



# Manual

## Mecway Finite Element Analysis

Version 14.0  
2021

## Contents

### *Chapter 1*

Welcome to FEA	7
----------------	---

### *Chapter 2*

Overview of Mecway	9
--------------------	---

- 2.1 Mesh, 9
- 2.2 Analysis, 9
- 2.3 Geometry, 10
- 2.4 Components & Materials, 10
- 2.5 Loads & Constraints, 10
- 2.6 Named Selections, 10
- 2.7 Solution, 10
- 2.8 Configurations, 10
- 2.9 Units, 11

### *Chapter 3*

Viewing and Selecting	12
-----------------------	----

- 3.1 Zoom, Pan, Rotate, 12
- 3.2 Display modes, 13
- 3.3 Selection, 14

### *Chapter 4*

Manual Meshing	16
----------------	----

- 4.1 Creating Tools, 16
- 4.2 Editing Tools, 19

- 4.3 Converting a 2D Mesh into a 3D Mesh, 22
- 4.4 Refinement Tools, 26
- 4.5 Tracing an Image, 27
- 4.6 Symmetry, 28
- 4.7 Mesh Information, 29
- 4.8 Modeling Errors, 29

## *Chapter 5*

### **CAD Models**

**34**

- 5.1 Introduction, 34
- 5.2 Loads and Constraints, 34
- 5.3 Meshing, 34
- 5.4 Gmsh, 35
- 5.5 Assemblies, 35

## *Chapter 6*

### **Analysis Types**

**36**

- 6.1 Static, 36
- 6.2 Nonlinear Static 3D, 37
- 6.3 Modal Vibration, 38
- 6.4 Dynamic Response, 39
- 6.5 Nonlinear Dynamic Response 3D, 40
- 6.6 Buckling, 40
- 6.7 Thermal, 42
- 6.8 Fluid, 42
- 6.9 DC Current Flow, 43
- 6.10 Acoustic Resonance, 43

## *Chapter 7*

### **Elements**

**45**

- 7.1 Plane Continuum Elements, 45
- 7.2 Axisymmetric Continuum Elements, 46
- 7.3 Solid Continuum Elements, 46
- 7.4 Shell, 47
- 7.5 Beam, 50
- 7.6 Truss, 52
- 7.7 Spring, 53
- 7.8 Damper, 53
- 7.9 Tension Only, 54
- 7.10 Fin, 54
- 7.11 Resistor, 54

## *Chapter 8*

## Materials

56

- 8.1 Library, 56
- 8.2 Defining a New Material, 56
- 8.3 Mixed Materials, 56
- 8.4 Mixed Elements, 56
- 8.5 Anisotropic Materials, 57
- 8.6 Temperature Dependent Properties, 57
- 8.7 Failure Criteria, 57
- 8.8 Nonlinear Materials, 58

## *Chapter 9*

---

## Loads and Constraints

59

- 9.1 Fixed support, 60
- 9.2 Frictionless support, 60
- 9.3 Pinned support, 60
- 9.4 Elastic support, 61
- 9.5 Compression only support, 61
- 9.6 Displacement, 61
- 9.7 Node rotation, 62
- 9.8 Bonded contact, 62
- 9.9 Contact, 64
- 9.10 Node-surface coupling, 64
- 9.11 Pre-tension section, 65
- 9.12 Flexible joint on beam, 66
- 9.13 Force, 67
- 9.14 Pressure, 68
- 9.15 Traction, 68
- 9.16 Line pressure, 69
- 9.17 Hydrostatic Pressure, 69
- 9.18 Moment, 70
- 9.19 Gravity, 71
- 9.20 Centrifugal force, 71
- 9.21 Mass, 72
- 9.22 Rotational inertia, 72
- 9.23 Temperature, 72
- 9.24 Node temperature, 73
- 9.25 Thermal stress, 73
- 9.26 Rayleigh damping, 73
- 9.27 Heat flow rate, 74
- 9.28 Heat flux, 74
- 9.29 Internal heat generation, 74
- 9.30 Convection, 75
- 9.31 Radiation, 75
- 9.32 Thermal contact conductance, 75
- 9.33 Velocity, 75
- 9.34 Fluid pressure, 76

- 9.35 Electric potential, 76
- 9.36 Charge, 76
- 9.37 Current, 76
- 9.38 Robin boundary condition, 77
- 9.39 Cyclic symmetry, 77
- 9.40 Constraint equation, 78

## *Chapter 10*

### **Results**

**79**

- 10.1 Display, 79
- 10.2 File Output, 82
- 10.3 Stress Linearization, 82
- 10.4 Mean and Volume Integral, 84
- 10.5 Surface integral, 84
- 10.6 Sum, 84
- 10.7 Formula, 84

## *Chapter 11*

### **Samples and Verification**

**86**

- 11.1 BeamBendingAndTwisting.liml, 86
- 11.2 CompositeBeam.liml, 87
- 11.3 CylinderLifting.liml, 89
- 11.4 PressureVesselAxisymmetric.liml, 90
- 11.5 TwistedBeam.liml, 91
- 11.6 MembraneActionPlate.liml, 92
- 11.7 SaggingCable.liml, 93
- 11.8 BucklingBeam.liml, 94
- 11.9 BucklingPlate.liml, 95
- 11.10 BucklingPlateNonlinear.liml, 96
- 11.11 PipeClip.liml, 97
- 11.12 BoltedJoint.liml, 98
- 11.13 FinConvection.liml, 100
- 11.14 ConductionConvectionRadiation.liml, 101
- 11.15 OscillatingHeatFlow.liml, 102
- 11.16 FluidCouette.liml, 104
- 11.17 FluidViscousCylinder.liml, 105
- 11.18 VibratingFreePlate.liml, 106
- 11.19 VibratingCantileverBeam.liml, 107
- 11.20 VibratingCantileverSolid.liml, 108
- 11.21 VibratingTrussTower.liml, 110
- 11.22 VibratingMembrane.liml, 111
- 11.23 Impeller.liml, 112
- 11.24 VibratingString.liml, 114
- 11.25 PandSWaves.liml, 115
- 11.26 DampedVibratingStrip.liml, 117

- 11.27 WheatstoneBridge.liml, 118
- 11.28 Capacitor.liml, 119
- 11.29 PiezoelectricStack.liml, 120

## **Chapter 12**

---

### **File Formats** 124

- 12.1 Liml, 124
- 12.2 STEP (.step/.stp), 124
- 12.3 DXF, 124
- 12.4 STL, 125
- 12.5 Gmsh (.msh), 125
- 12.6 UNV, 125
- 12.7 Netgen (.vol), 126
- 12.8 Polygon File Format (.ply), 126
- 12.9 JPEG, PNG, BMP, 126
- 12.10 XYZ, 126
- 12.11 ANSYS command file (.txt), 126
- 12.12 CalculiX (.inp), 126
- 12.13 CalculiX results (.frd), 131

## **Chapter 13**

---

### **CalculiX Solver** 132

- 13.1 Custom model definition, 132
- 13.2 Custom step contents, 133
- 13.3 Modify keyword, 133
- 13.4 Don't generate STEP, 133
- 13.5 No extra contact step, 133
- 13.6 Custom element type, 133

## **Chapter 14**

---

### **Automation** 134

- 14.1 Command Line Parameters, 134
- 14.2 Scripts, 134

## **Chapter 15**

---

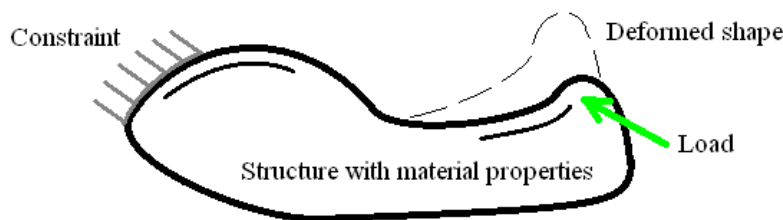
### **License Agreements** 136

- 15.1 Mecway, 136
- 15.2 ARPACK, 136
- 15.3 SlimDX, 136
- 15.4 Bouncy Castle, 136
- 15.5 Netgen, Pthreads-win32, ZedGraph, and OCCT version 7.3, 136

- 15.6 GNU Lesser General Public License, 137
- 15.7 OCC CAD Kernel, 139

## Welcome to FEA

Suppose you want to solve a physical problem such as finding the stresses in an object when some prescribed forces are applied. This is a typical problem for FEA: some type of load is applied to an object and the response calculated subject to specified constraints.

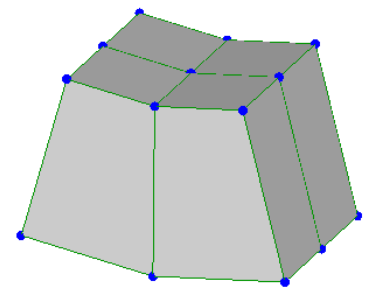


In a mechanics problem, the object might be a gear wheel, the load might be a force applied from another gear, the response might be the stresses throughout the gear wheel. The constraint is that the gear must remain on the shaft.

In a thermal problem, the object might be an electronics enclosure, the load might be heat flow into its interior surface and convective heat flow from its exterior surface. The response would be the temperature distribution in the material.

The finite element method is a numerical technique for gaining an approximate answer to the problem by representing the object by an assembly of simple shapes – the finite elements. Each of these elements is given material properties and is connected to adjacent elements at nodes – special points on the edges of the element. This assemblage of elements connected at their nodes is called a mesh.

Because the elements can easily be assembled into complicated shapes, FEA is a popular and powerful method for realistically predicting the behavior of many engineering structures and components.



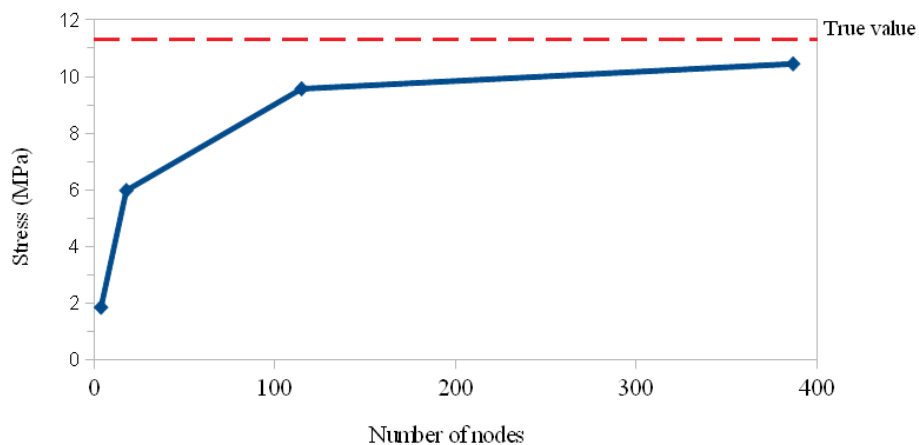
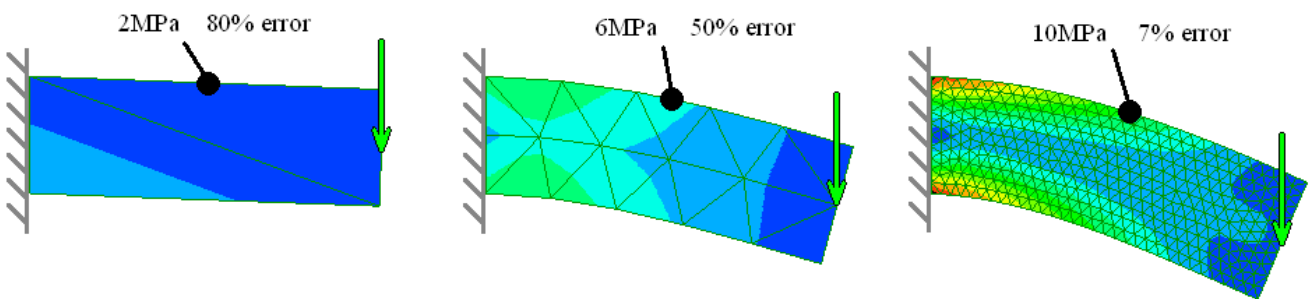
*A mesh consisting of 4 elements and 18 nodes*

The process of using the finite element method is usually iterative – you should solve the model several times to estimate the error in the results and reduce it to an acceptable level. This is called a mesh convergence study.

### 1. Build the model:

- Either place individual nodes and elements one by one or
- use the mesh creating and editing tools to make it easier or
- import a CAD model and use the automeshing tool to generate a mesh for you.

- Assign material properties to the elements and
  - apply loads and constraints to the mesh.
2. Solve the model:
- Define the type of analysis you want e.g. static mechanical, vibrational modes, dynamic response with time, etc.
  - Let Mecway's solver do the work.
3. Refine the mesh and solve again until the results don't change much:
- Record the previous solution values at the points of interest.
  - Refine the mesh, increasing the density of elements in the region of interest.
  - Solve again and repeat until the solution values are similar to the previous solution.

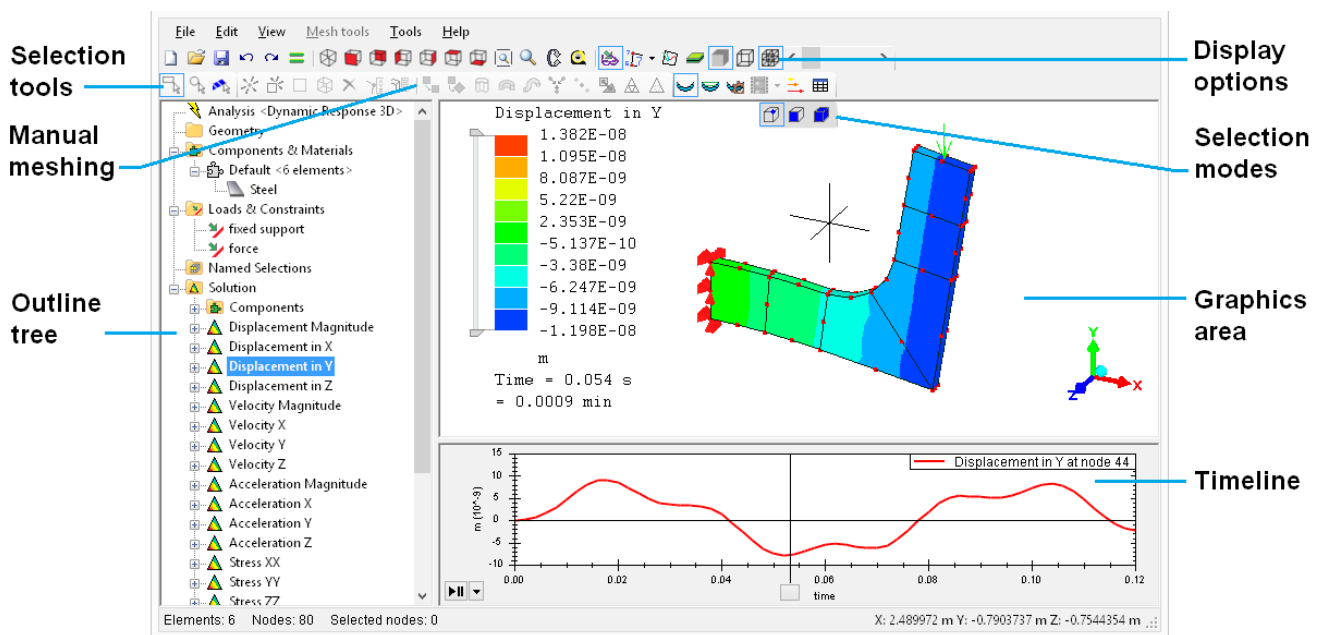


This mesh convergence plot shows a stress component at the top edge of the bar for each of 4 different mesh densities. With around 400 nodes we might consider the error is acceptable. Other element shapes can give faster convergence with fewer nodes.



## Overview of Mecway

This chapter introduces you to the Mecway work-flow for accomplishing your finite element analysis.



### 2.1 Mesh

A finite element mesh consists of nodes (points) and elements (shapes which link the nodes together). Elements represent material so they should fill the volume of the object being modeled. The mesh is displayed in the graphics area which occupies most of the Mecway window.

You can edit a mesh using the **Mesh tools** menu or by selecting parts and using the right-click context menu to access the mesh editing tools.

The other contents of a model are shown in the outline tree on the left hand side of the window. It has several groups containing different types of item listed below. Most of these items can be modified through a context menu which you can access by right-clicking on them.

### 2.2 Analysis

You can change global properties such as analysis type, physical constants, solver settings and output options by editing the Analysis item in the outline tree

## 2.3 Geometry

If you generate a mesh from a STEP file exported from CAD then these files are shown in the Geometry group. Each geometry item can be auto-meshed to generate a mesh.

## 2.4 Components & Materials

A component is an exclusive collection of elements. Every element must belong to exactly one component.

Components are used for assigning materials and controlling the appearance (color and visibility) of elements. All elements in a component share the same material and color. For complex models with logically different parts or features it can be helpful to assign each part to a component to aid in working on the mesh.

Each component containing some elements must have a material assigned to it. The same material can be shared between several components.

You can convert a component to a named selection by selecting its elements then creating a new element selection or adding them to an existing named selection. A named selection containing elements can be converted to a component in a similar way.

## 2.5 Loads & Constraints

This group contains all the loads and constraints in the model. Loads which are applied to named selections show their named selections as child nodes in the outline tree.

## 2.6 Named Selections

A named selection is a non-exclusive collection of nodes, elements or faces. A face is a face or edge of an element. You can use named selections for applying loads and constraints or just to help organize the model. For example, to apply a force to the surface of an object, instead of applying it directly to the faces, you can put all the faces in a named selection and apply the force to the named selection.

## 2.7 Solution

After solving, the results are shown under the **Solution** branch in the outline tree. You can click on a field variable to display a colored contour plot of it. To add more field variables, right click **Solution** or open the **Solution** menu on the menu bar and choose new ones. Some can be generated and displayed immediately while others require you to solve the model again to generate them. If you solve a model when no field variables are listed under Solution, it will produce the minimum set needed to obtain most of the others from without solving again. It also includes von Mises stress where applicable.

## 2.8 Configurations

Configurations allow you to define several different sets of loads and constraints in the same model. For example, you may have one configuration for dead loads and another for dead loads and live loads together. Each configuration has its own separate solution so you can also use them to keep multiple solutions available while changing something in the model.

Use **Edit** → **New configuration** to create a new configuration. Then suppress and unsuppress loads and constraints independently in each configuration or for all configurations at once.

Click the tabs above the outline tree to select the active configuration that will be displayed and solved.

## 2.9 Units

Each numerical value has its own unit. You can change the unit of any quantity by selecting it from the drop-down box next to the number. This will also convert the number into the new unit so it maintains the same physical meaning. An exception is input tables (eg. temperature dependent material properties and time dependent loads) and formulas which are left unchanged.

Density	<input type="text" value="0.28"/>	<div>lb/in<sup>3</sup> ▼</div> <div>kg/m<sup>3</sup></div> <div>g/mm<sup>3</sup></div> <div>lbf.s<sup>2</sup>/in<sup>4</sup></div> <div>lbf.s<sup>2</sup>/ft<sup>4</sup></div> <div>lb/in<sup>3</sup></div>
Thermal expansion coefficient	<input type="text" value="0"/>	
Speed of sound	<input type="text" value="0"/>	


The default units for newly created items are determined by previously used units so it learns your preferences.


Pounds-force and pounds-mass are distinguished by the symbols lbf and lb respectively.

## Viewing and Selecting

### 3.1 Zoom, Pan, Rotate

#### 3.1.1 Tool buttons

 Rotate to isometric view.

 Rotate the view so one set of axes is parallel to the screen.

 Fit to window

 Zoom with the left mouse button. Use this if you don't have a mouse wheel.

 Rotate with the left mouse button. Use this if you don't have a middle button.

