The FElt System:

User’s Guide and Reference Manua

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FElt: User's Guide and Reference Manual

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Foreword

FElt is intended as a tool for teaching finite element analysis methods. There are better tools, there are bigger tools, there are tools that can do many, many things that FElt cannot do. Those tools are for the most part not free and if they are, they're usually 20 years old. FElt is a new, continually evolving system that tries to provide lots of modern workstation type features and of course, it's completely free.

About this Manual

This manual documents **version 3.05** of FElt. It is part of a futile attempt to provide comprehensive, accurate documentation for the FElt system. We do our best to try to keep it up-to-date with the latest version of the software; we make no guarantees however, and chances are good that there are things wrong in here. If you find something that behaves differently than the way this document says it should behave then please let us know.

Organization of this Manual

This manual is organized in the following way:

Chapter 1 gives you an introduction to the types of problems that FElt can solve.

Chapter 2 details some of the underlying mathematics for each of the analysis types supported by FElt.

Chapter 3 discusses the basic structure of a FElt input file and how a general finite element problem is translated into the FElt language.

Chapter 4 describes the currently available elements within FElt.

Chapter 5 introduces you to the simplest user interface to the FElt system, the command line application *felt*.

Chapter 6 covers the *WinFElt* graphical encapsulator for the MS-Windows based environments.

Chapter 7 introduces *velvet*, a full-featured graphical environment for the FElt system.

Chapter 8 covers problem solutions and post-processing options within *velvet*

Chapter 9 discusses the syntax and usage of *corduroy*, the command line mesh generator program for FElt.

Chapter 10 introduces the powerful and flexible interactive environment *burlap*.

Chapter 11 describes the syntax of *burlap* in detail.

Chapter 12 describes some of the algorithms that FElt uses in solving an arbitrary problem.

Chapter 13 is an attempt at teaching you how to add elements to the FElt library.

Appendix A discusses building, installing and administering the FElt system. A must for potential administrators.

Appendix B provides a list of Geompack error codes. You'll want to keep this handy if you find yourself doing a lot of mesh generation.

Appendix C contains a copy of the GNU General Public License, the terms under which FElt is distributed.

Typographical Conventions

In writing this guide, a number of typographical conventions were employed to mark buttons, command names, menu options, screen interaction, etc.

Bold Font	Used to mark **buttons**, and **menu options** in graphical environments.
Italics Font	Used to indicate an application program name, e.g. *felt*.
`Typewriter Font`	Used to represent screen interaction, either with the *velvet* command line, or the shell prompt. Also used for example input files, keywords that belong in input files and code examples.
Key	Represents a key (or key combination) to press, as in press Return to continue.

Acknowledgements

We would like to acknowledge the work of the following people or groups. Different bits and pieces of their work have either made it possible for us to develop FElt or have contributed to making FElt a more functional and powerful system.

- Everyone who has ever worked on the Linux, GNU, X11, and XFree86 projects. We worked almost exclusively under Linux using gcc as a compiler. The X11 project provided a powerful and flexible graphical environment and the folks at XFree86 made it possible for us to use X11 on our Linux boxes.
- Barry Joe developed the Geompack code for triangular mesh generation that we used in earlier versions of the program. The new triangular mesh generator is Triangle by Jonathan Shewchuk.
- Some of the ideas for 3d structure plots are based on the way *gnuplot* (by Thomas Williams and Colin Kelley) does it.
- The code to generate PostScript graphics files is based on the code from *xmgr* by Paul J. Turner. The basic look of time-displacement plots is also based on the way that *xmgr* would have drawn them because we've always liked the way results from *xmgr* looked.
- XWD dumps are produced using the same code as in the actual *xwd* application. The man page says it was authored by Tony Della Fera and William F. Wyatt.
- Encapsulated PostScript image files are created using code from *pnmtops* which is part of Jef Poskanzer's fabulous PBMPLUS image format toolkit.
- The bivariate interpolation routines are hand translations into C of Fortran code originally written by Hiroshi Akima. The Fortran version is readily available as one of the ACM-TOMS algorithms.
- The routines to do Gibbs-Poole-Stockmeyer/Gibbs-King node renumbering are also hand translations of Fortran code that was originally published in ACM-TOMS.

Chapter 1

Introduction to the FElt System

1.1 Intentions

FElt is a package for introductory level Finite ELemenT analysis. It is centered around a mathematical engine designed to simply, effectively, accurately, and flexibly solve most types of structural / mechanical problems that would be encountered in an introductory course in finite element analysis. It was developed in an overzealous fit of “we can do better than that” based on some antiquated Fortran code that we had been using for just such a course.

Our intention was to design a system capable of doing everything that code could do in the mathematical sense, but bring it into the 90’s (or at least the late 80’s) in terms of input syntax (including error checking, problem debugging, etc.), flexibility, and most importantly, a graphical user interface. We hope we’ve succeeded, or at the least, are on the right track; FElt is a work-in-progress and chances are that if a feature isn’t there now, there are at least pipe dreams of it somewhere in the backs of our minds.

Of course all our good intentions are for nought if no one uses FElt. If you find something wrong, let us know (our email addresses are in the legal notice). Alternatively (and an even better option) you can subscribe to the FElt mailing list by sending a one line email message that says “subscribe felt-l” (without the quotes) to listserv@mecheng.fullfeed.com. If something is not there that you think should be, let us know. We make no promises, but we do try to respond and gear development toward user feedback. If you’re happily using FElt and have no complaints, let us know that too. At least then we know that we’re on the right track.

1.2 FElt: What it can do for you

FElt currently has support for the following element types: three-dimensional truss or bar, one-dimensional spring, two- and three-dimensional Euler beam, two-dimensional Timoshenko beam, constant strain triangles (CSTs) for both plane stress and plane strain analysis, planar isoparametric elements (four to nine node and a separate type for simple four node quadrilaterals), again for both plane stress or plane strain analysis, a linear axisymmetric triangular element, an HTK plate bending element, an isoparametric eight-node brick element, and a rod and a constant temperature gradient triangle for thermal analysis. FElt allows for an arbitrary mixing of element types within a problem. The syntax for the FElt input file is based on a high-level grammar which frees you from comma-delimited lists of numbers and hours of debugging due to not having the right number after the right comma; both the parser and the mathematical routines do extensive error checking and report what we hope are informative and useful error messages.

FElt offers several user interface options. The basic *felt* application gives you the capability to define an input file in a favorite editor and then solve the problem from the shell command line. Graphics for the *felt* applications can be handled by any number of graphing packages. Under Windows, *WinFElt* provides a text editor and encapsulator environment with some post-processing capability. *velvet* is the full-featured graphical user interface to the FElt system. *velvet* (which is smoother than *felt*) knows most of what there is to know about the FElt system and provides a consistent, CAD-like interface for drawing, defining, solving, and visualizing everything about a problem. Though it's probably not always the best way to set-up a problem (it's hard to beat *vi* for quick-and-dirty problems), a user working strictly with two-dimensional problems (three-dimensional graphics are only partially supported in *velvet*) need never actually see the internals of a FElt file or run the *felt* application from the shell command line; *velvet* provides access to all of the two-dimensional functionality of FElt in one completely stand-alone application. *velvet's* current post-processing capabilities include plotting the displaced shape, two-dimensional color contours of stress and displacement for planar elements, animation of dynamic structural simulations, line plots of for time and frequency domain results, and graphical presentation of mode shapes.

Automated element generation for a FElt problem is provided for both simple grids of line, quadrilateral and brick elements and arbitrary meshes of triangular planar elements. The latter capability is derived from J.R. Shewchuk's Triangle mesh generation routine. You can interface this functionality either graphically through *velvet* or through a separate command line application called *corduroy* that has its own input file syntax much like the regular syntax for FElt problems.

Support applications are provided for file format conversion and unit conversion and problem scaling. *patchwork* can translate between the standard FElt syntax and several other common graphical description formats. *yardstick* can be used to scale numerical quantities within a FElt file, including special options for conversion between different types of units.

FElt should be able to handle most types of linear static and dynamic problems that you throw at it, but there are no guarantees. Most elements allow arbitrary oriented distributed line loads. Displacement (e.g., settlement of support) and force (e.g., nodal hinge) boundary conditions are also allowed. Time varying force and boundary conditions can be expressed either as continuous or discrete functions.

1.3 FElt**: What it cannot do for you**

As of this release, FElt can only handle linear static and dynamic problems. We realize the shortcomings that this presents for some people and we have some vague plans for non-linear analysis, but nothing is here yet. *velvet* can't really draw in 3-d (at least in the problem definition stage) and thus isn't a terribly good way to define 3-d problems; it will always assume that it should work in the x-y plane (z = 0). This is probably going to stay this way for a long time.

There are certainly other shortcoming as well, depending on just what you would like the package to do. What it really comes down to is that FElt was never intended to solve everybody's real-world or cutting-edge research problems, so we're probably never going to incorporate lots of different analysis types, etc. If you want to take a crack at modifying FElt for your own local needs, however, then we encourage you to do so; we'll even help out where possible.

Chapter 2

FElt **Analysis Types**

2.1 Introduction

Our intent in writing this chapter was to give some basic details on the kinds of analyses that are intrinsic to FElt. It is not at all meant to be an introduction to the finite element method, but rather it is meant to provide a background on what types of physical and mathematical models we are talking about when we talk about the various analysis types. There is of course some very basic information on just how these models work in the context of FEA, but we really do suggest that you try some of the excellent textbooks ([9, 16, 13, 1, 14]) that are out there if you want any sort of real background information.

2.2 Static structural analysis

The basic equation for static structural analysis can be seen as a generalization of Hooke's law for the deformation of a linear spring, $f = kx$. For a spring, if we know the applied force and the spring stiffness, k, then we can find the deformation as $x = f/k$. We can generalize this to multiple degrees of freedom by considering the static equilibrium of the general spring in figure 2.1. If the spring is in static equilibrium then at point a we must have

$$(x_a - x_b)\,k = F_a, \tag{2.1}$$

and at point b

$$(x_b - x_a)\,k = F_b. \tag{2.2}$$

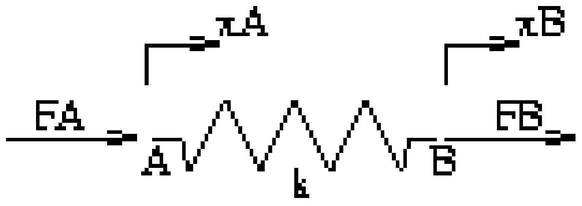


Figure 2.1: Static equilibrium of a general linear spring.

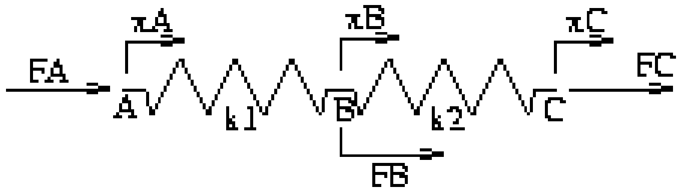


Figure 2.2: Static equilibrium of two springs in series.

These two conditions form a linear system of equations in two unknowns. We can rewrite this system in matrix notation as

$$\begin{bmatrix} k & -k \\ -k & k \end{bmatrix} \begin{bmatrix} x_a \\ x_b \end{bmatrix} = \begin{bmatrix} F_a \\ F_b \end{bmatrix}. \tag{2.3}$$

The problem is still essentially the same as our simple Hooke's law calculation – we know the applied force and the structural stiffness, and we want to solve for the deformations. Now, however, our applied force is a vector, our structural stiffness is a matrix and to solve for the deformations we must now solve a linear system of equations.

The *stiffness matrix* on the left hand side of equation 2.3 is just the element stiffness matrix in the finite element method. When we want to solve a problem of static equilibrium for a structure that is more complex than our simple spring all we have to do is stick a bunch of springs together and then linearly superpose the contributions from each element's stiffness matrix into our *global* stiffness matrix. Consider the case of two springs in series as in figure 2.2. Each individual spring has a stiffness matrix like the one on the left hand side of equation 2.3. The system now has three degrees of freedom (x_a, x_b, x_c) and thus we know that our global stiffness matrix will be 3×3. We assemble the global stiffness matrix (construct the superposition of all the element stiffness matrices) by considering which degrees of freedom each spring affects – spring 1 affects the deformations x_a and x_b; spring 2 affects x_b and x_c. We know then that the stiffness at b in our global stiffness matrix will have contributions from both spring 1 and spring 2. The global stiffness matrix is

$$K = \begin{bmatrix} k_1 & -k_1 & 0 \\ -k_1 & k_1 + k_2 & -k_2 \\ 0 & -k_2 & k_2 \end{bmatrix}. \tag{2.4}$$

The two boxes indicate exactly how the individual element stiffness matrices were placed into the global stiffness matrix. Our equation for static equilibrium is

$$\begin{bmatrix} k_1 & -k_1 & 0 \\ -k_1 & k_1 + k_2 & -k_2 \\ 0 & -k_2 & k_2 \end{bmatrix} \begin{bmatrix} x_a \\ x_b \\ x_c \end{bmatrix} = \begin{bmatrix} F_a \\ F_b \\ F_c \end{bmatrix}. \tag{2.5}$$

Of course, one-dimensional springs are not the most useful element in terms of modeling complex structural behavior. The concepts of static equilibrium, superposition, and assembly, however, are identical no matter what types of elements we are using. All that changes the is nature of the element stiffness matrices – rather than simple linear springs we take into account bending and torsional stiffness, three-dimensional solid deformations, plate bending motion, etc. In structural analysis there are only six possible degrees of freedom (translations along and rotations about the x, y, and, z axes) and thus if we know how each element affects each of these degrees of freedom we can even mix different element types into the same global stiffness matrix. Given a global stiffness matrix, K, which represents the contributions from an arbitrary number of individual elements and a vector, F, of the force at each global degree of freedom, then the general form of equation 2.5 is simply

$$Kx = F \tag{2.6}$$

where x is a vector of the displacements at each global degree of freedom which we solve for using matrix techniques for linear systems of equations. Note the direct analogy between this matrix equation and the simple linear spring relationship, $f = kx$, cited earlier.

2.3 Transient structural analysis

Just like static structural analysis is an extension of simple static equilibrium for a spring, we can draw an analogy between dynamic structural analysis and a simple spring-mass-dashpot oscillator. For the system shown in figure 2.3, a sum of forces at the mass and Newton's law gives

$$m\ddot{x} + c\dot{x} + kx = f(t) \tag{2.7}$$

where the overdot indicates differentiation in time. To generalize this to multiple degrees of freedom we take the same steps as in the static case, recognizing that we can assemble

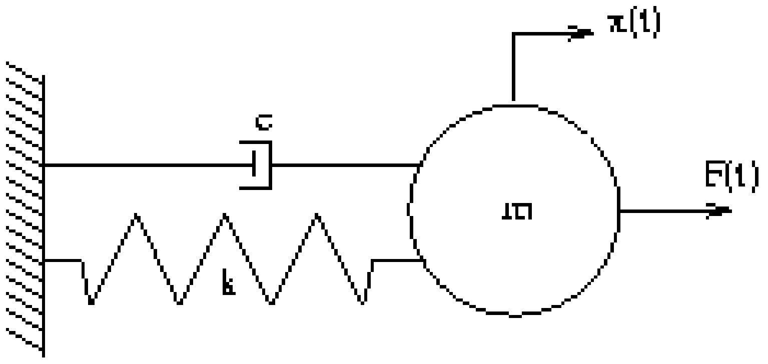


Figure 2.3: Dynamic equilibrium of a spring-mass-dashpot system.

global mass and damping matrices from elemental constructs just as we did for the stiffness matrix in the static case. In that case, the matrix equation of motion becomes

$$M\ddot{x} + C\dot{x} + Kx = F(t) \tag{2.8}$$

where M, C, and K are now global mass, damping and stiffness matrices, x is a vector of displacements at each DOF just as before and $F(t)$ is a time-dependent vector of forces at each degree of freedom.

The damping matrix is the most difficult quantity to estimate in equation 2.8; both the stiffness and mass matrices are relatively simple to derive for a wide variety of elements (see chapter 4). There is no explicit way to specify dashpot constants in FElt. Instead, damping is based on a Rayleigh model whereby the damping matrix is simply a linear combination of the mass and stiffness matrices

$$C = R_m M + R_k K \tag{2.9}$$

where R_m and R_k are user specified constants of proportionality.

2.4 Static thermal analysis

2.5 Transient thermal analysis

2.6 Modal analysis

There are two basic levels to modal analysis. The first is the eigenvalue problem which determines the natural frequencies and mode shapes of our structure in free ($F(t) = 0$), undamped ($C = [0]$) vibration. If we assume a solution to the undamped, unforced form of equation 2.8 of the form

$$x(t) = ue^{i\omega t} \tag{2.10}$$

(where u is a vector of displacement amplitudes) and substitute this into the equation of motion, we find that

$$\left[K - M\omega^2\right] u = 0. \tag{2.11}$$

Linear algebra tells us that this has a non-trivial solution only if

$$\det\left[K - M\omega^2\right] = 0. \tag{2.12}$$

Evaluating the determinant leads to a polynomial of order n (where n is the number of DOF in the problem) in ω^2. The roots of this polynomial give us the free vibration natural frequencies, ω_i^2, $i = 1\ldots n$. If we substitute each of these ω_i into equation 2.11 we can solve for n mode shape vectors, $u^{(i)}$.

By itself, natural frequency and mode shape information can be very useful; the natural frequencies are an immediate indication of where the resonances for this system will be (even when we put damping back into the system). The second level of modal analysis, however, uses the fact that the eigenvectors (mode shapes) form an orthogonal basis set to decouple an arbitrarily complex multiple degree of freedom system into n single degree of freedom systems. If we form a matrix, U, of all the eigenvectors, $u^{(i)}$, then we can compute the modal mass and stiffness matrices as

$$\hat{M} = U^{\mathrm{T}} M U \tag{2.13}$$
$$\hat{K} = U^{\mathrm{T}} K U. \tag{2.14}$$

The modal matrices have the remarkable property that they are diagonal and thus if we form a transformed system of coordinates,

$$q = U^{-1} x, \tag{2.15}$$

and a transformed force vector

$$Q = U^{\mathrm{T}} F, \tag{2.16}$$

we can write our matrix equation of motion as n uncoupled single degree of freedom equations

$$\hat{M}_i \ddot{q}_i + \hat{K}_i q_i = Q_i, \qquad i = 1\ldots n. \tag{2.17}$$

We say that $\hat{M}_i$ and $\hat{K}_i$ are the modal mass and stiffness in the ith mode. They are simple the ith entries along the diagonals of the modal mass and stiffness matrices. We can further simplify the resulting equations of motion by making use of the fact that the eigenvectors can be arbitrarily normalized (they are only fixed to within an arbitary multiplicative constant by equation 2.11). If we choose an appropriate normalization then we can fix it so that $\hat{M} = I$, the identity matrix. The appropriately normalized mode shapes are called orthonormal modes.

Because FElt uses a Rayleigh damping model, we can also construct a diagonal damping matrix,

$$\hat{C} = U^{\mathrm{T}} C U = R_m \hat{M} + R_k \hat{K}. \tag{2.18}$$

Also, because the motion in each mode is now just like a single degree of freedom motion, we can use concepts from the theory of single degree of freedom oscillators to help us choose R_k and R_m. The damping ratio in the ith mode is simply

$$\zeta_i = \frac{\hat{C}_i}{2\hat{M}_i \omega_i} \tag{2.19}$$

We can fix the damping ratio in two modes, i and j, simply by substituting

$$\hat{C}_i = R_m \hat{M}_i + R_k \hat{K}_i, \tag{2.20}$$

$$\hat{C}_j = R_m \hat{M}_j + R_k \hat{K}_j, \tag{2.21}$$

into equation 2.19 and solving the resulting linear system for R_m and R_k

$$\begin{bmatrix} \frac{1}{\omega_i} & \omega_i \\ \frac{1}{\omega_j} & \omega_j \end{bmatrix} \begin{bmatrix} R_m \\ R_k \end{bmatrix} = \begin{bmatrix} 2\zeta_i \\ 2\zeta_j \end{bmatrix}. \tag{2.22}$$

After solving the n uncoupled single degree of freedom equations (either as initial value problems or steady state oscillation problems) for the individual q_i, we can form the solution in physical coordinates as the superposition of the motion in each mode

$$x(t) = \sum_{i=1}^{n} u^{(i)} q_i(t) = Uq. \tag{2.23}$$

2.7 Spectral analysis

Spectral analysis is intended to give us information about the response of our structural system in the frequency domain. In a way, we can think of it as a direct way to calculate the results that we would get from estimating a power spectrum of our time domain results using a fast fourier transform. If we assume that the force vector in equation 2.8 is harmonic in time and of the form,

$$F(t) = F_0 e^{i\omega t} \tag{2.24}$$

then we can also assume the solution, $x(t)$, is of the form[1]

$$x(t) = \hat{x}_0 \mathrm{e}^{i\omega t}. \tag{2.25}$$

Note that $\hat{x}_0$ is a complex constant which incorporates both the magnitude and phase of the output motion

$$\hat{x}_0 = x_0 \mathrm{e}^{-i\phi}. \tag{2.26}$$

If we differentiate, substitute into equation 2.8, and divide out the time dependent harmonic term then we find that

$$F_0 = \left[-\omega^2 M + i\omega C + K\right] \hat{x}_0. \tag{2.27}$$

If this were a single degree of freedom system we would recognize the term in square brackets on the right hand side as the inverse of the transfer function. In a multiple degree of freedom system, this term is known as the impedance matrix,

$$Z = -\omega^2 M + i\omega C + K, \tag{2.28}$$

and its inverse is equivalent to a matrix of single degree of freedom transfer functions

$$H = Z^{-1}. \tag{2.29}$$

In essence, the transfer function matrix predicts the amount of response per unit force at frequency ω.

Given entries from the transfer function matrix $H_{ij}(\omega)$, we can calculate the output spectrum at DOF i due to system inputs as

$$S_i^{(\mathrm{out})} = \sum_{k=1}^{n} |H_{ik}(\omega)|^2 S_k^{(in)}, \tag{2.30}$$

where the input spectra $S_k^{(\mathrm{in})}$ are simply the power spectra of the applied forces.

2.8 Nonlinear static analysis

2.9 Nonlinear dynamic analysis

[1]It is a property of linear systems that the output will always be at the same frequency as the input.

Chapter 3

Structure of a FElt Problem

3.1 Input file syntax

A FElt problem is defined by an input file which you must create using a text-editor such as *vi* or from within *velvet* (though in this case you do not actually need to see the resulting file). The input file contains a complete description of everything that defines a problem: the nodes, the elements, analysis parameters, the constraints and forces on the nodes, and the distributed loads and material properties of the elements. An informal description of the file format and these objects is given below. For a complete, formal definition of the syntax, you should refer to the *felt*(4fe) manual page.

3.1.1 General rules

The input file for a typical FElt problem will consist of nine sections. In general, each object within a problem (`node`, `element`, `force`, `material`, `distributed load`, `constraint`) has a symbolic name. For nodes and elements these names are positive integers; for everything else the name is a user assigned string, e.g., “steel”, “point_load”, “roller”. Each type of object is defined within its own section, each section containing a list of definitions for objects of that type.

As a rule of thumb white space can occur anywhere and the definition sections can be given in any order (and repeated even). The exceptions to this are the `problem description` section and the `end` statement, which must come first and last in an input file, respectively, and cannot be repeated.

Comments are denoted as in the C programming language; anything between `/*` and

`*/` will be ignored as a comment no matter where it appears in the file.

3.1.2 Expressions

3.1.2.1 Continuous functions

As a convenience, wherever a numeric value is required for a material characteristic, magnitude of a force, load or displacement boundary condition, or a nodal coordinate, you can specify an arbitrary mathematical expression, including the operators `+, -, *, /, %` and the standard mathematical library functions *sin*, *cos*, *tan*, *sqrt*, *hypot*, *pow*, *exp*, *log*, *log10*, *floor*, *ceil*, *fabs* and *fmod*. Note that arguments to the trigonometric functions should be given in terms of radians just as if you were calling them from a C program using the standard math library. Other than this difference, these functions should be used and should behave as they are described in the manual pages for your local mathematics library.

In a transient analysis problem the symbol `t` denotes the time variable in expressions for force magnitudes or time-varying boundary conditions. These expressions will be dynamically evaulated throughout the course of the simulation. For other parameters (loads, nodal coordinates, etc.) and for all parameters in a static problems, these expressions will simply be evaluated as if `t=0`. For spectral inputs, the symbol `w` can be substituted for the independent variable for clarity and to distinguish frequency domain force spectra from time domain forces.

Expressions can also contain the ternary conditional operator as in the C programming language: “if a then b else c” is symbolized in a FElt input file as `a ? b : c` where `a`, `b`, and `c` are all valid expressions. The logical operators to use in constructing `a` are the same as those in C (`==, &&, ||, <=, <, >, >=, !=`). The conditional construct is particularly useful in defining things like discontinuous dynamic forces.

3.1.2.2 Discrete functions

Because some forcing functions are easier to express in a discretized (as opposed to continuous) form (e.g., earthquake records), the FElt syntax also includes a mechanism for specifying a discrete representation of a transient forcing function. The basic specification consists of a series of time magnitude pairs of the form `(t, F)` where `F` is the value of the function at time `t`. In evaluating the function, FElt will linearly interpolate between each adjacent pair for times that fall between two pairs. A single time magnitude pair will be interpreted as an impulse response function at the given time. Note that the pairs must be given in order of increasing time.

You can express a periodic discrete function simply by defining one period and then entering a + symbol at the end of the expression. You could represent a simple sawtooth forcing function having a period of 2 seconds and a maximum magnitude of 1000 with an expression like

```
forces
sawtooth Fx=(0,0.0) (2,1000.0)+
```

Figure 3.1 illustrates different ways to define some common types of loading functions with either continuous or discrete representations.

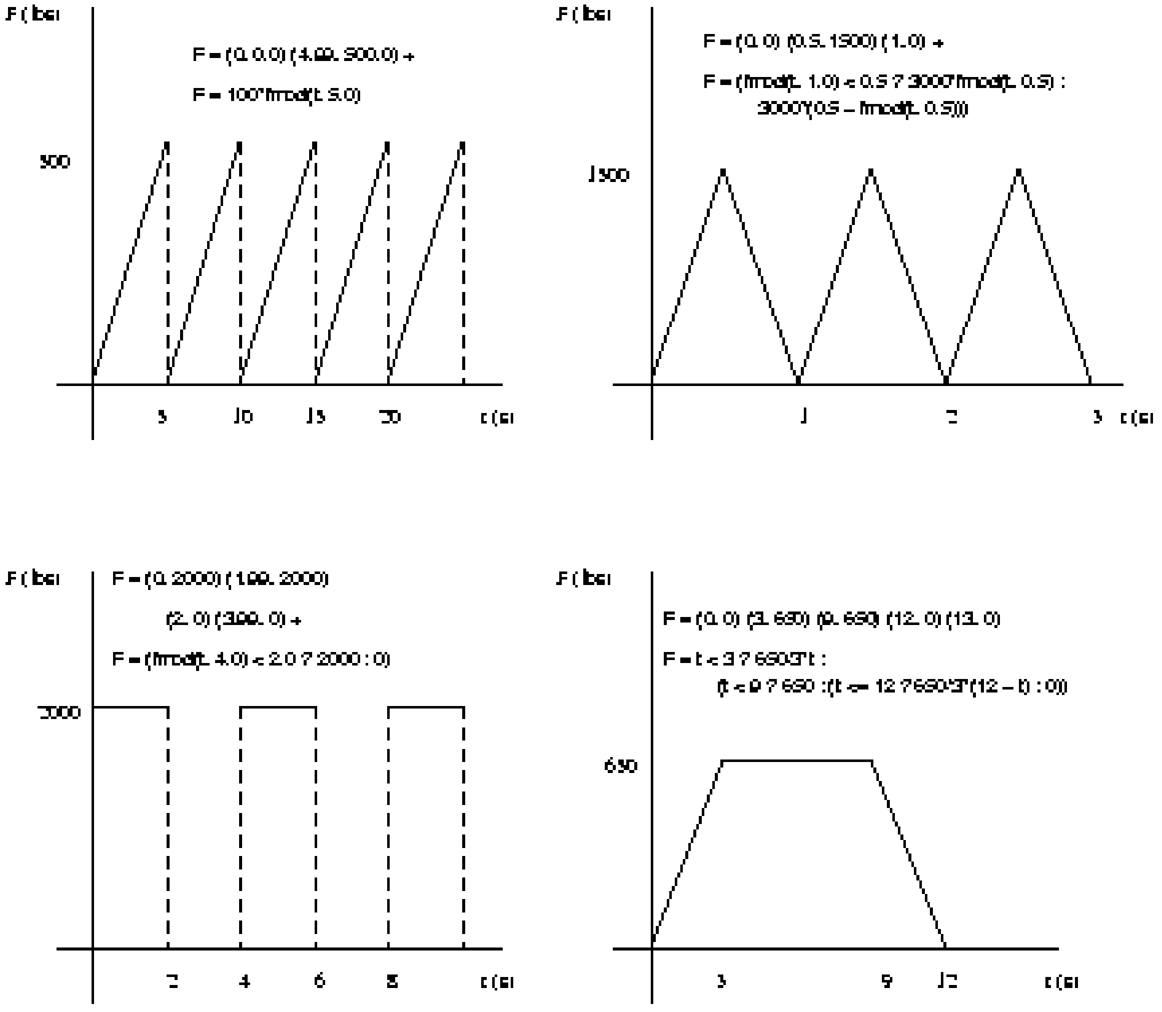


Figure 3.1: Example loading functions

3.1.3 Units

There are no set units for the dimensional quantities that you specify in defining a problem for FElt. The important thing is to remain consistent in the units that you use; numerical results will then be consistent with the input dimensions. Some examples of consistent units would be nodal coordinates in meters, forces in Newtons, elastic moduli in Pascals (N m^{-2}); moments of inertia would be in m^{-4}. The displacement results for a problem like this would be in meters; stresses would be in Pascals. Convenient English units are often pounds, inches, and psi (pounds per square inch) or kips (kilopounds), inches, and ksi.

The FElt application, *yardstick*, is intended for simple scaling of the numerical quantities in a FElt input file. As a special case of this, unit conversion of files is given special treatment and made particularly easy. To convert an input file that was originally specified in kips and feet to Newtons and meters, the *yardstick* command line would simply look like

```
% yardstick -if kips -il feet -of N -ol m foo.flt > foo_si.flt
```

The *yardstick* manual page details the list of units that the program recognizes and all of the available options.

3.1.4 A simple example

An input file for a simple cantilever beam problem with an end point-load and considering self-weight might look something like this:

```
problem description
title="Cantilever Beam Sample Problem" nodes=2 elements=1 analysis=static

nodes
1  x=0.0 y=0.0 constraint=fixed
2 x=10.0 y=0.0 constraint=free force=end_load

beam elements
1  nodes=[1,2] material=steel load=self_weight

material properties
steel A=10.0 E=30e6 Ix=357 /* properties can also be lowercase */

distributed loads
self_weight direction=perpendicular values=(1,2000) (2,2000)
```

```
constraints
fixed Tx=C Ty=C Rz=C /* column alignment is unimportant */
free Tx=U Ty=U Rz=U /* I could have used tx, ty and rz */

forces
end_load Fy=-1000

end
```

3.2 Sections of a FElt input file

3.2.1 Problem description

The `problem description` section is used to define the problem title and the number of nodes and elements in the problem. These numbers will be used for error checking so the specifications given here must match the actual number of nodes and elements given in the definition sections. Note that the definitions for nodes and elements do not have to be given in numerical order, as long as nodes 1 ... *m* and elements 1 ... *n* (where *m* is the number of nodes and *n* is the number of elements) all get defined in one of the element and node definition sections in the file. The `analysis=` statement defines the type of problem that you wish to solve. Currently it can either be `static`, `transient`, `static-substitution`, `modal`, `static-thermal`, `transient-thermal`, or `spectral`. If you do not specify anything, static analysis will be assumed. The `problem description` section is the only section which you cannot repeat within a given input file.

3.2.2 Nodes

The `nodes` section(s) must define all of the nodes given in the problem. Each node must be located with an x, y, and z coordinate using `x=`, `y=`, and `z=` assignments. Coordinates are taken as 0.0 if they are not otherwise defined. If a coordinate is left unspecified for a given node, it takes the value for that coordinate from the previous node. A node must also have a constraint assigned to it by a `constraint=` statement. The default constraint leaves the node completely free in all six degrees of freedom. Like coordinates, if a constraint is left unspecified, the node will be assigned the same constraint as the previous node. Forces (applied point loads and moments) on a node are optional and are applied using the `force=` statement; if a force is not specified there will be no force applied to that node. You can also specify an optional lumped mass at a given node with a `mass=` statement.

3.2.3 Elements

An element definition section begins with the keywords `xxxxx elements` where `xxxxx` is the symbolic name of a type of element. All elements under this section heading will be taken to be the given element type. There could be multiple sections that defined beam elements, but each must begin with the keywords `beam elements`. If there were truss elements in the same problem, they would have to be defined in sections which began with the keywords `truss elements`. Currently available types are `spring`, `truss`, `beam`, `beam3d`, `timoshenko`, `CSTPlaneStrain`, `CSTPlaneStress`, `iso2d_PlaneStrain`, `iso2d_PlaneStress`, `quad_PlaneStrain`, `quad_PlaneStress`, `htk`, `brick`, `rod`, and `ctg`. A FElt problem is not limited to one element type; the routines for assembling the global stiffness matrix takes care of getting the right parts of the right element stiffness matrices into the global matrix.

Each element must have a list of nodes to which it is attached. The node list is defined with the `nodes=[ ...]` statement. The length of the list inside the square brackets varies with element type, but must always contain the full number of nodes which the element type definition requires (see chapter 4 for a complete definition of what each type requires). A material property must also be assigned to every element with a `material=` statement. If the material is never specified, an element will take the same material property as the previous element. If nothing ever gets assigned to an element, the default material property will have zeros for all of its characteristics. Chances are this is not what you want. Finally, an element can have up to three optional distributed loads. Each load is assigned with a separate `load=` assignment. Each element type may treat a distributed load differently so you should be careful that the name given for the loads on a given element match the names of distributed loads which are defined in a manner conformant with what that element type is expecting.

3.2.4 Material properties

The `material properties` section(s) is quite simple. Each material has a name followed by a list of characteristics. Currently available characteristics are listed in table 3.1.

Note that no element types require a material property with every one of these characteristics defined. In fact, most only use three or four characteristics. You should consult the element definitions (chapter 4) for a complete list of what each element type requires from a material property. Density (`rho`) is always necessary for the materials in a transient, modal, or spectral analysis problem if you actually want your elements to have any inertia. Different element types can certainly use the same material as long as that material

property name	description
`E`	Young's modulus or elastic modulus
`A`	cross-sectional area
`t`	thickness
`rho`	density
`nu`	Poisson's ratio
`G`	bulk or shear modulus
`J`	torsional stiffness
`Ix`	I_{xx} moment of inertia
`Iy`	I_{yy} moment of inertia
`Iz`	I_{zz} moment of inertia
`kappa`	shear force correction factor
`Rk`	Rayleigh stiffness damping coefficient
`Rm`	Rayleigh mass damping coefficient
`Kx`	thermal conductivity along the x-direction
`Ky`	thermal conductivity along the y direction
`Kz`	thermal conductivity along the y direction
`c`	heat capacitance

Table 3.1: Symbolic names used to define material properties

definition contains the right characteristics for each type of element that uses it.

3.2.5 Constraints

The `constraints` section(s) must define all of the named constraints within the problem. Each constraint is defined by a name followed by a list of DOF specifications of the form `Tx=? Ty=? Tz=? Rx=? Ry=? Rz=?` where the `?` can either be `c` for constrained, `u` for unconstrained or a valid, possibly time-dependent, expression for cases where a displacement boundary condition is required (e.g., settlement of support, or time-varying temperature along a boundary in a transient thermal analysis problem). Note that specifiying `Tx=c` is equivalent to `Tx=0.0`. The `T` and `R` refer to translation and rotation, respectively and the subscripted axis letter indicates that the specification refers to translation along, or rotation about, that axis. An additional specification of `h` or `hinged` is allowed for the rotational DOFs for cases where it is necessary to model a nodal (momentless) hinge. Note that a hinge specification currently has meaning only on `beam`, `beam3d` and `timoshenko` elements.

If a specification is never made for a DOF, then the problem is assumed to be unconstrained in that degree of freedom. Getting the constraints right is an important part of getting a reasonable solution out of a finite element problem so you should be aware of what DOF are active in a given problem (this will depend on which types of elements are being used ... even in a 2-d problem, the global stiffness matrix will take displacement in the z direction into account if there are 3-d elements in use, consequently, those displacements should be constrained).

In order to account for initial conditions in transient analysis problems, a constraint specification may also include the initial (at time $t = 0$) displacements, and velocity and acceleration in the translational DOFs. Initial displacements are given with `ITx=`, `IRz=`, etc. You can specify initial accelerations with `Ax=`, `Ay=`, and `Az=`. Initial velocities are given by `Vx=`, `Vy=`, and `Vz=`. Unspecified velocities and displacements will be taken as 0.0. If there are no initial accelerations specified (i.e., none of the nodes have a constraint with an acceleration assigned) then the initial acceleration vector will be solved for by the mathematical routines based on the initial force and velocity vectors. If any of the nodes have a constraint which has an acceleration component defined (even if that component is assigned to 0.0) then the mathematical routines will not solve for an initial acceleration vector; they will build one based on the constraint information, assigning 0.0 to any component that was not specified. What this means is that you cannot specify the initial acceleration for only a few nodes and expect the mathematical routines to simply solve for the rest of them. If you specify any of the initial accelerations then you are effectively specifying all of them.

3.2.6 Forces

The `forces` section(s) defines all of the point loads used in the problem and actually looks a lot like the `constraints` section. The magnitudes of the forces in the six directions are specified by `Fx=?  Fy=?  Fz=?  Mx=?  My=?  Mz=?`. If the value for a given force component is not given it is assumed to be zero. The directions for both forces and the constraints as defined above should be given in the global coordinate system (right-hand Cartesian).

If you are doing transient analysis, the force definitions can be more complicated than a simple numerical assignment or expression. FElt allows you to specify a transient force as either a series of time-magnitude pairs (a discrete function in time) or as an actual continuous function of time. This latter fact means that you can define a force as `Fx=sin(t)` rather than having to discretize the sine function. These continuous forcing functions are a special case of expressions as discussed above.

For spectral analysis problems, you can explicitly specify input spectra using `Sfx=`, `Smy=`, etc. if you want to to compute the actual power spectrum of the output. These

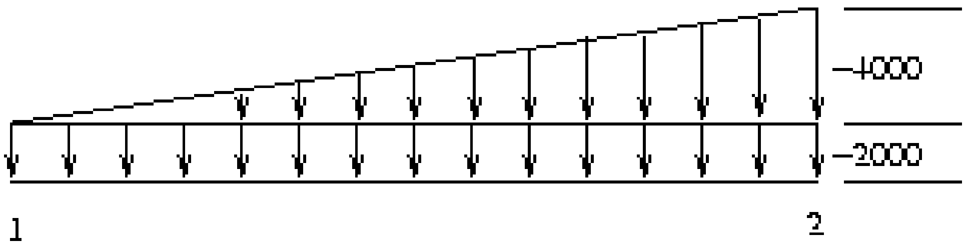


Figure 3.2: An example of a complex distributed load.

spectra can be analytic functions of `w` or discrete frequency, power pairs.

3.2.7 Distributed loads

The `distributed loads` section(s) must contain a definition for each distributed load that was assigned in the element definition section(s). A valid definition for a distributed load is a symbolic name followed by the keywords `direction=xxx` and `values=(n,x) (m,y) ...`. The direction assignment must be set to one of `parallel, perpendicular, LocalX, LocalY, LocalZ, GlobalX, GlobalY, GlobalZ, radial, axial.` The valid directions for a given element type, and what those directions refer to for that element type are described in the individual element descriptions in chapter 4. The values assignment is used to assign a list of load pairs to the named load. A load pair is given in the form `(n, x)` where `n` is the local node number to which the magnitude given by `x` applies. Generally, two pairs will be required after the `values=` token. The imaginary line between the two magnitudes at the two nodes defines an arbitrarily sloping linearly distributed load. This allows you to specify many common load shapes: a constant distributed load of magnitude y, including cases of self-weight, a load which slopes from zero at one node to x at a second node, or a linear superposition of these two cases in which the load has magnitude y at node 1 and magnitude $x+y$ at node 2. This latter case is illustrated in figure 3.2. The definition of this load would be

```
distributed loads
load_case_1    direction=perpendicular    values=(1,-2000) (2,-6000)
```

In the future, higher-order load shapes may be supported by some elements and thus require the specification of more than two load pairs.

