & 0 \\ & & & 300. & 0. & 0. & -150. & 0. & 0. & 0 & 0 \\ & & & & 245. & 0. & 0. & -22.5 & 45. & 0 & -200. \\ & & & & & 240. & 0. & -45. & 60. & 0 & 0 \\ & & & & & & 175.6 & 19.2 & 0. & -25.6 & -19.2 \\ & & & & & & & 36.9 & -45. & -19.2 & -14.4 \\ & & & & & & & & 120. & 0 & 0 \\ & & & & & & & & & 51.2 & 0. \\ \text{symm} & & & & & & & & & & 228.8 \end{bmatrix} \quad (25.33)$$

The diagonal location pointers of (25.33) are defined by the table

$$\text{DLT} = \{ 0, 1, 3, 6, 10, 15, 21, 25, 30, 36, 46, 57 \} \quad (25.34)$$

Examination of the skyline template (25.33) reveals that some additional zero entries could be removed from the template; for example K_{13} . The fact that those entries are exactly zero is, however, fortuitous. It comes from the fact that some elements such as the beams are aligned along x , which decouples axial and bending stiffnesses.

§25.3.2. A Mathematica Implementation

Cell 25.3 gives a *Mathematica* implementation of the assembly process for the frame-truss structure just described. This is done by module `AssembleMasterStiffnessPlaneFrameTruss`. The two subordinate modules `Stiffness2DBar` and `Stiffness2DBeamColumn`, which implement the stiffness matrix calculation for a plane bar and beam-column, respectively, are omitted to save space. These are the same used in Chapter 21.

Cell 25.3 - A Mathematica Assembler for the Frame-Truss Example of Figure 25.4

```

AssembleMasterStiffnessPlaneFrameTruss[nodcoor_,eletyp_,
  elenod_,elemat_,elefab_,eleopt_,NFCT_] :=Module[
{numele=Length[elenod],numnod=Length[nodcoor],
 numdof,e,eNL,eftab,n,ni,nj,i,j,k,m,mi,mj,
 ncoor,mprop,fprop,opt,NFMT,Ke,K}, NFMT=Table[0,{numnod}];
 m=0; For [n=1,n<=numnod,n++, k=NFCT[[n]];
   If [k>0, NFMT[[n]]=m; m+=k, NFMT[[n]]=-1]]; numdof=m;
 K=Table[0,{numdof},{numdof}];
 For [e=1, e<=numele, e++,
   eNL=elenod[[e]]; {ni,nj}=eNL; mi=NFMT[[ni]]; mj=NFMT[[nj]];
   ncoor={nodcoor[[ni]],nodcoor[[nj]]};
   mprop=elemat[[e]]; fprop=elefab[[e]]; opt=eleopt;
   If [eletyp[[e]]=="Bar",
     {ni,nj}=eNL; mi=NFMT[[ni]]; mj=NFMT[[nj]];
     ncoor={nodcoor[[ni]],nodcoor[[nj]]};
     eftab={mi+1,mi+2,mj+1,mj+2};
     Ke=Stiffness2DBar[ncoor,mprop,fprop,opt]];
   If [eletyp[[e]]=="BeamColumn",
     {ni,nj}=eNL; mi=NFMT[[ni]]; mj=NFMT[[nj]];
     ncoor={nodcoor[[ni]],nodcoor[[nj]]};
     eftab={mi+1,mi+2,mi+3,mj+1,mj+2,mj+3};
     Ke=Stiffness2DBeamColumn[ncoor,mprop,fprop,opt]];
   neldof=Length[Ke];
   For [i=1, i<=neldof, i++, ii=eftab[[i]];
     For [j=i, j<=neldof, j++, jj=eftab[[j]];
       K[[jj,ii]]=K[[ii,jj]]+Ke[[i,j]]
     ];
   ];
 ];
 Return[K]
];

(* Stiffness2DBar and Stiffness2DBeamColumn listings omitted to save space *)
nodcoor={{-4,3},{0,3},{4,3},{0,0}};
eletyp= {"BeamColumn","BeamColumn","Bar","Bar","Bar"};
elenod= {{1,2},{2,3},{1,4},{2,4},{3,4}};
elemat= Join[Table[{30000,0,0,0},{2}],Table[{200000,0,0,0},{3}]];
elefab= {{0.02,0.004},{0.02,0.004},{0.001},{0.003},{0.001}};
eleopt= {True}; NFCT= {3,3,3,2};
K=AssembleMasterStiffnessPlaneFrameTruss[nodcoor,eletyp,
  elenod,elemat,elefab,eleopt,NFCT]; K=Chop[K];
Print["Master Stiffness of Example Frame-Truss:"];
Print[K//MatrixForm];
Print["Eigs of K=",Chop[Eigenvalues[N[K]]]];

```

Comparing the logic of AssembleMasterStiffnessPlaneFrameTruss to that of Cell 25.1, an increase in complexity is evident. This assembler has two additional arguments:

eletyp A list of element type identifiers provided as character strings: "Bar" for a bar element or "BeamColumn" for a beam-column element. For the example frame-truss structure

Cell 25.4 - Output from the Program of Cell 25.3

Master Stiffness of Example Frame-Truss:										
175.6	-19.2	0.	-150.	0.	0.	0	0	0	-25.6	19.2
-19.2	36.9	45.	0.	-22.5	45.	0	0	0	19.2	-14.4
0.	45.	120.	0.	-45.	60.	0	0	0	0	0
-150.	0.	0.	300.	0.	0.	-150.	0.	0.	0	0
0.	-22.5	-45.	0.	245.	0.	0.	-22.5	45.	0	-200.
0.	45.	60.	0.	0.	240.	0.	-45.	60.	0	0
0	0	0	-150.	0.	0.	175.6	19.2	0.	-25.6	-19.2
0	0	0	0.	-22.5	-45.	19.2	36.9	-45.	-19.2	-14.4
0	0	0	0.	45.	60.	0.	-45.	120.	0	0
-25.6	19.2	0	0	0	0	-25.6	-19.2	0	51.2	0.
19.2	-14.4	0	0	-200.	0	-19.2	-14.4	0	0.	228.8
Eigs of K={460.456, 445.431, 306.321, 188.661, 146.761, 82.7415, 74.181, 25.4476, 0, 0, 0}										

eletyp is { "BeamColumn", "BeamColumn", "Bar", "Bar", "Bar" }.

NFCT The Node Freedom Count Table (NFCT) is an integer array with one entry per node. Entry $NFCT[[n]]=k$ if there are $k \geq 0$ degrees of freedom defined at node n . If k is zero, node n has not been defined. The NFCT for the example structure of Figure 25.4 is { 3, 3, 3, 2 }. In Cell 25.3 this array is directly set by the test statements. The construction of this table from more primitive node freedom data is the subject of Exercise 25.2.

The element fabrication list `eleftab` has minor modifications. A plane beam-column element requires two properties: { A, I_{zz} }, which are the cross section area and the moment of inertia about the local z axis. A bar element requires only the cross section area: { A }. For the example frame-truss structure `eleftab` is { { 0.02, 0.004 }, { 0.02, 0.004 }, { 0.001 }, { 0.003 }, { 0.001 } } in accordance with Figure 25.4. The element material property list `elemat` is modified on the account that the elastic moduli for the beam and bars is different. The other arguments are the same as in Cell 25.1.

Upon entry the assembler proceeds to build the Node Freedom Map Table or NFMT from NFCT. This is an array with one entry per node. Entry $NFMT[[n]]=i$, $i \geq 0$, if the global freedoms associated with node n are $i+1, \dots, i+k$, where $k=NFCT[[n]]$. If node n is undefined, $NFMT[[n]]=-1$. For example, the NFMT for the frame-truss example structure is { 0, 3, 6, 9 }. The total number of degrees of freedoms, `numdof`, is obtained as a byproduct, and used to initialize the master stiffness array.

The assembler then loops over the elements and unpacks data. It calls the appropriate element stiffness module according to element type. The NFMT is used to construct the Element Freedom Table (EFT) array. The merging logic (the two For loops at the end of the element loop) is common to both elements. Running the test statements shown at the end of Cell 25.3 produces the output shown in Cell 25.4. This reproduces the master stiffness (25.32). The eigenvalue analysis verifies that the assembled stiffness has the correct rank of $11 - 3 = 8$.

§25.4. *KINEMATIC RELEASES

Occasionally it is necessary to account for *kinematic releases* that preclude freedoms at a node to be the same. This occurs when modeling devices such as hinges and expansion joints. The treatment of these cases by freedom injection or node duplication is left to future version of these Notes.

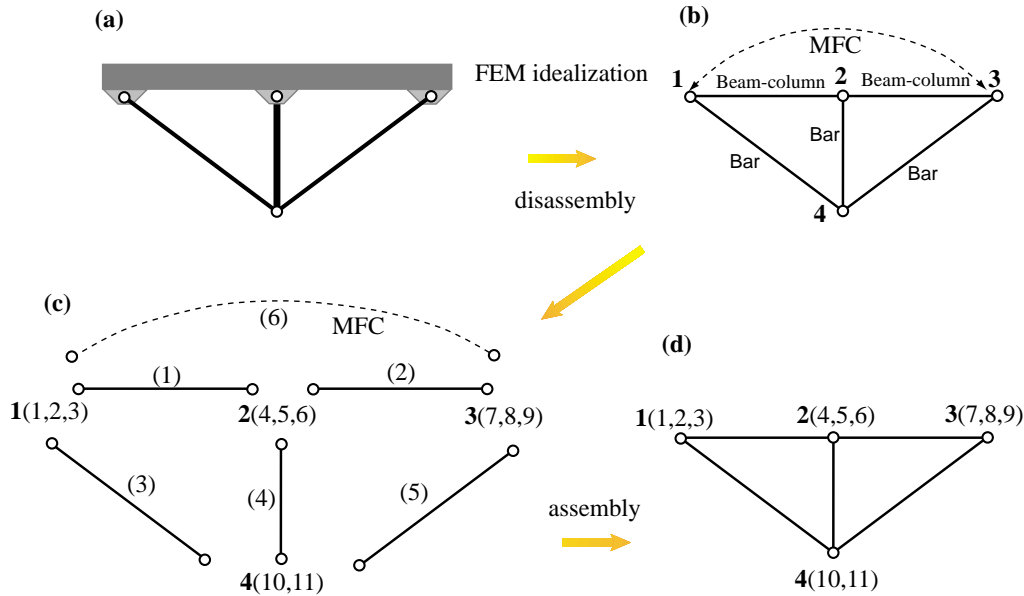


Figure 25.6. Finite element discretization, disassembly and assembly of frame-truss example structure with MFC $u_{y1} = u_{y3}$.

§25.5. *IMPOSING A MULTIFREEDOM CONSTRAINT

To see the effect of imposing an MFC through the Lagrange multiplier method on the configuration of the master stiffness matrix, suppose that the frame-truss example structure of the previous section is subjected to the constraint that nodes 1 and 3 must move vertically by the same amount. That is,

$$u_{y1} = u_{y3} \quad \text{or} \quad u_{y1} - u_{y3} = 0. \quad (25.35)$$

For assembly purposes (25.35) may be viewed as an element labeled as (6). See Figure 25.6.

The degrees of freedom of the assembled structure increase by one to 12. They are ordered as follows:

GDOF #:	1	2	3	4	5	6	7	8	9	10	11	12
DOF:	u_{x1}	u_{y1}	θ_{z1}	u_{x2}	u_{y2}	θ_{z2}	u_{x3}	u_{y3}	θ_{z3}	u_{x4}	u_{y4}	$\lambda^{(6)}$

(25.36)

The assembly of the first five elements proceeds as explained before, and produces the same master stiffness as (25.32), except for an extra zero row and column. Processing the MFC element (6) yields

$$\mathbf{K} = \begin{bmatrix} 175.6 & -19.2 & 0. & -150. & 0. & 0. & 0 & 0 & 0 & -25.6 & 19.2 & 0 \\ -19.2 & 36.9 & 45. & 0. & -22.5 & 45. & 0 & 0 & 0 & 19.2 & -14.4 & 1. \\ 0. & 45. & 120. & 0. & -45. & 60. & 0 & 0 & 0 & 0 & 0 & 0 \\ -150. & 0. & 0. & 300. & 0. & 0. & -150. & 0. & 0. & 0 & 0 & 0 \\ 0. & -22.5 & -45. & 0. & 245. & 0. & 0. & -22.5 & 45. & 0 & -200. & 0 \\ 0. & 45. & 60. & 0. & 0. & 240. & 0. & -45. & 60. & 0 & 0 & 0 \\ 0 & 0 & 0 & -150. & 0. & 0. & 175.6 & 19.2 & 0 & -25.6 & -19.2 & 0 \\ 0 & 0 & 0 & 0. & -22.5 & -45. & 19.2 & 36.9 & -45. & -19.2 & -14.4 & -1. \\ 0 & 0 & 0 & 0. & 45. & 60. & 0. & -45. & 120. & 0 & 0 & 0 \\ -25.6 & 19.2 & 0 & 0 & 0 & 0 & -25.6 & -19.2 & 0 & 51.2 & 0 & 0 \\ 19.2 & -14.4 & 0 & 0 & -200. & 0 & -19.2 & -14.4 & 0 & 0 & 228.8 & 0 \\ 0 & 1. & 0 & 0 & 0 & 0 & 0 & -1. & 0 & 0 & 0 & 0 \end{bmatrix} \quad (25.37)$$

in which the coefficients 1 and -1 associated with the MFC (25.35) end up in the last row and column.

Notes and Bibliography

The DSM assembly process for the simple cases enumerated in §25.2 is explained in any finite element. Few cover the complications that arise in more general situations. In a general purpose FEM code, the assembler typically requires thousands of code lines to cover all contingencies.

Homework Exercises for Chapter 25

The Assembly Process

EXERCISE 25.1

[A+C:5+10] The Freedom Activity Table, or FAT, is a two-dimensional data structure that marks which degrees of freedoms (DOF) are specified at each node. Entry $\text{FAT}[[n]]$ is a one-dimensional list of freedom activity tags. For example, suppose that¹ up to six DOF: three translations and three rotations ordered as

$$u_{xn}, u_{yn}, u_{zn}, \theta_{xn}, \theta_{yn}, \theta_{zn} \quad (\text{E25.1})$$

can be specified at each node $n = 1, 2, \dots$. Then each entry of the FAT If the j^{th} freedom is defined it is identified by a one, and if undefined by a zero. The configuration of the FAT for the plane truss example structure of Figure 25.2 is

$$\text{FAT} = \{ \{1, 1, 0, 0, 0, 0\}, \{1, 1, 0, 0, 0, 0\}, \{1, 1, 0, 0, 0, 0\}, \{1, 1, 0, 0, 0, 0\} \}$$

which says that only u_{xn} and u_{yn} are defined at each of the four nodes.

- (a) Show how the FAT of the frame-truss structure of Figure 25.4 looks like if this marking scheme is used. (Note: θ_n is the same as θ_{zn} .)
- (b) Suppose that the nodes of the frame-truss structure of Figure 25.4 are numbered 1, 2, 5 and 6 instead of 1, 2, 3 and 4. In other words, there is a “node gap” in that nodes 3 and 4 are undefined. Show how the FAT would then look like.

EXERCISE 25.2

[A+C:15] The Node Freedom Count Table, or NFCT, is an array with one entry per node. Entry $\text{NFCT}[[n]] = k$ if there are $k \geq 0$ defined DOFs at node n . If k is zero, node n has not been defined. For example, the NFCT for the frame-truss structure of Figure 25.4 is

$$\text{NFCT} = \{3, 3, 3, 2\} \quad (\text{E25.2})$$

Write a Mathematica module `NodeFreedomCountTable` that receives the FAT as only argument and returns NFCT. Test the module for the frame-truss example structure. Hint: a one-liner implementation is

```
NodeFreedomCountTable[FAT_] := Table[Sum[FAT[[n, i]], {i, 1, 6}], {n, 1, Length[FAT]}];
```

but this is difficult to figure out. Try a more readable form.

EXERCISE 25.3

[A+C:15] The Node Freedom Map Table, or NFMT, is an array with one entry per node. Entry $\text{NFMT}[[n]] = i$ with $i \geq 0$, if the global freedoms associated with node n are $i+1, \dots, i+k$, where $k = \text{NFCT}[[n]]$. If node n is undefined, $\text{NFMT}[[n]] = -1$.² For example, the NFMT for the frame-truss structure of Figure 25.4 is

$$\text{NFMT} = \{0, 3, 6, 9\} \quad (\text{E25.3})$$

Write a Mathematica module `NodeFreedomMapTable` that receives the NFCT as only argument and returns NFMT. Test the module for this example structure. Hint: the logic can be copied from code in Chapter 25 if you know where to look.

¹ This is actually the scheme implemented in the NASTRAN commercial FEM code

² Other schemes are used in FEM codes; this is just one of several possible choices.

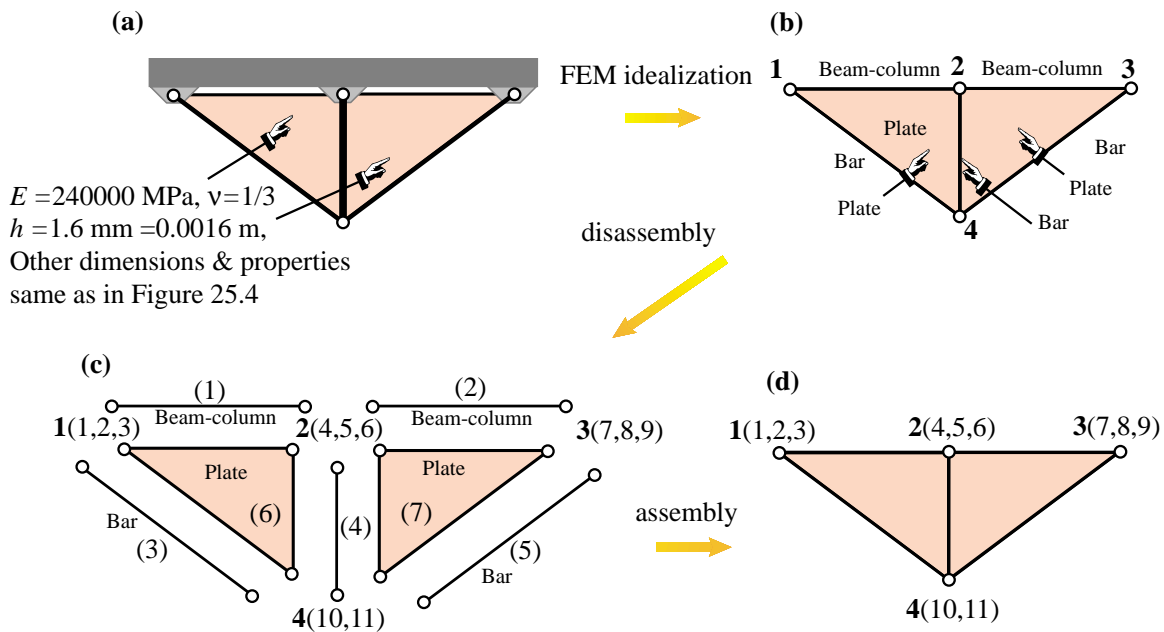


Figure E25.1. Plate-reinforced frame-truss structure for Exercise 25.4.

EXERCISE 25.4

[A/C:25] The frame-truss structure of Figure 25.4 is reinforced with two triangular steel plates attached as shown in Figure E25.1(a). The plate thickness is $1.6 \text{ mm} = 0.0016 \text{ m}$; the material is isotropic with $E = 240000 \text{ MPa}$ and $\nu = 1/3$. The reinforcing plates are modeled with two plane stress 3-node linear triangles numbered (6) and (7), as illustrated in Figure E25.1(b,c). Compute the master stiffness matrix \mathbf{K} of the structure.³

This exercise may be done through *Mathematica*. For this download Notebook `ExampleAssemblers.nb` from this Chapter index, and complete Cell 3 by writing the assembler.⁴ The plate element identifier is "Trig3". Debugging hint: check that matrix (25.32) is obtained if the plate thickness h_{plate} is (temporarily) set to zero.

Target:

$$\mathbf{K} = \begin{bmatrix} 337.6 & -19.2 & 0 & -312. & -72. & 0 & 0 & 0 & 0 & -25.6 & 91.2 \\ -19.2 & 90.9 & 45. & -72. & -76.5 & 45. & 0 & 0 & 0 & 91.2 & -14.4 \\ 0 & 45. & 120. & 0 & -45. & 60. & 0 & 0 & 0 & 0 & 0 \\ -312. & -72. & 0 & 816. & 0 & 0 & -312. & 72. & 0 & -192. & 0 \\ -72. & -76.5 & -45. & 0 & 929. & 0 & 72. & -76.5 & 45. & 0 & -776. \\ 0 & 45. & 60. & 0 & 0 & 240. & 0 & -45. & 60. & 0 & 0 \\ 0 & 0 & 0 & -312. & 72. & 0 & 337.6 & 19.2 & 0 & -25.6 & -91.2 \\ 0 & 0 & 0 & 72. & -76.5 & -45. & 19.2 & 90.9 & -45. & -91.2 & -14.4 \\ 0 & 0 & 0 & 0 & 45. & 60. & 0 & -45. & 120. & 0 & 0 \\ -25.6 & 91.2 & 0 & -192. & 0 & 0 & -25.6 & -91.2 & 0 & 243.2 & 0 \\ 91.2 & -14.4 & 0 & 0 & -776. & 0 & -91.2 & -14.4 & 0 & 0 & 804.8 \end{bmatrix} \quad (\text{E25.4})$$

³ This structure would violate the compatibility requirements stated in Chapter 19 unless the beams are allowed to deflect laterally independently of the plates. This is the fabrication sketched in Figure E25.1(a).

⁴ Cells 1 and 2 contain working assemblers for plane truss and plane frames, respectively, which may be used as templates.

26

Solving FEM Equations

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§26.1. MOTIVATION FOR SPARSE SOLVERS

In the Direct Stiffness Method (DSM) of finite element analysis, the element stiffness matrices and consistent nodal force vectors are immediately assembled to form the *master stiffness matrix* and *master force vector*, respectively, by the process called *merge*. The basic rules that govern the assembly process are described in Chapter 25. For simplicity the description that follows assumes that no MultiFreedom Constraints (MFCs) are present.

The end result of the assembly process are the master stiffness equations

$$\mathbf{K}\mathbf{u} = \mathbf{f} \quad (26.1)$$

where \mathbf{K} is the master stiffness matrix, \mathbf{f} the vector of node forces and \mathbf{u} the vector of node displacements. Upon imposing the displacement boundary conditions, the system (26.1) is solved for the unknown node displacements. The solution concludes the main phase of DSM computations.

In practical applications the order of the stiffness system (26.1) can be quite large. Systems of order 1000 to 10000 are routinely solved in commercial software. Larger ones (say up to 100000 equations) are not uncommon and even millions of equations are being solved on supercomputers. Presently the record is about 50 million on parallel computers.¹

In *linear* FEM analysis the cost of solving this system of equations rapidly overwhelms other computational phases. Much attention has therefore been given to matrix processing techniques that economize storage and solution time by taking advantage of the special structure of the stiffness matrix.

The master force vector is stored as a conventional one-dimensional array of length equal to the number N of degrees of freedom. This storage arrangement presents no particular difficulties even for very large problem sizes. Handling the master stiffness matrix, however, presents computational difficulties.

§26.1.1. The Curse of Fullness

If \mathbf{K} is stored and processed as if it were a *full* matrix, the storage and processing time resources rapidly become prohibitive as N increases. This is illustrated in Table 26.1, which summarizes the storage and factor-time requirements for orders $N = 10^4$, 10^5 and 10^6 .²

As regards memory needs, a full square matrix stored without taking advantage of symmetry, requires storage for N^2 entries. If each entry is an 8-byte, double precision floating-point number, the required storage is $8N^2$ bytes. Thus, a matrix of order $N = 10^4$ would require 8×10^8 bytes or 800 MegaBytes (MB) for storage.

For large N the solution of (26.1) is dominated by the factorization of \mathbf{K} , an operation discussed in §26.2. This operation requires approximately $N^3/6$ floating point operation units. [A floating-point operation unit is conventionally defined as a (multiply,add) pair plus associated indexing and data movement operations.] Now a fast workstation can typically do 10^7 of these operations per second, whereas a supercomputer may be able to sustain 10^9 or more. These times assume that the entire

¹ For fluid equations solved by fluid-volume methods the number is over 100 million.

² The factor times given in that table reflect 1998 computer technology. To update to 2003, divide times by 10 to 20.

Table 26.1 Storage & Solution Time for a Fully-Stored Stiffness Matrix

Matrix order N	Storage (double prec)	Factor op.units (FLOPS)	Factor time workstation (or fast PC)	Factor time supercomputer
10^4	800 MB	$10^{12}/6$	3 hrs	2 min
10^5	80 GB	$10^{15}/6$	4 mos	30 hrs
10^6	8 TB	$10^{18}/6$	300 yrs	3 yrs

Table 26.2 Storage & Solution Time for a Skyline Stored Stiffness Matrix
Assuming $B = \sqrt{N}$

Matrix order N	Storage (double prec)	Factor op.units (FLOPS)	Factor time workstation (or fast PC)	Factor time supercomputer
10^4	8 MB	$10^8/2$	5 sec	0.05 sec
10^5	240 MB	$10^{10}/2$	8 min	5 sec
10^6	8000 MB	$10^{12}/2$	15 hrs	8 min

matrix is kept in high-speed memory; for otherwise the elapsed time may increase by factors of 10 or more due to I/O transfer operations. The elapsed times estimated given in Table 26.1 illustrate that for present computer resources, orders above 10^4 would pose significant computational difficulties.

§26.1.2. The Advantages of Sparsity

Fortunately a very high percentage of the entries of the master stiffness matrix \mathbf{K} are zero. Such matrices are called *sparse*. There are clever programming techniques that take advantage of sparsity that fit certain patterns. Although a comprehensive coverage of such techniques is beyond the scope of this course, we shall concentrate on a particular form of sparse scheme that is widely used in FEM codes: *skyline* storage. This scheme is simple to understand, manage and implement, while cutting storage and processing times by orders of magnitude as the problems get larger.

The skyline storage format is a generalization of its widely used predecessor called the *band storage* scheme. A matrix stored in accordance with the skyline format will be called a *skymatrix* for short. Only symmetric skymatrices will be considered here, since the stiffness matrices in linear FEM are symmetric.

If a skymatrix of order N can be stored in S memory locations, the ratio $B = S/N$ is called the *mean bandwidth*. If the entries are, as usual, 8-byte double-precision floating-point numbers, the storage requirement is $8NB$ bytes. The factorization of a skymatrix requires approximately $\frac{1}{2}NB^2$ floating-point operation units. In two-dimensional problems B is of the order of \sqrt{N} . Under this assumption, storage requirements and estimated factorization times for $N = 10^4$, $N = 10^5$ and $N = 10^6$ are reworked in Table 26.2. It is seen that by going from full to skyline storage significant

reductions in computer resources have been achieved. For example, now $N = 10^4$ is easy on a workstation and trivial on a supercomputer. Even a million equations do not look far-fetched on a supercomputer as long as enough memory is available.

In preparation for assembling \mathbf{K} as a skymatrix one has to set up several auxiliary arrays related to nodes and elements. These auxiliary arrays are described in the previous Chapter. Knowledge of that material is useful for understanding the following description.

§26.2. SPARSE SOLUTION OF STIFFNESS EQUATIONS

§26.2.1. Skyline Storage Format

The skyline storage arrangement for \mathbf{K} is best illustrated through a simple example. Consider the 6×6 stiffness matrix

$$\mathbf{K} = \begin{bmatrix} K_{11} & 0 & K_{13} & 0 & 0 & K_{16} \\ & K_{22} & 0 & K_{24} & 0 & 0 \\ & & K_{33} & K_{34} & 0 & 0 \\ & & & K_{44} & 0 & K_{46} \\ & & & & K_{55} & K_{56} \\ \text{symm} & & & & & K_{66} \end{bmatrix} \quad (26.2)$$

Since the matrix is symmetric only one half, the upper triangle in the above display, need to be shown.

Next we define the *envelope* of \mathbf{K} as follows. From each diagonal entry move *up* the corresponding column until the last nonzero entry is found. The envelope separates that entry from the rest of the upper triangle. The remaining zero entries are conventionally removed:

$$\mathbf{K} = \begin{bmatrix} K_{11} & & K_{13} & & K_{16} \\ & K_{22} & 0 & K_{24} & 0 \\ & & K_{33} & K_{34} & 0 \\ & & & K_{44} & K_{46} \\ & & & & K_{55} & K_{56} \\ \text{symm} & & & & & K_{66} \end{bmatrix} \quad (26.3)$$

What is left constitute the *skyline profile* of *skyline template* of the matrix. A sparse matrix that can be profitably stored in this form is called a *skymatrix* for brevity. Notice that the skyline profile may include zero entries. During the factorization step discussed below these zero entries will in general become nonzero, a phenomenon that receives the name *fill-in*.

The key observation is that only *the entries in the skyline template need to be stored*, because *fill-in in the factorization process will not occur outside the envelope*. To store these entries it is convenient to use a one-dimensional *skyline array*:

$$\mathbf{s} : [K_{11}, K_{22}, K_{13}, 0, K_{33}, K_{24}, K_{34}, K_{44}, K_{55}, K_{16}, 0, 0, K_{46}, K_{56}, K_{66}] \quad (26.4)$$

This array is complemented by a $(N + 1)$ integer array \mathbf{p} that contains addresses of *diagonal locations*. The array has $N + 1$ entries. The $(i + 1)^{th}$ entry of \mathbf{p} has the location of the i^{th} diagonal

entry of \mathbf{K} in \mathbf{s} . For the example matrix:

$$\mathbf{p} : [0, 1, 2, 5, 8, 9, 15] \quad (26.5)$$

In the previous Chapter, this array was called the Global Skyline Diagonal Location Table, or GSDLT. Equations for which the displacement component is prescribed are identified by a *negative* diagonal location value. For example if u_3 and u_5 are prescribed displacement components in the test example, then

$$\mathbf{p} : [0, 1, 2, -5, 8, -9, 15] \quad (26.6)$$

REMARK 26.1

In Fortran it is convenient to dimension the diagonal location array as $\mathbf{p}(0:n)$ so that indexing begins at zero. In C this is the standard indexing.

§26.2.2. Factorization

The stiffness equations (26.1) are solved by a direct method that involves two basic phases: *factorization* and *solution*.

In the first stage, the skyline-stored symmetric stiffness matrix is factored as

$$\mathbf{K} = \mathbf{LDU} = \mathbf{LDL}^T = \mathbf{U}^T \mathbf{DU}, \quad (26.7)$$

where \mathbf{L} is a unit lower triangular matrix, \mathbf{D} is a nonsingular diagonal matrix, and \mathbf{U} and \mathbf{L} are the transpose of each other. The original matrix is overwritten by the entries of \mathbf{D}^{-1} and \mathbf{U} ; details may be followed in the program implementation. No pivoting is used in the factorization process. This factorization is carried out by *Mathematica* module `SymmSkyMatrixFactor`, which is described later in this Chapter.

§26.2.3. Solution

Once \mathbf{K} has been factored, the solution \mathbf{u} for a given right hand side \mathbf{f} is obtained by carrying out three stages:

$$\text{Forward reduction :} \quad \mathbf{Lz} = \mathbf{f}, \quad (26.8)$$

$$\text{Diagonal scaling :} \quad \mathbf{Dy} = \mathbf{z}, \quad (26.9)$$

$$\text{Back substitution :} \quad \mathbf{Uu} = \mathbf{y}, \quad (26.10)$$

where \mathbf{y} and \mathbf{z} are intermediate vectors. These stages are carried out by *Mathematica* modules `SymmSkyMatrixVectorSolve`, which is described later.

§26.2.4. Treating MFCs with Lagrange Multipliers

In *Mathematica* implementations of FEM, MultiFreedom Constraints (MFCs) are treated with Lagrange multipliers. There is one multiplier for each constraint. The multipliers are placed at the end of the solution vector.

Specifically, let the $\text{nummul} > 0$ MFCs be represented in matrix form as $\mathbf{C}\mathbf{u} = \mathbf{g}$, where \mathbf{C} and \mathbf{g} are given, and let the nummul multipliers be collected in a vector $\boldsymbol{\lambda}$. The multiplier-augmented master stiffness equations are

$$\begin{bmatrix} \mathbf{K} & \mathbf{C}^T \\ \mathbf{C} & \mathbf{0} \end{bmatrix} \begin{bmatrix} \mathbf{u} \\ \boldsymbol{\lambda} \end{bmatrix} = \begin{bmatrix} \mathbf{f} \\ \mathbf{g} \end{bmatrix} \quad (26.11)$$

or

$$\mathbf{A}\mathbf{x} = \mathbf{b}. \quad (26.12)$$

where the symmetric matrix \mathbf{A} , called a stiffness-bordered matrix, is of order $\text{numdof} + \text{nummul}$.

The stiffness bordered matrix is also stored in skyline form, and the previous solution procedure applies, as long as the skyline array is properly constructed as described in the previous Chapter.

The main difference with respect to the no-MFC case is that, because of the configuration (26.11), \mathbf{A} can no longer be positive definite. In principle pivoting should be used during the factorization of \mathbf{A} to forestall possible numerical instabilities. Pivoting can have a detrimental effect on solution efficiency because entries can move outside of the skyline template. However, by placing the $\boldsymbol{\lambda}$ at the end such difficulties will not be encountered if \mathbf{K} is positive definite, and the constraints are linearly independent (that is, \mathbf{C} has full rank). Thus pivoting is not necessary.

§26.3. A SKYSOLVER IMPLEMENTATION

The remaining sections of this revised Chapter describe a recent implementation of the skyline solver and related routines in *Mathematica*. This has been based on similar Fortran codes used since 1967.

§26.3.1. Skymatrix Representation

In what follows the computer representation in *Mathematica* of a symmetric skymatrix will be generally denoted by the symbol S . Such a representation consists of a list of two numeric objects:

$$S = \{ p, s \} \quad (26.13)$$

Here $p = \text{GSDLT}$ is the Global Skyline Diagonal Location Table introduced in §11.6, and s is the array of skymatrix entries, arranged as described in the previous section. This array usually consists of floating-point numbers, but it may also contain exact integers or fractions, or even symbolic entries.

For example, suppose that the numerical entries of the 6×6 skymatrix (26.10) are actually

$$\mathbf{K} = \begin{bmatrix} 11 & & 13 & & 16 \\ & 22 & 0 & 24 & 0 \\ & & 33 & 34 & 0 \\ & & & 44 & 46 \\ & & & & 55 & 56 \\ \text{symm} & & & & & 66 \end{bmatrix} \quad (26.14)$$

Cell 26.1 Factorization of a Symmetric SkyMatrix

```

SymmSkyMatrixFactor[S_,tol_]:= Module[
  {p,a,fail,i,j,k,l,m,n,ii,ij,jj,jk,jmj,d,s,row,v},
  row=SymmSkyMatrixRowLengths[S]; s=Max[row];
  {p,a}=S; n=Length[p]-1; v=Table[0,{n}]; fail=0;
  Do [jj=p[[j+1]]; If [jj<0|row[[j]]==0, Continue[]]; d=a[[jj]];
    jmj=Abs[p[[j]]]; jk=jj-jmj;
    Do [i=j-jk+k; v[[k]]=0; ii=p[[i+1]];
      If [ii<0, Continue[]]; m=Min[ii-Abs[p[[i]]],k]-1;
      ij=jmj+k; v[[k]]=a[[ij]];
      v[[k]]-=Take[a,{ii-m,ii-1}].Take[v,{k-m,k-1}];
      a[[ij]]=v[[k]]*a[[ii]],
    {k,1,jk-1}];
  d-=Take[a,{j+1,jmj+jk-1}].Take[v,{1,jk-1}];
  If [Abs[d]<tol*row[[j]], fail=j; a[[jj]]=Infinity; Break[] ];
  a[[jj]]=1/d,
  {j,1,n}];
  Return[{{p,a},fail}]
];

SymmSkyMatrixRowLengths[S_]:= Module[
  {p,a,i,j,n,ii,jj,m,d,row},
  {p,a}=S; n=Length[p]-1; row=Table[0,{n}];
  Do [ii=p[[i+1]]; If [ii<0, Continue[]]; m=ii-i; row[[i]]=a[[ii]]^2;
    Do [If [p[[j+1]]>0, d=a[[m+j]]^2; row[[i]]+=d; row[[j]]+=d,
      {j,Max[1,Abs[p[[i]]]-m+1],Min[n,i]-1}],
  {i,1,n}]; Return[Sqrt[row]];
];

```

Its *Mathematica* representation, using the symbols (26.13) is

$$\begin{aligned}
 p &= \{ 0, 1, 2, 5, 8, 9, 15 \}; \\
 s &= \{ 11, 22, 13, 0, 33, 24, 34, 44, 55, 16, 0, 0, 46, 56, 66 \}; \\
 S &= \{ p, s \};
 \end{aligned}
 \tag{26.15}$$

or more directly

$$S = \{ \{ 0, 1, 2, 5, 8, 9, 15 \}, \{ 11, 22, 13, 0, 33, 24, 34, 44, 55, 16, 0, 0, 46, 56, 66 \} \};
 \tag{26.16}$$

[The remaining sections on details of skyline processing logic, marked with a *, will not be covered in class. They are intended for a more advanced course.]