#### 3.1.2 Keyboard

+	Zoom-in
-	Zoom-out
Arrow Keys	Pan up/down/left/right
Alt + Arrow Keys	Rotate 5° about the horizontal or screen-normal axis
F4 and F5	Rotate 5° about the vertical axis

#### 3.1.3 Mouse

Rotate	Drag with the middle button. Click to position the center of rotation.
Zoom	Rotate the mouse wheel
Pan	Drag with the right button

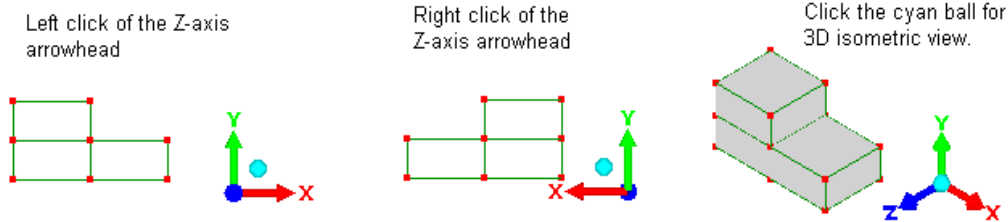
You can configure the middle and right buttons to have the opposite effect or reverse the mouse wheel zoom direction in **Tools** → **Options** → **View control** tab.

#### 3.1.4 3D mouse

A 3Dconnexion device such as SpaceMouse can be used for: 3D navigation (pan, zoom, rotate), Fit, 6 preset orthogonal views, ISO1 view, and QuickZoom.

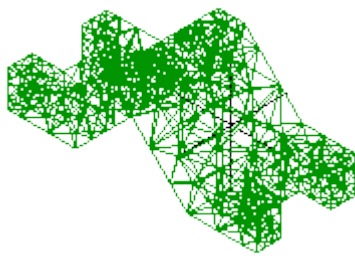
### 3.1.5 Triad

The triad at the bottom right of the screen can be used to quickly rotate the view parallel to the XY, YZ or ZX planes or isometrically. Left or right click on the arrowheads or the cyan ball to set the view parallel to the axes or to isometric.

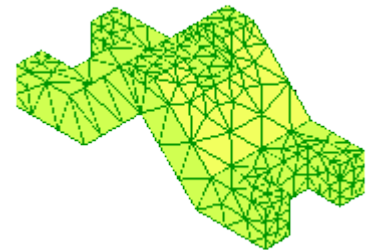


The default orientation has the positive Y direction upwards and positive Z towards the front. You can change to the other popular convention with positive Z upwards and positive Y towards the back using **Tools -> Options -> View orientation**. This option determines the orientations obtained by clicking the triad arrowheads, and the 6 default orthogonal views (front, back, etc). It has no effect on the model or solution.

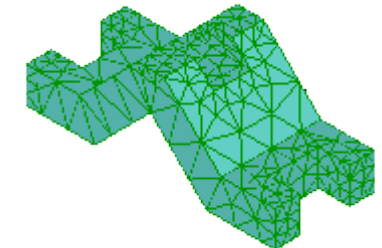
### 3.2 Display modes




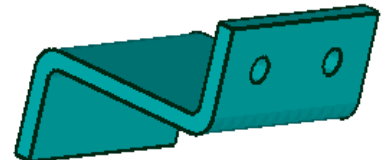
 Show element surfaces

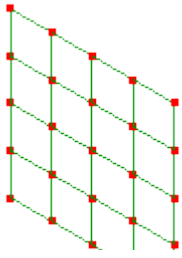


 Show element edges

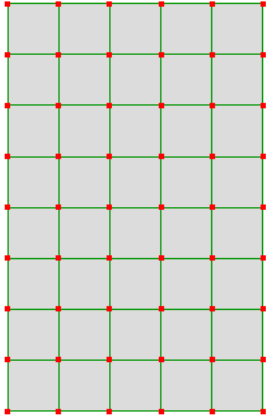
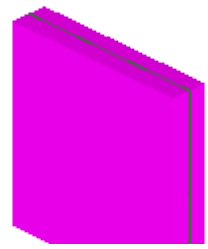


 Show model edges

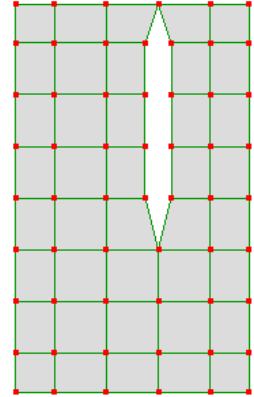




Toggle shell thickness display when element surfaces are displayed.



**Open cracks** in the **View** menu toggles this mode. It helps to show narrow gaps in the mesh where elements appear to be connected but are not sharing the same nodes. When open cracks mode is on, the outside surface of a mesh is shrunk, enlarging any gaps.

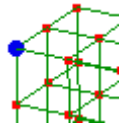


### 3.3 Selection

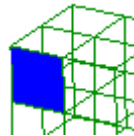
Mecway is selection driven which means to perform most mesh editing tasks you first have to select nodes, element faces or entire elements.



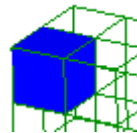
**Select nodes**



**Select faces**



**Select elements**



Changing between these three modes converts any existing selection to the new type.



**Rectangle selection**



**Circle selection**



**Paint selection** Drag the mouse to select faces bounded by sharp edges of the mesh.



**Edge detecting selection** Click on a face to quickly select the whole surface bounded by sharp edges of the mesh.

Hold **Ctrl** while selecting to add or remove items from the selection.

Hold the **Shift** key to disable node dragging while selecting nodes with mouse.

**Edit** → **Select nodes by formula** allows you to select nodes according to an inequality of the form

$$f(x, y, z) < g(x, y, z)$$

where  $f()$  and  $g()$  are mathematical functions of position. For example, to select nodes within a sphere of radius 3m, centered on the origin, enter  $x^2+y^2+z^2 < 3^2$  and set the **Position** unit to  $m$ .

## Manual Meshing

This chapter will explain how to use the tools that are available in Mecway for creating your finite element model. Unlike computer aided design (CAD) software which uses lines, surfaces and solids, finite element analysis software uses only nodes and elements. It is also possible to import CAD models into Mecway and create a suitable mesh with Mecway's automeshing tools, but this is described in Chapter 5 – CAD Models.

A finite element model is a mesh of elements. Each element has nodes which are simply points on the element. Elements can only be connected to other elements node-to-node. An element edge-to-element-node is no connection at all. Elements themselves have very simple shapes like lines, triangles, squares, cubes and pyramids.

Each element is formulated to obey a particular law of science. For example in static analysis, the elements are formulated to relate displacement and stress according to the theory of mechanics of materials. In the case of modal vibration the elements are formulated to obey deflection shapes and frequencies according to the theory of structural dynamics. Similarly, in thermal analysis the elements relate temperature and heat according to heat transfer theory. So it is essential that you have an understanding of the underlying physics theory before using finite element analysis software.

When beginning a new model, first check whether or not your choice of element shape is actually supported by the type of analysis. The element shapes that are available to each type of analysis are listed in chapter 6 – Analysis Types.



The manual meshing tools are grouped together by purpose in this manual:

- i. creating tools, that bring into existence a two dimensional mesh
- ii. editing tools, that form and modify the created two dimensional mesh
- iii. tools that will convert a two dimensional mesh into a three dimensional mesh
- iv. refinement tools for converging results

### 4.1 Creating Tools


This section describes in turn each of the tools for creating a mesh. Tools for modifying it are described in Section 4.2.

#### 4.1.1 Quick square / Quick cube

If you're making a simple orthogonal model or want to do a quick test on some feature in Mecway, use the **Mesh tools** → **Create** → **Quick square** or  and the **Mesh tools** → **Create** → **Quick cube** or . They have side lengths of 1 and can be used as building blocks for a model by scaling, re-positioning and refining.



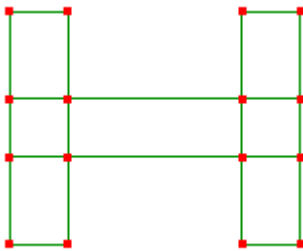
#### 4.1.2 New node


Use the **Mesh tools** → **Create** → **Node...** or  to place nodes by mouse clicks or entering coordinates. If you use mouse clicks, the nodes are placed on a plane parallel to the screen which passes through the origin so it's helpful to use one of the orthogonal views.



You can create a uniform line, arc or helix by checking **Relative to previous** and defining the position with an offset in Cartesian or polar coordinates. This option only becomes available after creating the first node.

#### 4.1.3 New element



To create individual elements use **Mesh tools** → **Create** → **Element...** or . You can draw elements by clicking on empty space to place each node. If you click an existing node, the element will be linked to that.

The order in which the nodes are clicked will affect the direction in which element subdivisions take effect when using the editing tools. So be consistent in how you are clicking the nodes. For example, you can choose to click the nodes by going counter-clockwise starting at the lower left corner.

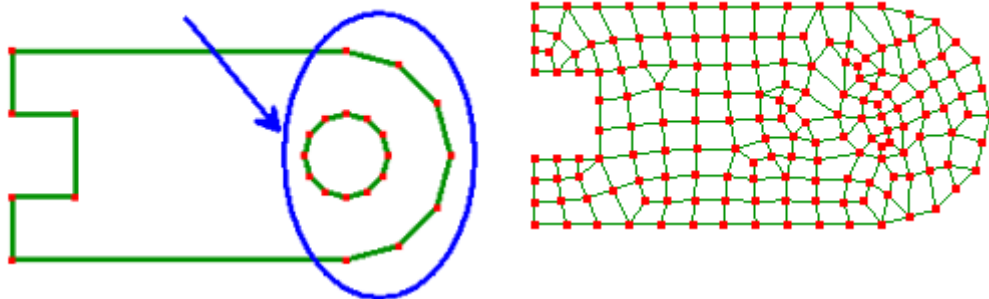
#### 4.1.4 Insert node between

- Select two or more nodes then use the **Mesh tools** → **Insert node between** to create a node at the centroid of the selected nodes. This is useful when laying out nodes for a coarse mesh.

#### 4.1.5 Curve generator

The **Mesh tools** → **Create** → **Curve generator...** creates line element curves defined by 3D parametric equations. If these are boundaries of flat closed shapes, then you can use the two dimensional automesh, **Mesh tools** → **Automesh 2D...** to fill in the shapes.

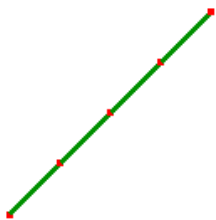
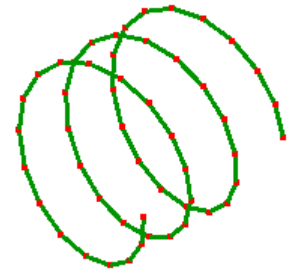
The circle and the arc was made using the curve generator.



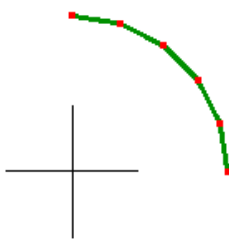
It can also be used to generate 3D shapes made from line2 or curved line3 elements such as a helix using these parameters:

**p initial**      0  
**p final**         $3*2*\pi$       (3 loops)  
**X=**                 $\cos(p)*1$       (radius is 1)  
**Y=**                 $\sin(p)*1$       (radius is 1)  
**Z=**                 $p/10$           (pitch is 10)

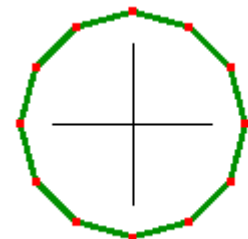
**Number of elements** 50



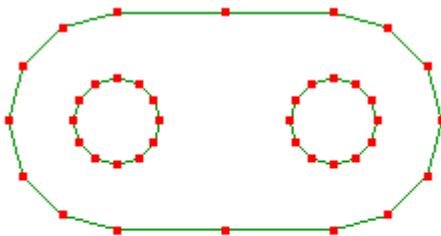
Several shapes have predefined equations so you can create them quickly. These are a straight line, arc, circle, ellipse and parabola.



The most common use of the curve generator is to create arcs and circles. Arcs can be created using either the center, start and end points or by specifying the start, end and any point lying on the arc.



#### 4.1.6 Automesh 2D

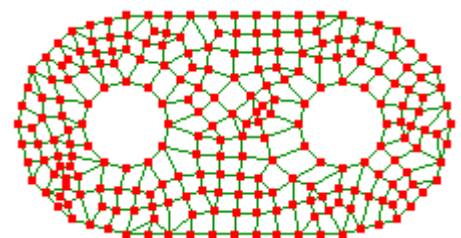


The **Mesh tools** → **Automesh 2D...** is used to fill an area bounded by line elements or formed by plane elements with either quadrilateral or triangle or a mix of both element shapes. After a successful automesh the original elements will no longer exist. It can only create elements that lie in a plane, which can be in any orientation.

The automesher will fill the entire bounding area with elements including any holes.

You will then have to manually delete the elements in the hole areas.

Curved edges can be defined using quadratic elements (line3, tri6, or quad8) so they don't become faceted when the mesh is very fine.




Depending on how you created and edited your model, you may have places where two parts of the bounding lines appear to be joined but are not. For example these two line elements look as though they are connected with each other, but on displaying their node numbers, it's clear that there are actually two overlapping nodes. This means the line elements are not connected to each other.



If there are any unconnected line elements, the automesher will fail.

Therefore, before running the automesher, use the editing tool **Mesh tools** → **Merge nearby nodes...** to replace any overlapping nodes with a shared node, thereby connecting all elements.

If the automesher default values make a mesh with only a few large elements, re-run the automesher using a smaller value for the **Maximum element size**. If you don't know what maximum size value to specify, use the tape measure tool  to measure the smallest line segment. It will give a dynamic read-out as you click and drag from one node to another.

By default, quadrilateral elements are set to be the dominant element of the mesh. If you have a good reason for using triangle elements, you may uncheck **Quad dominant**.

You can also use 2<sup>nd</sup> order elements with midside nodes (tri6 and quad8) by checking **Quadratic elements**.

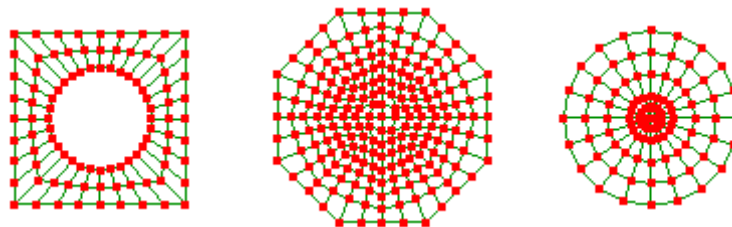
#### 4.1.7 Automesh 3D

**Mesh tools** → **Automesh 3D...** remeshes a solid part. It can also fill in the volume of a closed shell mesh and can include spherical regions of local refinement.

A limitation is that information about named selections and multiple components is lost so loads have to be reapplied afterwards.

#### 4.1.8 Plate mesh

In **Mesh tools** → **Create** → **Plate mesh** there are templates for creating simple shapes like a circular, square and octagonal plates with or without holes. These templates are simple to use and are self-explanatory.



These shapes can be extruded or revolved to generate three dimensional solids.

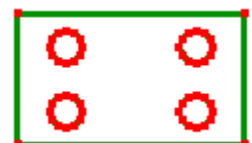
## 4.2 Editing Tools

### 4.2.1 Move

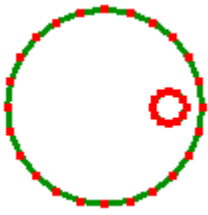
The **Mesh tools** → **Move/copy...** is used to reposition or duplicate nodes or elements. You can choose a coordinate system from Cartesian, cylindrical or spherical polar coordinates. The polar coordinates allow you to change the radial size of an object while keeping its thickness constant.



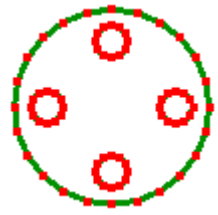
Note the copy check box. If this is ticked, the selection is both moved and duplicated. Bear in mind that the copies are not connected to each other. Use the **Mesh Tools** → **Merge nearby nodes...** to connect them.



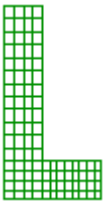
#### 4.2.2 Rotate



The **Mesh tools** → **Rotate/copy...** is used to rotationally reposition nodes or elements. With the copy option selected, nodes and elements can be duplicated.

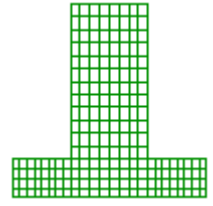


#### 4.2.3 Mirror

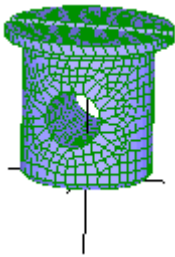


The **Mesh tools** → **Mirror/copy...** is used to reposition nodes or elements by reflecting them about either the XY, YZ or ZX planes or about a single node. When the **Copy** option is selected, it can be used to mirror meshes.

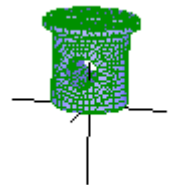
At the mirror joint the elements will not be connected so you will have to use the **Mesh tools** → **Merge nearby nodes...** to make it a continuous mesh.



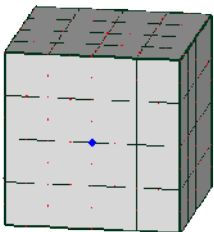
#### 4.2.4 Scale



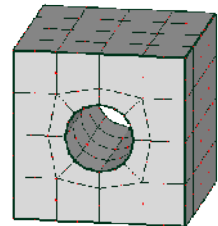
The **Mesh tools** → **Scale...** is used to re-size either the entire mesh or the selected items. If you're not re-sizing the entire mesh but only a selected portion of the model, you should move it so that it is centered at the origin. This is because scaling is done relative to the origin.



#### 4.2.5 Hole



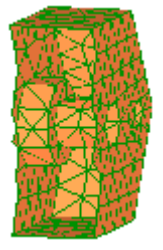
**Mesh tools** → **Hole...** cuts a circular hole through a solid or shell mesh. Define the center point by selecting a node. Choose the direction as either normal to the surface, or parallel to the X, Y or Z axis. Define the diameter by entering its value. It works best on regular hex or quad meshes and may produce some badly shaped elements with more complex meshes.



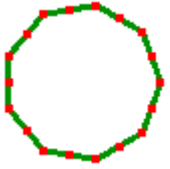
#### 4.2.6 Hollow



The **Mesh tools** → **Hollow** is used to convert a solid mesh into a shell mesh.

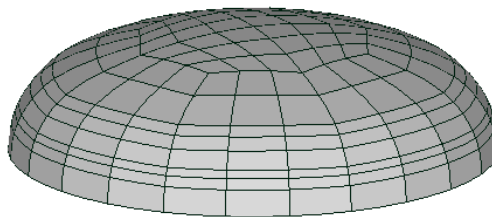
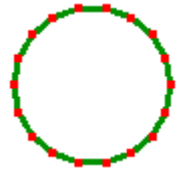


#### 4.2.7 Fit to curved surface

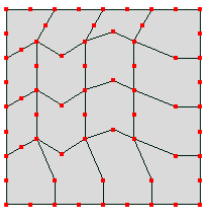


The **Mesh tools** → **Fit to curved surface...** is used to deform a mesh into a smooth curved shape of sphere, torispherical dome, cylinder or cone.

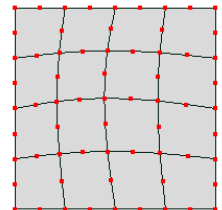
You can use it to create smooth curves on a mesh that has become faceted after refining, to round a corner or create a pressure vessel end from a flat disk.



#### 4.2.8 Smooth



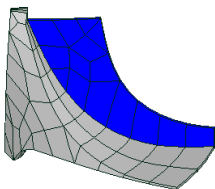
**Mesh tools** → **Smooth** improves the shapes of distorted elements by moving each node towards the mean position of its neighbors. The shapes of existing surfaces are preserved and edges are unchanged. You should avoid repeated use of this tool on coarse curved surfaces because moving the nodes on the surface slightly modifies the shape of the new surface defined by those nodes.



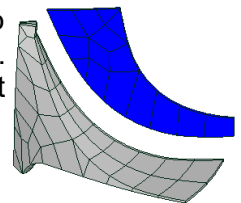
#### 4.2.9 Transfer displacements from solution

**Mesh tools** → **Transfer displacements from solution** deforms the model according to the displacement field variables in the currently selected solution data set. You can specify a scale factor which each displacement value will be multiplied by.

#### 4.2.10 Disconnect elements



To separate part of the mesh from the rest, select the elements to be separated then use **Mesh tools** → **Disconnect elements**. There is no immediate graphical change but you can verify that they are disconnected using **View** → **Open cracks**.