3.2.8 Analysis parameters

The `analysis parameters` section is required only if you are doing some type of transient, modal, or spectral analysis (e.g., `analyis=transient`, `analysis=spectral` in the `problem description` section). For modal analysis it is simply used to set the type of element mass matrices that will be formed, but for transient and spectral analyses it contains information that further defines the problem and the parameters for the numerical integration in time. The variables that you can define in this section include: `start=` for the start of the frequency range of interest in spectral analyis; `stop=` for the end of the time (transient analysis) or frequency (spectral analysis) range of interest (`duration=` is an alias for `stop=`); `step=` for the time or frequency step to be used between the start and stop points (`dt=` is an alias for `step=`); `beta=`, `gamma=`, and `alpha=` for integration parameters in the structural and thermal dynamic integration schemes; `mass-mode=` for the types of element mass matrices that should be formed, either `lumped` or `consistent` (note again that this is the only assignment that is required in this section for a modal analysis problem); `nodes=[ ... ]` defines a list of nodes which you are interested in seeing output for; `dofs=[ ... ]` defines the list of local DOF that you are interested in for the nodes that you are interested in. The list of nodes should just be a comma delimited list of node numbers. The list of DOF should be a list of symbolic DOF names (`Tx, Ty, Tz, Rx, Ry, Rz`). You will get solution output for each of these DOF at each of the nodes that you specified in the node list. Finally, you can specify global Rayleigh damping parameters with `Rk=` and `Rm=`. If either of these parameters is non-zero then the global damping matrix will be formed using these two parameters and the global mass and stiffness matrices as opposed to being formed from elemental Rayleigh parameters and elemental mass and stiffness matrices.

You should refer to the chapter on algorithmic details (section 12.4.7) for a little more insight on the meanings of the integration parameters. The individual element descriptions in chapter 4 discuss the different mass matrices for that element. The analysis parameters and their meaning are summarized in table 3.2.

3.2.9 Load cases

3.3 An illustrated example

Figure 3.3 illustrates a slightly more complicated problem, a simply supported beam with two triangular loads applied and a spring support at the center. We can model the spring as a slender truss element such that $EA/L = k$, the spring stiffness.

The input file for this problem would probably look something like this.

parameter	description	example	r/o
start	frequency range start (SP), load range start (LR)	0.0	r
stop	time- (T, TT), frequency- (SP) or load- (LR) range end	10.0	r
step	time- (T, TT), frequency- (SP) or load- (LR) step	0.05	r
alpha	α in HHT-α and transient thermal integration (T, TT)	0.5	o
gamma	γ in HHT-α integration (T)	0.25	o
beta	β in HHT-α integration (T)	0.5	o
mass-mode	element mass matrices to use (T, TT, SP, M)	`lumped`	r
nodes	list of nodes for which you want results (T, TT, SP, LR, LC)	[1,4,5]	r
dofs	list of DOF at each result node (T, TT, SP, LR, LC)	[Tx, Rz]	r
Rk	global Rayleigh damping constant for stiffness (T, M, SP)	3.0	o
Rm	global Rayleigh damping constant for mass (T, M, SP)	1.4	o
input-node	node to be parametrically excited (LR)	4	r
input-dof	DOF at the input node to be excited (LR)	Ty	r
tolerance	convergence tolerance (SUB)	1e-3	r
iterations	maximum permitted number of iterations (SUB)	100	r
relaxation	under- (over-) relaxation factor (SUB)	0.8	r
load-steps	number of incremental steps to full load (SUB)	10	r

Table 3.2: Meaning given to the various analysis parameters. T indicates transient analysis; TT is transient-thermal; SP is spectral; M is modal; LR is static analysis (linear or nonlinear) with load ranges; LC is static analysis (linear or nonlinear) with load cases; SUB is non-linear static substitution analysis. The r/o column indicates whether the parameter is (r)equired for the given analyses or if the (o)ptional default value of 0.0 will be used if nothing is specified.

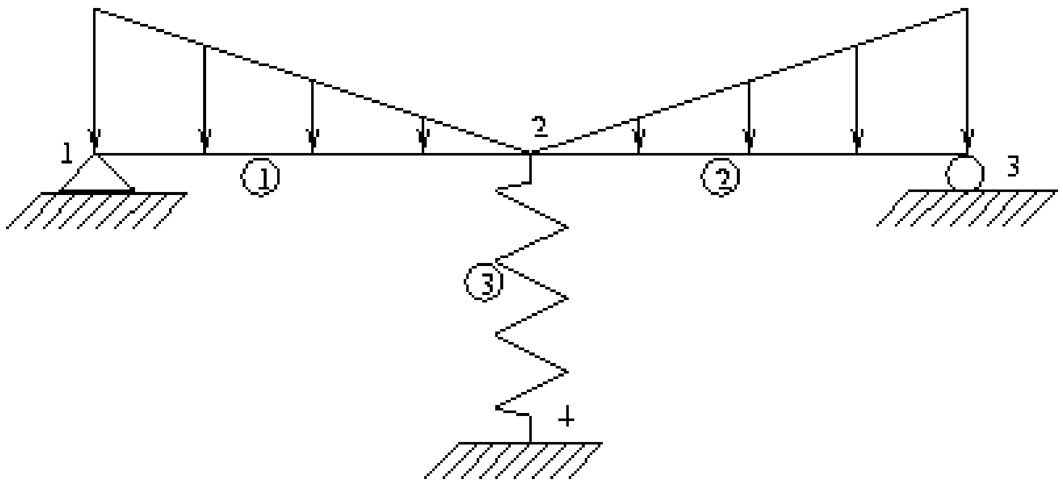


Figure 3.3: A mixed element problem with distributed loads.

```
problem description
title="Mixed Element Sample" nodes=4 elements=3

nodes
1     x=0     y=0     constraint=pin
2     x=6     y=0     constraint=free
3     x=12    y=0     constraint=roller
4     x=6     y=-10   constraint=fixed

beam elements
1 nodes=[1,2]     material=steel     load=left_side
2 nodes=[2,3]                        load=right_side

truss elements
3 nodes=[2,4]     material=spring

material properties
steel    A=1          E=210e9     Ix=4e-4
spring   A=4.76e-7    E=210e9 /* k = 10 kN/m */

constraints
pin     Tx=c Ty=c Tz=c Rz=u /* we better constrain Tz because */
free    Tx=u Ty=u Tz=c Rz=u /* there is a truss element!      */
roller  Tx=u Ty=c Tz=c Rz=u
fixed   Tx=c Ty=c Tz=c Rz=c
```

```
distributed loads
left_side       direction=perpendicular   values=(1,10000) (2,0)
right_side        direction=perpendicular   values=(1,0) (2,10000)

end
```

3.4 An example of a transient analysis problem

Figure 3.4 illustrates a simple frame problem subjected to a time-dependent loading. Our input file for this problem differs from our above example in only one significant way; we must include an `analysis parameters` section in this case. Also we need to remember to define the density of all the materials so we can get a proper mass matrix for each element. In this case we have also calculated what the self-weight effect of this mass will be assuming bays on 15 ft centers (depth of frame into page) and assigned it to each element as a distributed load in terms of pounds per inch.

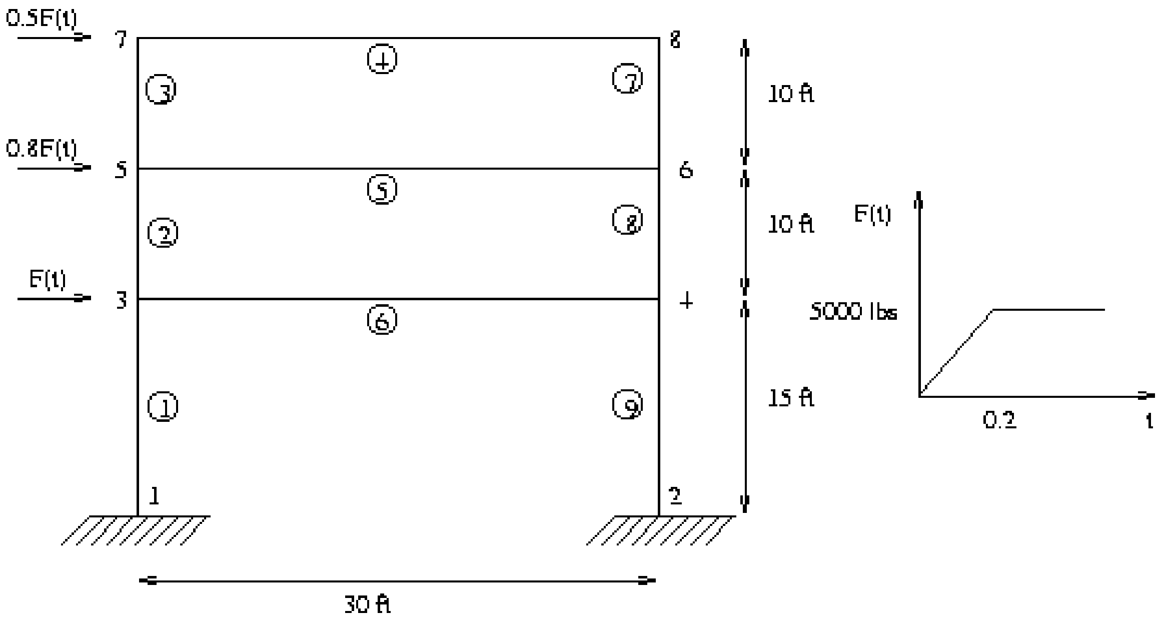


Figure 3.4: Transient analysis of a simple frame structure.

The input file for this case could look something like the following.

```
problem description
title="Dynamic Frame Analysis" nodes=8 elements=9 analysis=transient
```

```
analysis parameters
dt=0.05    duration=0.8
beta=0.25  gamma=0.5  alpha=0.0
nodes=[8]  dofs=[Tx]  mass-mode=lumped

nodes
1 x=0   y=0   constraint=fixed
2 x=360 y=0
3 x=0   y=180 constraint=free   force=f1
4 x=360
5 x=0   y=300                   force=f2
6 x=360
7 x=0   y=420                   force=f3
8 x=360

beam elements
1 nodes=[1,3] material=wall_bottom
2 nodes=[3,5] material=wall_top
3 nodes=[5,7]
4 nodes=[7,8] material=floor_top      load=top_wt
5 nodes=[5,6] material=floor_bottom   load=bottom_wt
6 nodes=[3,4]                         load=bottom_wt
7 nodes=[8,6] material=wall_top
8 nodes=[6,4]
9 nodes=[4,2] material=wall_bottom

material properties
wall_bottom  A=13.2 Ix=249 E=30e6 rho=0.0049    /* rho is in lb-s^2/in^4 */
wall_top     A=6.2  Ix=107 E=30e6 rho=0.0104
floor_top    A=12.3 Ix=133 E=30e6 rho=0.01315
floor_bottom A=24.7 Ix=237 E=30e6 rho=0.0136

distributed loads
top_wt    direction=perpendicular    values=(1,-62.5) (2,-62.5)
bottom_wt direction=perpendicular    values=(1,-130) (2,-130)

forces
f1 Fx=1000*(t < 0.2 ? 25*t : 5)
f2 Fx=800*(t < 0.2 ? 25*t : 5)
f3 Fx=500*(t < 0.2 ? 25*t : 5)

constraints
fixed Tx=c Ty=c Rz=c
```

```
free  Tx=u Ty=u Rz=u

end
```

3.5 Format conversion

Any problem that you want to solve with FElt must be formatted according to the rules that we have just described. However, we recognize that there are countless other formats that provide for descriptions of these same kinds of problems (at least the geometric aspects of the problem). Because of this, FElt provides a mechanism for translating back and forth between FElt and alternative syntaxes.

3.5.1 Conversion basics

A common example of a popular geometric description syntax is DXF (drawing interchange format). DXF files can be produced by any number of CAD and geometric modeling programs. The FElt conversion tool, *patchwork*, can translate DXF files to and from the FElt syntax. *patchwork* is invoked with a command-line like:

```
% patchwork -dxf bridge.dxf -felt bridge.flt
```

This example would convert the geometric description given in the file `bridge.dxf` to the geometric basics of a standard FElt file, `bridge.flt`. Geometric basics in this case means that only the nodes and elements sections will be translated; the DXF file cannot contain any information about material properties, forces, loads, constraints, and therefore this information will be left incomplete in the resulting FElt file.

patchwork can also translate FElt files into other formats. You might want to do this if you had used some of FElt's mesh generation capabilities for the bulk of your geometric problem description, but wanted to do some refinement using your favorite CAD program. The *patchwork* command line to do something like that is simply the reverse of above:

```
% patchwork -felt mesh.flt -dxf mesh.dxf
```

3.5.2 *patchwork* details

patchwork operates by providing routines to read and write the different file formats based around a single data structure. The input formatted file is translated into the uniform database and then the output formatted file is generated from this same database. The ability to read or write any given format is dependent on whether or not routines exist to go back and forth between the file syntax and the uniform data structure. Currently, *patchwork* can read (i.e., the allowable input formats are) DXF files, standard FElt files, and data files formatted for use in graphing applications like *gnuplot*; *patchwork* can write (i.e., allowable output formats are) FElt, DXF, *gnuplot*, and files formatted for the software that is distributed with Logan's introductory finite element text [13]. Capabilities for additional formats will be added as time permits and demand warrants. Note that the DXF handling routines are limited to line and polygonal polyline entities.

When invoking *patchwork*, the input format and file always come first. The syntax is `patchwork -[iformat] ifile -[oformat] ofile`, where `[iformat]` can currently be one of `dxf, felt, graph` and `[oformat]` can be one `dxf, felt, graph, logan`. An input or output file name of `-` (a hyphen) indicates that input should be read from standard input and/or output should be written to standard output, respectively.

Chapter 4

The FElt Element Library

4.1 Introduction

The FElt element library contains line, plane and solid elements. Line elements include a one-dimensional spring (`spring`), a bar or truss element (`truss`), two- and three-dimensional Euler-Bernoulli beams (`beam` and `beam3d`), and a two-dimensional beam based on Timoshenko theory, (`timoshenko`). The set of planar elements consists of constant strain triangles (`CSTPlaneStrain` and `CSTPlaneStress`), isoparametric quadrilaterals (`iso2d_PlaneStress`, `iso2d_PlaneStrain`, `quad_PlaneStrain`, `quad_PlaneStress`) and an HTK plate bending element (`htk`). The only solid element in the library is an eight node brick (`brick`). There is an axisymmetric triangular element (`axisymmetric`). The only elements available for thermal analysis are a simple one-dimensional line element (`rod`) and a constrant temperature gradient triangular element (`ctg`). In the above list, the names in parentheses indicate the actual symbolic type names of the elements.

The number of nodes and the material properties needed for defining each element and the DOF that the element affects are summarized in the table below. A brief description for each element follows the table; detailed derivations of element stiffness matrices can be found in any number of finite element textbooks.

Element type name	nodes	Material properties	DOFs
`truss`	2	`E,A,(rho)`	`Tx,Ty,Tz`
`spring`	2	`E,A,(rho)`	`Tx`
`beam`	2	`E,A,Ix,(rho)`	`Tx,Ty,Rz`
`beam3d`	2	`E,A,Iy,Iz,J,G,(rho)`	`Tx,Ty,Tz,Rx,Ry,Rz`
`timoshenko`	2	`E,A,Ix,G,kappa,(rho)`	`Tx,Ty,Rz`
`CSTPlaneStress`	3	`E,t,nu,(rho)`	`Tx,Ty`
`CSTPlaneStrain`	3	`E,t,nu,(rho)`	`Tx,Ty`
`quad_PlaneStress`	4	`E,t,nu,(rho)`	`Tx,Ty`
`quad_PlaneStrain`	4	`E,t,nu,(rho)`	`Tx,Ty`
`iso2d_PlaneStress`	9	`E,t,nu,(rho)`	`Tx,Ty`
`iso2d_PlaneStrain`	9	`E,t,nu,(rho)`	`Tx,Ty`
`htk`	4	`E,t,nu,kappa,(rho)`	`Tz,Rx,Ry`
`brick`	8	`E,nu,(rho)`	`Tx,Ty,Tz`
`axisymmetric`	3	`E,nu,(rho)`	`Tx, Ty`
`rod`	2	`A,Kx,(c,rho)`	`Tx`
`ctg`	3	`t,Kx,Ky,(c,rho)`	`Tx`

Table 4.1: Description of the materials required for and local DOF affected by the various element types. The material properties in () indicate properties that are only required during a non-static analysis.

4.2 Structural analysis elements

4.2.1 Truss and spring elements

Truss or bar elements are the simplest type of element in the library. They are two-node, three-dimensional linear elements which are assumed to deform in the axial direction only. They might find application in a simple bridge simulation (the classic example of truss elements) or as springs (as in the above example where the spring stiffness, k, is given by EA/L of the truss element). The truss element in the FElt library uses the classical linear shape functions, $N_1 = x/L$ and $N_2 = 1 - x/L$. Any introductory text on finite element analysis (for mechanics at least) probably starts out by deriving the stiffness matrix for just such an element; see Chapter 3 of [13] for instance. The stiffness matrix for this type of element in the local element two-dimensional reference frame is the familiar

$$\hat{k} = \begin{bmatrix} EA/L & -EA/L \\ -EA/L & EA/L \end{bmatrix}. \tag{4.1}$$

The lumped mass formulation of the local mass matrix is given by

$$\hat{m}_l = \frac{\rho AL}{2} \begin{bmatrix} 1 & 0 \\ 0 & 1 \end{bmatrix}, \tag{4.2}$$

and the consistent mass formulation by

$$\hat{m}_c = \frac{\rho AL}{6}\begin{bmatrix} 2 & 1 \\ 1 & 2 \end{bmatrix}. \tag{4.3}$$

The 2×2 matrices in local element coordinates are transformed into the global coordinate system (and into 6×6 form) according to

$$k = T^{\mathrm{T}}\hat{k}T, \tag{4.4}$$

where the transformation matrix is given by

$$T = \begin{bmatrix} \cos\theta_x & \cos\theta_y & \cos\theta_z & 0 & 0 & 0 \\ 0 & 0 & 0 & \cos\theta_x & \cos\theta_y & \cos\theta_z \end{bmatrix}. \tag{4.5}$$

The direction cosines are simply the projections of the local coodinate axes onto the global coordinate axes,

$$\cos\theta_x = \frac{x_1 - x_2}{L}, \tag{4.6}$$

$$\cos\theta_y = \frac{y_1 - y_2}{L}, \tag{4.7}$$

$$\cos\theta_z = \frac{z_1 - z_2}{L}. \tag{4.8}$$

The stress calculated for truss elements is the axial stress (not load). A positive quantity indicates tension, negative indicates compression. If a distributed load is assigned to a truss element it is assumed to be a linearly distributed axial loading condition (i.e., it must be directed `parallel`). Nothing else makes much sense since a truss element cannot carry bending or shear forces along its length. The sign convention for the magnitude of the load pairs is positive for loads pointing from local node 1 to local node 2.

The spring element included in FElt is simply the truss element described above without the transformation from local element coordinates to global coordinates. What this means is that the element matrices are only 2×2 and because of this they must be defined only along the global x-axis. Distributed loads are not supported by spring elements.

4.2.2 Euler-Bernoulli beam elements

4.2.2.1 Special case two-dimensional element

A beam element is a two-dimensional, two-node linear element which can carry in-plane axial, shear and bending forces. The moment of inertia used in stiffness calculations (specified with `Ix`) is the bending moment of inertia about the cross-section x-x axis (which in global coordinates is bending about the z-axis). Like the truss element described above, the beam element in the FElt library is also a standard in many textbooks; the following stiffness matrix should also look relatively familiar,

$$\hat{k} = \begin{bmatrix} AE/L & 0 & 0 & -AE/L & 0 & 0 \\ 0 & 12EI/L^3 & 6EI/L^2 & 0 & -12EI/L^3 & 6EI/L^2 \\ 0 & 6EI/L^2 & 4EI/L & 0 & -6EI/L^2 & 2EI/L \\ -AE/L & 0 & 0 & AE/L & 0 & 0 \\ 0 & -12EI/L^3 & -6EI/L^2 & 0 & 12EI/L^3 & -6EI/L^2 \\ 0 & 6EI/L^2 & 2EI/L & 0 & -6EI/L^2 & 4EI/L \end{bmatrix}. \tag{4.9}$$

The lumped mass matrix for the beam element is defined as

$$\hat{m}_l = \frac{\rho AL}{2} \begin{bmatrix} 1 & 0 & 0 & 0 & 0 & 0 \\ 0 & 1 & 0 & 0 & 0 & 0 \\ 0 & 0 & \frac{L^2}{12} & 0 & 0 & 0 \\ 0 & 0 & 0 & 1 & 0 & 0 \\ 0 & 0 & 0 & 0 & 1 & 0 \\ 0 & 0 & 0 & 0 & 0 & \frac{L^2}{12} \end{bmatrix}, \tag{4.10}$$

and the consistent mass matrix as

$$\hat{m}_c = \frac{\rho AL}{420} \begin{bmatrix} 140 & 0 & 0 & 70 & 0 & 0 \\ 0 & 156 & 22L & 0 & 54 & -13L \\ 0 & 22L & 4L^2 & 0 & 13L & -3L^2 \\ 70 & 0 & 0 & 140 & 0 & 0 \\ 0 & 54 & 13L & 0 & 156 & -22L \\ 0 & -13L & -3L^2 & 0 & -22L & 4L^2 \end{bmatrix}. \tag{4.11}$$

The basis for these mass matrices should also be readily available in many textbooks. The transformation from local element coordinates into global coordinates is of the same form as equation 4.4; the actual transform matrix for two-dimensional beams is given by

$$T = \begin{bmatrix} \cos\theta_x & \cos\theta_y & 0 & 0 & 0 & 0 \\ -\cos\theta_y & \cos\theta_x & 0 & 0 & 0 & 0 \\ 0 & 0 & 1 & 0 & 0 & 0 \\ 0 & 0 & 0 & \cos\theta_x & \cos\theta_y & 0 \\ 0 & 0 & 0 & -\cos\theta_y & \cos\theta_x & 0 \\ 0 & 0 & 0 & 0 & 0 & 1 \end{bmatrix}, \tag{4.12}$$

where the direction cosines, $\cos\theta_x$ and $\cos\theta_y$, are defined by equations 4.6 and 4.7.

A distributed load assigned to a beam element can either be directed `perpendicular` to the element (as in the self-weight case or the linearly sloping loads in the second example

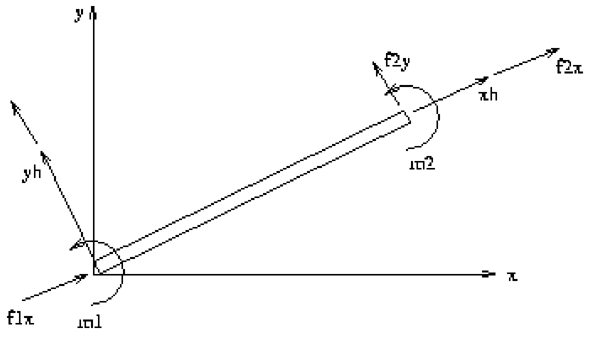


Figure 4.1: Sign convention for local forces on a beam element.

above), `parallel` to the element (just like a linearly distributed axial load in the truss case), or in the `GlobalX` or `GlobalY` directions. Note that loads given in the `LocalY` and `LocalX` directions will be taken as equivalent to the `perpendicular` and `parallel` cases, respectively. For `perpendicular` loads a positive magnitude indicates that the load points in the direction of positive $\hat{y}$. A positive `parallel` loads point from node 1 to node 2 as in the truss case. Similarly, the sign convention for loads in the global directions follow the sign convention of the global coordinate axes. Each beam element is limited to two applied distributed loads.

After displacements are found, six internal force quantities are computed for each beam element. These quantities are the axial force, shear force and bending moment at both nodes. The sign convention for these forces is shown in Figure 4.1. This convention is based on a coordinate system in which the local x-axis, $\hat{x}$, points from node 1 to node 2. As z and $\hat{z}$ always coincide for the 2d beam element, and we define the positive z-axis to point out of the page, $\hat{y} = z \times \hat{x}$.

The three beam type elements (`beam`, `beam3d` and `timoshenko`) all are capable of resolving hinged boundary conditions for the rotational degrees of freedom. The adjustment is made to the element stiffness matrix in global coordinates according to the following procedure. Given a hinged DOF, dof, then for all entries in k (the element stiffness matrix) not associated with dof (all entries not in row or column dof),

$$k(i,j) = k(i,j) - \frac{k(dof,j)}{k(dof,dof)} k(i,dof). \tag{4.13}$$

The inherent problem in this method of dealing with hinged conditions is that we cannot calculate any displacements associated with the hinged DOF and thus, the internal forces calculated for any element with a hinged node will not be correct. Displacements other than at the hinged DOF will be accurate.

4.2.2.2 Arbitrarily oriented three-dimensional element

Like their 2-d special case cousins, three-dimensional beam elements are two-node linear elements. 3-d beams however, can carry forces in any of the six DOFs - axial (local x-direction), vertical shear (local y-direction), horizontal shear (local z-direction), rotation about the x-axis (torsion), rotation about the y-axis (out-of-plane bending) and rotation about the z-axis (in-plane bending). Both the elastic and shear moduli (`E` and `G`) must be specified for a 3-d beam element. The cross-sectional area (`A`), torsional moment of inertia (`J`), I_{yy} moment of inertia (`Iy`) and I_{zz} moment of inertia (`Iz`) must also be specified. `beam3d` elements can define either a lumped or consistent local mass matrix. The lumped formulation includes entries at every DOF (i.e., there is an inertia entry at rotational DOF) just like the two-dimensional beam.

The local coordinate system used for the FElt 3-d beam element is based on the element geometry presented in [13]. Positive $\hat{x}$ points from node 1 to node 2. Then, $\hat{y}$ is defined by the cross product of z and $\hat{x}$ (global z and local x), i.e., $\hat{y} = z \times \hat{x}$. $\hat{z}$ is selected such that it is orthogonal to the $\hat{x}$-$\hat{y}$ plane, $\hat{z} = \hat{x} \times \hat{y}$. Given this geometry, clockwise moments and rotations about $\hat{y}$ are positive; for moments and rotations about $\hat{z}$, counter-clockwise is defined as positive.

Local internal forces calculated for each 3-d beam element include all six forces at both nodes (twelve total forces for each element). A distributed load on a 3-d beam can be directed `parallel` (`LocalX` is equivalent) or in the `LocalY`, `LocalZ`, `GlobalX`, `GlobalY`, or `GlobalZ` directions. A `parallel` load will be taken as a linearly distributed axial force. Like `beam` elements, the sign convention for distributed loads is based on the appropriate coordinate axes system. Positive `parallel` loads point from node 1 to node 2. Positive loads in the local directions point in the direction of of the positive $\hat{y}$ and $\hat{z}$ axes, respectively. The sign conventions for loads in the global directions follow the global coordinate axes. `beam3d` elements are limited to at most three applied distributed loads.

4.2.3 Timoshenko beam element

The Timoshenko beam element currently in the library is really intended as a well-worked example of how to add an element to the FElt system (see Chapter 13). It is limited to in-plane behavior and does not support an axial degree of freedom.

There are lots of approaches to defining an element using Timoshenko beam theory. Classic examples can be found in [10, 15]. The formulation we use is from [4]. In all formulations, the stiffness matrix is defined as

$$k = \frac{EI}{(1+\phi)L^3} \begin{bmatrix} 12 & 6L & -12 & 6L \\ 6L & (4+\phi)L^2 & -6L & (2-\phi)L^2 \\ -12 & -6L & 12 & -6L \\ 6L & (2-\phi)L^2 & -6L & (4+\phi)L^2 \end{bmatrix}. \tag{4.14}$$

In the above equation, ϕ is defined as the ratio of the bending stiffness to shear stiffness,

$$\phi = \frac{EI}{\kappa GA}. \tag{4.15}$$

κ is the shear coefficient from Timoshenko beam theory. Cowper [2] provides an approximation for κ based on Poisson's ratio,

$$\kappa = \frac{10(1+\nu)}{12+11\nu}, \tag{4.16}$$

if a better estimate is not available. This approximation will automatically be assumed if you provide `nu` rather than `kappa` in the material property for a Timoshenko beam element. Because there is no axial DOF in this formulation, you need to be careful about horizontally oriented elements; nothing will be assembled into the translational x DOF in these cases so you should be extra careful about constraints. The lumped mass matrix for the `timoshenko` element looks just like that for the 2-d Euler-Bernoulli element (eq. 4.10), minus the axial DOF of course. The definition of the consistent mass matrix varies from one formulation of Timoshenko theory to the next. The definition in the formulation that we are using is considerably more complicated than the Euler-Bernoulli formulation; see [4] for details.

Distributed loads on `timoshenko` elements can only be directed in the `perpendicular` (equivalent to `LocalY`) direction. (There are no axial DOF after all). The sign conventions for these loads and for internal forces is the same as that for the standard beam element. The internal forces calculated will be the shear forces and bending moment at each end.

4.2.4 Constant Strain Triangular (CST) elements

Two different CST elements are in the FElt library - one for plane stress analysis and one for plane strain analysis. This means that the only difference between the two is that the constitutive matrix, D, used in the element stiffness formulation is different for the two cases. A CST element is a three-node, two-dimensional, planar element. Each CST element should exist completely in a single x-y plane. The node numbers must be assigned to a CST element in counter-clockwise order to avoid the element having a negative area.

The lumped mass matrix for a CST element is formed simply by dividing the mass of the element equally between the three nodes, i.e.,

$$m_l = \frac{\rho A t}{3} \begin{bmatrix} 1 & 0 & 0 & 0 & 0 & 0 \\ 0 & 1 & 0 & 0 & 0 & 0 \\ 0 & 0 & 1 & 0 & 0 & 0 \\ 0 & 0 & 0 & 1 & 0 & 0 \\ 0 & 0 & 0 & 0 & 1 & 0 \\ 0 & 0 & 0 & 0 & 0 & 1 \end{bmatrix}, \tag{4.17}$$

where A is the computed planar area of the element (not the area as defined by the material property `A=`). There is no consistent mass formulation available for the current set of CST elements.

Six stress quantities are computed for each CST element: σ_x, σ_y, τ_{xy}, σ_1, σ_2, θ, where σ_x, σ_y, τ_{xy}, are the stresses in the global coordinate system, σ_1, σ_2 are the principal streses, and θ is the orientation of the principal stress axis. A distributed load on a CST element is taken as an in-plane surface traction. Valid directions for loads are `GlobalX` and `GlobalY`. The sign convention for load magnitude follows the orientation of the global axes. Loads which are perpendicular or parallel to element sides which are not parallel to one of the global axes must be broken down into components which are parallel to the axes and then specified as two separate loads.

4.2.5 Two-dimensional isoparametric elements

4.2.5.1 General four to nine node element

Like CST elements, isoparametric elements are available for either plane strain or plane stress analysis. Isoparametric elements are 2-d planar, quadrilateral elements with nine nodes. Any of the last five nodes are optional, however and the fourth node can be the third node repeated. The numbering convention is shown in Figure 4.2. If any of nodes 5 - 9 are left out, their place must be filled with a zero in the `nodes=[ ...]` specification for that element (i.e. you must always specify nine numbers, it is simply that any or all of the last five might be specified as zero). None of the first four nodes can be zero. If the fourth and third nodes are the same, then the element will be degenerated to a triangle. In this case nodes 5 - 9 must be zero.

Currently, no stresses are computed for the generalized isoparametric element. Distributed loads on the generalized isoparametric element are ignored and do not affect the problem solution in any way.

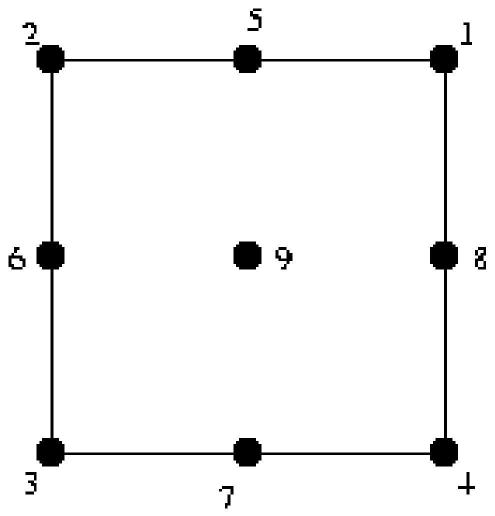


Figure 4.2: Node numbering scheme for nine node isoparametric planar element.

4.2.5.2 Simple four node element

`quad_PlaneStress` and `quad_PlaneStrain` elements are just like the generalized isoparametric elements described above, but they can only have the first four nodes. They are intended to make problem definition easier in cases where the added accuracy that the additional nodes offer is not worth the extra effort. Like the generalized case, if the fourth node is the same as the third node the element will be degenerated into a triangle. This allows for easy mixing and matching of element shapes without changing element types (though this too would be allowed).

Distributed loads on the four node isoparametric elements work exactly as they do in the CST case. The stresses computed for each four node isoparametric are also the same as those computed for CSTs.

Like CST elements, only a lumped mass formulation is available for the mass matrices of isoparametric quadrilateral elements. The lumped formulation is generated by lumping the total mass of the element equally at the four nodes (or the three nodes if the element is being degenerated into a triangle).

4.2.6 Plate bending element

The plate bending element in FElt is a simple four node isoparametric quadrilateral that uses selective reduced integration to prevent shear locking. The classic reference for this

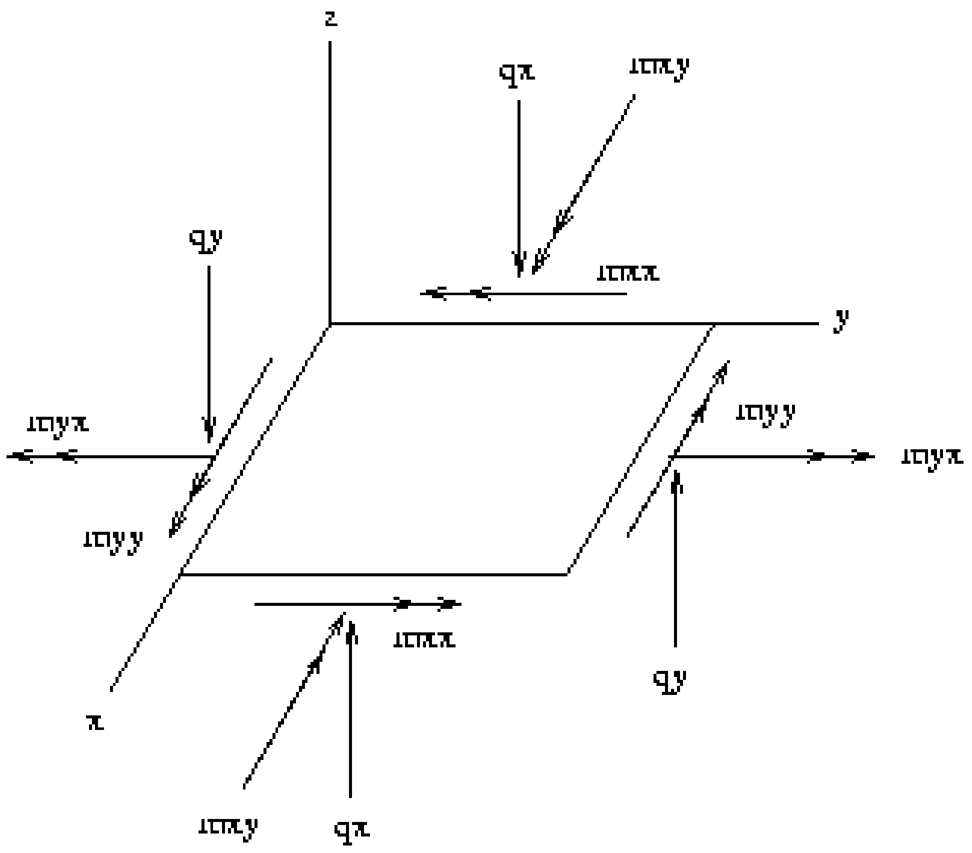


Figure 4.3: Sign convention for force resultants on an htk element.

approach is [10]. The effect is achieved simply by using one-point Gaussian quadrature to under-integrate the shear contribution to the stiffness where two-point quadrature is used on the bending contributions.

`htk` elements must exist entirely in the x-y plane; the three DOF at each node each represent out of plane deformation. Rotations are positive in the right-hand sense. You can generate plate bending triangular elements by using `htk` elements with the third and fourth nodes being equal (just like you can generate triangles from isoparametric quadrilateral elements). Mass matrices for `htk` elements also work the exact same way as the mass matrices for the in-plane `quad_PlaneStress` and `quad_PlaneStrain` elements.

Figure 4.3 illustrates the sign convention for the internal force resultants calculated for each `htk` element.

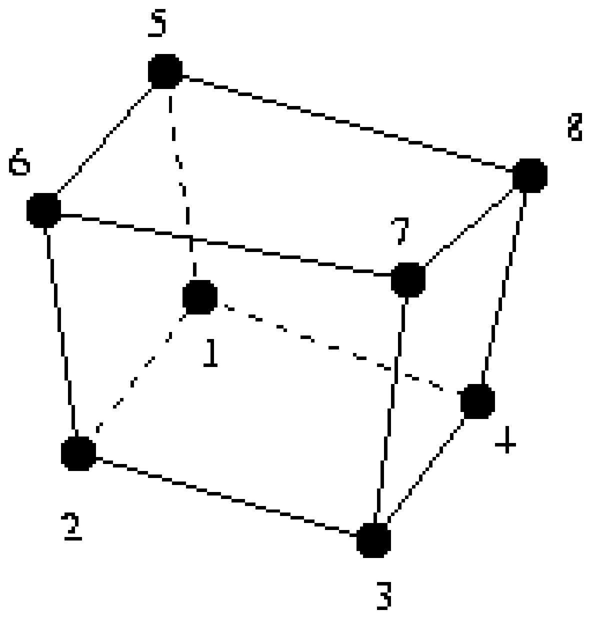


Figure 4.4: An example of valid node ordering on a brick element.

4.2.7 Solid brick element

The eight-node solid brick element in FElt is based on an isoparametric formulation using linear shape functions. The element incorporates all three translational DOF at each node for a total of 24 local degrees of freedom. The 24×24 element stiffness matrix is evaluated using a fourth-order accurate $2 \times 2 \times 2$ Gaussian quadrature rule. This formulation results in stress calculations at eight integration points. Figure 4.4 shows the local node numbering convention for bricks. As the figure illustrates, nodes 1-4 and nodes 5-8 must define opposite faces.

All six independent components of the stress tensor are calculated for brick elements. Brick elements are still somewhat limited, however, in that distributed loads are not handled and element definition routines do not calculate a mass matrix. Also, note that because brick elements are the only solid elements in the FElt library, they are also the only elements which are not fully supported by *velvet*, both in terms of problem definition and in terms of post-processing.

4.2.8 Axisymmetric elements

The axisymmetric element is a three node triangular element based on linear shape functions. The axis of symmetry/revolution is always the y-axis. Distributed loads can be directed in either the `axial` (y) or `radial` (x) directions.

4.3 Thermal analysis elements

Thermal elements only have one degree of freedom per node, the nodal temperature. Prescribed and initial temperatures are assigned using constraints, `Tx=`, and `iTx=`. Heat sources are specified using nodal forces (`Fx=`). For uniform heat sources, simply apply the same force to all nodes.

4.3.1 Rod element

The rod element is a two node straight line element for thermal analysis. Convection surfaces are specified using distributed loads. The direction of the load does not matter and does not need to be specified. The node, magnitude pairs of the `values` definition of the load are used to specify the surface on which the convection is taking place and the convection coefficient and free stream temperature. For example `values=(1,20) (2,50)` specifies a convection coefficient of 20, a free stream temperature of 50, and that the convection acts over the circumferential surface area of the rod. If the node values are equal in the two pairs then the convecting surface is taken to be an exposed end of the rod.

4.3.2 Constant Temperature Gradient (CTG) element

The `ctg` element is a thermal analog to the CST triangular elements for mechanical problems. The temperature is assumed to vary linearly over the element. As with `rod` elements, convection on `ctg` elements is specified using distributed loads. Only the three edges are handled. The node, magnitude pairs of the `values` definition of the load are used to specify the edge on which the convection is taking place and, as for the rod, the convection coefficient and free stream temperature. For a `ctg element` the definition `values=(1,20) (2,50)` specifies a convection coefficient of 20, a free stream temperature of 50, and that the convection acts on the edge defined by local element node numbers 1 and 2.

Chapter 5

The *felt* Application

5.1 Using *felt*

The basic way to solve a problem with *felt* is simply to type `felt foo.flt` on the command line, where `foo.flt` is the name of a FElt input file. There are several command line options, however, which offer additional capabilities.

`-version`	display the current version number of *felt* and exit.
`-help`	display a brief help message which lists all of the available command line options.
`-debug`	will generate a FElt file, that if all is working well, should look exactly like the original input file. The generated file represents what the application thinks it was given.
`-preview`	produces an ASCII rendering of the problem geometry. The graphics may not be great but the result is often good enough for a simple sanity check.
`-renumber`	will invoke a Gibbs-Poole-Stockmeyer/Gibbs-King automatic node renumbering algorithm for bandwidth/profile reduction of the global stiffness matrix (and mass and damping matrices in transient analysis). The renumbering will only affect solution time and memory requirements; results will be referenced to the originally assigned node numbers. Generally there is no benefit in using this capability for problems with small numbers of nodes.
`+table`	disables tabular output for transient and spectral analysis problems.