Cell 26.2 Factorization Test Input

```

ClearAll[n]; n=5; SeedRandom[314159];
p=Table[0,{n+1}]; Do[p[[i+1]]=p[[i]]+
    Max[1,Min[i,Round[Random[]*i]]],{i,1,n}];
a=Table[1.,{i,1,p[[n+1]]}];
Print["Mean Band=",N[p[[n+1]]/n]];
S={p,a};
Sr=SymmSkyMatrixLDUReconstruct[S];
Print["Reconstructed SkyMatrix:"]; SymmSkyMatrixLowerTrianglePrint[Sr];
SymmSkyMatrixLowerTriangleMap[Sr];
Print["eigs=",Eigenvalues[SymmSkyMatrixConvertToFull[Sr]]];
x=Table[{N[i],3.,(-1)^i*N[n-i]},{i,1,n}];
Print["Assumed x=",x];
b=SymmSkyMatrixColBlockMultiply[Sr,x];
(*x=Transpose[x]; b=SymmSkyMatrixRowBlockMultiply[Sr,x];*)
Print["b=Ax=",b];
Print[Timing[{F,fail}=SymmSkyMatrixFactor[Sr,10.^(-12)]]];
If [fail!=0, Print["fail=",fail]; Abort[]];
Print["F=",F]; Print["fail=",fail];
Print["Factor:"];
SymmSkyMatrixLowerTrianglePrint[F];
x=SymmSkyMatrixColBlockSolve[F,b];
(*x=SymmSkyMatrixRowBlockSolve[F,b];*)
Print["Computed x=",x//InputForm];

```

§26.3.2. *Skymatrix Factorization

Module `SymmSkyMatrixFactor`, listed in Cell 26.1, factors a symmetric skymatrix into the product **LDU** where **L** is the transpose of **U**. No pivoting is used. The module is invoked as

$$\{ Sf, fail \} = \text{SymmSkyMatrixFactor}[S, tol]$$

The input arguments are

- | | |
|-----|---|
| S | The skymatrix to be factored, stored as the two-object list $\{p, s\}$; see previous subsection. |
| tol | Tolerance for singularity test. The appropriate value of <code>tol</code> depends on the kind of skymatrix entries stored in <code>s</code> . |

If the skymatrix entries are floating-point numbers handled by default in double precision arithmetic, `tol` should be set to $8\times$ or $10\times$ the machine precision in that kind of arithmetic. The factorization aborts if, when processing the j -th row, $d_j \leq tol * r_j$, where d_j is the computed j^{th} diagonal entry of **D**, and r_j is the Euclidean norm of the j^{th} skymatrix row.

If the skymatrix entries are exact (integers, fractions or symbols), `tol` should be set to zero. In this case exact singularity is detectable, and the factorization aborts only on that condition.

The outputs are:

Cell 26.3 Output from Program of Cells 26.1 and 26.2

```

Mean Band=1.6
Reconstructed SkyMatrix:

      Col 1   Col 2   Col 3   Col 4   Col 5
Row 1   1.0000
Row 2           1.0000
Row 3           1.0000   2.0000
Row 4                   1.0000
Row 5                   1.0000   3.0000

      1 2 3 4 5

1  +
2  +
3  + +
4  +
5  + + +
eigs={3.9563, 2.20906, 1., 0.661739, 0.172909}
Assumed x={{1., 3., -4.}, {2., 3., 3.}, {3., 3., -2.}, {4., 3., 1.},
           {5., 3., 0.}}
b=Ax={{1., 3., -4.}, {5., 6., 1.}, {13., 12., -1.}, {9., 6., 1.},
      {22., 15., -1.}}
{0.0666667 Second, {{0, 1, 2, 4, 5, 8},
                    {1., 1., 1., 1., 1., 1., 1., 1.}}, 0}}
F={{0, 1, 2, 4, 5, 8}, {1., 1., 1., 1., 1., 1., 1., 1.}}
fail=0
Factor:

      Col 1   Col 2   Col 3   Col 4   Col 5
Row 1   1.0000
Row 2           1.0000
Row 3           1.0000   1.0000
Row 4                   1.0000
Row 5                   1.0000   1.0000
Computed x={{1., 3., -4.}, {2., 3., 3.}, {3., 3., -2.}, {4., 3., 1.},
           {5., 3., 0.}}

```

Sf If fail is zero on exit, Sf is the computed factorization of S. It is a two-object list {p, du}, where du stores the entries of \mathbf{D}^{-1} in the diagonal locations, and of \mathbf{U} in its strict upper triangle.

fail A singularity detection indicator. A zero value indicates that no singularity was detected. If

Cell 26.4 Solving for a Single RHS

```

SymmSkyMatrixVectorSolve[S_,b_]:= Module[
  {p,a,n,i,j,k,m,ii,jj,bi,x},
  {p,a}=S; n=Length[p]-1; x=b;
  If [n!=Length[x], Print["Inconsistent matrix dimensions in",
    " SymmSkyMatrixVectorSolve"]; Return[Null]];
  Do [ii=p[[i+1]];If [ii>=0, Continue[]]; ii=-ii; k=i-ii+Abs[p[[i]]]+1;
    bi=x[[i]]; If [bi==0, Continue[]];
    Do [jj=p[[j+1]], If [jj<0, Continue[]];
      m=j-i; If [m<0, x[[j]]-=a[[ii+m]]*bi; Break[]];
      ij=jj-m; If [ij>Abs[p[[j]]], x[[j]]-=a[[ij]]*bi],
      {j,k,n}],
    {i,1,n}];
  Do [ii=p[[i+1]]; If [ii<0, x[[i]]=0; Continue[]];
    imi=Abs[p[[i]]]; m=ii-imi-1;
    x[[i]]-=Take[a,{imi+1,imi+m}].Take[x,{i-m,i-1}],
    {i,1,n}];
  Do [ii=Abs[p[[i+1]]]; x[[i]]*=a[[ii]], {i,1,n}];
  Do [ii=p[[i+1]]; If [ii<0, x[[i]]=b[[i]]; Continue[]];
    m=ii-Abs[p[[i]]]-1;
    Do [ x[[i-j]]-=a[[ii-j]]*x[[i]], {j,1,m}],
    {i,n,1,-1}];
  Return[x]
];

```

fail returns $j > 0$, the factorization was aborted at the j -th row. In this case Sf returns the aborted factorization with ∞ stored in d_j .

A test of `SymmSkyMatrixFactor` on the matrix (26.14) is shown in Cells 26.2 and 26.3. The modules that print and produce skyline maps used in the test program are described later in this Chapter.

§26.3.3. *Solving for One or Multiple RHS

Module `SymmSkyMatrixVectorSolve`, listed in Cell 26.4, solves the linear system $\mathbf{Ax} = \mathbf{b}$ for \mathbf{x} , following the factorization of the symmetric skymatrix \mathbf{A} by `SymmSkyMatrixFactor`. The module is invoked as

$$\mathbf{x} = \text{SymmSkyMatrixVectorSolve}[Sf, \mathbf{b}]$$

The input arguments are

- | | |
|--------------|--|
| Sf | The factored matrix returned by <code>SymmSkyMatrixFactor</code> . |
| \mathbf{b} | The right-hand side vector to be solved for, stored as a single-level (one dimensional) list. If the i -th entry of \mathbf{x} is prescribed, the known value must be supplied in this vector. |

The outputs are:

- | | |
|--------------|---|
| \mathbf{x} | The computed solution vector, stored as a single-level (one-dimensional) list. Prescribed solution components return the value of the entry in \mathbf{b} . |
|--------------|---|

Cell 26.5 Solving for a Block of Right Hand Sides

```

SymmSkyMatrixColBlockSolve[S_,b_]:= Module[
  {p,a,n,nrhs,i,j,k,m,r,ii,jj,bi,x},
  {p,a}=S; n=Length[p]-1; x=b;
  If [n!=Dimensions[x][[1]], Print["Inconsistent matrix dimensions in",
    " SymmSkyMatrixBlockColSolve"]; Return[Null]]; nrhs = Dimensions[x][[2]];
  Do [ii=p[[i+1]];If [ii>0, Continue[]]; ii=-ii; k=i-ii+Abs[p[[i]]]+1;
    Do [bi=x[[i,r]]; If [bi==0, Continue[]];
      Do [jj=p[[j+1]], If [jj<0, Continue[]];
        m=j-i; If [m<0,x[[j,r]]-=a[[ii+m]]*bi; Break[]];
        ij=jj-m; If [ij>Abs[p[[j]]], x[[j,r]]-=a[[ij]]*bi],
        {j,k,n}],
    {r,1,nrhs}],
  {i,1,n}];
  Do [ii=p[[i+1]]; If [ii<0, Do[x[[i,r]]=0,{r,1,nrhs}];Continue[]];
    imi=Abs[p[[i]]]; m=ii-imi-1;
    Do [ Do [ x[[i,r]]-=a[[imi+j]]*x[[i-m+j-1,r]], {j,1,m}], {r,1,nrhs}],
    {i,1,n}];
  Do [ii=Abs[p[[i+1]]]; Do[x[[i,r]]*=a[[ii]], {r,1,nrhs}], {i,1,n}];
  Do [ii=p[[i+1]]; If [ii<0, Do[x[[i,r]]=b[[i,r]],{r,1,nrhs}];Continue[]];
    m=ii-Abs[p[[i]]]-1;
    Do [ Do [ x[[i-j,r]]-=a[[ii-j]]*x[[i,r]], {j,1,m}], {r,1,nrhs}],
    {i,n,1,-1}];
  Return[x]
];

SymmSkyMatrixRowBlockSolve[S_,b_]:= Module[
  {p,a,n,nrhs,i,j,k,m,r,ii,jj,bi,x},
  {p,a}=S; n=Length[p]-1; x=b;
  If [n!=Dimensions[x][[2]], Print["Inconsistent matrix dimensions in",
    " SymmSkyMatrixBlockRowSolve"]; Return[Null]]; nrhs = Dimensions[x][[1]];
  Do [ii=p[[i+1]];If [ii>0, Continue[]]; ii=-ii; k=i-ii+Abs[p[[i]]]+1;
    Do [bi=x[[r,i]]; If [bi==0, Continue[]];
      Do [jj=p[[j+1]], If [jj<0, Continue[]];
        m=j-i; If [m<0,x[[j,r]]-=a[[ii+m]]*bi; Break[]];
        ij=jj-m; If [ij>Abs[p[[j]]], x[[r,j]]-=a[[ij]]*bi],
        {j,k,n}],
      {r,1,nrhs}],
    {i,1,n}];
  Do [ii=p[[i+1]]; If [ii<0, Do[x[[r,i]]=0,{r,1,nrhs}];Continue[]];
    imi=Abs[p[[i]]]; m=ii-imi-1;
    Do [ Do [ x[[r,i]]-=a[[imi+j]]*x[[r,i-m+j-1]], {j,1,m}], {r,1,nrhs}],
    {i,1,n}];
  Do [ii=Abs[p[[i+1]]]; Do[x[[r,i]]*=a[[ii]], {r,1,nrhs}], {i,1,n}];
  Do [ii=p[[i+1]]; If [ii<0, Do[x[[r,i]]=b[[r,i]],{r,1,nrhs}];Continue[]];
    m=ii-Abs[p[[i]]]-1;
    Do [ Do [ x[[r,i-j]]-=a[[ii-j]]*x[[r,i]], {j,1,m}], {r,1,nrhs}],
    {i,n,1,-1}];
  Return[x]
];

```

Cell 26.6 Multiplying Skymatrix by Individual Vector

```

SymmSkyMatrixVectorMultiply[S_,x_]:= Module[
  {p,a,n,i,j,k,m,ii,b},
  {p,a}=S; n=Length[p]-1;
  If [n!=Length[x], Print["Inconsistent matrix dimensions in",
    " SymmSkyMatrixVectorMultiply"]; Return[Null]];
  b=Table[a[[ Abs[p[[i+1]]] ]]*x[[i]], {i,1,n}];
  Do [ii=Abs[p[[i+1]]]; m=ii-Abs[p[[i]]]-1; If [m<=0,Continue[]];
    b[[i]]+=Take[a,{ii-m,ii-1}].Take[x,{i-m,i-1}];
    Do [b[[i-k]]+=a[[ii-k]]*x[[i]],{k,1,m}],
  {i,1,n}];
  Return[b]
];

(*
ClearAll[n]; n=10; SeedRandom[314159];
p=Table[0,{n+1}]; Do[p[[i+1]]=p[[i]]+
  Max[1,Min[i,Round[Random[]*i]]],{i,1,n}];
a=Table[1.,{i,1,p[[n+1]]}];
Print["Mean Band=",N[p[[n+1]]/n]];
S={p,a};
Sr=SymmSkyMatrixLDUReconstruct[S];
Print["Reconstructed SkyMatrix:"]; SymmSkyMatrixLowerTrianglePrint[Sr];
SymmSkyMatrixLowerTriangleMap[Sr];
x=Table[1.,{i,1,n}];
b=SymmSkyMatrixVectorMultiply[Sr,x];
Print["b=Ax=",b];*)

```

Sometimes it is necessary to solve linear systems for multiple ($m > 1$) right hand sides. One way to do that is to call `SymmSkyMatrixVectorSolve` repeatedly. Alternatively, if m right hand sides are collected as columns of a rectangular matrix **B**, module `SymmSkyMatrixColBlockSolve` may be invoked as

$$\mathbf{X} = \text{SymmSkyMatrixVectorSolve}[\mathbf{Sf}, \mathbf{B}]$$

to provide the solution **X** of $\mathbf{SX} = \mathbf{B}$. This module is listed in Cell 26.5. The input arguments and function returns have the same function as those described for `SymmSkyMatrixVectorSolve`. The main difference is that **B** and **X** are matrices (two-dimensional lists) with the right-hand side and solution vectors as columns. There is a similar module `SymmSkyMatrixRowBlockSolve`, notlisted here, which solves for multiple right hand sides stored as rows of a matrix.

§26.3.4. *Matrix-Vector Multiply

For various applications it is necessary to form the matrix-vector product

$$\mathbf{b} = \mathbf{Sx} \tag{26.17}$$

where **S** is a symmetric skymatrix and **x** is given.

Cell 26.7 Multiplying Skymatrix by Vector Block

```

SymmSkyMatrixColBlockMultiply[S_,x_]:= Module[
  {p,a,n,nrhs,i,j,k,m,r,ii,aij,b},
  {p,a}=S; n=Length[p]-1;
  If [n!=Dimensions[x][[1]], Print["Inconsistent matrix dimensions in",
    " SymmSkyMatrixColBlockMultiply"]; Return[Null]];
  nrhs = Dimensions[x][[2]]; b=Table[0,{n},{nrhs}];
  Do [ii=Abs[p[[i+1]]]; m=ii-Abs[p[[i]]]-1;
    Do [b[[i,r]]=a[[ii]]*x[[i,r]], {r,1,nrhs}];
    Do [j=i-k; aij=a[[ii-k]]; If [aij==0, Continue[]];
      Do [b[[i,r]]+=aij*x[[j,r]]; b[[j,r]]+=aij*x[[i,r]], {r,1,nrhs}],
      {k,1,m}],
  {i,1,n}];
  Return[b]
];

SymmSkyMatrixRowBlockMultiply[S_,x_]:= Module[
  {p,a,n,nrhs,i,j,k,m,r,ii,aij,b},
  {p,a}=S; n=Length[p]-1;
  If [n!=Dimensions[x][[2]], Print["Inconsistent matrix dimensions in",
    " SymmSkyMatrixRowBlockMultiply"]; Return[Null]];
  nrhs = Dimensions[x][[1]]; b=Table[0,{nrhs},{n}];
  Do [ii=Abs[p[[i+1]]]; m=ii-Abs[p[[i]]]-1;
    Do [b[[r,i]]=a[[ii]]*x[[r,i]], {r,1,nrhs}];
    Do [j=i-k; aij=a[[ii-k]]; If [aij==0, Continue[]];
      Do [b[[r,i]]+=aij*x[[r,j]]; b[[r,j]]+=aij*x[[r,i]], {r,1,nrhs}],
      {k,1,m}],
  {i,1,n}];
  Return[b]
];

```

This is done by module `SymmSkyMatrixVectorMultiply`, which is listed in Cell 26.6. Its arguments are the skymatrix S and the vector x . The function returns Sx in b .

Module `SymmSkyMatrixColBlockMultiply` implements the multiplication by a block of vectors stored as columns of a rectangular matrix X :

$$B = SX \quad (26.18)$$

This module is listed in Cell 26.7. Its arguments are the skymatrix S and the rectangular matrix X . The function returns SX in B .

There is a similar module `SymmSkyMatrixRowBlockMultiply`, also listed in Cell 26.7, which postmultiplies a vector block stored as rows.

§26.3.5. *Printing and Mapping

Module `SymmSkyMatrixUpperTrianglePrint`, listed in Cell 26.8, prints a symmetric skymatrix in upper triangle form. Is is invoked as

`SymmSkyMatrixUpperTrianglePrint[S]`

where S is the skymatrix to be printed. For an example of use see Cells 262-3.

The print format resembles the configuration depicted in Section 26.1. This kind of print is useful for program development and debugging although of course it should not be attempted with a very large matrix.³

There is a similar module called `SymmSkyMatrixLowerTrianglePrint`, which displays the skymatrix entries in lower triangular form. This module is also listed in Cell 26.8.

Sometimes one is not interested in the actual values of the skymatrix entries but only on how the skyline template looks like. Such displays, called *maps*, can be done with just one symbol per entry. Module `SymmSkyMatrixUpperTriangleMap`, listed in Cell 26.9, produces a map of its argument. It is invoked as

`SymmSkyMatrixUpperTriangleMap[S]`

The entries within the skyline template are displayed by symbols +, - and 0, depending on whether the value is positive, negative or zero, respectively. Entries outside the skyline template are blank.

As in the case of the print module, there is module `SymmSkyMatrixLowerTriangleMap` which is also listed in Cell 26.9.

§26.3.6. *Reconstruction of SkyMatrix from Factors

In testing factorization and solving modules it is convenient to have modules that perform the “inverse process” of the factorization. More specifically, suppose that U , D (or D^{-1}) are given, and the problem is to reconstruct the skymatrix that have them as factors:

$$S = LDU, \quad \text{or} \quad S = LD^{-1}U \quad (26.19)$$

in which $L = U^T$. Modules `SymmSkyMatrixLDUReconstruction` and `SymmSkyMatrixLDinvUReconstruction` perform those operations. These modules are listed in Cell 26.10. Their argument is a factored form of S : U and D in the first case, and U and D^{-1} in the second case.

³ Computer oriented readers may notice that the code for the printing routine is substantially more complex than those of the computational modules. This is primarily due to the inadequacies of *Mathematica* in handling tabular format output. The corresponding Fortran or C implementations would be simpler because those languages provide much better control over low-level display.

Cell 26.8 Skymatrix Printing

```

SymmSkyMatrixLowerTrianglePrint[S_]:= Module[
  {p,a,cycle,i,ii,ij,it,j,jj,j1,j2,jref,jbeg,jend,jt,kcmax,kc,kr,m,n,c,t},
  {p,a}=S; n=Dimensions[p][[1]]-1; kcmax=5; jref=0;
  Label[cycle]; Print[" "];
  jbeg=jref+1; jend=Min[jref+kcmax,n]; kc=jend-jref;
  t=Table[" ",{n-jref+1},{kc+1}];
  Do [If [p[[j+1]]>0,c=" ",c="*"];
    t[[1,j-jref+1]]=StringJoin[c,"Col",ToString[PaddedForm[j,3]]],
    {j,jbeg,jend}]; it=1;
  Do [ii=Abs[p[[i+1]]]; m=ii-Abs[p[[i]]]-1; j1=Max[i-m,jbeg]; j2=Min[i,jend];
    kr=j2-j1+1; If [kr<=0, Continue[]]; If [p[[i+1]]>0,c=" ",c="*"];
    it++; t[[it,1]]=StringJoin[c,"Row",ToString[PaddedForm[i,3]]];
    jt=j1-jbeg+2; ij=j1+ii-i;
    Do[t[[it,jt++]]=PaddedForm[a[[ij++]]//FortranForm,{7,4}],{j,1,kr}],
    {i,jbeg,n}];
  Print[TableForm[Take[t,it],TableAlignments->{Right,Right},
    TableDirections->{Column,Row},TableSpacing->{0,2}]];
  jref=jend; If[jref<n,Goto[cycle]];
];

SymmSkyMatrixUpperTrianglePrint[S_]:= Module[
  {p,a,cycle,i,ij,it,j,j1,j2,jref,jbeg,jend,kcmax,kc,m,n,c,t},
  {p,a}=S; n=Dimensions[p][[1]]-1; kcmax=5; jref=0;
  Label[cycle]; Print[" "];
  jbeg=jref+1; jend=Min[jref+kcmax,n]; kc=jend-jref;
  t=Table[" ",{jend+1},{kc+1}];
  Do [If [p[[j+1]]>0,c=" ",c="*"];
    t[[1,j-jref+1]]=StringJoin[c,"Col",ToString[PaddedForm[j,3]]],
    {j,jbeg,jend}]; it=1;
  Do [it++; If [p[[i+1]]>0,c=" ",c="*"];
    t[[it,1]]=StringJoin[c,"Row",ToString[PaddedForm[i,3]]]; j=jref;
    Do [j++; If [j<i, Continue[]]; ij=Abs[p[[j+1]]]+i-j;
      If [ij<=Abs[p[[j]]], Continue[]];
      t[[it,k+1]]=PaddedForm[a[[ij]]//FortranForm,{7,4}],
      {k,1,kc}],
    {i,1,jend}];
  Print[TableForm[Take[t,it],TableAlignments->{Right,Right},
    TableDirections->{Column,Row},TableSpacing->{0,2}]];
  jref=jend; If[jref<n,Goto[cycle]];
];

Sr={{0, 1, 3, 6}, {1., 2., 7., 4., 23., 97.}};
SymmSkyMatrixLowerTrianglePrint[Sr];SymmSkyMatrixUpperTrianglePrint[Sr];

```

Cell 26.9 Skymatrix Mapping

```

SymmSkyMatrixLowerTriangleMap[S_]:=Module[
{p,a,cycle,i,ii,ij,it,itop,j,jj,j1,j2,jref,jbeg,jend,jt,kcmax,kc,kr,m,n,c,t},
{p,a}=S; n=Dimensions[p][[1]]-1; kcmax=40; jref=0;
Label[cycle]; Print[" "];
  jbeg=jref+1; jend=Min[jref+kcmax,n]; kc=jend-jref;
  itop=2; If[jend>9,itop=3]; If[jend>99,itop=4]; If[jend>999,itop=5];
  t=Table[" ",{n-jref+itop},{kc+1}]; it=0;
  If [itop>=5, it++; Do [m=Floor[j/1000];
    If [m>0,t[[it,j-jref+1]]=ToString[Mod[m,10]], {j,jbeg,jend}]];
  If [itop>=4, it++; Do [m=Floor[j/100];
    If [m>0,t[[it,j-jref+1]]=ToString[Mod[m,10]], {j,jbeg,jend}]];
  If [itop>=3, it++; Do [m=Floor[j/10];
    If [m>0,t[[it,j-jref+1]]=ToString[Mod[m,10]], {j,jbeg,jend}]];
  it++; Do[t[[it,j-jref+1]]=ToString[Mod[j,10]],{j,jbeg,jend}];
  it++; Do[If[p[[j+1]]<0,t[[it,j-jref+1]]="*"],{j,jbeg,jend}];
  Do [ii=Abs[p[[i+1]]]; m=ii-Abs[p[[i]]]-1; j1=Max[i-m,jbeg]; j2=Min[i,jend];
    kr=j2-j1+1; If [kr<=0, Continue[]]; If [p[[i+1]]>0,c=" ",c="*"];
    it++; t[[it,1]]=StringJoin[ToString[PaddedForm[i,2]],c];
    jt=j1-jbeg+2; ij=j1+ii-i;
    Do [ c=" 0"; If[a[[ij]]>0,c=" +"]; If[a[[ij++]]<0,c=" -"];
      t[[it,jt++]]=c, {j,1,kr},
    {i,jbeg,n}];
  Print[TableForm[Take[t,it],TableAlignments->{Right,Right},
    TableDirections->{Column,Row},TableSpacing->{0,0}]];
  jref=jend; If[jref<n,Goto[cycle]];
];

SymmSkyMatrixUpperTriangleMap[S_]:=Module[
{p,a,cycle,i,ij,it,itop,j,j1,j2,jref,jbeg,jend,kcmax,kc,m,n,c,t},
{p,a}=S; n=Dimensions[p][[1]]-1; kcmax=40; jref=0;
Label[cycle]; Print[" "];
  jbeg=jref+1; jend=Min[jref+kcmax,n]; kc=jend-jref;
  itop=2; If[jend>9,itop=3]; If[jend>99,itop=4]; If[jend>999,itop=5];
  t=Table[" ",{jend+itop},{kc+1}]; it=0;
  If [itop>=5, it++; Do [m=Floor[j/1000];
    If [m>0,t[[it,j-jref+1]]=ToString[Mod[m,10]], {j,jbeg,jend}]];
  If [itop>=4, it++; Do [m=Floor[j/100];
    If [m>0,t[[it,j-jref+1]]=ToString[Mod[m,10]], {j,jbeg,jend}]];
  If [itop>=3, it++; Do [m=Floor[j/10];
    If [m>0,t[[it,j-jref+1]]=ToString[Mod[m,10]], {j,jbeg,jend}]];
  it++; Do[t[[it,j-jref+1]]=ToString[Mod[j,10]],{j,jbeg,jend}];
  it++; Do[If[p[[j+1]]<0,t[[it,j-jref+1]]="*"],{j,jbeg,jend}];
  Do [it++; If [p[[i+1]]>0,c=" ",c="*"];
    t[[it,1]]=StringJoin[ToString[PaddedForm[i,2]],c]; j=jref;
    Do [j++; If [j<i, Continue[]]; ij=Abs[p[[j+1]]]+i-j;
      If [ij<=Abs[p[[j]]], Continue[]]; c=" 0";
      If[a[[ij]]>0,c=" +"]; If[a[[ij++]]<0,c=" -"]; t[[it,k+1]]=c,
    {k,1,kc},
    {i,1,jend}];
  Print[TableForm[Take[t,it],TableAlignments->{Right,Right},
    TableDirections->{Column,Row},TableSpacing->{0,0}]];
  jref=jend; If[jref<n,Goto[cycle]];
];

```


Cell 26.10 Skymatrix Reconstruction from Factors

```

SymmSkyMatrixLDUReconstruct[S_]:= Module[
  {p,ldu,a,v,n,i,ii,ij,j,jj,jk,jmj,k,m},
  {p,ldu}=S; a=ldu; n=Length[p]-1; v=Table[0,{n}];
  Do [jmj=Abs[p[[j]]]; jj=p[[j+1]]; If [jj<0, Continue[]];
    jk=jj-jmj; v[[jk]]=ldu[[jj]];
    Do [ij=jmj+k; i=j+ij-jj; ii=p[[i+1]]; If [ii<0, v[[k]]=0; Continue[]];
      If [i!=j, v[[k]]=ldu[[ij]]*ldu[[ii]]];
      m=Min[ii-Abs[p[[i]]],k]; a[[ij]]= v[[k]];
      a[[ij]]+=Take[ldu,{ii-m+1,ii-1}].Take[v,{k-m+1,k-1}],
      {k,1,jk}],
    {j,1,n}]; Return[{p,a}];
];

SymmSkyMatrixLDinvUReconstruct[S_]:= Module[
  {p,ldu,a,v,n,i,ii,ij,j,jj,jk,jmj,k,m},
  {p,ldu}=S; a=ldu; n=Length[p]-1; v=Table[0,{n}];
  Do [jmj=Abs[p[[j]]]; jj=p[[j+1]]; If [jj<0, Continue[]];
    jk=jj-jmj; v[[jk]]=1/ldu[[jj]];
    Do [ij=jmj+k; i=j+ij-jj; ii=p[[i+1]]; If [ii<0, v[[k]]=0; Continue[]];
      If [i!=j, v[[k]]=ldu[[ij]]/ldu[[ii]]];
      m=Min[ii-Abs[p[[i]]],k]; a[[ij]]= v[[k]];
      a[[ij]]+=Take[ldu,{ii-m+1,ii-1}].Take[v,{k-m+1,k-1}],
      {k,1,jk}],
    {j,1,n}]; Return[{p,a}];
];

p={0,1,2,5,8,9,15}; s={11,22,13,0,33,24,34,44,55,16,0,0,46,56,66};
S={p,s};
Sr=SymmSkyMatrixLDinvUReconstruct[S]; Print[Sr//InputForm];
Print[SymmSkyMatrixFactor[Sr,0]];

```

§26.3.7. *Miscellaneous Utilities

Finally, Cell 26.11 lists three miscellaneous modules. The most useful one is probably `SymmSkyMatrixConvertToFull`, which converts its skymatrix argument to a fully stored symmetric matrix. This is useful for things like a quick and dirty computation of eigenvalues:

```
Print[Eigenvalues[SymmSkyMatrixConvertToFull[S]]];
```

because *Mathematica* built-in eigensolvers require that the matrix be supplied in full storage form.

Cell 26.11 Miscellaneous Skymatrix Utilities

```

SymmSkyMatrixConvertToFull[S_]:= Module[
  {p,a,aa,n,j,jj,jmj,k},
  {p,a}=S; n=Length[p]-1; aa=Table[0,{n},{n}];
  Do [jmj=Abs[p[[j]]]; jj=Abs[p[[j+1]]]; aa[[j,j]]=a[[jj]];
    Do [aa[[j,j-k]]=aa[[j-k,j]]=a[[jj-k]],{k,1,jj-jmj-1}],
  {j,1,n}]; Return[aa];
];

SymmSkyMatrixConvertUnitUpperTriangleToFull[S_]:= Module[
  {p,ldu,aa,n,j,jj,jmj,k},
  {p,ldu}=S; n=Length[p]-1; aa=Table[0,{n},{n}];
  Do [jmj=Abs[p[[j]]]; jj=Abs[p[[j+1]]]; aa[[j,j]]=1;
    Do [aa[[j-k,j]]=ldu[[jj-k]],{k,1,jj-jmj-1}],
  {j,1,n}]; Return[aa];
];

SymmSkyMatrixConvertDiagonalToFull[S_]:= Module[
  {p,ldu,aa,n,i,j,jj,jmj,k},
  {p,ldu}=S; n=Length[p]-1; aa=Table[0,{n},{n}];
  Do [jj=Abs[p[[j+1]]]; aa[[j,j]]=ldu[[jj]],
  {j,1,n}]; Return[aa];
];

```

Homework Exercises for Chapter 26
Solving FEM Equations

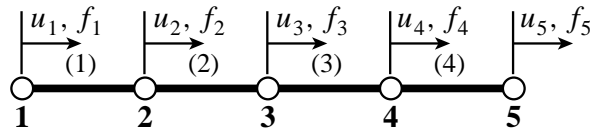


Figure 26.1. Structure for Exercise 26.1

EXERCISE 26.1

[A/C:10+10+15] Consider the 4-element assembly of bar elements shown in Figure 26.1. The only degree of freedom at each node is a translation along x . The element stiffness matrix of each element is

$$\mathbf{K}^{(e)} = \begin{bmatrix} 1 & -1 \\ -1 & 1 \end{bmatrix} \quad (\text{E26.1})$$

- (a) Assemble the 5×5 master stiffness matrix \mathbf{K} showing it as a full symmetric matrix. Hint: the diagonal entries are 1, 2, 2, 2, 1.
- (b) Show \mathbf{K} stored as a skyline matrix using a representation like illustrated in (26.15). Hint $\mathbf{p} = \{ 0, 1, 3, 5, 7, 9 \}$.
- (c) Perform the symmetric factorization $\mathbf{K} = \mathbf{LDL}^T$ of (26.7), where \mathbf{K} is stored as a full matrix to simplify hand work.⁴ Show that the entries of \mathbf{D} are 1, 1, 1, 1 and 0, which mean that \mathbf{K} is singular. Why?

⁴ You can do this with *Mathematica* using the function `LUDecomposition`, but hand work is as quick.

27

A Complete Plane Stress FEM Program

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§27.1. INTRODUCTION

This Chapter describes a complete finite element program for analysis of plane stress problems. Unlike the previous chapters the description is top down, i.e. starts from the main driver down to more specific modules.

§27.2. ANALYSIS STAGES

As in all FEM programs, the analysis of plane stress problems by the Direct Stiffness Method involves three major stages: (I) preprocessing or model definition, (II) processing, and (III) post-processing.

The preprocessing portion of the plane stress analysis is done by the driver program, which directly sets the data structures. These are similar to those outlined in Chapter 22 for frame/truss structures:

- I.1 Model definition by direct setting of the data structures.
- I.2 Plot of the FEM mesh, including nodes and element labels.

The processing stage involves three steps:

- II.1 Assembly of the master stiffness matrix, with a subordinate element stiffness module.
- II.2 Application of displacement BC. This is done by the equation modification method that keeps the same number and arrangement of degrees of freedom.
- II.3 Solution of the modified equations for displacements. The built in *Mathematica* function `LinearSolve` is used for this task.

Upon executing the processing steps, the nodal displacement solution is available. The following postprocessing steps follow:

- III.1 Recovery of forces including reactions, done through the built-in matrix multiplication.
- III.2 Computation of element stresses and interelement averaging to get nodal stresses.
- III.3 Contour plotting of displacement and stress fields.

§27.3. MODEL DEFINITION

The model-definition data may be broken down into three sets, which are listed below by order of appearance:

Model definition	$\left\{ \begin{array}{l} \text{Geometry data: node coordinates} \\ \text{Element data: type, connectivity, material and fabrication} \\ \text{Degree of freedom data: forces and support BCs} \end{array} \right.$
------------------	---

Note that the element data is broken down into four subsets: type, connectivity, material and fabrication, each of which has its own data structure. The degree of freedom data is broken into two subsets: tag and value. In addition there are miscellaneous process options, which are conveniently collected in a separate data set.

Accordingly, the model-definition input to the plane stress FEM program consists of eight data structures, which are called `NodeCoordinates`, `ElemTypes`, `ElemNodeLists`, `ElemMaterial`, `ElemFabrication`, `FreedomTags`, `FreedomValues` and `ProcessingOptions`

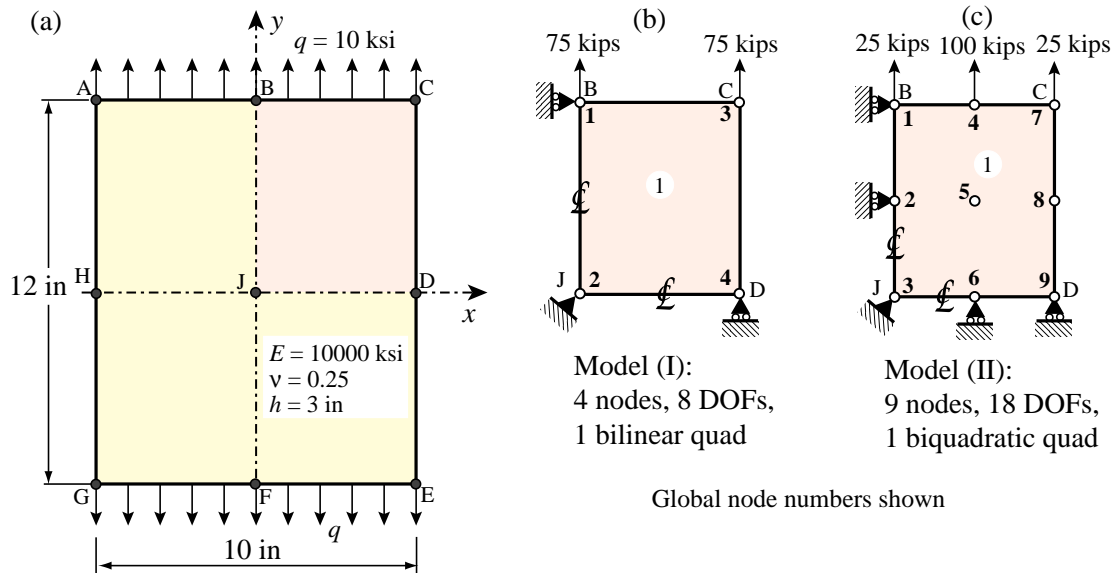


Figure 27.1. Rectangular plate under uniaxial loading, and two one-element FEM discretizations of quadrant BCDJ.

These data sets are described in the following subsections with reference to the problems and discretizations of Figures 27.1 to 27.3.

§27.3.1. Demo Problems

Figure 27.1 is a rectangular plate in plane stress under uniform uniaxial loading. Its analytical solution is $\sigma_{yy} = q$, $\sigma_{xx} = \sigma_{xy} = 0$, $u_y = qy/E$, $u_x = -\nu qx/E$. This problem should be solved exactly by *any* finite element mesh. In particular, the two one-element models shown on the right of that figure.

A similar but more complicated problem is shown in Figure 27.2: the axially-loaded rectangular plate dimensioned and loaded as that of Figure 27.1, but now with a central circular hole. This problem is defined in Figure 27.2. A FEM solution is to be obtained using the two quadrilateral element models (4-node and 9-node) depicted in Figure 27.3. The main result sought is the stress concentration factor on the hole boundary.

§27.3.2. Node Coordinates

The geometry data is specified through NodeCoordinates. This is a list of node coordinates configured as

$$\text{NodeCoordinates} = \{ \{x_1, y_1\}, \{x_2, y_2\}, \dots, \{x_N, y_N\} \} \quad (27.1)$$

where N is the number of nodes. Coordinate values should be floating point numbers; use the N function to insure that property if necessary. Nodes must be numbered consecutively and no gaps are permitted.