#### 4.2.11 Merge nearby nodes

The **Mesh tools** → **Merge nearby nodes** will ensure that the elements are connected node-to-node by replacing overlapping nodes with a single shared node. Meshing operations such as **Mesh tools** →

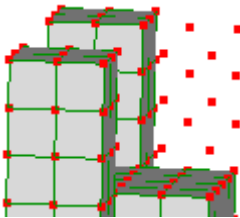
**Refine** → **Custom...** or **Mesh tools** → **Move/Rotate** with the Copy option will create meshes that are not connected. Separate files assembled using **File** → **Import...** will also not be automatically connected to each other at the mating surface.

If some nodes are selected before using this tool then only the selected nodes will be considered. However if you check **Merge other nodes into selected nodes** then the selected nodes will not be moved but any other nodes within the tolerance distance will be merged with them.


The **View** → **Open cracks** tool will expose unconnected elements. It shrinks elements slightly to open up any existing gap between adjacent faces of unconnected elements.

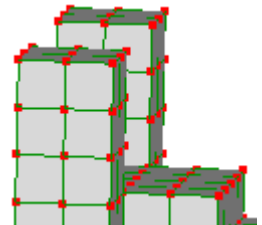
To eliminate these gaps use the **Mesh tools** → **Merge nearby nodes** to delete overlapping nodes. You have to specify a radial distance within which two or more nodes will be replaced by a single node. Too small a value and some overlapped nodes will not be eliminated. Too large a value and you risk collapsing elements as they lose a node. Use the **Tape Measure** tool  to determine the smallest distance between two nodes in your mesh, then use a value smaller than this so that elements don't collapse. You will notice a reduction in the number of nodes shown in the status bar after using this command if it merged any nodes.

#### 4.2.12 Delete unused nodes



The **Mesh tools** → **Delete unused nodes** will remove any node not belonging to an element or constraint equation.

If you use **Edit** → **Delete elements and retain nodes**, the nodes will be left behind. If you can't see them, activate the node select mode .



#### 4.2.13 Invert

Solid elements can sometimes be formed inside-out. This will cause the solver fails with a message about incorrect element topology. You can select the affected elements and use **Mesh tools** → **Invert** to fix their topology. Invert can also be useful for flipping shell elements so their top faces are all on the same side.

#### 4.2.14 Deleting items

To delete elements, first select them or any of their faces then press **Del**. This will also delete any unused nodes left behind. To delete elements without deleting their nodes press **Ctrl + Del**.


To delete a component along with all its elements and their nodes, right click the component in the outline tree and select **Delete**.

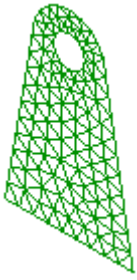
To delete nodes, first select the nodes then press **Del**.

### 4.3 Converting a 2D Mesh into a 3D Mesh

Once a 2D plane mesh has been created it can be extruded, revolved or lofted to create a 3D model.

#### 4.3.1 Extrude

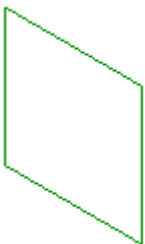
Extrusions can only be done on faces. Select faces using **Select faces** .



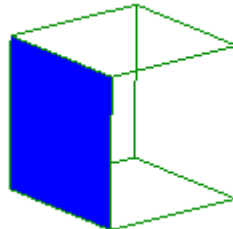
Use **Mesh tools** → **Extrude...** to create a 3D solid mesh.



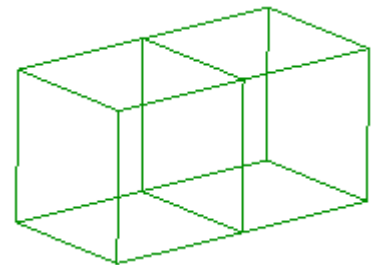
Line element and shell edge faces extrude into shells



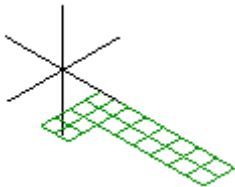
Shell or solid faces




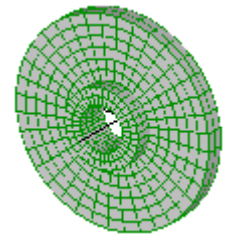
extrude into solids



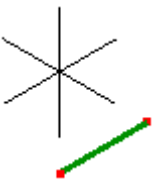
#### 4.3.2 Revolve



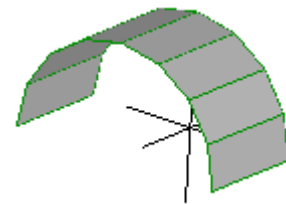
Revolve can only be used on faces. Select faces using **Select faces** 

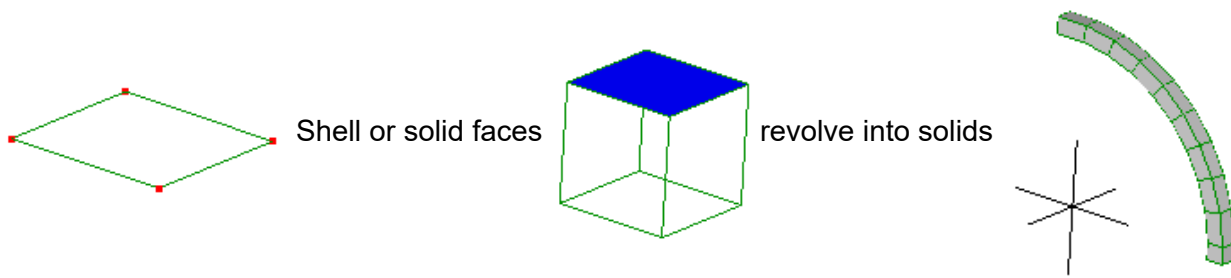


Use **Mesh tools** → **Revolve...** to create a 3D solid mesh. Any nodes on the axis of revolution will be duplicated for each element, making a disconnected mesh. To fix this problem, use **Mesh tools** → **Merge nearby nodes** afterwards.



Line elements and shell edge faces revolve into shells

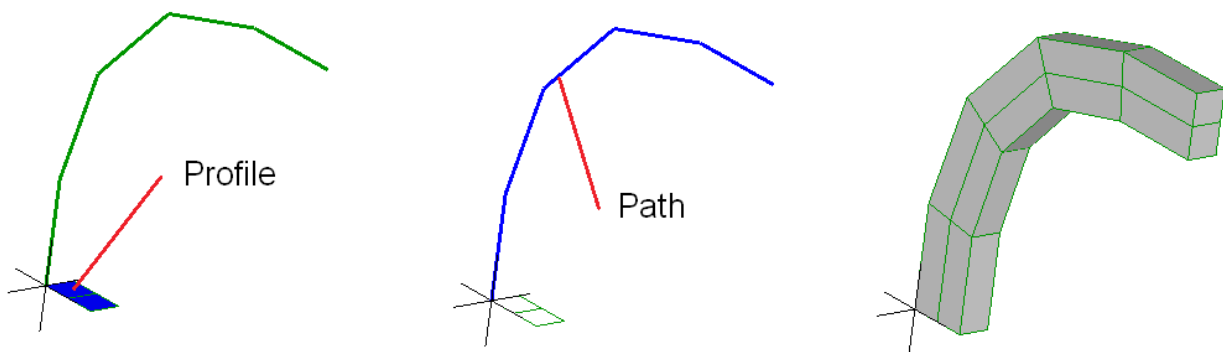




#### 4.3.3 Sweep

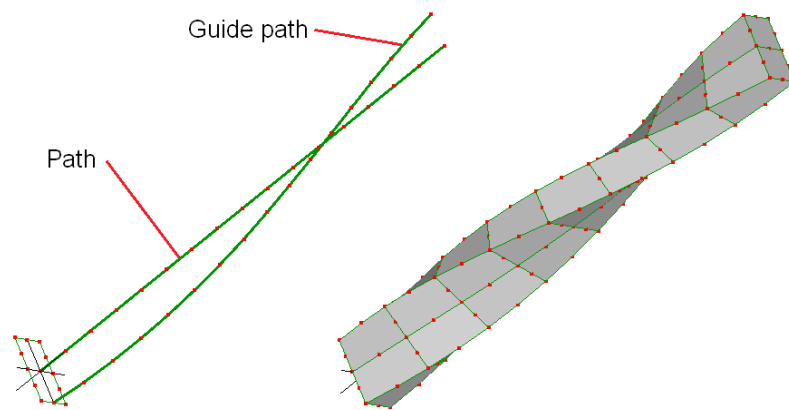
The sweep tool performs extrusion along a defined path which could be curved or have non-uniform element spacing.

After starting the tool with **Mesh tools** → **Sweep...**, select the faces that define the profile then press the corresponding **Accept** button. If the profile faces are planar they will be swept into solids. If they are the edges of shells or lines, they will be swept into shells. Next, select the faces that define the path followed by **Accept**. The path must consist of the edge faces of line or shell elements and must have one end node in common with the profile.

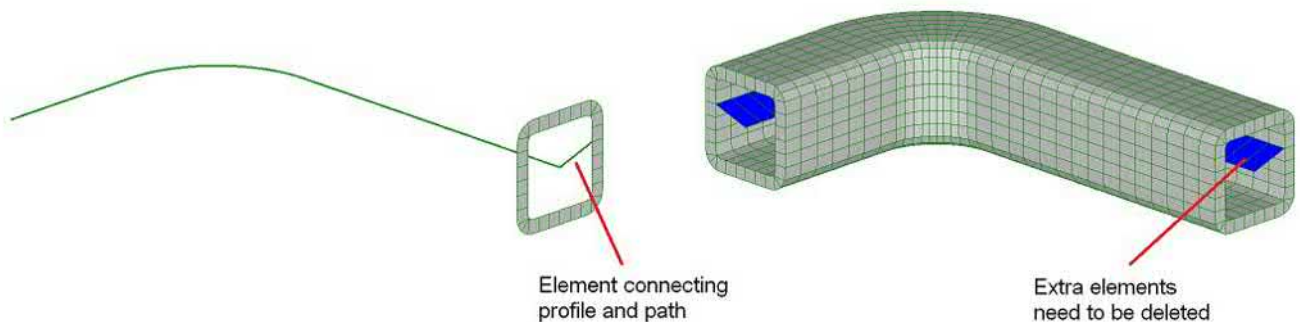


If you need to control the orientation of the profile as it is extruded, for example to make a helix or twisted shaft, you can define a guide path. Each face in the guide path determines the orientation of the profile at the corresponding face of the path. The guide path must have one end node in common with the profile. If it contains more faces than the path, the additional faces are ignored. If it contains fewer faces than the path, the orientation beyond the last guide path face remains the same as it was at the last guide path face.

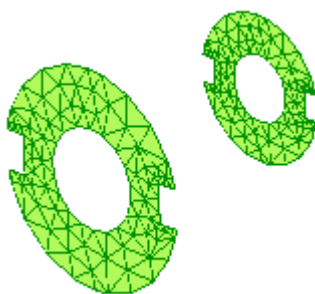




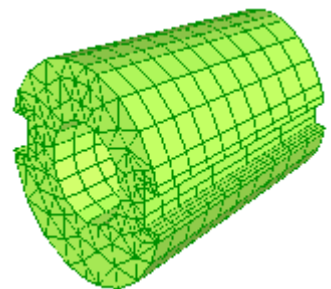
If the path or guide path doesn't intersect the profile, you can add a line element connecting them with its face being part of the profile. This will result in extra elements that you need to delete after performing the sweep.



#### 4.3.4 Loft



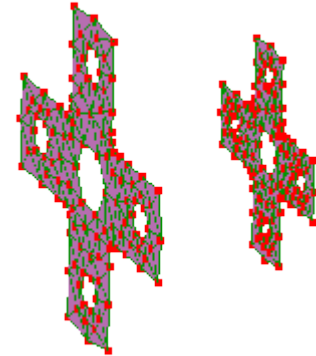
The loft tool fills the gap between two profiles having matching nodal patterns with solid elements. This can be used for creating tapered parts. The order of the node numbers must be identical on each profile with the only difference being a constant offset. You can use **Mesh tools** → **Move/copy** to generate a 2<sup>nd</sup> mesh which satisfies this requirement.



## 4.4 Refinement Tools

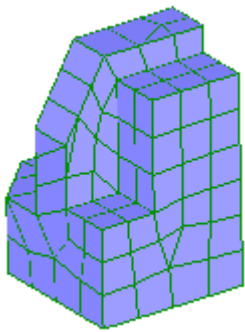
An individual element cannot accurately represent a complicated change in the field values across it. It approximates it as either a constant, linear or quadratic function. Therefore you need smaller elements in regions of the mesh where the results change rapidly.


To determine the areas that need mesh refinement you will first need to solve a coarse mesh and look at the color contour plot of the solution. If you find the field value is changing by a large amount within a single element, that tells you the area may need more elements.

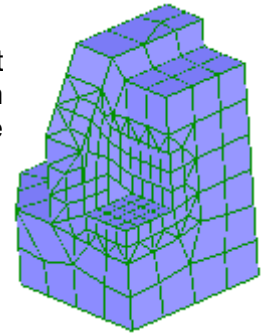


Checking for mesh convergence is a crucial part of finite element analysis because without an estimate of the error, the results may be wildly wrong. To do this, you should solve the same model with several different levels of mesh refinement and note down the important field values for each mesh. If additional refinement leads to only a small change in the results, such as 3%, then it is said to have converged and no further refinement is needed.

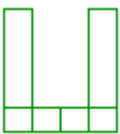
### 4.4.1 Refine x2



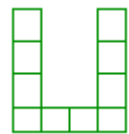
**Mesh tools** → **Refine** → **x2** or  replaces each element with two elements along each edge. It also inserts transition elements so that the refined part of the mesh will be correctly connected to the remaining unrefined part.



### 4.4.2 Refine Custom



**Mesh tools** → **Refine** → **Custom...** is used to subdivide elements by specifying the number of subdivisions along each of three directions. If no elements are selected, it uses the entire mesh.



### 4.4.3 Refine x3 (1D/2D elements only)

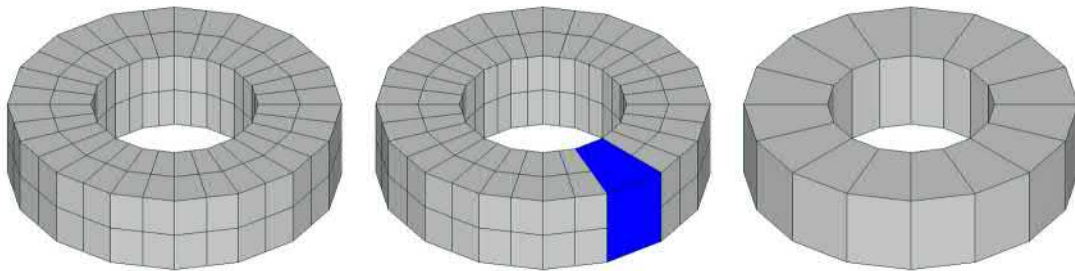
**Mesh tools** → **Refine** → **x3 (1D/2D elements only)** refines the selected nodes or elements by subdividing each element into three elements along each direction. It can be used with line2/3, tri3/6 and quad4/8 elements.

#### 4.4.4 Unrefine x2

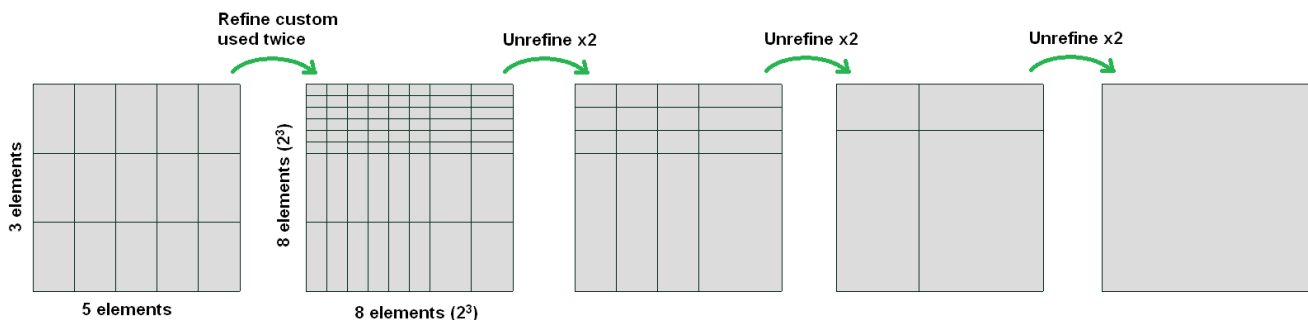
**Mesh tools** → **Refine** → **Unrefine x2** reverses the effect of a globally applied Refine x2 to produce the original coarser mesh. It supports all element shapes except pyramid and wedge (pyr5, pyr13, wedge6, wedge15).

Loads and constraints, named selections and components are maintained where they are the same across all the elements or faces that make up a single unrefined element or face respectively.

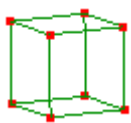
The unrefine algorithm requires a corner node to start from. A corner here is a node that's used by only one element of the same shape. Some meshes have no corners, such as a cylinder or sphere. In those cases, select a subset of the elements which does have a corner and unrefine them first. Then select the remaining elements (**Edit** → **Invert selection**) and unrefine them as shown below.



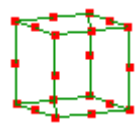
If the mesh has a different density along each edge, such as produced by Refine Custom, use **Mesh tools** → **Refine** → **Custom** on some of the elements so that each edge of a region has  $2^n$  elements then apply the **Mesh tools** → **Unrefine x2** tool  $n$  times as shown below.



#### 4.4.5 Change element shape



Higher order elements are often more efficient. Convert between linear and quadratic elements using **Mesh tools** → **Change element shape...**



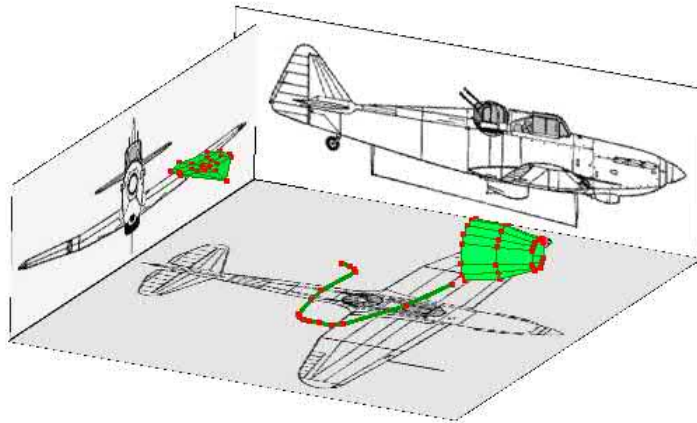
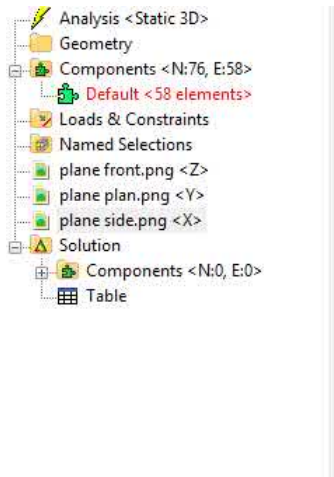
It can also replace some element shapes with templates of different shapes, such as converting a quadrilateral into two triangles. The choice of element shapes will be enabled or disabled according to the element shapes present, so you might have to repeat this step to get the final desired element.

### 4.5 Tracing an Image

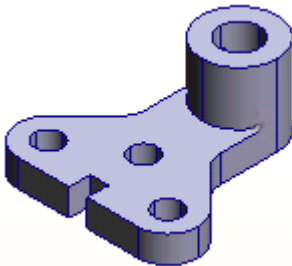
If you have a projected image (jpg, png or bmp) of a part, you can open it with Mecway and it will be displayed in the background on one of the 3 coordinate planes so you can manually place nodes in

the correct locations. The image file will be linked to the model rather than embedded in it. If you move the liml file, you should move the images with it.

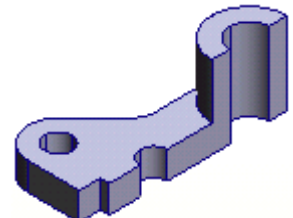
For example, to trace a 3D outline from a 3-view orthographic projection drawing, import (**File** → **Import**) an image file containing each view and place it in a different orientation by choosing **XY plane**, **YZ plane** or **ZX plane** in the dialog box that appears. If an image is misaligned, you can adjust its position by right clicking the file name in the outline tree and entering a distance for **Horizontal offset** (rightward) or **Vertical offset** (upward). After the images are set up, switch to an orthogonal view by clicking an arrowhead of the orientation triad and use the **New element** or **New node** tool to place nodes by mouse clicks.