`-plot`	generates an ASCII character based plot (or plots) of results for transient or spectral analysis problems.
`-transfer`	only calculate the transfer functions during spectral analysis. The output can be graphical and/or tabular depending on the `plot` and `table` options.
`-eigen`	in a modal analysis problem, only calculate the eigenvalues (natural frequencies) and eigenvectors (mode shapes) before quitting (i.e., do not calculate modal mass, damping and stiffness matrices).
`-orthonormal`	use orthonormal mode shapes to calculate modal mass, damping, and stiffness matrices in a modal analysis problem.
`-matrices`	will print the global matrices that are appropriate to this problem. In static analysis this would simply be the stiffness matrix; in transient analysis this is the stiffness, damping, and mass matrices; in modal analysis, this is the condensed stiffness, damping, and mass matrices.
`-summary`	will generate a summary of all the material properties that were used in the problem. The number of elements using a material, total length, surface area and mass of that material (if masses are desired, then the material properties definition must include a density) and the total mass of the structure are given.
`-graphics foo1`	will create a graphics file called foo1 so the user can visualize the structure with a standard graphing package to make sure that everything is connected where it should be. The format of this file will be a list of coordinate triplets connecting each element, with blank lines delimiting the end of an element.
`-cpp filename`	substitute `filename` for the pre-processor to run on the input file.
`-nocpp`	do not run the input file through the pre-processor.
`-Idirectory`	add `directory` to the standard search path for include files in the pre-processor.
`-Uname`	undefine the macro `name` in the pre-processor.

`-Dname=value`

define `name` to be the macro `value` in the pre-processor.

Additionally, every FElt file that is run through *felt* is pre-processed by the C preprocessor (except in DOS). This allows for the use of macro definitions and include files within a FElt input file. An example of this capability is the international translation files which allow you to specify a FElt input file in a language other than English. These translation files are nothing more than include files which contain `#define` statements which map the non-English terms to what FElt actually expects (the regular English terms). An input file that takes advantage of these powerful features might look like this:

```
#include "german.trn"
#define map(x) ((x)*(cos((x))*cos((x)) + sin((x))*sin((x))))

problembeschreibung
titel="A cantilever beam" knoten=3 elemente=2

knoten
1 x=0        y=0    krafte=point_load      zwangsbedingung=pin
2 x=map(240) y=0

beam elemente
1 knoten=[1,2] material=steel

materialeigenschaften
steel e=30e6 ix=100 a=10

kraefte
point_load fy=-1000

zwangsbedingungen
pin tx=c ty=c rz=u

ende
```

The C preprocessor options and capabilities are described more fully in the *felt*(1fe) manual page. The international translation capability is discussed more fully (including current translation tables) in appendix A.

5.2 Solving a problem

As mentioned above the easiest way to solve a problem is to simply do

```
% felt foo.flt
```

at the shell prompt . If you want to save the output it is easy to simply re-direct standard output to a file by doing

```
% felt foo.flt > foo.results
```

Th file `foo.results` could then be printed using whatever printer facilities exist at your site (*lp*, *lpr*, *print*, etc.) If you want to plot the structure, the command line might look like this:

```
% felt -graphics foo.graph foo.flt > foo.results
% myplotter foo.graph
```

The file `foo.graph` could then be used with a program like *gnuplot*, *xmgr* or *xgraph* (here we used a mystical program called *myplotter*) to visualize the structure and make sure that all the elements were connected properly.

5.3 Interpreting the output from *felt*

The output from *felt* is the most basic form of output that all applications in the FElt system produce. That is, no matter how you specify and solve a FElt problem (either with *felt* or *velvet*) the basic output will look the same.

The main output will vary depending on the analysis type defined in the input file. In addition to the results from the specific analysis, supplemental kinds of output which are available include material usage statistics, debugging information, and print-outs of the global stiffness and mass matrices. Supplemental output from *felt* is controlled via command line switches (see section 5.1).

5.3.1 Static analysis

For a static analysis problem there are three sections of output: nodal displacements, element stresses and reaction forces. The nodal displacements are given in a table with the six global DOF running across the top and the list of nodes going down. The displacement of each node in each DOF is printed. Of course in most problems not all DOF will be active so many of the displacements will simply come up as zero.

The information in the element stress table will vary depending on the element type. Each row contains up to six columns of information; if an element calculates more than six stresses or loads, the information for that element will take up more than one row (beam3d elements for instance have 12 loads calculated and thus the information for each element takes up two rows). The output format for each element type is summarized in Table 5.1.

Element	col 1	col 2	col 3	col 4	col 5	col 6
truss	σ_x					
beam	f_x^1	f_y^1	m_z^1	f_x^2	f_y^2	m_z^2
beam3d	f_x^1	f_y^1	f_z^1	m_x^1	m_y^1	m_z^1
	f_x^2	f_y^2	f_z^2	m_x^2	m_y^2	m_z^2
timoshenko	f_y^1	m_z^1	f_y^2	m_z^2		
CSTPlaneStress	σ_x	σ_y	τ_{xy}	σ_1	σ_2	θ
CSTPlaneStrain	σ_x	σ_y	τ_{xy}	σ_1	σ_2	θ
quad_PlaneStress	σ_x^1	σ_y^1	τ_{xy}^1	σ_1^1	σ_2^1	θ^1
	$\vdots$	$\vdots$	$\vdots$	$\vdots$	$\vdots$	$\vdots$
	σ_x^4	σ_y^4	τ_{xy}^4	σ_1^4	σ_2^4	θ^4
quad_PlaneStrain	σ_x^1	σ_y^1	τ_{xy}^1	σ_1^1	σ_2^1	θ^1
	$\vdots$	$\vdots$	$\vdots$	$\vdots$	$\vdots$	$\vdots$
	σ_x^4	σ_y^4	τ_{xy}^4	σ_1^4	σ_2^4	θ^4
htk	m_{xx}	m_{yy}	m_{xy}	q_x	q_y	
brick	σ_{xx}^1	σ_{yy}^1	σ_{zz}^1	τ_{xy}^1	τ_{yz}^1	τ_{zx}^1
	$\vdots$	$\vdots$	$\vdots$	$\vdots$	$\vdots$	$\vdots$
	σ_{xx}^8	σ_{yy}^8	σ_{zz}^8	τ_{xy}^8	τ_{yz}^8	τ_{zx}^8

Table 5.1: Contents of the stress vector for different element types.

The reaction force section contains the global forces calculated at each constrained DOF. The first column indicates the global node number at which this reaction occurs, the second column contains the DOF at which this force is applied and the last column gives the magnitude of this reaction force.

If we were to save the input file from the detailed example of section 3.3 to a file `mixed.flt` then the command

```
% felt -summary mixed.flt > mixed.result
```

would produce a file, `mixed.result`, which looked like the following.

```
** Mixed Element Sample **

Nodal Displacements
-----------------------------------------------------------------------------
Node #      DOF 1       DOF 2       DOF 3       DOF 4       DOF 5       DOF 6
-----------------------------------------------------------------------------
  1             0           0           0           0           0  -0.0032019
  2             0   -0.011522           0           0           0           0
  3             0           0           0           0           0   0.0032019
  4             0           0           0           0           0           0

Element Stresses
----------------------------------------------------------------------------
  1:            0      29942           0          0      57.587           0
  2:            0     57.587      -59654          0       29942           0
  3:  -2.4196e+08

Reaction Forces
-----------------------------------
Node #     DOF     Reaction Force
-----------------------------------
  1        Tx                 0
  1        Ty             29942
  1        Tz                 0
  2        Tz                 0
  3        Ty             29942
  3        Tz                 0
  4        Tx                 0
  4        Ty            115.17
  4        Tz                 0
  4        Mz                 0

Material Usage Summary
--------------------------
Material: steel
Number:   2
Length:    12.0000
Weight:     0.0000

Material: spring
```

```
Number:   1
Length:    10.0000
Mass:      0.0000

Total mass:     0.0000
```

5.3.2 Transient analysis

The output for transient analysis is much simpler. It can consist of a table listing times and nodal displacements for all the nodes and DOF which were specified in the `analysis parameters` section (or multiple tables if everything will not comfortably fit across one screen width) and/or a time-displacement ASCII character plot relating the same information in a graphical sense (or mulitple plots if there are more than ten nodes/DOF). Tabular output is the default. You can disable it with the command line switch `+table`. To see graphical output use `-plot`.

If we were to save the input file from the detailed example of section 3.4 to a file `transient.flt` then the command

```
% felt -plot transient.flt > transient.result
```

would produce a file, `transient.result`, which looked like the following.

```
------------------------------------------------------------------
       time        Tx(8)
------------------------------------------------------------------
          0            0
       0.05   0.00077749
        0.1     0.028323
       0.15      0.10137
        0.2      0.23788
       0.25      0.45923
        0.3      0.76125
       0.35       1.1366
        0.4       1.5518
       0.45       1.9716
        0.5        2.352
       0.55       2.6652
        0.6       2.8977
       0.65       3.0442
        0.7       3.1074
```

```
    0.75         3.08
     0.8       2.9652

     0                            1.5537                          3.1074
     +-------------------------------+-----------------------------------+
     |
   0 x
     |
0.05 x
     |
 0.1 x
     |
0.15 | x
     |
 0.2 |    x
     |
0.25 |        x
     |
 0.3 |               x
     |
0.35 |                       x
     |
 0.4 |                                x
     |
0.45 |                                          x
     |
 0.5 |                                                  x
     |
0.55 |                                                         x
     |
 0.6 |                                                              x
     |
0.65 |                                                                 x
     |
 0.7 |                                                                  x
     |
0.75 |                                                                  x
     |
 0.8 |                                                               x
     |

   x x x    Tx(8)
```

5.3.3 Modal analysis

The output from a modal analysis of a simple one story frame is shown below. Note that the `-matrices` command-line switch was used to generate supplemental output (the matrices prints before the results from the actual analysis) and that the `-eigen` switch was not activated so full modal analysis was performed. The actual modal analysis results consist of a listing of modal frequencies, a table of mode shape vectors (the eigenvectors)[1], the modal mass, damping, and stiffness matrices and the damping ratios in each mode.

```
M =


      15.62          0          0          0          0          0
          0      15.62          0          0          0          0
          0          0  8.558e+04          0          0          0
          0          0          0      15.62          0          0
          0          0          0          0      15.62          0
          0          0          0          0          0  8.558e+04

C =


      194.8          0        720     -184.6          0          0
          0      801.7      170.4          0     -1.049      170.4
        720      170.4  1.123e+05          0     -170.4  1.846e+04
     -184.6          0          0      194.8          0        720
          0     -1.049     -170.4          0      801.7     -170.4
          0      170.4  1.846e+04        720     -170.4  1.123e+05

K =


  9.711e+04          0    3.6e+05 -9.231e+04          0          0
          0  4.005e+05  8.521e+04          0     -524.4  8.521e+04
    3.6e+05  8.521e+04  5.446e+07          0 -8.521e+04  9.231e+06
 -9.231e+04          0          0  9.711e+04          0    3.6e+05
          0     -524.4 -8.521e+04          0  4.005e+05 -8.521e+04
          0  8.521e+04  9.231e+06    3.6e+05 -8.521e+04  5.446e+07

** Modal Analysis Example **

Modal frequencies (rad/sec)
------------------------
Mode #      Frequency
------------------------
```

[1]The current version of *felt* cannot generate a graphical representation of the mode shapes. See the discussion of mode shape plotting in chapter 8 for an example of how *velvet* can generate these kinds of plots.

```
  1        12.049  (     1.9177 Hz)
  2        22.807  (     3.6299 Hz)
  3         30.09  (     4.7889 Hz)
  4        110.14  (     17.529 Hz)
  5           160  (     25.465 Hz)
  6        160.21  (     25.499 Hz)

Mode shapes
--------------------------------------------------------------------------
   Mode    1     Mode    2     Mode    3     Mode    4     Mode    5     Mode    6
--------------------------------------------------------------------------
         1           1           1           1           1           1
   0.003005  7.4866e-16   -0.011436 -6.6597e-17 -1.9981e+16       13775
  -0.007032    -0.50358    0.025963  0.00036256   -0.012534      1.1007
         1          -1           1          -1       3.667           1
  -0.003005  -4.565e-15    0.011436  1.0303e-16 -1.9981e+16      -13775
  -0.007032     0.50358    0.025963 -0.00036256   -0.041164      1.1007

modal M =

     39.71          0          0          0          0          0
         0  4.344e+04          0          0          0          0
         0          0      146.6          0          0          0
         0          0          0      31.27          0          0
         0          0          0          0  1.248e+34          0
         0          0          0          0          0   5.93e+09

modal K =

      5766          0          0          0          0          0
         0  2.259e+07          0          0          0          0
         0          0  1.328e+05          0          0          0
         0          0          0  3.794e+05          0          0
         0          0          0          0  3.194e+38          0
         0          0          0          0          0  1.522e+14

modal C =

     13.12          0          0          0          0          0
         0  4.693e+04          0          0          0          0
         0          0      271.4          0          0          0
         0          0          0        760          0          0
         0          0          0          0  6.393e+35          0
         0          0          0          0          0  3.047e+11
```

```
------------------------
Modal damping ratios
------------------------
  1        0.01371
  2        0.02368
  3        0.03075
  4        0.11032
  5        0.16013
  6        0.16034
```

5.3.4 Thermal analysis

The result from a static thermal analysis is a simple table showing the node number and corresponding steady state temperature value; thermal stresses are not computed.

Output for transient thermal analysis can be tabular and/or graphical. The tabulated results simply show the temperature as a function of time for whatever nodes were listed in the `nodes=` specification of the analysis parameters (no `dofs=` specification is necessary in transient thermal analysis). If you specify the `-plot` option on the command line then *felt* will generate a simple ASCII plot of time versus temperature for these same nodes of interest. Just as with transient structural analysis, tabular output can be turned off witht the `+table` command-line switch.

5.3.5 Spectral analysis

Depending on how *felt* is invoked and the nature of the problem, the output for a spectral analysis problem can either consist of transfer functions or of output spectra. If the `-transfer` switch is given on the command line then all output will be for transfer functions only. The transfer functions computed will be for all combinations of output at the nodes and DOF defined by the `nodes=` and `dofs=` statements in the `analysis parameters` section of the input file and input at all DOF with a force applied. For example, if you define nodes 2 and 4 and DOF Tx and Ty as the relevant DOF in the analysis parameters and there is force object with Fx and Fy non-zero applied to both nodes 3 and 7, then you will get sixteen transfer functions as results. If the input forcing is defined in a spectral sense (e.g., with `Sfx=`) then you can also compute the output spectra at the output DOF due to input at any forced DOF with spectral inputs. This is the default behavior when the `-transfer` switch is not specified on the command-line. If none of the forces in a problem have non-zero spectral input components then the only solution that is non-trivial is with the `-transfer` switch invoked.

Just like for transient analysis, the output for spectral analysis (either transfer functions or output spectra) can be tabular and/or plotted. ASCII plots of transfer functions or output spectra (whichever are appropriate for the current problem) are generated if the `-plot` switch is specified. Tabular output is generated by default and can be turned off with the `+table` switch.

Chapter 6

Using *WinFElt*

6.1 Introduction to *WinFElt*

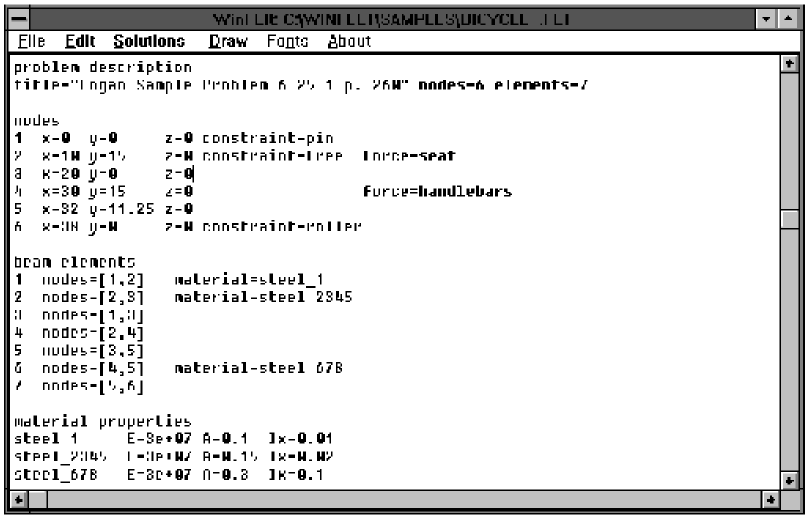
The primary feature of the main *WinFElt* window is a standard text editor.

6.2 Solving a problem

Depending on what you want your solution to include, there are a couple of different ways to go about solving a FElt problem from within *WinFElt*. In addition to the standard tabular type FElt output you can also have *WinFElt* generate two-dimensional color shaded stress or displacement plots and/or plots of the structure with magnified displacements applied. For transient analysis problems graphical time-displacement or time-temperature plots are available; in spectral analysis problems, plots of the transfer functions or ouput spectra are available.

All of the *WinFElt* output windows have a **File** menu from which you can save or print the results in that window. Line graphs are saved in Windows Metafile format (WMF) and color contour plots are saved as Windows bitmaps (BMP) Text output can be saved as an ASCII file. Once an output window is closed there is no way to bring the window back up without resolving the problem.

One way to solve a problem is to select the **solve** option from the **Solutions** menu. By default, this simply generates the standard FElt tabular output in a text output window (figure 6.2). This window will contain either the mathematical solution of your problem or the syntax errors encountered in solving the problem.


WinFElt C:\WINFELT\SAMPLES\BICYCLE.FLT
File Edit Solutions Draw Fonts About

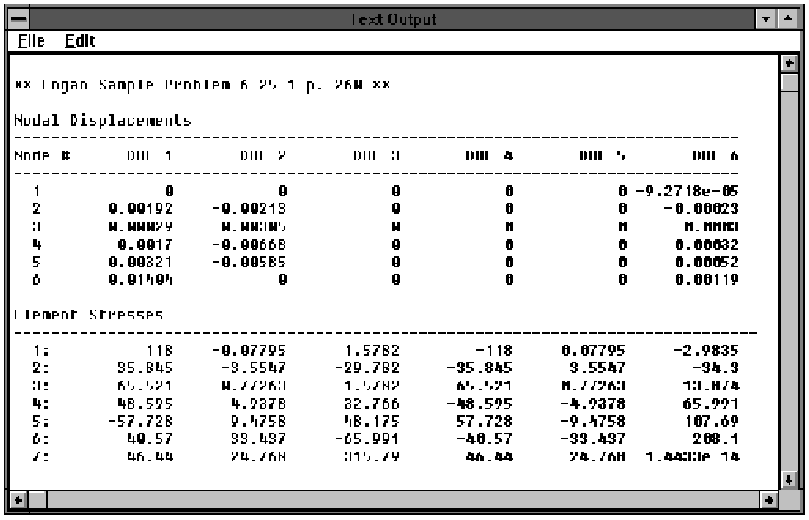
```
problem description
title="Logan Sample Problem 6.25.1 p. 260" nodes=6 elements=7

nodes
1  x=0  y=0      z=0 constraint=pin
2  x=10 y=15     z=0 constraint=free  force=seat
3  x=20 y=0      z=0
4  x=30 y=15     z=0                  force=handlebars
5  x=32 y=11.25 z=0
6  x=38 y=0      z=0 constraint=roller

beam elements
1  nodes=[1,2]   material=steel_1
2  nodes=[2,3]   material=steel_2345
3  nodes=[1,3]
4  nodes=[2,4]
5  nodes=[3,5]
6  nodes=[4,5]   material=steel_678
7  nodes=[5,6]

material properties
steel_1      E=3e+07 A=0.1  Ix=0.01
steel_2345   E=3e+07 A=0.15 Ix=0.02
steel_678    E=3e+07 A=0.3  Ix=0.1
```

Figure 6.1: Main *WinFElt* editor with a sample problem.

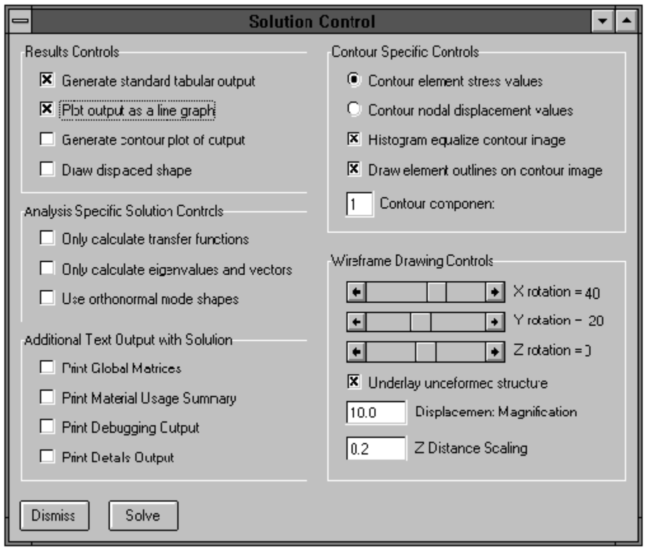


```
** Logan Sample Problem 6.25.1 p. 260 **

Nodal Displacements
------------------------------------------------------------------------------
Node #       DOF 1        DOF 2        DOF 3        DOF 4        DOF 5        DOF 6
------------------------------------------------------------------------------
  1              0            0            0            0            0 -9.2718e-05
  2        0.00192     -0.00213            0            0            0    -0.00023
  3        0.00029      0.00305            0            0            0 [illegible]
  4         0.0017     -0.00668            0            0            0     0.00032
  5        0.00321     -0.00585            0            0            0     0.00052
  6        0.01404            0            0            0            0     0.00119

Element Stresses
--------------------------------------------------------------------------------
  1:           118     -0.07795       1.5782         -118      0.07795     -2.9835
  2:        35.845      -3.5547      -29.782      -35.845       3.5547       -34.3
  3:        65.521      0.77263       1.5782       65.521      0.77263      13.874
  4:        48.595       4.9378       32.766      -48.595      -4.9378      65.991
  5:       -57.728       9.4758       48.175       57.728      -9.4758      107.69
  6:         40.57       33.437      -65.991       -40.57      -33.437       200.1
  7:         46.44       24.768       315.79        46.44       24.768  1.4433e-14
```

Figure 6.2: The text output box.



Results Controls
- [✓] Generate standard tabular output
- [✓] Plot output as a line graph
- [ ] Generate contour plot of output
- [ ] Draw displaced shape

Analysis Specific Solution Controls
- [ ] Only calculate transfer functions
- [ ] Only calculate eigenvalues and vectors
- [ ] Use orthonormal mode shapes

Additional Text Output with Solution
- [ ] Print Global Matrices
- [ ] Print Material Usage Summary
- [ ] Print Debugging Output
- [ ] Print Details Output

Contour Specific Controls
- ● Contour element stress values
- ○ Contour nodal displacement values
- [✓] Histogram equalize contour image
- [✓] Draw element outlines on contour image
- 1 Contour component

Wireframe Drawing Controls
- X rotation = 40
- Y rotation = 20
- Z rotation = 0
- [✓] Underlay undeformed structure
- 10.0 Displacement Magnification
- 0.2 Z Distance Scaling

Figure 6.3: The solution control box.

For more control over what gets generated whenever you solve a problem you can use the output control dialog box (figure 6.3), available by selecting **Controls** from the **Solutions** menu. The toggle buttons and controls in this dialog box control all of the available solution and output options, both text and graphical.

The toggles in the top-left corner determine the basic type of output that you would like to generate – tabular (available for all problem types), line graphs (available for transient and spectral problems), color contour plots (for static problems with planar elements), and wireframe drawings (for static problem with any type of element). If appropriate, multiple toggles can be selected for any given problem solution.

The next grouping of toggles are specific to spectral and modal analysis and control exactly what types of results get generated during those analyses.

The final grouping of toggle buttons on the left side of figure 6.3 allows for the inclusion of ancillary information in the tabular text-based output. Not all of these toggle buttons will have an effect on solutions of every analysis type – for instance, global matrices are not available in the solution of a nonlinear problem and the static analyses currently do not print any details.

6.3 Text output

The text output from *WinFElt* is the most basic form of output that all applications in the FElt system produce. That is, no matter how you specify and solve a FElt problem (either with *felt*, *velvet*, or *WinFElt*) the same kind of text output is available. Chapter 5 details the interpretation of this basic output form.

6.4 Graphical output

6.4.1 Contour plots

The controls for color contour plots are available on the right side of the main controls dialog (6.3). The two radio buttons control whether element stress or nodal displacements are used as the basis for the interpolation. For stress plots, the interpretation of the component number is element specific and must be a valid index in the stress vector for each element type in the problem. Consult Table 5.1 for details on what the stress vector consists of for each element type. For displacement plots, the component is simply the displacement DOF that you want to see plotted. In general, this should be one of the active global DOF for the current problem.

Other contouring controls include toggles for histogram equalization and element boundary overlay. Histogram equalization is a standard technique in image processing for enhancing the contrast of images. If the overlay elements toggle is checked then the outline of all the elements in the problem will be drawn in black on top of the color image. An example of stress contours (rendered here in greyscale) for the wrench example supplied with *WinFElt* is shown in figure 6.4. Note that both histogram equalization and element overlay were enabled for this plot.

6.4.2 Line graphs

6.4.3 Wireframe drawings

Most of the controls for wireframe drawing are specific to three-dimensional visualization. The three sliders control the rotation of the drawing about the three spatial axes. The magnification determines the multiplicative factor by which nodal displacements will be increased before the nodes are plotted. z scaling controls the front to back aspect ratio of the resulting plot.

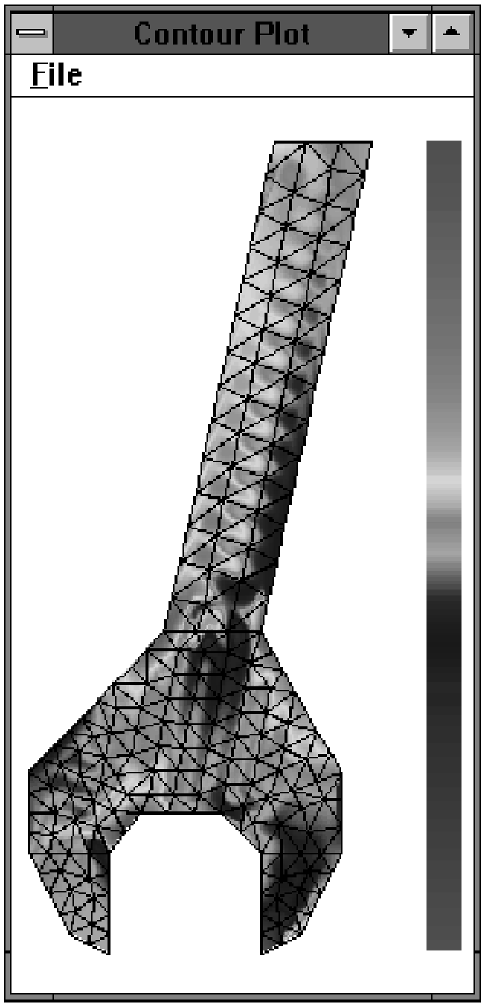


Figure 6.4: A color contour plot showing the principal stress component in a static problem using constant strain triangular elements

.

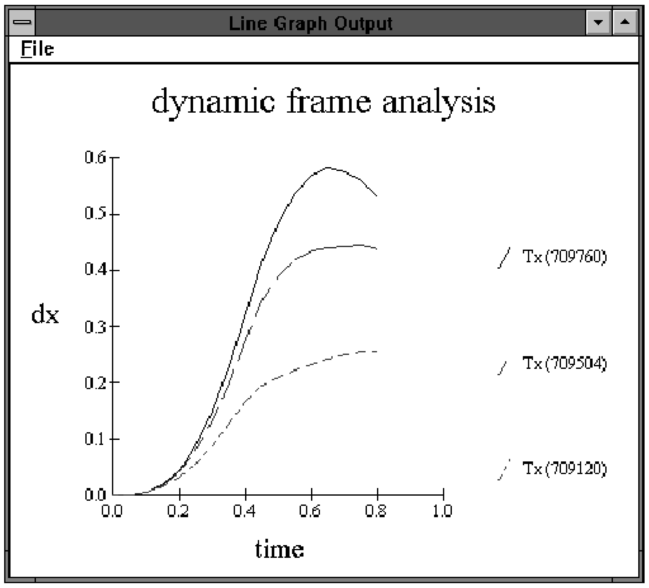


Figure 6.5: A time-displacement plot in *WinFElt*

.

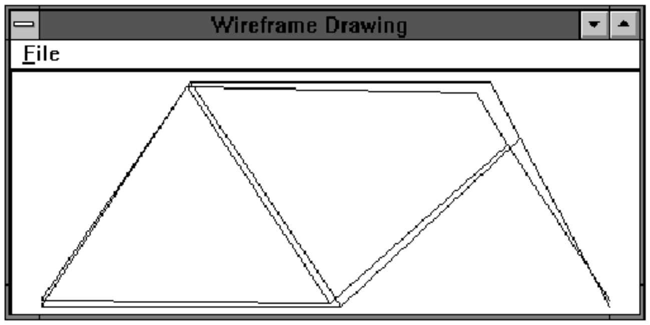


Figure 6.6: An example of a displaced structure plot.

i

Chapter 7

The *velvet* Application

7.1 Introduction to the *velvet* GUI

velvet was designed as the primary user interface to the FElt system. Generally, if the FElt system can solve a given problem, you will find it easier to specify, solve and analyze the results for that problem right within the *velvet* GUI. *velvet* embodies all of the mathematical features of FElt, as well as graphical pre- and post-processing and element generation.

The main *velvet* window consists of three major areas (Figure 7.1). The first is the list of menu buttons down the left side of the window. This is the main control area for all of the operations in *velvet*. The second is the command/status line across the bottom of the window. This line is used to display messages and for keyboard based input. The third is the drawing area, which occupies most of the window. This is the area in which you can dynamically define and interact with the FElt problem.

The menus are used to define everything about a problem (nodes, elements, forces, constraints, material properties, distributed loads), to configure *velvet*, to use the drawing tools, to configure the drawing area and to save, open and solve problems.

7.2 General features of the interface

Many of the interface modules that you will see in *velvet* look very similar to one another. Becoming familiar with a few basic operations that are common across all of these modules will allow you to use *velvet* much more easily and flexibly.

In general, clicking an **accept** button will register the current state of a dialog with the

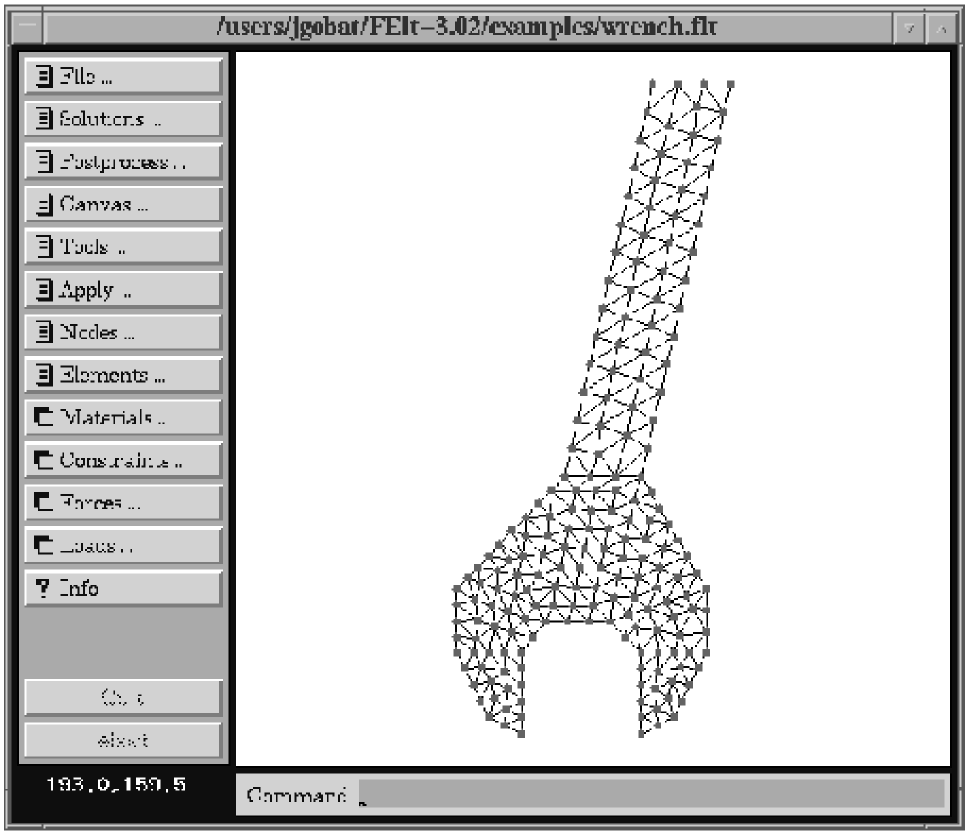


Figure 7.1: Main *velvet* drawing area with an interesting sample problem.

rest of the application. Say you have a dialog with a few toggle buttons and a couple of entry blanks. You could change the button states and type things into the entry blanks all you wanted without actually affecting the rest of *velvet*. Once you click **accept**, however, all of your changes become effective. This behavior is similar to an **okay** button in many GUIs; the difference is that **accept** will not cause the dialog box to disappear. You will need to click the **dismiss** button to do that.

Where there is a **help** button, you can click and hold it to view a brief help message about that dialog. In some dialogs, additional help is available by clicking and holding over special labels. This type of help will be discussed in more detail later.

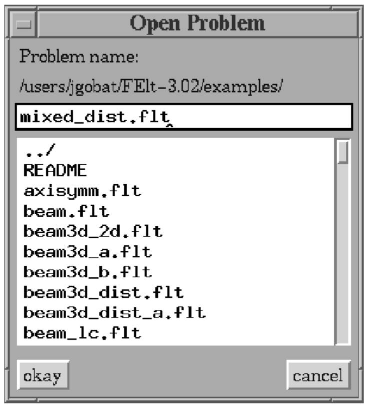
A final factor that is common across GUI components is the idea of multiple mechanisms. In general, there are three different ways that you can perform an action in *velvet*. The first is by the standard point, click, and drag operations of the mouse. The second is through keyboard shortcuts. The third is by entering commands or data into the command window at the bottom of the main drawing area. For the most part, we will limit our initial discussion to the mouse operations. Section 7.10 presents a summary of the equivalencies between the different interface methods for those of you who want to become *velvet* power users. For now, it should be enough to know that whenever *velvet* prompts you for a coordinate you can either click in the drawing area or type in an `x,y` pair in the command window (followed by a $\boxed{\text{return}}$ of course). Similarly if you are being asked to select an element or node; you can either type in the appropriate number or click on the appropriate object.

7.3 Working with files

If you already have a FElt input file defined, the easiest way to start working with that file is by specifying the filename on the shell command line when you invoke *velvet*. For example,

```
    % velvet foo.flt
```

Will start *velvet* and automatically load in the file `foo.flt`. The drawing area will be initially configured to some reasonable values to allow you to view the entire problem. If this file was saved from *velvet* then it will contain special sections that describe all of the figures and the configuration of the drawing area when the file was saved. *velvet* will use this information to reconstruct all of these things exactly as you left them.



Problem name:
/users/jgobat/FElt-3.02/examples/
mixed_dist.flt

```
../
README
axisymm.flt
beam.flt
beam3d_2d.flt
beam3d_a.flt
beam3d_b.flt
beam3d_dist.flt
beam3d_dist_a.flt
beam_lc.flt
```

Figure 7.2: The *velvet* file selection mechanism.

Alternatively, you can invoke *velvet* with no filename and then use the standard *velvet* file selection mechanism to load a file. Select **open** from the **file** menu. This will bring up a dialog box that looks something like the one shown in Figure 7.2. This dialog is the standard way for you to specify filenames (either for loading or saving problem files or graphical dumps). Within this dialog you can choose a file either by typing into the entry box at the top of the dialog or by clicking on a filename in the list with the first mouse button. Clicking **okay** will finalize your selection. You can maneuver through directories by selecting them and clicking **okay** just like files. An alternative to selecting an entry then clicking the **okay** button is to click on list entries with the middle mouse button. This will instantly select that entry (or move to that directory). Clicking the **cancel** button will dismiss the file selection dialog without making a selection.

Additional entries on the **file** menu should appear familiar to most users of modern GUI applications. Choosing **new** will erase the current problem and allow you to start with a blank slate. You'll be prompted if there is a chance that you might lose some unsaved changes to the current problem. **save** does just what it says; if the current problem does not have a filename already associated with it, choosing **save** acts like **save as**, otherwise it will silently save the current state of the problem to the associated file. The current filename is always displayed as the title in the window manager title bar. **save as** allows you to name the file that the problem will be saved to. After saving through a **save as** action, this filename becomes the current filename for this problem. The **restore** option is equivalent to selecting **open** and choosing the current input file (i.e., it simply reloads the currently named input file). **exit** quits *velvet* entirely; you will be prompted if you have unsaved changes.

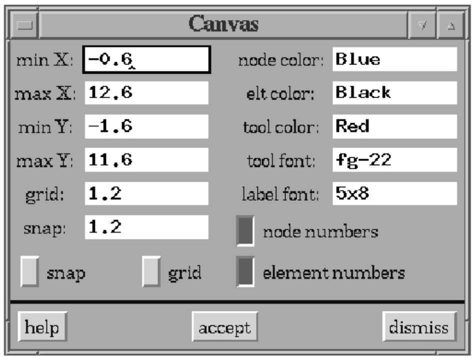
Note that when *velvet* saves a file, it will consist of the standard sections of a FElt file plus special sections that describe the appearance of the drawing area. These ancillary sections contain information that allows *velvet* to reconstruct the drawing area (including tool figures, canvas configuration options and zoom state) when the corresponding FElt file is loaded back into *velvet*. You should never have to worry about whether or not such information exists. If a file with these sections is run through *felt* the information simply will be ignored. If the information does not exist and you load the file into *velvet* then *velvet* simply will make a best guess as to how to configure the main canvas area. The advantage to having such a plain text description of the drawing area is that you can change this information outside of *velvet* if you want to make quick changes to the problem appearance but do not want to load *velvet* explicitly. Such an arrangement also allows you to easily create defaults files which you can use to start-up *velvet* with your preferred configuration.

7.4 Configuring the drawing area

7.4.1 Basic controls

The primary means for setting up the drawing area is through the **configure** dialog under the **canvas** menu (Figure 7.3). The left side of this dialog deals primarily with the minimum and maximum coordinates of the drawing area, the snap size and the grid size. The right side allows you to specify the colors and fonts to use for the different objects that appear in the drawing area. The label font is used for node and element numbers. Also included on the **configure** dialog are toggle buttons to control the status of snap, grid, element numbering and node numbering. Note that these four toggles are also available by selecting the appropriate entry right on the **canvas** menu. You must click **accept** to make any of your changes active.

As mentioned above, the **snap** and **grid** options act as toggle switches for these common drawing aids. For those unfamiliar with these terms, **grid** produces a ruled grid across the drawing area to help you locate yourself as you work and **snap** insures that mouse selected coordinates will fall on regular fractions (i.e., a snap size of 0.25 insures that all of your selections will be rounded to the nearest quarter unit). Note that the grid size (the spacing of the rulings on the screen) and the snap size (the fraction to which your selected coordinates will be automatically rounded) are controlled separately by entries in the **configure** dialog. The **node numbers** option acts as a toggle switch for node numbering - in a crowded problem, visible node numbers may not be helpful as they can tend to clutter the display.



```
min X: -0.6      node color: Blue
max X: 12.6      elt color: Black
min Y: -1.6      tool color: Red
max Y: 11.6      tool font: fg-22
grid: 1.2        label font: 5x8
snap: 1.2        node numbers
snap    grid     element numbers
```

Figure 7.3: The configuration dialog box.

The same is true for the **element numbers** option. A check mark will appear next to the menu entry if the option is enabled.

7.4.2 Object coloring

Beyond the basic node color and element color controls provided in the canvas configuration dialog (Figure 7.3), *velvet* also allows you to assign unique colors to individual forces, constraints, material properties and distributed loads. Selecting **color control** from the **canvas** menu will pop-up the colors dialog shown in Figure 7.4. The four smaller scrolled lists on the left list all of the attributable objects (materials, distributed loads, constraints, forces; see section 7.5) defined in the current problem. The larger list running down the right side lists the available colors. By highlighting an entry in one of the four object lists you can assign a color to that object.

In Figure 7.4, we see from the highlighted entries that the color for the material steel is red. Any elements with material property steel will be rendered in red on the main drawing canvas. If we wanted to change the color for steel to yellow, we could simply click on yellow in the colors list. There is no **accept** button on this dialog; color assignments take effect as soon as you select a color from the colors list. For any color changes to be reflected in the drawing area, however, you need to click on **recolor** on the bottom of the color control dialog. Because recoloring the drawing area can be computationally intensive it is a good idea to make all your color assignments first and then select **recolor** only once when you are finished.

The precedence of coloring, from highest to lowest, is force, constraint, default for

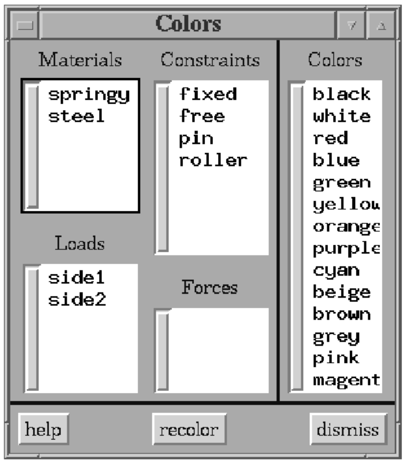


Figure 7.4: The object coloring control box.

nodes and distributed loads, material, default for elements. What this means is that if a node has a force applied and that force has a color assigned then the node will be colored according to the force color. If the force does not have a color assigned, but the constraint for that node does have one then the node will be colored according to the constraint color. If neither force nor constraint has a color assigned, then the node will be colored according to the standard node color defined via the regular canvas configuration dialog (Figure 7.3). Element coloring works the same way.