Example 1. For Model (I) of Figure 27.1:

$$\text{NodeCoordinates} = N[\{0,6\},\{0,0\},\{5,6\},\{5,0\}];$$

Example 2. For Model (II) of Figure 27.1:

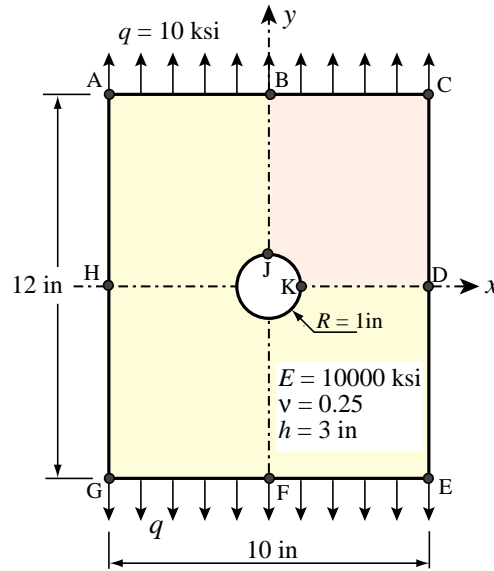


Figure 27.2. Rectangular plate with central circular hole.

```
NodeCoordinates = N[{0,6},{0,3},{0,0},{5/2,6},{5/2,3},
                    {5/2,0},{5,6},{5,3},{5,0}];
```

Example 3. For Model (I) of Figure 27.3, using a bit of coordinate generation along lines:

```
s={1,0.70,0.48,0.30,0.16,0.07,0.0};
xy1={0,6}; xy7={0,1}; xy8={2.5,6}; xy14={Cos[3*Pi/8],Sin[3*Pi/8]};
xy8={2.5,6}; xy21={Cos[Pi/4],Sin[Pi/4]}; xy15={5,6};
xy22={5,2}; xy28={Cos[Pi/8],Sin[Pi/8]}; xy29={5,0}; xy35={1,0};
NodeCoordinates=Table[{0,0},{35}];
Do[NodeCoordinates[[n]]=N[s[[n]]*xy1+(1-s[[n]])*xy7],{n,1,7}];
Do[NodeCoordinates[[n]]=N[s[[n-7]]*xy8+(1-s[[n-7]])*xy14],{n,8,14}];
Do[NodeCoordinates[[n]]=N[s[[n-14]]*xy15+(1-s[[n-14]])*xy21],{n,15,21}];
Do[NodeCoordinates[[n]]=N[s[[n-21]]*xy22+(1-s[[n-21]])*xy28],{n,22,28}];
Do[NodeCoordinates[[n]]=N[s[[n-28]]*xy29+(1-s[[n-28]])*xy35],{n,29,35}];
```

The result of this generation is that some of the interior nodes are not in the same positions as sketched in Figure 27.3, but this slight change hardly affects the results.

§27.3.3. Element Type

Element type is a label that specifies the type of element to be used.

$$\text{ElemTypes} = \{\{\text{etyp}^{(1)}\}, \{\text{etyp}^{(2)}\}, \dots \{\text{etyp}^{(N_e)}\}\} \quad (27.2)$$

Here $\text{etyp}(e)$ is the type descriptor of the e -th element specified as a character string. Allowable types are listed in Table 27.1.

For example, for Model (I) in Figure 27.3:

```
ElemTypes=Table[{"Quad4"},{numele}];
```

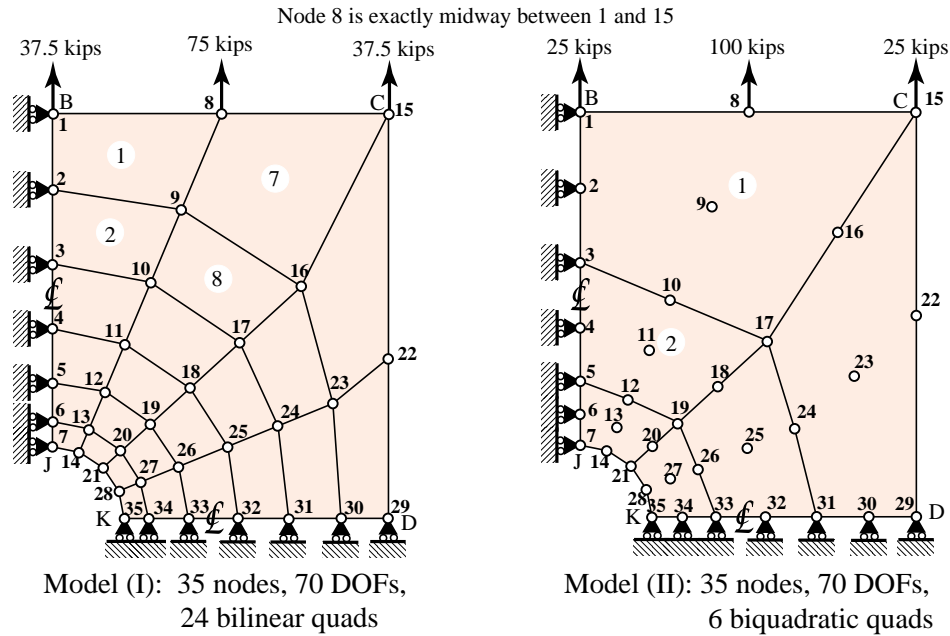



Figure 27.3. Two FEM discretizations of quadrant BCDKJ of the plate of Figure 27.2. Only a few element numbers are shown to reduce clutter.

and for Model (II):

```
ElemTypes=Table[{"Quad9"},{numele}];
```

Here `numele` is the number of elements. This can be extracted as `numele=Length[ElemNodeLists]`, where `ElemNodeLists` is defined below.

Table 27.1 Element Type Descriptors

Identifier	Plane Stress Model
"Trig3"	3-node linear triangle
"Trig6"	6-node quadratic triangle
"Trig10"	10-node cubic triangle
"Quad4"	4-node bilinear quad
"Quad9"	9-node biquadratic quad

§27.3.4. Element Connectivity

Element connectivity information specifies how the elements are connected.¹ This information is stored in `ElemNodeLists`, which is a list of element nodelists:

$$\text{ElemNodeLists} = \{ \{ \text{enL}^{(1)} \}, \{ \text{enL}^{(2)} \}, \dots \{ \text{enL}^{(N_e)} \} \} \quad (27.3)$$

¹ Some FEM programs call this the "topology" data.

where $eNL^{(e)}$ denotes the lists of nodes of the element e (given by nglobal node numbers) and N_e is the total number of elements.

Element boundaries must be traversed counterclockwise but you can start at any corner. Numbering elements with midnodes requires more care: first list corners counterclockwise, followed by midpoints (first midpoint is the one that follows first corner when going counterclockwise). When elements have an interior node, as in the 9-node biquadratic quadrilateral, that node goes last.

Example 1. For Model (I) of Figure 27.1, which has only one element:

```
ElemNodeLists= {{1,2,4,3}};
```

Example 2. For Model (II) of Figure 27.1, which has only one element:

```
ElemNodeLists= {{1,3,9,7,2,6,8,4,5}};
```

Example 3. For Model (I) of Figure 27.3, numbering the elements from top to bottom and from left to right:

```
ElemNodeLists=Table[{0,0,0,0},{24}];
ElemNodeLists[[1]]={1,2,9,8};
Do [ElemNodeLists[[e]]=ElemNodeLists[[e-1]]+{1,1,1,1},{e,2,6}];
ElemNodeLists[[7]]=ElemNodeLists[[6]]+{2,2,2,2};
Do [ElemNodeLists[[e]]=ElemNodeLists[[e-1]]+{1,1,1,1},{e,8,12}];
ElemNodeLists[[13]]=ElemNodeLists[[12]]+{2,2,2,2};
Do [ElemNodeLists[[e]]=ElemNodeLists[[e-1]]+{1,1,1,1},{e,14,18}];
ElemNodeLists[[19]]=ElemNodeLists[[18]]+{2,2,2,2};
Do [ElemNodeLists[[e]]=ElemNodeLists[[e-1]]+{1,1,1,1},{e,20,24}];
```

Example 4. For Model (II) of Figure 27.3, numbering the elements from top to bottom and from left to right:

```
ElemNodeLists=Table[{0,0,0,0,0,0,0,0,0,0},{6}];
ElemNodeLists[[1]]={1,3,17,15,2,10,16,8,9};
Do [ElemNodeLists[[e]]=ElemNodeLists[[e-1]]+{2,2,2,2,2,2,2,2,2},{e,2,3}];
ElemNodeLists[[4]]=ElemNodeLists[[3]]+{10,10,10,10,10,10,10,10,10};
Do [ElemNodeLists[[e]]=ElemNodeLists[[e-1]]+{2,2,2,2,2,2,2,2,2},{e,5,6}];
```

Since this particular mesh only has 6 elements, it would be indeed faster to write down the six nodelists.

§27.3.5. Material Properties

Data structure ElemMaterial lists the elastic modulus E and the Poisson's ratio ν for each element:

$$\text{ElemMaterial} = \{ \{ E^{(1)}, \nu^{(1)} \}, \{ E^{(2)}, \nu^{(2)} \}, \dots, \{ E^{(N_e)}, \nu^{(N_e)} \} \} \quad (27.4)$$

In the common case in which all elements have the same E and ν , this list can be easily generated by a Table instruction.

for all models shown in Figures 27.1 and 27.3,

```
ElemMaterial=Table[{Em,nu},{numele}],
```

in which $\text{numele} = \text{Length}[\text{ElemNodeLists}]$ is the number of elements (24 there) and the values of E_m and ν are typically declared separately.

§27.3.6. Fabrication Properties

`ElemMaterial` lists the plate thickness for each element:

$$\text{ElemFabrication} = \{ \{h(1)\}, \{h^{(2)}\}, \dots \{h^{(N_e)}\} \} \quad (27.5)$$

where N_e is the number of elements.

If all elements have the same thickness, this list can be easily generated by a `Table` instruction. For example, for the model of Figure 27.3, `ElemFabrication=Table[th,{ numele }]`, Here numele is the number of elements (24 in that case; this can be extracted as the `Length` of `ElemNodeList`) and the value of `th=3` is set at the start of the input cell.

§27.3.7. Freedom Tags

`FreedomTags` labels each node degree of freedom as to whether the load or the displacement is specified. The configuration of this list is similar to that of `NodeCoordinates`:

$$\text{FreedomTags} = \{ \{ \text{tag}_{x1}, \text{tag}_{y1} \}, \{ \text{tag}_{x2}, \text{tag}_{y2} \}, \dots \{ \text{tag}_{xN}, \text{tag}_{yN} \} \}$$

The tag value is 0 if the force is specified and 1 if the displacement is specified. When there are a lot of nodes, the quickest way to specify this list is to start from all zeros, and then insert the boundary conditions appropriately. For example, for the Model (I) of Figure 27.3:

```
numnod=Length[NodeCoordinates];
FreedomTags=Table[{0,0},{numnod}]; (* initialize and reserve space *)
Do[FreedomTags[[n]]={1,0},{n,1,7}]; (* vroller @ nodes 1 through 7 *)
Do[FreedomTags[[n]]={0,1},{n,29,35}]; (* hroller @ nodes 29 through 35 *)
```

This scheme works well because typically the number of supported nodes is small compared to the total number.

§27.3.8. Freedom Values

`FreedomValues` has the same node by node configuration as `FreedomTags`. It lists the specified values of the applied node force component if the corresponding tag is zero, and of the prescribed displacement component if the tag is one. Typically most of the list entries are zero. For example, in the model (I) of Figure 27.3 only 3 values (for the y forces on nodes 1, 8 and 15) will be nonzero:

```
numnod=Length[NodeCoordinates];
FreedomValues=Table[{0,0},{numnod}]; (* initialize and reserve space *)
FreedomValues[[1]]=FreedomValues[[15]]={0,37.5};
FreedomValues[[8]]={0,75}; (* y nodal loads *)
```

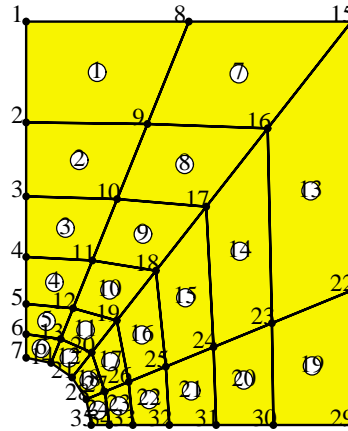


Figure 27.4. Mesh plot showing node and element numbers for Model (I) of Figure 27.3.

§27.3.9. Processing Options

Array `ProcessingOptions` should normally set as follows:

```
ProcessingOptions={True};
```

This specifies floating point numerical computations.

§27.3.10. Mesh Display

Nodes and elements of Model (I) of Figure 27.3 may be plotted by the following statement:

```
aspect=6/5;
Plot2DElementsAndNodes[NodeCoordinates,ElemNodeLists,aspect,
  "Plate with circular hole - 4-node quad model",True,True];
```

Here `aspect` is the plot frame aspect ratio (y dimension over x dimension), and the last two `True` argument values specify that node labels and element labels, respectively, be shown. The output is shown in Figure 27.4.

```

MembraneSolution[nodcoor_,eletyp_,elenod_,elemat_,
  elefab_,eleopt_,doftag_,dofval_] := Module[{K,Kmod,u,f,sig,j,n,ns,
  supdof,supval,numnod=Length[nodcoor],numele=Length[elenod]},
  u=f=sig={};
  K=MembraneMasterStiffness[nodcoor,
    eletyp_,elenod_,elemat_,elefab_,eleopt_]; K=N[K];
  ns=0; Do [Do [If[doftag[[n,j]]>0,ns++],{j,1,2}],{n,1,numnod}];
  supdof=supval=Table[0,{ns}];
  k=0; Do [Do [If[doftag[[n,j]]>0,k++;supdof[[k]]=2*(n-1)+j;
    supval[[k]]=dofval[[n,j]]],{j,1,2}],{n,1,numnod}];
  f=ModifiedNodeForces[supdof,supval,K,Flatten[dofval]];
  Kmod=ModifiedMasterStiffness[supdof,K];
  u=Simplify[Inverse[Kmod].f]; u=Chop[u];
  f=Simplify[K.u]; f=Chop[f];
  sig=MembraneNodalStresses[nodcoor,
    eletyp_,elenod_,elemat_,elefab_,eleopt_,u];
  sig=Chop[sig];
  Return[{u,f,sig}];
];

```

Figure 27.5. Module to drive the analysis of the plane stress problem.

§27.4. PROCESSING

The static solution is carried out by calling module `LinearSolutionOfPlaneStressModel` shown in Figure 27.5. The function call is

```

{u,f,sig}=MembraneSolution[NodeCoordinates,ElemTypes,
  ElemNodeLists,ElemMaterial,ElemFabrication,
  ProcessingOptions,FreedomTags,FreedomValues];

```

The function begins by assembling the free-free master stiffness matrix K by calling the module `MembraneMasterStiffness`. this module is listed in Figure 27.5. As a study of its code reveals, it can handle the five element types described in the previous section. The modules that compute the element stiffness matrices have been studied in previous Chapters and need not be listed here.

The displacement boundary conditions are applied by modules `ModifiedmasterStiffness` and `ModifiedNodeForces`, which return the modified stiffness matrix \hat{K} and the modified force vector \hat{f} in `Khat` and `fhat`, respectively. The logic of these modules has been explained in Chapter 22 and need not be described again here.

The unknown node displacements u are then obtained through the built in `LinearSolve` function, as `u=LinearSolve[Khat,fhat]`. This solution strategy is of course restricted to very small systems, but it has the advantages of simplicity.

The function returns arrays u , f and p , which are lists of length 12, 12 and 13, respectively. Array u contains the computed node displacements ordered $u_{x1} < u_{y1}, u_{x2}, \dots, u_{y8}$. Array f contains the node forces recovered from $f = Ku$; this includes the reactions f_{x1}, f_{y1} and f_{y8} .

Finally, array sig contains the nodal stresses $\sigma_{xx}, \sigma_{yy}, \sigma_{xy}$ at each node, recovered from the displacement solution. This computation is driven by module `MembraneNodeStresses`, which is

```

MembraneMasterStiffness[nodcoor_,eletyp_,elenod_,
  elemat_,elefab_,eleopt_]:=
Module[{numele=Length[elenod],numnod=Length[nodcoor],numer,
  ne,eNL,eftab,neldof,i,n,Em,v,Emat,th,ncoor,Ke,K},
K=Table[0,{2*numnod},{2*numnod}]; numer=eleopt[[1]];
For [ne=1,ne<=numele,ne++,
  {type}=eletyp[[ne]];
  If [type!="Trig3"&&type!="Trig6"&&type!="Trig10"&&
    type!="Quad4"&&type!="Quad9",
    Print["Illegal element type,ne=",ne]; Return[Null]];
  eNL=elenod[[ne]]; n=Length[eNL];
  eftab=Flatten[Table[{2*eNL[[i]]-1,2*eNL[[i]]},{i,n}]];
  ncoor=Table[nodcoor[[eNL[[i]]]},{i,n}];
  {Em,v}=elemat[[ne]];
  Emat=Em/(1-v^2)*{{1,v,0},{v,1,0},{0,0,(1-v)/2}};
  {th}=elefab[[ne]];
  If [type=="Trig3", Ke=Trig3IsoPMembraneStiffness[ncoor,
    {Emat,0,0},{th},{numer}]]];
  If [type=="Quad4", Ke=Quad4IsoPMembraneStiffness[ncoor,
    {Emat,0,0},{th},{numer,2}]]];
  If [type=="Trig6", Ke=Trig6IsoPMembraneStiffness[ncoor,
    {Emat,0,0},{th},{numer,3}]]];
  If [type=="Quad9", Ke=Quad9IsoPMembraneStiffness[ncoor,
    {Emat,0,0},{th},{numer,3}]]];
  If [type=="Trig10",Ke=Trig10IsoPMembraneStiffness[ncoor,
    {Emat,0,0},{th},{numer,3}]]];
  neldof=Length[Ke];
  For [i=1,i<=neldof,i++,ii=eftab[[i]];
    For [j=i,j<=neldof,j++,jj=eftab[[j]];
      K[[jj,ii]]=K[[ii,jj]]+=Ke[[i,j]]
    ];
  ];
];
Return[K];
];

```

Figure 27.6. Module to assemble the master stiffness matrix.

listed in Figure 27.7. This computation is actually part of the postprocessing stage. It is not described here since stress recovery is treated in more detail in a subsequent Chapter.

§27.5. POSTPROCESSING

Postprocessing are activities undertaken on return from `membraneSoution`. They include optional printout of u , f and p , plus some simple graphic displays described below.

§27.5.1. Displacement Field Contour Plots

Contour plots of the displacement components u_x and u_y interpolated over elements from the computed node displacements can be obtained by the following script:

```

aspect=6/5; Nsub=4; ux=uy=Table[0,{numnod}];
Do[ux[[n]]=u[[2*n-1]]; uy[[n]]=u[[2*n]], {n,1,numnod}];
uxmax=uymax=0;

```

```

MembraneNodalStresses[nodcoor_,eletyp_,elenod_,
  elemat_,elefab_,eleopt_,u_]:= Module[{numele=Length[elenod],
  numnod=Length[nodcoor],numer,type,ne,eNL,eftab,i,ii,j,k,n,
  Em,v,Emat,th,ncoor,ue,ncount,esig,sige,sig},
  ncount=Table[0,{numnod}]; {numer}=eleopt;
  sige= Table[0,{numele}]; sig=Table[{0,0,0},{numnod}];
  For [ne=1,ne<=numele,ne++,
    {type}=eletyp[[ne]];
    eNL=elenod[[ne]]; n=Length[eNL]; ue=Table[0,{2*n}];
    eftab=Flatten[Table[{2*eNL[[i]]-1,2*eNL[[i]]},{i,1,n}]];
    ncoor=Table[nodcoor[[eNL[[i]]]],{i,n}];
    Do [ii=eftab[[i]];ue[[i]]=u[[ii]],[i,1,2*n]];
    {Em,v}=elemat[[ne]];
    Emat=Em/(1-v^2)*{{1,v,0},{v,1,0},{0,0,(1-v)/2}};
    th=elefab[[ne]];
    If [type=="Trig3", esig=Trig3IsoPMembraneStresses[ncoor,
      {Emat,0,0},{th},{numer},ue] ];
    If [type=="Quad4", esig=Quad4IsoPMembraneStresses[ncoor,
      {Emat,0,0},{th},{numer,2},ue] ];
    If [type=="Trig6", esig=Trig6IsoPMembraneStresses[ncoor,
      {Emat,0,0},{th},{numer,3},ue] ];
    If [type=="Quad9", esig=Quad9IsoPMembraneStresses[ncoor,
      {Emat,0,0},{th},{numer,3},ue] ];
    If [type=="Trig10",esig=Trig10IsoPMembraneStresses[ncoor,
      {Emat,0,0},{th},{numer,3},ue] ];
    esig=Chop[esig]; sige[[ne]]=esig;
    For [i=1,i<=n,i++,k=eNL[[i]]; ncount[[k]]++;
      Do[ sig[[k,j]]+=esig[[i,j]],[j,3] ]];
  ];
  For [n=1,n<=numnod,n++,k=ncount[[n]];If [k>1,sig[[n]]/=k]
  ];
  Return[sig];
];

```

Figure 27.7. Module to compute averaged nodal stresses.

```

Do [uxmax=Max[Abs[ux[[n]]],uxmax]; uymax=Max[Abs[uy[[n]]],uymax],
  {n,1,numnod}];
ContourPlotNodeFuncOver2DMesh[NodeCoordinates,ElemNodeLists,ux,
  uxmax,Nsub,aspect,"Displacement component ux"];
ContourPlotNodeFuncOver2DMesh[NodeCoordinates,ElemNodeLists,uy,
  uymax,Nsub,aspect,"Displacement component uy"];

```

§27.5.2. Stress Field Contour Plots

Contour plots of the stress components σ_{xx} , σ_{yy} and σ_{xy} returned by MembraneSolution can be produced by the following script:

```

aspect=6/5; Nsub=4; sxx=syy=sxy=Table[0,{numnod}];
Do[{sxx[[n]],syy[[n]],sxy[[n]]}=sig[[n]},{n,1,numnod}];
sxxmax=syymax=sxymax=0;
Do[sxxmax=Max[Abs[sxx[[n]]],sxxmax];
  syymax=Max[Abs[syy[[n]]],syymax];

```

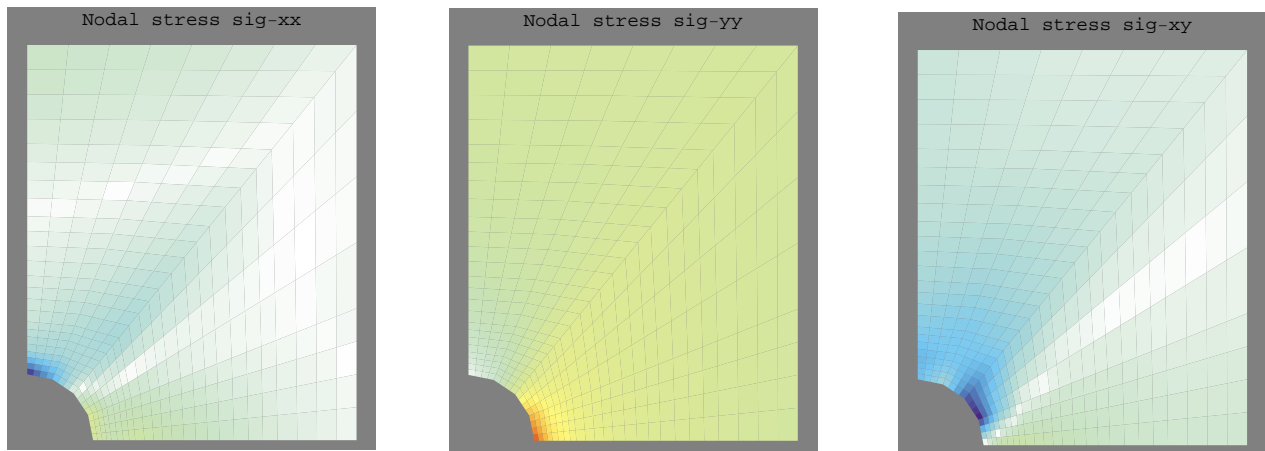


Figure 27.8. Stress contour plots for Model (I) of Figure 27.3.

```
sxymax=Max[Abs[sxy[[n]]],sxymax], {n,1,numnod}];
ContourPlotNodeFuncOver2DMesh[NodeCoordinates,ElemNodeLists,
  sxx,sxxmax,Nsub,aspect,"Nodal stress sig-xx"];
ContourPlotNodeFuncOver2DMesh[NodeCoordinates,ElemNodeLists,
  syy,symax,Nsub,aspect,"Nodal stress sig-yy"];
ContourPlotNodeFuncOver2DMesh[NodeCoordinates,ElemNodeLists,
  sxy,sxymax,Nsub,aspect,"Nodal stress sig-xy"]
```

The stress plots are shown in Figure 27.8.

§27.5.3. Animation

Sometimes it is useful to animate results such as stress fields when a load changes as a function of a time-like parameter t . this can be done by performing a sequence of analyses in a loop.

Such sequence produces a series of plot frames which may be animated by doubly clicking on one of them. The speed of the animation may be controlled by pressing on the rightmost buttons at the bottom of the *Mathematica* window. The second button from the left, if pressed, changes the display sequence to back and forth motion.

§27.6. A COMPLETE DRIVER CELL

A complete driver cell, (which is Cell 11 in the plane stress demo program placed on the web) is listed in Figures 27.9 through 27.11. This driver cell does the one-element test of Figure 27.1(b), which involves only one four node quadrilateral element.

The driver cell is broken down into three segments for reading convenience. Figure 27.9 lists the model definition script followed by a mesh plot command. Figure 27.10 lists the processing script and printout of displacements, forces and stresses. Figure 27.12 lists the postprocessing script to produce contour plots of displacements and stresses.

Of course one element should give the exact solution of the problem of Figure 27.1(a) with one element. That is precisely the test.


```

ClearAll[Em,v,th];
Em=10000; v=.25; th=3; aspect=6/5; Nsub=4;

(* Define FEM model *)

ElemTypes=Table[{"Quad4"},{numele}];
NodeCoordinates=N[{{0,6},{0,0},{5,6},{5,0}}];
ElemNodeLists= {{1,2,4,3}};
numnod=Length[NodeCoordinates]; numele=Length[ElemNodeLists];
ElemMaterial= Table[{Em,v},{numele}];
ElemFabrication=Table[{th},{numele}];
FreedomValues=FreedomTags=Table[{0,0},{numnod}];
FreedomValues[[1]]=FreedomValues[[3]]={0,75}; (* nodal loads *)
FreedomTags[[1]]={1,0}; (* vroller @ node 1 *)
FreedomTags[[2]]={1,1}; (* fixed node 2 *)
FreedomTags[[4]]={0,1}; (* hroller @ node 4 *)
ElemOptions={True};

Plot2DElementsAndNodes[NodeCoordinates,ElemNodeLists,aspect,
  "One element mesh - 4-node quad",True,True];

```

Figure 27.9. Model definition script for problem of Figure 27.1(b).

```

(* Solve problem and print results *)

{u,f,sig}=MembraneSolution[
  NodeCoordinates,ElemTypes,ElemNodeLists,
  ElemMaterial,ElemFabrication,
  ElemOptions,FreedomTags,FreedomValues];

Print["Computed displacements=",u];
Print["Recovered forces=",f];
Print["Avg nodal stresses=",sig];

```

Figure 27.10. Processing script for problem of Figure 27.1(b).

Other driver cell examples for more complicated problems may be studied in the `PlaneStress.nb` Notebook posted on the course web site. It can be observed that the processing and postprocessing scripts are largely the same. What changes is the model definition script portion, which often benefits from node and element generation constructs.

```

(* Plot Displacement Components Distribution *)

ux=uy=Table[0,{numnod}];
Do[ux[[n]]=u[[2*n-1]]; uy[[n]]=u[[2*n]],{n,1,numnod}];
uxmax=uymax=0;
Do[uxmax=Max[Abs[ux[[n]]],uxmax]; uymax=Max[Abs[uy[[n]]],uymax],
  {n,1,numnod}];
ContourPlotNodeFuncOver2DMesh[NodeCoordinates,ElemNodeLists,ux,
  uxmax,Nsub,aspect,"Displacement component ux"];
ContourPlotNodeFuncOver2DMesh[NodeCoordinates,ElemNodeLists,uy,
  uymax,Nsub,aspect,"Displacement component uy"];

(* Plot Averaged Nodal Stresses Distribution *)

sxx=syy=sxy=Table[0,{numnod}];
Do[{sxx[[n]],syy[[n]],sxy[[n]]}=sig[[n]],{n,1,numnod}];
sxxmax=syymax=sxymax=0;
Do[sxxmax=Max[Abs[sxx[[n]]],sxxmax];
  syymax=Max[Abs[syy[[n]]],syymax];
  sxymax=Max[Abs[sxy[[n]]],sxymax],{n,1,numnod}];
ContourPlotNodeFuncOver2DMesh[NodeCoordinates,ElemNodeLists,
  sxx,sxxmax,Nsub,aspect,"Nodal stress sig-xx"];
ContourPlotNodeFuncOver2DMesh[NodeCoordinates,ElemNodeLists,
  syy,syymax,Nsub,aspect,"Nodal stress sig-yy"];
ContourPlotNodeFuncOver2DMesh[NodeCoordinates,ElemNodeLists,
  sxy,sxymax,Nsub,aspect,"Nodal stress sig-xy"];

```

Figure 27.11. Postprocessing script for problem of Figure 27.1(b).

Notes and Bibliography

Few FEM books show a complete program. More common is the display of snippets of code. These are left dangling chapter after chapter, with no attempt at coherent interfacing. This historical problem comes from reliance on low-level languages such as Fortran. Even the simplest program would runs to thousands of code lines. At 50 lines/page that becomes difficult to fit snugly in a textbook.

Another historical problem is plotting. Languages such as C or Fortran do not include plotting libraries for the simple reason that low-level plotting standards never existed. The situation changes when using high-level languages like *Matlab* or *Mathematica*. The language engine provides the necessary fit to available computer hardware, and plotting scripts are transportable.

28

Stress Recovery

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§28.1. INTRODUCTION

In this lecture we study the recovery of stress values for two-dimensional plane-stress elements.¹

This analysis step is sometimes called *postprocessing* because it happens after the main processing step — the calculation of nodal displacements — is completed. Stress calculations are of interest because in structural analysis and design the stresses are often more important to the engineer than displacements.

In the stiffness method of solution discussed in this course, the stresses are obtained from the computed displacements, and are thus *derived quantities*. The accuracy of derived quantities is generally lower than that of primary quantities (the displacements), an informal statement that may be mathematically justified in the theory of finite element methods. For example, if the accuracy level of displacements is 1% that of the stresses may be typically 10% to 20%, and even lower at boundaries.

It is therefore of interest to develop techniques that enhance the accuracy of the computed stresses. The goal is to “squeeze” as much accuracy from the computed displacements while keeping the computational effort reasonable. These procedures receive the generic name *stress recovery techniques* in the finite element literature. In the following sections we cover the simplest stress recovery techniques that have been found most useful in practice.

§28.2. CALCULATION OF ELEMENT STRAINS AND STRESSES

In elastic materials, stresses σ are directly related to strains \mathbf{e} at each point through the elastic constitutive equations $\sigma = \mathbf{E}\mathbf{e}$. It follows that the stress computation procedure begins with strain computations, and that the accuracy of stresses depends on that of strains. Strains, however, are seldom saved or printed.

In the following we focus our attention on two-dimensional isoparametric elements, as the computation of strains, stresses and axial forces in bar elements is straightforward.

Suppose that we have solved the master stiffness equations

$$\mathbf{K}\mathbf{u} = \mathbf{f}, \quad (28.1)$$

for the node displacements \mathbf{u} . To calculate strains and stresses we perform a loop over all defined elements. Let e be the element index of a specific two-dimensional isoparametric element encountered during this loop, and $\mathbf{u}^{(e)}$ the vector of computed element node displacements. Recall from §15.3 that the strains at any point in the element may be related to these displacements as

$$\mathbf{e} = \mathbf{B}\mathbf{u}^{(e)}, \quad (28.2)$$

where \mathbf{B} is the strain-displacement matrix (14.18) assembled with the x and y derivatives of the element shape functions evaluated at the point where we are calculating strains. The corresponding stresses are given by

$$\sigma = \mathbf{E}\mathbf{e} = \mathbf{E}\mathbf{B}\mathbf{u} \quad (28.3)$$

¹ This Chapter needs rewriting to show the use of Mathematica for stress computation. To be done in the future.

Table 28.1 Natural Coordinates of Bilinear Quadrilateral Nodes

Corner node	ξ	η	ξ'	η'	Gauss node	ξ	η	ξ'	η'
1	-1	-1	$-\sqrt{3}$	$-\sqrt{3}$	1'	$-1/\sqrt{3}$	$-1/\sqrt{3}$	-1	-1
2	+1	-1	$+\sqrt{3}$	$-\sqrt{3}$	2'	$+1/\sqrt{3}$	$-1/\sqrt{3}$	+1	-1
3	+1	+1	$+\sqrt{3}$	$+\sqrt{3}$	3'	$+1/\sqrt{3}$	$+1/\sqrt{3}$	+1	+1
4	-1	+1	$-\sqrt{3}$	$+\sqrt{3}$	4'	$-1/\sqrt{3}$	$+1/\sqrt{3}$	-1	+1
Gauss nodes, and coordinates ξ' and η' are defined in §28.4 and Fig. 28.1									

In the applications it is of interest to evaluate and report these stresses at the *element nodal points* located on the corners and possibly midpoints of the element. These are called *element nodal point stresses*.

It is important to realize that the stresses computed at the same nodal point from adjacent elements *will not generally be the same*, since stresses are not required to be continuous in displacement-assumed finite elements. This suggests some form of stress averaging can be used to improve the stress accuracy, and indeed this is part of the stress recovery technique further discussed in §28.5. The results from this averaging procedure are called *nodal point stresses*.

For the moment let us see how we can proceed to compute element nodal stresses. Two approaches are followed in practice:

1. Evaluate directly σ at the element node locations by substituting the natural coordinates of the nodal points as arguments to the shape function modules. These modules return \mathbf{q}_x and \mathbf{q}_y and direct application of (28.2)-(28.4) yields the strains and stresses at the nodes.
2. Evaluate σ at the Gauss integration points used in the element stiffness integration rule and then extrapolate to the element node points.

Empirical evidence indicates that the second approach generally delivers better stress values for *quadrilateral* elements whose geometry departs substantially from the rectangular shape. This is backed up by “superconvergence” results in finite element approximation theory. For rectangular elements there is no difference.

For isoparametric *triangles* both techniques deliver similar results (identical if the elements are straight sided with midside nodes at midpoints) and so the advantages of the second one are marginal. Both approaches are covered in the sequel.

§28.3. DIRECT STRESS EVALUATION AT NODES

This approach is straightforward and need not be discussed in detail.

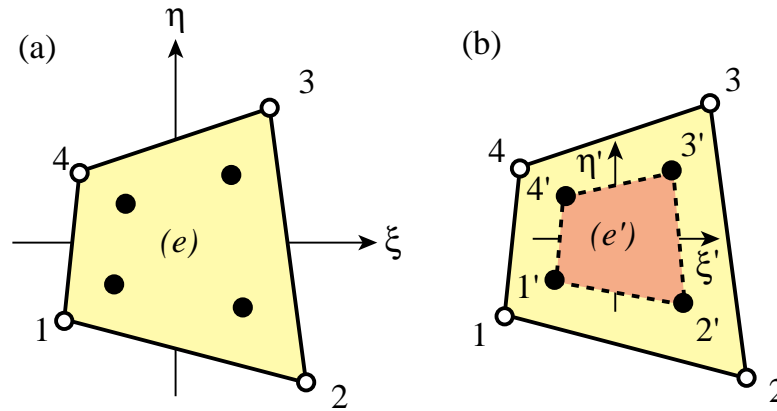


Figure 28.1. Extrapolation from 4-node quad Gauss points: (a) 2×2 rule, (b) Gauss element (e')

§28.4. EXTRAPOLATION FROM GAUSS POINTS

This will again be explained for the four-node bilinear quadrilateral. The normal Gauss integration rule for element stiffness evaluation is 2×2 , as illustrated in Figure 28.1.

The stresses are calculated at the Gauss points, which are identified as $1'$, $2'$, $3'$ and $4'$ in Figure 28.1. Point i' is closest to node i so it is seen that Gauss point numbering essentially follows element node numbering in the counterclockwise sense. The natural coordinates of these points are listed in Table 28.1. The stresses are evaluated at these Gauss points by passing these natural coordinates to the shape function subroutine. Then each stress component is “carried” to the corner nodes 1 through 4 through a bilinear extrapolation based on the computed values at $1'$ through $4'$.