## 4.6 Symmetry



If the geometry, loads & constraints are symmetric (*mirror symmetry*), a model size can be reduced to half or quarter.



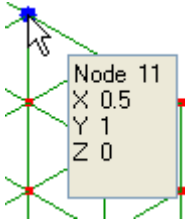
When taking advantage of *mirror symmetry* you must enforce constraints at the plane of symmetry. In static analysis the nodes in the plane of symmetry must be constrained so that they do not move out of that plane otherwise a gap or penetration will occur which in reality is not present in the full model. For elements with rotational degrees of freedom like shells and beams, each node that lies in the plane of symmetry should be constrained to have no rotation about either of the two axes that also lie in the plane of symmetry. In thermal analysis there should be no heat flow across a plane of symmetry, which is a condition that is automatically enforced where no other boundary conditions are specified. The same concept extends to DC current flow analysis.

Take care when assuming mirror symmetry for modal vibration or buckling problems because non-symmetric modes will be missed.

*Cyclic symmetry* which occurs in turbines, fans, etc can be taken advantage of by modeling only a segment containing the cyclic feature rather than the whole wheel. The node patterns must match on both sides of the segment.

## 4.7 Mesh Information

### 4.7.1 Nodal co-ordinates



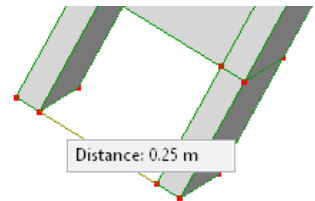
Click a node for a readout of its co-ordinates.

### 4.7.2 Length



Activate the tape measure tool, click and drag from one node to another.

In the solution where displacement field values exist, it also reports the deformed distance by adding the displacements to the node coordinates.



### 4.7.3 Surface area

**Tools** → **Surface area** reports the area of selected faces. It also reports the area after deformation if the solution is displayed, contains displacement field variables, and is not a mode shape.

### 4.7.4 Volume

**Tools** → **Volume** will report the volume of selected elements. If no elements are selected, it will report the full mesh volume. It also reports the volume after deformation if the solution is displayed, contains displacement field variables, and is not a mode shape.

### 4.7.5 Mass

**Tools** → **Mass** calculates the mass of the model or selected nodes or elements. It includes both density and point mass.

### 4.7.6 Center of mass

**Tools** → **Center of mass** displays the center of mass (center of gravity) of the model or selected nodes or elements. It includes both density and point mass.

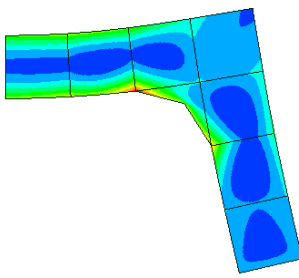
### 4.7.7 Find duplicate elements

Sometimes two or more elements share the same nodes so they appear as one and cause unexpectedly high stiffness or other errors. **Tools** → **Find duplicate elements** finds elements that mutually have all their corner nodes belonging to another element. It selects all the additional elements so you can see where they are and delete them with the **Del** key.

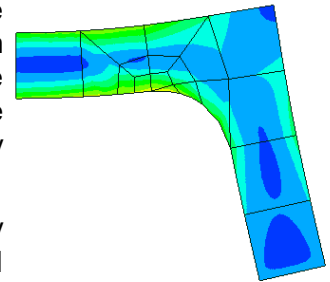
## 4.8 Modeling Errors

Results can only be as accurate as your model. Use rough estimates from hand calculations, experiment or experience to check whether or not the results are reasonable. If the results are not as expected, your model may have serious errors which need to be identified.

#### 4.8.1 Too coarse a mesh



A finite element solution gets more accurate as the mesh is refined. An efficient way of refining a mesh is to concentrate the mesh refinement in those areas where the accuracy can be improved, while leaving unchanged those areas that are already accurate.



You will have to run at least one model to identify the areas where the values are changing a lot and the areas where values are remaining more or less the same. The second run will be your refined model.

Refine areas that see large changes in value. Do not refine areas where values are more or less the same; it will only bloat the size of the model.

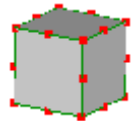
#### 4.8.2 Wrong choice of elements

Bending problems with plate-like geometries such as walls, where the thickness is less in comparison to its other dimensions, should be modeled with either shell elements or quadratic solid elements like the 20 node hexahedron or the 10 node tetrahedron. Shell, beam and membrane elements should not be used where their simplified assumptions do not apply. For example beams that are too thick, membranes that are too thick for plane stress and too thin for plane strain, or shells that are initially twisted out of their plane. In each of these cases solid elements should be used.

#### 4.8.3 Linear elements



Linear elements (elements with no mid-side nodes) are too stiff in bending so they typically have to be refined more than quadratic elements (elements with mid-side nodes) for results to converge.

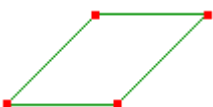


#### 4.8.4 Severely distorted elements

Element shapes that are compact and regular give the greatest accuracy. The ideal triangle is equilateral, the ideal quadrilateral is square, the ideal hexahedron is a cube, etc. Distortions tend to reduce accuracy by making the element stiffer than it would be otherwise, usually degrading stresses more than displacements. However, slight to moderate distortions do not have an appreciable effect on accuracy. The reality is that shape distortions will occur in FE modeling because it is quite impossible to represent structural geometry with perfectly shaped elements. Any deterioration in accuracy will only be in the vicinity of the badly shaped elements and will not propagate through the model (St. Venant's principle).



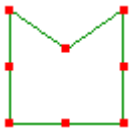
Avoid large aspect ratios. A length to breadth ratio of generally not more than 3.



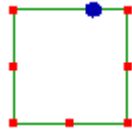
Highly skewed. A skewed angle of generally not more than 30 degrees.



A quadrilateral should not look almost like a triangle.

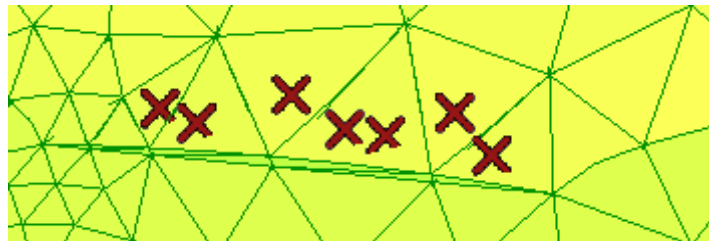


Avoid strongly curved sides in quadratic elements.



Off center mid-side nodes.

If an element is too severely distorted to solve, it will usually be shown with a red cross in it. This happens if it is inverted, self-intersecting, or collapsed.

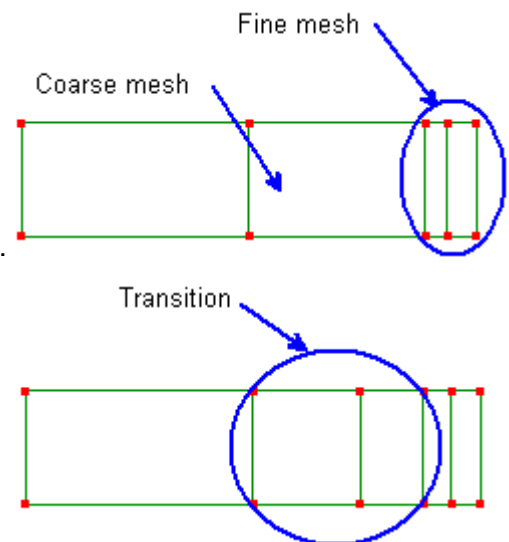


If these bad elements appear after using the automesher, they are usually quadratic elements with edges that are too steeply curved. Two solutions are:

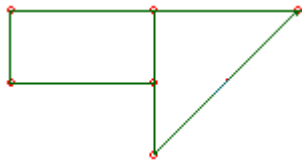
- If you're meshing a STEP file, increase the **Min. number of elements per curve** under **Geometry** → **file name** → **Meshing Parameters**. Then generate the mesh again. A value of 1.5 or 2 may be sufficient but can also increase the total number of elements.
- Use linear elements (ie tet4) then convert them to quadratic elements (ie tet10) if needed using **Mesh tools** → **Change element shape**. This will straighten all the edges.

#### 4.8.5 Mesh discontinuities

Element sizes should not change abruptly from fine to coarse. Rather they should make the transition gradually.

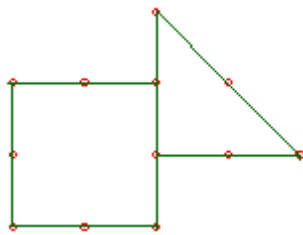
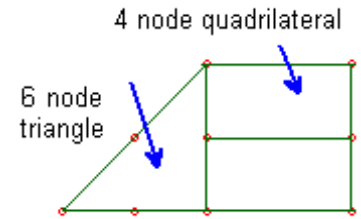






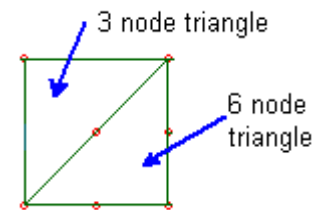
Nodes cannot be connected to element edges. Such arrangements will result in gaps and penetrations that do not occur in reality.

Linear elements (no mid-side node) should not be connected to the mid-side nodes of quadratic elements, because the edge of the quadratic element deforms quadratically whereas the edges of the linear element deform linearly.



Corner nodes of quadratic elements should not be connected to mid-side nodes. Although both edges deform quadratically, they are not deflecting in sync with each other.

Avoid using linear elements with quadratic elements as the mid side node will open a gap or penetrate the linear element.



None of these is a fatal error. Each will simply cause discontinuities in the results which should not be mistaken as being present in the actual part. These effects will be localized and not propagate through the mesh. You can also use bonded contact to connect incompatible meshes without causing the problems described above.

#### 4.8.6 Improper constraints

Fixed supports will result in less deformation than simple supports which permit material to move within the plane of support.

#### 4.8.7 Rigid body motion

In static analysis, for a structure to be stressed all rigid body motion must be eliminated. For 2D problems there are two translational (along the X- & Y-axes) and one rotational (about the Z-axis) rigid body motions. For 3D problems there are three translational (along the X-, Y- & Z-axes) and three rotational (about the X-,Y- & Z-axes) rigid body motions.

Rigid body motion can be eliminated by applying constraints such as **fixed support**, **displacement** and **node rotation**.

Modal vibration and dynamic response do not need to have all their rigid body motions eliminated. However the first few modes would be rigid body modes. For example, if you don't apply any



constraints in a 3D modal vibration problem then the first 6 modes would be for the 6 rigid body motions. The 7th mode onwards would be the structure's deformation modes.

## CAD Models

### 5.1 Introduction

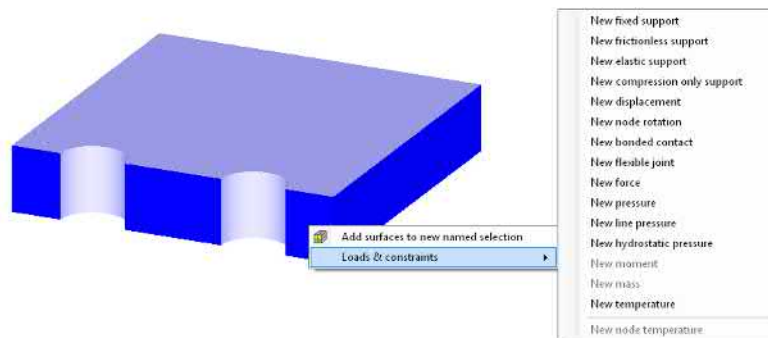
Mecway can open STEP files which can be output by most CAD applications. It displays the parts and can generate a mesh of them (automesh). Links to CAD models appear in the **Geometry** group in the outline tree. Each geometry item in this group must contain a single part so you must split assemblies up into a separate file for each part. If you want to add a CAD file without replacing the currently open model, use **File** → **Import** instead of **Open**.

STL (stereolithography) format files can also be opened and saved by Mecway. An STL file only contains a set of triangles so these are imported as tri3 elements in Mecway without any automeshing. Typically, STL files generated by CAD applications contain highly distorted elements so you should use **Automesh 3D** from the **Mesh tools** menu to improve the shape and convert the shells into a solid object.

### 5.2 Loads and Constraints

For imported STEP files, you should apply loads and constraints to surfaces of the geometry rather than the mesh so that they are linked to the geometry and you can regenerate the mesh without losing them. To do this, change to the geometry view by clicking **Geometry** in the outline tree, then right click a surface and choose **Loads & Constraints**.

To select multiple surfaces for a single load or constraint, hold the **Ctrl** key and click the surfaces.

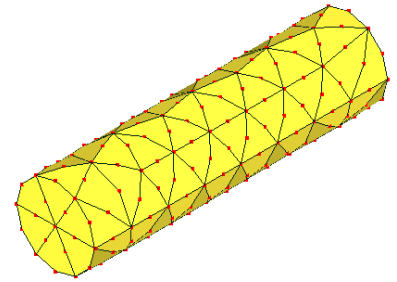
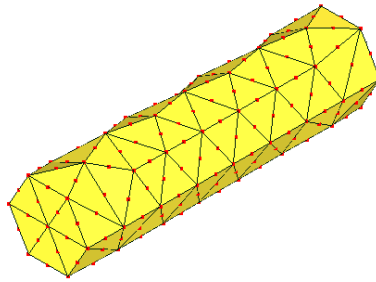


### 5.3 Meshing

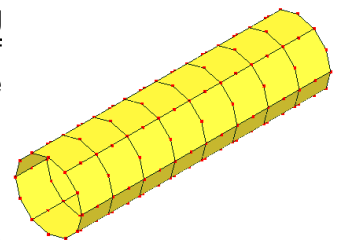
**Meshing parameters** allow local or global refinement by limiting the maximum element size within spherical regions or over the whole geometry. The size gradient of elements can also be controlled. An aggressive size gradient means each element can be much larger or smaller than its immediate

neighbors leading to a low mesh density in large featureless regions and a high density near small details. A gradual size gradient means each element must be a similar size to its immediate neighbors.

The **Fit midside nodes to geometry** option causes the midside nodes of quadratic elements to be located on the surface of the geometry rather than at the midpoints of the corner nodes. This typically improves accuracy but sometimes the midside node will be located too far from the midpoint and lead to a bad element that gives erroneous high stress or fails to solve.



To generate a shell mesh, set the **Surface mesh** option under **Meshing parameters**. If the STEP file contains only surfaces, this can work directly. If it contains a solid body, this will produce shell elements in the shape of the solid body's surface.



## 5.4 Gmsh

You can use the open source Gmsh mesher from within Mecway. This is often faster and can mesh some files that the built-in Netgen mesher cannot. Download and install Gmsh, then set the path to it in **Tools** → **Options** → **Gmsh**. You can then use it by selecting the **Gmsh** option in **Meshing parameters**.

## 5.5 Assemblies

Only a single object is allowed in each CAD file. To model an assembly, import each part as a separate file then use a bonded contact constraint to join their adjacent surfaces. When opening the files with **File** -> **Import** or **Geometry** -> **Import STEP file**, you can select multiple files at the same time by holding **Ctrl** while selecting them.

## Analysis Types



The analysis type determines what physical phenomena are modeled. Mecway starts up with Static 3D as the default. Double click **Analysis** or right click it and select **Analysis settings** to switch to another type of analysis such as thermal or modal vibration.

### 6.1 Static

Static analysis finds the steady state deformation and stress in a structure whose material has a linear stress-strain relationship.

For 2D and 3D, it can also model electrostatic fields in insulating dielectric materials. In this case, each node has an electric potential DOF so electric potential should be constrained at at least one node to ensure a unique solution. The solver then uses the electric potential field value to obtain the electric field and flux density.

#### 6.1.1 Static 2D

In Static 2D analysis, all nodes should lie in the XY plane because the Z coordinates are ignored by the solver. Each node has either 2 or 3 DOFs: Nodes of beams have displacement in X, displacement in Y and rotation about Z while nodes of plane, truss and spring elements have only the two displacement DOFs.

#### 6.1.2 Static 3D

In Static 3D analysis, solid, truss and spring elements have 3 DOFs on each node: displacement in X, Y and Z. Shells and beams have 6 DOFs on each node: 3 displacements and also rotation about X, Y and Z. You can combine all the different element types in the same model.

#### Piezoelectric effect

Solid elements can incorporate the piezoelectric effect in static 3D analysis. To use it, you must specify 3 material properties:

- *Elasticity* – either as a 6x6 material stiffness matrix, a 6x6 compliance matrix, 9 orthotropic constants or isotropic Young's modulus and Poisson ratio. These are measured at constant electric field (short circuit).
- *Permittivity* – either a 3x3 matrix, 3 orthotropic constants or 1 isotropic constant. They can be measured at either constant strain or constant stress.
- *Piezoelectric coupling constants* – either a 3x6 stress matrix (e) or 3x6 strain matrix (d).

Matrices with shear terms are in Voigt form, ordered using the popular mapping from tensor indices 11,22,33,23,31,12 to matrix indices 1,2,3,4,5,6 respectively. Caution: Some other software uses different ordering conventions so you may need to swap rows and columns if transcribing data from them.

The piezoelectric effect cannot be used in the same model as thermal stress or cyclic symmetry.

### 6.1.3 Static Axisymmetric

Only plane elements can be used here and they will be treated as axisymmetric elements. The Y axis is the axis of symmetry and the X axis is the radial direction. Each node must lie in the two positive X quadrants of the XY plane and have zero Z coordinates.

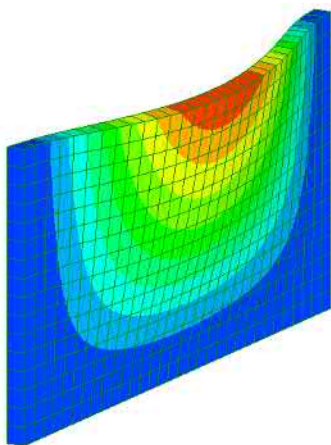
All nodes have two DOFs: displacement in X and displacement in Y.

## 6.2 Nonlinear Static 3D

### 6.2.1 Capabilities

Nonlinear static analysis can model more complex behavior than linear static analysis. It allows displacement to be a nonlinear function of load whereas in linear analysis, the displacement of each node is always in a straight line and proportional to the applied load. This greater capability comes at the expense of a longer solving time. It is only available with the CCX solver. Some examples of where nonlinear analysis is useful are:

**Large displacements and rotations.** For example, a thin strip can be rolled up into a circular shape or members of a mechanical linkage can rotate through various configurations.

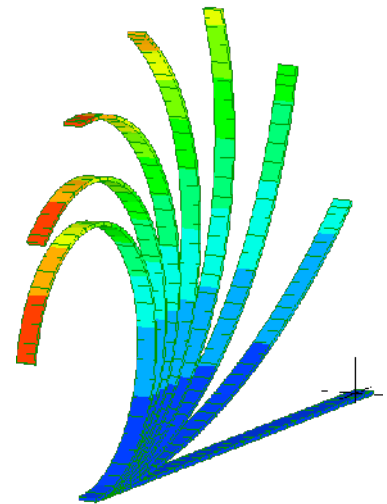


**Membrane action.** An initially flat plate fixed at the edges and bearing a pressure load will deform to a curved shape which becomes stiffer due to membrane stress in the plate.

**Contact between parts.**

**Plastic and hyperelastic materials.**

**Preloaded structures.** For example, a vertical beam hanging under the influence of gravity is in tension. This gives it a greater lateral stiffness than the same beam oriented horizontally.



**Cables.** A cable or rope sagging under the effect of gravity has a bending stiffness which is a function of its tensile force and almost independent of its material properties. This is a similar concept to membrane action. A cable can also have zero or near-zero stiffness in compression as it buckles but a large stiffness in tension. If you use a string of truss elements to model a cable, it must be initially straight and under tension to prevent rigid body motion.

**Buckling.** More general buckling cases are possible than with linear buckling analysis. For example an offset load on an Euler column or an initially bent structure may experience a combination of bending and buckling. A structure may also experience large displacements or rotations before

reaching a configuration in which it becomes unstable and then buckles. Mecway with CCX cannot model the post-buckling response of snap-through buckling.