7.4.3 Zooming

A model with a dense mesh can often be difficult to work with because it is difficult to position the cursor exactly on top of a given node or element. The way around this problem is to use the zoom controls to zoom in or out on a given part of the mesh. If you want to zoom in on a selected part of your drawing just use **zoom window** from the **canvas** menu. After selecting **zoom window** you can can define a bounding box by typing coordinate pairs in the command window or clicking and dragging out a box with the mouse. Once you release the mouse button or enter the second `x,y` pair, *velvet* will rescale the drawing such that the portion you selected will fill the entire drawing area.

Selecting the **zoom to fit** option under the **canvas** menu will rescale the drawing area such that the full extent of the problem can be viewed in the current window. If you want to move around a zoomed drawing you can either zoom out and zoom in on a new section (and repeat this as needed) or you can use the scroll bars on the left and bottom sides of the drawing area to move around a zoomed canvas.

7.4.4 Dumping the drawing area

Use the **save canvas** option under the **canvas** menu when you want to save a “snapshot” of the drawing area. Note that this will not be a dump of the entire *velvet* window as was shown in Figure 7.1. The drawing area only will be saved either as an XWD or a standard PostScript file. (Figure 7.11 illustrating the types of drawing tools available in *velvet* was dumped with the **save canvas** option.) You select the filename through the standard file dialog. Extra toggle buttons near the bottom of the dialog control whether the saved file will be in XWD or PostScript format. This file could then be rendered for a presentation or for inclusion in a report. Additional drawing tools (see section 7.7) are available which allow you to put a title, annotation or additional graphic objects in the drawing area in addition to the standard presentation of nodes and elements. Note that because of the nature of XWD’s, you need to make sure that no window is covering any part of the drawing area whenever you do a dump in XWD format. This includes the file selection dialog; make sure you move it out of the way before you click **okay**.

7.5 Drawing and defining a problem

Starting with a blank slate (either by invoking *velvet* without an input file or selecting **new** from the **file** menu) you need to do several things to completely define a new problem. Say for instance that you want to define the four node, three element, beam and truss example problem considered earlier (Section 3.3, Figure 3.3). This problem uses two different distributed loads, four different constraints, two element types and two materials. All of these things need to be defined in order to properly solve the problem. The material properties must be defined before elements are added because an element must have a material assigned to it. Likewise, at least one of the constraints must be defined before nodes can be added because nodes must have a constraint assigned. In general, the procedure is to setup whatever objects the problem might require (loads, forces, materials, constraints) before adding nodes and elements (generally nodes come first, since elements have to attach between them).

7.5.1 Defining attributable objects

The way that you work with these attributable objects (forces, loads, materials, constraints) is by setting one of each as the *active* object of that type. This is particularly important for materials and constraints. When you add a node it will automatically be assigned the active constraint and when you add an element it will automatically be assigned the active material. If you want to set the load on an element or the force on a node, or change the constraint on a node or material on an element, you can use the **apply** menu. For instance, choosing **loads** from this menu will allow you to apply the currently active distributed load on an element of your choosing. To apply an object simply choose the appropriate entry and then click on the nodes or elements which you wish to apply it to. *velvet* will keep prompting you for additional objects until you click on **quit** or **abort**. If you wish to apply an object to a group of nodes or elements (say for instance you wanted to apply a self-weight load to all elements in the problem) you could simply drag out a bounding box by clicking and holding the second mouse button. If you are applying forces or constraints, every node in this box will be assigned the active object of that type. If you are applying materials or loads, every element completely within the box will be assigned the active object.

You set the active object of a given type by selecting it in the appropriate dialog box. These dialogs can be popped-up by clicking on the **materials**, **forces**, **loads** or **constraints** buttons in the main *velvet* control area. A picture of the constraint dialog after defining all the constraints in our sample problem is shown in Figure 7.5. Several of the features that you see here are exactly the same in the other three dialogs. Each has a name field and a list of all the defined objects of that type. The six buttons (**help, accept, dismiss, delete, new, copy**) across the bottom are also present in all four dialogs. The main area to the right of the list and name is used for object specific definition.

Selecting an entry in the list will display the definition for that object and it will instantly make that object the active definition for that kind of object. It would remain active even if you were to dismiss the dialog box. You can use the name field to either set the name when you are defining a new object or to rename an already defined object. For example, if you bring up the material dialog in a brand new problem, you will be faced with a completely blank slate. You can name your first material `aluminum`, define some material properties and click **accept**. This would register `aluminum` as an available material and set it as the currently active material. Now you want to define another material property, say `springy`. You have two options. Clicking **new** will clear out all of the property fields and the name field and you can name the new material `springy` and define its properties. Alternatively, you can click **copy** and only the name field will be cleared. This is simply a convenient way

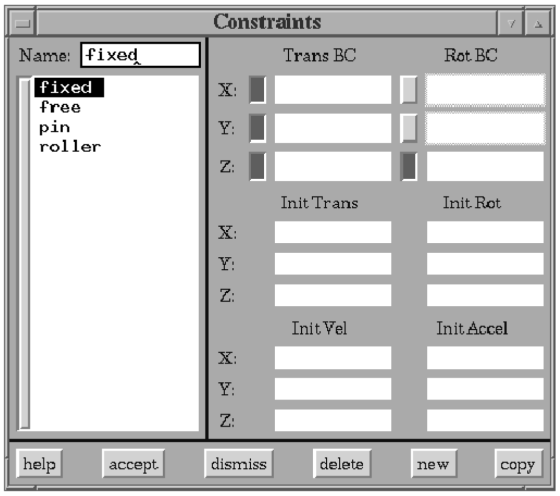
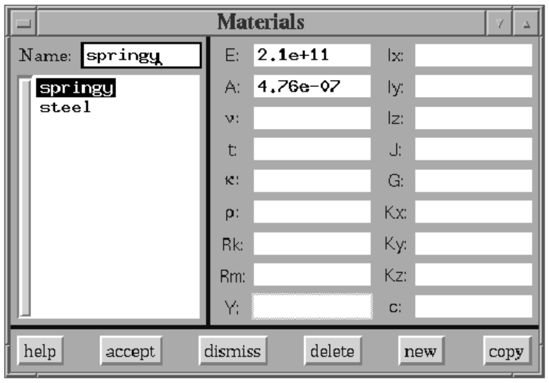


Figure 7.5: The constraint dialog box.

to define a new material with properties similar to the previous material. All you have to enter are a new name and any changed or additional material properties. In either case, you need to click **accept** to register `springy` as a new material. If you still want `aluminum` as the active material, you simply have to select `aluminum` in the list after you have accepted `springy`. Finally, if you really meant to refer to `aluminum` by the name `steel`, you can simply change the name in the field above the list and click **accept**. It is important to note that the basic behavior of these buttons is exactly the same in all four dialogs. Only the object specific information (discussed in more detail below) in the right side of each behaves differently.

To delete an object you can simply click **delete** in the appropriate dialog box. Note that *velvet* will not let you delete an object that is currently assigned to an element or node. Clicking and holding the **help** button will display a brief help message for the given dialog box.

The advantage to this mode of working with objects is that these dialogs can be left up throughout your work session. You can have them up and easily change the active object for a given type. There is no need to be constantly popping things up, selecting something or changing something and then popping it down again only to need it again in 30 seconds. If you're sure you won't be needing them after some initial setup, you can safely dismiss them if you don't like too much screen clutter.



Name: springy

springy
steel

E: 2.1e+11

A: 4.76e-07

Figure 7.6: The material dialog box.

7.5.1.1 The material dialog

The object specific information for a material property is simply entered through the labeled text boxes to the right of the material list (Figure 7.6). Additional help regarding each property, including a list of which elements require that property to be defined, can be gotten by clicking on the label for the given property (i.e., click on `E` for a description of Young's modulus). If a field is left blank, that property will be taken as 0.0.

7.5.1.2 The constraint dialog

The six toggle buttons shown in Figure 7.5 allow you to specify which DOFs are constrained by a boundary condition. If a toggle button is engaged, the corresponding text field will also be used in constructing the boundary condition definition. If a button is engaged and the text field is blank, that DOF will be fixed; if a button is engaged and that field contains the word `hinged` then that DOF will be treated as a hinge (remember, hinged DOFs only make sense on certain element types); if a button is engaged and the text field contains a number, then that DOF will be defined as a displacement boundary condition; if a button is engaged and the corresponding text field has a time-dependent expression then that DOF will be treated as time-varying BC. Finally, if a button is not engaged, then that DOF will remain completely unconstrained.

Initial conditions for transient analysis problems are defined with the initial displacement, velocity, and acceleration entries. For initial displacement and velocity conditions empty entries are taken as 0.0. For initial acceleration conditions, empty entries are un-

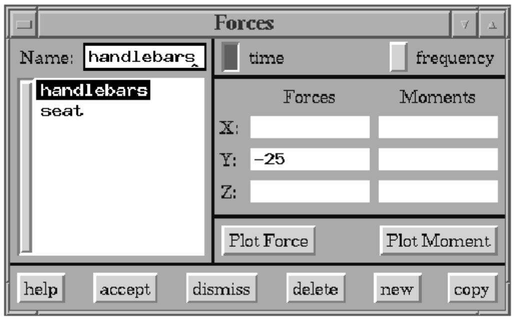


Figure 7.7: The force dialog box.

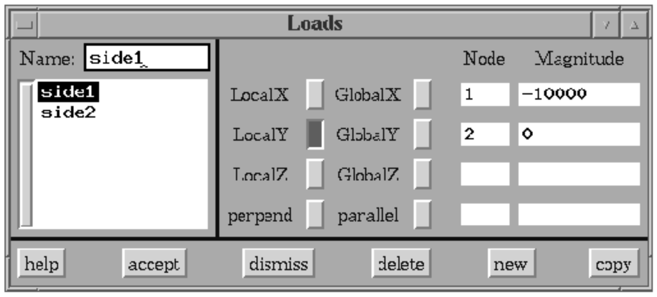
defined. The distinction is that if all of the initial accelerations are undefined then the mathematical engine will solve for an initial acceleration vector based on the initial displacement and velocity vectors. If any of the initial accelerations are specified in a problem then any others which were left undefined will be taken as 0.0 and the mathematical engine will use this vector as the initial acceleration.

7.5.1.3 The force dialog

The two toggle buttons at the top of the dialog toggle the display of force information between the time domain forces (`Fx ...Mz` in a FElt file) and frequency domain spectral inputs (`Sfx ...Smz` in a FElt file). Forces and moments (or spectra of forces and moments) in each of the three directions can be entered in the appropriate text field. If an entry is left blank that component of the force will be taken as 0.0. The dialog pictured in Figure 7.7 is from a model of a bicycle in which there are two static forces defined. Just like in a FElt file, forces can be a time- or frequency-dependent function using the independent variables `t` and `w`.

7.5.1.4 The load dialog

Figure 7.8 shows what the load dialog box looks like after we have defined both of the distributed loads used in our sample problem. The eight toggle buttons define the direction of the load. Note that these toggles function as a radio group, i.e., only one of them can be selected at any one time. The text fields allow you to enter node, magnitude pairs much the same way that you would define a load in a standard FElt input file. Remember that the node specification refers to the local element number (i.e., 1 or 2 for a beam, 1, 2 or 3 for a



Name: side1

side1
side2

Node	Magnitude
1	-10000
2	0

Figure 7.8: The load dialog box.

CST, etc.) Also, it is always your responsibility to make sure that the loads assigned to an element have a valid direction and node ordering for that element type.

7.5.2 Working with nodes and elements

The first step in actually laying out the geometry of a problem is usually to lay the nodes out (elements have to be attached to nodes after all). Let's assume that we went through everything above and defined all the materials, constraints, and loads in our sample problem (there are no forces). We made `steel` the active material and `free` the active constraint (we'll deal with the loads later). Now to add a node all we have to is select **add** from the **node** menu. You choose the location of the node simply by clicking in the drawing area. A node will be created at that location (or the nearest snapped location if snap is on). Alternatively, you can specify the exact coordinates within the command window. You do not need to select **add** for each individual node. Once selected, you are effectively in add node mode and can simply add additional nodes by repeatedly clicking or typing coordinates. To stop adding nodes click on the **quit** or **abort** button in the main control area. Most other command functions available from the main control area will be greyed out while you add nodes. In our sample problem all we need to do is click on the location of our four nodes. Nodes will be numbered consecutively in the order in which they were added.

Once the nodes are down, we need to remember that all four have the constraint `free` assigned to them (it was active when we added them). So now we change the active constraint to `pin` by clicking it in the constraint list, select **constraint** from the **apply** menu and click on node 1. We make node 3 a `roller` by selecting `roller` on the constraint list and applying it to the node via the apply mechanism. Similarly for the fixed condition on node 4. We could have gotten around this by simply changing the active constraint before we added each node. All we would have needed to do was leave the constraint dialog up and click on the appropriate entry in the constraint list before we added the node that used that constraint.

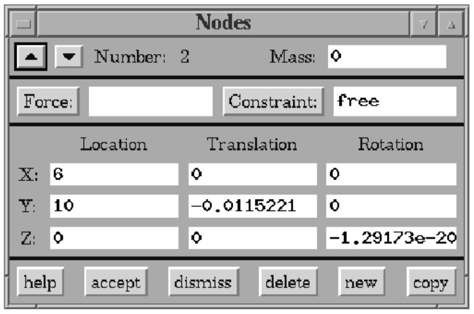
Before we can add the elements, we need to know how to set the element type. Like an active object, we have to set a current element type before we can actually add any elements. There is a distinction, however, as we cannot go back and apply a different element type to an element. The only way to change the type of a given element is to delete it and add a new element. We set the current element by choosing **set type** from the **elements** menu. This will bring up a dialog similar to the standard file selection mechanism. Like the object dialogs, the element selection dialog can be left up throughout your work session. The current element type can be set simply by clicking on the appropriate name in the list. The **accept** button is simply there in case you want to type in the element name. Typing `beam3d` in the text field and clicking **accept** is equivalent to simply clicking on the `beam3d` entry in the list.

Given this mechanism, the logical way to add the elements in our sample problem is to set the current element type to `beam` and add elements 1 and 2. Elements are added by selecting **add** from the **elements** menu and then clicking on nodes in the appropriate order, or entering node numbers in the command window. *velvet* will wait until you have selected the appropriate number of nodes for the current type of element before it prompts you to add the next element. Click **quit** to stop adding elements. Click **abort** if you goofed up on an element and just want to start over for that element (e.g., if you entered the first two nodes wrong on a CST element).

Once elements 1 and 2 are added we can change the element type to truss and add element 3 by clicking on nodes 2 and 4. Before we did this we would probably also change the active material to `springy` (or we could just change it with **material** from the **apply** menu later).

We also still need to apply the distributed loads. All we have to do is set `side1` active by clicking it in the load dialog and apply it to element 1 from the **apply** menu. Then we do the same thing for `side2` and element 2.

Deleting nodes and elements (via **delete** under the appropriate menu) must proceed in reverse order of adding nodes and elements. That is, a node can not be deleted if an element is still attached to it, so the element must be deleted first. Elements can be deleted freely. Deleting multiple nodes and elements is accomplished by clicking on one after the other (**quit** or **abort** will finish deleting) or by clicking and holding the middle mouse button and dragging out a box containing the nodes or elements to be deleted. As in deleting one at a



Number: 2 Mass: 0

Force: Constraint: free

	Location	Translation	Rotation
X:	6	0	0
Y:	10	-0.0115221	0
Z:	0	0	-1.29173e-20

Figure 7.9: The node information and editing dialog.

time, none of the selected nodes which are still attached to elements will be deleted.

7.6 More on nodes and elements

7.6.1 Editing nodes

In addition to simply adding and deleting nodes, *velvet* provides a powerful mechanism for displaying and editing all of the information about a node. The node dialog box is shown in Figure 7.9. You can raise this dialog either by selecting **edit** from the **node** menu or by clicking directly on the node which you want to edit with the left mouse button. You may want to use the former option (even though it seems a bit clumsier) in cases where you can't quite click on the right node. If you proceed from the menu option, you will be prompted to either select the node with the mouse or to type the node number into the command window. Once the dialog is up you can change the displayed node either with the arrow keys in the dialog box, by clicking on a different node in the drawing area or by selecting a new node through the **edit** option under the **node** menu. As a third option for editing nodes you can click on a node with the third mouse button – this will raise the node editing dialog and the constraint and force (if appropriate) dialogs with the assigned objects for the selected node already highlighted. This is an easy way to take a complete look at a given node; remember, however, that any changes you make to the node's objects will also affect any other nodes that have that object assigned to them.

The information available in the node dialog are the exact nodal coordinates, all six nodal displacements (valid only after a problem has been solved), the lumped mass for the node, the name of the constraint applied to the node and the name of the force (if any) applied to the node. The location, lumped mass, constraint and force can be modified simply by editing the information in this dialog box. To change the applied constraint, you can enter a new name into the text field or you can click on the **constraint** label to pop-up a menu that will let you choose from a list of the currently defined constraints. The same is true for forces by clicking on the **force** label. Note that while you can change the applied force on a node via the **force** option under the **apply** menu, editing the information in this dialog is the only way to completely remove the applied force on a node. Just like the object dialogs you need to click **accept** to make your changes effective. The rest of the buttons on the bottom of the node dialog also behave exactly the same as the buttons on the object dialogs.

velvet provides two ways for you to change a node's location. The first is by changing the location fields in the node dialog. The second is by selecting **move** from the **node** menu. You can then select the node that you want to move by clicking on it, move to the new location and click again to put it back down. Alternatively, just like you can click the first mouse button on a node to edit, you can click on a node with the second mouse button to pick it up, move to a new location and click again with the second mouse button to put it back down. Note that because *velvet* is basically a package for working with two-dimensional problems, you cannot change the z coordinate of a node in the coordinate entry fields of the node dialog.

7.6.2 Automatic node renumbering

velvet gives you two ways to optimize the node numbering in terms of reducing the profile of the global stiffness matrix. The first way is to select **renumber perm** from the **nodes** menu. This will actually rearrange all of the node numbers in the current problem - permanently. Generally you would choose this option where the node numbering was not particularly important to your own understanding of the problem. This might be the case if the elements had been automatically generated. If, on the other hand, you have a paper sketch of the problem which you used to setup the problem then you may not want to lose that particular node numbering scheme.

In this latter case what you can do is toggle temporary renumbering by selecting **renumber temp** from the **nodes** menu. This toggle is equivalent to the `-renumber` command line switch in the command line application *felt*. If this switch is on, the nodes will be renumbered internally during the computations, and then restored to their original numbers before you actually see any output, i.e. the output will reflect the original numbering scheme.

Both renumbering methods have their advantages. The permanent scheme reduces over-

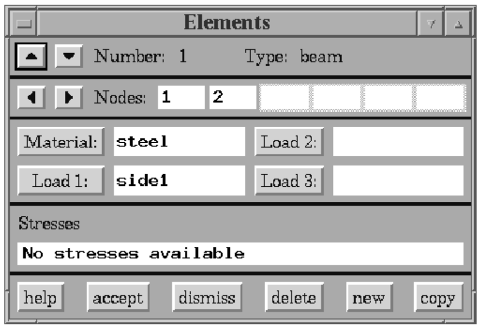


Figure 7.10: The element information and editing dialog.

head if you will be solving the problem numerous times; do it once and then why waste the time to figure out that it can't get any better every time you try to solve it. The temporary, internal only scheme keeps the problem and the results referenced the way that you had originally defined it. The additional overhead of doing the renumbering each time can be well worth it for large problems - both in terms of memory requirements and in solution speed. For both options, the same caveat applies as in the discussion of the command line switch in *felt* - for small problems, the algorithm probably won't be able to do much if anything in terms of profile reduction; in these cases it is probably easiest simply not to worry about either of the renumbering options.

7.6.3 Editing elements

Editing elements is very similar to editing nodes. You can either select **edit** from the **elements** menu (particularly in cases where you will want to choose the element by typing in its number) or click on an element (or its number) with the first mouse button to display the element dialog (Figure 7.10). Like the node dialog you can change the displayed elements either with the up and down arrow buttons or by clicking on a different element with the first mouse button. Also just like nodes, you can click on an element with third mouse button to raise the element dialog and the object dialogs (materials and loads) with the objects assigned to that element already highlighted. Again, remember that any changes to the assigned objects can affect other elements as well as the element currently displayed in the element dialog.

The information displayed in this dialog includes the element type, the nodes to which this element is attached, the material property and distributed loads assigned to this element and, if the problem has been solved, the stress results for this element. If an element uses more than six nodes, you can use the side to side arrow buttons next to the node list to scroll through them. The dialog gives you a simple mechanism for changing the nodes, material and loads for a given element. Editing the material and loads is just like the constraint and force fields in the node dialogs. You can either type into the text field or click on the appropriate label and choose from a menu of currently available objects. Like forces in the node dialog, changing the information in the element dialog is the only way to completely remove all the loads from an element. Once again of course, you need to click **accept** to make any changes effective. The other buttons across the bottom and their functions should be very familiar to you by now.

7.7 Using tools

The drawing tools are intended to be aids for meshing up or annotating a problem. For instance, you can draw a circle by selecting **circle** from the **tools** menu and then place nodes along the circle. The actual figure on the screen does not affect the solution of the problem in any way. *velvet* will prompt you in the command window for the necessary coordinates for each figure type. As usual, you can choose coordinates either with the mouse or by typing in the command window. In general, however, you must enter all of the coordinate locations for a given figure with the same input method; what this means is that if you enter the coordinates of the first point of a line with the keyboard then you will have to enter the second point with keyboard input as well or, alternatively, if you select the center of a circle with the mouse then you will have to select the radius with the mouse as well.

Titles and comments can be entered in the drawing area by selecting **text** from the **tools** menu option. As mentioned earlier, the drawing area could then be dumped to an XWD and the result could serve as documentation or a figure in a presentation, complete with color annotation. Figure 7.11 shows such a dump with examples of the available tools. Tool color and font is controlled from the **configure** option under the **canvas** menu. In addition to text and circles, the *velvet* drawing toolbox currently includes lines, arcs, polylines, and rectangles.

To draw a line with the mouse you have to click at the start point and hold the mouse button while you drag the line to the end point. Rectangles work the same way; click and hold while you drag out a box. For circles your mouse click will select a center point, and as you move the mouse while still holding the button, your circle will expand about that center. When drawing polylines with the mouse, use the first mouse button to select the

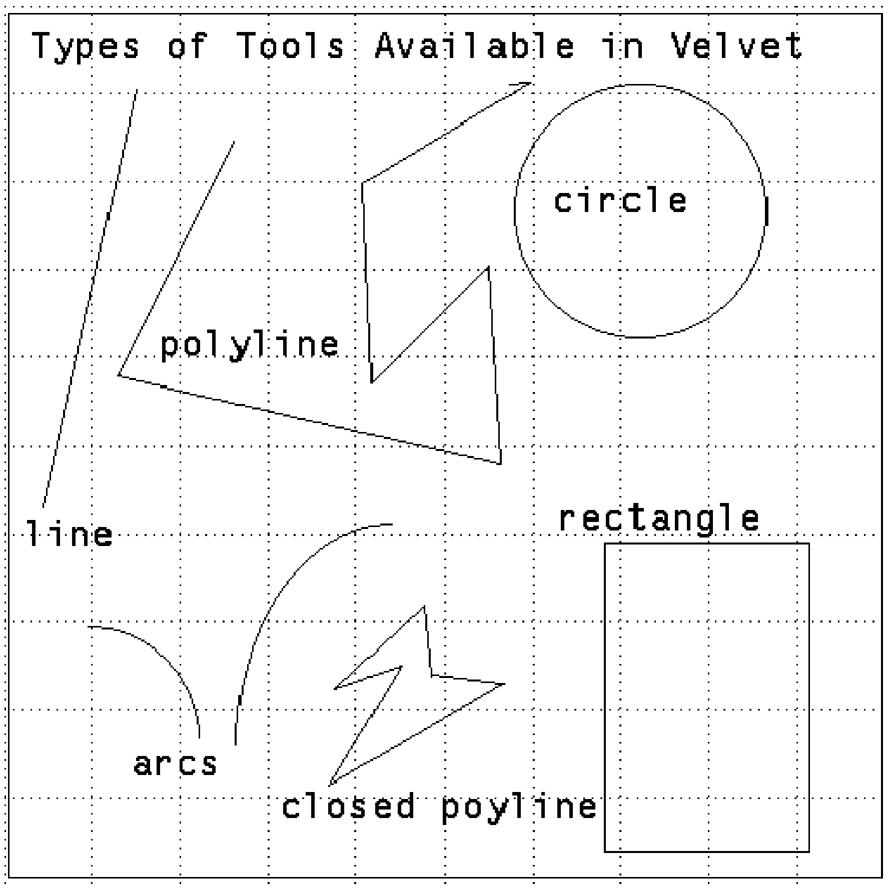


Figure 7.11: Examples of all the tools available in *velvet*.

starting point by a simple click (not a click and hold). Additional end points are defined by additional clicks of the first mouse button. You can stop adding endpoints by clicking the third mouse button. If you want a closed polyline, you can simply click the second mouse button and *velvet* will draw a line from your current position to the starting point of the polyline.

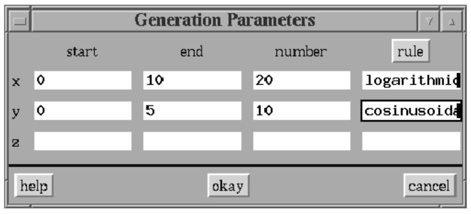
Tools can be deleted from the screen just like nodes and elements, either one at a time or with a window. Once in delete mode (select **delete** from the **tools** menu), individual figures can be selected by clicking with the left mouse button or a window can be drawn with the middle mouse button. The delete operation proceeds until **quit** or **abort** is clicked. Moving a tool figure that already is drawn on the screen is also straightforward; simply select **move** from the **tools** menu, click on the figure that you want to move and then click again once you have dragged it to its new location.

The ancillary sections in the FElt file that are created when you save a problem from *velvet* record all your tool figures. If these sections are available when you reload a problem, the tools will automatically be redrawn for you.

7.8 Material databases and defaults files

Often, you will find that certain objects or material properties are being used over and over again in different problems. Dummy FElt files can be used to make easy use of this repetitive information. Starting *velvet* with a FElt file that contains a couple of common constraint definitions means that those definitions will not need to be specified explicitly within *velvet* - they will be there for use on start-up. Similarly for other types of objects. Note that loading a defaults file from within *velvet* is just like opening a new input file, everything in the current problem will be lost. You can use the `canvas configuration` and `display list` sections of a FElt file to create a custom canvas configuration in a defaults file.

Material databases are also special cases of FElt input files. They are given special treatment within *velvet* because they are so convenient. Whole classes of material types and shapes (W-shape beams, standard diameter bars, etc.) can be stored in a material database and loaded into *velvet* at any time with the **open database** command under the **file** menu. Any changes to material properties or additions of new materials can be saved simply by selecting **update database** from the **file** menu. The current set of materials is the only thing that is retained when a new input file is loaded or a new problem is started within *velvet*.



Generation Parameters

	start	end	number	rule
x	0	10	20	logarithmic
y	0	5	10	cosinusoida
z				

Figure 7.12: The form for defining line and grid generation parameters.

7.9 Automated element generation

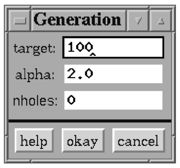
Velvet can generate grids and triangular meshes automatically. A grid is defined as an $n \times m$ array of two-dimensional line or quadrilateral elements. A triangular mesh is an arbitrary polygonal shape (possibly with interior holes) discretized into two-dimensional triangular elements. When you select **generate** from the **elements** menu, the type of generation is automatically determined from the current element type. Because both nodes and elements will be generated, both a constraint and a material property must be active. You should make sure that the active object of each type will be appropriate for the majority of nodes and elements that will be generated. This will save you from applying a lot of objects individually once the generation is complete.

7.9.1 Generating a grid of line or quadrilateral elements

Generating a grid requires that you complete the form pictured in figure 7.12. The entries in the form match parameters in the *corduroy* input syntax (see chapter 9). The x, y, and z start and end locations define the two extreme corners of the grid. The number entries set the number of elements to be generated along each direction and the rule entries set the spacing rule to be used along each axis. Rules can either be typed in or entered by giving the focus to one of the entries and selecting a rule from the pull-down menu available by clicking on the **Rule** button.

7.9.2 Generating a mesh of triangular elements

Generating a mesh of triangles requires a little more work on your part, but it is still a lot easier than meshing any sort of complicated geometry (and most simple geometries even) by hand. When triangles are being generated a form will pop-up (Figure 7.13) for you to define the parameters of the generation. The parameters are the target number of elements


Generation

target: 100

alpha: 2.0

nhole

Figure 7.13: The form for defining triangle generation parameters.

that you want to generate, the α area constraint factor and the number of holes in your generation region. The area constraint factor is defined as

$$A_{\text{elt}} \leq \frac{\alpha A_{\text{region}}}{\text{target}}, \tag{7.1}$$

where A_{region} is the total area of the generation region and A_{elt} is the maximum area of a generated element.

Once the parameters are defined, click **okay** to begin defining the boundaries of your problem. Regions are defined by putting down marker points in the drawing area (they must be put down in a counter-clockwise order). To finish the boundary, click on the **quit** button. If the number of holes was greater than zero then each hole must be defined just like the boundary was defined – by laying down a series of marker points – but in clockwise order. Each hole is finished by clicking the **quit** button. The generation process can be aborted by clicking the **abort** button. Typing shift-bkspc will delete the last marker point that was laid down, thus allowing you to correct mistakes without restarting the entire sequence. Once all the regions are defined, *velvet* will automatically begin generating the mesh.

7.10 Keyboard interface mechanisms

7.10.1 Keyboard shortcuts

As in most graphical environments, the pointer can only do so much for you; often it will be easier for you to use the keyboard. True, you could get away with using the mouse for everything, but in the long run you will work more efficiently if you learn the keyboard interface mechanisms.

There are two basic ways that you can use key presses in place of mouse clicks. The first are commonly called keyboard shortcuts. This means that rather than selecting **solve** from the **problem** menu, you can simply press ctrl-v. A complete list of these shortcuts is given in the quick reference table at the end of this section.

The second common way to use the keyboard is to move through groups of buttons and text fields with tab to change the input focus and use space to activate the button or change the toggle state. This is a very convenient way of working with dialog boxes like the object dialogs (forces, materials, constraints, loads), the file selection dialog, the node and element dialogs, the configure dialog, the triangular mesh parameter dialog and the solution dialog. When one of the dialogs has the window manager focus there are a few special keyboard shortcuts that you can use as well. ctrl-h is equivalent to **help**, return is **accept** or **okay**, esc is **dismiss** or **cancel**, ctrl-d is **delete**, ctrl-n is **new**, and ctrl-c is **copy**.

7.10.2 Command names

All of the commands that are available through menu options in the main control area are also available by typing a text command into the command window at the bottom of the main window. To use a text command simply type it into the command window and hit return. If you just type return with no command entered, *velvet* will repeat the last command that you entered. The following table lists the equivalencies for each command. The first column lists the name of the menu button in the control area, the second lists the applicable entry under that menu, if any, and the third lists the command name(s) that perform the same action. The fourth column provides additional ways that you can perform the same function, either through a keyboard shortcut or through mouse interaction with the drawing area.

Menu	Command	Text command	Other alternatives
File	New	`new`	Ctrl-f n
	Open	`open`	Ctrl-f o
	Save	`save`	Ctrl-f s
	Save as	`save as`	Ctrl-f w
	Restore	`restore`	Ctrl-f r
	Open database	`open database`	
	Update database	`update database`	
	Exit	`exit`	Ctrl-f x
Solutions	Solve	`solve`	Ctrl-v
	Animate	`animate`	
	Results/Output	`define output`	
	Problem/Analysis	`define problem`	
Postprocess	Plot stresses	`plot stresses`	
	Plot displacements	`plot displacements`	
	Plot structure	`plot structure`	
	Contouring	`contour`	
	Wireframe	`wireframe`	
Canvas	Configure	`configure`	
	Color control	`colors`	
	Zoom to fit	`zoom all`	Ctrl-Z
	Zoom window	`zoom`	Ctrl-z
	Save canvas	`dump`	
	Snap	`snap`	Ctrl-s
	Grid	`grid`	Ctrl-g
	node numbering	`node numbers`	Ctrl-n
	element numbering	`element numbers`	Ctrl-e
Apply	Materials	`apply material`	Ctrl-a m
	Forces	`apply force`	Ctrl-a f
	Constraints	`appply constraint`	Ctrl-a c
	Loads	`apply load`	Ctrl-a l
Tools	Circle	`draw circle`	Ctrl-t c
	Arc	`draw arc`	Ctrl-t a
	Rectangle	`draw rectangle`	Ctrl-t r
	Polygon	`draw polygon`	Ctrl-t p
	Line	`draw line`	Ctrl-t l
	Text	`draw text`	Ctrl-t t
	Delete	`delete tool`	
	Move	`move tool`	
Nodes	Add	`add node`	
	Edit	`edit node`	button 1 on node
	Delete	`delete node`	
	Move	`move node`	button 2 on node
	Lumped mass	`lumped mass`	
	Renumber perm	`renumber nodes`	
	Renumber temp	`toggle renumber`	
Elements	Add	`add element`	
	Edit	`edit element`	button 1 on element
	Set type	`set element`	
	Generate	`generate`	
Materials		`edit material`	Ctrl-d m
Constraints		`edit constraint`	Ctrl-d c
Forces		`edit force`	Ctrl-d f
Loads		`edit load`	Ctrl-d l
Quit			Esc or button 3
Abort			Ctrl-c

7.11 Command line options

Some of the above functionality can be enabled, disabled or altered via command line switches when you invoke *velvet*. In addition to the standard X toolkit options (`-display`, etc.) you can use any of the following to control the behavior and set the start-up condition of *velvet*. Each of these options can also be set through your Xresources using the associated resource name.

`-sensitive` Enable sensitive/insensitive menu switching. *velvet* has two major modes of operation, normal mode and edit mode. Each mode allows only certain operations to be performed. This option allows disabling of the menus controlling the inactive operations. This is the default behavior but may be inappropriate on a slow server. Associated resource: `*sensitiveMenus`.

`+sensitive` Disable sensitive/insensitive menu switching. Associated resource: `*sensitiveMenus`.

`-numbers` Enable element and node numbering on start-up. This is the default behavior. Associated resource: `*numbers`.

`+numbers` Disable the drawing of element and node numbering on start-up. On some servers the node and element numbers may take some extra time to draw on big problems and since you'll probably turn them off anyways to reduce clutter, you can use this switch to disable them from the start. Associate resource: `*numbers`.

`-nodeColor color`
Sets the color of nodes and node numbers. The default color is `blue`. Associated resource: `*nodeColor`.

`-elementColor color`
Sets the color of elements and element numbers. The default color is `black`. Associated resource: `*elementColor`.

`-toolColor color`
Sets the color of tools (rectangles, lines, circles) and interactive windows. The default color is `red`. Associated resource: `*toolColor`.

`-labelFont font`
Sets the font used for node and element numbers. The default font is `5x8`.

Associated resource: `*labelFont`.

`-toolFont font`
: Sets the font used for text drawn using the tools. The default font is `fg-22`. Associated resource: `*toolFont`.

`-cpp filename`
: substitute `filename` for the pre-processor to run on the input file.

`-nocpp` do not run the input file through the pre-processor.

`-Idirectory`
: add `directory` to the standard search path for include files in the pre-processor.

`-Uname` undefine the macro `name` in the pre-processor.

`-Dname=value`
: define `name` to be the macro `value` in the pre-processor.

Chapter 8

Post-processing with *velvet*

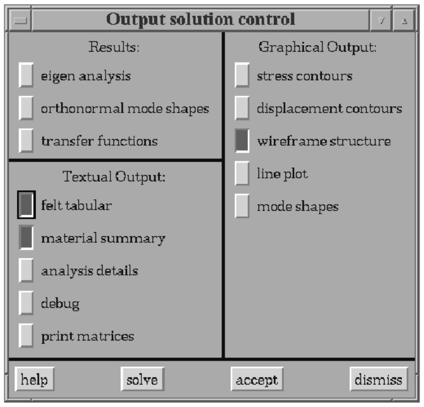
8.1 Solving a problem with *velvet*

Depending on what you want your solution to include, there are a couple of different ways to go about solving a FElt problem from within *velvet*. In addition to the standard tabular type FElt output (see section 5.3) you can also have *velvet* generate two-dimensional color shaded stress or displacement plots and/or plots of the structure with magnified displacements applied. For transient analysis problems graphical time-displacement or time-temperature plots replace the ASCII versions that *felt* produces. Transient structural analysis can also include an animation of the simulation. For spectral analysis, graphical plots of the transfer functions or ouput spectra replace *felt*'s ASCII versions. *velvet* can also draw graphical representations of mode shapes for modal analysis problems.

One way to solve a problem is to select the **solve** option from the **solutions** menu. By default, this simply generates the standard FElt output. This window will contain either the mathematical solution of your problem or the syntax errors encountered in solving the problem.

For more control over what gets generated whenever you solve a problem you can use the output control dialog box (Figure 8.1, available by selecting **results/output** from the **solutions** menu. The toggle buttons in this dialog box control all of the available solution and output options, both text and graphical.

The three toggles at the top–left of the dialog mimick three of the switches available in *felt*; they control the computations that are performed for modal or spectral analysis. The **felt output** toggle controls whether or not you want to see any of the standard textual output that *felt* would generate. If the **felt output** toggle is on then the other three textual



Output solution control

Results:
- [ ] eigen analysis
- [ ] orthonormal mode shapes
- [ ] transfer functions

Textual Output:
- [✓] felt tabular
- [✓] material summary
- [ ] analysis details
- [ ] debug
- [ ] print matrices

Graphical Output:
- [ ] stress contours
- [ ] displacement contours
- [✓] wireframe structure
- [ ] line plot
- [ ] mode shapes

Figure 8.1: The output control dialog box.

output toggles mimick command-line switches that are available in *felt* (see chapter 5) to control the printing of additional information within the textual output.

For graphical output, the toggles for stress, structure and displacement plots simply allow you to automatically invoke these post-processing options (see below) on solution. The other two options (time-displacement plots and mode shape plots) are only available during a problem solution (i.e., only by selecting the appropriate toggle in this dialog and then solving the problem). The **line plot** option generates a graphical line plot (as opposed to the ASCII plot that you would get with *felt* output) for transient or spectral results. A graphical time-displacement plot is shown in Figure 8.2. Figure 8.3 is an example of a mode shape plot. Note that *velvet* only draws a single mode at a time and that the regular problem geometry is drawn as an underlying dotted line; you can cycle forward and backward through the individual modes using the $<$ and $>$ buttons.

Given the settings in our example control box, the output for a static model of a bicycle would be the two windows shown in Figures 8.4 and 8.5. (The input file for this problem can be found as `bicycle_boys.flt` in the tests directory of the standard FElt distributions.)

You can save any of the *velvet* output by clicking **save** at the bottom of the output window. A file dialog will pop-up and allow you to name the file which you wish to save to. Depending on the type of output, you may have the option of saving in one of several different file formats. Time-displacement, wireframe structure, and mode shape plots can be saved in either XWD or PostScript format. Text output can be saved as an ASCII file. Color contours of stress and displacement can be saved in PPM or encapsulated PostScript (EPS) format.

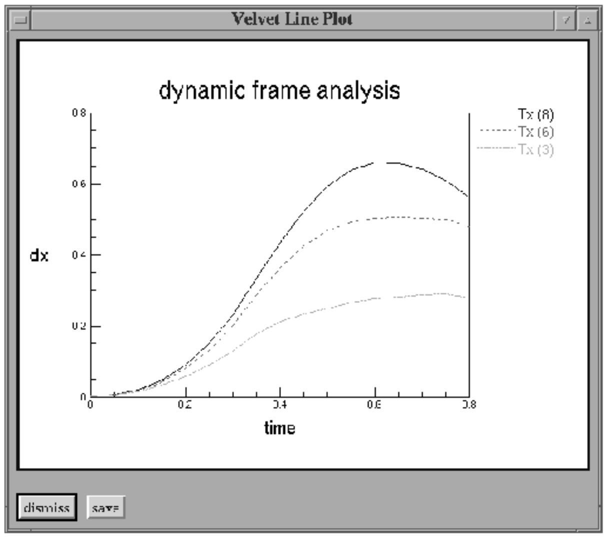


Figure 8.2: A time-displacement plot in *velvet*

.

You can leave all output windows up as long as you like. If you want to unview one at some point in your work, simply click **dismiss**.

For a transient analysis problem you can build a special case simulation for animation simply by selecting **animate** from the **solutions** menu. The solution that is constructred is special in that the displacement of all nodes in the x and y (and possibly z) translational DOF will be automatically recorded for each time step. (This is in contrast to the normal mode of solution for a transient analysis problem in which only specially requested nodes and DOF are recorded for output at the end of the simulation.) Once this complete table is built the animate window (Figure 8.6) will pop-up and you can use the playback controls to determine the speed and direction of the animation. The buttons that look like rewind and fast forward ($<<$ and $>>$) are really the speed controls. Holding them down will slow down or speed up the animation. The play backward and play forward buttons ($<$ and $>$) control which direction time moves during the animation. You can use the stop button to freeze the structure at a given point during the animation. The text indicator in the right bottom corner of the window keeps track of the elapsed time. The animation will repeat itself until it is either stopped or the animation window is dismissed.