To understand the extrapolation procedure more clearly it is convenient to consider the region bounded by the Gauss points as an “internal element” or “Gauss element”. This interpretation is depicted in Figure 28.1(b). The Gauss element, denoted by (e'), is also a four-node quadrilateral. Its quadrilateral (natural) coordinates are denoted by ξ' and η' . These are linked to ξ and η by the simple relations

$$\xi = \xi' / \sqrt{3}, \quad \eta = \eta' / \sqrt{3}, \quad \xi' = \xi \sqrt{3}, \quad \eta' = \eta \sqrt{3}. \quad (28.4)$$

Any scalar quantity w whose values w'_i at the Gauss element corners are known can be interpolated through the usual bilinear shape functions now expressed in terms of ξ' and η' :

$$w(\xi', \eta') = [w'_1 \quad w'_2 \quad w'_3 \quad w'_4] \begin{bmatrix} N_1^{(e')} \\ N_2^{(e')} \\ N_3^{(e')} \\ N_4^{(e')} \end{bmatrix}, \quad (28.5)$$

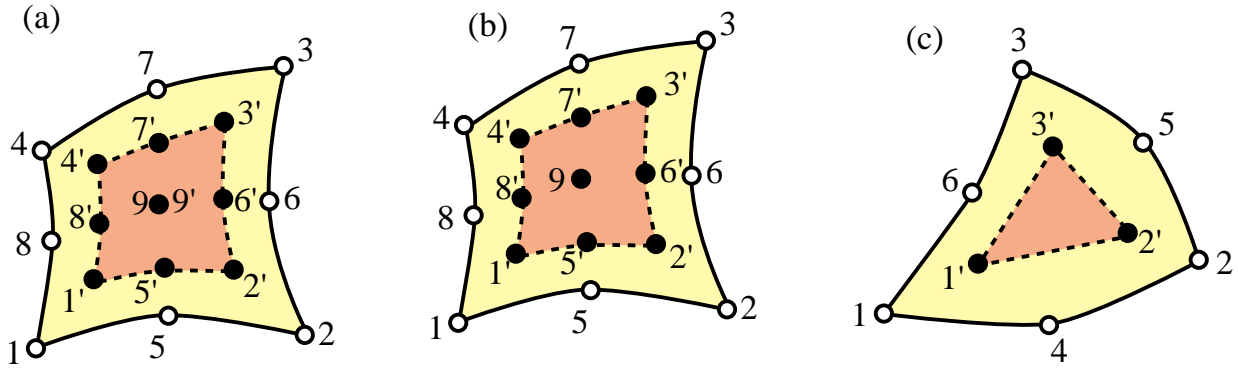


Figure 28.2. Gauss elements for higher order quadrilaterals and triangles:
 (a) 9-node element with 3×3 Gauss rule, (b) 8-node element with
 3×3 Gauss rule, (c) 6-node element with 3-interior point rule.

where (cf. §15.6.2)

$$\begin{aligned}
 N_1^{(e')} &= \frac{1}{4}(1 - \xi')(1 - \eta'), \\
 N_2^{(e')} &= \frac{1}{4}(1 + \xi')(1 - \eta'), \\
 N_3^{(e')} &= \frac{1}{4}(1 + \xi')(1 + \eta'), \\
 N_4^{(e')} &= \frac{1}{4}(1 - \xi')(1 + \eta').
 \end{aligned} \tag{28.6}$$

To extrapolate w to corner 1, say, we replace its ξ' and η' coordinates, namely $\xi' = \eta' = -\sqrt{3}$, into the above formula. Doing that for the four corners we obtain

$$\begin{bmatrix} w_1 \\ w_2 \\ w_3 \\ w_4 \end{bmatrix} = \begin{bmatrix} 1 + \frac{1}{2}\sqrt{3} & -\frac{1}{2} & 1 - \frac{1}{2}\sqrt{3} & -\frac{1}{2} \\ -\frac{1}{2} & 1 + \frac{1}{2}\sqrt{3} & -\frac{1}{2} & 1 - \frac{1}{2}\sqrt{3} \\ 1 - \frac{1}{2}\sqrt{3} & -\frac{1}{2} & 1 + \frac{1}{2}\sqrt{3} & -\frac{1}{2} \\ -\frac{1}{2} & 1 - \frac{1}{2}\sqrt{3} & -\frac{1}{2} & 1 + \frac{1}{2}\sqrt{3} \end{bmatrix} \begin{bmatrix} w'_1 \\ w'_2 \\ w'_3 \\ w'_4 \end{bmatrix} \tag{28.7}$$

Note that the sum of the coefficients in each row is one, as it should be. For stresses we apply this formula taking w to be each of the three stress components, σ_{xx} , σ_{yy} and τ_{xy} , in turn.

Extrapolation in Higher Order Elements

For eight-node and nine-node isoparametric quadrilaterals the usual Gauss integration rule is 3×3 , and the Gauss elements are nine-noded quadrilaterals that look as in Figure 28.2(a) and (b) above. For six-node triangles the usual quadrature is the 3-point rule with internal sampling points, and the Gauss element is a three-node triangle as shown in Figure 28.2(c).

§28.5. INTERELEMENT AVERAGING

The stresses computed in element-by-element fashion as discussed above, whether by direct evaluation at the nodes or by extrapolation, will generally exhibit jumps between elements. For printing and plotting purposes it is usually convenient to “smooth out” those jumps by computing *averaged nodal stresses*. This averaging may be done in two ways:

- (I) Unweighted averaging: assign same weight to all elements that meet at a node;
- (II) Weighted averaging: the weight assigned to element contributions depends on the stress component and the element geometry and possibly the element type.

Several weighted average schemes have been proposed in the finite element literature, but they do require additional programming.

A

Matrix Algebra: Vectors

§A.1 MOTIVATION

Matrix notation was invented¹ primarily to express linear algebra relations in *compact form*. Compactness enhances visualization and understanding of essentials. To illustrate this point, consider the following set of m linear relations between one set of n quantities, x_1, x_2, \dots, x_n , and another set of m quantities, y_1, y_2, \dots, y_m :

$$\begin{array}{cccccccccccl}
 a_{11}x_1 & + & a_{12}x_2 & + & \cdots & + & a_{1j}x_j & + & \cdots & + & a_{1n}x_n & = & y_1 \\
 a_{21}x_1 & + & a_{22}x_2 & + & \cdots & + & a_{2j}x_j & + & \cdots & + & a_{2n}x_n & = & y_2 \\
 \cdots & & \cdots & & \cdots & & \cdots & & \cdots & & \cdots & & \cdots \\
 a_{i1}x_1 & + & a_{i2}x_2 & + & \cdots & + & a_{ij}x_j & + & \cdots & + & a_{in}x_n & = & y_i \\
 \cdots & & \cdots & & \cdots & & \cdots & & \cdots & & \cdots & & \cdots \\
 a_{m1}x_1 & + & a_{m2}x_2 & + & \cdots & + & a_{mj}x_j & + & \cdots & + & a_{mn}x_n & = & y_m
 \end{array} \tag{A.1}$$

The subscripted a , x and y quantities that appear in this set of relations may be formally arranged as follows:

$$\begin{bmatrix} a_{11} & a_{12} & \cdots & a_{1j} & \cdots & a_{1n} \\ a_{21} & a_{22} & \cdots & a_{2j} & \cdots & a_{2n} \\ \vdots & \vdots & \ddots & \vdots & \ddots & \vdots \\ a_{i1} & a_{i2} & \cdots & a_{ij} & \cdots & a_{in} \\ \vdots & \vdots & \ddots & \vdots & \ddots & \vdots \\ a_{m1} & a_{m2} & \cdots & a_{mj} & \cdots & a_{mn} \end{bmatrix} \begin{bmatrix} x_1 \\ x_2 \\ \vdots \\ x_j \\ \vdots \\ x_n \end{bmatrix} = \begin{bmatrix} y_1 \\ y_2 \\ \vdots \\ y_i \\ \vdots \\ y_m \end{bmatrix} \tag{A.2}$$

Two kinds of mathematical objects can be distinguished in (A.2). The two-dimensional array expression enclosed in brackets is a *matrix*, which we call **A**. Matrices are defined and studied in Appendix B.

The one-dimensional array expressions in brackets are *column vectors* or simply *vectors*, which we call **x** and **y**, respectively. The use of **boldface** uppercase and lowercase letters to denote matrices and vectors, respectively, follows conventional matrix-notation rules in engineering applications.

Replacing the expressions in (A.2) by these symbols we obtain the *matrix form* of (A.1):

$$\mathbf{Ax} = \mathbf{y} \tag{A.3}$$

Putting **A** next to **x** means “matrix product of **A** times **x**”, which is a generalization of the ordinary scalar multiplication, and follows the rules explained later.

Clearly (A.3) is a more compact, “short hand” form of (A.1).

Another key practical advantage of the matrix notation is that it translates directly to the computer implementation of linear algebra processes in programming languages that offer array data structures.

¹ By Arthur Cayley at Cambridge (UK) in 1858.

§A.2 VECTORS

We begin by defining a *vector*, a set of n numbers which we shall write in the form

$$\mathbf{x} = \begin{bmatrix} x_1 \\ x_2 \\ \vdots \\ x_n \end{bmatrix} \quad (\text{A.4})$$

A vector of this type is called a *column vector*. We shall see later that although a vector may be viewed as a special case of a matrix it deserves treatment on its own. The symbol \mathbf{x} is the name of the vector.

If n numbers are arranged in a horizontal array, as in

$$\mathbf{z} = [z_1 \quad z_2 \quad \dots \quad z_n] \quad (\text{A.5})$$

then \mathbf{z} is called a *row vector*. If we use the term “vector” without a qualifier, it is understood to be a column vector such as (A.4).

§A.2.1 Notational Conventions

Typeset vectors will be designated by **bold** lowercase letters. For example:

$$\mathbf{a}, \quad \mathbf{b}, \quad \mathbf{c}, \quad \mathbf{x}, \quad \mathbf{y}, \quad \mathbf{z}$$

On the other hand, *handwritten or typewritten* vectors, which are those written on paper or on the blackboard, are identified by putting a wiggle or bar underneath the letter. For example:

$$\underline{a}$$

The subscripted quantities such as x_1 in (A.4) are called the *entries* or *components*² of \mathbf{x} , while n is called the *order* of the vector \mathbf{x} . Vectors of order one ($n = 1$) are called *scalars*. These are the usual quantities of analysis.

For compactness one sometimes abbreviates the phrase “vector of order n ” to just “ n -vector.” For example, \mathbf{z} in the example below is a 4-vector.

If the components are real numbers, the vector is called a *real vector*. If the components are complex numbers we have a *complex vector*. Linear algebra embraces complex vectors as easily as it does real ones; however, we rarely need complex vectors for this exposition. Consequently real vectors will be assumed unless otherwise noted.

² The term *component* is primarily used in mathematical treatments whereas *entry* is used more in conjunction with the computer implementation. The term *element* is also used in the literature but this will be avoided here as it may lead to confusion with finite elements.

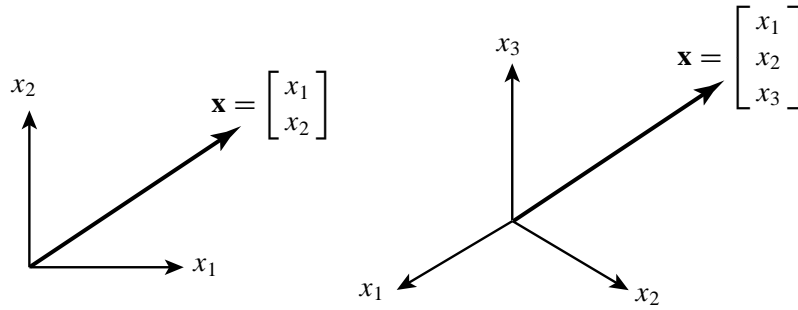


Figure A.1. Standard visualization of 2-vectors and 3-vectors as position vectors in 2-space and 3-space, respectively.

EXAMPLE A.1

$$\mathbf{x} = \begin{bmatrix} 4 \\ -2 \\ 3 \\ 0 \end{bmatrix}$$

is real column vector of order 4, or, briefly, a 4-vector.

EXAMPLE A.2

$$\mathbf{q} = [1 \quad 1 \quad 1 \quad 1 \quad 1 \quad 1]$$

is a real row vector of order 6.

Occasionally we shall use the short-hand “component notation”

$$\mathbf{x} = [x_i] \tag{A.6}$$

for a generic vector. This comes handy when it is desirable to show the scheme of notation for the components.

§A.2.2 Visualization

To aid visualization a two-dimensional vector \mathbf{x} ($n = 2$) can be depicted as a line segment, or arrow, directed from the chosen origin to a point on the Cartesian plane of the paper with coordinates (x_1, x_2) . See Figure A.1. In mechanics this is called a *position vector* in 2-space.

One may resort to a similar geometrical interpretation in three-dimensional Cartesian space ($n = 3$); see Figure A.1. Drawing becomes a bit messier, however, although some knowledge of perspective and projective geometry helps.

The interpretation extends to all Euclidean spaces of dimensions $n > 3$ but direct visualization is of course impaired.

§A.2.3 Special Vectors

The *null* vector, written $\mathbf{0}$, is the vector all of whose components are zero.

The *unit* vector, denoted by \mathbf{e}_i , is the vector all of whose components are zero, except the i^{th} component, which is one. After introducing matrices (Appendix B) a unit vector may be defined as the i^{th} column of the identity matrix.

The *unitary* vector, called \mathbf{e} , is the vector all of whose components are unity.

§A.3 VECTOR OPERATIONS

Operations on vectors in two-dimensional and three-dimensional space are extensively studied in courses on Mathematical Physics. Here we summarize operations on n -component vectors that are most useful from the standpoint of the development of finite elements.

§A.3.1 Transposition

The *transpose* of a column vector \mathbf{x} is the row vector that has the same components, and is denoted by \mathbf{x}^T :

$$\mathbf{x}^T = [x_1 \quad x_2 \quad \dots \quad x_n]. \quad (\text{A.7})$$

Similarly, the transpose of a row vector is the column vector that has the same components. Transposing a vector twice yields the original vector: $(\mathbf{x}^T)^T = \mathbf{x}$.

EXAMPLE A.3

The transpose of

$$\mathbf{a} = \begin{bmatrix} 3 \\ -1 \\ 6 \end{bmatrix} \quad \text{is} \quad \mathbf{b} = \mathbf{a}^T = [3 \quad -1 \quad 6]$$

and transposing \mathbf{b} gives back \mathbf{a} .

§A.3.2 Equality

Two column vectors \mathbf{x} and \mathbf{y} of equal order n are said to be *equal* if and only if their components are equal, $x_i = y_i$, for all $i = 1, \dots, n$. We then write $\mathbf{x} = \mathbf{y}$. Similarly for row vectors.

Two vectors of different order cannot be compared for equality or inequality. A row vector cannot be directly compared to a column vector of the same order (unless their dimension is 1); one of the two has to be transposed before a comparison can be made.

§A.3.3 Addition and Subtraction

The simplest operation acting on two vectors is *addition*. The sum of two vectors of same order n , \mathbf{x} and \mathbf{y} , is written $\mathbf{x} + \mathbf{y}$ and defined to be the vector of order n

$$\mathbf{x} + \mathbf{y} \stackrel{\text{def}}{=} \begin{bmatrix} x_1 + y_1 \\ x_2 + y_2 \\ \vdots \\ x_n + y_n \end{bmatrix}. \quad (\text{A.8})$$

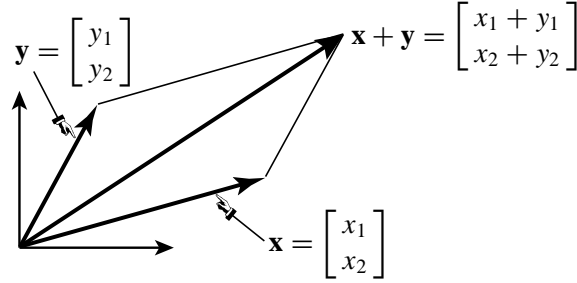


Figure A.2. In two and three-dimensional space, the vector addition operation is equivalent to the well known parallelogram composition law.

If \mathbf{x} and \mathbf{y} are not of the same order, the addition operation is undefined.

The operation is commutative: $\mathbf{x} + \mathbf{y} = \mathbf{y} + \mathbf{x}$, and associative: $\mathbf{x} + (\mathbf{y} + \mathbf{z}) = (\mathbf{x} + \mathbf{y}) + \mathbf{z}$.

Strictly speaking, the plus sign connecting \mathbf{x} and \mathbf{y} is not the same as the sign connecting x_i and y_i . However, since it enjoys the same analytical properties, there is no harm in using the same symbol in both cases.

The geometric interpretation of the vector addition operator for two- and three-dimensional vectors ($n = 2, 3$) is the well known parallelogram law; see Figure A.2. For $n = 1$ the usual scalar addition results.

For vector subtraction, replace $+$ by $-$ in the previous expressions.

EXAMPLE A.4

The sum of

$$\mathbf{a} = \begin{bmatrix} 3 \\ -1 \\ 6 \end{bmatrix} \quad \text{and} \quad \mathbf{b} = \begin{bmatrix} 2 \\ 1 \\ -4 \end{bmatrix} \quad \text{is} \quad \mathbf{a} + \mathbf{b} = \begin{bmatrix} 5 \\ 0 \\ 2 \end{bmatrix}$$

§A.3.4 Multiplication and Division by Scalar, Span

Multiplication of a vector \mathbf{x} by a scalar c is defined by means of the relation

$$c \mathbf{x} \stackrel{\text{def}}{=} \begin{bmatrix} cx_1 \\ cx_2 \\ \vdots \\ cx_n \end{bmatrix} \quad (\text{A.9})$$

This operation is often called *scaling* of a vector. The geometrical interpretation of scaling is: the scaled vector points the same way, but its magnitude is multiplied by c .

If $c = 0$, the result is the null vector. If $c < 0$ the direction of the vector is reversed. In particular, if $c = -1$ the resulting operation $(-1)\mathbf{x} = -\mathbf{x}$ is called *reflexion about the origin* or simply *reflexion*.

Division of a vector by a scalar $c \neq 0$ is equivalent to multiplication by $1/c$. The operation is written

$$\frac{\mathbf{x}}{c} \equiv \mathbf{x}/c \stackrel{\text{def}}{=} (1/c)\mathbf{x} \quad (\text{A.10})$$

The operation is not defined if $c = 0$.

Sometimes it is not just \mathbf{x} which is of interest but the “line” determined by \mathbf{x} , namely the collection $c\mathbf{x}$ of all scalar multiples of \mathbf{x} , including $-\mathbf{x}$. We call this $\text{Span}(\mathbf{x})$. Note that the span always includes the null vector.

§A.3.5 Inner Product, Norm and Length

The *inner product* of two column vectors \mathbf{x} and \mathbf{y} , of same order n , is a scalar function denoted by (\mathbf{x}, \mathbf{y}) — as well as two other forms shown below — and is defined by

$$(\mathbf{x}, \mathbf{y}) = \mathbf{x}^T \mathbf{y} = \mathbf{y}^T \mathbf{x} \stackrel{\text{def}}{=} \sum_{i=1}^n x_i y_i \stackrel{\text{sc}}{=} x_i y_i \quad (\text{A.11})$$

This operation is also called *dot product* and *interior product*. If the two vectors are not of the same order, the inner product is undefined. The two other notations shown in (A.11), namely $\mathbf{x}^T \mathbf{y}$ and $\mathbf{y}^T \mathbf{x}$, exhibit the inner product as a special case of the *matrix product* discussed in Appendix B.

The last expression in (A.11) applies the so-called *Einstein’s summation convention*, which implies sum on repeated indices (in this case, i) and allows the \sum operand to be dropped.

The inner product is commutative: $(\mathbf{x}, \mathbf{y}) = (\mathbf{y}, \mathbf{x})$ but not generally associative: $(\mathbf{x}, (\mathbf{y}, \mathbf{z})) \neq ((\mathbf{x}, \mathbf{y}), \mathbf{z})$. If $n = 1$ it reduces to the usual scalar product. The scalar product is of course associative.

EXAMPLE A.5

The inner product of

$$\mathbf{a} = \begin{bmatrix} 3 \\ -1 \\ 6 \end{bmatrix} \quad \text{and} \quad \mathbf{b} = \begin{bmatrix} 2 \\ 1 \\ -4 \end{bmatrix} \quad \text{is} \quad (\mathbf{a}, \mathbf{b}) = \mathbf{a}^T \mathbf{b} = 3 \times 2 + (-1) \times 1 + 6 \times (-4) = -19.$$

REMARK A.1

The inner product is not the only way of “multiplying” two vectors. There are other vector products, such as the cross or outer product, which are important in many applications such as fluid mechanics and nonlinear dynamics. They are not treated here because they are not defined when going from vectors to matrices. Furthermore they are not easily generalized for vectors of dimension $n \geq 4$.

The *Euclidean norm* or *2-norm* of a real vector \mathbf{x} is a scalar denoted by $\|\mathbf{x}\|$ that results by taking the inner product of the vector with itself:

$$\|\mathbf{x}\| \stackrel{\text{def}}{=} (\mathbf{x}, \mathbf{x}) = \sum_{i=1}^n x_i^2 \stackrel{\text{sc}}{=} x_i x_i \quad (\text{A.12})$$

Because the norm is a sum of squares, it is zero only if \mathbf{x} is the null vector. It thus provides a “meter” (mathematically, a norm) on the vector magnitude.

The *Euclidean length* or simply *length* of a real vector, denoted by $|\mathbf{x}|$, is the positive square root of its Euclidean norm:

$$|\mathbf{x}| \stackrel{\text{def}}{=} +\sqrt{\|\mathbf{x}\|}. \quad (\text{A.13})$$

This definition agrees with the intuitive concept of vector magnitude in two- and three-dimensional space ($n = 2, 3$). For $n = 1$ observe that the length of a scalar x is its absolute value $|x|$, hence the notation $|\mathbf{x}|$.

The Euclidean norm is not the only vector norm used in practice, but it will be sufficient for the present course.

Two important inequalities satisfied by vector norms (and not just the Euclidean norm) are the *Cauchy-Schwarz inequality*:

$$|(\mathbf{x}, \mathbf{y})| \leq |\mathbf{x}| |\mathbf{y}|, \quad (\text{A.14})$$

and the *triangle inequality*:

$$|\mathbf{x} + \mathbf{y}| \leq |\mathbf{x}| + |\mathbf{y}|. \quad (\text{A.15})$$

EXAMPLE A.6

The Euclidean norm of

$$\mathbf{x} = \begin{bmatrix} 4 \\ 3 \end{bmatrix}$$

is $\|\mathbf{x}\| = 4^2 + 3^2 = 25$, and its length is $|\mathbf{x}| = \sqrt{25} = 5$.

§A.3.6 Unit Vectors and Normalization

A vector of length one is called a *unit vector*. Any non-null vector \mathbf{x} can be scaled to unit length by dividing all components by its original length:

$$\mathbf{x}/|\mathbf{x}| = \begin{bmatrix} x_1/|\mathbf{x}| \\ x_2/|\mathbf{x}| \\ \vdots \\ x_n/|\mathbf{x}| \end{bmatrix} \quad (\text{A.16})$$

This particular scaling is called *normalization to unit length*. If \mathbf{x} is the null vector, the normalization operation is undefined.

EXAMPLE A.7

The unit-vector normalization of

$$\mathbf{x} = \begin{bmatrix} 4 \\ 3 \end{bmatrix} \quad \text{is} \quad \mathbf{x}/|\mathbf{x}| = \mathbf{x}/5 = \begin{bmatrix} 0.8 \\ 0.6 \end{bmatrix}$$

§A.3.7 Angles, Orthonormality and Projections

The *angle* in radians between two *unit* real vectors \mathbf{x} and \mathbf{y} , written $\angle(\mathbf{x}, \mathbf{y})$, is the real number θ satisfying $0 \leq \theta \leq \pi$ (or $0^\circ \leq \theta \leq 180^\circ$ if measured in degrees). The value is defined by the cosine formula:

$$\cos \theta = (\mathbf{x}, \mathbf{y}) \quad (\text{A.17})$$

If the vectors are not of unit length, they should be normalized to such, and so the general formula for two arbitrary vectors is

$$\cos \theta = \left(\frac{\mathbf{x}}{|\mathbf{x}|}, \frac{\mathbf{y}}{|\mathbf{y}|} \right) = \frac{(\mathbf{x}, \mathbf{y})}{|\mathbf{x}| |\mathbf{y}|} \quad (\text{A.18})$$

The angle θ formed by a non-null vector with itself is zero. Note that this will always be the case for $n = 1$. If one of the vectors is null, the angle is undefined.

The definition (A.18) agrees with that of the angle formed by *oriented* lines in two- and three-dimensional Euclidean geometry ($n = 2, 3$). The generalization to n dimensions is a natural one.

For some applications the *acute* angle ϕ between the line on \mathbf{x} and the line on \mathbf{y} (i.e., between the vector spans) is more appropriate than θ . This angle between $\text{Span}(\mathbf{x})$ and $\text{Span}(\mathbf{y})$ is the real number ϕ satisfying $0 \leq \phi \leq \pi/2$ and

$$\cos \phi = \frac{|(\mathbf{x}, \mathbf{y})|}{|\mathbf{x}| |\mathbf{y}|} \quad (\text{A.19})$$

Two vectors \mathbf{x} and \mathbf{y} connected by the relation

$$(\mathbf{x}, \mathbf{y}) = 0 \quad (\text{A.20})$$

are said to be *orthogonal*. The acute angle formed by two orthogonal vectors is $\pi/2$ radians or 90° .

The *projection* of a vector \mathbf{y} onto a vector \mathbf{x} is the vector \mathbf{p} that has the direction of \mathbf{x} and is orthogonal to $\mathbf{y} - \mathbf{p}$. The condition can be stated as

$$(\mathbf{p}, \mathbf{y} - \mathbf{p}) = 0, \quad \text{where} \quad \mathbf{p} = c \frac{\mathbf{x}}{|\mathbf{x}|} \quad (\text{A.21})$$

and it is easily shown that

$$c = \left(\mathbf{y}, \frac{\mathbf{x}}{|\mathbf{x}|} \right) = |\mathbf{y}| \cos \theta \quad (\text{A.22})$$

where θ is the angle between \mathbf{x} and \mathbf{y} . If \mathbf{x} and \mathbf{y} are orthogonal, the projection of any of them onto the other vanishes.

§A.3.8 Orthogonal Bases, Subspaces

Let \mathbf{b}^k , $k = 1, \dots, m$ be a set of n -dimensional *unit* vectors which are *mutually orthogonal*. Then if a particular n -dimensional vector \mathbf{x} admits the representation

$$\mathbf{x} = \sum_{k=1}^m c_k \mathbf{b}^k \quad (\text{A.23})$$

the coefficients c_k are given by the inner products

$$c_k = (\mathbf{x}, \mathbf{b}^k) = \sum_{i=1}^n x_i b_i^k \stackrel{\text{sc}}{=} x_i b_i^k. \quad (\text{A.24})$$

We also have *Parseval's equality*

$$(\mathbf{x}, \mathbf{x}) = \sum_{k=1}^m c_k^2 \stackrel{\text{sc}}{=} c_k c_k \quad (\text{A.25})$$

If the representation (A.23) holds, the set \mathbf{b}^k is called an *orthonormal basis* for the vector \mathbf{x} . The \mathbf{b}^k are called the *base vectors*.

The set of all vectors \mathbf{x} given by (A.24) forms a *subspace* of dimension m . The subspace is said to be *spanned* by the basis \mathbf{b}^k , and is called *Span*(\mathbf{b}^k).^{*} The numbers c_k are called the *coordinates* of \mathbf{x} with respect to that basis.

If $m = n$, the set \mathbf{b}^k forms a *complete orthonormal basis* for the n -dimensional space. (The qualifier “complete” means that all n -dimensional vectors are representable in terms of such a basis.)

The simplest complete orthonormal basis is of order n is the n -dimensional *Cartesian basis*

$$\mathbf{b}^1 = \begin{bmatrix} 1 \\ 0 \\ \vdots \\ 0 \end{bmatrix}, \quad \mathbf{b}^2 = \begin{bmatrix} 0 \\ 1 \\ \vdots \\ 0 \end{bmatrix}, \quad \dots \quad \mathbf{b}^n = \begin{bmatrix} 0 \\ 0 \\ \vdots \\ 1 \end{bmatrix} \quad (\text{A.26})$$

In this case the coordinates of \mathbf{x} are simply its components, that is, $c_k \equiv x_k$.

^{*} Note that if $m = 1$ we have the span of a single vector, which as noted before is simply a line passing through the origin of coordinates.

Homework Exercises for Appendix A: Vectors

EXERCISE A.1

Given the four-dimensional vectors

$$\mathbf{x} = \begin{bmatrix} 2 \\ 4 \\ 4 \\ -8 \end{bmatrix}, \quad \mathbf{y} = \begin{bmatrix} 1 \\ -5 \\ 7 \\ 5 \end{bmatrix} \quad (\text{EA.1})$$

- compute the Euclidean norms and lengths of \mathbf{x} and \mathbf{y} ;
- compute the inner product (\mathbf{x}, \mathbf{y}) ;
- verify that the inequalities (A.15) and (A.16) hold;
- normalize \mathbf{x} and \mathbf{y} to unit length;
- compute the angles $\theta = \angle(\mathbf{x}, \mathbf{y})$ and ϕ given by (A.19) and (A.20);
- compute the projection of \mathbf{y} onto \mathbf{x} and verify that the orthogonality condition (A.22) holds.

EXERCISE A.2

Given the base vectors

$$\mathbf{b}^1 = \frac{1}{10} \begin{bmatrix} 1 \\ -7 \\ 1 \\ 7 \end{bmatrix}, \quad \mathbf{b}^2 = \frac{1}{10} \begin{bmatrix} 5 \\ 5 \\ -5 \\ 5 \end{bmatrix} \quad (\text{EA.2})$$

- Check that \mathbf{b}^1 and \mathbf{b}^2 are orthonormal, i.e., orthogonal and of unit length.
- Compute the coefficients c_1 and c_2 in the following representation:

$$\mathbf{z} = \begin{bmatrix} 8 \\ -16 \\ -2 \\ 26 \end{bmatrix} = c_1 \mathbf{b}^1 + c_2 \mathbf{b}^2 \quad (\text{EA.3})$$

- Using the values computed in (b), verify that the coordinate-expansion formulas (A.24) and Parseval's equality (A.25) are correct.

EXERCISE A.3

Prove that Parseval's equality holds in general.

EXERCISE A.4

What are the angles θ and ϕ formed by an arbitrary non-null vector \mathbf{x} and the opposite vector $-\mathbf{x}$?

EXERCISE A.5

Show that

$$(\alpha \mathbf{x}, \beta \mathbf{y}) = \alpha \beta (\mathbf{x}, \mathbf{y}) \quad (\text{EA.4})$$

where \mathbf{x} and \mathbf{y} are arbitrary vectors, and α and β are scalars.

Homework Exercises for Appendix A - Solutions

EXERCISE A.1

(a)

$$\|\mathbf{x}\| = 2^2 + 4^2 + 4^2 + (-8)^2 = 100, \quad |\mathbf{x}| = 10$$

$$\|\mathbf{y}\| = 1^2 + (-5)^2 + 7^2 + 5^2 = 100, \quad |\mathbf{y}| = 10$$

(b)

$$(\mathbf{x}, \mathbf{y}) = 2 - 20 + 28 - 40 = -30$$

(c)

$$|(\mathbf{x}, \mathbf{y})| = 30 \leq |\mathbf{x}| |\mathbf{y}| = 100$$

$$|\mathbf{x} + \mathbf{y}| = \sqrt{3^2 + (-1)^2 + (-11)^2 + (-3)^2} = \sqrt{140} \leq |\mathbf{x}| + |\mathbf{y}| = 10 + 10 = 20$$

(d)

$$\mathbf{x}'' = \frac{\mathbf{x}}{10} = \begin{bmatrix} 0.2 \\ 0.4 \\ 0.4 \\ -0.8 \end{bmatrix}, \quad \mathbf{y}'' = \frac{\mathbf{y}}{10} = \begin{bmatrix} 0.1 \\ -0.5 \\ 0.7 \\ 0.5 \end{bmatrix}$$

(e)

$$\cos \theta = (\mathbf{x}'', \mathbf{y}'') = -0.30, \quad \theta = 107.46^\circ$$

$$\cos \phi = |(\mathbf{x}'', \mathbf{y}'')| = 0.30, \quad \phi = 72.54^\circ$$

(f)

$$c = |\mathbf{y}| \cos \theta = -3$$

$$\mathbf{p} = c \mathbf{x}'' = -3 \begin{bmatrix} 0.2 \\ 0.4 \\ 0.4 \\ -0.8 \end{bmatrix} = \begin{bmatrix} -0.6 \\ -1.2 \\ -1.2 \\ 2.4 \end{bmatrix}$$

$$\mathbf{y} - \mathbf{p} = \begin{bmatrix} 1.6 \\ -3.8 \\ 8.2 \\ 2.6 \end{bmatrix}$$

$$(\mathbf{p}, \mathbf{y} - \mathbf{p}) = -0.96 + 4.56 - 9.84 + 6.24 = 0$$

EXERCISE A.2

(a)

$$\|\mathbf{b}^1\| = 0.1^2 + (-0.7)^2 + 0.1^2 + 0.7^2 = 1, \quad |\mathbf{b}^1| = 1$$

$$\|\mathbf{b}^2\| = 0.5^2 + 0.5^2 + (-0.5)^2 + 0.5^2 = 1, \quad |\mathbf{b}^2| = 1$$

$$(\mathbf{b}^1, \mathbf{b}^2) = 0.05 - 0.35 - 0.05 + 0.35 = 0$$

(b)

$$c_1 = 30, \quad c_2 = 10$$

(by inspection, or solving a system of 2 linear equations)

(c)

$$c_1 = (\mathbf{z}, \mathbf{b}^1) = 0.8 + 11.2 - 0.2 + 18.2 = 30$$

$$c_2 = (\mathbf{z}, \mathbf{b}^2) = 4.0 - 8.0 + 1.0 + 13.0 = 10$$

$$c_1^2 + c_2^2 = 30^2 + 10^2 = 1000 = \|\mathbf{z}\|^2 = 8^2 + (-16)^2 + (-2)^2 + 26^2 = 1000$$

EXERCISE A.3

See any linear algebra book, e.g. Strang's.

EXERCISE A.4

$$\theta = 180^\circ, \quad \phi = 0^\circ$$

EXERCISE A.5

$$(\alpha \mathbf{x}, \beta \mathbf{y}) = \alpha x_i \beta y_i = \alpha \beta x_i y_i = \alpha \beta (\mathbf{x}, \mathbf{y})$$

(summation convention used)

B

Matrix Algebra: Matrices

§B.1 MATRICES

§B.1.1 Concept

Let us now introduce the concept of a *matrix*. Consider a set of scalar quantities arranged in a rectangular array containing m rows and n columns:

$$\begin{bmatrix} a_{11} & a_{12} & \cdots & a_{1j} & \cdots & a_{1n} \\ a_{21} & a_{22} & \cdots & a_{2j} & \cdots & a_{2n} \\ \vdots & \vdots & \ddots & \vdots & \ddots & \vdots \\ a_{i1} & a_{i2} & \cdots & a_{ij} & \cdots & a_{in} \\ \vdots & \vdots & \ddots & \vdots & \ddots & \vdots \\ a_{m1} & a_{m2} & \cdots & a_{mj} & \cdots & a_{mn} \end{bmatrix}. \quad (\text{B.1})$$

This array will be called a *rectangular matrix* of order m by n , or, briefly, an $m \times n$ matrix. Not every rectangular array is a matrix; to qualify as such it must obey the operational rules discussed below.

The quantities a_{ij} are called the *entries* or *components* of the matrix. Preference will be given to the latter unless one is talking about the computer implementation. As in the case of vectors, the term “matrix element” will be avoided to lessen the chance of confusion with finite elements. The two subscripts identify the row and column, respectively.

Matrices are conventionally identified by **bold uppercase** letters such as **A**, **B**, etc. The entries of matrix **A** may be denoted as A_{ij} or a_{ij} , according to the intended use. Occasionally we shall use the short-hand component notation

$$\mathbf{A} = [a_{ij}]. \quad (\text{B.2})$$

EXAMPLE B.1

The following is a 2×3 numerical matrix:

$$\mathbf{B} = \begin{bmatrix} 2 & 6 & 3 \\ 4 & 9 & 1 \end{bmatrix} \quad (\text{B.3})$$

This matrix has 2 rows and 3 columns. The first row is (2, 6, 3), the second row is (4, 9, 1), the first column is (2, 4), and so on.

In some contexts it is convenient or useful to display the number of rows and columns. If this is so we will write them underneath the matrix symbol. For the example matrix (B.3) we would show

$$\mathbf{B}_{2 \times 3} \quad (\text{B.4})$$

REMARK B.1

Matrices should not be confused with determinants. A determinant is a number associated with square matrices ($m = n$), defined according to the rules stated in Appendix C.

§B.1.2 Real and Complex Matrices

As in the case of vectors, the components of a matrix may be real or complex. If they are real numbers, the matrix is called *real*, and *complex* otherwise. For the present exposition all matrices will be real.

§B.1.3 Square Matrices

The case $m = n$ is important in practical applications. Such matrices are called *square matrices* of order n . Matrices for which $m \neq n$ are called non-square (the term “rectangular” is also used in this context, but this is fuzzy because squares are special cases of rectangles).

Square matrices enjoy certain properties not shared by non-square matrices, such as the symmetry and antisymmetry conditions defined below. Furthermore many operations, such as taking determinants and computing eigenvalues, are only defined for square matrices.

EXAMPLE B.2

$$\mathbf{C} = \begin{bmatrix} 12 & 6 & 3 \\ 8 & 24 & 7 \\ 2 & 5 & 11 \end{bmatrix} \quad (\text{B.5})$$

is a square matrix of order 3.

Consider a square matrix $\mathbf{A} = [a_{ij}]$ of order $n \times n$. Its n components a_{ii} form the *main diagonal*, which runs from top left to bottom right. The *cross diagonal* runs from the bottom left to upper right. The main diagonal of the example matrix (B.5) is $\{12, 24, 11\}$ and the cross diagonal is $\{2, 24, 3\}$.

Entries that run parallel to and above (below) the main diagonal form superdiagonals (subdiagonals). For example, $\{6, 7\}$ is the first superdiagonal of the example matrix (B.5).

§B.1.4 Symmetry and Antisymmetry

Square matrices for which $a_{ij} = a_{ji}$ are called *symmetric about the main diagonal* or simply *symmetric*.

Square matrices for which $a_{ij} = -a_{ji}$ are called *antisymmetric* or *skew-symmetric*. The diagonal entries of an antisymmetric matrix must be zero.