**Deformation dependent loading (follower loads).** For example, a pressure load applies a constant normal pressure which means the total force changes if surface area or orientation changes.

**Time dependent loading.** Loads can change with time or be turned on and off at different times.

### 6.2.2 Usage

No special settings are needed to use nonlinear analysis in Mecway. Just change the analysis type under **Analysis** → **Edit** and select the **CCX** solver.

If you want time-dependent loads, check the **Quasi-static** box and specify the **Time period** and **Time step** size under **Analysis** → **Edit**. Time here is a pseudo variable which is only used to control the sequence of applied loads. No time dependent effects like inertia or creep are being modeled.

With **Quasi-static** turned off, most loads are automatically ramped up from zero and only the final solution with the full load is shown in the post-processor. With **Quasi-static** turned on, loads that are specified as constant are not ramped and you must explicitly define any required time dependence using formulas or tables.

### 6.2.3 Convergence Failures

When the solver fails to converge on a solution, there are several possible courses of action:

- Ensure there are sufficient constraints to prevent rigid body motion at every time step. Contacts may be initially open so don't rely on them to act as constraints against rigid body motion. For parts that are only supported by contact, add artificial weak springs to support them gently until the contacts close. You can do this using elastic support.
- You can gain insight into how failures occur by turning on **Quasi-static** and defining the loads as formulas that are functions of time ( $t$ ) so that they ramp up gradually over the time period. That will show the converged solution up to the load at which it fails and you may see an unexpected displacement.
- If the model contains many nonlinear features, temporarily remove or suppress some of them to isolate what's causing the problem.
- Turn on **Automatic time stepping** in **Analysis settings**.

## 6.3 Modal Vibration

Free vibrations of a structure occur due to its own elastic properties when it is disturbed from its equilibrium state. These vibrations only occur at discrete natural frequencies. The two properties required for vibrational motion are:

- elasticity which returns the disturbed structure back to its equilibrium state, and
- inertia (from the mass) of the structure which makes it overshoot its equilibrium state.

Modal vibration analysis finds the natural frequencies of a structure and the corresponding deflected shapes (mode shapes). This is done without regard to how the vibration was initiated. All the nodes move with simple harmonic motion in phase with one another at the same frequency. Therefore all the time-dependent displacements reach their maximum magnitudes at the same instant of time.

The magnitudes of all the solution field variables are only relative to the other values in the same mode. Their absolute magnitude has no meaning.

The maximum number of modes is equal to the number of unconstrained DOF in the model. For example, if a model is a single hex8 element with a fixed support applied to one face, the maximum number of modes will be 12, which is the number of DOFs per node (3) multiplied by the number of unconstrained nodes (4). Unless there is shock loading, only the modes of the lowest frequencies are important in the structural response.

### 6.3.1 Modal Vibration 2D

With this analysis type, only two-dimensional elements can be used. Membrane elements must be in plane stress and are appropriate for finding the in-plane vibration modes of a thin sheet while ignoring the out-of-plane modes. Beam, truss and spring elements can be connected to the membrane elements or used on their own.

The model must be made in the XY plane. All Z-coordinates of nodes are ignored by the solver.

### 6.3.2 Modal Vibration 3D

Allows 3D elements such as solids, shells, and beams.

#### Stress stiffening

The stress state of a structure can influence its stiffness and vibration modes. This effect, called stress stiffening, is particularly apparent in a tensioned cable or guitar string.

When any loads are present, Mecway first performs a linear static analysis to determine the stress state then proceeds with the modal vibration analysis using a modified stiffness matrix.

Tensile stress tends to stiffen a structure and increase the natural frequencies. Compressive stress tends to reduce the frequencies. If compressive stress is too high, it may exceed the buckling load of a structure. In these cases Mecway may produce non-physical modes with zero frequencies.

The following features are not allowed with stress stiffening loads:

- Internal solver: Shells using laminate material, beams
- Both solvers: non-zero displacement constraints, non-zero rotation constraints, stress field variable in the solution

## 6.4 Dynamic Response

When a part or structure is subjected to a time varying load, it's stresses are amplified by an induced vibration. Dynamic response analysis takes this vibration into account when calculating the stresses & strains. It also calculates the velocities & accelerations in the model's response to the vibrating load.

Default initial conditions are zero displacement and velocity. You can impose an initial acceleration by applying a load at time zero. With the CCX solver, you can specify initial displacements.

The **Time step** affects the accuracy of the solution with smaller time step sizes being more accurate. You can choose a suitable time step size by first performing a modal vibration analysis to determine the period ( $1/f$ ) of the highest mode of interest, then starting from that value, repeatedly reduce it and solve the problem again until the solution doesn't change significantly. The total **Time period** should be at least the period of the lowest vibration mode. This ensures that all modes oscillate at least once.

Two solvers are available – the default is the Newmark method with constant average acceleration which is unconditionally stable. You can also choose **Mode superposition** which first finds the natural frequencies and mode shapes by solving an eigenvalue problem then generates nodal displacements, rotations and velocities at each time step. There are several other differences of the mode superposition option:

- You must choose the number of modes to include. Typically only the first few modes are important but for shock loading up to 2/3 of them may be needed. Too many modes can slow down the solver and cause it to run out of memory.
- It can be much faster when a large number of time steps are required.
- It is not available in the Axisymmetric analysis type.
- Damper elements cannot be used.
- Any Rayleigh damping must be applied uniformly to the whole model rather than selected elements, and must have a damping ratio less than 1 for every mode that's solved for.
- Rigid body motion is not allowed so the model must be fully constrained.
- Accelerations are not included in the results.

To reduce the number of time steps stored in the results, specify an integer value for **Keep solution every \_\_ time steps**. This reduces memory use and file size.

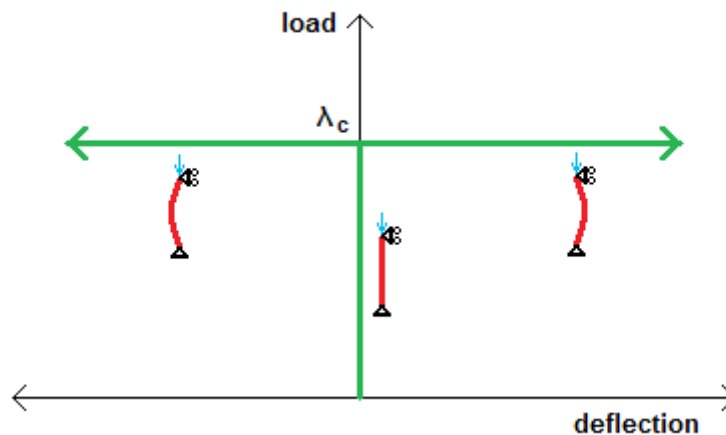
## 6.5 Nonlinear Dynamic Response 3D

This is similar to Dynamic Response 3D but it allows nonlinear behavior such as large displacements, contact, and plastic and hyperelastic materials. It can only be used with the CCX solver.

## 6.6 Buckling

Linear eigenvalue buckling analysis, such as that provided with Mecway, is only capable of describing bifurcation buckling with a constant, symmetric load-deflection relationship as shown below. An Euler column is used as an example, but the same curve can be applied to other structures. Here, deflection is the displacement perpendicular to the direction of the load. Symmetric means the structure must be equally able to buckle in two opposite directions. There should also be negligible displacement in any direction prior to buckling. If these conditions are not met, you should use nonlinear static analysis instead.





As the load is increased from 0 to the critical load  $\lambda_c$ , the structure remains in its original configuration with no deflection. When the load reaches  $\lambda_c$ , the deflection is indeterminate and increases with no further increase of the load.

Some real structures closely approximate this behavior, while others are so different that eigenvalue buckling analysis is of no use. You should take care to ensure that these assumptions are appropriate to the problem otherwise the buckling factors may be grossly in error even if the mode shapes are reasonable.

An important class of problems for which eigenvalue buckling analysis is usually unsuitable is limit point instability. Here, the structure continuously deflects by a finite amount as load is increased, until a 'limit point' of the load is reached, where it 'snaps through' into a different configuration. An example is a toggle mechanism.

These and other structures which appear to be buckling are in fact general non-linear problems. Another example is a column with an eccentric axial load. The deflection is non-zero for any finite load and there is no bifurcation point.

Eigenvalue buckling analysis assumes no imperfections in the material or loading. For this reason, it is non-conservative and typically overestimates the actual buckling loads.

Each mode has an associated buckling factor. You can think of this as the safety factor. Instability occurs when all the forces are multiplied by the buckling factor. For thermal loads, instability occurs when each node's temperature is

$$T_{cr} = (T_{node} - T_{reference}) \times \text{buckling factor} + T_{reference}$$

Centrifugal force loads are proportional to the square of the angular velocity. Therefore, the critical angular velocity,  $\omega_{cr}$ , is

$$\omega_{cr} = \omega_{specified} \times \sqrt{(\text{buckling factor})}$$

It can be convenient to specify unit loads in the model so that the buckling factor is equal to the critical load for forces. If some loads are constant, such as gravity, then you may need to perform several iterations to adjust the unknown loads until the buckling factor becomes 1.

The mode shape represents the relative movement of the nodes immediately after buckling occurs. The actual equilibrium shape of a structure after buckling cannot be found using linear eigenvalue buckling analysis.

You must specify the **Number of modes** and a **Shift point** (internal solver only). The shift point controls the stability of the eigenvalue solver. It must not be zero and should be between zero and the lowest buckling factor. The closer it is to the buckling factors, the greater their accuracy. However, modes with buckling factors below the shift point will not be found.

**Buckling 3D** can model global buckling of a truss structure due to elastic deformation of the individual elements. However, it does not consider buckling of individual truss elements. You can calculate these loads from a static analysis using the axial force values and the Euler column buckling formula.

## 6.7 Thermal

Thermal analysis uses a single temperature DOF for each node. The solver computes heat flux from the temperature field. All thermal analysis types are 3D. However, you can make a 2D model using shell elements of any thickness laid in a plane.

### 6.7.1 Thermal Steady State

This finds the equilibrium temperature distribution in a structure after any transients have dissipated.

### 6.7.2 Thermal Transient

Thermal Transient analysis produces a time history of the temperature field through a structure. You can specify time-dependent loads and temperature constraints as well as an initial temperature distribution. For nodes with no initial temperature specified, Mecway applies a default value of 0 K.

To reduce the number of time steps stored in the results, specify an integer value for **Keep solution every \_\_ time steps**. This reduces memory use and file size.

## 6.8 Fluid

Fluid is used to find steady state solutions to the Navier-Stokes equations, which can represent rotational, viscous flow. They are implemented according to (*C. Taylor & T. G. Hughes, Finite Element Programming of The Navier-Stokes Equations*).

The corner nodes of each element have three degrees of freedom  $u, p, v$  :

$u$  = nodal fluid velocity in x-direction

$v$  = nodal fluid velocity in y-direction

$p$  = nodal static pressure.

The midside nodes have only two degrees of freedom,  $u$  and  $v$ . This means that the velocities are interpolated with quadratic shape functions and pressures are interpolated linearly with the shape functions of tri3 and quad4.

Boundary conditions can be velocity or fluid pressure. The walls of a vessel should have both velocity components set to zero to prevent flow through the wall and enforce the no-slip condition.

Exterior faces with no specified boundary conditions are automatically subject to

$$\frac{\partial u}{\partial n} = 0 \quad \text{and} \quad \frac{\partial v}{\partial n} = 0$$

where  $\frac{\partial}{\partial n}$  is the partial derivative with respect to the face normal. For example a face perpendicular to the X axis with no other velocity constraints will have  $\frac{\partial u}{\partial x}=0$  and  $\frac{\partial v}{\partial x}=0$

The solver finishes when the relative change in the Euclidean norms of the nodal field values between subsequent iterations falls below the **Convergence tolerance**. Leaving this blank defaults to near machine precision.

Many cases where the solver fails or doesn't converge can be caused by its inability to represent unsteady flow which may be caused by turbulence or vortex shedding. Sharp changes in boundary conditions or geometry can lead to such failures. Insufficient constraints on both velocity and pressure can also prevent a solution being found.

## 6.9 DC Current Flow

This analysis type finds the electric potential (voltage relative to an arbitrary zero) distribution throughout a structure. This electric potential which is the single DOF at each node, is then used to obtain current density, resistive power loss (Joule heating or Ohmic heating) and current in resistor elements.

Temperature dependent conductivity can be used on all types of elements. You must also specify a temperature on every element that use the temperature dependent material. Piecewise constant temperatures be applied with the **Temperature** load or you can obtain a non-uniform temperature field from thermal analysis by first solving the same model as thermal analysis, then right clicking **Loads & Constraints** and choosing **Transfer temperatures from solution**.

To do thermal analysis with Joule heating, first solve it as DC Current Flow, then change to a thermal analysis type and use **Transfer internal heat generation from solution** under **Loads & Constraints**, then solve the thermal model.

## 6.10 Acoustic Resonance

This analysis type can find vibration modes of fluid in an enclosed cavity such as sound resonance in a room or vehicle cabin. It cannot model openings in the walls of the cavity such as windows or doors. If you need to model an opening in an object, you need to embed the object in a larger 'room' which itself is fully closed. However, you will then need to distinguish the acoustic behavior of the object from that of the enclosing room.

The modes are solutions to the Helmholtz equation

$$\nabla^2 p + (\omega/c)^2 p = 0$$

where  $p$  is the pressure relative to ambient,  $\omega$  is the angular frequency of the mode, and  $c$  the speed of sound in the medium.

Continuum elements represent the medium inside the cavity. Boundary faces without any constraints behave like hard surfaces with the following Neumann boundary condition

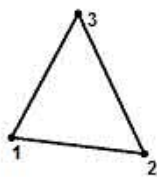
$$\nabla p \cdot \underline{n} = 0$$

where  $\underline{n}$  is the surface normal vector. This means some of the pressure anti-nodes will occur at the boundaries.

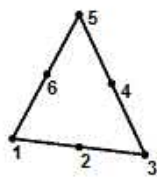
The mode shapes (pressure) shown in the solution are the instantaneous pressure of the standing waves which will oscillate sinusoidally with time. Their amplitudes are arbitrary. The mode frequencies shown in the solution are the angular frequencies  $\omega$ .

## Elements

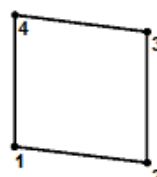
### 7.1 Plane Continuum Elements



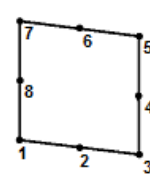
*Tri3*



*Tri6*



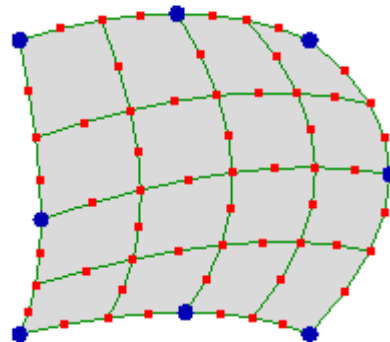
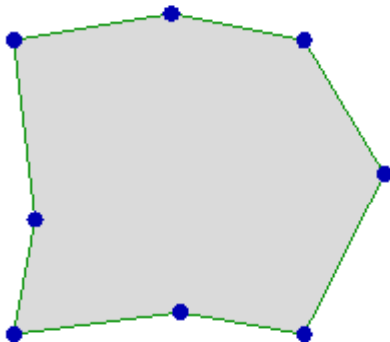
*Quad4*



*Quad8*

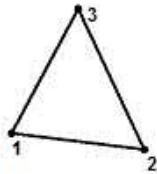
Plane elements can be used for various 2D analysis types to represent structures such as flat plates or prismatic rods. Their nodes should lie in the XY plane.

The quadratic elements (those with midside nodes) can have parabolically curved sides although they are displayed as being straight. You can see the curved shape by refining the element as shown below.

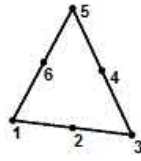


Quadratic elements typically perform better than the linear elements because the DOF field value can vary quadratically across the element whereas the linear elements only allow a linear variation. In mechanical analysis types, the linear elements, especially tri3 (constant strain triangle), have a further limitation of being too stiff in bending. To model bending accurately with linear elements you must refine the mesh so that each individual element experiences mostly tension or compression and less bending.

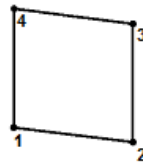
## 7.2 Axisymmetric Continuum Elements



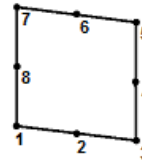
*Tri3*



*Tri6*



*Quad4*



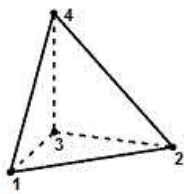
*Quad8*

Although these appear to be the same as plane continuum elements, each axisymmetric element actually represents an entire circular solid as shown below. The axis of symmetry is the Y-axis and elements must be located in the X-Y plane with all X-coordinates non-negative. Axisymmetric elements can only be used in the axisymmetric analysis types.

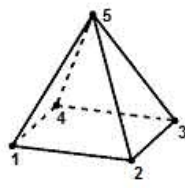


In mechanical analysis types, each node has 2 DOFs: displacement in X(radial) and displacement in Y(axial). Any nodes located at X=0 must be restrained to not be displaced in the X direction because it is physically unreasonable that the material should overlap itself or for a hole to appear. Also, rigid body motion can only occur by translation along the Y-axis, so only translational motion in the Y direction needs to be constrained to prevent rigid body motion.

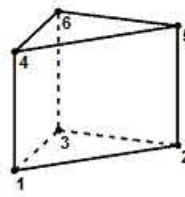
## 7.3 Solid Continuum Elements



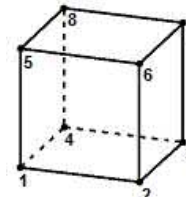
*Tet4*



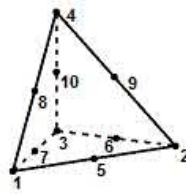
*Pyr5*



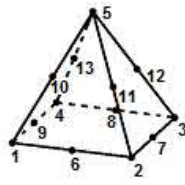
*Wedge6*



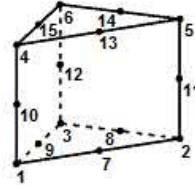
*Hex8*



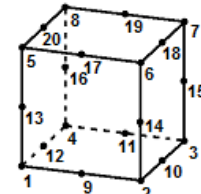
*Tet10*



*Pyr13*



*Wedge15*

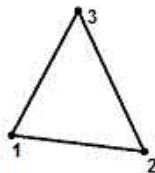


*Hex20*

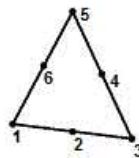
Solid elements are the most general and, in principle, can be used to model any shaped structure. However some geometries such as thin beams or plates can require such a large number of solid elements that the solver runs out of memory or takes too much time. In these cases you can further idealize the model by using shells, beams, fins, resistors, etc. instead of solid elements.

In mechanical analysis types, hex20 typically performs much better than all the other solid elements. This means you attain the same accuracy with fewer elements.

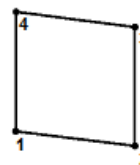
## 7.4 Shell



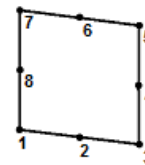
*Tri3*



*Tri6*



*Quad4*



*Quad8*

Shells are 3D elements that can model stress, heat flow or electric current in the plane of the element but not through the thickness. They are useful for thin parts where solid elements are too computationally expensive or in place of 2D elements where those are not available. Typical structures modeled with shell elements include sheet metal brackets and cabinets, thin metal platforms, pressure vessels, and body parts of motor vehicles.

### 7.4.1 Formulation

In the mechanical analysis types, shells are a combination of Mindlin thick plates which model out-of-plane bending and shear stiffness and membrane elements for in-plane deformation. Each node has 6 DOFs – displacement in X, Y and Z and rotation about X, Y and Z. However there is no drilling DOF which means each node is free to rotate about the shell's normal and is only resisted by an arbitrary small stiffness to ensure numerical stability. The tri3 shape is not available for static analysis and only tri6 and quad8 are available in modal vibration, dynamic response and buckling.

The rotational DOFs of shells mean that you can join two shell elements together at a common edge and they will transmit bending moments between each other. This is distinct from solid elements which cannot transmit bending moments when they're only connected by an edge. If the straight edge of a shell is joined to the edge of a solid element, then it will form a hinge joint. To make a stiff joint, use bonded contact.