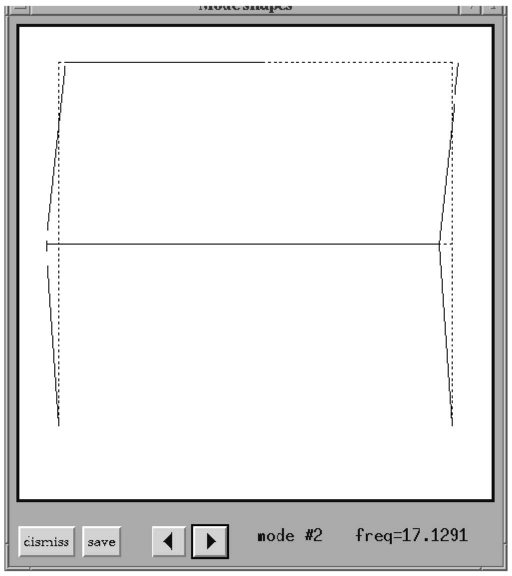
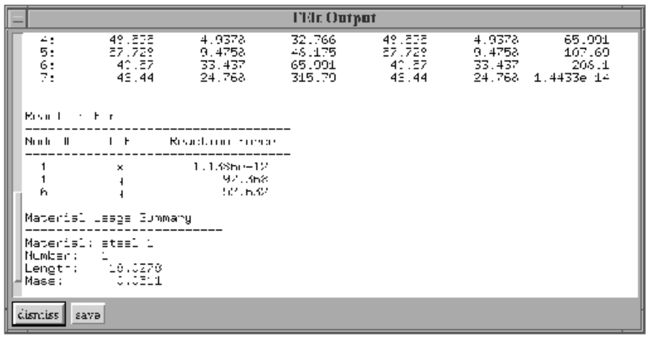


Figure 8.3: The mode shape plotting window

.



```
  4:        49.555        4.9378        32.766        49.555        4.9378        65.001
  5:        57.729        9.4758        46.175        57.729        9.4758        107.60
  6:         40.57        33.437        65.001         40.57        33.437         206.1
  7:         45.44        24.768        315.70         45.44        24.768   1.4433e 14


Reaction Forces
------------------------------------
Node #      DOF      Reaction Force
------------------------------------
  1          x          1.1486e-12
  1          y             [illegible]
  6          y             [illegible]

Material Usage Summary
--------------------------
Material: steel 1
Number:   1
Length:    10.0270
Mass:       0.0311
```

Figure 8.4: An example of textual output from *velvet*.

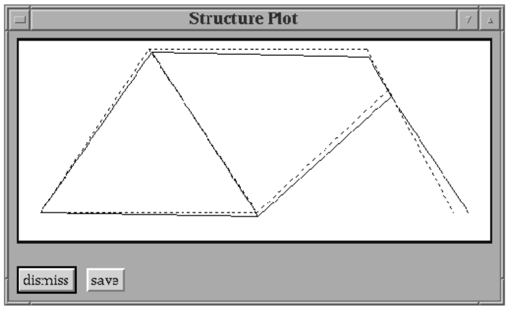


Figure 8.5: An example of a displaced structure plot.

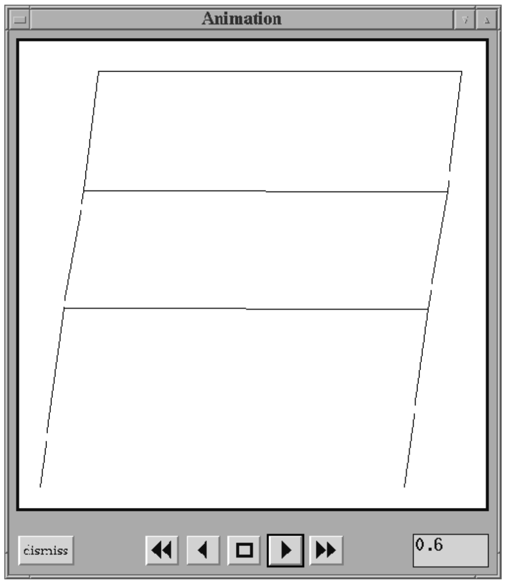


Figure 8.6: An animation in *velvet*

.

8.2 Problem description and analysis parameters

The `problem description` and `analysis parameters` section of a regular FElt input file are mimicked in *velvet* by the problem and analysis dialog box (Figure 8.7). This dialog is available by selecting **problem/analysis** from the **solutions** menu. The problem title text entry defines the header that will be used for textual output and on graphical line plots. The analysis mode is defined by engaging one of the toggle buttons just below the problem title at the top of the dialog. The mass-mode toggles below the analysis toggles allow you to select either consistent or lumped mass formulations for element mass matrices.

For a transient structural and thermal analysis problems, there are several parameters which control the numerical integration in time. The text entries down the left side of the dialog allow you to fill in values for these parameters in the γ, β, and α text entries. For both transient and spectral analysis the range of time or frequency over which to perform the computations is defined by the *start*, *stop*, and *step* text entries. Note that the value for *start* is ignored in transient analysis types. Values for the global Rayleigh damping proportionality constants are defined with the *Rk* and *Rm* text entries. Remember that if either of these values is non-zero then the Rayleigh damping for this problem will be based on these values and the global stiffness and mass matrices as opposed to elemental material values of *Rk* and *Rm* and the element stiffness and mass matrices.

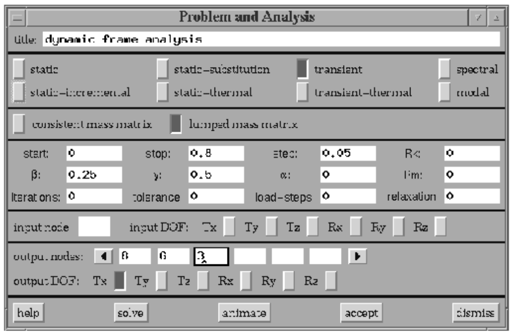
The DOF toggles and the node entries allow you to define the list of nodes and which DOF at each of those nodes that you would like to have included in the output when the problem is solved. Click on the left and right arrow keys to scroll through the complete list of nodes if you are interested in more than six of them.

The push buttons on the bottom of the dialog box are standard of course. The **solve** and **animate** buttons are simply there as a convenience; pushing either button will cause the current state of the dialog to be registered (as if you had pushed **accept**) and then the appropriate action to be taken just as if you had selected **solve** or **animate** from the main **solutions** menu.

8.3 Controlling the post-processing

8.3.1 Controlling contour plots

The controls for color contour plots are available by selecting **contouring** from the **post-processing** menu. On the dialog (Figure 8.8), there are identical controls for stress and displacement plots. On the stress side, the component specifies which stress component



Problem and Analysis

title: dynamic frame analysis

static, static-substitution, [✓ transient], spectral, static-incremental, static-thermal, transient-thermal, modal

consistent mass matrix, [✓ lumped mass matrix]

start:	0	stop:	0.8	step:	0.05	Rk:	0
β:	0.25	γ:	0.5	α:	0	Rm:	0
iterations:	0	tolerance	0	load-steps	0	relaxation	0

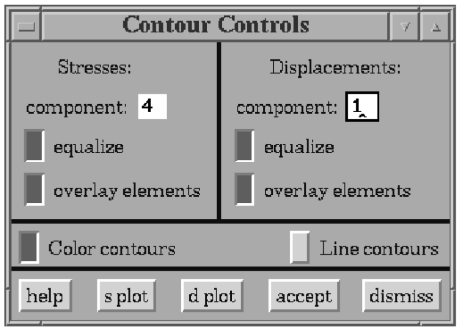
input node: | input DOF: Tx, Ty, Tz, Rx, Ry, Rz

output nodes: 8, 6, 3

output DOF: [✓ Tx], Ty, Tz, Rx, Ry, Rz

Figure 8.7: The analysis parameters dialog box

.



Contour Controls

Stresses: component: 4 | [✓ equalize] | [✓ overlay elements]

Displacements: component: 1 | [✓ equalize] | [✓ overlay elements]

[✓ Color contours] Line contours

Figure 8.8: The control dialog box for contour plots.

will be plotted. This number is element specific and must be a valid index in the stress vector for each element type in the problem. Consult Table 5.1 for details on what the stress vector consists of for each element type. For displacement plots, the component is simply the displacement DOF that you want to see plotted. In general, this should be one of the active global DOF for the current problem.

Other contouring controls include toggles for histogram equalization and element boundary overlay. Histogram equalization is a standard technique in image processing for enhancing the contrast of images. If the overlay elements toggle is checked then the outline of all the elements in the problem will be drawn in black on top of the color image. An example of stress contours (rendered here in greyscale) for the wrench example pictured earlier is shown in Figure 8.9. Note that both histogram equalization and element overlay were enabled for this plot.

In addition to the standard buttons for **help**, **accept**, and **dismiss**, you can use the **s plot** and **d plot** buttons as an alias for selecting either **plot stresses** or **plot displacements** from the main **postprocessing** menu item. Both buttons cause the dialog state to be accepted before the plot is generated.

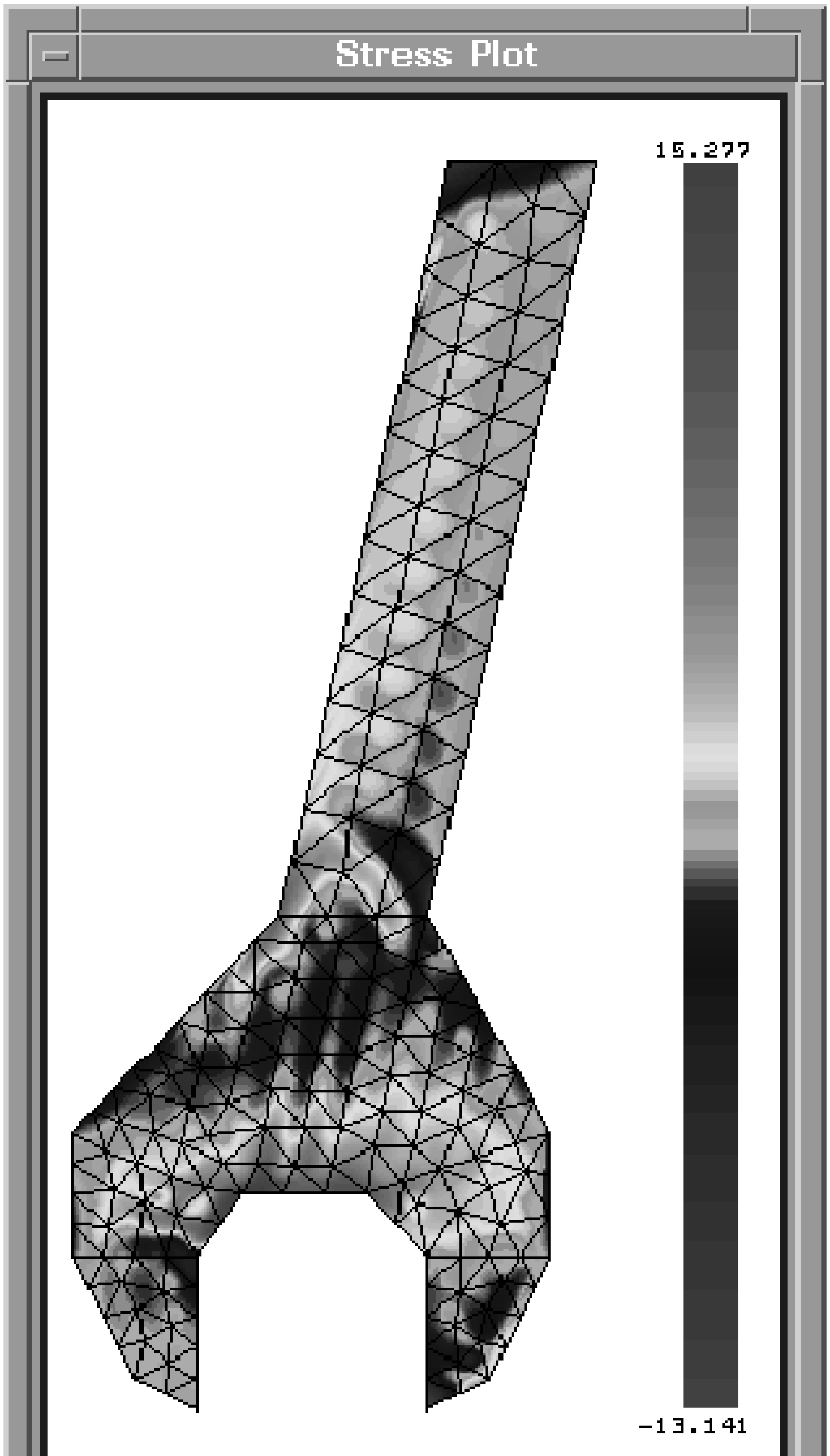
8.3.2 Controlling structure plots

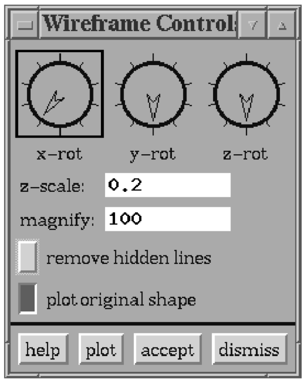
Additional control over structure plots is available through the wireframe dialog box (Figure 8.10). The majority of these controls are specific to three-dimensional visualization. The three dial widgets control the rotation of the drawing about the three spatial axes. The magnification determines the multiplicative factor by which nodal displacements will be increased before the nodes are plotted. z scaling controls the front to back aspect ratio of the resulting plot. The toggle for hidden line removal is non-functional in the current version of *velvet*. The defaults reflected in the dialog pictured in Figure 8.10 were used to generate our earlier example of the structural plot of the bicycle (Figure 8.5).

The extra button at the bottom of the dialog **plot** is equivalent to pressing **accept** and selecting **plot structure** from the **postprocessing** menu.

8.3.3 Controlling animation

The analysis parameters used in constructing a displacement table for an animation are the same as those that would be used in a normal solution for that problem in transient analysis mode, except for the nodes and DOF. As mentioned above, an animation will automatically solve for all nodes at the x and y (and z if the problem is 3d) translational DOF. The magnification of the displacements during the animation can be controlled via the same magnification control that is used in structure plots. Likewise for 3-d animations, the 3-d drawing parameters will be taken from the controls in the wireframe control dialog box.


Stress Plot
15.277
-13.141



z-scale: 0.2
magnify: 100
remove hidden lines
plot original shape

Figure 8.10: The wireframe plotting control dialog box.

Chapter 9

The *corduroy* Application

9.1 Introduction

corduroy is a command line application for automatically generating nodes and elements. It takes a text input file and generates FElt format nodes and elements for inclusion into a FElt problem. Like *felt* then, it is a command line interface to some of the same functionality that you can get in *velvet*. However, unlike *felt* and *velvet*, which share their file syntax (the standard FElt syntax), *corduroy* has its own special syntax to describe the way elements are generated.

9.2 The *corduroy* syntax

9.2.1 Specifying basic parameters

corduroy allows you to generate elements in five different ways – along a single line, as a three-dimensional grid of line elements, as a two-dimensional grid of four-node planar elements, as a three-dimensional grid of solid elements and as a two-dimensional mesh of triangular elements. Each type of generation is described in its own section; there can be multiple types and multiple sections of a given type in any given file. Besides these descriptive sections there are only a few basic parameters, all of which are optional. Order of specifications does not matter in a *corduroy* file except for numbering of generated nodes and elements and the `end` statement which must always come at the end of a file (and is not optional). As in regular FElt files, comments can be denoted with `/*` and `*/` as in the C programming language and symbolic expressions may be substituted for any non-integer value.

You can use `start-node=` and `start-element=` to specify the number for the first things that get generated. Normally, *corduroy* starts the numbering at one and then numbers sequentially as it moves from section to section of your input file. These two specifications allow you to override these starting points in situations where you may already have a few elements and nodes defined and you are going to attach the things that get generated after *corduroy* has done its work.

`constraint=` and `material=` will assign a constraint or material name to the initial node and element. This is simply a convenience; you will need to be sure to go back and actually define these objects after everything is generated and you are putting the finishing touches on the problem definition.

9.2.2 Generating elements along a line

The simplest case of element generation is a line which is divided up into an arbitrary number of elements. You might use something like this if you were examining the accuracy of a cantilever beam model and wanted to use successively higher numbers of elements from one case to the next. The *corduroy* specification for a `line` section looks like this

```
line
element-type = beam
start        = (0,0,0)
end          = (5,5,5)
number       = 20
```

This example would generate twenty beam elements all lying along the line from the origin to a point at x=5, y=5, z=5 in rectangular Cartesian coordinates. The default element type for both line and grid generation is truss. Valid type names are the same as in a standard FElt file (see chapter 3).

9.2.3 Generating a grid of line elements

Another relatively simple case is when you want to generate a three-dimensional grid of line elements. An excellent example of when this might be useful is for a model of a steel frame structure. The specification for a grid is

```
grid
element-type = beam
start        = (0,0,0)   end      = (100,500,100)
x-number     = 4         y-number = 20             z-number = 4
```

This example would generate a three-dimensional frame structure with four bays along both the x and z axes and 20 bays along y axis.

9.2.4 Generating a grid of quadrilateral planar elements

The case of a simple grid of planar four-node elements is almost identical to the grid of line elements described above. The only difference is that there are no specifications along the z direction for quadrilateral grids. An example of a mesh for a long rectangular plate using `htk` plate bending elements (the default element type for quadrilateral grids is quad_PlaneStress) might look like:

```
quadrilateral grid
element-type = htk
start        = (0,0)      end      = (10,3)
x-number     = 20         y-number = 6
```

The result would be a rectangular region meshed with 120 equi-sized square elements.

9.2.5 Generating a grid of solid brick elements

The case of a grid of solid eight-node elements is also very similar to grids of line and planar elements described in the preceding two sections. An example of a mesh for a long rectangular plate using `brick` elements (the default type for grids of solid elements) might look like:

```
brick grid
start        = (0,0,0)      end      = (8,12,6)
x-number     = 4            y-number = 6            z-number=3
```

The result would be a three-dimensional solid region three elements deep, four elements wide, and six elements tall.

9.2.6 Grid spacing rules

For all of the line and grid generation examples given above, we could also have directed that *corduroy* generate the elements with non-linear spacing rules. Such spacings are common when we know that we want a higher mesh density at one corner of the grid or only at the end of the line. The rules available are `linear` (this is the default, equi-spaced), `sinusoidal`, `cosinusoidal`, `logarithmic`, `parabolic`, `reverse-parabolic`, and `reverser-logarithmic`. If we are placing the sides of n elements along a line, then we need to locate $n+1$ nodes on the line. The coordinate x_i of each node is defined as follows for each of the spacing rules

$$\begin{aligned}
\text{linear}: \quad x_i &= L\beta & (9.1)\\
\text{cosinusoidal}: \quad x_i &= L - L\cos\left(\frac{\pi}{2}\beta\right) & (9.2)\\
\text{sinusoidal}: \quad x_i &= L\sin\left(\frac{\pi}{2}\beta\right) & (9.3)\\
\text{logarithmic}: \quad x_i &= L\log_{10}(1+9\beta) & (9.4)\\
\text{parabolic}: \quad x_i &= L\beta^2 & (9.5)\\
\text{reverse}-\text{logarithmic}: \quad x_i &= L - L\log_{10}(10-9\beta) & (9.6)\\
\text{reverse}-\text{parabolic}: \quad x_i &= L\sqrt{\beta} & (9.7)\\
& & (9.8)
\end{aligned}$$

where $\beta = (i-1)/n$.

9.2.7 Generating a triangular mesh

The triangular mesh generation capabilities of *corduroy* use Jonathan Shewchuk's Triangle routine. In addition to specifications that describe the boundary and the holes of your two-dimensional region, you must also define the approximate number of elements to generate and a constraint on the maximum area of any one generated element. The number of elements is specified with `target=`. The area constraint is given using `alpha=`, where α is defined such that

$$A_{\text{elt}} \leq \frac{\alpha A_{\text{total}}}{\text{total}}. \tag{9.9}$$

The entire *corduroy* input file to generate the nodes and elements for the wrench model pictured in Figure 7.1 would look like this.

```
start-node    = 1        /* unnecessary as this is the default */
start-element = 1

triangular mesh
element-type = CSTPlaneStress
angtol = 20    dmin = 0.5     min = 100 /* except for min and max */
angspc = 30    kappa = 0.25   max = 300 /* these are defaults     */

boundary = [
   (0,50)
   (20,10)
```

```
      (40,0)
      (40,50)
      (55,70)
      (95,70)
      (115,50)
      (115,0)
      (135,10)
      (155,50)
      (155,90)
      (115,160)
      (170,404)
      (122,404)
      (67,160)
      (0,90)
   ]

   end
```

To generate a mesh in a rectangular plate with two side by side rectangular holes in it, the input file could look like

```
   triangular mesh          /* use all defaults, including CSTPlaneStress */
   boundary = [
      (0,0)
      (20,0)
      (20,10)
      (0,10)
   ]

   hole = [
      (6,3)
      (8,3)
      (8,7)
      (6,7)
   ]

   hole = [
      (12,3)
      (14,3)
      (14,7)
      (12,7)
   ]
```

```
    end
```

Note that the hole definitions do not need to come before the boundary, but that points must always be specified in a counter-clockwise order.

9.3 Using *corduroy*

The only options to *corduroy* are the standard *cpp* options as in both *felt* and *velvet* (like those two programs, *corduroy* runs all of its input through the pre-processor before actually operating on it) and a `-debug` flag which causes *corduroy* to reproduce as output what it thinks it received for input. To generate, all you need to do is create an input file with a text editor and then run *corduroy* with the name of this file as the last argument on the command line. Output will be directed to standard out. So if you had a generation description in a file `foo.crd`, you could turn it into the basics of a FElt file called `foo.flt` with the following command

```
% corduroy foo.crd > foo.flt
```

9.4 Incorporating output into a FElt file

Because *corduroy* only generates the node and element sections of a FElt file, you still have some editing to do before you can call the problem completely defined and actually try to solve it with *felt* or *velvet*. In general, you will have to define forces and constraints to assign to nodes and material properties (and possibly distributed loads) to assign to elements. Remember, the `material=` and `constraint=` specification are only a convenience. These two names will simply be assigned to the first element and node, respectively (and all the following elements and nodes by default). You still need to actually define those objects and assign any different objects to appropriate nodes and elements. To do all this defining and assigning you can either use your favorite text editor or you can load the problem into *velvet* and do it all from there.

One thing to keep in mind whenever you use *corduroy* to generate a problem for you is that *corduroy* is not very good at node numbering. Using the renumbering features of either *felt* or *velvet* is highly recommended with *corduroy* generated problems. In general the problems that you will use *corduroy* for will be quite large and thus the reduction in memory and solution time would be significant even if node numbering had been intelligent to start with; the reductions for large problem with initially very bad numbering can be remarkable.

Chapter 10

The *burlap* Application

10.1 Introduction to the *burlap* environment

burlap is a mathematical environment designed for adding new element types to FElt and extending the current set of finite element analyses. *burlap* can best be described as a mathematical software package, similar to *matlab* or *octave*, but with the data structures of FElt. It was designed as an alternative to trying to understand and modify the C code of the FElt library, which was difficult for even us to do, much less engineers with little C programming experience.

burlap has its own syntax for manipulating matrices, like that of *matlab*. However, *burlap* also includes all of the FElt data types as well, such as nodes, elements, constraints, etc., which eliminates the need for having large arrays of numbers, where the third number is the material thickness and the fourth number is the cross-sectional area, etc.

Any analysis that can be performed with *felt* can be performed with *burlap*. Although the *burlap* code is slower than the compiled C code of the FElt library, *burlap*'s interactive environment makes prototyping new element types and new analyses much easier. At present, however, there is no support for executing *burlap* code from *velvet*.

10.2 Using *burlap*

10.2.1 Interacting with *burlap*

The syntax of *burlap* is designed to be intuitive and is best illustrated by examples. A detailed discussion of the syntax can be found in Chapter 11. The simplest way to start *burlap* is to simply type `burlap` on the command line, or `burlap foo.b` if you wish to execute the *burlap* file `foo.b`. You can also specify command line options that qualify what you want to do.

`-help | -h` Display a brief help message which lists all of the available command line options and then exits.

`-quiet | -q`
: Do not print the start-up message regarding copyright information.

`-alias | -a`
: Do not define the set of built-in aliases.

`-interactive | -i`
: Enter interactive mode by reading expressions from the terminal after processing the input files. Normally, *burlap* will simply exit after processing any input files.

`-no-interactive | -n`
: Do not enter interactive mode. This flag tells *burlap* to simply exit if no files are given. It is useful if you have `burlap` aliased to `burlap -i` and wish to override the effect of the `-i` flag.

`-source command-file | -s command-file`
: Read commands from `command-file` on start-up.

`-felt felt-file | -f felt-file`
: Use `felt-file` to define the current FElt problem.

Since *burlap* can load and process a FElt file, it also accepts the standard `-nocpp`, `-cpp`, `-I`, `-U`, and `-D` flags common to *felt* and *velvet*. If you simply type `burlap`, you will be presented with some copyright information and a prompt.

```
This is burlap, copyright 1995 by Jason I. Gobat and Darren C. Atkinson.
This is free software, and you are welcome to redistribute it under certain
conditions, but there is absolutely no warranty.  Type help ("copyright")
for details.  Use the -q option to suppress this message.

[1] _
```

If *burlap* was compiled with the GNU `readline` library, then you have complete command-line editing and history, as in *bash*. (A complete discussion of the editing capabilities can be found in the documentation of the `readline` library.) At the prompt, you can enter expressions to be evaluated.

```
[1] 1 + 2
[2] write (1 + 2)
3
[3] a = 1 + 2
[4] write (a)
3
[5] write ([1, 2, 3])

         1          2          3

[6] write ([1; 2; 3])

         1
         2
         3

[7] write ([1, 2, 3] + [4, 5, 6])

         5          7          9
```

As illustrated above, `write()` is used to print results and = is the assignment operator. Also notice from the first expression that *burlap* does not print the result of every expression like *matlab* does. But, just like *matlab*, matrices are delimited by square brackets with elements separated by commas and rows separated by semicolons.

If you type `alias` you should see the following list. (If *burlap* is not compiled with the `readline` library then the list will be slightly different.)

```
exit    exit ( )
h       history (20)
help    help ("!$")
ls      system ("ls !*")
quit    exit ( )
```

burlap has a very simple syntax with all computations being done through either operators, such as +, or functions, such as `write()`. But, since `exit()` and `help()` are rather non-intuitive, you can alias commands to be whatever you want, just like *csh*. So, you can just type `exit` or `quit` to leave *burlap*. Typing `help` will present you with a list of all the operators and functions that *burlap* provides. Typing `help foo` will give you detailed help on topic `foo`.

In short, *burlap* works just like most other interactive programs including *csh*, *gdb*, *gnuplot*, *matlab*, *octave*, and *gs*.

10.3 ***burlap* and** FElt

You can load a FElt file into *burlap* by either specifying the file on the command line with the `-felt` flag or by using the `felt()` function.

```
[1] felt ("truss.flt")
```

If everything goes okay, then the function should simply return silently. Several variables are now automatically defined, as shown in Table 10.1.

Variable	Description
`nodes`	array of nodes
`elements`	array of elements
`dofs_pos`	global DOF positions
`dofs_num`	global DOF numbers
`problem`	problem definition structure
`analysis`	analysis parameters structure

Table 10.1: Finite element related variables.

FElt maintains the global DOFs in two vectors. The first vector, `dofs_pos`, is an alias for `problem.dofs_pos`, and contains the position number for each of the six DOFs. A zero entry in the vector indicates that the DOF is not active. An example vector for a problem consisting entirely of beam elements would be

$$[1\ 2\ 0\ 0\ 0\ 3]$$

since a beam element allows translation along the x-axis and y-axis, and rotation about the z-axis. The `dofs_pos` vector can be used to determine if a given DOF is active and which global number it is assigned.

The second vector, `dofs_num`, is an alias for `problem.dofs_num`, and contains the numbers of the active DOFs. The length of the vector is equal to the number of active DOFs. The `dofs_num` vector corresponding to the `dofs_pos` vector given above would then be

$$[1\ 2\ 6]$$

since the first, second, and sixth DOFs are active. *burlap* ensures that the two vectors are always kept consistent.

All arrays start at one, so `nodes (1)` designates the first node in the problem, and `nodes (1).constraint` designates the constraint associated with that node. A period (`.`) is used to access a field of a FElt object. In general the field names are the same as the names used in the FElt input file. For example, `constraint` is the field name for accessing the constraint object assigned to a node with the `constraint=` syntax in the FElt file.

```
[2] write (nodes)
array of node
[3] write (elements)
array of element
[4] write (nodes (1))
node (1)
[5] write (nodes (1).constraint)
constraint (free)
[6] write (elements (1).material.A)
0.0004
[7] m = elements (1).material
[8] write ("A = ", m.A, " E = ", m.E)
A = 0.0004 E = 2.1e+11
```

10.3.1 Element objects

Elements have the same field names as those used in the FElt file with additional fields for holding the computed matrices and stress values, as shown in Table 10.2. For example, `e.number` will return the element number of the element `e` and `e.material.t` will return the thickness of the element's material; `e.nodes (1)` will return the first node of the element and `e.nodes (1).x` will return the x-coordinate of that node. Note that `loads` and `distributed` are synonymous, as are `num_loads` and `num_distributed`.

10.3.2 Node objects

Like elements, nodes have some fields that are defined through the FElt input file with additional fields to contain the result of computations, as shown in Table 10.3. As an example, `n.constraint` will return the constraint assigned to the node `n`, and `n.constraint.name` will return the name of that constraint; `n.dx` will return the displacement vector and `n.dx (1)` will return the first component of that vector. Note that the displacement vector may be accessed either as a vector by using `dx`, or as individual components, such as `Tx`. This

Field name	Description
`number`	element number
`nodes`	array of nodes
`K`	stiffness matrix
`M`	mass matrix
`material`	material object
`definition`	element type definition
`loads`	array of distributed loads
`num_loads`	number of loads
`stresses`	array of element stresses
`ninteg`	number of integration points
`distributed`	array of distributed loads (`loads`)
`num_distributed`	number of distributed loads (`num_loads`)

Table 10.2: Fields of an element object.

is simply a shorthand; assigning to an individual component is the same as assigning to the result of indexing the displacement vector. The equivalent force vector, `eq_force`, is used in transforming distributed loads on an element into equivalent forces on its nodes.

10.3.3 Material objects

Materials have the fields named by the FElt input syntax, as shown in Table 10.4. For example, `m.rho` will return the density of the material `m` and `m.A` will return its cross-sectional area.

10.3.4 Force objects

Force objects allow the forces and moments to be accessed as individual components or as a vector, as shown in Table 10.5. For example, `f.force` will return the force vector of `f` and `f.force (4)` will return the moment about the x axis, which is equivalent to `f.force.Mx`.

For transient problems, a force or moment may vary with time. In *felt*, this is described by an expression in terms of `t`. A component of a force may be assigned a string value which will be interpreted by *burlap* as an expression. Note that evaluating the component will evaluate the expression at time `t=0`. There is currently no way to determine if a component is assigned a time-varying expression.

Field name	Description
`number`	node number
`constraint`	constraint object
`force`	force object
`eq_force`	equivalent force vector
`dx`	displacement vector
`x`	x-coordinate
`y`	y-coordinate
`z`	z-coordinate
`m`	lumped mass
`Tx`	translation along x axis (`dx (1)`)
`Ty`	translation along y axis (`dx (2)`)
`Tz`	translation along z axis (`dx (3)`)
`Rx`	rotation about x axis (`dx (4)`)
`Ry`	rotation about y axis (`dx (5)`)
`Rz`	rotation about z axis (`dx (6)`)

Table 10.3: Fields of a node object.

```
[1] f.Fx = "cos (t)"
[2] write (f.Fx)
1
```

10.3.5 Constraint objects

Constraint objects also allow the various information to be accessed as vectors or as individual components, as shown in Table 10.6. If `c` is a constraint then `c.ix` is its initial displacement vector and `c.ix (1)` or `c.iTx` is its initial translation along the x axis. Components may be assigned time-varying expressions, as in the case of forces.

10.3.6 Distributed load objects

Distributed loads, or simply loads, have the fields named by the FElt file syntax, as shown in Table 10.7. The individual values can be accessed using `node` and `magnitude`. For example, `l.values (1).node` returns the node object of the first load point and `l.values (1).magnitude` returns its magnitude.

The `direction` field is used to indicate the direction of the distributed load. The direc-

Field name	Description
`name`	name of material
`E`	Young’s modulus
`Ix`	moment of inertia about x-x axis
`Iy`	moment of inertia about y-y axis
`Iz`	moment of inertia about z-z axis
`A`	cross-sectional area
`J`	polar moment of inertia
`G`	bulk (shear) modulus
`t`	thickness
`rho`	density
`nu`	Poisson’s ratio
`kappa`	shear force correction
`Rk`	Rayleigh damping coefficient (K)
`Rm`	Rayleigh damping coefficient (M)
`Kx`	conductivity along x axis
`Ky`	conductivity along y axis
`Kz`	conductivity along z axis
`c`	heat capacity

Table 10.4: Fields of a material object.

Field name	Description
`name`	name of force
`force`	force vector
`spectrum`	input spectra
`Fx`	force along x axis (`force (1)`)
`Fy`	force along y axis (`force (2)`)
`Fz`	force along z axis (`force (3)`)
`Mx`	moment about x axis (`force (4)`)
`My`	moment about y axis (`force (5)`)
`Mz`	moment about z axis (`force (6)`)
`Sfx`	spectra along x axis (`spectrum (1)`)
`Sfy`	spectra along y axis (`spectrum (2)`)
`Sfz`	spectra along z axis (`spectrum (3)`)
`Smx`	spectra about x axis (`spectrum (4)`)
`Smy`	spectra about y axis (`spectrum (5)`)
`Smz`	spectra about z axis (`spectrum (6)`)

Table 10.5: Fields of a force object.

tion is simply a scalar value as indicated in Table 10.8.

10.3.7 Element definition objects

An element definition object is one of the few object types not defined in a FElt file. An element definition, or simply definition, contains all information necessary for a particular element type, such as a beam, truss, etc. The fields given in Table 10.9 are discussed at length in Section 10.4.

10.3.8 Problem definition

The problem definition object contains all information about the current FElt problem, with the exception of the analysis parameters.

Most fields of the problem definition given in Table 10.10 are read-only. If you want to change the characteristics of a problem, you need to edit the input file that defines the problem, and load it with the `felt()` function. The only exceptions are the DOF arrays, `dofs_pos` and `dofs_num`. Since these arrays must be properly initialized before many of the built-in finite element functions may be called, *burlap* ensures that the arrays are kept

Field name	Description
`name`	name of constraint
`constraint`	constraint vector
`ix`	initial displacement vector
`vx`	initial velocity vector
`ax`	initial acceleration vector
`dx`	boundary displacement vector
`iTx`	initial displacement along x axis (`ix (1)`)
`iTy`	initial displacement along y axis (`ix (2)`)
`iTz`	initial displacement along z axis (`ix (3)`)
`iRx`	initial rotation about x axis (`ix (4)`)
`iRy`	initial rotation about y axis (`ix (5)`)
`iRz`	initial rotation about z axis (`ix (6)`)
`Vx`	initial velocity along x axis (`vx (1)`)
`Vy`	initial velocity along y axis (`vx (2)`)
`Vz`	initial velocity along z axis (`vx (3)`)
`Ax`	initial acceleration along x axis (`ax (1)`)
`Ay`	initial acceleration along y axis (`ax (2)`)
`Az`	initial acceleration along z axis (`ax (3)`)
`Tx`	translation along x axis (`dx (1)`)
`Ty`	translation along y axis (`dx (2)`)
`Tz`	translation along z axis (`dx (3)`)
`Rx`	translation along x axis (`dx (4)`)
`Ry`	translation along y axis (`dx (5)`)
`Rz`	translation along z axis (`dx (6)`)

Table 10.6: Fields of a constraint object.

Field name	Description
`name`	name of distributed load
`direction`	load direction
`num_values`	number of values
`values`	array of values ((`node`, `magnitude`) pairs)

Table 10.7: Fields of a distributed load object.

Direction	Value
`LocalX`	`&local_x`
`LocalY`	`&local_y`
`LocalZ`	`&local_z`
`GlobalX`	`&global_x`
`GlobalY`	`&global_y`
`GlobalZ`	`&global_z`
`Parallel`	`&parallel`
`Perpendicular`	`&perpendicular`

Table 10.8: Directions of a distributed load.

Field name	Description
`name`	name of definition
`setup`	element set-up function
`stress`	element stress computation function
`num_nodes`	number of nodes in element
`shape_nodes`	number of nodes that define shape
`num_dofs`	number of degrees of freedom
`dofs`	degrees of freedom
`num_stresses`	number of stresses
`retainK`	flag to retain stiffness matrix after assembly

Table 10.9: Fields of an element definition object.

Field name	Description
`mode`	analysis mode
`title`	problem title
`nodes`	array of node objects
`elements`	array of element object
`dofs_pos`	global DOF positions
`dofs_num`	global DOF numbers
`num_dofs`	number of active DOFs
`num_nodes`	number of nodes
`num_elements`	number of elements

Table 10.10: Fields of a problem definition object.

consistent. The individual components of the arrays are read-only; however, either array may be assigned a row vector to change the set of active DOFs. When one array is changed, both arrays and the number of active DOFs are updated. (We don't know why you would want to do this. Usually, `find_dofs()` should be used to initialize the arrays. Changing the DOFs yourself will probably just lead to non-singular matrices.)

10.3.9 Analysis parameters

The analysis parameters structure contains information related to a specific analysis type.

Field name	Description
`gamma`	parameter in Newmark integration
`beta`	parameter in Newmark integration
`alpha`	parameter in H-H-T
`mass_mode`	mass mode
`nodes`	array of node numbers of interest
`dofs`	array of dofs of interest
`start`	starting time or frequency
`stop`	ending time or frequency
`step`	time or frequency increment
`num_dofs`	number of dofs of interest
`num_nodes`	number of nodes of interest

Table 10.11: Fields of an analysis parameters object.

Unlike the problem definition structure, most of the fields in the analysis parameters structure are changeable. However, the `dofs` and `nodes` fields are similar to the `dofs_num` and `dofs_pos` fields of the problem definition. The individual components are read-only, but they may be assigned a row vector. If `analysis.dofs` is assigned a valid row vector then `analysis.num_dofs` is updated and is equivalent to `length (analysis.dofs)`. Similarly, if `analysis.nodes` is assigned a valid row vector then `analysis.num_nodes` is updated and is equivalent to `length (analysis.nodes)`. The `length()` function returns the number of items in an array, matrix, or string.

10.4 Adding new element types to *burlap*

Adding a new element type to FElt can be very time-consuming and error-prone. *burlap* allows the user to easily add new element definitions that can be quickly tested without having to recompile the *felt* or *velvet* applications. To add a new element to the set of existing elements, the `add_definition()` function is used. Similarly, to remove an existing element definition, the `remove_definition()` function is used.

Argument	Description	Type	Example
`name`	name of definition	`string`	`"truss"`
`set_up`	set up function (K and M)	`function`	`truss_set_up`
`stress`	stress computation function	`function`	`truss_stress`
`shape`	element shape	`scalar`	`&linear`
`num_nodes`	number of nodes in element	`scalar`	`2`
`shape_nodes`	number of nodes that define shape	`scalar`	`2`
`num_stresses`	number of stress values	`scalar`	`1`
`dofs`	vector of DOFS	`row vector`	`[1,2,3,0,0,0]`
`retainK`	retain stiffness matrix indicator	`scalar`	`&false`

Table 10.12: Arguments to the `add_definition()` function.

The `add_definition()` function requires a variety of arguments as shown in Table 10.12. For the reminder of the discussion regarding the arguments to the `add_definition()` function, we will use the truss element as an example.

The `name` of the element definition is a string by which the element type is referred. Once the element definition has been successfully added then new elements of that type can be created. In our example the name is `"truss"`, so a FElt file can now be loaded that contains `truss` elements. If an element definition of the same name already exists then `add_definition()` returns `1`, otherwise it returns `0`.

The `set_up` function is a *burlap* function that will compute the local stiffness matrix, K, and local mass matrix, M, for any element of the new definition. Our sample set-up function is shown in Figure 10.1. For our simple truss element, we are ignoring the mass matrix and the effect of distributed loads.

The set-up function will be called with two arguments. The first argument is the element whose stiffness and mass matrices are to be computed. The second argument is the mass mode, which is one of `&false`, `&lumped`, or `&consistent`, where `&false` indicates that no mass matrix is required. In our simple example, we have simply omitted the second argument, since we will not be considering mass matrices. Note that the `set_up` argument is a *burlap* function itself, not the name of the function. As discussed in Section 11.6, functions in *burlap* are simply variables that can be passed as arguments to functions and also returned from functions. Our simple function for truss elements computes the length of the element by calling the `length()` function, which is discussed in Section 11.5.4.

```
function truss_set_up (e)
    L = length (e)
    AEonL = e.material.A * e.material.E / e.material.L
    K = [AEonL, -AEonL; -AEonL, AEonL]

    cx = (e.node (2).x - e.node (1).x) / L
    cy = (e.node (2).y - e.node (1).y) / L
    cz = (e.node (2).z - e.node (1).z) / L

    T = [cx, cy, cz, 0, 0, 0; 0, 0, 0, cx, cy, cz]
    e.K = T'*K*T

    return 0
end
```

Figure 10.1: Set-up function for the truss element definition.