EXAMPLE B.3

The following is a symmetric matrix of order 3:

$$\mathbf{S} = \begin{bmatrix} 11 & 6 & 1 \\ 6 & 3 & -1 \\ 1 & -1 & -6 \end{bmatrix}. \quad (\text{B.6})$$

The following is an antisymmetric matrix of order 4:

$$\mathbf{W} = \begin{bmatrix} 0 & 3 & -1 & -5 \\ -3 & 0 & 7 & -2 \\ 1 & -7 & 0 & 0 \\ 5 & 2 & 0 & 0 \end{bmatrix}. \quad (\text{B.7})$$

§B.1.5 Are Vectors a Special Case of Matrices?

Consider the 3-vector \mathbf{x} and a 3×1 matrix \mathbf{X} with the same components:

$$\mathbf{x} = \begin{bmatrix} x_1 \\ x_2 \\ x_3 \end{bmatrix}, \quad \mathbf{X} = \begin{bmatrix} x_{11} \\ x_{21} \\ x_{31} \end{bmatrix}. \quad (\text{B.8})$$

in which $x_1 = x_{11}$, $x_2 = x_{21}$ and $x_3 = x_{31}$. Are \mathbf{x} and \mathbf{X} the same thing? If so we could treat column vectors as one-column matrices and dispense with the distinction.

Indeed in many contexts a column vector of order n may be treated as a matrix with a single column, i.e., as a matrix of order $n \times 1$. Similarly, a row vector of order m may be treated as a matrix with a single row, i.e., as a matrix of order $1 \times m$.

There are some operations, however, for which the analogy does not carry over, and one has to consider vectors as different from matrices. The dichotomy is reflected in the notational conventions of lower versus upper case. Another important distinction from a practical standpoint is discussed next.

§B.1.6 Where Do Matrices Come From?

Although we speak of “matrix algebra” as embodying vectors as special cases of matrices, in practice the quantities of primary interest to the structural engineer are vectors rather than matrices. For example, an engineer may be interested in displacement vectors, force vectors, vibration eigenvectors, buckling eigenvectors. In finite element analysis even stresses and strains are often arranged as vectors although they are really tensors.

On the other hand, matrices are rarely the quantities of primary interest: they work silently in the background where they are normally engaged in operating on vectors.

§B.1.7 Special Matrices

The *null* matrix, written $\mathbf{0}$, is the matrix all of whose components are zero.

EXAMPLE B.4

The null matrix of order 2×3 is

$$\begin{bmatrix} 0 & 0 & 0 \\ 0 & 0 & 0 \end{bmatrix}. \quad (\text{B.9})$$

The *identity matrix*, written \mathbf{I} , is a square matrix all of which entries are zero except those on the main diagonal, which are ones.

EXAMPLE B.5

The identity matrix of order 4 is

$$\mathbf{I} = \begin{bmatrix} 1 & 0 & 0 & 0 \\ 0 & 1 & 0 & 0 \\ 0 & 0 & 1 & 0 \\ 0 & 0 & 0 & 1 \end{bmatrix}. \quad (\text{B.10})$$

A *diagonal matrix* is a square matrix all of which entries are zero except for those on the main diagonal, which may be arbitrary.

EXAMPLE B.6

The following matrix of order 4 is diagonal:

$$\mathbf{D} = \begin{bmatrix} 14 & 0 & 0 & 0 \\ 0 & -6 & 0 & 0 \\ 0 & 0 & 0 & 0 \\ 0 & 0 & 0 & 3 \end{bmatrix}. \quad (\text{B.11})$$

A short hand notation which lists only the diagonal entries is sometimes used for diagonal matrices to save writing space. This notation is illustrated for the above matrix:

$$\mathbf{D} = \mathbf{diag} [14 \quad -6 \quad 0 \quad 3]. \quad (\text{B.12})$$

An *upper triangular* matrix is a square matrix in which all elements underneath the main diagonal vanish. A *lower triangular* matrix is a square matrix in which all entries above the main diagonal vanish.

EXAMPLE B.7

Here are examples of each kind:

$$\mathbf{U} = \begin{bmatrix} 6 & 4 & 2 & 1 \\ 0 & 6 & 4 & 2 \\ 0 & 0 & 6 & 4 \\ 0 & 0 & 0 & 6 \end{bmatrix}, \quad \mathbf{L} = \begin{bmatrix} 5 & 0 & 0 & 0 \\ 10 & 4 & 0 & 0 \\ -3 & 21 & 6 & 0 \\ -15 & -2 & 18 & 7 \end{bmatrix}. \quad (\text{B.13})$$

§B.2 ELEMENTARY MATRIX OPERATIONS

§B.2.1 Equality

Two matrices \mathbf{A} and \mathbf{B} of same order $m \times n$ are said to be *equal* if and only if all of their components are equal: $a_{ij} = b_{ij}$, for all $i = 1, \dots, m$, $j = 1, \dots, n$. We then write $\mathbf{A} = \mathbf{B}$. If the inequality test fails the matrices are said to be *unequal* and we write $\mathbf{A} \neq \mathbf{B}$.

Two matrices of different order cannot be compared for equality or inequality.

There is no simple test for greater-than or less-than.

§B.2.2 Transposition

The *transpose* of a matrix \mathbf{A} is another matrix denoted by \mathbf{A}^T that has n rows and m columns

$$\mathbf{A}^T = [a_{ji}]. \quad (\text{B.14})$$

The rows of \mathbf{A}^T are the columns of \mathbf{A} , and the rows of \mathbf{A} are the columns of \mathbf{A}^T .

Obviously the transpose of \mathbf{A}^T is again \mathbf{A} , that is, $(\mathbf{A}^T)^T = \mathbf{A}$.

EXAMPLE B.8

$$\mathbf{A} = \begin{bmatrix} 5 & 7 & 0 \\ 1 & 0 & 4 \end{bmatrix}, \quad \mathbf{A}^T = \begin{bmatrix} 5 & 1 \\ 7 & 0 \\ 0 & 4 \end{bmatrix}. \quad (\text{B.15})$$

The transpose of a square matrix is also a square matrix. The transpose of a symmetric matrix \mathbf{A} is equal to the original matrix, *i.e.*, $\mathbf{A} = \mathbf{A}^T$. The negated transpose of an antisymmetric matrix \mathbf{A} is equal to the original matrix, *i.e.* $\mathbf{A} = -\mathbf{A}^T$.

EXAMPLE B.9

$$\mathbf{A} = \begin{bmatrix} 4 & 7 & 0 \\ 7 & 1 & 2 \\ 0 & 2 & 3 \end{bmatrix} = \mathbf{A}^T, \quad \mathbf{W} = \begin{bmatrix} 0 & 7 & 0 \\ -7 & 0 & -2 \\ 0 & 2 & 0 \end{bmatrix} = -\mathbf{W}^T \quad (\text{B.16})$$

§B.2.3 Addition and Subtraction

The simplest operation acting on two matrices is *addition*. The sum of two matrices of the same order, \mathbf{A} and \mathbf{B} , is written $\mathbf{A} + \mathbf{B}$ and defined to be the matrix

$$\mathbf{A} + \mathbf{B} \stackrel{\text{def}}{=} [a_{ij} + b_{ij}]. \quad (\text{B.17})$$

Like vector addition, matrix addition is commutative: $\mathbf{A} + \mathbf{B} = \mathbf{B} + \mathbf{A}$, and associative: $\mathbf{A} + (\mathbf{B} + \mathbf{C}) = (\mathbf{A} + \mathbf{B}) + \mathbf{C}$. For $n = 1$ or $m = 1$ the operation reduces to the addition of two column or row vectors, respectively.

For matrix subtraction, replace $+$ by $-$ in the definition (B.17).

EXAMPLE B.10

The sum of

$$\mathbf{A} = \begin{bmatrix} 1 & -3 & 0 \\ 4 & 2 & -1 \end{bmatrix} \quad \text{and} \quad \mathbf{B} = \begin{bmatrix} 6 & 3 & -3 \\ 7 & -2 & 5 \end{bmatrix} \quad \text{is} \quad \mathbf{A} + \mathbf{B} = \begin{bmatrix} 7 & 0 & -3 \\ 11 & 0 & 4 \end{bmatrix}. \quad (\text{B.18})$$

§B.2.4 Scalar Multiplication

Multiplication of a matrix \mathbf{A} by a scalar c is defined by means of the relation

$$c\mathbf{A} \stackrel{\text{def}}{=} [ca_{ij}] \quad (\text{B.19})$$

That is, each entry of the matrix is multiplied by c . This operation is often called *scaling* of a matrix. If $c = 0$, the result is the null matrix. Division of a matrix by a nonzero scalar c is equivalent to multiplication by $(1/c)$.

EXAMPLE B.11

$$\text{If } \mathbf{A} = \begin{bmatrix} 1 & -3 & 0 \\ 4 & 2 & -1 \end{bmatrix}, \quad 3\mathbf{A} = \begin{bmatrix} 3 & -9 & 0 \\ 12 & 6 & -3 \end{bmatrix}. \quad (\text{B.20})$$

§B.3 MATRIX PRODUCTS

§B.3.1 Matrix by Vector

Before describing the general matrix product of two matrices, let us treat the particular case in which the second matrix is a column vector. This so-called *matrix-vector product* merits special attention because it occurs very frequently in the applications. Let $\mathbf{A} = [a_{ij}]$ be an $m \times n$ matrix, $\mathbf{x} = \{x_j\}$ a column vector of order n , and $\mathbf{y} = \{y_i\}$ a column vector of order m . The matrix-vector product is symbolically written

$$\mathbf{y} = \mathbf{Ax}, \quad (\text{B.21})$$

to mean the linear transformation

$$y_i \stackrel{\text{def}}{=} \sum_{j=1}^n a_{ij}x_j \stackrel{\text{sc}}{=} a_{ij}x_j, \quad i = 1, \dots, m. \quad (\text{B.22})$$

EXAMPLE B.12

The product of a 2×3 matrix and a vector of order 3 is a vector of order 2:

$$\begin{bmatrix} 1 & -3 & 0 \\ 4 & 2 & -1 \end{bmatrix} \begin{bmatrix} 1 \\ 2 \\ 3 \end{bmatrix} = \begin{bmatrix} -5 \\ 5 \end{bmatrix} \quad (\text{B.23})$$

This product definition is not arbitrary but emanates from the analytical and geometric properties of entities represented by matrices and vectors.

For the product definition to make sense, the column dimension of the matrix \mathbf{A} (called the pre-multiplicand) must equal the dimension of the vector \mathbf{x} (called the post-multiplicand). For example, the reverse product \mathbf{xA} does not make sense unless $m = n = 1$.

If the row dimension m of \mathbf{A} is one, the matrix formally reduces to a row vector (see §A.2), and the matrix-vector product reduces to the inner product defined by Equation (A.11). The result of this operation is a one-dimensional vector or scalar. We thus see that the present definition properly embodies previous cases.

The associative and commutative properties of the matrix-vector product fall under the rules of the more general matrix-matrix product discussed next.

§B.3.2 Matrix by Matrix

We now pass to the most general matrix-by-matrix product, and consider the operations involved in computing the product \mathbf{C} of two matrices \mathbf{A} and \mathbf{B} :

$$\mathbf{C} = \mathbf{AB}. \quad (\text{B.24})$$

Here $\mathbf{A} = [a_{ij}]$ is a matrix of order $m \times n$, $\mathbf{B} = [b_{jk}]$ is a matrix of order $n \times p$, and $\mathbf{C} = [c_{ik}]$ is a matrix of order $m \times p$. The entries of the result matrix \mathbf{C} are defined by the formula

$$c_{ik} \stackrel{\text{def}}{=} \sum_{j=1}^n a_{ij}b_{jk} \stackrel{\text{sc}}{=} a_{ij}b_{jk}, \quad i = 1, \dots, m, \quad k = 1, \dots, p. \quad (\text{B.25})$$

We see that the $(i, k)^{th}$ entry of \mathbf{C} is computed by taking the *inner product* of the i^{th} row of \mathbf{A} with the k^{th} column of \mathbf{B} . For this definition to work and the product be possible, *the column dimension of \mathbf{A} must be the same as the row dimension of \mathbf{B}* . Matrices that satisfy this rule are said to be *product-conforming*, or *conforming* for short. If two matrices do not conform, their product is undefined. The following mnemonic notation often helps in remembering this rule:

$$\underset{m \times p}{\mathbf{C}} = \underset{m \times n}{\mathbf{A}} \underset{n \times p}{\mathbf{B}} \quad (\text{B.26})$$

For the matrix-by-vector case treated in the preceding subsection, $p = 1$.

Matrix \mathbf{A} is called the pre-multiplicand and is said to *premultiply* \mathbf{B} . Matrix \mathbf{B} is called the post-multiplicand and is said to *postmultiply* \mathbf{A} . This careful distinction on which matrix comes first is a consequence of the absence of commutativity: even if \mathbf{BA} exists (it only does if $m = n$), it is not generally the same as \mathbf{AB} .

For *hand* computations, the matrix product is most conveniently organized by the so-called Falk's scheme:

$$\begin{bmatrix} a_{11} & \cdots & a_{1n} \\ \vdots & \ddots & \vdots \\ a_{i1} & \rightarrow & a_{in} \\ \vdots & \ddots & \vdots \\ a_{m1} & \cdots & a_{mn} \end{bmatrix} \begin{bmatrix} b_{11} & \cdots & b_{ik} & \cdots & b_{1p} \\ \vdots & \ddots & \downarrow & \ddots & \vdots \\ b_{n1} & \cdots & b_{nk} & \cdots & b_{np} \\ \vdots & & & & \\ \cdots & & c_{ik} & & \end{bmatrix}. \quad (\text{B.27})$$

Each entry in row i of \mathbf{A} is multiplied by the corresponding entry in column k of \mathbf{B} (note the arrows), and the products are summed and stored in the $(i, k)^{th}$ entry of \mathbf{C} .

EXAMPLE B.13

To illustrate Falk's scheme, let us form the product $\mathbf{C} = \mathbf{AB}$ of the following matrices

$$\mathbf{A} = \begin{bmatrix} 3 & 0 & 2 \\ 4 & -1 & 5 \end{bmatrix}, \quad \mathbf{B} = \begin{bmatrix} 2 & 1 & 0 & -5 \\ 4 & 3 & -1 & 0 \\ 0 & 1 & -7 & 4 \end{bmatrix} \quad (\text{B.28})$$

The matrices are conforming because the column dimension of \mathbf{A} and the row dimension of \mathbf{B} are the same (3). We arrange the computations as shown below:

$$\mathbf{A} = \begin{bmatrix} 3 & 0 & 2 \\ 4 & -1 & 5 \end{bmatrix} \begin{bmatrix} 2 & 1 & 0 & -5 \\ 4 & 3 & -1 & 0 \\ 0 & 1 & -7 & 4 \end{bmatrix} = \mathbf{B} \quad (\text{B.29})$$

$$\mathbf{A} = \begin{bmatrix} 3 & 0 & 2 \\ 4 & -1 & 5 \end{bmatrix} \begin{bmatrix} 6 & 5 & -14 & -7 \\ 4 & 6 & -34 & 0 \end{bmatrix} = \mathbf{C} = \mathbf{AB}$$

Here $3 \times 2 + 0 \times 4 + 2 \times 0 = 6$ and so on.

§B.3.3 Matrix Powers

If $\mathbf{A} = \mathbf{B}$, the product $\mathbf{A}\mathbf{A}$ is called the *square* of \mathbf{A} and is denoted by \mathbf{A}^2 . Note that for this definition to make sense, \mathbf{A} must be a square matrix.

Similarly, $\mathbf{A}^3 = \mathbf{A}\mathbf{A}\mathbf{A} = \mathbf{A}^2\mathbf{A} = \mathbf{A}\mathbf{A}^2$. Other positive-integer powers can be defined in an analogous manner.

This definition does not encompass negative powers. For example, \mathbf{A}^{-1} denotes the *inverse* of matrix \mathbf{A} , which is studied in Appendix C. The general power \mathbf{A}^m , where m can be a real or complex scalar, can be defined with the help of the matrix spectral form and require the notion of eigensystem.

A square matrix \mathbf{A} that satisfies $\mathbf{A} = \mathbf{A}^2$ is called *idempotent*. We shall see later that that equation characterizes the so-called projector matrices.

A square matrix \mathbf{A} whose p^{th} power is the null matrix is called *p-nilpotent*.

§B.3.4 Properties of Matrix Products

Associativity. The associative law is verified:

$$\mathbf{A}(\mathbf{B}\mathbf{C}) = (\mathbf{A}\mathbf{B})\mathbf{C}. \quad (\text{B.30})$$

Hence we may delete the parentheses and simply write \mathbf{ABC} .

Distributivity. The distributive law also holds: If \mathbf{B} and \mathbf{C} are matrices of the same order, then

$$\mathbf{A}(\mathbf{B} + \mathbf{C}) = \mathbf{A}\mathbf{B} + \mathbf{A}\mathbf{C}, \quad \text{and} \quad (\mathbf{B} + \mathbf{C})\mathbf{A} = \mathbf{B}\mathbf{A} + \mathbf{C}\mathbf{A}. \quad (\text{B.31})$$

Commutativity. The commutativity law of scalar multiplication does not generally hold. If \mathbf{A} and \mathbf{B} are square matrices of the same order, then the products \mathbf{AB} and \mathbf{BA} are both possible but in general $\mathbf{AB} \neq \mathbf{BA}$.

If $\mathbf{AB} = \mathbf{BA}$, the matrices \mathbf{A} and \mathbf{B} are said to *commute*. One important case is when \mathbf{A} and \mathbf{B} are diagonal. In general \mathbf{A} and \mathbf{B} commute if they share the same eigensystem.

EXAMPLE B.14

Matrices

$$\mathbf{A} = \begin{bmatrix} a & b \\ b & c \end{bmatrix}, \quad \mathbf{B} = \begin{bmatrix} a - \beta & b \\ b & c - \beta \end{bmatrix}, \quad (\text{B.32})$$

commute for any a, b, c, β . More generally, \mathbf{A} and $\mathbf{B} = \mathbf{A} - \beta\mathbf{I}$ commute for any square matrix \mathbf{A} .

Transpose of a Product. The transpose of a matrix product is equal to the product of the transposes of the operands taken in reverse order:

$$(\mathbf{AB})^T = \mathbf{B}^T \mathbf{A}^T. \quad (\text{B.33})$$

The general transposition formula for an arbitrary product sequence is

$$(\mathbf{ABC} \dots \mathbf{MN})^T = \mathbf{N}^T \mathbf{M}^T \dots \mathbf{C}^T \mathbf{B}^T \mathbf{A}^T. \quad (\text{B.34})$$

Congruential Transformation. If \mathbf{B} is a symmetric matrix of order m and \mathbf{A} is an arbitrary $m \times n$ matrix, then

$$\mathbf{S} = \mathbf{A}^T \mathbf{B} \mathbf{A}. \quad (\text{B.35})$$

is a symmetric matrix of order n . Such an operation is called a congruential transformation. It occurs very frequently in finite element analysis when changing coordinate bases because such a transformation preserves energy.

Loss of Symmetry. The product of two symmetric matrices is not generally symmetric.

Null Matrices may have Non-null Divisors. The matrix product \mathbf{AB} can be zero although $\mathbf{A} \neq \mathbf{0}$ and $\mathbf{B} \neq \mathbf{0}$. Similar, it is possible that $\mathbf{A} \neq \mathbf{0}$, $\mathbf{A}^2 \neq \mathbf{0}$, \dots , but $\mathbf{A}^p = \mathbf{0}$.

§B.4 BILINEAR AND QUADRATIC FORMS

Let \mathbf{x} and \mathbf{y} be two column vectors of order n , and \mathbf{A} a real square $n \times n$ matrix. Then the following triple product produces a *scalar* result:

$$s = \underset{1 \times n}{\mathbf{y}^T} \underset{n \times n}{\mathbf{A}} \underset{n \times 1}{\mathbf{x}} \quad (\text{B.36})$$

This is called a *bilinear form*.

Transposing both sides of (B.36) and noting that the transpose of a scalar does not change, we obtain the result

$$s = \mathbf{x}^T \mathbf{A}^T \mathbf{y}. \quad (\text{B.37})$$

If \mathbf{A} is symmetric and vectors \mathbf{x} and \mathbf{y} coalesce, *i.e.*

$$\mathbf{A}^T = \mathbf{A}, \quad \mathbf{x} = \mathbf{y}, \quad (\text{B.38})$$

the bilinear form becomes a *quadratic form*

$$s = \mathbf{x}^T \mathbf{A} \mathbf{x}. \quad (\text{B.39})$$

Transposing both sides of a quadratic form reproduces the same equation.

EXAMPLE B.15

The kinetic energy of a system consisting of three point masses m_1, m_2, m_3 is

$$T = \frac{1}{2}(m_1 v_1^2 + m_2 v_2^2 + m_3 v_3^2). \quad (\text{B.40})$$

This can be expressed as the quadratic form

$$T = \frac{1}{2} \mathbf{u}^T \mathbf{M} \mathbf{u} \quad (\text{B.41})$$

where

$$\mathbf{M} = \begin{bmatrix} m_1 & 0 & 0 \\ 0 & m_2 & 0 \\ 0 & 0 & m_3 \end{bmatrix}, \quad \mathbf{u} = \begin{bmatrix} u_1 \\ u_2 \\ u_3 \end{bmatrix}. \quad (\text{B.42})$$

Homework Exercises for Appendix B: Matrices

EXERCISE B.1

Given the three matrices

$$\mathbf{A} = \begin{bmatrix} 2 & 4 & 1 & 0 \\ -1 & 2 & 3 & 1 \\ 2 & 5 & -1 & 2 \end{bmatrix}, \quad \mathbf{B} = \begin{bmatrix} 2 & -2 \\ 1 & 0 \\ 4 & 1 \\ -3 & 2 \end{bmatrix}, \quad \mathbf{C} = \begin{bmatrix} 1 & -3 & 2 \\ 2 & 0 & 2 \end{bmatrix} \quad (\text{EB.1})$$

compute the product $\mathbf{D} = \mathbf{ABC}$ by hand using Falk's scheme. (*Hint:* do \mathbf{BC} first, then premultiply that by \mathbf{A} .)

EXERCISE B.2

Given the square matrices

$$\mathbf{A} = \begin{bmatrix} 1 & 3 \\ -4 & 2 \end{bmatrix}, \quad \mathbf{B} = \begin{bmatrix} 3 & 0 \\ 1 & -2 \end{bmatrix} \quad (\text{EB.2})$$

verify by direct computation that $\mathbf{AB} \neq \mathbf{BA}$.

EXERCISE B.3

Given the matrices

$$\mathbf{A} = \begin{bmatrix} 1 & 0 \\ -1 & 2 \\ 2 & 0 \end{bmatrix}, \quad \mathbf{B} = \begin{bmatrix} 3 & -1 & 4 \\ -1 & 2 & 0 \\ 4 & 0 & 0 \end{bmatrix} \quad (\text{EB.3})$$

(note that \mathbf{B} is symmetric) compute $\mathbf{S} = \mathbf{A}^T \mathbf{BA}$, and verify that \mathbf{S} is symmetric.

EXERCISE B.4

Given the square matrices

$$\mathbf{A} = \begin{bmatrix} 3 & -1 & 2 \\ 1 & 0 & 3 \\ 3 & -2 & -5 \end{bmatrix}, \quad \mathbf{B} = \begin{bmatrix} 3 & -6 & -3 \\ 7 & -14 & -7 \\ -1 & 2 & 1 \end{bmatrix} \quad (\text{EB.4})$$

verify that $\mathbf{AB} = \mathbf{0}$ although $\mathbf{A} \neq \mathbf{0}$ and $\mathbf{B} \neq \mathbf{0}$. Is \mathbf{BA} also null?

EXERCISE B.5

Given the square matrix

$$\mathbf{A} = \begin{bmatrix} 0 & a & b \\ 0 & 0 & c \\ 0 & 0 & 0 \end{bmatrix} \quad (\text{EB.5})$$

show by direct computation that $\mathbf{A}^2 \neq \mathbf{0}$ but $\mathbf{A}^3 = \mathbf{0}$.

EXERCISE B.6

Can a diagonal matrix be antisymmetric?

EXERCISE B.7

(Tougher) Prove (B.33). (*Hint*: call $\mathbf{C} = (\mathbf{AB})^T$, $\mathbf{D} = \mathbf{B}^T \mathbf{A}^T$, and use the matrix product definition (B.25) to show that the generic entries of \mathbf{C} and \mathbf{D} agree.)

EXERCISE B.8

If \mathbf{A} is an arbitrary $m \times n$ matrix, show: (a) both products $\mathbf{A}^T \mathbf{A}$ and $\mathbf{A} \mathbf{A}^T$ are possible, and (b) both products are square and symmetric. (*Hint*: for (b) make use of the symmetry condition $\mathbf{S} = \mathbf{S}^T$ and of (B.31).)

EXERCISE B.9

Show that \mathbf{A}^2 only exists if and only if \mathbf{A} is square.

EXERCISE B.10

If \mathbf{A} is square and antisymmetric, show that \mathbf{A}^2 is symmetric. (*Hint*: start from $\mathbf{A} = -\mathbf{A}^T$ and apply the results of Exercise B.8.)

Homework Exercises for Appendix B - Solutions

EXERCISE B.1

$$\begin{aligned}
 \mathbf{B} &= \begin{bmatrix} 2 & -2 \\ 1 & 0 \\ 4 & 1 \\ -3 & 2 \end{bmatrix} \quad \begin{bmatrix} 1 & -3 & 2 \\ 2 & 0 & 2 \end{bmatrix} = \mathbf{C} \\
 &\quad \begin{bmatrix} -2 & -6 & 0 \\ 1 & -3 & 2 \\ 6 & -12 & 10 \\ 1 & 9 & -2 \end{bmatrix} = \mathbf{BC} \\
 \mathbf{A} &= \begin{bmatrix} 2 & 4 & 1 & 0 \\ -1 & 2 & 3 & 1 \\ 2 & 5 & -1 & 2 \end{bmatrix} \quad \begin{bmatrix} 6 & -36 & 18 \\ 23 & -27 & 32 \\ -3 & 3 & -4 \end{bmatrix} = \mathbf{ABC} = \mathbf{D}
 \end{aligned}$$

EXERCISE B.2

$$\mathbf{AB} = \begin{bmatrix} 6 & -6 \\ -10 & -4 \end{bmatrix} \neq \mathbf{BA} = \begin{bmatrix} 3 & 9 \\ 9 & -1 \end{bmatrix}$$

EXERCISE B.3

$$\mathbf{S} = \mathbf{A}^T \mathbf{BA} = \begin{bmatrix} 23 & -6 \\ -6 & 8 \end{bmatrix}$$

which is symmetric, like \mathbf{B} .

EXERCISE B.4

$$\begin{aligned}
 \mathbf{A} &= \begin{bmatrix} 3 & -1 & 2 \\ 1 & 0 & 3 \\ 3 & -2 & -5 \end{bmatrix} \quad \begin{bmatrix} 3 & -6 & -3 \\ 7 & -14 & -7 \\ -1 & 2 & 1 \end{bmatrix} = \mathbf{B} \\
 &\quad \begin{bmatrix} 0 & 0 & 0 \\ 0 & 0 & 0 \\ 0 & 0 & 0 \end{bmatrix} = \mathbf{AB} = \mathbf{0}
 \end{aligned}$$

However,

$$\mathbf{BA} = \begin{bmatrix} -6 & 3 & 3 \\ -14 & 7 & 7 \\ 2 & -1 & -1 \end{bmatrix} \neq \mathbf{0}$$

EXERCISE B.5

$$\mathbf{A}^2 = \mathbf{AA} = \begin{bmatrix} 0 & 0 & ac \\ 0 & 0 & 0 \\ 0 & 0 & 0 \end{bmatrix}, \quad \mathbf{A}^3 = \mathbf{AAA} = \begin{bmatrix} 0 & 0 & 0 \\ 0 & 0 & 0 \\ 0 & 0 & 0 \end{bmatrix} = \mathbf{0}$$

EXERCISE B.6

Only if it is the null matrix.

EXERCISE B.7

To avoid “indexing indigestion” let us carefully specify the dimensions of the given matrices and their transposes:

$$\begin{aligned}\mathbf{A} &= [a_{ij}]_{m \times n}, & \mathbf{A}^T &= [a_{ji}]_{n \times m} \\ \mathbf{B} &= [b_{jk}]_{n \times p}, & \mathbf{B}^T &= [b_{kj}]_{p \times n}\end{aligned}$$

Indices i, j and k run over $1 \dots m, 1 \dots n$ and $1 \dots p$, respectively. Now call

$$\mathbf{C} = [c_{ki}]_{p \times m} = (\mathbf{AB})^T$$

$$\mathbf{D} = [d_{ki}]_{p \times m} = \mathbf{B}^T \mathbf{A}^T$$

From the definition of matrix product,

$$c_{ki} = \sum_{j=1}^n a_{ij} b_{jk}$$

$$d_{ki} = \sum_{j=1}^n b_{jk} a_{ij} = \sum_{j=1}^n a_{ij} b_{jk} = c_{ki}$$

hence $\mathbf{C} = \mathbf{D}$ for any \mathbf{A} and \mathbf{B} , and the statement is proved.

EXERCISE B.8

(a) If \mathbf{A} is $m \times n$, \mathbf{A}^T is $n \times m$. Next we write the two products to be investigated:

$$\mathbf{A}^T \mathbf{A}, \quad \mathbf{A} \mathbf{A}^T$$

$n \times m \quad m \times n$ $m \times n \quad n \times m$

In both cases the column dimension of the premultiplicand is equal to the row dimension of the postmultiplicand. Therefore both products are possible.

(b) To verify symmetry we use three results. First, the symmetry test: transpose equals original; second, transposing twice gives back the original; and, finally, the transposed-product formula proved in Exercise B.7.

$$(\mathbf{A}^T \mathbf{A})^T = \mathbf{A}^T (\mathbf{A}^T)^T = \mathbf{A}^T \mathbf{A}$$

$$(\mathbf{A} \mathbf{A}^T)^T = (\mathbf{A}^T)^T \mathbf{A}^T = \mathbf{A} \mathbf{A}^T$$

Or, to do it more slowly, call $\mathbf{B} = \mathbf{A}^T$, $\mathbf{B}^T = \mathbf{A}$, $\mathbf{C} = \mathbf{AB}$, and let's go over the first one again:

$$\mathbf{C}^T = (\mathbf{AB})^T = \mathbf{B}^T \mathbf{A}^T = \mathbf{AA}^T = \mathbf{AB} = \mathbf{C}$$

Since $\mathbf{C} = \mathbf{C}^T$, $\mathbf{C} = \mathbf{AA}^T$ is symmetric. Same mechanics for the second one.

EXERCISE B.9

Let \mathbf{A} be $m \times n$. For $\mathbf{A}^2 = \mathbf{AA}$ to exist, the column dimension n of the premultiplicand \mathbf{A} must equal the row dimension m of the postmultiplicand \mathbf{A} . Hence $m = n$ and \mathbf{A} must be square.

EXERCISE B.10

Premultiply both sides of $\mathbf{A} = -\mathbf{A}^T$ by \mathbf{A} (which is always possible because \mathbf{A} is square):

$$\mathbf{A}^2 = \mathbf{AA} = -\mathbf{AA}^T$$

But from Exercise B.8 we know that \mathbf{AA}^T is symmetric. Since the negated of a symmetric matrix is symmetric, so is \mathbf{A}^2 .

C

Matrix Algebra: Determinants, Inverses, Eigenvalues

This Chapter discusses more specialized properties of matrices, such as determinants, eigenvalues and rank. These apply only to *square* matrices unless extension to rectangular matrices is explicitly stated.

§C.1 DETERMINANTS

The *determinant* of a *square* matrix $\mathbf{A} = [a_{ij}]$ is a number denoted by $|\mathbf{A}|$ or $\det(\mathbf{A})$, through which important properties such as singularity can be briefly characterized. This number is defined as the following function of the matrix elements:

$$|\mathbf{A}| = \pm \prod a_{1j_1} a_{2j_2} \dots a_{nj_n}, \quad (\text{C.1})$$

where the column indices j_1, j_2, \dots, j_n are taken from the set $1, 2, \dots, n$ with no repetitions allowed. The plus (minus) sign is taken if the permutation $(j_1 j_2 \dots j_n)$ is even (odd).

EXAMPLE C.1

For a 2×2 matrix,

$$\begin{vmatrix} a_{11} & a_{12} \\ a_{21} & a_{22} \end{vmatrix} = a_{11}a_{22} - a_{12}a_{21}. \quad (\text{C.2})$$

EXAMPLE C.2

For a 3×3 matrix,

$$\begin{vmatrix} a_{11} & a_{12} & a_{13} \\ a_{21} & a_{22} & a_{23} \\ a_{31} & a_{32} & a_{33} \end{vmatrix} = a_{11}a_{22}a_{33} + a_{12}a_{23}a_{31} + a_{13}a_{21}a_{32} - a_{13}a_{22}a_{31} - a_{12}a_{21}a_{33} - a_{11}a_{23}a_{32}. \quad (\text{C.3})$$

REMARK C.1

The concept of determinant is not applicable to rectangular matrices or to vectors. Thus the notation $|\mathbf{x}|$ for a vector \mathbf{x} can be reserved for its magnitude (as in Appendix A) without risk of confusion.

REMARK C.2

Inasmuch as the product (C.1) contains $n!$ terms, the calculation of $|\mathbf{A}|$ from the definition is impractical for general matrices whose order exceeds 3 or 4. For example, if $n = 10$, the product (C.1) contains $10! = 3,628,800$ terms each involving 9 multiplications, so over 30 million floating-point operations would be required to evaluate $|\mathbf{A}|$ according to that definition. A more practical method based on matrix decomposition is described in Remark C.3.

§C.1.1 Some Properties of Determinants

Some useful rules associated with the calculus of determinants are listed next.

- I. Rows and columns can be interchanged without affecting the value of a determinant. That is

$$|\mathbf{A}| = |\mathbf{A}^T|. \quad (\text{C.4})$$

- II. If two rows (or columns) are interchanged the sign of the determinant is changed. For example:

$$\begin{vmatrix} 3 & 4 \\ 1 & -2 \end{vmatrix} = - \begin{vmatrix} 1 & -2 \\ 3 & 4 \end{vmatrix}. \quad (\text{C.5})$$

- III. If a row (or column) is changed by adding to or subtracting from its elements the corresponding elements of any other row (or column) the determinant remains unaltered. For example:

$$\begin{vmatrix} 3 & 4 \\ 1 & -2 \end{vmatrix} = \begin{vmatrix} 3+1 & 4-2 \\ 1 & -2 \end{vmatrix} = \begin{vmatrix} 4 & 2 \\ 1 & -2 \end{vmatrix} = -10. \quad (\text{C.6})$$

- IV. If the elements in any row (or column) have a common factor α then the determinant equals the determinant of the corresponding matrix in which $\alpha = 1$, multiplied by α . For example:

$$\begin{vmatrix} 6 & 8 \\ 1 & -2 \end{vmatrix} = 2 \begin{vmatrix} 3 & 4 \\ 1 & -2 \end{vmatrix} = 2 \times (-10) = -20. \quad (\text{C.7})$$

- V. When at least one row (or column) of a matrix is a linear combination of the other rows (or columns) the determinant is zero. Conversely, if the determinant is zero, then at least one row and one column are linearly dependent on the other rows and columns, respectively. For example, consider

$$\begin{vmatrix} 3 & 2 & 1 \\ 1 & 2 & -1 \\ 2 & -1 & 3 \end{vmatrix}. \quad (\text{C.8})$$

This determinant is zero because the first column is a linear combination of the second and third columns:

$$\text{column 1} = \text{column 2} + \text{column 3} \quad (\text{C.9})$$

Similarly there is a linear dependence between the rows which is given by the relation

$$\text{row 1} = \frac{7}{8} \text{row 2} + \frac{4}{5} \text{row 3} \quad (\text{C.10})$$

- VI. The determinant of an upper triangular or lower triangular matrix is the product of the main diagonal entries. For example,

$$\begin{vmatrix} 3 & 2 & 1 \\ 0 & 2 & -1 \\ 0 & 0 & 4 \end{vmatrix} = 3 \times 2 \times 4 = 24. \quad (\text{C.11})$$

This rule is easily verified from the definition (C.1) because all terms vanish except $j_1 = 1, j_2 = 2, \dots, j_n = n$, which is the product of the main diagonal entries. Diagonal matrices are a particular case of this rule.

- VII. The determinant of the product of two square matrices is the product of the individual determinants:

$$|\mathbf{AB}| = |\mathbf{A}| |\mathbf{B}|. \quad (\text{C.12})$$

This rule can be generalized to any number of factors. One immediate application is to matrix powers: $|\mathbf{A}^2| = |\mathbf{A}||\mathbf{A}| = |\mathbf{A}|^2$, and more generally $|\mathbf{A}^n| = |\mathbf{A}|^n$ for integer n .

- VIII. The determinant of the transpose of a matrix is the same as that of the original matrix:

$$|\mathbf{A}^T| = |\mathbf{A}|. \quad (\text{C.13})$$

This rule can be directly verified from the definition of determinant.

REMARK C.3

Rules VI and VII are the key to the practical evaluation of determinants. Any square nonsingular matrix \mathbf{A} (where the qualifier “nonsingular” is explained in §C.3) can be decomposed as the product of two triangular factors

$$\mathbf{A} = \mathbf{L}\mathbf{U}, \quad (\text{C.14})$$

where \mathbf{L} is unit lower triangular and \mathbf{U} is upper triangular. This is called a LU triangularization, LU factorization or LU decomposition. It can be carried out in $O(n^3)$ floating point operations. According to rule VII:

$$|\mathbf{A}| = |\mathbf{L}| |\mathbf{U}|. \quad (\text{C.15})$$

But according to rule VI, $|\mathbf{L}| = 1$ and $|\mathbf{U}| = u_{11}u_{22} \dots u_{nn}$. The last operation requires only $O(n)$ operations. Thus the evaluation of $|\mathbf{A}|$ is dominated by the effort involved in computing the factorization (C.14). For $n = 10$, that effort is approximately $10^3 = 1000$ floating-point operations, compared to approximately 3×10^7 from the naive application of (C.1), as noted in Remark C.1. Thus the LU-based method is roughly 30,000 times faster for that modest matrix order, and the ratio increases exponentially for large n .