Shells use consistent mass matrices with fictitious near-zero rotational inertia at the nodes. This simplification typically doesn't cause problems because the effect of nodal rotational inertia vanishes with mesh refinement.

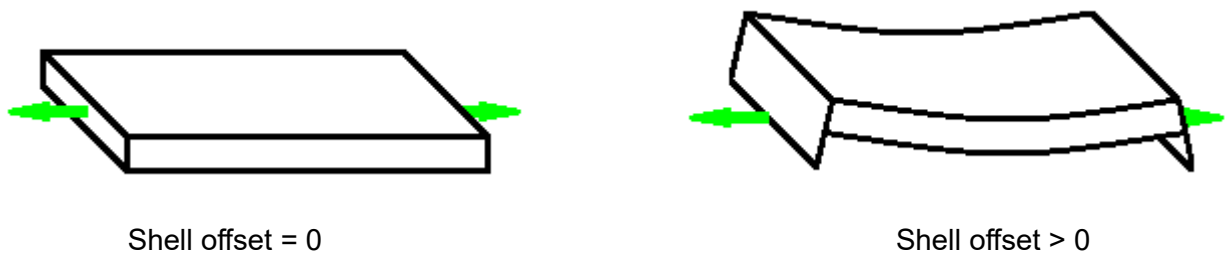
### 7.4.2 Orientation of Local Axes

Each element has a local right-handed coordinate system, U,V,W, which is used for orienting orthotropic and laminate materials and for displaying some stress results. The W axis is normal to the element's surface. By default U is parallel to the edge from nodes 1 to 2, or 1 to 3 for linear or quadratic elements respectively. You can define the direction of the U axis by specifying a vector in global X,Y,Z coordinates. The direction of the U axis is the projection of this vector onto the plane normal to W. The V axis is mutually perpendicular to U and W.

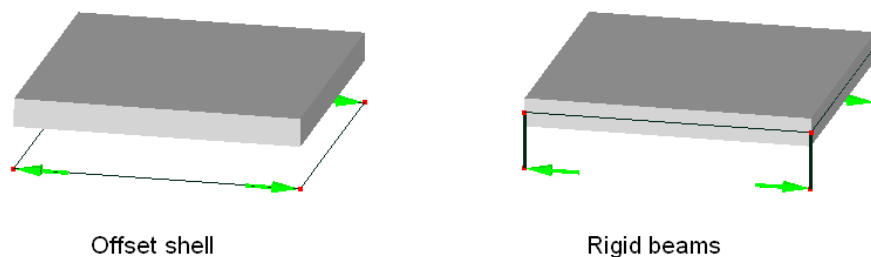
You can specify orientation in two places - **Element properties** and **Loads & Constraints** → **New element orientation**. Specifying orientation through **Loads & Constraints** allows you to enter a formula to define orientation as a function of element center position, and is applied to a selection of elements. Orientation specified through **Element properties** applies to individual elements and is usually not as convenient unless you are importing data with a different orientation for each element.

### 7.4.3 Shell Offset

By default, the nodes of a shell element are located on its midplane. However, in static analysis, you can specify a **Shell offset** in **Element properties** to locate the midplane away from the node locations. This number is a distance the shell is moved in the +W direction in its local coordinate system. Loads are still transmitted to the element at the node locations so extensional forces can result in bending and vice versa.



An offset shell behaves the same as if there were rigid beams connected between the nodes and the shell located at the midplane.



Face loads (force, pressure, line pressure, hydrostatic pressure) and body loads (gravity, centrifugal force) are applied to the offset locations of the faces or elements. This means they will be internally converted to both forces and moments at the nodes. Thermal stress loading is not allowed on offset shells.



Curved offset shells should have an offset distance which is small compared to the radius of curvature. This is because the internal solver rigidly moves each element in the offset direction without changing its size. The CCX solver changes the size by offsetting each node in its local normal direction, which affects loads that depend on surface area or volume.

Constraints on offset shells behave differently with the internal and CCX solvers. The CCX solver constrains the offset positions of the nodes whereas the internal solver constrains the positions they appear at in the modeler.

#### 7.4.4 Curved Elements

Shell elements can be curved out of their plane but should be approximately flat because they are internally projected onto a 2D plane by the solver. Because of this, warped or doubly curved elements in mechanical analysis types can have slow convergence or may fail to converge at all with mesh refinement as shown in the sample TwistedBeam.liml.

#### 7.4.5 Number of Nodes

The best accuracy and fastest convergence is typically obtained with the quad8 shells. Mecway uses various integration schemes for the different physical phenomena in each element shape as shown in the following table.

Number of integration points	tri3	tri6	quad4	quad8
membrane	N/A	7	4	9
bending	N/A	7	4	9
shear	N/A	7	1	4
mass	4	7	4	9
thermal	1	7	4	9
DC current	1	7	4	9

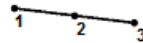
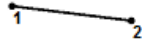
#### 7.4.6 Results

Stress results for shells in static analysis are obtained from bending moments and membrane forces. Out-of-plane shear forces are disregarded. Stresses are not reported for modal vibration.

Mecway reports **von Mises stress**, the two **in-plane principal stresses** and the three in-plane components of **stress in element coordinates** at three locations: the top face, the midplane and the bottom face of each element. The top face is the face with a more positive W coordinate. If you select the stress group's node in the outline tree, the display shows all three values simultaneously on their respective faces. Select the top, midplane or bottom branch to show only one.

**Moment per Length U**, **Moment per Length V** and **Moment per Length UV** are moments per unit length for out-of-plane bending and twisting. They are with respect to the local coordinate system of the element. Moment U is bending about the local V axis, and Moment V about the local U axis. Moment UV is the twisting (torsion) moment per unit length.

## 7.5 Beam



*Line2*

*Line3 (CCX)*

Mecway's beam elements use Euler beam theory which means they can model bending of slender, uniform, isotropic beams. They also have longitudinal and torsional stiffness allowing them to act as columns or shafts. Each node has both displacement and rotational DOFs so they can transmit bending and twisting moments between elements.

You can apply loads and constraints to the end faces or the side face.

Predefined cross-sectional shapes or an arbitrary shape (**General section**) can be used depending on the analysis type and solver:

### Internal solver

static 3D modal vibration 3D dynamic response 3D	I, rectangular bar, rectangular tube, circular bar, circular tube, general section
static 2D	C, T, I, rectangular bar, rectangular tube, circular bar, circular tube, general section
modal vibration 2D dynamic response 2D buckling 2D	general section

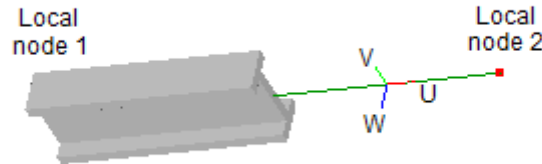
### CCX solver

static 3D nonlinear static 3D modal vibration 3D dynamic response 3D nonlinear dynamic response 3D buckling 3D	rectangular bar, rectangular tube, circular bar, circular tube
---	--

The orientation of the beam cannot be fully defined by the locations of the nodes alone so there are other options to control how the cross section is rotated about the beam's longitudinal axis. You can specify these in two places - **Element properties** and **Loads & Constraints** → **New element**

**orientation.** Specifying orientation through **Loads & Constraints** allows you to enter a formula to define orientation as a function of element center position, and is applied to a selection of elements. Orientation specified through **Element properties** applies to individual elements and is usually not as convenient unless you are importing data with a different orientation for each element.

In 3D, the element's local coordinate system, U, V, W defines the orientation of the cross-section.



The U axis is parallel to the element's length, the V axis is perpendicular to U and the W axis is mutually perpendicular to both U and V according to  $W=U \times V$ . These constraints leave the V axis free to rotate about the element's longitudinal axis. There are two options for defining the direction of the V axis:

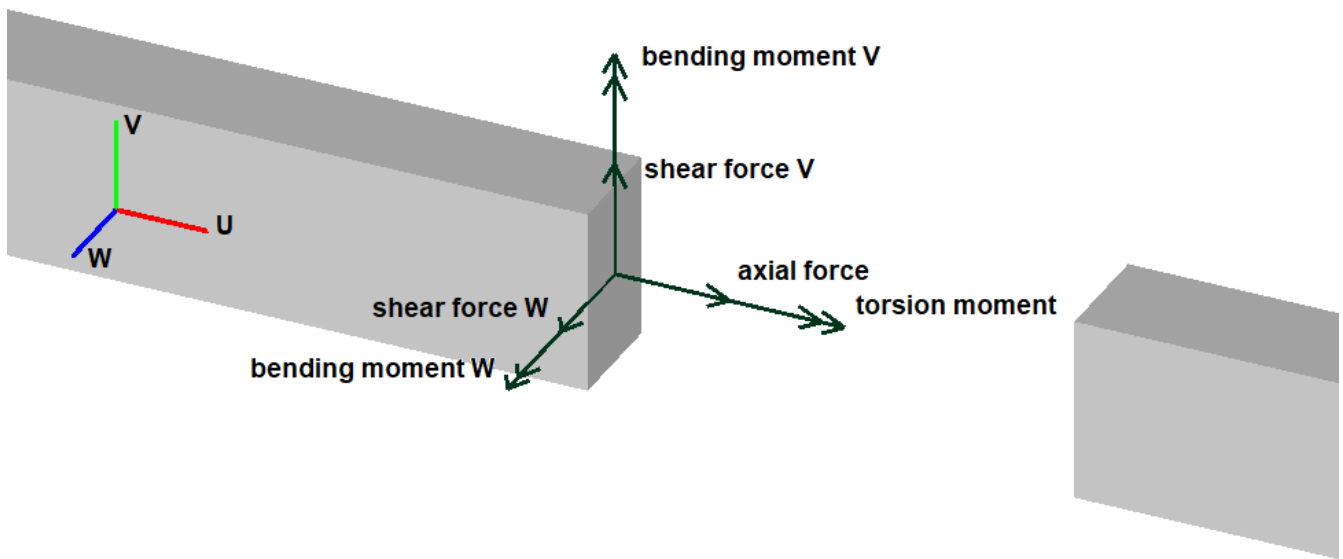
- **Default / none.** The V axis is perpendicular to the global Z axis and parallel to the XY plane. If the element is nearly parallel to the Z axis (gradient < 0.01%) then the V axis is made parallel to the Y axis too.
- **Direction of element V axis.** The V axis can be defined by a vector in the global X, Y, Z coordinate system. The specified vector might not meet the requirement of being perpendicular to U, so the V axis is the projection of this vector onto a plane perpendicular to U. This allows almost any vector to be used as long as it is not zero (0, 0, 0) and not be parallel to U. The magnitude of the vector is not used by Mecway.

After defining the vector you can also apply a further rotation of V about U by a specified angle (**Rotate by angle**). This feature is useful if you know the angle in which you want to orient the beam but don't want to manually convert it to vector form. It can also be helpful if you want to specify a common direction of W for many elements in different orientations. You can set the vector to the direction of W then rotate it by  $-90^\circ$ .

In 2D, the U, V, W element coordinate system is fully defined by the locations of the element's nodes, with W parallel to the global Z axis and in the same direction. You can flip the section upside-down by reversing the element's nodes with **Mesh tools** → **Invert**.

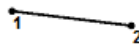
Longitudinal stresses (**stress in element coordinates**) are evaluated at each of 4 predefined points in the cross-section and one user-defined **Stress recovery point in principal coordinates**. These stress values are calculated only from the normal force and bending moments. They do not incorporate shear stress caused by torsion or shear forces.

The solution's internal forces and moments follow the sign convention shown in the diagram below.



The CCX solver (2.17) has some bugs with beam elements. Non-zero displacement constraints on beams can cause incorrect stress, and constrained rotational DOFs can cause incorrect external force. Mecway will show a warning in the outline tree if the conditions for these bugs are met.

## 7.6 Truss



*Line2*

A truss element is a beam hinged at both ends so that it can only resist axial compressive or tensile forces and has no bending or torsional stiffness. Each node has translational degrees of freedom in the X, Y and, if 3D, Z directions.

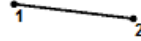
Truss elements can be used to model truss structures where no bending occurs in individual members, as well as to transmit forces between objects and create hinge joints between beam or shell elements.

To convert a beam element into a truss element, select the desired elements, right click on one of them, click **Element properties** and check **Truss**.

Only 3D models can use all the section shapes (C, T, L, I, rectangular bar, rectangular tube, circular bar, circular tube and general section) for trusses. Static 2D disallows L section, and in other analysis types only the general section can be used. No orientation needs to be specified like for beam elements.

The results in static and dynamic response show the axial force and stress at each end of each element.

## 7.7 Spring



### *Line2*

The spring element has stiffness only in the direction of its length and is hinged at its endpoints. It can be used anywhere that a truss element can. Each node has 2 or 3 translational degrees of freedom along the X, Y and Z axes depending on the number of dimensions (2D or 3D).

You can define a spring by applying the **Spring** mechanical material type to a line2 element.

The **Spring constant**  $k$  is the stiffness in Hooke's law

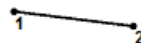
$$F = -k u$$

where  $F$  is the extending force exerted by the spring and  $u$  is the increase in length from the free length.

A spring element behaves identically to a truss element with zero density. The equivalent truss element satisfies the equation

$$\text{cross-sectional area} \times \text{Young's modulus} = \text{spring constant} \times \text{length}$$

## 7.8 Damper



### *Line2*

The damper element has damping only in the direction of its length and is hinged at its endpoints. Each node has 2 or 3 translational degrees of freedom along the X, Y and Z axes depending on the number of dimensions (2D or 3D).

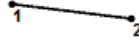
You can define a damper by applying the **Damper** mechanical material type to a line2 element.

**Viscous damping coefficient** ( $c$ ) causes a damping force of

$$F = -c \dot{u}$$

where  $\dot{u}$  is the relative axial velocity of the two nodes.

## 7.9 Tension Only

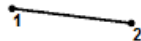


*Line2*

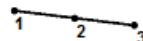
The tension only element is a truss element which has a specified stiffness when loaded in tension but zero stiffness when compressed. It can only be used in the Static 3D analysis type where it causes an iterative solution which may take longer than solving without tension only elements.

To define a tension only element, apply the **Tension only** mechanical material type to a line2 element and choose one of the section shapes in the geometric material types. You must also identify the element as a truss. To do this, select the desired elements, right click on one of them, click **Element properties** and check **Truss**.

## 7.10 Fin



*Line2*



*Line3*

The rate of heat removed by convection from a surface is increased by increasing the surface area for heat transfer using extended surfaces called fins. A simple example is a metal spoon placed in a glass of hot water, heat is conducted through the spoon to cause the handle to become warmer than the surrounding air. Heat is then transferred from the spoon handle to the air by convection. Similarly, industrial applications use fins in car radiators, double pipe heat exchangers, electronic equipment and compressors.

A fin element models heat conduction along its length but not across its width. It can also have convection and radiation heat transfer applied to its side face and/or end faces.

## 7.11 Resistor



### *Line2*

A resistor element has a linear change in potential over its length according to Ohm's law  $V=IR$ . Its resistance can be defined by either using the material's **Resistor** property **Resistance**, or by its conductivity together with the geometry of a beam. Any of the uniform section shapes (C, T, L, I, rectangular bar, rectangular tube, circular bar, circular tube, general section) can be used. Resistors can be useful for modeling linear electric circuits as well as for measuring current values in 3D continuum or shell models. To find the current flowing through a complex object, you can place a resistor element with near-zero resistance in series with the object. The solution will then show the current flowing in the resistor.

## Materials

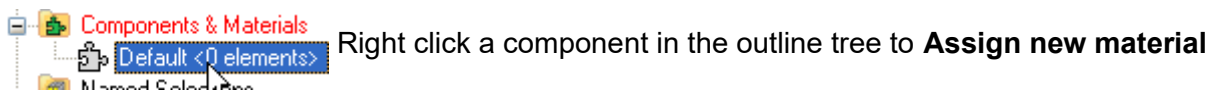
Elements are designed to mathematically represent various types of materials for both their physical properties and their geometric shapes.

### 8.1 Library

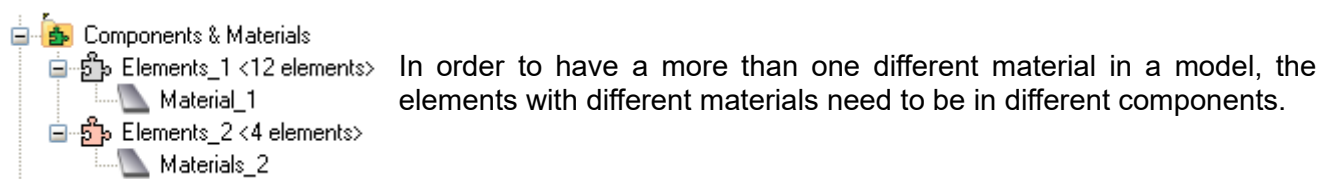
Mecway does not include a materials database but you can build one yourself. Create an empty .liml file with only a material defined in it, then when you need that material, open it using **Import** from the **File** menu. This will bring those materials into the currently open model.

As a convenience for managing such files, you can define a common location in **File** → **Import from library** → **Set library location...** and any .liml files put there will appear on that sub-menu as well as under the **Assign material from library** menu when right clicking on a component. These library files may contain other data such as meshes of common parts or solution formulas. A shortcut to save an individual material to a new library file is the **Save to library...** button in the **Material Properties** window.

### 8.2 Defining a New Material



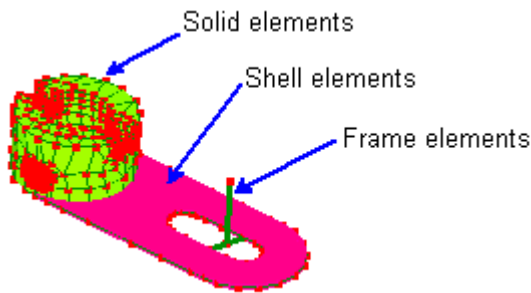
### 8.3 Mixed Materials



### 8.4 Mixed Elements

Element types can be mixed in the same model, but each should have its own material definition because their geometrical properties are not the same.






## 8.5 Anisotropic Materials

Orthotropic materials have mechanical properties that differ along directions which are perpendicular with each other. In Mecway you can use orthotropic materials for 3D solid, 2D membrane and shell elements. On shell elements, one axis (W) is always in the through thickness direction (normal to the surface).

There are two conventions for defining the Poisson's ratios of orthotropic materials. Mecway defines  $\nu_{xy}$  to be the negative ratio of strain in the y direction to strain in the x direction due to stress in the x direction. This is sometimes called a major Poisson's ratio. Some older sources of data transpose the subscripts (minor Poisson's ratio). You can check that Mecway's convention is being used by ensuring that this equation is satisfied:

$$\frac{\nu_{xy}}{\nu_{yx}} = \frac{E_x}{E_y}$$

The directions of the orthotropic axes are the local element axes (U, V, W). For solid elements, these are the same as the global X, Y, Z axes by default. You can change this for each element by defining the local U and V axes in either **Element properties** or **Loads & Constraints** → **New element orientation**. The direction you specify for the U axis is the same direction that the element will use. You must also specify the V axis to fully define the orientation in 3D space. It's often not easy to choose a V axis which is perfectly orthogonal to the U axis so Mecway will obtain the orthogonal V axis by projecting the vector you specify onto the plane normal to the U axis.

The local axes of all elements can be displayed graphically using this tool-button . With red, green and blue lines representing the U, V, and W axes respectively.

Anisotropic materials have mechanical properties that differ along arbitrary directions which are not perpendicular with each other. You can use these with solid elements in the Static 3D analysis type by specifying either a material stiffness matrix or compliance matrix (inverse of material stiffness matrix).

## 8.6 Temperature Dependent Properties

You can specify temperature dependent thermal conductivities and specific heat values. The solver linearly interpolates between the specified data points. If a temperature is outside of the range, the solver uses the nearest data point rather than attempting to extrapolate.

## 8.7 Failure Criteria

If you specify a failure criterion for a material with the static or dynamic response analysis types then the solution will include a factor of safety field variable.

The **von Mises** failure criterion can only be used on 2D continuum, axisymmetric, 3D shell, and 3D solid elements. On 3D shell elements, it ignores out-of-plane shear stress.