The set-up function assigns the stiffness and mass matrices to the element by assigning to the `K` and `M` fields of the element structure. If the set-up function is successful then it should return zero. Otherwise, it should return a non-zero value.

The `stress` function is a *burlap* function that will compute the stress values for any element of the new definition. Our sample set-up function is shown in Figure 10.2. The stress function assigns the stresses by assigning to the `stress` field of the element structure. If the stress function is successful then it should return zero. Otherwise, it should return a non-zero value.

Our simple truss element has only one integration point and only a single stress value at that point. Since an element may have an arbitrary number of integration points, the `ninteg` field of the element structure must be set before the `stress` vector can be accessed. The number of stresses and thus the length of the `values` vector is determined by the element definition. In the computation of the stress value for the truss element, the two displacement row vectors are subtracted and then multiplied by the column vector, `c`, to yield a scalar.

The remaining arguments to `add_definition()` are easily calculated. The shape of our truss element is linear and has two nodes, both of which are required and needed to define the shape of the element. The truss element permits displacements along the X, Y, and Z axes, but does not allow for rotation about any of the axes. Therefore, the first, second, and third DOFs are active. The entire call to `add_definition()` is:

```
function truss_stress (e)
    L = length (e)
    EonL = e.material.E / L
    c = zeros (6, 1)

    c(1) = (e.node (2).x - e.node (1).x) / L
    c(2) = (e.node (2).y - e.node (1).y) / L
    c(3) = (e.node (2).z - e.node (1).z) / L

    e.ninteg = 1
    e.stress (1).x = (e.node(1).x + e.node(2).x) / 2
    e.stress (1).y = (e.node(1).y + e.node(2).y) / 2
    e.stress (1).values (1) = EonL * (e.node(2).dx - e.node(1).dx) * c

    return 0
end
```

Figure 10.2: Stress function for the truss element definition.

```
add_definition ("truss", truss_set_up, truss_stress,
        &linear, 2, 2, 1, [1, 2, 3, 0, 0, 0])
```

We have neglected to pass a value for the last argument, which indicates whether the local stiffness matrix should be retained after assembly into the global stiffness matrix. The default value for this argument is `&false`, which is fine for our purposes. Only the last argument to `add_definition()` may be omitted; the remaining argument do not have default values. If we wish to remove the truss element definition, then the `remove_definition()` function must be used. The `remove_definition()` function takes a single argument that is the name of the element to be removed. In our case, the call would be `remove_definition ("truss")`.

10.5 Tips on using interactive mode

Most of the time, *burlap* will be used in interactive mode: expressions will be typed in at the prompt and evaluated. Functions are best created with an editor as a file which is then included.

```
[1] system ("vi foo.b")
```

```
[2] include ("foo.b")
[3] system ("vi foo.b")
[4] include ("foo.b")
```

It soon becomes tedious to retype the same expressions. If *burlap* is compiled with the `readline` library, then you can use the built-in history mechanism.

```
[1] system ("vi foo.b")
[2] include ("foo.b")
[3] !s
[4] !i
```

For those not familiar with `csh` or `bash`, `!s` executes the last line that begins with `s`. Similarly, `!i` executes the last line that begins with `i`. You can save yourself more typing by using the word completion abilities of the `readline` library. Pressing the tab key will complete the current word. If there is not a unique completion, then you will hear a bell and pressing the tab key again will list the completions.

```
[1] sy <Tab>
symmetric? system
[1] system ("vi foo.b")
[2] in <Tab>
in                    integrate_hyperbolic  inv
include               integrate_parabolic
[2] include ("foo.b")
```

The default set of built-in completions includes all keywords such as `function` and `return`, intrinsic functions such as `include` and `sin`, and enumeration constants such as `&linear` and `&null`. Inside a quoted string, the default set of completions is not used; instead, file names are completed.

```
[1] include ("Examples/f <Tab>
fe.b         find-dofs.b  foo.flt      foo_d.flt
[1] include ("Examples/f
```

Using the completion mechanism can save a lot of typing, especially when referring to file names. Another trick is to define your own aliases for common lines.

```
[1] alias v system ("vi foo.b")
[2] alias vi system ("vi !$")
[3] alias print write (!*)
[4] vi foo.b
```

The first alias defines `v` to be `system ("vi foo.b")`. The second is more general and defines `vi` to be `system ("vi !$")`. The string `!$` will be replaced with the last word on the line. A complete description of the history expansion can be found in the documentation for the `readline`library and for `csh`. The final alias, `print`, is simply a convenient wrapper for the `write` function.

If *burlap* is not compiled with the `readline` library, you still have access to a limited form of aliasing. The only expansion string legal in an alias is `%s` which will expand to the entire line except the first word. A similar alias to `vi` above would be `system ("vi %s")`.

An alternative to writing aliases is to define your own convenience functions. If you wish to be able to both edit and include a file then you can write a small function, called `edit`, and place it in a file called `edit.b` in your search path, as described in Section 11.4.14.

```
function edit (s)
    system (concat ("vi ", s))
    include (s)
end
```

Finally, it is important to realize that the `readline` library capabilities and aliases work on lines, not on expressions. History substitutions apply to the lines entered, not to the individual expressions on a line. Similarly, aliases and history expansion characters are not recognized when *burlap* is reading from a file.

10.6 Common error messages

This section attempts to explain some of the common error messages associated with using *burlap*. All error messages are prefixed with the name of the file and the number of the line that was found to be in error. An error on a line may not be discovered until several lines later, so the line number may not be the line that actually contained the error. If an error occurs in a function, then the function call is terminated and the error is propagated back to the outermost level. In this case, the calling sequence is shown to assist in tracking down the cause of the error. In the examples given, the file name and line number have been omitted.

`type error in expression:  function call to null`

This error indicates that the function call or array index operators were applied to a `null` expression such as `foo ( )`, where `foo` is `null`. This error often results from misspelling a variable name or from forgetting to declare a global variable using the `global` declaration in a function body. You can generally locate this error by searching for parentheses on the offending line.

`type error in expression:` *type* `has no such field`

The field index operator was applied to an object of the specified *type*, but the object has no such field. You can generally locate this error by searching for a period on the given line.

`type error in expression:` *type1 operator type2*

The specified *operator* does not allow operands of the specified types. The way to fix this error is to determine what operand types are allowed, as discussed in Section 11.4, and determine why one or more of your operands are of the incorrect type.

`parse error before` *string*

At some time at or before *string*, the expression syntax was illegal. Often, *string* is the cause of the error. You should now be back at the top level prompt, in which case you can correct your error and try again.

`size mismatch in expression: (`*a* `x` *b*`)` *operator* `(`*c* `x` *d*`)`

The types of the operands were legal for the specified *operator*, but the sizes of the operands were illegal. The legal operand sizes are discussed in Section 11.4.

Chapter 11

The *burlap* Syntax

burlap has a rich variety of operators and functions for manipulating matrices, scalars, strings, and FElt objects. Matrices are denoted by square brackets with matrix elements separated by commas and matrix rows separated by semicolons. Scalars are floating-point numbers such as `3.14159` or `123`. Any scalar can also be converted to a matrix with one row and one column. Strings are enclosed in double quotation marks. Special enumeration constants begin with an ampersand, as in `&local_x`. Variable names are a sequence of letters, digits, and the underscore character. Finally, comments are enclosed within `/* */`; a pound sign (`#`) may also be used to indicate a comment until the end of the current line.

```
[1] a = 3.14159
[2] b = "hello"
[3] c = [1, 2, 3; 4, 5, 6]
[4] write (a)
3.14159
[5] write (b)
hello
[6] write (c)

          1           2           3
          4           5           6
```

The following sections discuss the operators and functions of *burlap* in detail. They are intended to be the definitive word on the semantics of the operators, functions, matrix formers, etc.

11.1 Literals

It seems appropriate to begin the discussion with the literals since they are the building blocks of the *burlap* syntax. As previously mentioned, floating-point literals are numbers such as `123` or `3.14159`. Specifically, a numeric literal consists of a *base* followed by an optional *exponent*. The base consists of a series of digits with an optional decimal point. The exponent consists of the letter `E` or `e` followed by the digits of the base-10 exponent. The digits in the exponent may be preceded by a `+` or `-` to indicate a positive or negative exponent, respectively. String literals are enclosed within double quotation marks, as in `"hello"`. The following *escapes* may be used to introduce special characters within a string literal.

audible alert	`\a`	carriage return	`\r`
backspace	`\b`	horizontal tab	`\t`
formfeed	`\f`	vertical tab	`\v`
newline	`\n`	quote	`\"`

Matrices are delimited by square brackets with rows separated by semicolons and elements within a row separated by commas. An important thing to realize when writing matrices is that *burlap* may automatically insert a semicolon at the end of line. In particular, a semicolon is inserted at the end of a line in *interactive mode* when the last literal or operator on the line can end an expression.

```
[1] a = [1, 2, 3
[2] 4, 5, 6]
[3] write (a)

         1          2          3
         4          5          6

[4] a = [1, 2, 3 +
 4> 4, 5, 6]
[5] write (a)

         1          2          7          5          6
```

Note that since an expression cannot end with `+`, the next line is taken to be a continuation of the previous line. A matrix element may be an arbitrary expression; however, all expressions on the same row must have the same height. Additionally, all rows must have the same number of columns.

```
[1] x = 1
[2] a = [0, x, 2]
[3] write (a)

        0           1           2

[4] b = [a; 3, 4, 5]
[5] write (b)

        0           1           2
        3           4           5

[6] write ([b, b])

        0           1           2           0           1           2
        3           4           5           3           4           5

[7] write ([b, a])
stdin:7: inconsistent number of rows
```

11.2 Variables

Variables are named by a sequence of letters, digits, the underscore character, and the question mark symbol. A variable must begin with a letter or the underscore character. The question mark may only appear as the last character in a variable name. There is no limit on the length of a variable name.

Variables are assigned values either through the assignment operator, parameter passing, or function definitions. A variable is either *local* or *global*. Local variables are those variables referenced within a function body. Global variables are those variables referenced outside a function body. To explicitly reference a global variable inside a function body, it must be declared as global using the `global` declaration.

```
function foo (x)
    global y, z
    return x + y + z
end
```

The function `foo` references two global variables, `y` and `z`. If the function did not use the `global` declaration then `y` and `z` would be local variables. The only exception to this rule is a function call, as discussed in Section 11.4.14.

11.3 Constants

There are several predefined enumeration constants in *burlap*. Most of the constants correspond to keywords used in the FElt input file to define properties such as the direction of a load. The enumeration constants are listed in Table 11.1.

analysis types:	`&static`, `&transient`, `&modal`, `&static_thermal`, `&transient_thermal`, `&spectral`
load directions:	`&local_x`, `&local_y`, `&local_z`, `&global_x`, `&global_y`, `&global_z`, `&parallel`, `&perpendicular`
degrees of freedom:	`&tx`, `&ty`, `&tz`, `&rx`, `&ry`, `&rz`, `&fx`, `&fy`, `&fz`, `&mx`, `&my`, `&mz`
element shapes:	`&linear`, `&planar`, `&solid`
miscellaneous:	`&constrained`, `&unconstrained`, `&hinged`, `&lumped`, `&consistent`, `&null`, `&true`, `&false`

Table 11.1: Enumeration constants in *burlap*.

11.4 Operators

This section discusses the various operators available in *burlap*. The operators are discussed in order of precedence, with the lowest precedence first. In general, uppercase letters denote matrices and lowercase letters denote scalars. For many operators, a precise definition of the operator is given, along with a table specifying the legal types of the operands, and example *burlap* output.

11.4.1 Expression separators

```
X ; Y
X , Y
```

A semicolon is used to separate multiple expressions to be evaluated and to separate the rows of a matrix. A comma is used to separate the arguments to a function and to separate the elements within a single row of a matrix.

Since an expression usually ends at the end of a line, typing a semicolon at the end of every line to separate one expression from the next would be extremely tedious. Instead, *burlap* will automatically insert a semicolon for you at the end of a line *under certain conditions*. When reading from a terminal in interactive mode, if the last token on the line may end an expression then a semicolon is inserted. When reading from a file, if the last token on the line may end an expression *and* the first token on the next line may begin an expression then a semicolon will be inserted. A line can be explicitly continued by ending the line with a backslash character.

```
[1] write ("hello"); write ("there")
hello
there
[2] a = 1 +
 2> 2 + 3
[3] write (a)
6
[4] b = 1 \
 4> + 2
[5] write (b)
3
[6] x = [1, 2, 3
[7]      4, 5, 6]
[8] write (x)

         1          2          3
         4          5          6
```

Although these semantics may sound complicated, they are designed so that 99.9% of the time you don't need to worry about when to add a semicolon. However, if you find yourself with a nasty bug, you might want to review the rules for inserting semicolons or try putting in an explicit semicolon or backslash.

11.4.2 Assignment expressions

```
X = Y; X := Y
```

The assignment operator has two equivalent forms and groups from right-to-left. An assignment expression assigns Y to X and returns X. X must be a variable name, a submatrix, a field reference, or the result of a function call returning a global variable.

Generally, any expression can be assigned to a variable. However, in certain cases the expression on the right-hand side is restricted. For example, if X is a submatrix, created as the result of indexing a matrix, then the dimensions (number of rows and columns) of X and Y must agree.

If X is the result of a field reference of a FElt structure then there may be type constraints on Y. For example, a matrix cannot be assigned to the component of a force. Some fields are also read-only, such as the name of a constraint or the number of a node or element.

```
[1] a = [1, 2, 3]
[2] write (a)

         1          2          3

[3] a = b = 0
[4] write (a, " ", b)
0 0
[5] a = (b = 2) + 1
[6] write (a, " ", b)
3 2
[7] nodes (1).number = 3
stdin:7: type error in expression: changing a read-only variable
[8] nodes (1).force.Fx = 4
[9] nodes (1).force.Fy = [1, 2, 3]
stdin:9: type error in expression: scalar = matrix
[10] a = [1, 2, 3; 4, 5, 6]
[11] write (a)

         1          2          3
         4          5          6

[12] a (1, 2) = 0
[13] write (a)

         1          0          3
         4          5          6
```

11.4.3 Logical OR operator

```
x or y; x || y
```

The logical OR operator has two equivalent forms and groups from left-to-right. It returns `1` if either of its operands compares unequal to zero, and `0` otherwise. The OR operator guarantees left-to-right evaluation: x is first evaluated; if it is unequal to `0`, the value of the expression is `1`. Otherwise, y is evaluated, and if it is unequal to `0`, the result of the OR expression is `1`, otherwise the result is `0`.

$$x \text{ or } y \stackrel{\text{def}}{=} \begin{cases} 1 & \text{if } x \neq 0 \text{ or } y \neq 0 \\ 0 & \text{otherwise} \end{cases}$$

Both x and y must be scalar expressions, as shown in Table 11.2. It makes little sense to compare a matrix or string with zero, for example. There are, however, common ways of mapping a matrix to a scalar. Two such ways are provided by the `any?()` and `every?()` functions discussed in Section 11.5.3.

X	Y	Z	Z = X or Y
scalar	scalar	scalar	$z = x$ or y

Table 11.2: Type table for the logical OR operator.

```
[1] write (1 < 2 or 5 > 6)
1
[2] write (1 > 2 or 5 > 6)
0
[3] write (1 or 1 / 0)
1
[4] write (0 or 1 / 0)
stdin:4: exception in expression: right division by zero
[5] a = [1, 2, 3]
[6] write (a or 1)
stdin:6: type error in expression: matrix in conditional context
[7] write (any? (a) or 1)
1
```

11.4.4 Logical AND operator

```
x and y; x && y
```

The logical AND operator has two equivalent forms and groups from left-to-right. It returns `1` if both of its operands compare unequal to zero, and `0` otherwise. The AND operator guarantees left-to-right evaluation: x is first evaluated; if it is equal to `0`, the value of the expression is `0`. Otherwise, y is evaluated, and if it is equal to `0`, the result of the AND expression is `0`, otherwise the result is `1`.

$$x \text{ and } y \overset{\text{def}}{=} \begin{cases} 1 & \text{if } x \neq 0 \text{ and } y \neq 0 \\ 0 & \text{otherwise} \end{cases}$$

Both x and y must be scalar expressions, as shown in Table 11.3. It makes little sense to compare a matrix or string with zero, for example. There are, however, common ways of mapping a matrix to a scalar. Two such ways are provided by the `any?()` and `every?()` functions discussed in Section 11.5.3.

X	Y	Z	Z = X and Y
scalar	scalar	scalar	$z = x$ and y

Table 11.3: Type table for the logical AND operator.

```
[1] write (1 < 2 and 3 < 4)
1
[2] write (1 < 2 and 3 > 4)
0
[3] write (0 and 1 / 0)
0
[4] write (1 and 1 / 0)
stdin:4: exception in expression: right division by zero
```

11.4.5 Equality operators

```
X == Y
X != Y; X <> Y
```

The equality (==) and inequality (`!=` or <>) operators associate from left-to-right. The result of the expression is `1` if true, and `0` if false.

$$x = y \overset{\text{def}}{=} \begin{cases} 1 & \text{if } x = y \\ 0 & \text{otherwise} \end{cases}$$

$$x \mathrel{!=} y \overset{\text{def}}{=} \begin{cases} 1 & \text{if } x \neq y \\ 0 & \text{otherwise} \end{cases}$$

Unlike the previously discussed operators, these operators, like most of the operators in *burlap*, accept a variety of types as operands, as shown in Table 11.4.

X	Y	Z	Z = X op Y
matrix	matrix	matrix	$\forall i, j \mid Z_{i,j} = X_{i,j}$ op $Y_{i,j}$
matrix	scalar	matrix	$\forall i, j \mid Z_{i,j} = X_{i,j}$ op y
scalar	matrix	matrix	$\forall i, j \mid Z_{i,j} = x$ op $Y_{i,j}$
scalar	scalar	scalar	$z = x$ op y (scalar comparison)
string	string	scalar	$z = x$ op y (lexicographic comparison)
object	object	scalar	$z = x$ op y (object identity comparison)

Table 11.4: Type table for the equality operators.

Although the table might look confusing, it provides a precise semantics for all possible type operands. Essentially, *burlap* does element-by-element comparison for matrices and scalars, string (i.e., lexicographic) comparison for strings, and identity (i.e., pointer) comparison on all other type objects. When comparing two matrices, their dimensions must be identical.

```
[1]  write ([1, 2, 3] == 2)

         0          1          0

[2] write (2 != [1, 2, 3])

         1          0          1

[3] write ([1, 2, 3] == [3, 2, 3])

         0          1          1

[4] write ("hello" == "hello")
1
[5] write ("hello" == "there")
0
[6] write (nodes (1) == nodes (1))
1
[7] write (nodes (1) == elements (1))
stdin:7: type error in expression: node == element
```

11.4.6 Relational operators

```
X < Y
```

```
X > Y
X <= Y
X >= Y
```

The relational operators are similar to the equality operators, with the only difference being that they do not allow an arbitrary object type as an operand for comparison (Table 11.5).

X	Y	Z	Z = X op Y
matrix	matrix	matrix	$\forall i,j \mid Z_{i,j} = X_{i,j}$ op $Y_{i,j}$
matrix	scalar	matrix	$\forall i,j \mid Z_{i,j} = X_{i,j}$ op y
scalar	matrix	matrix	$\forall i,j \mid Z_{i,j} = x$ op $Y_{i,j}$
scalar	scalar	scalar	$z = x$ op y (scalar comparison)
string	string	scalar	$z = x$ op y (lexicographic comparison)

Table 11.5: Type table for the relational operators.

The result of the expression is `1` if the expression is true, and `0` if the expression is false. If two matrices are compared, then their dimensions must be identical.

```
[1] write ([1, 2, 3] < 3)

         1           1           0

[2] write ([1, 2, 3] >= [3, 2, 1])

         0           1           1

[3] write ("aardvark" < "aardwolf")
1
```

11.4.7 Range operator

```
x : z
x : y : z
```

The range operator produces a row vector starting at *x* and ending at *z*, with *y* used as the step between successive elements.

$$x:y:z \overset{\text{def}}{=} \begin{cases} \left[x\ x+y\ \cdots\ x+\left\lfloor\frac{z-x}{y}\right\rfloor y\right] & \text{if } (x \le z \text{ and } y > 0) \text{ or } (x \ge z \text{ and } y < 0) \\ \text{null} & \text{otherwise} \end{cases}$$

If *y* is not specified then `1` is used as the step value. Each of *x*, *y*, and *z* must be scalar expressions, as shown in Table 11.6.

X	Y	Z	W	W = X : Y : Z
scalar	scalar	scalar	matrix	(as defined above)

Table 11.6: Type table for the range operator.

```
[1] write (1:3)

         1           2           3

[2] write (3:1)
null
[3] write (1:2:10)

         1           3           5           7           9

[4] write (10:-2:1)

        10           8           6           4           2
```

As illustrated above, if the range is improperly specified then a `null` value is returned. The `null` value is illegal in most operations and is used to indicate an uninitialized value.

11.4.8 Additive operators

```
X + Y
X - Y
```

The type semantics of the additive operators are quite similar to the semantics of the relational and equality operators, except that only scalars and matrices are allowed. As shown in Table 11.7, adding two scalars produces a scalar, as expected; adding a matrix to a scalar simply adds the scalar to each element of the matrix; adding two matrices results in standard matrix addition, again as expected.

Like the equality and relational operators, the additive operators associate from left-to-right. Also, two matrices must have identical dimensions for addition or subtraction.

```
[1] write (1 + 2)
3
```

X	Y	Z	Z = X op Y
scalar	scalar	scalar	$z = x$ op y
scalar	matrix	matrix	$\forall i,j \mid Z_{i,j} = x$ op $Y_{i,j}$
matrix	scalar	matrix	$\forall i,j \mid Z_{i,j} = X_{i,j}$ op y
matrix	matrix	matrix	$\forall i,j \mid Z_{i,j} = X_{i,j}$ op $Y_{i,j}$

Table 11.7: Type table for the additive operators.

```
[2] write (1 + [2, 3, 4])

         3          4          5

[3] write ([3, 4, 5] - 1)

         2          3          4

[4] write ([1, 2] + [3, 4])

         4          6

[5] write ([1, 2, 3] + [4, 5])
stdin:5: size mismatch in expression: (1 x 3) + (1 x 2)
```

11.4.9 Multiplicative operators

```
X * Y
X / Y
X \ Y
X % Y
```

The multiplicative operators, which associate from left-to-right, are distinct from the other operators in that the results are not computed on an element-by-element basis.

As illustrated in Table 11.8, multiplying two scalars results in scalar multiplication, multiplying a matrix by a scalar is equivalent to scaling the matrix by the scalar, and multiplying two matrices results in standard matrix multiplication. In matrix multiplication the inner dimensions must agree.

```
[1] write (2 * 3)
6
```

X	Y	Z	Z = X * Y
scalar	scalar	scalar	$z = x \times y$
scalar	matrix	matrix	$\forall i,j \mid Z_{i,j} = x \times Y_{i,j}$
matrix	scalar	matrix	$\forall i,j \mid Z_{i,j} = X_{i,j} \times y$
matrix	matrix	matrix	$\forall i,j \mid Z_{i,j} = \sum_{k=1}^{n} X_{i,k} \times Y_{k,j}$

Table 11.8: Type table for the multiplication operator.

```
[2] write (2 * [1, 2, 3])

         2           4           6

[3] write ([1, 2, 3] * [1; 2; 3])
14
[4] write ([1; 2; 3] * [1, 2, 3])

         1           2           3
         2           4           6
         3           6           9

[5] write ([1, 2] * [3, 4])
stdin:5: size mismatch in expression: (1 x 2) * (1 x 2)
```

burlap supports two forms of division: left division and right division. The right division of X and Y is (approximately) equivalent to multiplying X by the inverse of Y.

X	Y	Z	Z = X / Y
scalar	scalar	scalar	$z = x/y$
scalar	matrix	matrix	$\forall i,j \mid Z_{i,j} = x \times Y_{i,j}^{-1}$
matrix	scalar	matrix	$\forall i,j \mid Z_{i,j} = X_{i,j}/y$
matrix	matrix	matrix	$Z \mid ZY = X$

Table 11.9: Type table for the right division operator.

The first three cases in Table 11.9 are as expected. Note that in the second case, where Y is a matrix, its inverse must be explicitly computed, which may result in numeric instability and thus should be avoided. In the fourth case, where two matrices are "divided", then an LU-factorization, combined with transposition, is used to compute the "inverse" without introducing numeric instability, as shown in Equation 11.1. In all cases, the number of rows of X must equal the number of columns of Y.

$$\begin{aligned} Z = X/Y &\equiv Z = XY^{-1} \\ &\equiv ZY = X \\ &\equiv (Y^T Z^T)^T = (X^T)^T \end{aligned} \tag{11.1}$$

```
[1] write (1 / 2)
0.5
[2] write (1 / 0)
stdin:2: exception in expression: right division by zero
[3] write ([1, 2, 3] / 10)

       0.1        0.2        0.3
```

The left division of X and Y is (approximately) equivalent to the inverse of X multiplied by Y. Left division is most often used to solve equations of the form $Ax = b$.

X	Y	Z	Z = X \ Y
scalar	scalar	scalar	$z = y/x$
scalar	matrix	matrix	$\forall i, j \mid Z_{i,j} = Y_{i,j}/x$
matrix	scalar	matrix	$\forall i, j \mid Z_{i,j} = X_{i,j}^{-1} \times y$
matrix	matrix	matrix	$Z \mid XZ = Y$

Table 11.10: Type table for the left division operator.

As shown in Table 11.10, the third case involves the explicit computation of the matrix inverse, and thus should be avoided. Left division is also known as a *backsolve*, since if performed on two matrices, an LU-decomposition (or Crout factorization for compact matrices) is computed followed by a backsolve operation. (This is easy to remember since the *back*slash is used to perform a *back*solve). The number of columns of X must equal the number of rows of Y.

```
[1] write (2 \ 1)
0.5
[2] write (0 \ 1)
stdin:2: exception in expression: left division by zero
[3] write (10 \ [1, 2, 3])

       0.1        0.2        0.3

[4] write ([1, 2; 3, 4] \ [5, 6])
```

```
stdin:4: size mismatch in expression: (2 x 2) \ (1 x 2)
[5] write ([1, 2; 3, 4] \ [5; 6])

          -4
         4.5
```

Finally, *burlap* also supports a remainder operation as shown in Table 11.11. In the left division, right division, and remainder operators, a division by zero will cause a mathematical exception.

$$x \bmod y \overset{\text{def}}{=} x - \lfloor x/y \rfloor y$$

X	Y	Z	Z = X % Y
scalar	scalar	scalar	$z = x \bmod y$
scalar	matrix	matrix	$\forall i,j \mid Z_{i,j} = x \bmod Y_{i,j}$
matrix	scalar	matrix	$\forall i,j \mid Z_{i,j} = X_{i,j} \bmod y$
matrix	matrix	matrix	$\forall i,j \mid Z_{i,j} = X_{i,j} \bmod Y_{i,j}$

Table 11.11: Type table for the remainder operator.

11.4.10 Exponentiation operator

```
x ** y; x ^ y
```

The exponentiation operator has two equivalent forms as associated from right-to-left. Both x and y must be scalar expressions, as shown in Table 11.12.

x	y	z	z = x ** y
scalar	scalar	scalar	$z = x^y$

Table 11.12: Type table for the exponentiation operator.

Additionally, x must be non-negative or y must be an integer; otherwise a mathematical exception will result.

```
[1] write (2 ** 3)
8
[2] write (2 ** .5)
1.41421
```

```
[3] write (-2 ** .5)
stdin:3: exception in expression: illegal base and exponent
```

11.4.11 Transposition operator

```
X'
```

The transposition operator returns the transpose of its operand and associates from left-to-right. The transpose of a scalar is defined to be itself, as shown in Table 11.13.

X	Y	Y = X'
scalar	scalar	$y = x$
matrix	matrix	$Y = X^T$

Table 11.13: Type table for the transposition operator.

11.4.12 Unary operators

```
- X
+ X
not X; ! X
```

The prefix unary operators associate from right-to-left. The unary minus operator simply returns the negative of X. The unary plus operator simply returns X, which must be a matrix or scalar expression. The logical negation operator (`not` or `!`) logically negates each element of X.

$$\text{not } x \overset{\text{def}}{=} \begin{cases} 1 & \text{if } x = 0 \\ 0 & \text{otherwise} \end{cases}$$

The type tables for the unary operators are shown in Table 11.14.

X	Y	X = op Y
scalar	scalar	$x = \text{ op } y$
matrix	matrix	$\forall i, j \mid X_{i,j} = \text{ op } Y_{i,j}$

Table 11.14: Type table for the unary operators.

11.4.13 Index expressions

```
X (I)
X (I,J)
```

An index expression is used to access the components of matrices. Each index must be an expression that is either a scalar, a vector whose elements are *contiguous and increasing*, or the special vector `:`, which is used to specify all rows or columns of a matrix and should not be confused with the range operator. If *I* or *J* is a vector then it specifies a range of indices that are to be used. Thus, `:` is a shorthand for writing `1 : rows (X)` for *I* or `1 : cols (X)` for *J*.

If *X* is a matrix then `X(i)` is its *i*th column and `X(i,j)` is the value of the element of its *i*th row and *j*th column. However, if *X* is a vector then `X(i)` is its *i*th element. Since a scalar is a matrix with one row and one column, it may also be indexed.

```
[1] A = [1, 2, 3; 4, 5, 6]
[2] x = [1; 2; 3]
[3] y = [1, 2, 3]
[4] write (A)

         1           2           3
         4           5           6

[5] write (A(2))

         2
         5

[6] write (A(2,3))
6
[7] write (x)

         1
         2
         3

[8] write (x(2))
2
[9] write (y)

         1           2           3
```

```
[10] write (y(1))
1
[11] write (A(:,[2,3]))

         2           3
         5           6
```

Thus, there is some ambiguity in using `X(i)` since if X is a column vector then the result will be its ith element and if X is a matrix then the result will be its ith column. Therefore, caution should be used; *octave* does not support indexing a matrix with a single index. However, the single index notation can be useful and is consistent with the semantics of the `for` expression of Section 11.7.3.

11.4.14 Function expressions

```
F (...)
```

A function call is an expression, F, followed by parentheses containing a possibly empty, comma-separated list of arguments, which may be arbitrary expressions. The argument list is evaluated in order from left to right. The result of the expression is the return value of the function named by F. If more arguments are specified than required by the function, the extra arguments are simply discarded. If too few arguments are specified, then `null` values are appended to the argument list. Thus, it is impossible to distinguish a `null` value due to an uninitialized variable from a missing argument.

If F evaluates to `null` and is an identifier, such as `write`, then the global scope is searched for a function called `F`. If a function exists then it is executed. Otherwise, the directories named by the environment variable `BURLAP_PATH` are searched in order for a file named first `F` and then `F.b`. If a file is found then it is processed just as if `include()` were called. After the file is processed the global scope is searched again. If the function still cannot be found then a type error results. This is the only case in which a global variable used inside a function does not need to be explicitly declared in a `global` declaration.

11.5 Intrinsic functions

The *intrinsic* functions are those functions that are *burlap* provides by default. They are, however, just variables and their values can be reassigned, although this is not typically done. The functions can be grouped into five basic categories:

- **mathematical functions:** `sin()`, `sqrt()`, `floor()`, etc. (Section 11.5.1)
- **matrix functions:** `lu()`, `qr()`, `eig()`, etc. (Section 11.5.2)
- **predicate functions:** `null?()`, `any?()`, `matrix?()`, etc. (Section 11.5.3)
- **finite element functions:** `assemble()`, `area()`, etc. (Section 11.5.4)
- **miscellaneous functions:** `concat()`, `write()`, `exit()`, etc. (Section 11.5.5)

11.5.1 Mathematical functions

burlap supports most of the functions available in the C math library, as shown in Table 11.15. Each of the functions can take either a scalar or a matrix as an argument. If a matrix is passed as an argument, then the result is also a matrix with the function applied to each element individually.

`abs (x)`	$\lvert x\rvert$	`hypot (x,y)`	$\sqrt{x^2+y^2}$
`ceil (x)`	$\lceil x\rceil$	`log (x)`	$\ln x$
`cos (x)`	$\cos x$	`log10 (x)`	$\log x$
`exp (x)`	e^x	`pow (x,y)`	x^y
`fabs (x)`	$\lvert x\rvert$	`sin (x)`	$\sin x$
`floor (x)`	$\lfloor x\rfloor$	`sqrt (x)`	$\sqrt{x}$
`fmod (x,y)`	$x \bmod y$	`tan (x)`	$\tan x$

Table 11.15: Intrinsic functions from the math library.

In the case of the `hypot()` function which requires two arguments, the type semantics are the same as that of the addition operator. The `fmod()` function is equivalent to the remainder operator (`%`); the `pow()` function is equivalent to the exponentiation operator (`**`).

11.5.2 Matrix functions

burlap provides a variety of functions for manipulating and factoring matrices. All of these functions, with the exception of `rows()` and `cols()` functions, could be written in *burlap* itself; however, the built-in versions will execute faster. Detailed explanations of the various factorizations can be found in any basic linear algebra text.

`chol (X)` Returns the cholesky factorization of X. The cholesky decomposition of a matrix X is a matrix B such that $BB^T = X$, provided that X is symmetric and positive definite. The cholesky decomposition of a scalar is simply its square root.

`cols (X)` Returns the number of columns of the matrix X. A scalar is defined to have a single column. A synonym for this function is `columns()`.

`compact (X)`

Returns a *compact-column* matrix whose elements are identical to those of X, provided that X is a symmetric matrix. Matrices are normally *full* matrices, with the size of an $m \times n$ full matrix requiring $O(mn)$ space. The space required by a compact-column matrix is approximately equal to the number of non-zero entries. Most of the FElt functions expect compact-column matrices. The compact representation of a scalar is itself.

`det (X)` Returns the determinant of X, $|X|$, provided that X is a nonsingular matrix. The determinant of a scalar is itself.

`eig (X, V)` Returns a column vector, Λ, containing the eigenvalues of X, provided that X is a square matrix. Therefore, $\forall i \; \exists y \mid Xy = \Lambda_i y$. If a variable V is specified *and* X is a symmetric matrix then V will contain the matrix of eigenvectors on return. Each column of V will contain a single eigenvector. Therefore, $\forall i \mid XV_i = \Lambda_i V_i$. If V is missing or X is not a symmetric matrix then only the eigenvalues are returned.

`eye (m, n)` Returns an identity matrix of size $m \times n$, I_{mn}. If n is omitted then an $m \times m$ matrix is returned. Both m and n must be scalar expressions that are greater than zero.

`inv (X)` Returns the inverse of X (X^{-1}), which must be either a nonsingular matrix or a non-zero scalar.

`lu (X, L, U, P)`

Computes the LU decomposition of X, which must be a nonsingular matrix. The return value is a row permuted superposition of L and U, with the diagonal of L not being stored since L is unit lower triangular. Thus, the result is not very useful. However, if L is specified as an argument, it will contain the unit lower triangular matrix on return. Similarly, U will contain the upper triangular matrix and P will contain the permutation matrix, such that $PLU = X$.

`norm (X, s)`

Returns the norm of X, $\|X\|$. The type of the norm depends upon the type of X and the value of s, which must be a string, as shown in Table 11.16.

X	s: `null`	`"1"`	`"2"`	`"fro"`
scalar	$\lvert X\rvert$	$\lvert X\rvert$	$\lvert X\rvert$	$\lvert X\rvert$
vector	$\|X\|_2$	$\|X\|_1$	$\|X\|_2$	$\|X\|_F$
matrix	$\|X\|_F$	$\|X\|_1$	illegal	$\|X\|_F$

Table 11.16: Argument table for the `norm()` function.

The notation $\|X\|_F$ indicates the *Frobenius* norm of X; the Frobenius norm of a vector X is equivalent to its 2-norm, $\|X\|_2$. Since a scalar value can always be converted to a string value, it is legal to use `norm (X, 1)` or `norm (X, 2)`, but not `norm (X, fro)` unless `fro` evaluates to `"1"`, `"2"` or `"fro"`.

`ones (m, n)`

Returns an $m \times n$ matrix whose elements are all `1`. If n is not specified then an $m \times m$ matrix is returned. Both m and n must evaluate to scalars greater than zero.

`qr (X, Q, R)`

Computes the QR factorization of X such that $Q^T X = R$ and $QQ^T = I$. The result is a right upper triangular matrix, R. The orthogonal matrix, Q, may be retrieved by specifying it as the second parameter; R may also be retrieved in this manner if desired. The matrix X must be overdetermined (i.e., tall and thin).

`rand (m, n, s)`

Returns an $m \times n$ matrix whose elements are randomly distributed in $[0,1)$. If n is not specified then an $m \times m$ matrix is returned. If both m and n are not specified then a scalar is returned. The third argument, s, will be used to seed the random number generator if it is specified and is a non-zero scalar. *burlap* initially seeds the generator with the current time.

`rows (X)` Returns the number of rows of the matrix X. A scalar is defined to have a single row.

`zeros (m, n)`
: Returns an $m \times n$ matrix whose elements are all `0`. If n is not specified then an $m \times m$ matrix is returned. A synonym for this function is `zeroes()`.

The `eye()`, `ones()`, `rand()`, and `zeros()` functions provide alternative ways of creating a matrix without using the matrix formers.

11.5.3 Predicate functions

Predicate functions (Table 11.17) return `1` if the predicate is true and `0` otherwise. The names of all predicate functions in *burlap* end with a question mark, `?`, but this is simply a convention and is not required should you write your own functions.

`any? (X)`	Is any element of X non-zero?
`compact? (X)`	Is X a compact-column matrix?
`every? (X)`	Is every element of X non-zero?
`null? (X)`	Is X null?
`matrix? (X)`	Is X a matrix?
`scalar? (X)`	Is X a scalar?
`symmetric? (X)`	Is X a symmetric matrix?

Table 11.17: Predicate functions available in *burlap*.

The `any?()`, `compact?()`, `every?()`, and `symmetric?()` functions require X to be either a scalar or a matrix. The remaining functions will accept any type of object as an argument.

```
[1] x = [1, 2, 3]
[2] y = 4
[3] write (scalar? (x), " ", scalar? (y))
0 1
[4] write (matrix? (x), " ", matrix? (y))
1 0
[5] z = x'*x
[6] write (symmetric? (z), " ", compact? (z))
1 0
[7] write (compact? (compact (z)))
1
```

11.5.4 Finite element functions

The finite element functions provide an interface to the functions in the FElt library. The functions can all be written in *burlap* itself, but the built-in functions will execute faster.

`add_definition (...)`

Adds a new element definition. After the definition is added, elements of that particular type can be loaded using the `felt()` function. The types of the arguments are given in order in Table 11.18. Examples and a more in-depth discussion of the arguments can be found in Section 10.4.

name of definition	`string`
set up function (K and M)	`function`
stress computation function	`function`
element shape	`scalar`
number of nodes in element	`scalar`
number of nodes that define shape	`scalar`
number of stress values	`scalar`
vector of DOFs	`row vector`
retain stiffness matrix indicator	`scalar`

Table 11.18: Arguments to the `add_definition()` function.

`area (e)` Returns the area of the element e. The element must be a planar element (i.e., `e.definition.shape` should be `&planar`).

`assemble (M, C)`

Computes and returns the global stiffness matrix, K, for the currently defined FElt problem by computing the local stiffness matrices and assembling them into the global matrix. For transient problems, if M is specified then it will contain the global mass matrix on return. Similarly, C will contain the global damping matrix. All matrices are compact-column matrices. The DOF-related fields of the `problem` structure must be initialized, as by calling the `find_dofs()` function.

`clear_nodes ( )`

Clears the displacements and equivalent nodal force vectors for all nodes defined in the current FElt problem. The return value is `null`.

`compute_modes (K, M, X)`
: Computes the modes for given stiffness matrix, K, and mass matrix, M. The result is the vector of eigenvalues, Λ. If X is specified then it will contain the matrix of eigenvectors upon return (see `eig()` in Section 11.5.2). The DOF-related fields of the `problem` structure must be initialized, as by calling `find_dofs()`.

`compute_stresses (e)`
: Computes the stresses for the element e. The return value is the return value of the element's stress function. The `stress` field of the element is initialized by this function. This function is merely a shorthand for `e.definition.stress (e)`.

`construct_forces (t)`
: Constructs and returns the global nodal force vector, f, for the current FElt problem. The vector is constructed based on all nodal forces and the global DOFs at those nodes. For transient problems, t may be a scalar expression used to specify the current time. If t is missing then it is assumed to be `0`. The DOF-related fields of the `problem` structure must be initialized, as by calling `find_dofs()`.

`felt (s)`
: Reads the FElt file named by s to define the current FElt problem. If s does not name an absolute path (i.e., starts with / or ˜) then the environment variable `FELT_PATH` is used to search for the file named by s. The variable should be a colon-separated list of directories. The directories are searched from left-to-right for a file first named s and then s`.flt`. If the variable `FELT_PATH` is not set, then only the current directory is searched. If s is `null` then an empty problem is defined.

`find_dofs ( )`
: Computes the set of active DOFs for the current problem. As a result, the DOF-related fields of the `problem` structure are initialized. The number of active DOFs is returned. This function must be called before most of the other finite element related functions can be called.

`global_dof (n, d)`
: Returns the global DOF corresponding to a local DOF. The local DOF is specified by its node, n, and its DOF, d. The node, n, may be specified as either a node object or a node number. The DOF, d, should be one of `&tx`,

`&ty`, `&tz`, `&rx`, `&ry`, or `&rz`. The DOF-related fields of the `problem` structure must be initialized, as by calling `find_dofs()`.