§C.1.2 Cramer's Rule

Cramer's rule provides a recipe for solving linear algebraic equations in terms of determinants. Let the simultaneous equations be as usual denoted as

$$\mathbf{A}\mathbf{x} = \mathbf{y}, \quad (\text{C.16})$$

where \mathbf{A} is a given $n \times n$ matrix, \mathbf{y} is a given $n \times 1$ vector, and \mathbf{x} is the $n \times 1$ vector of unknowns. The explicit form of (C.16) is Equation (A.1) of Appendix A, with $n = m$.

The explicit solution for the components x_1, x_2, \dots, x_n of \mathbf{x} in terms of determinants is

$$x_1 = \frac{\begin{vmatrix} y_1 & a_{12} & a_{13} & \dots & a_{1n} \\ y_2 & a_{22} & a_{23} & \dots & a_{2n} \\ \vdots & \vdots & \ddots & \ddots & \vdots \\ y_n & a_{n2} & a_{n3} & \dots & a_{nn} \end{vmatrix}}{|\mathbf{A}|}, \quad x_2 = \frac{\begin{vmatrix} a_{11} & y_1 & a_{13} & \dots & a_{1n} \\ a_{21} & y_2 & a_{23} & \dots & a_{2n} \\ \vdots & \vdots & \ddots & \ddots & \vdots \\ a_{n1} & y_n & a_{n3} & \dots & a_{nn} \end{vmatrix}}{|\mathbf{A}|}, \dots \quad (\text{C.17})$$

The rule can be remembered as follows: in the numerator of the quotient for x_j , replace the j^{th} column of \mathbf{A} by the right-hand side \mathbf{y} .

This method of solving simultaneous equations is known as *Cramer's rule*. Because the explicit computation of determinants is impractical for $n > 3$ as explained in Remark C.3, this rule has practical value only for $n = 2$ and $n = 3$ (it is marginal for $n = 4$). But such small-order systems arise often in finite element calculations at the *Gauss point level*; consequently implementors should be aware of this rule for such applications.

EXAMPLE C.3

Solve the 3×3 linear system

$$\begin{bmatrix} 5 & 2 & 1 \\ 3 & 2 & 0 \\ 1 & 0 & 2 \end{bmatrix} \begin{bmatrix} x_1 \\ x_2 \\ x_3 \end{bmatrix} = \begin{bmatrix} 8 \\ 5 \\ 3 \end{bmatrix}, \quad (\text{C.18})$$

by Cramer's rule:

$$x_1 = \frac{\begin{vmatrix} 8 & 2 & 1 \\ 5 & 2 & 0 \\ 3 & 0 & 2 \end{vmatrix}}{\begin{vmatrix} 5 & 2 & 1 \\ 3 & 2 & 0 \\ 1 & 0 & 2 \end{vmatrix}} = \frac{6}{6} = 1, \quad x_2 = \frac{\begin{vmatrix} 5 & 8 & 1 \\ 3 & 5 & 0 \\ 1 & 3 & 2 \end{vmatrix}}{\begin{vmatrix} 5 & 2 & 1 \\ 3 & 2 & 0 \\ 1 & 0 & 2 \end{vmatrix}} = \frac{6}{6} = 1, \quad x_3 = \frac{\begin{vmatrix} 5 & 2 & 8 \\ 3 & 2 & 5 \\ 1 & 0 & 3 \end{vmatrix}}{\begin{vmatrix} 5 & 2 & 1 \\ 3 & 2 & 0 \\ 1 & 0 & 2 \end{vmatrix}} = \frac{6}{6} = 1. \quad (\text{C.19})$$

EXAMPLE C.4

Solve the 2×2 linear algebraic system

$$\begin{bmatrix} 2+\beta & -\beta \\ -\beta & 1+\beta \end{bmatrix} \begin{bmatrix} x_1 \\ x_2 \end{bmatrix} = \begin{bmatrix} 5 \\ 0 \end{bmatrix} \quad (\text{C.20})$$

by Cramer's rule:

$$x_1 = \frac{\begin{vmatrix} 5 & -\beta \\ 0 & 1+\beta \end{vmatrix}}{\begin{vmatrix} 2+\beta & -\beta \\ -\beta & 1+\beta \end{vmatrix}} = \frac{5+5\beta}{2+3\beta}, \quad x_2 = \frac{\begin{vmatrix} 2+\beta & 5 \\ -\beta & 0 \end{vmatrix}}{\begin{vmatrix} 2+\beta & -\beta \\ -\beta & 1+\beta \end{vmatrix}} = \frac{5\beta}{2+3\beta}. \quad (\text{C.21})$$

§C.1.3 Homogeneous Systems

One immediate consequence of Cramer's rule is what happens if

$$y_1 = y_2 = \dots = y_n = 0. \quad (\text{C.22})$$

The linear equation systems with a null right hand side

$$\mathbf{Ax} = \mathbf{0}, \quad (\text{C.23})$$

is called a *homogeneous system*. From the rule (C.17) we see that if $|\mathbf{A}|$ is nonzero, all solution components are zero, and consequently the only possible solution is the trivial one $\mathbf{x} = \mathbf{0}$. The case in which $|\mathbf{A}|$ vanishes is discussed in the next section.

§C.2 SINGULAR MATRICES, RANK

If the determinant $|\mathbf{A}|$ of a $n \times n$ square matrix $\mathbf{A} \equiv \mathbf{A}_n$ is zero, then the matrix is said to be *singular*. This means that at least one row and one column are linearly dependent on the others. If this row and column are removed, we are left with another matrix, say \mathbf{A}_{n-1} , to which we can apply the same criterion. If the determinant $|\mathbf{A}_{n-1}|$ is zero, we can remove another row and column from it to get \mathbf{A}_{n-2} , and so on. Suppose that we eventually arrive at an $r \times r$ matrix \mathbf{A}_r whose determinant is nonzero. Then matrix \mathbf{A} is said to have *rank* r , and we write $\text{rank}(\mathbf{A}) = r$.

If the determinant of \mathbf{A} is nonzero, then \mathbf{A} is said to be *nonsingular*. The rank of a nonsingular $n \times n$ matrix is equal to n .

Obviously the rank of \mathbf{A}^T is the same as that of \mathbf{A} since it is only necessary to transpose “row” and “column” in the definition.

The notion of rank can be extended to rectangular matrices as outlined in section §C.2.4 below. That extension, however, is not important for the material covered here.

EXAMPLE C.5

The 3×3 matrix

$$\mathbf{A} = \begin{bmatrix} 3 & 2 & 2 \\ 1 & 2 & -1 \\ 2 & -1 & 3 \end{bmatrix}, \quad (\text{C.24})$$

has rank $r = 3$ because $|\mathbf{A}| = -3 \neq 0$.

EXAMPLE C.6

The matrix

$$\mathbf{A} = \begin{bmatrix} 3 & 2 & 1 \\ 1 & 2 & -1 \\ 2 & -1 & 3 \end{bmatrix}, \quad (\text{C.25})$$

already used as an example in §C.1.1 is singular because its first row and column may be expressed as linear combinations of the others through the relations (C.9) and (C.10). Removing the first row and column we are left with a 2×2 matrix whose determinant is $2 \times 3 - (-1) \times (-1) = 5 \neq 0$. Consequently (C.25) has rank $r = 2$.

§C.2.1 Rank Deficiency

If the square matrix \mathbf{A} is supposed to be of rank r but in fact has a smaller rank $\bar{r} < r$, the matrix is said to be *rank deficient*. The number $r - \bar{r} > 0$ is called the *rank deficiency*.

EXAMPLE C.7

Suppose that the *unconstrained* master stiffness matrix \mathbf{K} of a finite element has order n , and that the element possesses b independent rigid body modes. Then the expected rank of \mathbf{K} is $r = n - b$. If the actual rank is less than r , the finite element model is said to be rank-deficient. This is an undesirable property.

EXAMPLE C.8

An illustration of the foregoing rule, consider the two-node, 4-DOF plane beam element stiffness derived in Chapter 13:

$$\mathbf{K} = \frac{EI}{L^3} \begin{bmatrix} 12 & 6L & -12 & 6L \\ & 4L^2 & -6L & 2L^2 \\ & & 12 & -6L \\ \text{symm} & & & 4L^2 \end{bmatrix} \quad (\text{C.26})$$

where EI and L are nonzero scalars. It can be verified that this 4×4 matrix has rank 2. The number of rigid body modes is 2, and the expected rank is $r = 4 - 2 = 2$. Consequently this model is rank sufficient.

§C.2.2 Rank of Matrix Sums and Products

In finite element analysis matrices are often built through sum and product combinations of simpler matrices. Two important rules apply to “rank propagation” through those combinations.

The rank of the product of two square matrices \mathbf{A} and \mathbf{B} cannot exceed the smallest rank of the multiplicand matrices. That is, if the rank of \mathbf{A} is r_a and the rank of \mathbf{B} is r_b ,

$$\text{Rank}(\mathbf{AB}) \leq \min(r_a, r_b). \quad (\text{C.27})$$

Regarding sums: the rank of a matrix sum cannot exceed the sum of ranks of the summand matrices. That is, if the rank of \mathbf{A} is r_a and the rank of \mathbf{B} is r_b ,

$$\text{Rank}(\mathbf{A} + \mathbf{B}) \leq r_a + r_b. \quad (\text{C.28})$$

§C.2.3 Singular Systems: Particular and Homogeneous Solutions

Having introduced the notion of rank we can now discuss what happens to the linear system (C.16) when the determinant of \mathbf{A} vanishes, meaning that its rank is less than n . If so, the linear system (C.16) has either no solution or an infinite number of solution. Cramer's rule is of limited or no help in this situation.

To discuss this case further we note that if $|\mathbf{A}| = 0$ and the rank of \mathbf{A} is $r = n - d$, where $d \geq 1$ is the *rank deficiency*, then there exist d nonzero independent vectors $\mathbf{z}_i, i = 1, \dots, d$ such that

$$\mathbf{A}\mathbf{z}_i = \mathbf{0}. \quad (\text{C.29})$$

These d vectors, suitably orthonormalized, are called *null eigenvectors* of \mathbf{A} , and form a basis for its *null space*.

Let \mathbf{Z} denote the $n \times d$ matrix obtained by collecting the \mathbf{z}_i as columns. If \mathbf{y} in (C.13) is in the *range* of \mathbf{A} , that is, there exists a nonzero \mathbf{x}_p such that $\mathbf{y} = \mathbf{A}\mathbf{x}_p$, its general solution is

$$\mathbf{x} = \mathbf{x}_p + \mathbf{x}_h = \mathbf{x}_p + \mathbf{Z}\mathbf{w}, \quad (\text{C.30})$$

where \mathbf{w} is an arbitrary $d \times 1$ weighting vector. This statement can be easily verified by substituting this solution into $\mathbf{A}\mathbf{x} = \mathbf{y}$ and noting that $\mathbf{A}\mathbf{Z}$ vanishes.

The components \mathbf{x}_p and \mathbf{x}_h are called the *particular* and *homogeneous* part, respectively, of the solution \mathbf{x} . If $\mathbf{y} = \mathbf{0}$ only the homogeneous part remains.

If \mathbf{y} is not in the range of \mathbf{A} , system (C.13) does not generally have a solution in the conventional sense, although least-square solutions can usually be constructed. The reader is referred to the many textbooks in linear algebra for further details.

§C.2.4 Rank of Rectangular Matrices

The notion of rank can be extended to rectangular matrices, real or complex, as follows. Let \mathbf{A} be $m \times n$. Its *column range space* $\mathcal{R}(\mathbf{A})$ is the subspace spanned by $\mathbf{A}\mathbf{x}$ where \mathbf{x} is the set of all complex n -vectors. Mathematically: $\mathcal{R}(\mathbf{A}) = \{\mathbf{A}\mathbf{x} : \mathbf{x} \in C^n\}$. The rank r of \mathbf{A} is the dimension of $\mathcal{R}(\mathbf{A})$.

The *null space* $\mathcal{N}(\mathbf{A})$ of \mathbf{A} is the set of n -vectors \mathbf{z} such that $\mathbf{A}\mathbf{z} = \mathbf{0}$. The dimension of $\mathcal{N}(\mathbf{A})$ is $n - r$.

Using these definitions, the product and sum rules (C.27) and (C.28) generalize to the case of rectangular (but conforming) \mathbf{A} and \mathbf{B} . So does the treatment of linear equation systems $\mathbf{A}\mathbf{x} = \mathbf{y}$ in which \mathbf{A} is rectangular; such systems often arise in the fitting of observation and measurement data.

In finite element methods, rectangular matrices appear in change of basis through congruential transformations, and in the treatment of multifreedom constraints.

§C.3 MATRIX INVERSION

The *inverse* of a square nonsingular matrix \mathbf{A} is represented by the symbol \mathbf{A}^{-1} and is defined by the relation

$$\mathbf{A}\mathbf{A}^{-1} = \mathbf{I}. \quad (\text{C.31})$$

The most important application of the concept of inverse is the solution of linear systems. Suppose that, in the usual notation, we have

$$\mathbf{A}\mathbf{x} = \mathbf{y} \quad (\text{C.32})$$

Premultiplying both sides by \mathbf{A}^{-1} we get the inverse relationship

$$\mathbf{x} = \mathbf{A}^{-1}\mathbf{y} \quad (\text{C.33})$$

More generally, consider the matrix equation for multiple (m) right-hand sides:

$$\underset{n \times n}{\mathbf{A}} \underset{n \times m}{\mathbf{X}} = \underset{n \times m}{\mathbf{Y}}, \quad (\text{C.34})$$

which reduces to (C.32) for $m = 1$. The inverse relation that gives \mathbf{X} as function of \mathbf{Y} is

$$\mathbf{X} = \mathbf{A}^{-1}\mathbf{Y}. \quad (\text{C.35})$$

In particular, the solution of

$$\mathbf{AX} = \mathbf{I}, \quad (\text{C.36})$$

is $\mathbf{X} = \mathbf{A}^{-1}$. Practical methods for computing inverses are based on directly solving this equation; see Remark C.4.

§C.3.1 Explicit Computation of Inverses

The explicit calculation of matrix inverses is seldom needed in large matrix computations. But occasionally the need arises for the explicit inverse of small matrices that appear in element computations. For example, the inversion of Jacobian matrices at Gauss points, or of constitutive matrices.

A general formula for elements of the inverse can be obtained by specializing Cramer's rule to (C.36). Let $\mathbf{B} = [b_{ij}] = \mathbf{A}^{-1}$. Then

$$b_{ij} = \frac{A_{ji}}{|\mathbf{A}|}, \quad (\text{C.37})$$

in which A_{ji} denotes the so-called *adjoint* of entry a_{ij} of \mathbf{A} . The adjoint A_{ji} is defined as the determinant of the submatrix of order $(n-1) \times (n-1)$ obtained by deleting the j^{th} row and i^{th} column of \mathbf{A} , multiplied by $(-1)^{i+j}$.

This direct inversion procedure is useful only for small matrix orders: 2 or 3. In the examples below the inversion formulas for second and third order matrices are listed.

EXAMPLE C.9

For order $n = 2$:

$$\mathbf{A} = \begin{bmatrix} a_{11} & a_{12} \\ a_{21} & a_{22} \end{bmatrix}, \quad \mathbf{A}^{-1} = \frac{1}{|\mathbf{A}|} \begin{bmatrix} a_{22} & -a_{12} \\ -a_{21} & a_{11} \end{bmatrix}, \quad (\text{C.38})$$

in which $|\mathbf{A}|$ is given by (C.2).

EXAMPLE C.10

For order $n = 3$:

$$\mathbf{A} = \begin{bmatrix} a_{11} & a_{12} & a_{13} \\ a_{21} & a_{22} & a_{23} \\ a_{31} & a_{32} & a_{33} \end{bmatrix}, \quad \mathbf{A}^{-1} = \frac{1}{|\mathbf{A}|} \begin{bmatrix} b_{11} & b_{12} & b_{13} \\ b_{21} & b_{22} & b_{23} \\ b_{31} & b_{32} & b_{33} \end{bmatrix}, \quad (\text{C.39})$$

where

$$\begin{aligned} b_{11} &= \begin{vmatrix} a_{22} & a_{23} \\ a_{32} & a_{33} \end{vmatrix}, & b_{21} &= -\begin{vmatrix} a_{12} & a_{13} \\ a_{32} & a_{33} \end{vmatrix}, & b_{31} &= \begin{vmatrix} a_{12} & a_{13} \\ a_{22} & a_{23} \end{vmatrix}, \\ b_{12} &= -\begin{vmatrix} a_{21} & a_{23} \\ a_{31} & a_{33} \end{vmatrix}, & b_{22} &= \begin{vmatrix} a_{11} & a_{13} \\ a_{31} & a_{33} \end{vmatrix}, & b_{32} &= -\begin{vmatrix} a_{11} & a_{13} \\ a_{21} & a_{23} \end{vmatrix}, \\ b_{13} &= \begin{vmatrix} a_{21} & a_{22} \\ a_{31} & a_{32} \end{vmatrix}, & b_{23} &= -\begin{vmatrix} a_{11} & a_{12} \\ a_{31} & a_{32} \end{vmatrix}, & b_{33} &= \begin{vmatrix} a_{11} & a_{12} \\ a_{21} & a_{22} \end{vmatrix}, \end{aligned} \quad (\text{C.40})$$

in which $|\mathbf{A}|$ is given by (C.3).

EXAMPLE C.11

$$\mathbf{A} = \begin{bmatrix} 2 & 4 & 2 \\ 3 & 1 & 1 \\ 1 & 0 & 1 \end{bmatrix}, \quad \mathbf{A}^{-1} = -\frac{1}{8} \begin{bmatrix} 1 & -4 & 2 \\ -2 & 0 & 4 \\ -1 & 4 & -10 \end{bmatrix}. \quad (\text{C.41})$$

If the order exceeds 3, the general inversion formula based on Cramer's rule becomes rapidly useless as it displays combinatorial complexity. For numerical work it is preferable to solve the system (C.38) after \mathbf{A} is factored. Those techniques are described in detail in linear algebra books; see also Remark C.4.

§C.3.2 Some Properties of the Inverse

I. The inverse of the transpose is equal to the transpose of the inverse:

$$(\mathbf{A}^T)^{-1} = (\mathbf{A}^{-1})^T, \quad (\text{C.42})$$

because

$$(\mathbf{A}\mathbf{A}^{-1}) = (\mathbf{A}\mathbf{A}^{-1})^T = (\mathbf{A}^{-1})^T \mathbf{A}^T = \mathbf{I}. \quad (\text{C.43})$$

II. The inverse of a symmetric matrix is also symmetric. Because of the previous rule, $(\mathbf{A}^T)^{-1} = \mathbf{A}^{-1} = (\mathbf{A}^{-1})^T$, hence \mathbf{A}^{-1} is also symmetric.

III. The inverse of a matrix product is the reverse product of the inverses of the factors:

$$(\mathbf{AB})^{-1} = \mathbf{B}^{-1}\mathbf{A}^{-1}. \quad (\text{C.44})$$

This is easily verified by substituting both sides of (C.39) into (C.31). This property generalizes to an arbitrary number of factors.

IV. For a diagonal matrix \mathbf{D} in which all diagonal entries are nonzero, \mathbf{D}^{-1} is again a diagonal matrix with entries $1/d_{ii}$. The verification is straightforward.

V. If \mathbf{S} is a block diagonal matrix:

$$\mathbf{S} = \begin{bmatrix} \mathbf{S}_{11} & \mathbf{0} & \mathbf{0} & \cdots & \mathbf{0} \\ \mathbf{0} & \mathbf{S}_{22} & \mathbf{0} & \cdots & \mathbf{0} \\ \mathbf{0} & \mathbf{0} & \mathbf{S}_{33} & \cdots & \mathbf{0} \\ \vdots & \vdots & \vdots & \ddots & \vdots \\ \mathbf{0} & \mathbf{0} & \mathbf{0} & \cdots & \mathbf{S}_{nn} \end{bmatrix} = \text{diag} [\mathbf{S}_{ii}], \quad (\text{C.45})$$

then the inverse matrix is also block diagonal and is given by

$$\mathbf{S}^{-1} = \begin{bmatrix} \mathbf{S}_{11}^{-1} & \mathbf{0} & \mathbf{0} & \cdots & \mathbf{0} \\ \mathbf{0} & \mathbf{S}_{22}^{-1} & \mathbf{0} & \cdots & \mathbf{0} \\ \mathbf{0} & \mathbf{0} & \mathbf{S}_{33}^{-1} & \cdots & \mathbf{0} \\ \vdots & \vdots & \vdots & \ddots & \vdots \\ \mathbf{0} & \mathbf{0} & \mathbf{0} & \cdots & \mathbf{S}_{nn}^{-1} \end{bmatrix} = \text{diag} [\mathbf{S}_{ii}^{-1}]. \quad (\text{C.46})$$

VI. The inverse of an upper triangular matrix is also an upper triangular matrix. The inverse of a lower triangular matrix is also a lower triangular matrix. Both inverses can be computed in $O(n^2)$ floating-point operations.

REMARK C.4

The practical numerical calculation of inverses is based on triangular factorization. Given a nonsingular $n \times n$ matrix \mathbf{A} , calculate its LU factorization $\mathbf{A} = \mathbf{L}\mathbf{U}$, which can be obtained in $O(n^3)$ operations. Then solve the linear triangular systems:

$$\mathbf{U}\mathbf{Y} = \mathbf{I}, \quad \mathbf{L}\mathbf{X} = \mathbf{Y}, \quad (\text{C.47})$$

and the computed inverse \mathbf{A}^{-1} appears in \mathbf{X} . One can overwrite \mathbf{I} with \mathbf{Y} and \mathbf{Y} with \mathbf{X} . The whole process can be completed in $O(n^3)$ floating-point operations. For symmetric matrices the alternative decomposition $\mathbf{A} = \mathbf{L}\mathbf{D}\mathbf{L}^T$, where \mathbf{L} is unit lower triangular and \mathbf{D} is diagonal, is generally preferred to save computing time and storage.

§C.4 EIGENVALUES AND EIGENVECTORS

Consider the special form of the linear system (C.13) in which the right-hand side vector \mathbf{y} is a multiple of the solution vector \mathbf{x} :

$$\mathbf{A}\mathbf{x} = \lambda\mathbf{x}, \quad (\text{C.48})$$

or, written in full,

$$\begin{array}{cccccc} a_{11}x_1 & + & a_{12}x_2 & + & \cdots & + & a_{1n}x_n & = & \lambda x_1 \\ a_{21}x_1 & + & a_{22}x_2 & + & \cdots & + & a_{2n}x_n & = & \lambda x_2 \\ \cdots & & \cdots & & \cdots & & \cdots & & \cdots \\ a_{n1}x_1 & + & a_{n2}x_2 & + & \cdots & + & a_{nn}x_n & = & \lambda x_n \end{array} \quad (\text{C.49})$$

This is called the standard (or classical) *algebraic eigenproblem*. System (C.48) can be rearranged into the homogeneous form

$$(\mathbf{A} - \lambda\mathbf{I})\mathbf{x} = \mathbf{0}. \quad (\text{C.50})$$

A nontrivial solution of this equation is possible if and only if the coefficient matrix $\mathbf{A} - \lambda\mathbf{I}$ is singular. Such a condition can be expressed as the vanishing of the determinant

$$|\mathbf{A} - \lambda\mathbf{I}| = \begin{vmatrix} a_{11} - \lambda & a_{12} & \cdots & a_{1n} \\ a_{21} & a_{22} - \lambda & \cdots & a_{2n} \\ \vdots & \vdots & \ddots & \vdots \\ a_{n1} & a_{n2} & \cdots & a_{nn} - \lambda \end{vmatrix} = 0. \quad (\text{C.51})$$

When this determinant is expanded, we obtain an algebraic polynomial equation in λ of degree n :

$$P(\lambda) = \lambda^n + \alpha_1\lambda^{n-1} + \cdots + \alpha_n = 0. \quad (\text{C.52})$$

This is known as the *characteristic equation* of the matrix \mathbf{A} . The left-hand side is called the *characteristic polynomial*. We know that a polynomial of degree n has n (generally complex) roots $\lambda_1, \lambda_2, \dots, \lambda_n$. These n numbers are called the *eigenvalues*, *eigenroots* or *characteristic values* of matrix \mathbf{A} .

With each eigenvalue λ_i there is an associated vector \mathbf{x}_i that satisfies

$$\mathbf{A}\mathbf{x}_i = \lambda_i\mathbf{x}_i. \quad (\text{C.53})$$

This \mathbf{x}_i is called an *eigenvector* or *characteristic vector*. An eigenvector is unique only up to a scale factor since if \mathbf{x}_i is an eigenvector, so is $\beta\mathbf{x}_i$ where β is an arbitrary nonzero number. Eigenvectors are often *normalized* so that their Euclidean length is 1, or their largest component is unity.

§C.4.1 Real Symmetric Matrices

Real symmetric matrices are of special importance in the finite element method. In linear algebra books dealing with the algebraic eigenproblem it is shown that:

- (a) The n eigenvalues of a real symmetric matrix of order n are real.
- (b) The eigenvectors corresponding to distinct eigenvalues are orthogonal. The eigenvectors corresponding to multiple roots may be orthogonalized with respect to each other.
- (c) The n eigenvectors form a complete orthonormal basis for the Euclidean space E_n .

§C.4.2 Positivity

Let \mathbf{A} be an $n \times n$ square *symmetric* matrix. \mathbf{A} is said to be *positive definite* (p.d.) if

$$\mathbf{x}^T \mathbf{A} \mathbf{x} > 0, \quad \mathbf{x} \neq 0 \quad (\text{C.54})$$

A positive definite matrix has rank n . This property can be checked by computing the n eigenvalues λ_i of $\mathbf{A}\mathbf{z} = \lambda\mathbf{z}$. If all $\lambda_i > 0$, \mathbf{A} is p.d.

\mathbf{A} is said to be *nonnegative*¹ if zero equality is allowed in (C.54):

$$\mathbf{x}^T \mathbf{A} \mathbf{x} \leq 0, \quad \mathbf{x} \neq 0 \quad (\text{C.55})$$

A p.d. matrix is also nonnegative but the converse is not necessarily true. This property can be checked by computing the n eigenvalues λ_i of $\mathbf{A}\mathbf{z} = \lambda\mathbf{z}$. If r eigenvalues $\lambda_i > 0$ and $n - r$ eigenvalues are zero, \mathbf{A} is nonnegative with rank r .

A symmetric square matrix \mathbf{A} that has at least one negative eigenvalue is called *indefinite*.

¹ A property called *positive semi-definite* by some authors.

§C.4.3 Normal and Orthogonal Matrices

Let \mathbf{A} be an $n \times n$ real square matrix. This matrix is called *normal* if

$$\mathbf{A}^T \mathbf{A} = \mathbf{A} \mathbf{A}^T \quad (\text{C.56})$$

A normal matrix is called *orthogonal* if

$$\mathbf{A}^T \mathbf{A} = \mathbf{A} \mathbf{A}^T = \mathbf{I} \quad \text{or} \quad \mathbf{A}^T = \mathbf{A}^{-1} \quad (\text{C.57})$$

All eigenvalues of an orthogonal matrix have modulus one, and the matrix has rank n .

The generalization of the orthogonality property to complex matrices, for which transposition is replaced by conjugation, leads to *unitary* matrices. These are not required, however, for the material covered in the text.

§C.5 THE SHERMAN-MORRISON AND RELATED FORMULAS

The Sherman-Morrison formula gives the inverse of a matrix modified by a rank-one matrix. The Woodbury formula extends the Sherman-Morrison formula to a modification of arbitrary rank. In structural analysis these formulas are of interest for problems of *structural modifications*, in which a finite-element (or, in general, a discrete model) is changed by an amount expressible as a low-rank correction to the original model.

§C.5.1 The Sherman-Morrison Formula

Let \mathbf{A} be a square $n \times n$ invertible matrix, whereas \mathbf{u} and \mathbf{v} are two n -vectors and β an arbitrary scalar. Assume that $\sigma = 1 + \beta \mathbf{v}^T \mathbf{A}^{-1} \mathbf{u} \neq 0$. Then

$$(\mathbf{A} + \beta \mathbf{u} \mathbf{v}^T)^{-1} = \mathbf{A}^{-1} - \frac{\beta}{\sigma} \mathbf{A}^{-1} \mathbf{u} \mathbf{v}^T \mathbf{A}^{-1}. \quad (\text{C.58})$$

This is called the Sherman-Morrison formula² when $\beta = 1$. Since any rank-one correction to \mathbf{A} can be written as $\beta \mathbf{u} \mathbf{v}^T$, (C.58) gives the rank-one change to its inverse. The proof is by direct multiplication as in Exercise C.5.

For practical computation of the change one solves the linear systems $\mathbf{A} \mathbf{a} = \mathbf{u}$ and $\mathbf{A} \mathbf{b} = \mathbf{v}$ for \mathbf{a} and \mathbf{b} , using the known \mathbf{A}^{-1} . Compute $\sigma = 1 + \beta \mathbf{v}^T \mathbf{a}$. If $\sigma \neq 0$, the change to \mathbf{A}^{-1} is the dyadic $-(\beta/\sigma) \mathbf{a} \mathbf{b}^T$.

§C.5.2 The Woodbury Formula

Let again \mathbf{A} be a square $n \times n$ invertible matrix, whereas \mathbf{U} and \mathbf{V} are two $n \times k$ matrices with $k \leq n$ and β an arbitrary scalar. Assume that the $k \times k$ matrix $\Sigma = \mathbf{I}_k + \beta \mathbf{V}^T \mathbf{A}^{-1} \mathbf{U}$, in which \mathbf{I}_k denotes the $k \times k$ identity matrix, is invertible. Then

$$(\mathbf{A} + \beta \mathbf{U} \mathbf{V}^T)^{-1} = \mathbf{A}^{-1} - \beta \mathbf{A}^{-1} \mathbf{U} \Sigma^{-1} \mathbf{V}^T \mathbf{A}^{-1}. \quad (\text{C.59})$$

This is called the Woodbury formula.³ It reduces to (C.58) if $k = 1$, in which case $\Sigma \equiv \sigma$ is a scalar. The proof is by direct multiplication.

² J. Sherman and W. J. Morrison, Adjustment of an inverse matrix corresponding to changes in the elements of a given column or a given row of the original matrix, *Ann. Math. Statist.*, **20**, (1949), 621. For a history of this remarkable formula and its extensions, which are quite important in many applications such as statistics, see the paper by Henderson and Searle in *SIAM Review*, **23**, 53–60.

³ M.A. Woodbury, Inverting modified matrices, *Statist. Res. Group, Mem. Rep.* No. 42, Princeton Univ., Princeton, N.J., 1950

§C.5.3 Formulas for Modified Determinants

Let $\tilde{\mathbf{A}}$ denote the adjoint of \mathbf{A} . Taking the determinants from both sides of $\mathbf{A} + \beta \mathbf{u}\mathbf{v}^T$ one obtains

$$|\mathbf{A} + \beta \mathbf{u}\mathbf{v}^T| = |\mathbf{A}| + \beta \mathbf{v}^T \tilde{\mathbf{A}} \mathbf{u}. \quad (\text{C.60})$$

If \mathbf{A} is invertible, replacing $\tilde{\mathbf{A}} = |\mathbf{A}| \mathbf{A}^{-1}$ this becomes

$$|\mathbf{A} + \beta \mathbf{u}\mathbf{v}^T| = |\mathbf{A}| (1 + \beta \mathbf{v}^T \mathbf{A}^{-1} \mathbf{u}). \quad (\text{C.61})$$

Similarly, one can show that if \mathbf{A} is invertible, and \mathbf{U} and \mathbf{V} are $n \times k$ matrices,

$$|\mathbf{A} + \beta \mathbf{U}\mathbf{V}^T| = |\mathbf{A}| |\mathbf{I}_k + \beta \mathbf{V}^T \mathbf{A}^{-1} \mathbf{U}|. \quad (\text{C.62})$$

Exercises for Appendix C: Determinants, Inverses, Eigenvalues

EXERCISE C.1

If \mathbf{A} is a square matrix of order n and c a scalar, show that $\det(c\mathbf{A}) = c^n \det \mathbf{A}$.

EXERCISE C.2

Let \mathbf{u} and \mathbf{v} denote real n -vectors normalized to unit length, so that $\mathbf{u}^T \mathbf{u} = 1$ and $\mathbf{v}^T \mathbf{v} = 1$, and let \mathbf{I} denote the $n \times n$ identity matrix. Show that

$$\det(\mathbf{I} - \mathbf{u}\mathbf{v}^T) = 1 - \mathbf{v}^T \mathbf{u} \quad (\text{EC.1})$$

EXERCISE C.3

Let \mathbf{u} denote a real n -vector normalized to unit length, so that $\mathbf{u}^T \mathbf{u} = 1$ and \mathbf{I} denote the $n \times n$ identity matrix. Show that

$$\mathbf{H} = \mathbf{I} - 2\mathbf{u}\mathbf{u}^T \quad (\text{EC.2})$$

is orthogonal: $\mathbf{H}^T \mathbf{H} = \mathbf{I}$, and idempotent: $\mathbf{H}^2 = \mathbf{H}$. This matrix is called a *elementary Hermitian*, a *Householder matrix*, or a *reflector*. It is a fundamental ingredient of many linear algebra algorithms; for example the QR algorithm for finding eigenvalues.

EXERCISE C.4

The *trace* of a $n \times n$ square matrix \mathbf{A} , denoted $\text{trace}(\mathbf{A})$ is the sum $\sum_{i=1}^n a_{ii}$ of its diagonal coefficients. Show that if the entries of \mathbf{A} are real,

$$\text{trace}(\mathbf{A}^T \mathbf{A}) = \sum_{i=1}^n \sum_{j=1}^n a_{ij}^2 \quad (\text{EC.3})$$

EXERCISE C.5

Prove the Sherman-Morrison formula (C.59) by direct matrix multiplication.

EXERCISE C.6

Prove the Sherman-Morrison formula (C.59) for $\beta = 1$ by considering the following block bordered system

$$\begin{bmatrix} \mathbf{A} & \mathbf{U} \\ \mathbf{V}^T & \mathbf{I}_k \end{bmatrix} \begin{bmatrix} \mathbf{B} \\ \mathbf{C} \end{bmatrix} = \begin{bmatrix} \mathbf{I}_n \\ \mathbf{0} \end{bmatrix} \quad (\text{EC.4})$$

in which \mathbf{I}_k and \mathbf{I}_n denote the identity matrices of orders k and n , respectively. Solve (C.62) two ways: eliminating first \mathbf{B} and then \mathbf{C} , and eliminating first \mathbf{C} and then \mathbf{B} . Equate the results for \mathbf{B} .

EXERCISE C.7

Show that the eigenvalues of a real symmetric square matrix are real, and that the eigenvectors are real vectors.

EXERCISE C.8

Let the n real eigenvalues λ_i of a real $n \times n$ symmetric matrix \mathbf{A} be classified into two subsets: r eigenvalues are nonzero whereas $n - r$ are zero. Show that \mathbf{A} has rank r .

EXERCISE C.9

Show that if \mathbf{A} is p.d., $\mathbf{A}\mathbf{x} = \mathbf{0}$ implies that $\mathbf{x} = \mathbf{0}$.

EXERCISE C.10

Show that for any real $m \times n$ matrix \mathbf{A} , $\mathbf{A}^T \mathbf{A}$ exists and is nonnegative.

EXERCISE C.11

Show that a triangular matrix is normal if and only if it is diagonal.

EXERCISE C.12

Let \mathbf{A} be a real orthogonal matrix. Show that all of its eigenvalues λ_i , which are generally complex, have unit modulus.

EXERCISE C.13

Let \mathbf{A} and \mathbf{T} be real $n \times n$ matrices, with \mathbf{T} nonsingular. Show that $\mathbf{T}^{-1}\mathbf{A}\mathbf{T}$ and \mathbf{A} have the same eigenvalues. (This is called a similarity transformation in linear algebra).

EXERCISE C.14

(Tough) Let \mathbf{A} be $m \times n$ and \mathbf{B} be $n \times m$. Show that the nonzero eigenvalues of \mathbf{AB} are the same as those of \mathbf{BA} (Kahan).

EXERCISE C.15

Let \mathbf{A} be real skew-symmetric, that is, $\mathbf{A} = -\mathbf{A}^T$. Show that all eigenvalues of \mathbf{A} are purely imaginary or zero.

EXERCISE C.16

Let \mathbf{A} be real skew-symmetric, that is, $\mathbf{A} = -\mathbf{A}^T$. Show that $\mathbf{U} = (\mathbf{I} + \mathbf{A})^{-1}(\mathbf{I} - \mathbf{A})$, called a Cayley transformation, is orthogonal.

EXERCISE C.17

Let \mathbf{P} be a real square matrix that satisfies

$$\mathbf{P}^2 = \mathbf{P}. \quad (\text{EC.5})$$

Such matrices are called *idempotent*, and also *orthogonal projectors*. Show that all eigenvalues of \mathbf{P} are either zero or one.

EXERCISE C.18

The necessary and sufficient condition for two square matrices to commute is that they have the same eigenvectors.