The **Tsai-Wu** failure criterion can only be used with 3D shell elements and it assumes plane stress, ignoring out-of-plane shear stress. It requires 6 parameters – tensile strength in two directions, compressive strength in two directions, shear strength, and an interaction coefficient  $F^*_{12}$  between -1 and 1 inclusive. For isotropic and orthotropic materials, the factor of safety is evaluated at both sides of the shell (top and bottom) and only the lowest of these two factors of safety is reported for each node. For laminates, the factor of safety is evaluated at both sides of each layer and the minimum factor of safety at each node is reported (first-ply failure) together with the number of the layer (**First Layer to Fail**) and the position within the layer that it occurs on (**Failure Position in Layer**). A position of 1 means the top side (+W direction) and -1 means the bottom side (-W direction) and these are the only two positions where it is evaluated.

The **Factor of Safety** (strength ratio) reported in the solution is the value which all stress components can be multiplied by to cause the stress state to lie on the failure surface. For example, a value between 0 and 1 indicates failure and a value of 2 indicates that the structure can sustain twice the stress before failing. The factor of safety is also the value which all loads can be multiplied by to cause failure. The scale is limited to a maximum of 15 by default but you can change that in **Tools** → **Options** → **Contour plot** → **Factor of safety scale maximum**.

## 8.8 Nonlinear Materials

Several non-linear materials become available with the Nonlinear Static 3D and Nonlinear Dynamic Response 3D analysis types. They are:

- Neo Hooke

- Mooney-Rivlin

- Plastic with isotropic hardening

- Plastic with kinematic hardening

- Ramberg-Osgood

## Loads and Constraints

To apply a load or constraint, right click the **Loads & constraints** branch of the outline tree and choose what to add. Most loads and constraints need to be applied to nodes, faces, elements or named selections. Some are restricted to only nodes, only faces or other combinations. If you select some of these items before you create the new load or constraint then it will automatically use them by default.

Where loads or constraints are distributed over surfaces, apply them to faces rather than nodes. Mesh refinements will automatically transfer face loads and constraints to the newly created elements, whereas loads and constraints applied to nodes remain at the same point locations.

To temporarily disable a load or constraint so that it is ignored by the solver and not displayed in the graphics area, right click it and choose **Suppress in active configuration**. It's name will then be grayed out until you **Unsuppress in active configuration**.

Some loads and constraints can vary as a function of time. There are two ways to do this - with a table or a formula. Using a **Table**, you specify a table of time and value pairs which can be entered directly or pasted from a spreadsheet. The solver performs a linear interpolation between data points in the table to determine the value at each time step. If a time step is outside the range of times in the table then constant extrapolation is performed so that the final value will be applied at all subsequent times and the first value will be applied at all preceding times. You can also enter a **Formula** with the variable  $t$  to represent time and it will be evaluated at each time step.

For the CCX solver, formulas are converted to piecewise linear functions. It samples each function at 10001 uniformly spaced time values and retains only those points which define a piecewise linear function whose error is no more than 0.01% of the range of the function. Determination of the error and range are made using the 10001 samples.

You can assign a name to each load or constraint. They are sorted by name in the outline tree and a name allows them to be accessed by scripts using the Python API. To set the name, right click the item in the outline tree and choose **Rename**.

You can organize loads and constraints into groups. Groups can be suppressed or collapsed to make working with large numbers of loads and constraints easier. To create a group and place a load or constraint in it, right click it and choose **Move to group** → **New group**.

## 9.1 Fixed support

Analysis types: Static, Nonlinear Static, Modal Vibration, Dynamic Response, Nonlinear Dynamic Response, Buckling

Solvers: Internal, CCX

A fixed support fully constrains the nodes or faces against displacement and rotation in any direction. It can be applied to nodes of beam elements or any element faces except the end faces of truss, spring, and damper elements. Only those DOFs supported by each face or node will be constrained so a fixed support on the face of a solid element won't constrain the rotational DOF of an adjacent shell element.

## 9.2 Frictionless support

Analysis types: Static 3D, Nonlinear Static, Modal Vibration 3D, Dynamic Response 3D, Nonlinear Dynamic Response, Buckling 3D

Solvers: Internal, CCX

A frictionless support constrains displacement in the direction normal to the faces it is applied to. This allows the nodes to slide along the surface but not lift off or penetrate it.

On curved surfaces, the normal direction at each node is the average of the normals of all adjacent faces which the frictionless support is applied to.

On shells and the end faces of beams, frictionless support also constrains node rotation about any of the axes in the plane of the surface.

On the edge faces of beam and truss elements, frictionless support constrains the elements as if they were able to slide inside a tube. They can slide along and rotate about their longitudinal axes, but not move or bend laterally.

In linear static analysis, you can specify a **Normal displacement** which is the distance the surface is displaced. A positive value means displacement in the direction into the surface, as indicated by the red arrowhead constraint symbols.

## 9.3 Pinned support

Analysis types: Static 3D, Nonlinear Static, Modal Vibration 3D, Dynamic Response 3D, Nonlinear Dynamic Response, Buckling 3D

Solvers: Internal, CCX

A pinned support constrains mean displacement to zero in all three directions. This allows the nodes or faces to rotate freely about their centroid.

Faces or multiple nodes with a pinned support cannot have other constraints in some cases.

It can only be applied to faces of solid elements or nodes of any element type.

## 9.4 Elastic support

Analysis types: Static 3D, Nonlinear Static, Modal Vibration 3D, Dynamic Response 3D, Nonlinear Dynamic Response, Buckling 3D

Solvers: Internal, CCX

An elastic support constrains each face through a finite stiffness distributed over the surface. You can independently specify normal and tangential stiffness per unit area. With only a normal stiffness, this is also called a Winkler foundation.

It can only be applied to faces with an area defined by their nodes – that is faces of solid elements and the front and back of shell elements.

## 9.5 Compression only support

Analysis types: Static 3D

Solvers: Internal

A compression only support behaves similar to a frictionless support at nodes where the support is in compression, and applies no constraint to nodes that would put the support in tension. This is a non-linear feature so the solver will perform several iterations to determine which nodes should be constrained and which are free to displace. It stops when there's no change in the status of any node between two successive iterations.

It can only be applied to 2D faces, this includes faces of solid elements and the surfaces of shells.

## 9.6 Displacement

Analysis types: Static, Nonlinear Static, Modal Vibration, Dynamic Response, Nonlinear Dynamic Response, Buckling

Solvers: Internal, CCX

A displacement constraint enforces the specified displacement on every node it is applied to. The displacement is only constrained in the specified direction and the nodes are free to move in perpendicular directions.

A displacement can be applied to nodes, faces or elements.

To constrain displacement in an arbitrary direction, specify the X, Y and Z components of a vector describing the direction. Each node can have any number of constraints in any directions. Mecway will internally reduce them to a minimum equivalent set of orthogonal constraints in a rotated coordinate system.

The direction vector's components can be functions of position. This allows, for example, radial or tangential displacement in a cylindrical coordinate system. With nonlinear analysis types, the position is the initial modeled position of each node, even if they move during the solution.

Displacement must be zero with the internal solver and any analysis type except Static. It can be non-zero and a function of time with the CCX solver.

## 9.7 Node rotation

Analysis types: Static 2D, Static 3D, Nonlinear Static, Dynamic Response 2D, Dynamic Response 3D, Nonlinear Dynamic Response, Modal Vibration, Buckling 2D Beam

Solvers: Internal, CCX

Node rotation fixes the rotation angle of each of its nodes about the specified axis. The angle is measured in degrees. You can only use it on nodes or faces of elements having the corresponding rotational DOFs. This includes beam and shell elements but not solid, plane continuum, truss, spring or damper elements. If you need to constrain rotation of these types of elements, use a pair of displacement constraints instead. For instance, fixing displacement=0 in the Z direction at  $x=-1$  and  $x=+1$  will prevent rotation about the Y axis.

To constrain rotation about an arbitrary axis, specify the X, Y and Z components of a vector describing the direction of the axis. Each node can have any number of constraints about any axes. Mecway will internally reduce them to a minimum equivalent set of orthogonal constraints in a rotated coordinate system.

Node rotation angle must be zero with the internal solver and any analysis type except Static.

## 9.8 Bonded contact

Analysis types: Static 3D, Nonlinear Static, Modal Vibration 3D, Dynamic Response 3D, Nonlinear Dynamic Response, Buckling 3D, Thermal, DC Current Flow, Acoustic Resonance 3D

Solvers: Internal, CCX

Bonded contact joins two sets of faces together. It can be useful for connecting parts of an assembly which have incompatible meshes, for making a transition between a coarse and fine mesh or between linear and quadratic elements or for connecting shells and beams to solid elements without creating a pin joint. The faces in one set (the master) must be 2D such as any face of a 3D solid element or the surface of a shell. The faces in the slave set can be of any type, including the end of a beam or the edge of a shell.

Three formulations are available: default, **\*TIE**, and **Elastic** which have different limitations in each solver as shown in the following table:

	Solid element face	Shell surface	Shell edge (slave only)	Beam end (slave only)	Beam edge (slave only)	Nonlinear large rotations
Default	CCX and internal	internal	internal	internal	internal	none
*TIE	CCX	none	none	none	none	CCX
Elastic	CCX	CCX	CCX	CCX	none	CCX

### Default

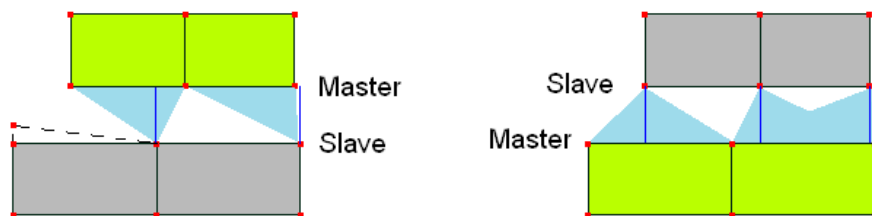
It is implemented as a node-to-face contact using constraint equations. One surface is designated the master surface (also called independent or target surface) and the other the slave surface (also called dependent or contact surface). Gaps and penetrations between the surfaces are allowed. Mecway projects each slave node onto at most one master face. The slave node is then constrained to move and rotate along with its projected point. These constraints introduce no artificial stiffness parallel to the master face and they are rigid in the normal direction which prevents the two surfaces from separating. For other analysis types (thermal, DC current flow, and acoustic resonance), the single DOF of the slave node is constrained to equal the value at the projected point on the master face.

There are several factors to consider when deciding which surface to make the master and which the slave:

- The surface with the coarser mesh should be the master. If the master mesh density is too fine compared to the slave then some element faces of the master surface might not have any slave nodes projected onto them and they will be disconnected.



- The surface with the larger area should be the master. If the slave surface overhangs the edge of the master surface then slave nodes which are too far from the edge won't be connected. However a small tolerance of about 4% of an element face's width in its in-plane direction is allowed. Slave nodes within this distance of the edge of a master face are still connected.



- Other constraints are not allowed on nodes of the slave surface and only some types of constraint are allowed on nodes of the master surface. These should be in all 3 directions such as fixed support. Frictionless support typically can't be used if any of its nodes are part of a bonded contact, however you can use elastic support with a large stiffness to achieve the same effect as frictionless support sharing either master or slave nodes.

To display the slave-master connections, click on the bonded contact in the outline tree. It will show a line from each slave node to the point it connects with, if any, on the master surface.

**\*TIE.** Nodes of the slave surface are moved in a strain-free way so they lie on the master surface. If the surfaces are not already coincident, specify a **Position tolerance** greater than the maximum separation between them. Nodes outside this tolerance are excluded from the bonded contact.

**Elastic.** This creates elastic springs between the surfaces. As with \*TIE, the nodes of the slave surface are moved in a strain-free way so they lie on the master surface.

## 9.9 Contact

Analysis types: Nonlinear Static, Nonlinear Dynamic Response

Solvers: CCX

Contact allows contact between two surfaces to be detected so that they don't pass through each other but they can separate.

Contact uses the recommended most robust options of face-to-face contact with linear contact springs and a default stiffness per unit area chosen according to the adjacent material's Young's modulus. It also allows friction.

**Contact stress**, which includes contact pressure, and **contact displacement** are available in the solution. They only appear on slave nodes and are zero everywhere else. The tangential components of contact displacement are only present if friction is defined, otherwise they appear as zero. The normal component of contact displacement (clearance) is only present where penetration occurs and is zero at slave nodes that do not penetrate the master surface. That means only negative clearances appear.

## 9.10 Node-surface coupling

Analysis types: Static 3D, Nonlinear Static, Modal Vibration 3D, Dynamic Response 3D, Nonlinear Dynamic Response, Buckling 3D

Solvers: CCX

**Node-surface coupling** connects a single node (the reference node) to a surface consisting of element faces. It's useful for applying remote loads, constraints and point mass to an entire surface, and creating a rigid body. Three options are:

**\*RIGID BODY** and **\*KINEMATIC** are mostly the same and fix all nodes of the surface and reference node together so that they behave as a single rigid body with 6 degrees of freedom. Those 6 degrees of freedom are also the 6 DOF of the reference node. While CCX's **\*RIGID BODY** requires a separate node for rotational DOFs (ROT NODE), Mecway automatically generates this node and transfers any rotational constraints and moments to it before solving.

**\*DISTRIBUTING** allows the surface to deform. This option is not recommended because it is unreliable and the location of the reference node is neglected so there is no moment arm between it and the centroid of the surface. Alternatives to \*DISTRIBUTING are **pinned support** or **moment** applied to faces.

Various limitations of each option are listed in the **node-surface coupling** window in Mecway.



## 9.11 Pre-tension section

Analysis types: Static 2D, Static 3D, Nonlinear Static, Modal Vibration 3D, Dynamic Response 3D, Nonlinear Dynamic Response, Buckling 3D

Solvers: Internal, CCX

**Pre-tension section** applies an axial tension or compression to a prismatic mesh made from volume elements (CCX solver only) or a single spring element (internal solver only). A typical application is bolt pre-tension. You can define the load as either a force or a length adjustment:

### Force

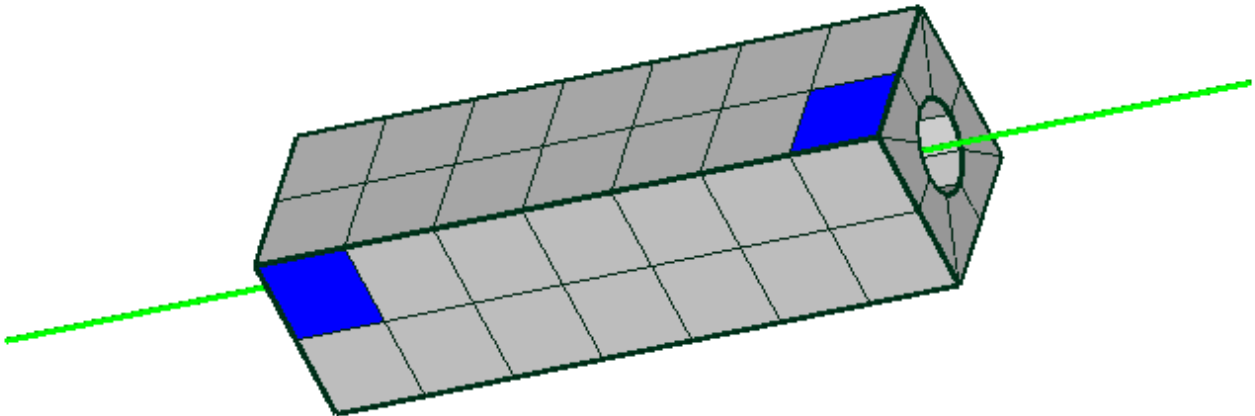
This applies the specified tension (positive) or compression (negative) to the prism. It can be constant or time dependent.

### Length adjustment

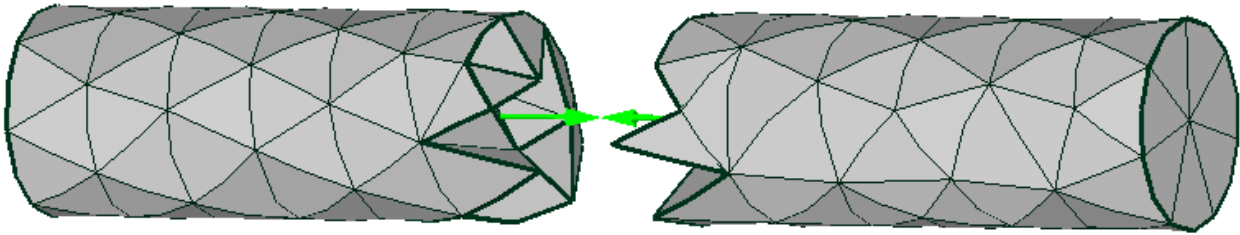
The prism or spring's free length is changed by the specified amount, where a positive value is a shortening. It can be constant or time dependent.

### Prismatic mesh

Apply it to at least two exterior faces oriented at different angles on the surface. These define the orientation of the prism's axis, its extent in the axial direction, and which elements belong to it. The prism's axis is the direction mutually orthogonal to their normal vectors. If you select more than 2 faces, Mecway finds the axes defined by every pair of faces up to a maximum of 1000 faces and uses their median as the axis orientation. It's typically OK to select the entire prismatic surface which can be easier than choosing individual faces. The axial extent of the prism is defined by the maximum axial extent of these faces so you must choose two faces at opposite ends of the region that will contain the pre-tension. The elements within the prism are all the volume elements within its axial extent and connected to any of the selected faces or to any other elements within the prism.



Mecway finds a surface through the cross-section of the prism and CCX uses that to separate the mesh into two parts connected by multi-point constraints. Mecway also inserts a thin layer of hex or wedge elements at the surface to ensure compliance with CCX's requirement that no elements on one part are connected to the surface by only an edge or node.



Limitations include:

- All elements in the prism should belong to the same component, have the same loads and constraints, and have the same element axis orientations specified in Element properties.
- Any loads and constraints on faces or nodes are not extended to the inserted layer of elements. Usually, the error this causes is negligible because the inserted layer is very thin.
- Mecway may be unable to find a surface through the cross-section if the mesh is too irregular. This can happen on unstructured tet meshes with a large number of elements.
- It is not a follower load so if the prism undergoes large rotation, the direction of the pre-tension will not rotate with it.
- Only available with the CCX solver and the following analysis types: Static 3D, Nonlinear Static 3D, Dynamic Response 3D, Nonlinear Dynamic Response 3D.

### Spring element

Apply it to the single edge face of the element. This can only be used with the **Length adjustment** option, the internal solver, and one of the analysis types: Static 2D, Static 3D, Modal Vibration 3D, Buckling 3D.

## 9.12 Flexible joint on beam

Analysis types: Static 3D

Solvers: Internal

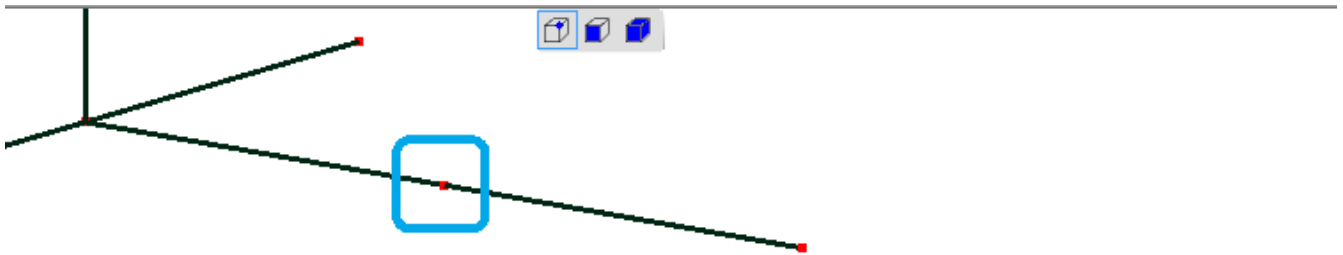
This releases the rotational stiffness of the end of a beam element allowing it to rotate freely even when it's connected to another element or has constrained rotation.

Flexible joint on beam can only be applied to end faces of beam elements.