`integrate_hyperbolic (K, M, C, p)`
: Solve the discrete equation of motion, $Ma+Cv+Kd=f$, using Newmark's method with the Hilbert-Hughes-Taylor α-correction for improved accuracy with numerical damping. The return value is a matrix, D, of nodal displacements, with each column of D corresponding to a single time step. If the nodes of the FElt problem have been renumbered then p should be used to specify the permutation vector, as returned by the `renumber_nodes()` function. The sizes of the matrices must be consistent with the definition of the current problem. Compact-column matrices are expected, but full symmetric matrices will be accepted by coercing them to compact-column matrices. The DOF-related fields of the `problem` structure must be initialized, as by calling `find_dofs()`.

`integrate_parabolic (K, M, p)`
: Solves the discrete parabolic differential equation $Mv+Kd=f$ using a generalized trapezoidal method. If the nodes have been renumbered then p should be used to specify the permutation vector. The sizes of the matrices must be consistent with the definition of the current problem. Compact-column matrices are expected, but full symmetric matrices will be accepted by coercing them to compact-column matrices. The DOF-related fields of the `problem` structure must be initialized, as by calling `find_dofs()`.

`length (e)`
: Returns the length of the element e. The element must be a linear element (i.e., `e.definition.shape` should be `&linear`).

`local_dof (g, l)`
: Returns the number of the node corresponding to the global DOF, g. If l, is specified then it will contain the local DOF on return. The number of the node is returned rather than the node object itself since the nodes of the problem may have been renumbered. The DOF-related fields of the `problem` structure must be initialized, as by calling `find_dofs()`.

`remove_constrained (K)`
: Removes the rows and columns of K at all DOFs with a fixed boundary condition and returns the new matrix. K itself is not modified. K should be either a symmetric matrix or a column vector; the size of K must be

consistent with the definition of the current problem. The DOF-related fields of the `problem` structure must be initialized, as by calling `find_dofs()`.

`remove_definition (n)`
: Removes the element definition named by *n*. If the definition is successfully removed then `0` is returned. Otherwise, `1` is returned. The definition must not be in use by any elements.

`renumber_nodes ( )`
: Renumbers the nodes of the current problem using the Gibbs-Poole-Stockmeyer and Gibbs-King node renumbering algorithms for bandwidth and profile reduction. The result is a permutation vector, *p*, of the node numbers.

`restore_numbers (p)`
: Restores the original node numbers of the current problem. The permutation vector, as returned by the `renumber_nodes()` function, is specified by *p*. The return value is `null`.

`set_up (e, s)`
: Calls the set-up function for element *e*. The argument *s* may be used to specify the mass mode and should be either `&lumped` or `&consistent`. If *s* is not specified then no mass matrix will be computed for *e*. This function is merely a shorthand for `e.definition.set_up (e, s)`.

`solve_displacements (K, f)`
: Solves the linear system $Kd = f$ for the vector of global nodal displacements, d. The sizes of the inputs must be consistent with the definition of the problem. Additionally, K and f should both be condensed; K is expected to be compact. The DOF-related fields of the `problem` structure must be initialized, as by calling `find_dofs()`.

`volume (e)`
: Returns the volume of the element *e*. The element must be a solid element (i.e., `e.definition.shape` should be `&solid`).

`zero_constrained (K)`
: Zeroes the rows and columns of K at all DOFs with a fixed boundary condition and returns the new matrix, K_{cond}. K itself is not modified. K should be either a symmetric matrix or a column vector. If K is a symmetric matrix then a `1` is placed on the diagonal of each zeroed row/column. The size of K

must be consistent with the definition of the problem. The DOF-related fields of the `problem` structure must be initialized, as by calling `find_dofs()`.

These functions alone are sufficient to solve all static and transient problems that *felt* or *velvet* can solve. As shown in Figure 11.1, it is relatively simple to solve a FElt problem using the finite element intrinsic functions of *burlap*.

11.5.5 Miscellaneous functions

The remaining intrinsic functions are concerned with a variety of topics, including file access, input/output, and the user-interface.

`concat (s, t)`
: Returns the concatenation of *s* and *t*, both of which must be strings.

`eval (s)`
: Evaluates the string *s* as a *burlap* expression. The return value of the function is the return value of the evaluated expression.

`exit (n)`
: Exits *burlap* with exit code *n*. Normally, an exit code of `0` indicates success and a non-zero exit code indicates an error. If *n* is missing then `0` is used. Note that the built-in aliases define `exit` as `exit (!*)`, so from the prompt, typing `exit` will exit *burlap* with exit code `0`. There is no return value from the function.

`help (s)`
: Requests help on topic *s*. If *s* is not specified or is the empty string (`""`) then a list of help topics is printed. If *s* is the name of an intrinsic function then the help topic for that function is printed. Note that the built-in aliases define `help` as `help ("!$")`, so from the prompt you need to type `help foo` rather than `help "foo"` to retrieve information on topic `foo`; in fact, `help ("foo")` will not work since it will expand to `help ("(")`.

`history (n)`
: Prints the command history list. If *n* is given then only the last *n* events are printed. The built-in set of aliases define `h` as `history (20)`.

`include (s)`
: Includes the file named by *s*. The file is interpreted just as if it it were specified on the command line. Therefore, it is executed in the global scope and *not* within a function body, even if the `include()` should be inside a

```
find_dofs ( )

if problem.mode == &static then
    K = assemble ( )
    F = construct_forces ( )
    Kc = zero_constrained (K)
    Fc = zero_constrained (F)
    d = solve_displacements (Kc, Fc)

else if problem.mode == &transient then
    K = assemble (M, C)
    D = integrate_hyperbolic (K, M, C)

else if problem.mode == &modal then
    K = assemble (M, C)
    Kc = remove_constrained (K)
    Mc = remove_constrained (M)
    Cc = remove_constrained (C)
    l = compute_modes (K, M, X)

else if problem.mode == &static_thermal then
    K = assemble ( )
    F = construct_forces ( )
    Kc = zero_constrained (K)
    Fc = zero_constrained (F)
    d = solve_displacements (Kc, Fc)

else if problem.mode == &transient_thermal then
    K = assemble (M)
    D = integrate_parabolic (K, M)
end
```

Figure 11.1: Solving problems with the finite element intrinsic functions.

function body. If *s* does not name an absolute path (i.e., starts with `/` or `~`) then the environment variable `BURLAP_PATH` is used to search for the file named by *s*. The variable should be a colon-separated list of directories. The directories are searched from left-to-right for a file first named *s* and then *s*`.b`. If the variable `BURLAP_PATH` is not set, then only the current directory is searched.

`length (X)` Returns the length of the object X. If X is a matrix then it returns the number of elements in X. If X is a string then it returns the number of characters in X.

`load (s)` Loads a file, *s*, of saved variables created with the `save()` function. At present, this function is not implemented.

`read ( )` Reads a line from standard input and returns it as a string. Any newline character is discarded. The function returns `null` upon end of file.

`reads ( )` Reads a string (i.e., a sequence of characters separated by spaces) from standard input and returns it as a string. A `null` value is returned upon end of file.

`save (s)` Saves the current set of global variables in the file named by *s*. At present, this function is not implemented.

`system (s)` Executes the UNIX command named by *s*. The command is executed in its own subshell. The return value of the function is the return status of the command.

`type (X)` Returns a string containing the type of X, which may be of any type.

`write (...)`

Writes its arguments to standard output with no intervening spaces but *with* a newline at the end of the output. The arguments may be of any type. A matrix is always written on a series of separate lines. The return value is always `0`.

`writes (...)`

Writes its arguments to standard output with no intervening spaces and *without* a newline at the end of the output. The arguments may be of any type. A matrix is always written on a series of separate lines. The return value is always `0`.

11.6 User-defined functions

> `function` *name* `(` [*argument* [`,` *argument*] ...] `)`
>   *expression-list*
> `end`

burlap provides a simple syntax for allowing users to define their own functions. A function definition creates a new function with the specified *name*, overwriting any previously assigned value. Function names are simply variables just as in any of the previous examples. Functions can be passed as parameters to other functions or returned from functions, just like ordinary scalar variables.

The *expression-list* is a semicolon separated list of expressions that comprise the body of the function. The return value of the function is the value of the last *expression*, although usually a RETURN expression is used to explicitly return a value from the function (Section 11.7.4).

Each *argument* specifies the name of a formal parameter to the function (Section 11.4.14). By default, all arguments are passed *by value*; the actual parameter is evaluated and a copy is passed to the function with assignment to the formal parameter having no effect on the caller of the function. If the keyword `shared` is placed before the name of the formal parameter (e.g., `shared foo`) then the data associated with the formal is shared with the data of the actual. An assignment to one is implicitly an assignment to both. *burlap* uses this ability in several intrinsic functions (e.g., `lu()`, `eig()`) to "return" more than one value from a function.

```
function swap (x, y)
    t = x; x = y; y = t
end

[1] a = 1; b = 2
[2] swap (a, b)
[3] write (a, " ", b)
1 2
```

The values are not swapped as desired since they are passed by value. For the `swap()` function to work correctly the data of the formal and actual parameters must be shared.

```
function swap (shared x, shared y)
    t = x; x = y; y = t
end
```

```
[1] a = 1; b = 2
[2] swap (a, b)
[3] write (a, " ", b)
2 1
```

This type of parameter passing is also known as *call-by-reference* or *var-parameters*. Fortran passes all parameters by reference; C passes all parameters by value. *burlap* adopts the term *shared parameters* to illustrate that the space associated with the formal and actual parameters is shared.

11.7 Control-flow constructs

burlap supports the standard control-flow constructs found in most programming languages. The principal difference is that control-flow constructs in *burlap* are expressions, not statements, so they can be used anywhere.

11.7.1 IF expressions

```
  if expression then
      expression-list
[ else if expression then
      expression-list ] ...
[ else
      expression-list ]
  end
```

First, the `if` *expression* is evaluated. If the result is non-zero, the `then` *expression-list* is executed. Otherwise each successive `if` *expression* is evaluated in turn, and if its result is non-zero, the corresponding `then` *expression-list* is executed. If none of the `if` *expressions* evaluates to non-zero, the `else` *expression-list* is executed. Each `if` *expression* must evaluate to a scalar. The IF expression has the highest precedence, so `x := if y < z then y else z` works as expected.

The *expression-list* is a semicolon separated list of expressions. The result of the *expression-list* is the result of the last expression in the list. The `else if` and `else` constructs are optional. The `else if` syntax is combined into one keyword that allows the entire IF expression to end with a single `end` keyword. To have `else if` treated as two keywords, thus requiring multiple `end` keywords, place a semicolon or other expression between the `else` and the `if`.

If an *expression-list* is missing then it evaluates to `null`. Similarly, if the final `else` is missing and none of the `if` *expressions* evaluated to non-zero, the result of the entire IF expression will be `null`.

```
function max (x, y)
    if x > y then x else y end
end

[1] write (max (1, 2))
2
```

In this example, the IF expression returns x if $x > y$ and returns y if $x \leq y$. Since the return value of a function is the value of the last expression, the function `max()` returns the result of the IF expression.

```
function max (x, y)
    if x > y then return x
    else return y
    end
end
```

This function illustrates a more conventional way of writing the `max()` function. The result is the same regardless of which function is used.

11.7.2 WHILE expressions

`while` *expression* `do`
    expression-list
`end`

As long as the `while` *expression* evaluates to non-zero, the *expression-list* is evaluated. The *expression-list* is a semicolon separated list of expressions. If the `while` *expression* evaluates to zero, then WHILE construct terminates and the result is a `null` value. The BREAK expression of Section 11.7.4 may be used to alter the result of the WHILE. The `while` *expression* must evaluate to a scalar.

```
function sum_of_first (n)
    i = 0
    s = 0
    while i <= n do
        s = s + i
        i = i + 1
    end
    return s
end

[1] write (sum_of_first (3))
6
[2] write (sum_of_first (10))
55
[3] write (while 0 do end)
null
```

This example uses a WHILE expression to compute the sum of the first *n* integers. The last `write()` function illustrates the result of the WHILE loop. The WHILE expression has the highest precedence, although this rarely matters.

11.7.3 FOR expressions

```
for expression in expression do
    expression-list
end
```

The FOR construct first evaluates the `for` *expression* and then the `in` *expression*. For each element in the `in` *expression*, the element is assigned to the `for` *expression* and the *expression-list* is executed. The *expression-list* is a semicolon separated list of expressions. The result of the FOR expression is a `null` value unless a BREAK expression (Section 11.7.4) is used to explicitly produce a result. The FOR expression has the highest precedence, although this rarely matters.

The `for` *expression* must be an expression that can be assigned a value (i.e., can appear on the left-hand side of an assignment expression). The `in` *expression* must be a matrix, vector, scalar, or the `null` value. If it is a matrix then each column of the matrix is assigned to the `for` *expression*. If it is a vector then each element is assigned. If it is scalar then the scalar is assigned, and the *expression-list* is evaluated only once. If the `in` *expression* is `null`, then the *expression-list* is not executed.

```
function sum_of_first (n)
    s = 0
```

```
        for i in 1 : n do
            s = s + i
        end
        return s
    end

    [1] write (sum_of_first (3))
    6
    [2] write (sum_of_first (10))
    55
    [3] write (sum_of_first (-1))
    0
```

This last `write()` expression works correctly since the range `1 : -1` produces a `null` value and thus the body of the FOR expression is not executed.

```
    function sum (x)
        s = 0
        for v in x do
            for i in v do
                s = s + i
            end
        end
        return s
    end
```

This example correctly computes the sum of all elements of *x* for scalars, vectors, and matrices. If *x* is simply `3` then *v* is simply `3` and *i* is also `3`. If *x* is the vector `[1, 2, 3]` then *v* is successively assigned `1`, `2`, and `3` and *i* is simply assigned each value of *v*. If *x* is the matrix `[1, 2, 3; 4, 5, 6]` then *v* is successively assigned the column vectors `[1; 4]`, `[2; 5]`, and `[3; 6]` and *i* is assigned the values `1`, `4`, `2`, `5`, `3`, and `6` in that order.

11.7.4 BREAK, NEXT, and RETURN expressions

`return` [*expression*]
`break` [*expression*]
`next`

The RETURN expression may be used to return a value from a function. If the *expression* is absent then a `null` value is returned. A RETURN expression may only occur inside a function body and has the lowest possible precedence.

The BREAK expression may be used to explicitly exit and return a value from a FOR or WHILE loop. If the *expression* is absent then a `null` value is returned. A BREAK expression is executed by first evaluating the *expression* if it is present, and then transferring control out of the nearest enclosing FOR or WHILE loop. The BREAK expression has the lowest possible precedence.

```
function search (x, a)
    found = 0
    for v in x do
        if v == a then
            found = 1
            break
        end
    end
    write ("found = ", found)
end
```

If the value *a* is found within the vector *x* then there is no point in continuing with the search after setting the `found` indicator.

The NEXT expression may be used to skip to the next iteration of the nearest enclosing FOR or WHILE loop, and has the highest possible precedence.

Chapter 12

The Algorithms Behind FElt

12.1 Some background

In general, there is a massive base of experience in the numerical analysis community that relates to programming the finite element method. Any time you see the method implemented, chances are that the implementation is in some piece of software. So why is it that good books on computational techniques are hard to come by and good books on the underlying theory and mathematics abound? Go figure. As we mentioned in chapter 2, if you need some references to the latter we like Hughes' [9] and Zienkiewicz and Taylor's [16] "classic" textbooks. Logan's [13] book is a good introductory text. He approaches the method from the same direction that many of us come to finite elements, through classical matrix structural analysis. He's a bit lean on the mathematical basis and theory, but that's probably why it makes a good introductory text. It is probably our ignorance, but we are not aware of any definitive texts on the algorithmic implementation of the finite element method. Texts that have been recommended, but that we have not really worked with include Segerlind [14], a very well-recommended introductory text by Burnett [1] and a book by Hinton and Owen [8].

12.2 Elementary C programming

The advantages to coding FElt entirely in C rather than the more traditional (at least for finite element analysis) choice of Fortran are: 1. keeping the package consistent – the system and graphical interface stuff required working in C, having all of the mathematics in C made interfacing the two aspects a lot easier. 2. C's flexibility in working with data structures makes it really easy to implement some of the concepts of finite element analysis in the code – elements have pointers to materials and loads, nodes have pointers to forces and constraints. There is no need to keep track of lots of separate arrays of things; generally speaking, passing the node and element arrays around is all the information that a routine will need (or even just passing the elements around as they contain pointers into the nodes array).

For those unfamiliar with C, here's a quick and dirty lesson that might help you read the code with a little more comprehension. Basically, in C you can declare new types of variables, including entire data structures. What this means is that we can define a structure called an Element which has several different members (think of each member as a different field or record in a database). When we declare a variable to be of type Element, that variable then contains all these different fields and we can get at all the information for that element through that one variable. For instance, in FElt an element structure contains all of the following:

```
Element {
    unsigned    number;          /* this element's global number          */
    Node        node [ ];        /* array of pointers to element's nodes   */
    Matrix      K;               /* element stiffness matrix               */
    Definition  definition;      /* definition structure (type) of element */
    Material    material;        /* pointer to material property          */
    Distributed distributed [4]; /* array of distributed loads             */
    unsigned    numdistributed;  /* number of distributed loads assigned   */
    Stress      stress [ ];      /* element stresses                       */
    unsigned    ninteg;          /* number of integration points          */
}
```

Several of these members are structures themselves: `Node`, `Matrix`, `Distributed`, `Definition`, `Material` and `Stress` are other structures used in FElt.

Because most of the FElt routines access the structures as pointers (that is usually when the code references a structure it is actually referencing the address of that structure in memory ... this is something you shouldn't need to worry about to simply read and understand the mathematics), the usual way that you will see code that refers to elements is like this: `element -> node[1] -> x`, or `element -> material -> E`. The first of these examples gets the value of the x-coordinate of the element's first node (not necessarily globally numbered node 1). The second gets the value of the Young's modulus of the material assigned to this element. If the array of elements is under consideration, you might seem something like this: `element[3] -> stress[1] -> values[1]` . This will be the value of the first stress component at the first integration point on element 3. The other common data structures in FElt are:

```
/* A reaction force */

typedef struct reaction {
    double   force;                     /* reaction force             */
    unsigned node;                      /* node number                */
    unsigned dof;                       /* affected degree of freedom */
} *Reaction;

/* Element stress */

typedef struct stress {
    double  x;                          /* x coordinate           */
    double  y;                          /* y coordinate           */
    double  z;                          /* z coordinate           */
    double  values [ ];                 /* computed stress values */
} *Stress;

/* An element definition */

typedef struct definition {
    char      *name;             /* element symbolic name              */
    int      (*setup) ( );       /* element initialization function    */
    int      (*stress) ( );      /* stress computation function        */
    Shape     shape;             /* element dimensional shape          */
    unsigned  numnodes;          /* number of nodes in element         */
    unsigned  shapenodes;        /* number of nodes which define shape */
    unsigned  numstresses;       /* number of computed stress values   */
    unsigned  numdofs;           /* number of degrees of freedom       */
    unsigned  dofs [7];          /* degrees of freedom                 */
    unsigned  retainK;           /* retain element K after assemblage  */
} *Definition;

/* A distributed load */

typedef struct distributed {
    char     *name;             /* name of distributed load */
    Direction direction;        /* direction of load        */
    unsigned  nvalues;          /* number of values         */
    Pair      value [ ];        /* nodes and magnitudes     */
} *Distributed;

/* A force */
```

```
typedef struct force {
    char     *name;                /* name of force           */
    VarExpr   force [7];           /* force vector            */
    VarExpr   spectrum [7];        /* vector of input spectra */
} *Force;

/* A constraint */

typedef struct constraint {
    char   *name;                /* name of constraint            */
    char    constraint [7];      /* constraint vector             */
    double  ix [7];              /* initial displacement vector   */
    double  ax [4];              /* initial acceleration vector   */
    double  vx [4];              /* initial velocity vector       */
    VarExpr dx [7];              /* boundary displacement vector  */
} *Constraint;

/* A material */

typedef struct material {
    char  *name;               /* name of material                          */
    double E;                  /* Young's modulus                           */
    double Ix;                 /* moment of inertia about x-x axis          */
    double Iy;                 /* moment of inertia about y-y axis          */
    double Iz;                 /* moment of inertia about z-z axis          */
    double A;                  /* cross-sectional area                      */
    double J;                  /* polar moment of inertia                   */
    double G;                  /* bulk (shear) modulus                      */
    double t;                  /* thickness                                 */
    double rho;                /* density                                   */
    double nu;                 /* Poisson's ratio                           */
    double kappa;              /* shear force correction                    */
    double Rk;                 /* Rayleigh stiffness damping coefficient */
    double Rm;                 /* Rayleigh mass damping coefficient         */
    double Kx;                 /* conductivity in x direction               */
    double Ky;                 /* conductivity in y direction               */
    double Kz;                 /* conductivity in z direction               */
    double c;                  /* heat capacity                             */
} *Material;

/* A node */

typedef struct node {
        unsigned   number;          /* node number                     */
        Constraint constraint;      /* constrained degrees of freedom */
        Force      force;           /* force acting on node           */
        double     m;               /* lumped mass at this node       */
        double     eq_force [ ];    /* equivalent force               */
        double     dx [7];          /* displacement                   */
        double     x;               /* x coordinate                   */
        double     y;               /* y coordinate                   */
        double     z;               /* z coordinate                   */
    } *Node;
```

Hopefully the organization of the data structures is intuitive enough to someone familiar with finite elements that a detailed understanding of the mechanics of the C language will not be necessary.

12.3 Introduction to the general FElt routines

Throughout FElt, simplicity and readability have generally won out over speed and efficiency (at least where the mathematics are concerned ... the system and GUI code are another story entirely). What this means is that FElt does not make use of many of the algorithmic tricks that have been developed over the years for working with finite elements. The most glaring example of this is probably the fact that the FElt matrix manipulation library is fairly simple and does not make use of symmetry in most cases.

12.4 Details of a few general FElt routines

12.4.1 Finding the active DOF

The first step in the solution of a FElt problem is to find all the degrees of freedom that the different types of elements in the problem use. This information gets stored in two different six-element arrays. In the `dofs_pos` array a non-zero value in the *i*th position indicates that the *i*th DOF is active. The actual number in the *i*th position indicates which DOF this is for the current problem (i.e. in a problem consisting solely of beams, a 3 in the sixth position indicates that rotation about the z-axis is the third DOF in the current problem). The second array is `dofs_num`. In this array, the entries from one through the number of active DOF are all non-zero. A value of *j* in the *i*th position indicates that the standard DOF *i* is the *j*th DOF for this problem (i.e., for our example problem consisting of beams only, a 6 in the third position indicates that the third active DOF for this problem is rotation about the z-axis). Both of these arrays are available globally through the `analysis` structure.

12.4.2 Node renumbering

We all know that the profile and bandwidth of a global stiffness or mass matrix – critical factors in determing memory requirements and ultimate solution speed – are a function of the global node numbering. The contribution from each element is assembled into the global matrices based on the global numbers of the elements' nodes. Better node numbering schemes take this into account and try to keep the contribution from each element as near to the diagonal of the global matrix as possible. Given the compact column storage scheme described below, a matrix with most entries very near the diagonal can use significantly less storage than a very full matrix. The linear equation solver can also make use of this reduced matrix and operate a lot faster by not having to operate on a lot of zero entries.

There are numerous algorithms available to try to optimize node numbering with just such a goal in mind. The one that we use in FElt is popularly known as Gibbs-King [6], the profile reduction variant of the Gibbs-Poole-Stockmeyer algorithm [7] which primarily tries to minimize bandwidth. Other popular algorithms include Cuthill-Mckee [3] and Reverse Cuthill-McKee [5].

12.4.3 Assembling the global stiffness matrix

This step of the process in FElt actually accomplishes three things. For each element, this routine calls an element setup routine based on element type. Besides filling out `element -> K` most element setup routines will also calculate the equivalent nodal forces on the element's nodes if appropriate (i.e., if there is a distributed load on the element).

Assembling the stiffness and mass matrices in a transient analysis problem works the same way. For transient analysis, the element setup routines calculate a mass matrix according to the the mass-mode set in the analysis parameters section. The global stiffness and mass matrices are formed exactly as in the static case. Nodal lumped masses are superposed after the element mass matrices have been assembled.

The damping matrix can be formed in either of two ways – both of them based on a Rayleigh damping model. In the first method, the damping matrix is assembled at the same time as the stiffness and mass matrices by creating and assembling element damping matrices based on the Rayleigh damping coefficient for each element's material,

$$C_e = {R_k}^e K_e + {R_m}^e M_e. \tag{12.1}$$

Where ${R_k}^e$ and ${R_m}^e$ are the Rayleigh damping coefficients for the given material. Alternatively, if non-zero R_k and R_m values are given in the analysis parameters of the FElt file, then these coefficients will be applied to form a global damping matrix only after the global mass and stiffness matrices have been completely assembled,

$$C = R_k K + R_m M. \tag{12.2}$$

12.4.4 Compact column representation

All of the global matrices are assembled directly into a compact column representation. This representation tries to minimize storage requirements by taking each column of the matrix and only storing the entries from the first non-zero entry to the diagonal. In this scheme the following 6×6 would require a vector of length 14 to store as opposed to a matrix with 36 entries,

$$\begin{bmatrix} x & 1 & 0 & 0 & 0 & 0 \\ & x & 1 & 0 & 1 & 0 \\ & & x & x & 1 & 0 \\ & & & x & 0 & 1 \\ & & & & x & x \\ & & & & & x \end{bmatrix}.$$

Assembly is done in two passes through the elements. The first pass is used to optimize the storage scheme by finding out just how tall each column needs to be. Given this information, vectors are actually allocated to hold the global matrices. This first pass is also used to construct a vector of diagonal addresses. For each row of what would be the full matrix, this vector contains the position in the compact form of the diagonal of that row. For our above example, that vector would be

$$\begin{bmatrix} 1 & 3 & 5 & 7 & 11 & 14 \end{bmatrix}.$$

The second pass of the assembly process actually inserts the appropriate contribution from each element into these vectors.

Any routine that needs to access the global stiffness or mass matrices can behave just as if they were full two-dimensional matrix representations. We achieve this transparency through calls to a function which uses the information in the diagonal addresses array (which is attached as a member of the standard matrix data structure) to convert from a row-column location to a single address in the global vector representations.

12.4.5 Dealing with boundary conditions

FElt can handle several types of boundary conditions. The simplest is a fixed DOF at a given node. In this case, the row and column of the global stiffness matrix corresponding to the fixed DOF are simply filled with zeros and a one is placed on the diagonal at that location; a zero is placed in the appropriate DOF of the global force vector just for clarity.

The second type of condition is a displacement condition such as might be found in a settlement of support problem. In this case, before we eliminate the rows and columns of the stiffness matrix, we need to adjust the force vector to account for the displacement. Given a globally numbered DOF, *n*, which has a displacement condition, dx, global force vector, F, and global stiffness matrix, K, then for each DOF, *i*, in the problem,

$$F(i) = F(i) - K(i,n)dx. \tag{12.3}$$

After this adjustment, the rows and columns of K associated with DOF n can be zeroed out as in the ordinary fixed case. The magnitude of the displacement condition, dx, is placed in the $F(n)$ to insure that dx will re-appear exactly in the final solution for nodal displacements.

The final type of condition is a hinge and because the primary adjustments that this condition requires were made in the element stiffness matrix routines, all we do here is zero out rows and columns of the stiffness matrix and force vector as in the fixed case.

12.4.6 Solving for nodal displacements

The FElt routine to solve for nodal displacements (or to solve any linear system of equations) takes the compact column representation and solves $Kd = F$ for the global displacement vector using a skyline solver. This procedure basically involves a Crout factorization and forward and backward substitution on the compact column representation of the global stiffness matrix.

12.4.7 Time integrating in transient structural analysis

Time-stepping in a transient structural analysis problem is done with the Hilbert-Hughes-Taylor alpha variant of Newmark integration. There are three critical parameters in this algorithm. The first two, β and γ, are the standard parameters of Newmark integration. Depending on the values for these two parameters we can implement several well-known algorithms. Setting $\beta = \frac{1}{4}$ and $\gamma = \frac{1}{2}$ results in an unconditionally stable average acceleration (trapezoidal rule) implementation. $\beta = \frac{1}{6}$ and $\gamma = \frac{1}{2}$ results in a linear acceleration algorithm.

The HHT-α algorithm adds a third parameter, α, to account for the decrease in order of accuracy that results when you introduce numerical damping into the Newmark method. Setting $\alpha = 0$ reduces the problem to a standard Newmark method. Choosing $\alpha \in [-\frac{1}{3}, 0]$, $\gamma = (1-2\alpha)/2$ and $\beta = (1-\alpha)^2/4$ results in an unconditionally stable, second-order accurate algorithm [9].

With these three parameters, the time-discrete equation of motion in the HHT-α method is written as

$$Ma_{i+1} + (1+\alpha)Cv_{i+1} - \alpha Cv_i + (1+\alpha)Kd_{i+1} - \alpha Kd_i = F\left(t_{i+1} + \alpha\Delta t\right). \tag{12.4}$$

The standard Newmark finite difference formulas are used as approximations for d_{i+1} and v_{i+1},

$$\begin{aligned} d_{i+1} &= \tilde{d}_{i+1} + \beta\Delta t^2 a_{i+1}, && (12.5)\\ v_{i+1} &= \tilde{v}_{i+1} + \gamma\Delta t a_{i+1}, && (12.6) \end{aligned}$$

where the predictor variables, $\tilde{d}_{i+1}$ and $\tilde{v}_{i+1}$ are defined as

$$\begin{aligned} \tilde{d}_{i+1} &= d_i + \Delta t v_i + \Delta t^2 \left(\frac{1}{2} - \beta\right) a_i, && (12.7)\\ \tilde{v}_{i+1} &= v_i + (1-\gamma)\Delta t a_i. && (12.8) \end{aligned}$$

We can form an implicit update equation with d_{i+1} as the only unknown by rearranging equation 12.5 and substituting the result along with equation 12.6 into the equation of motion (equation 12.4). If we denote the left hand coefficient matrix as K' and the right hand contributions not due to the force vector as F' then the update equation is

$$K'd_{i+1} = F'_{i+1} + F\left(t_{i+1} + \alpha\Delta t\right). \tag{12.9}$$

where

$$K' = M + \gamma\Delta t C + (1+\alpha)\beta\Delta t^2 K, \tag{12.10}$$

and

$$F'_{i+1} = \left[M + \gamma\Delta t C\right]\tilde{d}_{i+1} - (1+\alpha)\beta\Delta t^2 C\tilde{v}_{i+1} + \alpha\beta\Delta t^2\left[Cv_i + Kd_i\right]. \tag{12.11}$$

12.4.8 Time integrating in transient thermal analysis

As in transient structural analysis, the time-stepping in a transient thermal analysis problem is done using an approach similar to Newmark integration. In this case, however, there is only one integration parameter, α, because of the simpler nature of the governing ODE in the thermal analysis case. Using a generalized trapezoidal rule we express the temperature vector update equation as

$$T_{i+1} = T_i + \Delta t \left[(1-\alpha)\dot{T}_i + \alpha \dot{T}_{i+1} \right]. \tag{12.12}$$

Through some algebraic manipulations of the governing equation at time steps i and $i+1$ and subsitution of equation 12.12 to eliminate time derivative terms, we can write the implicit update equation for T as

$$K' T_{i+1} = [M - \Delta t (1-\alpha) K] T_i + (1-\alpha)\Delta t F_i + \alpha \Delta t F_{i+1}, \tag{12.13}$$

where

$$K' = M + \alpha \Delta t K, \tag{12.14}$$

and everything on the right-hand side is known.

Logan [13] gives the following summary of the methods that result from various choices of α: $\alpha = 0$, simple forward difference scheme which is only conditionally stable; $\alpha = \frac{1}{2}$ Crank-Nicolson or trapezoidal rule which is unconditionally stable; $\alpha = \frac{2}{3}$, Galerkin which is also unconditionally stable; $\alpha = 1$, backward difference which is unconditionally stable.

12.4.9 Solving the eigenvalue problem

FElt uses a relatively unsophisticated procedure to solve for the eigenvalues and eigenvectors in a modal analysis problem. The first step is to actually remove constrained DOF from the global stiffness and mass matrices. This procedure is less efficient than the simple zeroing out of constrained DOF used in transient and static analysis, but it is necessary for the actual numerical algorithm used to solve the eigenvalue problem.

In terms of the reduced mass, M, and stiffness, K, matrices, the generalized eigenvalue problem can be written

$$Kx = \lambda M, \tag{12.15}$$

where λ and x are the eigenvalue and eigenvector in a given mode. We can transform this to standard form,

$$(A - \lambda I)x = 0, \tag{12.16}$$

by forming the Cholesky factorization of the mass matrix, $M = QQ^T$, and realizing that $Q^{-T}MQ^{-1} = I$. With this factorization, we just need to make the substitution $x = Q^{-1}\hat{x}$ and pre-multiply both sides of equation 12.15 by Q^{-T}; the problem then becomes

$$Q^{-T}KQ^{-1}\hat{x} = \lambda I\hat{x} \tag{12.17}$$

which is now in standard form. FElt solves this standard eigenvalue problem by applying the QL method to the tridiagonal form of the transformed coefficient matrix.

Note that this procedure will not work if there are zeros on the diagonal of the mass matrix (the global mass matrix must be positive-definite for the Cholesky factorization to be non-singular). This kind of condition often occurs in problems with lumped mass formulations of the element mass matrices.

Chapter 13

Adding Elements to FElt

13.1 How to get started

Adding additional element types to FElt is meant to be fairly straightforward. Coding an element definition is always an excellent way to really understand the ins and outs of finite element analysis.

A good starting point is probably to look at the source code for the currently available elements, not so much because they are mathematical / algorithmic wonders, but simply because they provide examples of how the analyst has to interface the mathematical code with the system code – elements take up memory so memory will need to be allocated; the generalized analysis routines will expect that the element routines take certain parameters and return certain values, or perhaps initialize certain arrays. Besides simply defining an element stiffness matrix, it is the responsibility of the element routines to calculate equivalent nodal forces due to distributed loads and to calculate their own stresses once the general routines have computed the nodal displacements. If you want to be able to do transient analysis, the same element set-up routine that initializes the element stiffness matrix must also build the element mass matrix.

If you're unfamiliar with the C programming language, you should probably start there. You might be able to get away with just hacking something together based on existing elements, but efficiency and elegance will probably suffer. The material in [12] is considered by many to be the definitive word on C; you may want to start there. Other good books on C undoubtedly abound. Another simple starting point for C novices would be section 12.2 on understanding the data structures used in FElt.

13.2 Necessary definitions and functions

The general routines will expect three things to be defined. Aside from these three things the element routines can use whatever local functions and methods that they want. There must be a definition struct defined for the element type. This structure contains information that everything else needs to know about elements of a given type: symbolic name, shape, number of nodes, number of nodes which define the shape, number of degrees of freedom per node, the degrees of freedom which this element affects, and what functions to use for element setup and stress calculations. Secondly, there must be a routine to create and define the element stiffness matrix for elements of this type. The setup function should also calculate and fill-in the equivalent nodal forces if there is a distributed load on an element and create the element mass matrix if necessary for the current analysis. Lastly, there has to be a routine which calculates the stress or internal loads on an element. If no stress information will be computed there must still be a routine which tells the global routines that no stresses will be computed.

13.2.1 The definition structure

From our previous discussion of the structures in FElt you might remember that each element had a pointer to a definition structure which looked like

```
typedef struct definition {
    char    *name;            /* element name                       */
    int    (*setup) ( );      /* initialization function            */
    int    (*stress) ( );     /* stress computation function        */
    Shape    shape;           /* element dimensional shape          */
    unsigned numnodes;        /* number of nodes in element         */
    unsigned shapenodes;      /* number of nodes which define shape */
    unsigned numstresses;     /* number of computed stress values   */
    unsigned numdofs;         /* number of degrees of freedom       */
    unsigned dofs [7];        /* degrees of freedom                 */
    unsigned retainK;         /* retain element K after assemblage  */
} *Definition;
```

For an arbitrary element, foo, the definition structure might be filled out to look something like this:

```
struct definition fooDefinition = {
    "foo", fooEltSetup, fooEltStress,
    Linear, 2, 2, 3, {0, 1, 2, 3, 0, 0 , 0}, 0
};
```

This defines an element for which the symblic name will be `foo`. It is a linear element with two total nodes, two nodes which define its shape and three DOF per node. The general FElt routines will call the functions `fooEltSetup` and `fooEltStress` for element setup and stress calculations. The DOF array always starts with a zero in the 0th position. The rest of the array tells the general FElt routines that foo elements affect global DOF 1, 2, and 3 (`Tx`, `Ty`, and `Tz`) and that `Tx` is its first DOF, `Ty` its second and `Tz` its third. The last member of the structure is the flag to retain or destroy `element -> K` after it has been assembled. In this case we don't need it around so we set `element -> retainK` to zero (false). If you don't quite see how this all works yet, it may help you to know that foo elements have the same definition as truss elements in the actual FElt library.

A slightly more complicated example is the definition structure for a beam.

```
struct definition beamDefinition = {
    "beam", beamEltSetup, beamEltStress,
    Linear, 2, 2, 3, {0, 1, 2, 0, 0, 0, 3}, 1
};
```

The difference between this definition and that for the foo element is that while beams affect three DOF, the third DOF is `Rz` (position six in the DOF array). Also, we'll want to keep the element stiffness matrix around after we assemble it into the global stiffness matrix (beams use it for internal force calculations), so we set `element -> retainK` to 1 (true).

For the general four to nine node isoparametric element (for plane stress), the definition structure is:

```
struct definition iso2d_PlaneStressDefinition = {
    "iso2d_PlaneStress", iso2d_PlaneStressEltSetup, iso2d_PlaneStressEltStress,
    Planar, 9, 4, 2, {0, 1, 2, 0, 0, 0, 0}, 0
};
```

Here, the basic type is planar rather than linear, the total number of nodes is nine, the number of nodes which define the element shape is four (i.e., the first four nodes completely define the geometric shape of the element) and the number of DOF per node is two. Two-dimensional isoparametric elements only affect translations in the x and y directions (positions one and two in the DOFs array). There is no need to retain the element stiffness matrix after assembling so `element -> retainK` is set to 0 (false).

The definition structure is the only tricky part of setting up a new element. Once FElt knows how an element interacts and "looks" all that remains is for you to tell it how to define the element stiffness and mass matrices and how to compute element stresses (or internal forces) given nodal displacements.

13.2.2 Inside the element setup functions

For our mythical foo element the setup routine would look like this.

```
int fooEltSetup (element, mass_mode)
    Element    element;
    char       mass_mode;
{
}
```

The name of the element setup function must match the function name given in the element definition structure. The setup function must return an integer (the number of errors encountered in performing their respective function, zero on success) and must take a single element and a flag indicating the mass matrix to compute as input. If this is a static problem that flag will be zero. If transient analysis is being performed then the flag will either be 'c' or 'l' depending on whether the setup routine should calculate a consistent or lumped mass matrix. If no element mass matrix is required then the mass_mode will be zero.

Given the element, you have access to the material property, distributed loads, and nodes assigned to that element. The `fooEltSetup` routine's primary responsibility is to allocate and fill out the element stiffness matrix (and mass matrix if necessary) for this element. However, if there are distributed loads on this element, then this routine must also compute the affects of those loads on the element's nodes' equivalent forces.

13.2.3 Inside the element stress function

Like the `fooEltSetup` routine, the `fooEltStress` function must also be defined in a standard way:

```
int fooEltStress (element)
    Element    element;
{
}
```

The fooEltStress routine's only responsibility is to fill a Stress structure for that element. This will likely involve computations with the nodal displacements (accessible via `element -> node[j] -> dx[k]`) and element geometry. If necessary, element stresses or internal forces can be adjusted for equivalent nodal loads.

There is no pre-allocated space for element stresses because the number of stresses and the number of magnitudes for each stress structure vary so greatly from element to element. For this reason, you as the element writer are responsible for allocating an array of stress structures (basically you will need one structure for each point within the element where you will compute stresses; you should also set `element -> ninteg` to this number) and for the stress magnitude array within each of these structures. For example for a three-dimensional beam element, six stress components are computed at both ends (the six possible internal forces). In this case then, we need to allocate space for two stress structures, set `element -> ninteg` to two, and allocate space for six magnitudes in each stress structure.

13.3 The FElt matrix and memory allocation routines

When looking through any of the FElt code, you will notice that we make use of several convenience functions and macros. The most important of these deal with two additional variable types that FElt defines. Besides the general finite element type data structures, FElt makes available a `Matrix` and a `Vector` type. Because these types are really structures, a reliable way to get at the data is `MatrixData (a) [1][1]` or `VectorData (a) [7]`. These macros simply expand to `a -> data[1][1]` and `a -> data[7][1]`. The macro method is preferrable simply because the actual definition of the types may change as FElt develops and the macros will make any such changes transparent.

The following list illustrates how each of the matrix routines can be used. In general, `a`, `b`, and `c` represent variables of type `Matrix` or `Vector`, `width` and `height` define dimensions and status is an integer error code. Note that the actual operational functions (transpose, multiply, add, etc. as opposed to create, delete) can take either matrices or vectors interchangeably. Appropriate dimensions must always match of course. All matrices that go into the matrix routines (both source and destination matrices) must be previously created by one of the matrix creation routines. The operational matrix routines (as opposed to create and destroy functions) all return integer status codes with zero indicating a successful operation; non-zero return values are defined in `include/status.h`. Finally, `Vector` and `Matrix` types are unit offset. The create and destroy routines should make this transparent to the developer.