EXERCISE C.19

A matrix whose elements are equal on any line parallel to the main diagonal is called a Toeplitz matrix. (They arise in finite difference or finite element discretizations of regular one-dimensional grids.) Show that if \mathbf{T}_1 and \mathbf{T}_2 are any two Toeplitz matrices, they commute: $\mathbf{T}_1\mathbf{T}_2 = \mathbf{T}_2\mathbf{T}_1$. Hint: do a Fourier transform to show that the eigenvectors of any Toeplitz matrix are of the form $\{e^{i\omega n h}\}$; then apply the previous Exercise.

D

Matrix Calculus

In this Appendix we collect some useful formulas of matrix calculus that often appear in finite element derivations.

§D.1 THE DERIVATIVES OF VECTOR FUNCTIONS

Let \mathbf{x} and \mathbf{y} be vectors of orders n and m respectively:

$$\mathbf{x} = \begin{bmatrix} x_1 \\ x_2 \\ \vdots \\ x_n \end{bmatrix}, \quad \mathbf{y} = \begin{bmatrix} y_1 \\ y_2 \\ \vdots \\ y_m \end{bmatrix}, \quad (\text{D.1})$$

where each component y_i may be a function of all the x_j , a fact represented by saying that \mathbf{y} is a function of \mathbf{x} , or

$$\mathbf{y} = \mathbf{y}(\mathbf{x}). \quad (\text{D.2})$$

If $n = 1$, \mathbf{x} reduces to a scalar, which we call x . If $m = 1$, \mathbf{y} reduces to a scalar, which we call y . Various applications are studied in the following subsections.

§D.1.1 Derivative of Vector with Respect to Vector

The derivative of the vector \mathbf{y} with respect to vector \mathbf{x} is the $n \times m$ matrix

$$\frac{\partial \mathbf{y}}{\partial \mathbf{x}} \stackrel{\text{def}}{=} \begin{bmatrix} \frac{\partial y_1}{\partial x_1} & \frac{\partial y_2}{\partial x_1} & \cdots & \frac{\partial y_m}{\partial x_1} \\ \frac{\partial y_1}{\partial x_2} & \frac{\partial y_2}{\partial x_2} & \cdots & \frac{\partial y_m}{\partial x_2} \\ \vdots & \vdots & \ddots & \vdots \\ \frac{\partial y_1}{\partial x_n} & \frac{\partial y_2}{\partial x_n} & \cdots & \frac{\partial y_m}{\partial x_n} \end{bmatrix} \quad (\text{D.3})$$

§D.1.2 Derivative of a Scalar with Respect to Vector

If y is a scalar,

$$\frac{\partial y}{\partial \mathbf{x}} \stackrel{\text{def}}{=} \begin{bmatrix} \frac{\partial y}{\partial x_1} \\ \frac{\partial y}{\partial x_2} \\ \vdots \\ \frac{\partial y}{\partial x_n} \end{bmatrix}. \quad (\text{D.4})$$

§D.1.3 Derivative of Vector with Respect to Scalar

If x is a scalar,

$$\frac{\partial \mathbf{y}}{\partial x} \stackrel{\text{def}}{=} \begin{bmatrix} \frac{\partial y_1}{\partial x} & \frac{\partial y_2}{\partial x} & \cdots & \frac{\partial y_m}{\partial x} \end{bmatrix} \quad (\text{D.5})$$

REMARK D.1

Many authors, notably in statistics and economics, define the derivatives as the transposes of those given above.¹ This has the advantage of better agreement of matrix products with composition schemes such as the chain rule. Evidently the notation is not yet stable.

EXAMPLE D.1

Given

$$\mathbf{y} = \begin{bmatrix} y_1 \\ y_2 \end{bmatrix}, \quad \mathbf{x} = \begin{bmatrix} x_1 \\ x_2 \\ x_3 \end{bmatrix} \quad (\text{D.6})$$

and

$$\begin{aligned} y_1 &= x_1^2 - x_2 \\ y_2 &= x_3^2 + 3x_2 \end{aligned} \quad (\text{D.7})$$

the partial derivative matrix $\partial \mathbf{y} / \partial \mathbf{x}$ is computed as follows:

$$\frac{\partial \mathbf{y}}{\partial \mathbf{x}} = \begin{bmatrix} \frac{\partial y_1}{\partial x_1} & \frac{\partial y_1}{\partial x_2} & \frac{\partial y_1}{\partial x_3} \\ \frac{\partial y_2}{\partial x_1} & \frac{\partial y_2}{\partial x_2} & \frac{\partial y_2}{\partial x_3} \end{bmatrix} = \begin{bmatrix} 2x_1 & -1 & 0 \\ 0 & 3 & 2x_3 \end{bmatrix} \quad (\text{D.8})$$

§D.1.4 Jacobian of a Variable Transformation

In multivariate analysis, if \mathbf{x} and \mathbf{y} are of the same order, the determinant of the square matrix $\partial \mathbf{x} / \partial \mathbf{y}$, that is

$$J = \left| \frac{\partial \mathbf{x}}{\partial \mathbf{y}} \right| \quad (\text{D.9})$$

is called the *Jacobian* of the transformation determined by $\mathbf{y} = \mathbf{y}(\mathbf{x})$. The inverse determinant is

$$J^{-1} = \left| \frac{\partial \mathbf{y}}{\partial \mathbf{x}} \right|. \quad (\text{D.10})$$

¹ One authors puts it this way: “When one does matrix calculus, one quickly finds that there are two kinds of people in this world: those who think the gradient is a row vector, and those who think it is a column vector.”

EXAMPLE D.2

The transformation from spherical to Cartesian coordinates is defined by

$$x = r \sin \theta \cos \psi, \quad y = r \sin \theta \sin \psi, \quad z = r \cos \theta \quad (\text{D.11})$$

where $r > 0$, $0 < \theta < \pi$ and $0 \leq \psi < 2\pi$. To obtain the Jacobian of the transformation, let

$$\begin{aligned} x &\equiv x_1, & y &\equiv x_2, & z &\equiv x_3 \\ r &\equiv y_1, & \theta &\equiv y_2, & \psi &\equiv y_3 \end{aligned} \quad (\text{D.12})$$

Then

$$\begin{aligned} J = \left| \frac{\partial \mathbf{x}}{\partial \mathbf{y}} \right| &= \begin{vmatrix} \sin y_2 \cos y_3 & \sin y_2 \sin y_3 & \cos y_2 \\ y_1 \cos y_2 \cos y_3 & y_1 \cos y_2 \sin y_3 & -y_1 \sin y_2 \\ -y_1 \sin y_2 \sin y_3 & y_1 \sin y_2 \cos y_3 & 0 \end{vmatrix} \\ &= y_1^2 \sin y_2 = r^2 \sin \theta. \end{aligned} \quad (\text{D.13})$$

The foregoing definitions can be used to obtain derivatives to many frequently used expressions, including quadratic and bilinear forms.

EXAMPLE D.3

Consider the quadratic form

$$y = \mathbf{x}^T \mathbf{A} \mathbf{x} \quad (\text{D.14})$$

where \mathbf{A} is a square matrix of order n . Using the definition (D.3) one obtains

$$\frac{\partial y}{\partial \mathbf{x}} = \mathbf{A} \mathbf{x} + \mathbf{A}^T \mathbf{x} \quad (\text{D.15})$$

and if \mathbf{A} is symmetric,

$$\frac{\partial y}{\partial \mathbf{x}} = 2\mathbf{A} \mathbf{x}. \quad (\text{D.16})$$

We can of course continue the differentiation process:

$$\frac{\partial^2 y}{\partial \mathbf{x}^2} = \frac{\partial}{\partial \mathbf{x}} \left(\frac{\partial y}{\partial \mathbf{x}} \right) = \mathbf{A} + \mathbf{A}^T, \quad (\text{D.17})$$

and if \mathbf{A} is symmetric,

$$\frac{\partial^2 y}{\partial \mathbf{x}^2} = 2\mathbf{A}. \quad (\text{D.18})$$

The following table collects several useful vector derivative formulas.

\mathbf{y}	$\frac{\partial \mathbf{y}}{\partial \mathbf{x}}$
$\mathbf{A} \mathbf{x}$	\mathbf{A}^T
$\mathbf{x}^T \mathbf{A}$	\mathbf{A}
$\mathbf{x}^T \mathbf{x}$	$2\mathbf{x}$
$\mathbf{x}^T \mathbf{A} \mathbf{x}$	$\mathbf{A} \mathbf{x} + \mathbf{A}^T \mathbf{x}$

§D.2 THE CHAIN RULE FOR VECTOR FUNCTIONS

Let

$$\mathbf{x} = \begin{bmatrix} x_1 \\ x_2 \\ \vdots \\ x_n \end{bmatrix}, \quad \mathbf{y} = \begin{bmatrix} y_1 \\ y_2 \\ \vdots \\ y_r \end{bmatrix} \quad \text{and} \quad \mathbf{z} = \begin{bmatrix} z_1 \\ z_2 \\ \vdots \\ z_m \end{bmatrix} \quad (\text{D.19})$$

where \mathbf{z} is a function of \mathbf{y} , which is in turn a function of \mathbf{x} . Using the definition (D.2), we can write

$$\left(\frac{\partial \mathbf{z}}{\partial \mathbf{x}} \right)^T = \begin{bmatrix} \frac{\partial z_1}{\partial x_1} & \frac{\partial z_1}{\partial x_2} & \cdots & \frac{\partial z_1}{\partial x_n} \\ \frac{\partial z_2}{\partial x_1} & \frac{\partial z_2}{\partial x_2} & \cdots & \frac{\partial z_2}{\partial x_n} \\ \vdots & \vdots & & \vdots \\ \frac{\partial z_m}{\partial x_1} & \frac{\partial z_m}{\partial x_2} & \cdots & \frac{\partial z_m}{\partial x_n} \end{bmatrix} \quad (\text{D.20})$$

Each entry of this matrix may be expanded as

$$\frac{\partial z_i}{\partial x_j} = \sum_{q=1}^r \frac{\partial z_i}{\partial y_q} \frac{\partial y_q}{\partial x_j} \quad \begin{cases} i = 1, 2, \dots, m \\ j = 1, 2, \dots, n. \end{cases} \quad (\text{D.21})$$

Then

$$\begin{aligned} \left(\frac{\partial \mathbf{z}}{\partial \mathbf{x}} \right)^T &= \begin{bmatrix} \sum \frac{\partial z_1}{\partial y_q} \frac{\partial y_q}{\partial x_1} & \sum \frac{\partial z_1}{\partial y_q} \frac{\partial y_q}{\partial x_2} & \cdots & \sum \frac{\partial z_1}{\partial y_q} \frac{\partial y_q}{\partial x_n} \\ \sum \frac{\partial z_2}{\partial y_q} \frac{\partial y_q}{\partial x_1} & \sum \frac{\partial z_2}{\partial y_q} \frac{\partial y_q}{\partial x_2} & \cdots & \sum \frac{\partial z_2}{\partial y_q} \frac{\partial y_q}{\partial x_n} \\ \vdots & \vdots & & \vdots \\ \sum \frac{\partial z_m}{\partial y_q} \frac{\partial y_q}{\partial x_1} & \sum \frac{\partial z_m}{\partial y_q} \frac{\partial y_q}{\partial x_2} & \cdots & \sum \frac{\partial z_m}{\partial y_q} \frac{\partial y_q}{\partial x_n} \end{bmatrix} \\ &= \begin{bmatrix} \frac{\partial z_1}{\partial y_1} & \frac{\partial z_1}{\partial y_2} & \cdots & \frac{\partial z_1}{\partial y_r} \\ \frac{\partial z_2}{\partial y_1} & \frac{\partial z_2}{\partial y_2} & \cdots & \frac{\partial z_2}{\partial y_r} \\ \vdots & \vdots & & \vdots \\ \frac{\partial z_m}{\partial y_1} & \frac{\partial z_m}{\partial y_2} & \cdots & \frac{\partial z_m}{\partial y_r} \end{bmatrix} \begin{bmatrix} \frac{\partial y_1}{\partial x_1} & \frac{\partial y_1}{\partial x_2} & \cdots & \frac{\partial y_1}{\partial x_n} \\ \frac{\partial y_2}{\partial x_1} & \frac{\partial y_2}{\partial x_2} & \cdots & \frac{\partial y_2}{\partial x_n} \\ \vdots & \vdots & & \vdots \\ \frac{\partial y_r}{\partial x_1} & \frac{\partial y_r}{\partial x_2} & \cdots & \frac{\partial y_r}{\partial x_n} \end{bmatrix} \\ &= \left(\frac{\partial \mathbf{z}}{\partial \mathbf{y}} \right)^T \left(\frac{\partial \mathbf{y}}{\partial \mathbf{x}} \right)^T = \left(\frac{\partial \mathbf{y}}{\partial \mathbf{x}} \frac{\partial \mathbf{z}}{\partial \mathbf{y}} \right)^T. \end{aligned} \quad (\text{D.22})$$

On transposing both sides, we finally obtain

$$\frac{\partial \mathbf{z}}{\partial \mathbf{x}} = \frac{\partial \mathbf{y}}{\partial \mathbf{x}} \frac{\partial \mathbf{z}}{\partial \mathbf{y}}, \quad (\text{D.23})$$

which is the *chain rule* for vectors. If all vectors reduce to scalars,

$$\frac{\partial z}{\partial x} = \frac{\partial y}{\partial x} \frac{\partial z}{\partial y} = \frac{\partial z}{\partial y} \frac{\partial y}{\partial x}, \quad (\text{D.24})$$

which is the conventional chain rule of calculus. Note, however, that when we are dealing with vectors, the chain of matrices builds “toward the left.” For example, if \mathbf{w} is a function of \mathbf{z} , which is a function of \mathbf{y} , which is a function of \mathbf{x} ,

$$\frac{\partial \mathbf{w}}{\partial \mathbf{x}} = \frac{\partial \mathbf{y}}{\partial \mathbf{x}} \frac{\partial \mathbf{z}}{\partial \mathbf{y}} \frac{\partial \mathbf{w}}{\partial \mathbf{z}}. \quad (\text{D.25})$$

On the other hand, in the ordinary chain rule one can indistinctly build the product to the right or to the left because scalar multiplication is commutative.

§D.3 THE DERIVATIVE OF SCALAR FUNCTIONS OF A MATRIX

Let $\mathbf{X} = (x_{ij})$ be a matrix of order $(m \times n)$ and let

$$y = f(\mathbf{X}), \quad (\text{D.26})$$

be a scalar function of \mathbf{X} . The derivative of y with respect to \mathbf{X} , denoted by

$$\frac{\partial y}{\partial \mathbf{X}}, \quad (\text{D.27})$$

is defined as the following matrix of order $(m \times n)$:

$$\mathbf{G} = \frac{\partial y}{\partial \mathbf{X}} = \begin{bmatrix} \frac{\partial y}{\partial x_{11}} & \frac{\partial y}{\partial x_{12}} & \cdots & \frac{\partial y}{\partial x_{1n}} \\ \frac{\partial y}{\partial x_{21}} & \frac{\partial y}{\partial x_{22}} & \cdots & \frac{\partial y}{\partial x_{2n}} \\ \vdots & \vdots & & \vdots \\ \frac{\partial y}{\partial x_{m1}} & \frac{\partial y}{\partial x_{m2}} & \cdots & \frac{\partial y}{\partial x_{mn}} \end{bmatrix} = \left[\frac{\partial y}{\partial x_{ij}} \right] = \sum_{i,j} \mathbf{E}_{ij} \frac{\partial y}{\partial x_{ij}}, \quad (\text{D.28})$$

where \mathbf{E}_{ij} denotes the elementary matrix* of order $(m \times n)$. This matrix \mathbf{G} is also known as a *gradient matrix*.

EXAMPLE D.4

Find the gradient matrix if y is the trace of a square matrix \mathbf{X} of order n , that is

$$y = \text{tr}(\mathbf{X}) = \sum_{i=1}^n x_{ii}. \quad (\text{D.29})$$

Obviously all non-diagonal partials vanish whereas the diagonal partials equal one, thus

$$\mathbf{G} = \frac{\partial y}{\partial \mathbf{X}} = \mathbf{I}, \quad (\text{D.30})$$

where \mathbf{I} denotes the identity matrix of order n .

* The elementary matrix \mathbf{E}_{ij} of order $m \times n$ has all zero entries except for the (i, j) entry, which is one.

§D.3.1 Functions of a Matrix Determinant

An important family of derivatives with respect to a matrix involves functions of the determinant of a matrix, for example $y = |\mathbf{X}|$ or $y = |\mathbf{A}\mathbf{X}|$. Suppose that we have a matrix $\mathbf{Y} = [y_{ij}]$ whose components are functions of a matrix $\mathbf{X} = [x_{rs}]$, that is $y_{ij} = f_{ij}(x_{rs})$, and set out to build the matrix

$$\frac{\partial |\mathbf{Y}|}{\partial \mathbf{X}}. \quad (\text{D.31})$$

Using the chain rule we can write

$$\frac{\partial |\mathbf{Y}|}{\partial x_{rs}} = \sum_i \sum_j \mathbf{Y}_{ij} \frac{\partial |\mathbf{Y}|}{\partial y_{ij}} \frac{\partial y_{ij}}{\partial x_{rs}}. \quad (\text{D.32})$$

But

$$|\mathbf{Y}| = \sum_j y_{ij} \mathbf{Y}_{ij}, \quad (\text{D.33})$$

where \mathbf{Y}_{ij} is the *cofactor* of the element y_{ij} in $|\mathbf{Y}|$. Since the cofactors $\mathbf{Y}_{i1}, \mathbf{Y}_{i2}, \dots$ are independent of the element y_{ij} , we have

$$\frac{\partial |\mathbf{Y}|}{\partial y_{ij}} = \mathbf{Y}_{ij}. \quad (\text{D.34})$$

It follows that

$$\frac{\partial |\mathbf{Y}|}{\partial x_{rs}} = \sum_i \sum_j \mathbf{Y}_{ij} \frac{\partial y_{ij}}{\partial x_{rs}}. \quad (\text{D.35})$$

There is an alternative form of this result which is occasionally useful. Define

$$a_{ij} = \mathbf{Y}_{ij}, \quad \mathbf{A} = [a_{ij}], \quad b_{ij} = \frac{\partial y_{ij}}{\partial x_{rs}}, \quad \mathbf{B} = [b_{ij}]. \quad (\text{D.36})$$

Then it can be shown that

$$\frac{\partial |\mathbf{Y}|}{\partial x_{rs}} = \text{tr}(\mathbf{A}\mathbf{B}^T) = \text{tr}(\mathbf{B}^T \mathbf{A}). \quad (\text{D.37})$$

EXAMPLE D.5

If \mathbf{X} is a nonsingular square matrix and $\mathbf{Z} = |\mathbf{X}|\mathbf{X}^{-1}$ its cofactor matrix,

$$\mathbf{G} = \frac{\partial |\mathbf{X}|}{\partial \mathbf{X}} = \mathbf{Z}^T. \quad (\text{D.38})$$

If \mathbf{X} is also symmetric,

$$\mathbf{G} = \frac{\partial |\mathbf{X}|}{\partial \mathbf{X}} = 2\mathbf{Z}^T - \text{diag}(\mathbf{Z}^T). \quad (\text{D.39})$$

§D.4 THE MATRIX DIFFERENTIAL

For a scalar function $f(\mathbf{x})$, where \mathbf{x} is an n -vector, the ordinary differential of multivariate calculus is defined as

$$df = \sum_{i=1}^n \frac{\partial f}{\partial x_i} dx_i. \quad (\text{D.40})$$

In harmony with this formula, we define the differential of an $m \times n$ matrix $\mathbf{X} = [x_{ij}]$ to be

$$d\mathbf{X} \stackrel{\text{def}}{=} \begin{bmatrix} dx_{11} & dx_{12} & \dots & dx_{1n} \\ dx_{21} & dx_{22} & \dots & dx_{2n} \\ \vdots & \vdots & & \vdots \\ dx_{m1} & dx_{m2} & \dots & dx_{mn} \end{bmatrix}. \quad (\text{D.41})$$

This definition complies with the multiplicative and associative rules

$$d(\alpha\mathbf{X}) = \alpha d\mathbf{X}, \quad d(\mathbf{X} + \mathbf{Y}) = d\mathbf{X} + d\mathbf{Y}. \quad (\text{D.42})$$

If \mathbf{X} and \mathbf{Y} are product-conforming matrices, it can be verified that the differential of their product is

$$d(\mathbf{XY}) = (d\mathbf{X})\mathbf{Y} + \mathbf{X}(d\mathbf{Y}). \quad (\text{D.43})$$

which is an extension of the well known rule $d(xy) = y dx + x dy$ for scalar functions.

EXAMPLE D.6

If $\mathbf{X} = [x_{ij}]$ is a square nonsingular matrix of order n , and denote $\mathbf{Z} = |\mathbf{X}|\mathbf{X}^{-1}$. Find the differential of the determinant of \mathbf{X} :

$$d|\mathbf{X}| = \sum_{i,j} \frac{\partial |\mathbf{X}|}{\partial x_{ij}} dx_{ij} = \sum_{i,j} \mathbf{X}_{ij} dx_{ij} = \text{tr}(|\mathbf{X}|\mathbf{X}^{-1})^T d\mathbf{X} = \text{tr}(\mathbf{Z}^T d\mathbf{X}), \quad (\text{D.44})$$

where \mathbf{X}_{ij} denotes the cofactor of x_{ij} in \mathbf{X} .

EXAMPLE D.7

With the same assumptions as above, find $d(\mathbf{X}^{-1})$. The quickest derivation follows by differentiating both sides of the identity $\mathbf{X}^{-1}\mathbf{X} = \mathbf{I}$:

$$d(\mathbf{X}^{-1})\mathbf{X} + \mathbf{X}^{-1}d\mathbf{X} = \mathbf{0}, \quad (\text{D.45})$$

from which

$$d(\mathbf{X}^{-1}) = -\mathbf{X}^{-1}d\mathbf{X}\mathbf{X}^{-1}. \quad (\text{D.46})$$

If \mathbf{X} reduces to the scalar x we have

$$d\left(\frac{1}{x}\right) = -\frac{dx}{x^2}. \quad (\text{D.47})$$

F

Geometric Applications of Matrices

In this Appendix we summarize some geometric applications of matrices. The space is 3-dimensional. Indexed homogeneous Cartesian coordinates¹ $\{x_0, x_1, x_2, x_3\}$ are used. To pass to physical coordinates, divide x_1, x_2 and x_3 by x_0 : $\{x_1/x_0, x_2/x_0, x_3/x_0\}$. If $x_0 = 0$ the physical coordinates are at infinity.

§F.1 POINTS, PLANES

Points in 3D space will be identified by X, Y, P, Q , etc. Their coordinates are put in the 4-vectors

$$\mathbf{x} = [x_0 \ x_1 \ x_2 \ x_3]^T, \quad \mathbf{y} = [y_0 \ y_1 \ y_2 \ y_3]^T, \quad \mathbf{p} = [p_0 \ p_1 \ p_2 \ p_3]^T, \text{ etc} \quad (\text{F.1})$$

Planes in 3D space will be identified by A, B, C , etc. The equation of plane A is written $a_0x_0 + a_1x_1 + a_2x_2 + a_3x_3 = 0$ or

$$\mathbf{a}^T \mathbf{x} = 0, \quad \text{or} \quad \mathbf{x}^T \mathbf{a} = 0 \quad (\text{F.2})$$

The plane A is thus defined by the 4-vector \mathbf{a} .

EXAMPLE F.1

Find the coordinates of the point where line joining points P and Q intersects plane A .

Solution. Any point of PQ is R where

$$\mathbf{r} = \lambda \mathbf{p} + \mu \mathbf{q} \quad (\text{F.3})$$

If R is on plane A , then $\mathbf{a}^T \mathbf{x} = 0$ so that $\lambda \mathbf{a}^T \mathbf{p} + \mu \mathbf{a}^T \mathbf{q} = 0$. Absorbing a suitable multiplier into the coordinates of R we obtain its vector in the form

$$\mathbf{r} = (\mathbf{a}^T \mathbf{q}) \mathbf{p} - (\mathbf{a}^T \mathbf{p}) \mathbf{q}. \quad (\text{F.4})$$

§F.2 LINES

Let \mathbf{x} and \mathbf{y} be the coordinates of points X and Y on a given line L . The 4×4 antisymmetric matrix

$$\mathbf{L} = \mathbf{xy}^T - \mathbf{yx}^T \quad (\text{F.5})$$

is called the *coordinate matrix* of the line. It can be shown that \mathbf{L} determines the line L to within a scale factor.

Let A and B be two planes through L . The 4×4 antisymmetric matrix

$$\mathbf{L}^* = \mathbf{ab}^T - \mathbf{ba}^T \quad (\text{F.6})$$

is called the *dual coordinate matrix* of line L . It can be shown that²

$$\mathbf{L}^* \mathbf{L} = \mathbf{0}. \quad (\text{F.7})$$

¹ These coordinates were independently invented in 1827 by Möbius and Feuerbach, and further developed in 1946 by E. A. Maxwell at Cambridge. See E. A. Maxwell, *General Homogeneous Coordinates in Space of Three Dimensions*, Cambridge University Press, 1951.

² See E. A. Maxwell, loc. cit., page 150.

EXAMPLE F.2

Find where line L defined by two points X and Y meets a plane A .

Solution. Consider the vector

$$\mathbf{p} = \mathbf{L}\mathbf{a} = (\mathbf{xy}^T - \mathbf{yx}^T)\mathbf{a} = (\mathbf{y}^T\mathbf{a})\mathbf{x} - (\mathbf{x}^T\mathbf{a})\mathbf{y} \quad (\text{F.8})$$

From the last form the point P of coordinates \mathbf{p} must lie on the line that joins X and Y . Moreover since \mathbf{L} is antisymmetrical, $\mathbf{a}^T\mathbf{L}\mathbf{a} = 0$ so that $\mathbf{a}^T\mathbf{p} = 0$. Thus P is the intersection of line L and plane A .

EXAMPLE F.3

Find the plane A that joins line L to a point X .

Solution.

$$\mathbf{a} = \mathbf{L}^* \mathbf{x} \quad (\text{F.9})$$

where \mathbf{L}^* is the dual of \mathbf{L} . The demonstration is trivial.

Two immediate corollaries: (i) Line L lies in the plane A if $\mathbf{L}\mathbf{a} = \mathbf{0}$; (ii) Line L passes through the point X if $\mathbf{L}^*\mathbf{x} = \mathbf{0}$.

EXAMPLE F.4

Consider two lines L_1 and L_2 with coordinate matrices $\mathbf{L}_1, \mathbf{L}_2$ and dual matrices $\tilde{\mathbf{L}}_1$ and $\tilde{\mathbf{L}}_2$, respectively. Find the conditions for the lines to intersect.

Solution. Any of the four equivalent conditions

$$\mathbf{L}_1\mathbf{L}_2^*\mathbf{L}_1 = \mathbf{0}, \quad \mathbf{L}_1^*\mathbf{L}_2\mathbf{L}_1^* = \mathbf{0}, \quad \mathbf{L}_2\mathbf{L}_1^*\mathbf{L}_2 = \mathbf{0}, \quad \mathbf{L}_2^*\mathbf{L}_1\mathbf{L}_2^* = \mathbf{0}. \quad (\text{F.10})$$

For the proof see E. A. Maxwell, loc. cit, page 154.

G

Oldies but Goodies

The following list is fairly comprehensive until about 1989. Since then many more books as well as revised editions of previous ones have appeared. Newer books usually emphasize the mathematical interpretation and thus are of limited usefulness to engineers. In fact the field by now can be safely characterized, to paraphrase T. S. Eliot, as an engineer's wasteland.

Recommendation: stick with the oldies. Unfortunately many are out of print.

§G.1 MATHEMATICALLY ORIENTED

A. K. Aziz (ed.), *The Mathematical Foundations of the Finite Element Method with Applications to Partial Differential Equations*, Academic Press, New York, 1972.

G. F. Carey and J. T. Oden, *Finite Elements IV: Mathematical Aspects*, Prentice Hall, Englewood Cliffs, N. J., 1983.

P. G. Ciarlet, *The Finite Element Method for Elliptic Problems*, North Holland, Amsterdam, 1978.

D. F. Griffiths (ed.), *The Mathematical Basis of Finite Element Methods*, Clarendon Press, Oxford, UK 1984.

A. R. Mitchell and R. Wait, *The Finite Element Analysis in Partial Differential Equations*, Wiley, New York, 1977.

J. T. Oden, *An Introduction to the Mathematical Theory of Finite Elements*, Wiley, New York, 1976.

G. Strang and G. J. Fix, *An Analysis of the Finite Element Method*, Prentice Hall, Englewood Cliffs, N. J., 1973.

R. Wait and A. R. Mitchell, *Finite Element Analysis and Applications*, Wiley, Chichester, UK, 1985.

E. L. Wachpress, *A Rational Finite Element Basis*, Wiley, New York, 1976.

O. C. Zienkiewicz and K. Morgan, *Finite Element and Approximations*, Wiley, New York, 1983.

§G.2 APPLICATIONS ORIENTED

K. J. Bathe, *Finite Element Procedures in Engineering Analysis*, Prentice Hall, Englewood Cliffs, N. J., 1982. Second edition entitled *Finite Element Analysis: From Concepts to Applications* has appeared in 1996.

K. J. Bathe and E. L. Wilson, *Numerical Methods in Finite Element Analysis*, Prentice Hall, Englewood Cliffs, N. J., 1976.

E. B. Becker, G. F. Carey and J. T. Oden, *Finite Elements III: Computational Aspects*, Prentice Hall, Englewood Cliffs, N. J., 198

G. F. Carey and J. T. Oden, *Finite Elements II: A Second Course*, Prentice Hall, Englewood Cliffs, N. J., 1981.

M. V. K. Chari and P. P. Silvester, *Finite Elements in Electrical and Magnetic Field Problems*, Wiley, Chichester, UK, 1984.

T. H. Chung, *The Finite Element Method in Fluid Mechanics*, McGraw-Hill, New York, 1978.

- R. D. Cook, D. S. Malkus and M. E. Plesha, *Concepts and Application of Finite Element Methods*, 3rd ed., Wiley, New York, 1989.
- A. J. Davies, *The Finite Element Method*, Clarendon Press, Oxford, UK, 1980.
- C. S. Desai, *Elementary Finite Element Method*, Prentice Hall, Englewood Cliffs, N. J., 1979.
- C. S. Desai and J. F. Abel, *Introduction to the Finite Element Method*, Van Nostrand, New York, 1972.
- G. Dhatt and G. Touzot, *The Finite Element Method Displayed*, Wiley, Chichester, UK, 1984.
- R. H. Gallagher, *Finite Element Analysis*, Prentice Hall, Englewood Cliffs, N. J., 1975.
- I. Holand and K. Bell (eds), *Finite Element Methods in Stress Analysis*, Tapir, Trondheim, Norway, 1969.
- K. H. Huebner, *The Finite Element Method for Engineers*, Wiley, New York, 1975.
- T. J. R. Hughes, *The Finite Element Method: Linear Static and Dynamic Finite Element Analysis*, Prentice Hall, Englewood Cliffs, N. J., 1987.
- B. Irons and S. Ahmad, *Techniques of Finite Elements*, Ellis Horwood Ltd, Chichester, UK, 1980.
- H. C. Martin and G. F. Carey, *Introduction to Finite Element Analysis*, McGraw-Hill, New York, 1973.
- J. T. Oden, *Finite Elements of Nonlinear Continua*, Wiley, New York, 1972.
- J. T. Oden and G. F. Carey, *Finite Elements I: An Introduction*, Prentice Hall, Englewood Cliffs, N. J., 1981.
- J. T. Oden and G. F. Carey, *Finite Elements V: Special Problems in Solid Mechanics*, Prentice Hall, Englewood Cliffs, N. J., 1984.
- J. S. Przemieniecki, *Theory of Matrix Structural Analysis*, McGraw-Hill, New York, 1968 (also in Dover ed).
- S. S. Rao, *The Finite Element Method in Engineering*, Pergamon Press, Oxford, 1982.
- J. Robinson, *Integrated Theory of Finite Element Methods*, Wiley, London, 1973.
- K. C. Rockey, H. R. Evans, D. W. Griffiths and D. A. Nethercot, *The Finite Element Method: A Basic Introduction for Engineers*, Collins, London, 1983.
- L. J. Segerlind, *Applied Finite Element Analysis*, Wiley, New York, 1976.
- P. Tong and J. N. Rosettos, *Finite Element Method*, MIT Press, London, 1977.
- O. C. Zienkiewicz, *The Finite Element Method in Engineering Sciences*, 3rd ed., McGraw-Hill, London, 1977. Partly superseded by Vol I of Zienkiewicz and Taylor, McGraw-Hill, 1988. Vol II appeared in 1993.

§G.3 SOFTWARE ORIENTED

J. E. Akin, *Application and Implementation of Finite Element Methods*, Academic Press, New York, 1982.

E. Hinton and D. R. J. Owen, *An Introduction to Finite Element Computations*, Pineridge Press, Swansea, 1979.

I. M. Smith, *Programming the Finite Element Method*, Wiley, Chichester, UK, 1982.

§G.4 RECOMMENDED BOOKS FOR LINEAR FEM

Basic level (reference): Zienkiewicz and Taylor (1988) Vols I (1988), II(1993). This is a comprehensive upgrade of the 1977 edition. Primarily an encyclopedic reference work that gives panoramic coverage of FEM, as well as a comprehensive list of references. Not a textbook. A fifth edition has appeared recently.

Basic level (textbook): Cook, Malkus and Plesha (1989). This third edition is fairly comprehensive in scope and fairly up to date although the coverage is more superficial than Zienkiewicz and Taylor.

Intermediate level: Hughes (1987). It requires substantial mathematical expertise on the part of the reader.

Mathematically oriented: Strang and Fix (1973). Still the most readable mathematical treatment for engineers, although outdated in many subjects.

Best value for the \$\$\$: Przeminiecki (Dover edition, 1985, about \$14). A reprint of a 1966 Mac-Graw-Hill book. Although woefully outdated in many respects (the word “finite element” does not appear anywhere), it is a valuable reference for programming simple elements. It contains a comprehensive bibliography up to 1966.

Most fun (if you appreciate British “humor”): Irons and Ahmad (1980)

For buying out-of-print books through web services, check the search engine in <http://www.bookarea.com/>

H

An Outline of MSA History

A Historical Outline of Matrix Structural Analysis: A Play in Three Acts

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Abstract

The evolution of Matrix Structural Analysis (MSA) from 1930 through 1970 is outlined. Highlighted are major contributions by Collar and Duncan, Argyris, and Turner, which shaped this evolution. To enliven the narrative the outline is configured as a three-act play. Act I describes the pre-WWII formative period. Act II spans a period of confusion during which matrix methods assumed bewildering complexity in response to conflicting demands and restrictions. Act III outlines the cleanup and consolidation driven by the appearance of the Direct Stiffness Method, through which MSA completed morphing into the present implementation of the Finite Element Method.

Keywords: matrix structural analysis; finite elements; history; displacement method; force method; direct stiffness method; duality

§H.1 INTRODUCTION

Who first wrote down a stiffness or flexibility matrix?

The question was posed in a 1995 paper [1]. The educated guess was “somebody working in the aircraft industry of Britain or Germany, in the late 1920s or early 1930s.” Since then the writer has examined reports and publications of that time. These trace the origins of Matrix Structural Analysis to the aeroelasticity group of the National Physics Laboratory (NPL) at Teddington, a town that has now become a suburb of greater London.

The present paper is an expansion of the historical vignettes in Section 4 of [1]. It outlines the major steps in the evolution of MSA by highlighting the fundamental contributions of four individuals: Collar, Duncan, Argyris and Turner. These contributions are lumped into three milestones:

Creation. Beginning in 1930 Collar and Duncan formulated discrete aeroelasticity in matrix form. The first two journal papers on the topic appeared in 1934-35 [2,3] and the first book, couthored with Frazer, in 1938 [4]. The representation and terminology for discrete dynamical systems is essentially that used today.

Unification. In a series of journal articles appearing in 1954 and 1955 [5] Argyris presented a formal unification of Force and Displacement Methods using dual energy theorems. Although practical applications of the duality proved ephemeral, this work systematized the concept of assembly of structural system equations from elemental components.

FEMinization. In 1959 Turner proposed [6] the Direct Stiffness Method (DSM) as an efficient and general computer implementation of the then embryonic, and as yet unnamed, Finite Element Method.

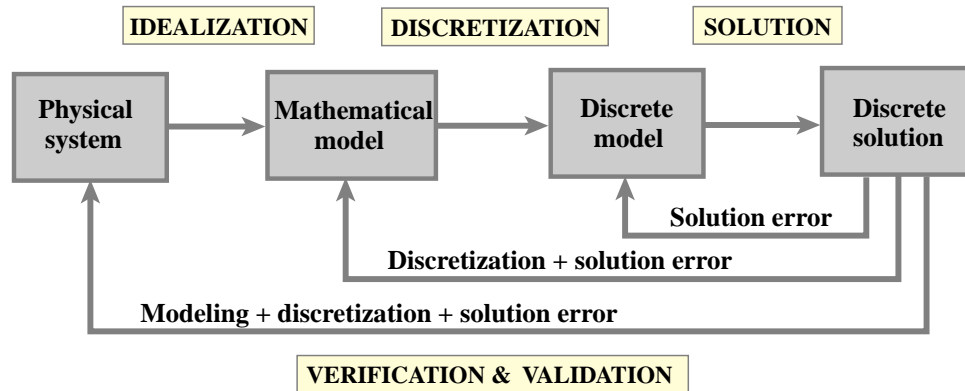


Figure 1. Flowchart of model-based simulation (MBS) by computer.

This technique, fully explained in a follow-up article [7], naturally encompassed structural and continuum models, as well as nonlinear, stability and dynamic simulations. By 1970 DSM had brought about the demise of the Classical Force Method (CFM), and become the dominant implementation in production-level FEM programs.

These milestones help dividing MSA history into three periods. To enliven and focus the exposition these will be organized as three acts of a play, properly supplemented with a “matrix overture” prologue, two interludes and a closing epilogue. Here is the program:

Prologue - *Victorian Artifacts*: 1858–1930.

Act I - *Gestation and Birth*: 1930–1938.

Interlude I - *WWII Blackout*: 1938–1947.

Act II - *The Matrix Forest*: 1947–1956.

Interlude II - *Questions*: 1956–1959.

Act III - *Answers*: 1959–1970.

Epilogue - *Revisiting the Past*: 1970–date.

Act I, as well as most of the Prologue, takes place in the U.K. The following events feature a more international cast.

§H.2 BACKGROUND AND TERMINOLOGY

Before departing for the theater, this Section offers some general background and explains historical terminology. Readers familiar with the subject should skip to Section 3.

The overall schematics of model-based simulation (MBS) by computer is flowcharted in Figure 1. For mechanical systems such as structures the Finite Element Method (FEM) is the most widely used discretization and solution technique. Historically the ancestor of FEM is MSA, as illustrated in Figure 2. The morphing of the MSA from the pre-computer era — as described for example in the first book [4] — into the first programmable computers took place, in wobbly gyrations, during the transition period herein called Act II. Following a confusing interlude, the young FEM begin to settle, during the early 1960s, into the configuration shown on the right of Figure 2. Its basic components have not changed since 1970.

MSA and FEM stand on three legs: mathematical models, matrix formulation of the discrete equations,

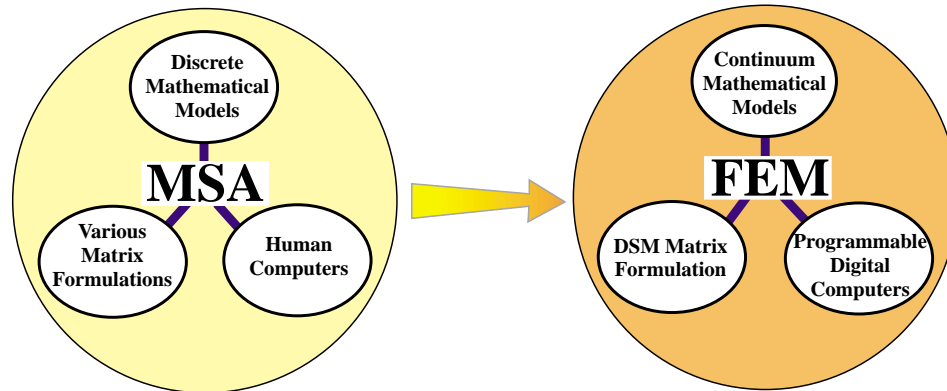


Figure 2. Morphing of the pre-computer MSA (before 1950) into the present FEM. On the left “human computer” means computations under direct human control, possibly with the help of analog devices (slide rule) or digital devices (desk calculator). The FEM configuration shown on the right settled by the mid 1960s.

and computing tools to do the numerical work. Of the three legs the latter is the one that has undergone the most dramatic changes. The “human computers” of the 1930s and 1940s morphed by stages into programmable computers of analog and digital type. The matrix formulation moved like a pendulum. It began as a simple displacement method in Act I, reached bewildering complexity in Act II and went back to conceptual simplicity in Act III.

Unidimensional structural models have changed little: a 1930 beam is still the same beam. The most noticeable advance is that pre-1955 MSA, following classical Lagrangian mechanics, tended to use spatially discrete energy forms from the start. The use of space-continuum forms as basis for multidimensional element derivation was pioneered by Argyris [5], successfully applied to triangular geometries by Turner, Clough, Martin and Topp [8], and finalized by Melosh [9] and Irons [10,11] with the precise statement of compatibility and completeness requirements for FEM.

Matrix formulations for MSA and FEM have been traditionally classified by the choice of *primary unknowns*. These are those solved for by the human or digital computer to determine the system state. In the Displacement Method (DM) these are physical or generalized displacements. In the Classical Force Method (CFM) these are amplitudes of redundant force (or stress) patterns. (The qualifier “classical” is important because there are other versions of the Force Method, which select for example stress function values or Lagrange multipliers as unknowns.) There are additional methods that involve combinations of displacements, forces and/or deformations as primary unknowns, but these have no practical importance in the pre-1970 period covered here.

Appropriate mathematical names for the DM are *range-space method* or *primal method*. This means that the primary unknowns are the same type as the primary variables of the governing functional. Appropriate names for the CFM are *null-space method*, *adjoint method*, or *dual method*. This means that the primary unknowns are of the same type of the adjoint variables of the governing functional, which in structural mechanics are forces. These names are not used in the historical outline, but are useful in placing more recent developments, as well as nonstructural FEM applications, within a general framework.

The terms Stiffness Method and Flexibility Method are more diffuse names for the Displacement and Force Methods, respectively. Generally speaking these apply when stiffness and flexibility matrices, respectively, are important part of the modeling and solution process.

§H.3 PROLOG - VICTORIAN ARTIFACTS: 1858-1930

Matrices — or “determinants” as they were initially called — were invented in 1858 by Cayley at Cambridge, although Gibbs (the co-inventor, along with Heaviside, of vector calculus) claimed priority for the German mathematician Grassmann. Matrix algebra and matrix calculus were developed primarily in the U.K. and Germany. Its original use was to provide a compact language to support investigations in mathematical topics such as the theory of invariants and the solution of algebraic and differential equations. For a history of these early developments the monograph by Muir [12] is unsurpassed. Several comprehensive treatises in matrix algebra appeared in the late 1920s and early 1930s [13–15].

Compared to vector and tensor calculus, matrices had relatively few applications in science and technology before 1930. Heisenberg’s 1925 matrix version of quantum mechanics was a notable exception, although technically it involved infinite matrices. The situation began to change with the advent of electronic desk calculators, because matrix notation provided a convenient way to organize complex calculation sequences. Aeroelasticity was a natural application because the stability analysis is naturally posed in terms of determinants of matrices that depend on a speed parameter.

The non-matrix formulation of Discrete Structural Mechanics can be traced back to the 1860s. By the early 1900s the essential developments were complete. A readable historical account is given by Timoshenko [16]. Interestingly enough, the term “matrix” never appears in this book.

§H.4 ACT I - GESTATION AND BIRTH: 1930-1938

In the decade of World War I aircraft technology began moving toward monoplanes. Biplanes disappeared by 1930. This evolution meant lower drag and faster speeds but also increased disposition to flutter. In the 1920s aeroelastic research began in an international scale. Pertinent developments at the National Physical Laboratory (NPL) are well chronicled in a 1978 historical review article by Collar [17], from which the following summary is extracted.

§H.4.1 The Source Papers

The aeroelastic work at the Aerodynamics Division of NPL was initiated in 1925 by R. A. Frazer. He was joined in the following year by W. J. Duncan. Two years later, in August 1928, they published a monograph on flutter [18], which came to be known as “The Flutter Bible” because of its completeness. It laid out the principles on which flutter investigations have been based since. In January 1930 A. R. Collar joined Frazer and Duncan to provide more help with theoretical investigations. Aeroelastic equations were tedious and error prone to work out in long hand. Here are Collar’s own words [17, page 17] on the motivation for introducing matrices:

“Frazer had studied matrices as a branch of applied mathematics under Grace at Cambridge; and he recognized that the statement of, for example, a ternary flutter problem in terms of matrices was neat and compendious. He was, however, more concerned with formal manipulation and transformation to other coordinates than with numerical results. On the other hand, Duncan and I were in search of numerical results for the vibration characteristics of airscrew blades; and we recognized that we could only advance by breaking the blade into, say, 10 segments and treating it as having 10 degrees of freedom. This approach also was more conveniently formulated in matrix terms, and readily expressed numerically. Then we found that if we put an approximate mode into one side of the equation, we calculated a better approximation on the other; and the matrix iteration procedure was born. We published our method in two papers in *Phil. Mag.* [2,3]; the first, dealing with conservative systems,

in 1934 and the second, treating damped systems, in 1935. By the time this had appeared, Duncan had gone to his Chair at Hull.”

The aforementioned papers appear to be the earliest *journal publications* of MSA. These are amazing documents: clean and to the point. They do not feel outdated. Familiar names appear: mass, flexibility, stiffness, and dynamical matrices. The matrix symbols used are $[m]$, $[f]$, $[c]$ and $[D] = [c]^{-1}[m] = [f][m]$, respectively, instead of the \mathbf{M} , \mathbf{F} , \mathbf{K} and \mathbf{D} in common use today. A general inertia matrix is called $[a]$. As befit the focus on dynamics, the displacement method is used. Point-mass displacement degrees of freedom are collected in a vector $\{x\}$ and corresponding forces in vector $\{P\}$. These are called $[q]$ and $[Q]$, respectively, when translated to generalized coordinates.

The notation was changed in the book [4] discussed below. In particular matrices are identified in [4] by capital letters without surrounding brackets, in more agreement with the modern style; for example mass, damping and stiffness are usually denoted by A , B and C , respectively.

§H.4.2 The MSA Source Book

Several papers on matrices followed, but apparently the traditional publication vehicles were not viewed as suitable for description of the new methods. At that stage Collar notes [17, page 18] that

“Southwell [Sir Richard Southwell, the “father” of relaxation methods] suggested that the authors of the various papers should be asked to incorporate them into a book, and this was agreed. The result was the appearance in November 1938 of “Elementary Matrices” published by Cambridge University Press [4]; it was the first book to treat matrices as a branch of applied mathematics. It has been reprinted many times, and translated into several languages, and even now after nearly 40 years, still sells in hundreds of copies a year — mostly paperback. The interesting thing is that the authors did not regard it as particularly good; it was the book we were instructed to write, rather than the one we would have liked to write.”

The writer has copies of the 1938 and 1963 printings. No changes other than minor fixes are apparent. Unlike the source papers [2,3] the book feels dated. The first 245 pages are spent on linear algebra and ODE-solution methods that are now standard part of engineering and science curricula. The numerical methods, oriented to desk calculators, are obsolete. That leaves the modeling and application examples, which are not coherently interweaved. No wonder that the authors were not happy about the book. They had followed Southwell’s “merge” suggestion too literally. Despite these flaws its direct and indirect influence during the next two decades was significant. Being first excuses imperfections.

The book focuses on dynamics of a complete airplane and integrated components such as wings, rudders or ailerons. The concept of *structural element* is primitive: take a shaft or a cantilever and divide it into segments. The assembled mass, stiffness or flexibility is given directly. The source of damping is usually aerodynamic. There is no static stress analysis; pre-WWII aircraft were overdesigned for strength and typically failed by aerodynamic or propulsion effects.

Readers are reminded that in aeroelastic analysis stiffness matrices are generally unsymmetric, being the sum of a symmetric elastic stiffness and an unsymmetric aerodynamic stiffness. This clean decomposition does not hold for flexibility matrices because the inverse of a sum is not the sum of inverses. The treatment of [4] includes the now called load-dependent stiffness terms, which represent another first.

On reading the survey articles by Collar [17,19] one cannot help being impressed by the lack of pretension. With Duncan he had created a tool for future generations of engineers to expand and improve upon. Yet he appears almost apologetic: “I will complete the matrix story as briefly as

possible” [17, page 17]. The NPL team members shared a common interest: to troubleshoot problems by understanding the physics, and viewed numerical methods simply as helpers.

§H.5 INTERLUDE I - WWII BLACKOUT: 1938-1947

Interlude I is a “silent period” taken to extend from the book [4] to the first journal publication on the matrix Force Method for aircraft [20]. Aeroelastic research continued. New demands posed by high strength materials, higher speeds, combat maneuvers, and structural damage survival increased interest in stress analysis. For the beam-like skeletal configurations of the time, the traditional flexibility-based methods such as CFM were appropriate. Flexibilities were often measured experimentally by static load tests, and fitted into the calculations. Punched-card computers and relay-calculators were increasingly used, and analog devices relied upon to solve ODEs in guidance and ballistics. Precise accounts of MSA work in aerospace are difficult to trace because of publication restrictions. The blackout was followed by a 2-3 year hiatus until those restrictions were gradually lifted, R&D groups restaffed, and journal pipelines refilled.

§H.6 ACT II - THE MATRIX FOREST: 1947-1956

As Act II starts MSA work is still mainly confined to the aerospace community. But the focus has shifted from dynamics to statics, and especially stress, buckling, fracture and fatigue analysis. Turbines, supersonic flight and rocket propulsion brought forth thermomechanical effects. The Comet disasters forced attention on stress concentration and crack propagation effects due to cyclic cabin pressurization. Failsafe design gained importance. In response to these multiple demands aircraft companies staffed specialized groups: stress, aerodynamics, aeroelasticity, propulsion, avionics, and so on. A multilevel management structure with well defined territories emerged.

The transition illustrated in Figure 2 starts, driven by two of the legs supporting MSA: new computing resources and new mathematical models. The matrix formulation merely reacts.

§H.6.1 Computers Become Machines

The first electronic commercial computer: Univac I, manufactured by a division of Remington-Rand, appeared during summer 1951. The six initial machines were delivered to US government agencies [21]. It was joined in 1952 by the Univac 1103, a scientific-computation oriented machine built by ERA, a R-R acquisition. This was the first computer with a drum memory. T. J. Watson Sr., founder of IBM, had been once quoted as saying that six electronic computers would satisfy the needs of the planet. Turning around from that prediction, IBM launched the competing 701 model in 1953.

Big aircraft companies began purchasing or leasing these expensive wonders by 1954. But this did not mean immediate access for everybody. The behemoths had to be programmed in machine or assembly code by specialists, who soon formed computer centers allocating and prioritizing cycles. By 1956 structural engineers were still likely to be using their slides rules, Marchants and punched card equipment. Only after the 1957 appearance of the first high level language (Fortran I, offered on the IBM 704) were engineers and scientists able (and allowed) to write their own programs.

§H.6.2 The Matrix CFM Takes Center Stage

In static analysis the non-matrix version of the Classical Force Method (CFM) had enjoyed a distinguished reputation since the source contributions by Maxwell, Mohr and Castigliano. The method provides directly the internal forces, which are of paramount interest in stress-driven design. It offers considerable scope of ingenuity to experienced structural engineers through clever selection of

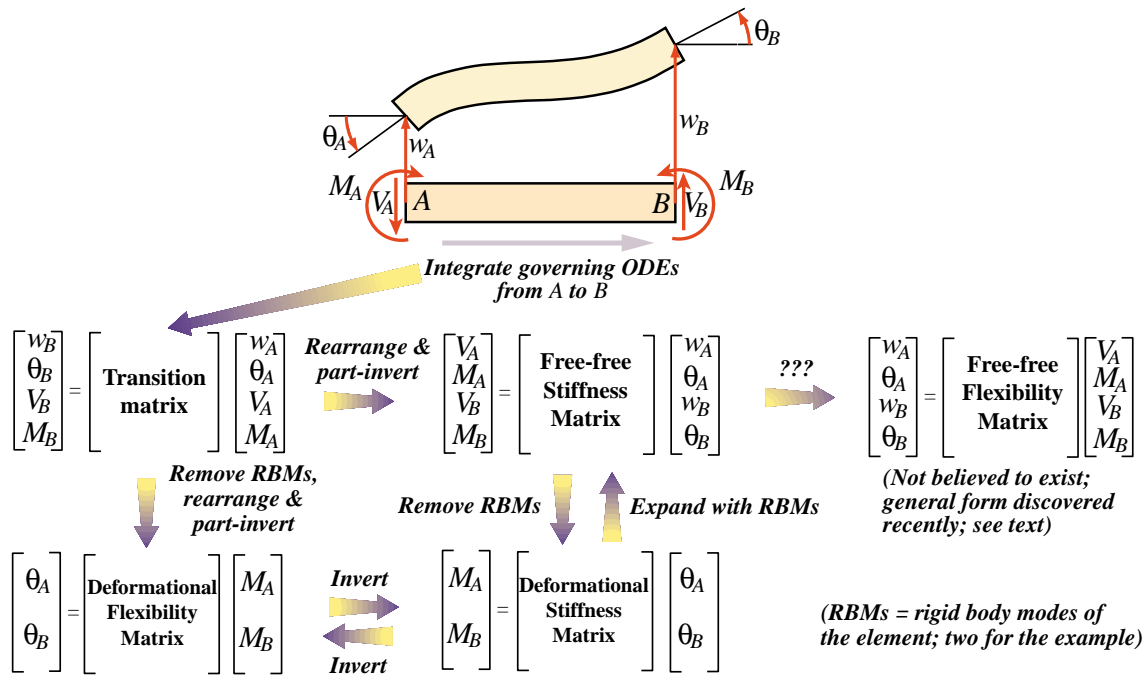


Figure 3. Transition, flexibility and stiffness matrices for unidimensional linear structural elements, such as the plane beam depicted here, can be obtained by integrating the governing differential equations, analytically or numerically, over the member to relate end forces and displacements. Clever things were done with this “method of lines” approach, such as including intermediate supports or elastic foundations.

redundant force systems. It was routinely taught to Aerospace, Civil and Mechanical Engineering students.

Success in hand-computation dynamics depends on “a few good modes.” Likewise, the success of CFM depends crucially on the selection of good redundant force patterns. The structures of pre-1950 aircraft were a fairly regular lattice of ribs, spars and panels, forming beam-like configurations. If the panels are ignored, the selection of appropriate redundants was well understood. Panels were modeled conservatively as inplane shear-force carriers, circumventing the difficulties of two-dimensional elasticity. With some adjustments and experimental validations, sweptback wings of high aspect ratio were eventually fitted into these models.

A matrix framework was found convenient to organize the calculations. The first journal article on the matrix CFM, which focused on sweptback wing analysis, is by Levy [20], followed by publications of Rand [22], Langefors [23], Wehle and Lansing [24] and Denke[25]. The development culminates in the article series of Argyris [5] discussed in Section 6.5.

§H.6.3 The Delta Wing Challenge

The Displacement Method (DM) continued to be used for vibration and aeroelastic analysis, although as noted above this was often done by groups separated from stress and buckling analysis. A new modeling challenge entered in the early 1950s: delta wing structures. This rekindled interest in stiffness methods.

The traditional approach to obtain flexibility and stiffness matrices of unidimensional structural members such as bars and shafts is illustrated in Figure 3. The governing differential equations are integrated, analytically or numerically, from one end to the other. The end quantities, grouping forces and displacements, are thereby connected by a transition matrix. Using simple algebraic manipulations three more matrices shown in Figure 3 can be obtained: deformational flexibility, deformational stiffness and free-free stiffness. This well known technique has the virtue of reducing the number of unknowns since the integration process can absorb structural details that are handled in the present FEM with multiple elements.

Notably absent from the scheme of Figure 3 is the free-free flexibility. This was not believed to exist since it is the inverse of the free-free stiffness, which is singular. A general closed-form expression for this matrix as a Moore-Penrose generalized stiffness inverse was not found until recently [26,27].

Modeling delta wing configurations required two-dimensional panel elements of arbitrary geometry, of which the triangular shape, illustrated in Figure 4, is the simplest and most versatile. Efforts to follow the ODE-integration approach lead to failure. (One particularly bizarre proposal, for solving exactly the wrong problem, is mentioned for fun in the label of Figure 4.) This motivated efforts to construct the stiffness matrix of the panel directly. The first attempt in this direction is by Levy [28]; this was only partly successful but was able to illuminate the advantages of the stiffness approach.

The article series by Argyris [5] contains the derivation of the 8×8 free-free stiffness of a flat rectangular panel using bilinear displacement interpolation in Cartesian coordinates. But that geometry was obviously inadequate to model delta wings. The landmark contribution of Turner, Clough, Martin and Topp [8] finally succeeded in directly deriving the stiffness of a triangular panel. Clough [29] observes that this paper represents the delayed publication of 1952-53 work at Boeing. It is recognized as one of the two sources of present FEM implementations, the second being the DSM discussed later. Because of the larger number of unknowns compared to CFM, competitive use of the DM in stress analysis had necessarily to wait until computers become sufficiently powerful to handle hundreds of simultaneous equations.

§H.6.4 Reduction Fosters Complexity

For efficient digital computation on present computers, data organization (in terms of fast access as well as exploitation of sparseness, vectorization and parallelism) is of primary concern whereas raw problem size, up to certain computer-dependent bounds, is secondary. But for hand calculations minimal problem size is a key aspect. Most humans cannot comfortably solve by hand linear systems of more than 5 or 6 unknowns by direct elimination methods, and 5–10 times that through problem-oriented “relaxation” methods. The first-generation digital computers improved speed and reliability, but were memory strapped. For example the Univac I had 1000 45-bit words and the IBM 701, 2048 36-bit words. Clearly solving a full system of 100 equations was still a major challenge.

It should come as no surprise that problem reduction techniques were paramount throughout this period, and exerted noticeable influence until the early 1970s. In static analysis reduction was achieved by elaborated *functional groupings* of static and kinematic variables. Most schemes of the time can be

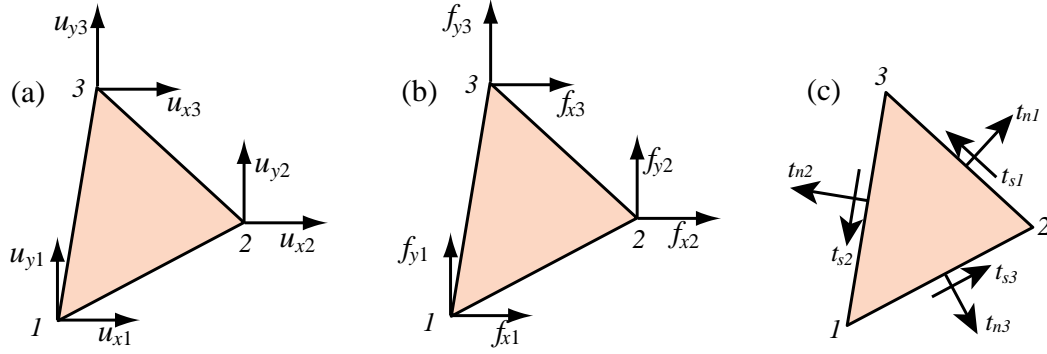


Figure 4. Modeling delta wing configurations required panel elements of arbitrary geometry such as the triangles depicted here. The traditional ODE-based approach of Figure 3 was tried by some researchers who (seriously) proposed finding the corner displacements in (a) produced by the concentrated corner forces in (b) on a supported triangle from the elasticity equations solved by numerical integration! Bad news: those displacements are infinite. Interior fields assumptions were inevitable, but problems persisted. A linear inplane displacement field is naturally specified by corner displacements, whereas a constant membrane force field is naturally defined by edge tractions (c). Those quantities “live” on different places. The puzzle was first solved in [8] by lumping edge tractions to node forces on the way to the free-free stiffness matrix.

understood in terms of the following classification:

$$\begin{aligned}
 &\text{generalized forces} \quad \left\{ \begin{array}{l} \text{primary} \quad \left\{ \begin{array}{l} \text{applied forces } \mathbf{f}_a \\ \text{redundant forces } \mathbf{y} \end{array} \right. \\ \text{secondary} \quad \left\{ \begin{array}{l} \text{condensable forces } \mathbf{f}_c = \mathbf{0} \\ \text{support reactions } \mathbf{f}_s \end{array} \right. \end{array} \right. \\
 &\text{generalized displacements} \quad \left\{ \begin{array}{l} \text{primary} \quad \left\{ \begin{array}{l} \text{applied displacements } \mathbf{u}_a \\ \text{redundant displacements } \mathbf{z} \end{array} \right. \\ \text{secondary} \quad \left\{ \begin{array}{l} \text{condensable displacements } \mathbf{u}_c \\ \text{support conditions } \mathbf{u}_s = \mathbf{0} \end{array} \right. \end{array} \right. \quad (\text{H.1})
 \end{aligned}$$

Here *applied forces* are those acting with nonzero values, that is, the ones visibly drawn as arrows by an engineer or instructor. In reduction-oriented thinking zero forces on unloaded degrees of freedom are classified as *condensable* because they can be removed through static condensation techniques. Similarly, nonzero *applied displacements* were clearly differentiated from zero-displacements arising from support conditions because the latter can be thrown out while the former must be retained. Redundant displacements, which are the counterpart of redundant forces, have been given many names, among them “kinematically indeterminate displacements” and “kinematic deficiencies.”

Matrix formulation evolved so that the unknowns were the force redundants \mathbf{y} in the CFM and the displacement redundants \mathbf{z} in the DM. Partitioning matrices in accordance to (H.1) fostered exuberant growth culminating in the *matrix forest* that characterizes works of this period.

To a present day FEM programmer familiar with the DSM, the complexity of the matrix forest would strike as madness. The DSM master equations can be assembled without functional labels. Boundary conditions are applied on the fly by the solver. But the computing limitations of the time must be kept in mind to see the method in the madness.

§H.6.5 Two Paths Through the Forest

A series of articles published by J. H. Argyris in four issues of *Aircraft Engrg.* during 1954 and 1955 collectively represents the second major milestone in MSA. In 1960 the articles were collected in a book, entitled “Energy Theorems and Structural Analysis” [5]. Part I, sub-entitled General Theory, reprints the four articles, whereas Part II, which covers additional material on thermal analysis and torsion, is co-authored by Argyris and Kelsey. Both authors are listed as affiliated with the Aerospace Department of the Imperial College at London.

The dual objectives of the work, stated in the Preface, are “to generalize, extend and unify the fundamental energy principles of elastic structures” and “to describe in detail practical methods of analysis of complex structures — in particular for aeronautical applications.” The first objective succeeds well, and represents a key contribution toward the development of continuum-based models. Part I carefully merges classical contributions in energy and work methods with matrix methods of discrete structural systems. The coverage is methodical, with numerous illustrative examples. The exposition of the Force Method for wing structures reaches a level of detail unequalled for the time.

The Displacement Method is then introduced by duality — called “analogy” in this work:

“The analogy between the developments for the flexibilities and stiffnesses ... shows clearly that parallel to the analysis of structures with forces as unknowns there must be a corresponding theory with deformations as unknowns.”

This section credits Ostenfeld [30] with being the first to draw attention to the parallel development. The duality is exhibited in a striking Form in Table II, in which both methods are presented side by side with simply an exchange of symbols and appropriate rewording. The steps are based on the following decomposition of internal deformation states \mathbf{g} and force patterns \mathbf{p} :

$$\mathbf{p} = \mathbf{B}_0 \mathbf{f}_a + \mathbf{B}_1 \mathbf{y}, \quad \mathbf{g} = \mathbf{A}_0 \mathbf{u}_a + \mathbf{A}_1 \mathbf{z}, \quad (\text{H.2})$$

Here the \mathbf{B}_i and \mathbf{A}_i denote system equilibrium and compatibility matrices, respectively. The vector symbols on the right reflect a particular choice of the force-displacement decomposition (H.1), with kinematic deficiencies taken to be the condensable displacements: $\mathbf{z} \equiv \mathbf{u}_c$.

This unification exerted significant influence over the next decade, particularly on the European community. An excellent textbook exposition is that of Pestel and Leckie [31]. This book covers both paths, following Argyris’ framework, in Chapters 9 and 10, using 83 pages and about 200 equations. These chapters are highly recommended to understand the organization of numeric and symbolic hand computations in vogue at that time, but it is out of print. Still in print (by Dover) is the book by Przemieniecki [32], which describes the DM and CFM paths in two Chapters: 6 and 8. The DM coverage is strongly influenced, however, by the DSM; thus duality is only superficially used.

§H.6.6 Dubious Duality

A key application of the duality in [5] was to introduce the DM by analogy to the then better known CFM. Although done with good intentions this approach did not anticipate the direct development of continuum-based finite elements through stiffness methods. These can be formulated from the total potential energy principle via shape functions, a technique not fully developed until the mid 1960s.

The side by side presentation of Table II of [5] tried to show that CFM and DM were going through exactly the same sequence of steps. Some engineers, eventually able to write Fortran programs, concluded that the methods had similar capabilities and selecting one or the other was a matter of

taste. (Most structures groups, upholding tradition, opted for the CFM.) But the few engineers who tried implementing both noticed a big difference. And that was before the DSM, which has no dual counterpart under the decomposition (H.2), appeared.

The paradox is explained in Section 4 of [1]. It is also noted there that (H.2) is not a particularly useful state decomposition. A better choice is studied in Section 2 of that paper; that one permits all known methods of Classical MSA, including the DSM, to be derived for skeletal structures as well as for a subset of continuum models.

§H.7 INTERLUDE II - QUESTIONS: 1956-1959

Interlude I was a silent period dominated by the war blackout. Interlude II is more vocal: a time of questions. An array of methods, models, tools and applications is now on the table, and growing. Solid-state computers, Fortran, ICBMs, the first satellites. So many options. Stiffness or flexibility? Forces or displacements? Do transition matrix methods have a future? Is the CFM-DM duality a precursor to general-purpose programs that will simulate everything? Will engineers be allowed to write those programs?

As convenient milestone this outline takes 1959, the year of the first DSM paper, as the beginning of Act III. Arguments and counter-arguments raised by the foregoing questions will linger, however, for two more decades into diminishing circles of the aerospace community.

§H.8 ACT III - ANSWERS: 1959-1970

The curtain of Act III lifts in Aachen, Germany. On 6 November 1959, M. J. Turner, head of the Structural Dynamics Unit at Boeing and an expert in aeroelasticity, presented the first paper on the Direct Stiffness Method to an AGARD Structures and Materials Panel meeting [6]. (AGARD is NATO's Advisory Group for Aeronautical Research and Development, which had sponsored workshops and lectureships since 1952. Bound proceedings or reports are called AGARDographs.)

§H.8.1 A Path Outside the Forest

No written record of [6] seem to exist. Nonetheless it must have produced a strong impression since published contributions to the next (1962) panel meeting kept referring to it. By 1960 the method had been applied to nonlinear problems [33] using incremental techniques. In July 1962 Turner, Martin and Weikel presented an expanded version of the 1959 paper, which appeared in an AGARDograph volume published by Pergamon in 1964 [7]. Characteristic of Turner's style, the Introduction goes directly to the point:

"In a paper presented at the 1959 meeting of the AGARD Structures and Material Panel in Aachen, the essential features of a system for numerical analysis of structures, termed the direct-stiffness method, were described. The characteristic feature of this particular version of the displacement method is the assembly procedure, whereby the stiffness matrix for a composite structure is generated by direct addition of matrices associated with the elements of the structure."

The DSM is explained in six text lines and three equations:

"For an individual element e the generalized nodal force increments $\{\Delta X^e\}$ required to maintain a set of nodal displacement increments $\{\Delta u\}$ are given by a matrix equation

$$\{\Delta X^e\} = K^e \{\Delta u\} \quad (\text{H.3})$$

in which K^e denotes the stiffness matrix of the individual element. Resultant nodal force increments acting on the complete structure are

$$\{\Delta X\} = \sum \{\Delta X^e\} = K \{\Delta u\} \quad (\text{H.4})$$

wherein K , the stiffness of the complete structure, is given by the summation

$$K = \sum K^e \quad (\text{H.5})$$

which provides the basis for the matrix assembly procedure noted earlier.”

Knowledgeable readers will note a notational glitch. For (H.3)-(H.5) to be correct matrix equations, K^e must be an element stiffness fully expanded to global (in that paper: “basic reference”) coordinates, a step that is computationally unnecessary. A more suggestive notation used in present DSM expositions is $K = \sum (L^e)^T K^e L^e$, in which L^e are Boolean localization matrices. Note also the use of Δ in front of u and X and their identification as “increments.” This simplifies the extension to nonlinear analysis, as outlined in the next paragraph:

“For the solution of linear problems involving small deflections of a structure at constant uniform temperature which is initially stress-free in the absence of external loads, the matrices K^e are defined in terms of initial geometry and elastic properties of the materials comprising the elements; they remain unchanged throughout the analysis. Problems involving nonuniform heating of redundant structures and/or large deflections are solved in a sequence of linearized steps. Stiffness matrices are revised at the beginning of each step to account for changes in internal loads, temperatures and geometric configurations.”

Next are given some computer implementation details, including the first ever mention of user-defined elements:

“Stiffness matrices are generally derived in local reference systems associated with the elements (as prescribed by a set of subroutines) and then transformed to the basic reference system. It is essential that the basic program be able to accommodate arbitrary additions to the collection of subroutines as new elements are encountered. Associated with these are a set of subroutines for generation of stress matrices S^e relating matrices of stress components σ^e in the local reference system of nodal displacements:

$$\{\sigma^e\} = S^e \{\bar{u}\} \quad (\text{H.6})$$

The vector $\{\bar{u}\}$ denotes the resultant displacements relative to a local reference system which is attached to the element. ... Provision should also be made for the introduction of numerical stiffness matrices directly into the program. This permits the utilization and evaluation of new element representations which have not yet been programmed. It also provides a convenient mechanism for introducing local structural modifications into the analysis.”

The assembly rule (H.3)-(H.5) is insensitive to element type. It work the same way for a 2-node bar, or a 64-node hexahedron. To do dynamics and vibration one adds mass and damping terms. To do buckling one adds a geometric stiffness and solves the stability eigenproblem, a technique first explained in [33]. To do nonlinear analysis one modifies the stiffness in each incremental step. To apply multipoint constraints the paper [7] advocates a master-slave reduction method.

Some computational aspects are missing from this paper, notably the treatment of simple displacement boundary conditions, and the use of sparse matrix assembly and solution techniques. The latter were first addressed in Wilson’s thesis work [34,35].

§H.8.2 The Fire Spreads

DSM is a paragon of elegance and simplicity. The writer is able to teach the essentials of the method in three lectures to graduate and undergraduate students alike. Through this path the old MSA and the young FEM achieved smooth confluence. The matrix formulation returned to the crispness of the source papers [2,3]. A widely referenced MSA correlation study by Gallagher [36] helped dissemination. Computers of the early 1960s were finally able to solve hundreds of equations. In an ideal world, structural engineers should have quickly razed the forest and embraced DSM.

It did not happen that way. The world of aerospace structures split. DSM advanced first by word of mouth. Among the aerospace companies, only Boeing and Bell (influenced by Turner and Gallagher, respectively) had made major investments in DSM by 1965. Among academia the Civil Engineering Department at Berkeley became a DSM evangelist through Clough, who made his students — including the writer — use DSM in their thesis work. These codes were freely disseminated into the non-aerospace world since 1963. Martin established similar traditions at Washington University, and Zienkiewicz, influenced by Clough, at Swansea. The first textbook on FEM [37], which appeared in 1967, makes no mention of force methods. By then the application to non-structural field problems (thermal, fluids, electromagnetics, ...) had begun, and again the DSM scaled well into the brave new world.

§H.8.3 The Final Test

Legacy CFM codes continued, however, to be used at many aerospace companies. The split reminds one of Einstein's answer when he was asked about the reaction of the old-guard school to the new physics: "we did not convince them; we outlived them." Structural engineers hired in the 1940s and 1950s were often in managerial positions in the 1960s. They were set in their ways. How can duality fail? All that is needed are algorithms for having the computer select good redundants automatically. Substantial effort was spent in those "structural cutters" during the 1960s [32,38].

That tenacity was eventually put to a severe test. The 1965 NASA request-for-proposal to build the NASTRAN finite element system called for the simultaneous development of Displacement and Force versions [39]. Each version was supposed to have identical modeling and solution capabilities, including dynamics and buckling. Two separate contracts, to MSC and Martin, were awarded accordingly. Eventually the development of the Force version was cancelled in 1969. The following year may be taken as closing the transition depicted in Figure 2, and as marking the end of the Force Method as a serious contender for general-purpose FEM programs.

§H.9 EPILOGUE - REVISITING THE PAST: 1970-DATE

Has MSA, now under the wider umbrella of FEM, attained a final form? This seems the case for general-purpose FEM programs, which by now are truly “1960 heritage” codes.

Resurrection of the CFM for special uses, such as optimization, was the subject of a speculative technical note by the writer [40]. This was motivated by efforts of numerical analysts to develop sparse null-space methods [41–45]. That research appears to have been abandoned by 1990. Section 2 of [26] elaborates on why, barring unexpected breakthroughs, a resurrection of the CFM is unlikely.

A more modest revival involves the use of non-CFM *flexibility methods* for multilevel analysis. The structure is partitioned into subdomains or substructures, each of which is processed by DSM; but the subdomains are connected by Lagrange multipliers that physically represent node forces. A key driving application is massively parallel processing in which subdomains are mapped on distributed-memory processors and the force-based interface subproblem solved iteratively by FETI methods [46]. Another set of applications include inverse problems such as system identification and damage detection. Pertinent references and a historical sketch may be found in a recent article [47] that presents a hybrid variational formulation for this combined approach.

The true duality for structural mechanics is now known to involve displacements and stress functions, rather than displacements and forces. This was discovered by Fraeijs de Veubeke in the 1970s [48]. Although extendible beyond structures, the potential of this idea remains largely unexplored.

§H.10 CONCLUDING REMARKS

The patient reader who has reached this final section may have noticed that this essay is a critical overview of MSA history, rather than a recital of events. It reflects personal interpretations and opinions. There is no attempt at completeness. Only what are regarded as major milestones are covered in some detail. Furthermore there is only spotty coverage of the history of FEM itself as well as its computer implementation; this is the topic of an article under preparation for Applied Mechanics Reviews.

This outline can be hopefully instructive in two respects. First, matrix methods now in disfavor may come back in response to new circumstances. An example is the resurgence of flexibility methods in massively parallel processing. A general awareness of the older literature helps. Second, the sweeping victory of DSM over the befuddling complexity of the “matrix forest” period illustrates the virtue of Occam’s proscription against multiplying entities: when in doubt chose simplicity. This dictum is relevant to the present confused state of computational mechanics.

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