`a = CreateMatrix (width,height)`
: Creates matrix of the given width and height. This function allocates space and initializes fields within the matrix structure.

`a = CreateVector (height)`
: Creates a vector of length *height*.

`DestroyMatrix (a)`
: This will free the memory associated with a previously created matrix. Once destroyed a matrix variable should not be used again until it is recreated.

`DestroyVector (a)`
: This will free the memory associated with a previously created vector.

`status = MultiplyMatrices (a,b,c)`
: Multiplies matrices `b` and `c` and stores the result in `a`. This is one of the few functions in which your destination matrix `a` cannot be one of your source matrices, `b` or `c`.

`status = TransposeMatrix (a,b)`
: Transposes matrix `b` and puts the result into `a`. Source and destination matrices cannot be the same.

`status = MirrorMatrix (a,b)`
: This function is useful for filling in symmetric matrices. Given a symmetric matrix `b` with only the diagonal and above diagonal terms filled in, this function will complete all entries below the diagonal. The result, `a` will be the complete symmetric matrix.

`status = ZeroMatrix (a)`
: Fills all entries in matrix `a` with zeros.

`status = ScaleMatrix (a,b,x,y)`
: Given a matrix `b`, multiplies all terms by the scalar factor `x` and then adds `y`. The result is stored in `a`.

`status = AddMatrices (a,b,c)`
: Adds matrices `b` and `c` and stores the result in `a`.

`status = SubtractMatrices (a,b,c)`
: Subtracts matrix `c` from matrix `b` and stores the result in `a`.

In addition to the matrix functions, we often use several convenient macros to make memory allocation for non-matrix types a little easier. Examples of these macros are

`ptr = Allocate(type,size)`
: This is equivalent to `ptr = (type *) malloc (sizeof(type)*size)`, i.e., this creates space for `ptr` to hold `size` items each of type `type`.

`UnitOffset (ptr)`
: The allocation macro creates a matrix or vector that is zero offset. Since most FElt arrays are one offset, we use this macro to make the change.

`ZeroOffset (ptr)`
: If `ptr` has been previously unit offset, this macro will reset it zero offset. You must use this macro before de-allocating a pointer that has previously been unit offset.

`Deallocate(ptr)`
: This frees any memory associated with `ptr`.

13.4 Element library convenience functions

The file `lib/Elements/misc.c` contains several functions which are useful across a wide variety of element types. If you want to use these functions you should `#include "misc.h"` in your source file. These functions are

`D = PlaneStressD (element)`
: Returns a constitutive matrix suitable for use in plane stress analysis.

`D = PlaneStrainD (element)`
: Returns a constitutive matrix suitable for use in plane strain analysis.

`l = ElementLength (element, n)`
: Given a dimension, `n`, this function returns the length of an element in `n` dimensions.

`GaussPoints (n, points, weights)`
: Given the number of Gaussian integration points, `n` and the addresses of two pointers to double, `weights` and `points`, this will fill these two arrays with the appropriate values for use in Gaussian quadrature. Currently, this

function only knows the values for 1, 2, and 3 point Gaussian quadrature.

`ResolveHingeConditions (element)`

If you want your element to deal with the possibility of hinged DOFs then you can use this routine to modify the element stiffness matrix appropriately. Generally, you would call this routine only after completely defining `element -> K`. The given element's nodes are checked for hinged DOF; if a DOF is hinged the the coefficients of the element stiffness matrix are adjusted according to the following procedure: given a hinged DOF, dof, then for all entries in k (the element stiffness matrix) not associated with dof (all entries not in row or column dof),

$$k(i,j) = k(i,j) - \frac{k(dof,j)}{k(dof,dof)} k(i,dof). \tag{13.1}$$

The inherent problem in this method of dealing with hinged conditions is that we cannot calculate any displacements associated with the hinged DOF. In general, these displacements will not be zero.

`SetupStressMemory (element)`

After settting `element -> ninteg` to determine the number of points in your element for which you want to calculate stresses, you can call this routine to allocate all of the necessary memory for stress structures and the arrays of stress values in those structures.

`SetEquivalentForceMemory (element)`

This routine will allocate space for the `eq_force[]` array on an element's nodes if such an allocation has not already been done from a previous element.

`MultiplyAtBA (C, A, B)`

Given matrices A, B and a pre-allocated destination matrix C, this routine will form the matrix product of A(transpose)*B*A. This function is particularly useful for doing the multiplication TtKT without any temporary storage and without ever explicitly forming the transpose.

Finally, in order to insure a consistent error protocol across the different interfaces to FElt, there are a few error routines which your routines should call when they encounter trouble (either of the recoverable or irrecoverable variety).

`error (error_message, var1, var2, ...)`
: This is the general routine for reporting a non-fatal (recoverable) error. The calling syntax is exactly like that for `printf`. There is a format string (`error_message`) and a variable length list of variables which are substituted into the message.

`Fatal (error_message, var1, var2, ...)`
: This is the routine for non-recoverable errors. Generally we use this routine when memory allocation fails. After displaying the message (which again should look like the syntax for `printf`) the program will automatically exit.

`AllocationError (element, string)`
: This is a simple little convenience routine for fatal errors during memory allocation in the element routines. The message displayed will be `allocation error computing element` ***`n string`*** where ***`n`*** is the element number and ***`string`*** is some short descriptor of the program location (i.e., `stiffness matrix`).

13.5 Putting it all together

Once the routines have been written, you'll need to update the Makefile for the element library and the initialization routine that tells FElt applications what kinds of elements are available by default. Add the filename of your new element to the `OBJS=` line in `lib/Elements/Makefile`. You can type `make` in that directory to make sure the code compiles. Once you think you've got all the bugs worked out and the new element library is made, you need to add two lines to the initialization procedures in `lib/Felt/initialize.c`. To the list of `extern struct` declarations in that file add a line that looks like

```
extern struct definition fooDefinition;
```

and to the list of `AddDefinition` function calls in the function `add_all_definitions` add a line that looks like

```
AddDefinition (&fooDefinition);
```

(replace `fooDefinition` with whatever variable name you gave to the definition structure for your element of course). Once those changes are made just do a make in `lib/Felt` and then a make in each application directory (e.g., `src/Velvet`, `src/Felt`) to relink the applications with the modified libraries. Your new element should now be ready for use. Writing up a simple test file that uses the new element and running this through the command line application *felt* is usually the easiest way to check numerical results.

13.6 A detailed example

If everything above is still a bit unclear, fear not; it's really not as bad as it looks. Let's walk through a step-by-step description of how the Timoshenko beam element was added to the FElt library. This is a very simple element. We only consider bending and shear deformation (no axial stiffness is taken into account) and we don't allow distributed loads applied in the global DOF. For some basic mathematical details for this element refer to section 4.2.3. More details are available in [4, 10, 15].

In general, all the code for a given element type will be contained in one source file. Hopefully the routines in this file will be completely specific to that element type. In a perfect world, common functionality would be available as convenience functions. This is the basic model that we will follow here. All of the following code is taken from the file `timoshenko.c` in the directory `lib/Elements`.

We start by including the necessary header files, prototyping our local functions (we make them `static` to insure that they will be local to this file), and setting up the `definition` structure.

```
/************************************************************************
 * File:        timoshenko.c                                            *
 *                                                                      *
 * Description: This file contains the definition structure and         *
 *              set-up functions for a Timoshenko beam element.         *
 *                                                                      *
 * History:     V1.4 by Jason Gobat and Darren C. Atkinson              *
 ************************************************************************/

# include <math.h>
# include "allocate.h"
# include "element.h"
# include "misc.h"

        /*
         * Here's the definition structure.  This is a very simple
         * implementation, 2 nodes, possible effect on 3 global DOF
         * per node.  We need to prototype the setup and stress functions
         * first thing so we can use them in the definition declaration.
         */

int timoshenkoEltSetup ( );
int timoshenkoEltStress ( );

struct definition timoshenkoDefinition = {
    "timoshenko",              /* the symbolic name used in input files   */
    timoshenkoEltSetup,        /* the element setup function              */
    timoshenkoEltStress,       /* the element stress function             */
    Linear,  /* The shape of this element    */
    2,  /* 2 nodes per element    */
    2,  /* 2 nodes define the shape (it's a line!) */
    2,  /* 2 magnitudes in each stress structure   */
    3,  /* 3 global DOF / node    */
   {0, 1, 2, 6, 0, 0, 0},      /* DOF 1 is Tx, DOF 2 is Ty DOF 3 is Rz .. */
    1 /* retain stiffness after assembling    */
};

        /*
         * We'll declare these three functions as static because other
         * people might use these same names for their element.  The
         * static declaration makes them private to this file.
         * There is nothing magical about them.  They could be called
         * anything, your element may not use any local functions,
         * etc., etc.  It's all a matter of preference and style.
         */

static Matrix LocalK ( );
static Matrix TransformMatrix ( );
static Matrix LumpedMassMatrix ( );
static Matrix ConsistentMassMatrix ( );
static int    EquivNodalForces ( );
```

The next thing we want to do is define the element setup routine which will define `element -> K` for every Timoshenko beam element. All this function really does is set some memory, call a few functions and do a few matrix multiplications. The brute force filling in of matrices occurs in our own private routines.

```
        /*
         * The element setup function (the one that the general
         * routines actually call to define element -> K for
         * Timoshenko beams).  We'll break it up a little more
         * for our own internal purposes and call some functions
         * of our own to actually fill out the guts of the thing.
         */

int timoshenkoEltSetup (element, mass_mode)
    Element        element;
    char           mass_mode;
{
    int            count;       /* a count of errors encountered           */
    Matrix         T;           /* transform matrix                        */
    Matrix         khat;        /* local coordinate stiffness matrix       */
    Matrix         mhat;        /* local coordinate mass matrix            */

        /*
         * Since we're nice and we like to do as much error checking
         * as possible, we'll also check to make sure that all necessary
         * material properties are set for this element
         */

    count = 0;
    if (element -> material -> E == 0.0) {
        error ("timoshenko element %d has 0.0 for Young's modulus (E)",
               element -> number);
        count++;
    }
    if (element -> material -> Ix == 0.0) {
        error ("timoshenko element %d has 0.0 for moment of inertia (Ix)",
               element -> number);
        count++;
    }
    if (element -> material -> G == 0.0) {
        error ("timoshenko element %d has 0.0 for bulk modulus (G)",
               element -> number);
        count++;
    }
    if (element -> material -> A == 0.0) {
        error ("timoshenko element %d has 0.0 for bulk modulus (G)",
               element -> number);
        count++;
    }

        /*
         * nu and kappa are somewhat special because we have to have
         * at least one.  If we have nu we’ll use it to estimate
         * element -> kappa according to Cowper’s (1966) approximation.
         * If we have kappa we will of course always use it.  If
         * we have neither, it’s an error
         */

    if (element -> material -> kappa == 0.0) {
        if (element -> material -> nu == 0.0) {
            error ("timoshenko element %d has 0.0 for Poisson’s ratio (nu)",
                   element -> number);
            count++;
        }
        else {
            element -> material -> kappa =
                 10.0*(1.0 + element -> material -> nu)/
                 (12.0 + 11.0*element -> material -> nu);
        }
    }

        /*
         * if we’ve had any errors there is no point in continuing
         */

    if (count)
        return count;

        /*
         * get the local stiffness matrix and the transform matrix.
         * we never allocated any memory for these two because the
         * functions that we are calling will do that.
         */

    khat = LocalK (element);
    if (khat == NullMatrix)
       return 1;

    T = TransformMatrix (element);

        /*
         * We can form the element stiffness matrix now through
         * some simple matrix multiplications.  The special multiply
         * function here just saves us having to allocate
         * some temporary space and actually transposing the transform
         * matrix, it will simply carry out k = T(trans) * K * T
```

```
          */

    element -> K = CreateMatrix (6,6);
    if (element -> K == NullMatrix)
        AllocationError (element, "stiffness matrix");

    MultiplyAtBA (element -> K, T, khat);

        /*
         * Things can get a little tricky here; we'll check if there
         * are any distributed loads - if there are we need to resolve
         * them and modify this element's node's equivalent nodal forces.
         * If not we're home free.  In this case I have
         * relegated all the distributed load handling to a separate
         * little module.
         */

    if (element -> numdistributed > 0) {
        count = EquivNodalForces (element, Tt, NULL, 1);
        if (count);
            return count;
    }

        /*
         * there's also the possibility that some of this element's nodes
         * have a hinged DOF ... that's easy to deal with because we have a
         * convenience routine to do all the checking and modifying for us.
         */

    ResolveHingeConditions (element);

/*
 * check to see if we need to form a mass matrix.  If we
 * need to we call a local function just like we did for
 * the local stiffness matrix (depending on the mass_mode)
 * then use the same multiplication function to transform
 * to global coordinates.
 */

    if (mass_mode) {
        if (mass_mode == 'c')
           mhat = ConsistentMassMatrix (element);
        else if (mass_mode == 'l')
           mhat = LumpedMassMatrix (element);
```

```
            if (mhat == NullMatrix)
               return 1;

            element -> M = CreateMatrix (6,6);
            if (element -> M == NullMatrix)
                AllocationError (element, "mass matrix");

            MultiplyAtBA (element -> M, T, mhat);
        }

            /*
             * we made it here, everything must have worked!
             */

        return 0;
    }
```

Not bad, right? Note our use of `ResolveHingeConditions` in the above because we want these elements to be able to deal with hinged DOF. For many elements, hinges are meaningless and this check would not be necessary.

The next function we need is the element stress routine. Again, this is pretty simple because we do a lot of the work elsewhere in this case. In order to retrieve the four internal forces that we are interested in, all we need to do is transform the element stiffness matrix back to local coordinates, transform the nodal displacements back to local coordinates, multiply and subtract any equivalent nodal forces.

```
            /*
             * The element stress function that actually gets called
             * to fill in the element's stress structures.  I realize
             * that a lot of this seems awfully inefficient ... beam type
             * elements are a bit of an anomaly because they need their
             * stiffness matrix back and a bunch of local<->global transforms.
             */

    int timoshenkoEltStress (element)
        Element         element;
    {
        unsigned        i;                  /* loop index                   */
        int             count;              /* count of errors              */
        static Vector   dlocal = NULL;      /* local nodal displacements    */
```

```
static Vector   d;                  /* global nodal displacements    */
static Vector   f;                  /* actual internal forces        */
Vector          equiv;              /* equivalent nodal forces       */
Matrix          T;                  /* transform matrix              */
static Matrix   khat;               /* local stiffness matrix        */
static Matrix   Tt;                 /* transpose of transform        */
static Matrix   temp;               /* temporary matrix for multiply */

    /*
     * our usual trick to set-up the matrices and vectors that
     * we need memory for, but that are really just local to
     * this function.
     */

if (dlocal == NULL) {
    dlocal = CreateVector (4);
    d = CreateVector (6);
    f = CreateVector (4);
    khat = CreateMatrix (4,4);
    Tt = CreateMatrix (6,4);
    temp = CreateMatrix (4,6);

    if (dlocal == NullMatrix || d == NullMatrix || f == NullMatrix ||
        khat == NullMatrix || Tt == NullMatrix)
        AllocationError (element, "stresses");
}

    /*
     * set the number of points where we will calculate stresses.
     * In this case it's two (one at each end).
     */

element -> ninteg = 2;

    /*
     * Fill out a vector with the element's nodal displacements.
     * These are in global coordinates of course.  We need to
     * do a transformation to get them into local coordinates.
     */

VectorData (d) [1] = element -> node[1] -> dx[Tx];
VectorData (d) [2] = element -> node[1] -> dx[Ty];
VectorData (d) [3] = element -> node[1] -> dx[Rz];
VectorData (d) [4] = element -> node[1] -> dx[Tx];
VectorData (d) [5] = element -> node[2] -> dx[Ty];
VectorData (d) [6] = element -> node[2] -> dx[Rz];

T = TransformMatrix (element);

MultiplyMatrices (dlocal, T, d);

    /*
     * We already have the element stiffness matrix because we
     * set element -> retainK = 1 in the definition structure.  This
     * means that the global stiffness assembly routine didn't
     * trash element -> K after it was done with it and we can
     * use it again.  We will have to transform it back to local
     * coordinates, however.
     */

TransposeMatrix (Tt, T);
MultiplyMatrices (temp, T, element -> K);
MultiplyMatrices (khat, temp, Tt);

    /*
     * we can get the internal force vector through a simple
     * matrix multiplication.
     */

MultiplyMatrices (f, khat, dlocal);

    /*
     * Of course, we may need to modify that for equiv nodal forces
     */

if (element -> numdistributed > 0) {
    count = EquivNodalForces (element, NULL, &equiv, 2);
    if (count)
        return count;

    for (i = 1; i <= 4; i++)
        VectorData (f) [i] -= VectorData (equiv) [i];
}

    /*
     * set-up some memory for the stress structure and for the values
     * in the stress structure.  We'll just use a quicky little
     * convenience routine to do it for us.  It's important to
```

```
         * set element -> ninteg before we call this function.
         */

    SetupStressMemory (element);

        /*
         * establish the location of the stresses and the magnitudes
         * of the stresses at each point.  This particular loop
         * only works because there are two stress points and two
         * stress values at each point.
         */


    for (i = 1; i <= 2; i++) {
        element -> stress[i] -> x = element -> node[i] -> x;
        element -> stress[i] -> y = element -> node[i] -> y;

        element -> stress[1] -> values[i] = VectorData (f)[i];
        element -> stress[2] -> values[i] = VectorData (f)[i+2];
    }

    return 0;
}
```

Now we get into the few routines that are local to this file (i.e., private routines which we would only call when we were defining Timoshenko beam elements). The first of these simply fills in the 4×4 local stiffness matrix. There is no fancy integration here because we know how it all turns out so we save ourselves some computations by plugging straight into the entries in the matrix. The second of these two routines will compute a transformation matrix for these elements.

```
        /*
         * Our own function to define the stiffness matrix in
         * local coordinates.
         */

static Matrix LocalK (element)
    Element       element;
{
    static Matrix k = NULL;     /* the local stiffness matrix          */
    double        L;            /* the element length                  */
    double        phi;          /* bending stiffness / shear stiffness */
```

```
   double       factor;      /* common factor in stiffness matrix   */

       /*
        * We use a trick to make sure we only allocate this memory
        * once and then use it over and over again each time we need to
        * create an element of this kind.
        */

   if (k == NULL) {
       k = CreateMatrix (4,4);

       if (k == NullMatrix)
           AllocationError (element, "local stiffness");
   }

   L = ElementLength (element, 2);
   if (L <= TINY) {
       error ("length of element %d is zero to machine precision",
              element -> number);
       return NullMatrix;
   }

   phi = 12.0/(L*L)*(element -> material -> E*element -> material -> Ix/
                  (element -> material -> kappa*
                   element -> material -> G*element -> material -> A));

       /*
        * We know how the integration works out for the stiffness
        * matrix so we're just going to fill it out an entry at
        * a time.  For some element types this wouldn't be possible and
        * we would do some integrating right here to fill in k.
        * Also, because this is a symmetric matrix we'll just
        * fill in everything above the diagonal and then use MirrorMatrix
        */

  MatrixData (k) [1][1] = 12.0;
  MatrixData (k) [1][2] = 6.0*L;
  MatrixData (k) [1][3] = -12.0;
  MatrixData (k) [1][4] = 6.0*L;
  MatrixData (k) [2][2] = (4.0 + phi)*L*L;
  MatrixData (k) [2][3] = -6.0*L;
  MatrixData (k) [2][4] = (2.0 - phi)*L*L;
  MatrixData (k) [3][3] = 12.0;
  MatrixData (k) [3][4] = -6*L;
```

```
   MatrixData (k) [4][4] = (4.0 + phi)*L*L;

   MirrorMatrix (k,k);

        /*
         * the above numbers aren't quite right, we've got a term out
         * front of the matrix that we need to scale the entire
         * matrix by
         */

   factor = (element -> material -> E*element -> material -> Ix)/
            ((1.0 +phi)*L*L*L);

   ScaleMatrix (k, k, factor, 0.0);

        /*
         * that's all for this part
         */

   return k;
}

        /*
         * a simple little function to compute the transform matrix
         * for a simple 2d beam element with no axial DOF.
         * This should be a convenience routine, but none of the other
         * elements actually use this one because they are more complicated.
         */

static Matrix TransformMatrix (element)
    Element        element;
{
    double         s,c;         /* direction cosines                   */
    static Matrix  T = NULL;    /* transform matrix to return          */
    double         L;           /* element length                      */

        /*
         * no surprise here, we only want to allocate memory for this
         * guy once!
         */

    if (T == NULL) {
       T = CreateMatrix (4,6);
```

```
            if (T == NullMatrix)
                AllocationError (element, "transform matrix");
        }

             /*
              * This is a pretty sparse matrix so we'll just zero it out
              * then fill in the few relevant entries.
              */

        ZeroMatrix (T);

        L = ElementLength (element, 2);
        c = (element -> node[2] -> x - element -> node[1] -> x) / L;
        s = (element -> node[2] -> y - element -> node[1] -> y) / L;

        MatrixData (T) [1][1] = -s;
        MatrixData (T) [1][2] = c;
        MatrixData (T) [2][3] = 1.0;
        MatrixData (T) [3][4] = -s;
        MatrixData (T) [3][5] = c;
        MatrixData (T) [4][6] = 1.0;

        return T;
    }
```

Now we need two local functions to create the two different kinds of mass matrices. They look an awful lot like the local stiffness matrix because once again we don't actually need to do any numerical integration; we know how it all turns out so we just have to fill in some matrix entries.

```
    /*
     * much like the local K function above all we do here is fill in
     * the mass matrix - this function fills it out for consistent
     * mass, the following function is used if the user wanted a lumped
     * mass
     */

    static Matrix ConsistentMassMatrix (element)
        Element    element;
    {
        static Matrix m = NULL;    /* the local stiffness matrix           */
        double        L;           /* the element length                   */
        double        phi;         /* bending stiffness / shear stiffness  */
    double        phi2;        /* phi squared                           */
    double        const1;      /* constant term for rotational mass    */
    double        const2;      /* constant term for translational mass */

    if (m == NULL) {
        m = CreateMatrix (4, 4);

        if (m == NullMatrix)
            AllocationError (element, "mass matrix");
    }

/*
 * the constants that we'll need, including the constant terms
 * in front of the rotational (first terms) and translational
 * (second terms) portions of the matrix.
 */

    L = ElementLength (element, 2);
    phi = 12.0/(L*L)*(element -> material -> E*element -> material -> Ix/
                      (element -> material -> kappa*
                       element -> material -> G*
                       element -> material -> A));
    phi2 = phi*phi;
    const1 = element -> material -> rho *
             element -> material -> Ix /
             (30.0*(1.0 + phi)*(1.0 + phi)*L);
    const2 = element -> material -> rho *
             element -> material -> A * L /
             (210.0*(1.0 + phi)*(1.0 + phi));

/*
 * fill out the top half of the mass matrix (no need to
 * explicitly integrate of course)
 */

   MatrixData (m) [1][1] = 36.0*const1 +
                           (70.0*phi2 + 147.0*phi + 78)*const2;
   MatrixData (m) [1][2] = -L*(15.0*phi - 3.0)*const1 +
                           (35.0*phi2 + 77.0*phi + 44.0)*L/4.0*const2;
   MatrixData (m) [1][3] = -36.0*const1 +
                           (35.0*phi2 + 63.0*phi + 27.0)*const2;
   MatrixData (m) [1][4] = -L*(15.0*phi - 3.0)*const1 -
                           (35.0*phi2 + 63.0*phi + 26.0)*L/4.0*const2;
   MatrixData (m) [2][2] = (10.0*phi2 + 5.0*phi + 4)*L*L*const1 +
                        (7.0*phi2 + 14.0*phi + 8.0)*L*L/4.0*const2;
   MatrixData (m) [2][3] = -MatrixData (m) [1][4];
   MatrixData (m) [2][4] = (5.0*phi2 - 5.0*phi - 1.0)*L*L*const1 -
                        (7.0*phi2 + 14.0*phi + 6.0)*L*L/4.0*const2;
   MatrixData (m) [3][3] = 36.0*const1 + (70.0*phi2 +
                        147.0*phi + 78.0)*const2;
   MatrixData (m) [3][4] = -MatrixData (m) [1][2];
   MatrixData (m) [4][4] = (10.0*phi2 + 5.0*phi + 4.0)*L*L*const1 +
                        (7.0*phi2 + 14.0*phi + 8.0)*L*L/4.0*const2;

/*
 * complete it by mirroring
 */

   MirrorMatrix (m, m);

/*
 * and we're done;
 */

   return m;
}

static Matrix LumpedMassMatrix (element)
    Element    element;
{
    static Matrix m = NULL;       /* the local stiffness matrix  */
    double        factor;         /* constant term               */
    double        I_factor;       /* inertia term for rotation   */
    double        L;              /* element length              */

    if (m == NULL) {
        m = CreateMatrix (4, 4);

        if (m == NullMatrix)
            AllocationError (element, "mass matrix");

        ZeroMatrix (m);
    }

    L = ElementLength (element, 2);
    factor = L * element -> material -> rho * element -> material -> A / 2;
    I_factor = factor*L*L/12;
    MatrixData (m) [1][1] = factor;
    MatrixData (m) [2][2] = I_factor;
    MatrixData (m) [3][3] = factor;
    MatrixData (m) [4][4] = I_factor;

    return m;
}
```

The last local function is the routine to compute equivalent nodal loads based on distributed loads which the user has applied to this element. These routines can get kind of tricky because there is a lot of error checking and lots of different possibilities that we have to deal with (directions, node ordering, etc.). Furthermore, this one needs to take care of two different ways in which it might be called. The stiffness function calls this routine to set the equivalent forces on its nodes. We get these forces by transforming our equivalent force vector to global coordinates and then adding the terms of this global vector onto the `eq_force[]` arrays of the element's nodes. The stress function calls this routine because it needs to adjust the internal force vector to account for equivalent nodal forces. It only needs the basic vector in local coordinates.

```
             /*
              * We need to compute the equivalent nodal load
              * vector here. Just for convenience we are going to call
              * this function in two different ways (mode=1 and mode=2).
              * The first way is for the element stiffness function
              * which just wants to get the forces applied to the
              * element's nodes.  The second is for the stress routine
              * which actually needs the equiv force vector in local coordinates.
              * There are lots of ways to handle all these cases;
              * see the Bernoulli beam elements for example. In mode 1,
              * eq_stress can be NULL, in mode 2, Tt can be NULL.
              */

static int EquivNodalForces (element, Tt, eq_stress, mode)
    Element        element;
    Matrix         Tt;                 /* passing it in saves a few FLOPs */
    Vector         *eq_stress;         /* vector pointer to set in mode 2 */
    int            mode;               /* mode of operation               */
{
    static Vector  equiv = NULL;       /* the equiv vector in local coord */
    static Vector  eq_global;          /* equiv in global coordinates     */
    double         wa, wb;             /* values of load at nodes         */
    double         L;                  /* the element length              */
```

```
unsigned       i,j;               /* some loop conuters              */
double         factor;            /* constant factor for sloped load */
double         phi;               /* bending / shear stiffness       */
int            count;             /* error count                     */

if (equiv == NULL) {
    equiv     = CreateVector (4);
    eq_global = CreateVector (6);

    if (equiv == NullMatrix || eq_global == NullMatrix)
        AllocationError (element, "equivalent nodal loads");
}

ZeroMatrix (equiv);

count = 0;

    /*
     * Again, we want to do as much error checking and descriptive
     * error reporting as possible.  Seem like overkill?  It probably
     * is, but it’s not hurting anybody either :-)
     */

if (element -> numdistributed > 2) {
   error ("Timoshenko beam element %d has more than 2 distributed loads",
          element -> number);
   count++;
}

L = ElementLength (element, 2);
if (L <= TINY) {
    error ("length of element %d is zero to machine precision",
           element -> number);
    count++;
}

for (i = 1; i <= element -> numdistributed; i++) {
    if (element -> distributed[i] -> nvalues != 2) {
        error ("Timoshenko beam element %d must have 2 values for load",
               element -> number);
        count++;
    }

    /*
```

```
         * We only want to deal with loads in the perpendicular (LocalY)
         * direction ... this is a very simple instantiation of this
         * element after all.
         */

        if (element -> distributed[i] -> direction != LocalY &&
            element -> distributed[i] -> direction != Perpendicular) {

            error ("invalid direction for element %d distributed load",
                   element -> number);
            count++;
        }

        /*
         * make sure that the user isn’t try to apply part of this
         * load to a non-existent node (some local node other than
         * number 1 or 2)
         */

        for (j = 1 ;j <= element -> distributed[i] -> nvalues; j++) {
            if (element -> distributed[i] -> value[j].node < 1 ||
                element -> distributed[i] -> value[j].node > 2) {

                error ("invalid node numbering for elt %d distrib load %s",
                       element -> number,element -> distributed[i] -> name);
                count++;
            }
        }

        if (element -> distributed[i] -> value[1].node ==
            element -> distributed[i] -> value[2].node) {

            error ("incorrect node numbering for elt %d distributed load %s",
                   element -> number, element -> distributed[i] -> name);
            count++;
        }
    }

        /*
         * Have we had any errors? If so bail out.
         */

    if (count)
        return count;
```

```
phi = 12.0/(L*L)*(element -> material -> E*element -> material -> Ix/
                  (element -> material -> kappa*
                   element -> material -> G*element -> material -> A));

    /*
     * loop over all of the applied distributed loads, superposing
     * the effects of each
     */

for (i = 1 ; i <= element -> numdistributed ; i++) {

    /*
     * First we have to sort out what order the load values
     * were supplied in.  We need to get it so that wa is
     * the value on element node 1 and wb is the value on
     * element node 2.
     */

    if (element -> distributed[i] -> value[1].node == 1) {
        wa = element -> distributed[i] -> value[1].magnitude;
        wb = element -> distributed[i] -> value[2].magnitude;
    }
    else if (element -> distributed[i] -> value[1].node == 2) {
        wb = element -> distributed[i] -> value[1].magnitude;
        wa = element -> distributed[i] -> value[2].magnitude;
    }

    /*
     * Again, since we know how the integration turns out, we'll
     * just go head and plug straight into the entries in the equiv
     * vector.  The order of entries in equiv is Fy1,Mz1,Fy2,Mz2.
     * There are three cases we need to deal with.  The first is
     * a uniform load.  The second two are sloped loads which we'll
     * treat as the superposition of the uniform case and a case
     * in which the load can be treated as q(x) = q0*(1 - x/L)
     * (i.e., a load which goes from q0 to 0)
     */

    if (wa == wb) {                     /* uniform distributed load   */
        VectorData (equiv)[1] += -wa*L/2.0;
        VectorData (equiv)[3] += -wa*L/2.0;
        VectorData (equiv)[2] += -wa*L*L/12.0;
        VectorData (equiv)[4] += wa*L*L/12.0;
        }
        else if (fabs(wa) > fabs(wb)) {     /* load sloping node 1-node 2 */
            factor = (wa - wb)*L/120.0/(1.0 + phi);
            VectorData (equiv)[1] += -wb*L/2.0 - factor*(42.0 + 40.0*phi);
            VectorData (equiv)[3] += -wb*L/2.0 - factor*(18.0 + 20.0*phi);
            VectorData (equiv)[2] += -wb*L*L/12.0 - factor*(6.0 + 5.0*phi)*L;
            VectorData (equiv)[4] += wb*L*L/12.0 + factor*(4.0 + 5.0*phi)*L;
        }
        else if (fabs (wa) < fabs (wb)) {   /* load sloping node 2-node 1 */
            factor = (wb - wa)*L/120.0/(1.0 + phi);
            VectorData (equiv)[1] += -wa*L/2.0 - factor*(18.0 + 20.0*phi);
            VectorData (equiv)[3] += -wa*L/2.0 - factor*(42.0 + 40.0*phi);
            VectorData (equiv)[2] += -wa*L*L/12.0 - factor*(4.0 + 5.0*phi)*L;
            VectorData (equiv)[4] += wa*L*L/12.0 + factor*(6.0 + 5.0*phi)*L;
        }
    }

        /*
         * if this is mode 2, we're done, just hand the equiv vector
         * back by setting eq_stress.
         */

    if (mode == 2) {
        *eq_stress = equiv;
        return 0;
    }

        /*
         * We have the load vector in local coordinates now.
         * All of this is taken care of by a convenience routine.
         * What it is doing is checking if the eq_force array has been
         * allocated for this element's nodes.  If it hasn't it will set
         * it up.  If it has it will do nothing and simply return
         * to us.  It has to allocate space for six doubles (even
         * though we will only ever use two entries for Timoshenko
         * elements) because other element types may try to insert
         * something into this array in different locations. Also,
         * remember that we will access it as a standard array,
         * it's not a Vector or Matrix type.
         */

    SetEquivalentForceMemory (element);

        /*
        * The equiv vector has four things in it.  We need to transform
        * these to global coordinate and then add them
        * incrementally into the eq_force [] array on the nodes
        * because some other element may have also already added
        * something onto this node.  Note the use of Tx, Ty and Rz
        * to access the eq_force array.  These are just enumerated
        * so that they expand to 2 and 6 ... no real magic there, it
        * is just little more intuitive to look at.
        */

    MultiplyMatrices (eq_global, Tt, equiv);
    element -> node[1] -> eq_force[Tx] += VectorData (eq_global) [1];
    element -> node[1] -> eq_force[Ty] += VectorData (eq_global) [2];
    element -> node[1] -> eq_force[Rz] += VectorData (eq_global) [3];
    element -> node[2] -> eq_force[Tx] += VectorData (eq_global) [4];
    element -> node[2] -> eq_force[Ty] += VectorData (eq_global) [5];
    element -> node[2] -> eq_force[Rz] += VectorData (eq_global) [6];

    return 0;
}
```

Now that we have written the source file `timoshenko.c` in the directory `lib/Elements` we need to change the `OBJS=` line in `Makefile` in this directory to include our new file. That line should now read like:

```
OBJS    = beam.o beam3d.o cst.o truss.o iso_2d.o iso_quad.o misc.o \
          timoshenko.o
```

All we have to do now is type `make` at the shell prompt and of course because we do such good work there are no errors. On the off chance that you're not so lucky and you do have a few errors, all you have to do is fix up your source file and keep trying `make` until they go away. Once we have the new element library made, we need to update the initialization procedure which will define the default set of elements for a FElt application. We do this by editing `lib/Felt/initialize.c` and adding `extern struct definition` and `AddDefinition` lines just like the lines already in that file. With that change made do a make in `lib/Felt`.

Now that all of the necessary libraries have been built to be aware of the new element we need to re-link an application with the updated libraries. *felt* is the easiest because it is so simple. Go to the `bin/Felt` directory and do a make. Then, create a simple test file that uses your new element and run it through *felt*. If the numbers don't come out right you can modify the source in `lib/Elements`, do a `make` in that directory and a make in `bin/Felt` to re-link your changes. Repeat this process until everything comes out to your satisfaction. Eventually you should do a make in the directory for each separate application so that your new element is accessible from all of the application in the FElt system.

Appendix A

Installing and Administering FElt

A.1 Building the FElt system from source

The FElt package is intended to be easily portable. It should build on most reasonable Un*x systems without any modifications. To start the build `cd` to the toplevel directory of the FElt source tree. From there do a `./configure`. This will try to automatically determine where to find relevant include files and libraries on your system and create a new `etc/Makefile.conf` file. If you feel comfortable with it you can go in and tweak the `Makefile.conf` file by hand if something does not go right. After configuring, do a `make clean` followed by a `make all`. To install the package after it has been successfully built do a `make install`.

The entire package has been compiled and tested under SunOS (4.x and 5.x), and Linux (the OS under which it was developed). This version or earlier versions compiled under HP-UX 8.0 and 9.0 SystemV386 (R3.2.2), and various SGI, DEC, and IBM workstations (using Irix, OSF/1 and Ultrix, and AIX) with little or no problem. The files *felt.exe* and *feltvu.exe* (a graphing application) are available as pre-compiled executables for the DOS environment.

The most recent version of the source code for FElt should always be available via anonymous ftp at felt.sourceforge.net in the directory `/pub/FElt`. We also try to make pre-compiled binaries available for a wide variety of platforms and operating systems. If you want to be made aware of each revision of the FElt system then we strongly encourage you to subscribe to the FElt mailing list. To subscribe visit the list information page at `http://lists.sourceforce.net/mailman/listinfo/felt-announce`. Only major revisions and changes will be announced broadly via Usenet.

A.2 Translation files

The translation files which map non-English terms to the English terms which FElt expects should be installed in the library directory which you specified during the build. This path will automatically be searched when *cpp* goes looking for include files (which is really what the translation files are). If they are not in some standard place, users need to specify a `-I` flag on the *felt* command line to tell *cpp* where to look for them. The current filenames are in English, you'll obviously want to link them or move them to something reasonable on your system and make users aware of what the appropriate file to include is called.

The following tables list the currently available translations (that we have at least). You can look at the current `german.trn` and `spanish.trn` files if you want to change something or provide a new one. If you do write a new one, please send it to us. This is only a first cut at internationalization, but as we get more translations we will be in a better position to see how we can make it all work a little more fluently. Also, remember that because this is a first cut all you can really do is type in a FElt file by hand using these translations. Anything *velvet* saves or even the file that *felt* writes with the `-debug` command will be in English even if the input file was given in another language. All output is still in English. Lastly, if you use these translations, you have to use them exactly as they are given here. They are not case insensitive, and if there are underscores in the place of spaces, you must use the underscores. Things that are user specifiable to begin with (object names, problem title) can still be whatever you want of course. If a keyword is not listed in a table below, use the original English keyword (the English and non-English were probably equivalent).

For German, use . . .	**Instead of . . .**
problembeschreibung	problem description
titel	title
knoten	nodes
elemente	elements
krafte	force
kraefte	forces
zwangsbedingung	constraint
zwangsbedingungen	constraints
materialeigenschaften	material properties
richtung	direction
senkrecht	perpendicular
GlobalesX	GlobalX
GlobalesY	GlobalY
GlobalesZ	GlobalZ
LokalesX	LocalX
LokalesY	LocalY
LokalesZ	LocalZ
werte	values
verteilte	distributed
lasten	loads
last	load
gelenk	hinge
ende	end

For Spanish, use . . .	**Instead of . . .**
descripcion_del_problema	problem title
titulo	title
nodos	nodes
elementos	elements
fuerza	force
fuerzas	forces
restriccion	constraint
restricciones	constraints
propriedades_del_material	material properties
direccion	direction
paralela	parallel
valores	values
cargas_distribuidas	distributed loads
carga	load
articulacion_plana	hinge
final	end

A.3 Defaults and material databases

Though the defaults files and material databases are not a necessity for *velvet*, they are convenient and it might be desirable to install them in a location where all users have easy access to them, probably the same library directory as the translation files. These files too are really nothing more than include files which can be included as a convenience for the user.

Appendix B

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11. BECAUSE THE PROGRAM IS LICENSED FREE OF CHARGE, THERE IS NO WARRANTY FOR THE PROGRAM, TO THE EXTENT PERMITTED BY APPLICABLE LAW. EXCEPT WHEN OTHERWISE STATED IN WRITING THE COPYRIGHT HOLDERS AND/OR OTHER PARTIES PROVIDE THE PROGRAM “AS IS” WITHOUT WARRANTY OF ANY KIND, EITHER EXPRESSED OR IMPLIED, INCLUDING, BUT NOT LIMITED TO, THE IMPLIED WARRANTIES OF MERCHANTABILITY AND FITNESS FOR A PARTICULAR PURPOSE. THE ENTIRE RISK AS TO THE QUALITY AND PERFORMANCE OF THE PROGRAM IS WITH YOU. SHOULD THE PROGRAM PROVE DEFECTIVE, YOU ASSUME THE COST OF ALL NECESSARY SERVICING, REPAIR OR CORRECTION.

B.3 Appendix: How to Apply These Terms to Your New Programs

If you develop a new program, and you want it to be of the greatest possible use to the public, the best way to achieve this is to make it free software which everyone can redistribute and change under these terms.

To do so, attach the following notices to the program. It is safest to attach them to the start of each source file to most effectively convey the exclusion of warranty; and each file should have at least the “copyright” line and a pointer to where the full notice is found.

Also add information on how to contact you by electronic and paper mail.

If the program is interactive, make it output a short notice like this when it starts in an interactive mode:

```
Gnomovision version 69, Copyright (C) 19yy name of author Gnomovision
comes with ABSOLUTELY NO WARRANTY; for details type 'show w'.  This
is free software, and you are welcome to redistribute it under certain
conditions; type 'show c' for details.
```

The hypothetical commands 'show w' and 'show c' should show the appropriate parts of the General Public License. Of course, the commands you use may be called something other than 'show w' and 'show c'; they could even be mouse-clicks or menu items–whatever suits your program.

You should also get your employer (if you work as a programmer) or your school, if any, to sign a "copyright disclaimer" for the program, if necessary. Here is a sample; alter the names:

> Yoyodyne, Inc., hereby disclaims all copyright interest in the program 'Gnomovision' (which makes passes at compilers) written by James Hacker.
>
> ⟨*signature of Ty Coon*⟩, 1 April 1989 Ty Coon, President of Vice

This General Public License does not permit incorporating your program into proprietary programs. If your program is a subroutine library, you may consider it more useful to permit linking proprietary applications with the library. If this is what you want to do, use the GNU Library General Public License instead of this License.

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