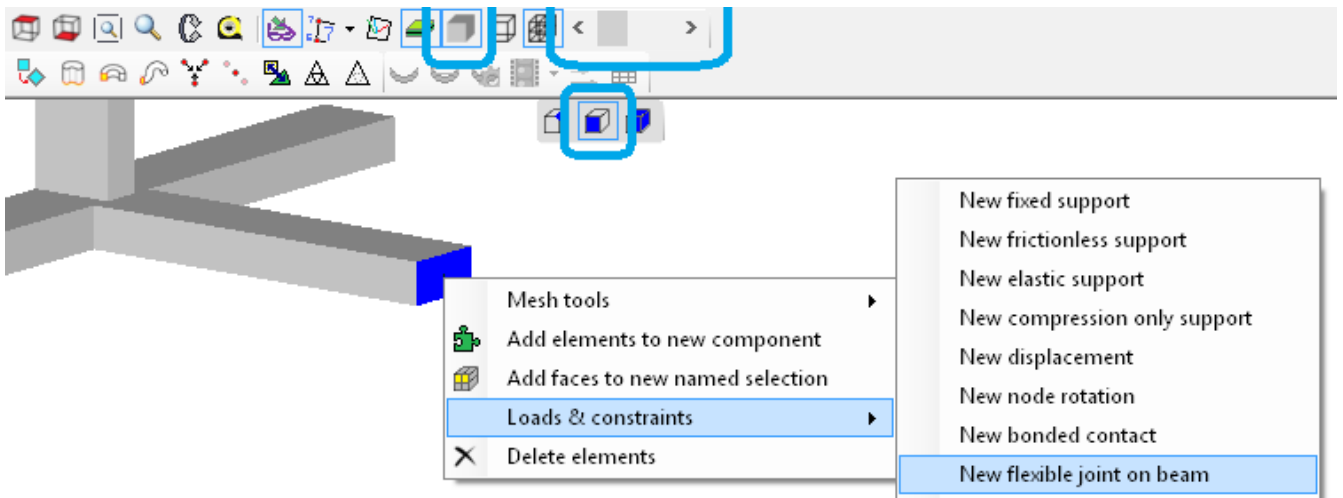
The joint can allow rotation about any combination of the three local axes (U,V,W) of the beam element. The meaning of some combinations are:

- U – The element can rotate about its own axis like being connected to a roller bearing.
- V or W – A pin joint which allows the element to pivot about one axis.
- V and W – A universal joint which transmits torque but not bending moments.
- U, V and W – A ball joint which allows the element to rotate in any direction.

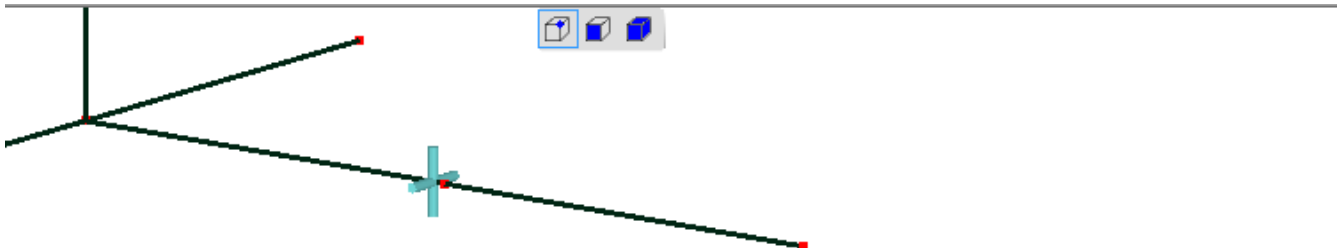
To put a flexible joint at an internal node like this



, turn on **Show element surfaces**, change to **Select faces** mode, and use the **Cutting plane** slider to hide the foreground element(s). Then right click the exposed end face and choose **Loads & Constraints** → **New flexible joint on beam**.



The graphics area then shows pins representing the axes of free rotation.



## 9.13 Force

Analysis types: Static, Nonlinear Static, Modal Vibration 3D, Dynamic Response, Nonlinear Dynamic Response, Buckling

Solvers: Internal, CCX

A force applied to faces or elements is uniformly distributed over them. A force applied to nodes is divided equally between the nodes.

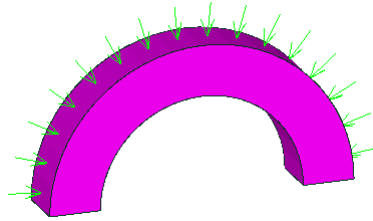
It can be constant or time dependent. The force vector must be specified in global X, Y, Z coordinates.

## 9.14 Pressure

Analysis types: Static, Nonlinear Static, Modal Vibration 3D, Dynamic Response, Nonlinear Dynamic Response, Buckling

Solvers: Internal, CCX

Pressure applies a normal force proportional to the area of the face. A positive value directs the force in the opposite direction to the face normal, which is inwards for volume elements.



A pressure can only be applied to faces. It can be constant or time dependent. It can be used with most types of element used in the mechanical analysis types: plane continuum, solid continuum, axisymmetric continuum, shell, beam and truss but not spring or damper elements because they have no surface area.

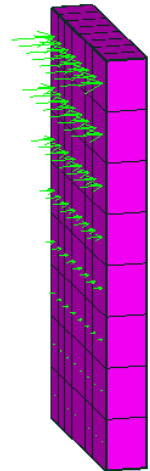
In nonlinear static analysis, pressure is a follower load, meaning that if the orientation or area of the faces changes, the total force will change to maintain constant pressure.

Pressure can also be defined as a mathematical function of position. To do this, enter a formula for its value using the variables  $x$ ,  $y$  and  $z$  for position. For example, to apply a quadratic pressure load to a vertical (+Y) wall, where the pressure varies from 0 psi at  $Y=0$  to 12 psi at  $Y=2$  ft, enter a pressure of:

$$3*y^2$$

and select units of **ft** for position and **lbf/in<sup>2</sup>** for pressure.

If you define pressure with a formula then on each element face, the pressure function is evaluated at 1, 7 or 9 sampling points for line/point, triangle or quadrilateral faces respectively. This means that rapid changes in pressure might not be captured accurately and may need mesh refinement.

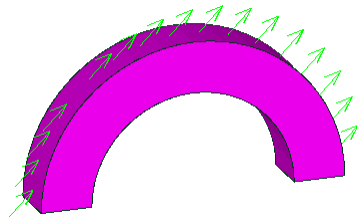


## 9.15 Traction

Analysis types: Static, Nonlinear Static, Modal Vibration 3D, Dynamic Response, Nonlinear Dynamic Response, Buckling

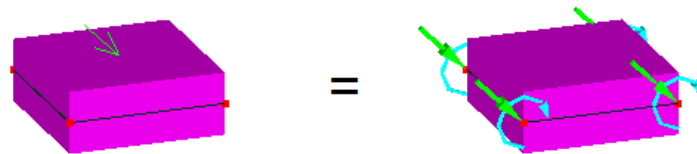
Solvers: Internal, CCX

Traction applies a force proportional to the area of the surface and in a direction specified in global X, Y, Z coordinates.



It can be constant or time dependent and can be used with most types of element used in the mechanical analysis types: plane continuum, solid continuum, axisymmetric continuum, shell, beam and truss but not spring or damper elements because they have no surface area. It can also be defined as a mathematical function of position in the same way as pressure.

On a shell element, the face is offset from the midplane by half the element thickness so a traction applied to the face generates moments at the nodes as well as forces.



If you don't want these moments, apply it to the elements instead of the faces. Then it acts on the midplane. You should always do this for curved shells with the CCX solver because those moments usually include a normal component which isn't allowed by CCX.

## 9.16 Line pressure

Analysis types: Static, Modal Vibration 3D, Dynamic Response, Buckling

Solvers: Internal

Line pressure applies a uniform force per unit length to each face or line2 element. The direction of the force can be specified in either global X, Y, Z coordinates or element U, V, W coordinates when applied to the edge faces of beam or truss elements.

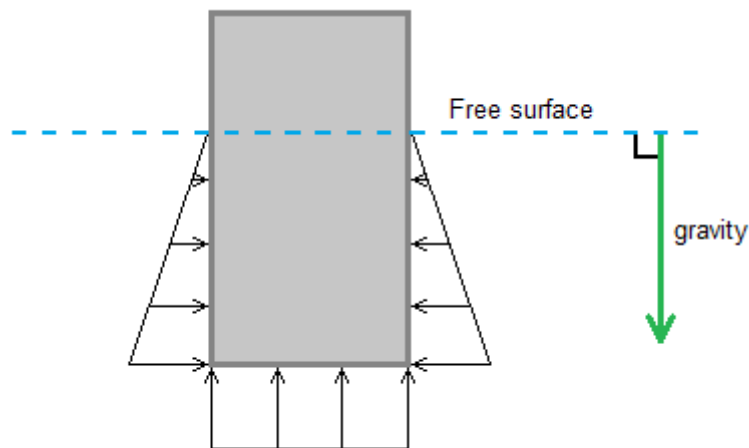
A line pressure can only be applied to line2 elements or faces that are locally 1-dimensional. This includes the edges of plane and line elements such as shells and beams. It can be constant or time dependent.

## 9.17 Hydrostatic Pressure

Analysis types: Static 3D, Modal Vibration 3D, Dynamic Response 3D, Buckling 3D

Solvers: Internal, CCX

Hydrostatic pressure is a triangular or trapezoidal pressure load that simulates the pressure gradient exerted on a surface by a body of fluid under the influence of gravity.



To use it, you must specify a point on the free surface, the density of the fluid and a separate gravity load.

Mecway converts the pressure into equivalent nodal forces by numerically integrating the pressure function over the area of each face it is applied to. If the entire face is above the free surface then no pressure will be applied to it. If the entire face is below the free surface then the integration will give exact nodal forces. However, if the face intersects the free surface then the integration will typically introduce a small error of less than 1% for that face. To reduce the effect of this error, you can refine the mesh at the free surface or align the edges of elements with the free surface.

A hydrostatic pressure can only be applied to element faces that are locally 2-dimensional. This includes the surfaces of shells and all faces of solid elements.

You can achieve the same effect as hydrostatic pressure by using a pressure load with a formula to define pressure as a function of position. The main difference is that hydrostatic pressure can more accurately capture the abrupt change in pressure that occurs if the free surface intersects an element's face because it uses more sampling points and a piecewise linear pressure function.

## 9.18 Moment

Analysis types: Static 2D, Static 3D, Nonlinear Static, Dynamic Response 2D, Dynamic Response 3D, Modal Vibration 3D, Nonlinear Dynamic Response, Buckling

Solvers: Internal, CCX

Moment applies a moment to either a single node with rotational degrees of freedom (beam or shell node), or as forces distributed over a selection of 2D faces (shell or solid faces). The direction of the moment vector is specified in global X, Y, Z coordinates.

Each component can be constant or time dependent.

## 9.19 Gravity

Analysis types: Static, Nonlinear Static, Modal Vibration 3D, Dynamic Response, Nonlinear Dynamic Response, Buckling

Solvers: Internal, CCX

Gravity gives a distributed force to all massive parts of a model. This includes elements whose material has density and nodes with mass loads applied to them. It also includes the fluid in any hydrostatic pressure loads.

Specify a vector which is the force per unit mass or acceleration due to gravity. For Earth gravity you can use  $Y=-9.81 \text{ m/s}^2$ . It can be either constant or time-dependent.

## 9.20 Centrifugal force

Analysis types: Static, Nonlinear Static, Modal Vibration 3D, Buckling

Solvers: Internal, CCX

Centrifugal Force puts the model in a rotating reference frame to simulate rotation of the entire model along with the global coordinate system. This gives every massive part (node or element) an outward force whose magnitude is

$$F = m \omega^2 r$$

where  $m$  is the part's mass,  $\omega$  is the angular velocity and  $r$  is the distance from the axis of rotation. The axis of rotation must be one of X, Y or Z.

Angular velocity can be constant or time dependent in nonlinear static analysis. However, since this is a quasi-static simulation, it can't model the forces induced by angular acceleration.

Centrifugal force can increase the radius of rotation of part of a structure, thus further increasing the centrifugal force. This effect is called spin softening and can sometimes have a significant effect on natural frequencies and critical buckling loads. It is automatically taken into account with the Modal Vibration, Buckling and Nonlinear Static analysis types but not linear Static.

You can include a 2<sup>nd</sup> centrifugal force in the Static 3D analysis type with the internal solver. This applies its own loads as described above but the combination of the two also causes the gyroscopic effect of a distributed moment about their mutually perpendicular axis. The overall load at each point with position vector  $\mathbf{r}$  is

$$\mathbf{F} = -m[\boldsymbol{\omega} \times (\boldsymbol{\omega} \times \mathbf{r}) + 2\boldsymbol{\Omega} \times (\boldsymbol{\omega} \times \mathbf{r}) + \boldsymbol{\Omega} \times (\boldsymbol{\Omega} \times \mathbf{r})]$$

where  $\boldsymbol{\Omega}$  is the angular velocity vector of the "outer" centrifugal force load and  $\boldsymbol{\omega}$  is the angular velocity vector of the inner centrifugal force load. You must identify one of them as being outer. The meaning of outer and inner is that the model rotates about the inner rotation axis and the inner rotation axis rotates about the outer rotation axis.

Beam elements that have non-zero density can not be used in the same model as two centrifugal force loads.

## 9.21 Mass

Analysis types: Static, Nonlinear Static, Modal Vibration, Dynamic Response, Nonlinear Dynamic Response, Buckling

Solvers: Internal, CCX

Mass is a lumped mass at a node. It can be used instead of density or to idealize a massive object without modeling it with elements. This mass contributes to dynamic behavior, gravity and centrifugal force. When you apply a single mass load to several nodes, the specified mass is divided equally between each node.



## 9.22 Rotational inertia

Analysis types: Dynamic Response, Modal Vibration

Solvers: Internal

A rotational inertia load is a point rotational inertia about one of the X, Y or Z axes. It can only be applied to the nodes of beam and shell elements. You can use rotational inertia to idealize massive parts such as flywheels without modeling them using elements.



## 9.23 Temperature

Analysis types: Static, Nonlinear Static, Nonlinear Dynamic Response, Modal Vibration 3D, Buckling, Thermal, DC Current Flow

Solvers: Internal, CCX

In a thermal analysis a temperature constraint constrains the temperature to the specified value on the items it is applied to. Heat is allowed to flow into or out of the model at these locations to maintain the specified temperature.

In a static or buckling analysis, the temperature is used for thermal stress where a thermal stress load must also be applied.

Temperature can be applied to nodes, faces or elements.

The temperature can be constrained within a specified time interval in a thermal transient analysis. Within this interval, a constant or time-dependent temperature can be specified. In steady state analysis, temperature must be specified as a single constant without any time interval.

It can also be specified as a function of position using a formula.



## 9.24 Node temperature

Analysis types: Static, Nonlinear Static, Modal Vibration 3D, Buckling, Thermal, DC Current Flow

Solvers: Internal, CCX

Node temperature fixes the temperature of each node individually. It cannot be applied to named selections and is appropriate for a temperature field whose value is different on every node.

A common use for node temperature is thermal stress analysis where the values are generated by the solver in a thermal analysis then used as input for a static analysis.

## 9.25 Thermal stress

Analysis types: Static, Nonlinear Static, Nonlinear Dynamic Response, Modal Vibration 3D, Buckling

Solvers: Internal, CCX

When present, thermal stress enables thermal loading. A single reference temperature must be specified as well as temperatures on parts of the model and thermal expansion coefficients in the materials.

The difference between each element's average specified temperature and the reference temperature is used to determine the thermal expansion/contraction loads. Any nodes that don't have a specified temperature are assumed to be at the reference temperature.

Only one thermal stress can be applied in a model.

## 9.26 Rayleigh damping

Analysis types: Dynamic Response

Solvers: Internal, CCX

Rayleigh damping, also called proportional damping, is a type of viscous damping that's distributed over the elements. It uses a damping matrix (C) formed from a linear combination of the mass (M) and stiffness (K) matrices

$$C = \alpha M + \beta K$$

You can specify values for  $\alpha$  and  $\beta$  to control its behavior. A non-zero  $\alpha$  causes each node to be connected to a fixed external point through a damper whose damping coefficient is related to the mass associated with that node. A non-zero  $\beta$  causes internal damping of deformation of the elements.

You can apply Rayleigh damping to either the whole model or a selection of elements. This allows you to assign different damping properties to different materials or regions of a structure. However, with the CCX solver or mode superposition option, only the whole model is allowed. If you apply more than one Rayleigh damping load to the same element then the damping matrices are summed which is equivalent to summing the damping ratios.

The modal vibration solver uses a damping ratio for each mode instead of a damping matrix. The damping matrix defined above is equivalent to a set of modal damping ratios

$$\zeta_i = \frac{1}{2} \left( \frac{\alpha}{\omega_i} + \beta \omega_i \right)$$

where  $i$  is the mode number and  $\omega_i$  is the angular frequency of that mode.  $\zeta_i$  must be less than 1 for every mode in the solution.

You can choose values for  $\alpha$  and  $\beta$  by solving the above equation at two points with damping ratios  $\zeta_1$  and  $\zeta_2$  and frequencies  $\omega_1$  and  $\omega_2$  respectively, as shown below. With the mode superposition option,  $\zeta$  for every mode must be less than 1 – i.e. the structure must be underdamped.

$$\alpha = \frac{2 \omega_1 \omega_2 (\zeta_2 \omega_1 - \zeta_1 \omega_2)}{\omega_1^2 - \omega_2^2} \quad \beta = \frac{2 (\zeta_1 \omega_1 - \zeta_2 \omega_2)}{\omega_1^2 - \omega_2^2}$$

## 9.27 Heat flow rate

Analysis types: Thermal

Solvers: Internal, CCX

Heat is added to the model at the specified rate. It is distributed uniformly across the volume or surfaces it is applied to. If it is applied to nodes then the specified heat flow rate is divided equally between each node.

Heat Flow Rate can be applied to nodes, faces or elements.

A time-dependent Heat Flow Rate can be specified as a formula or table of time and value pairs.

## 9.28 Heat flux

Analysis types: Thermal

Solvers: Internal, CCX

Heat is added to the model at the specified rate per unit area. It is distributed uniformly across the surfaces it is applied to.

Heat flux can only be applied to faces.

Time and position dependent values can be specified as a formula or as a table for time dependence only. With the CCX solver, it cannot be a function of both time and position in the same formula.

## 9.29 Internal heat generation

Analysis types: Thermal

Solvers: Internal, CCX

Heat is added to the model at a uniform rate over the volume it applies to. The value specifies the rate at which heat is added to the model per unit volume.

Internal Heat Generation can only be applied to elements.

Time and position dependent values can be specified as a formula or as a table for time dependence only. With the CCX solver, it cannot be a function of both time and position in the same formula.

## 9.30 Convection

Analysis types: Thermal

Solvers: Internal, CCX

Convection allows heat flow into or out of a face according to a specified heat transfer coefficient and ambient temperature.

Time and position dependent values can be specified as a formula or as a table for time dependence only. With the CCX solver, it cannot be a function of both time and position in the same formula.

## 9.31 Radiation

Analysis types: Thermal

Solvers: Internal, CCX

Radiation allows heat flow into or out of a face according to a specified emissivity and ambient temperature. It cannot model heat transfer between two parts of a model.

Time and position dependent values can be specified as a formula or as a table for time dependence only. With the CCX solver, it cannot be a function of both time and position in the same formula.

## 9.32 Thermal contact conductance

Analysis types: Thermal

Solvers: CCX

Thermal contact conductance connects two surfaces with a thermally resistive layer. It's similar to bonded contact but with a finite conductance per unit area.

## 9.33 Velocity

Analysis types: Fluid

Solvers: Internal

This constraint fixes a component of a fluid's velocity in either the X, Y or Z direction.

In the Fluid analysis type, you can model a no-slip boundary condition by applying two velocity constraints, zero velocity in the X direction and Y directions, on all the nodes of the boundary. You can model a symmetry plane by setting only one component to zero to prevent fluid flow through the plane of symmetry. You can model a known flow rate in an arbitrary direction by applying two velocity constraints with arbitrary X and Y components.

### 9.34 Fluid pressure

Analysis types: Fluid, Acoustic Resonance

Solvers: Internal

In the Fluid analysis type, fluid pressure constrains the fluid's pressure at all the nodes, faces or elements it is applied to. Only the corner nodes of elements have pressure DOFs, so fluid pressure constraints applied to mid-side nodes will be ignored by the solver.

In the Acoustic Resonance analysis type, you can use fluid pressure=0 to enforce a pressure node. Node here means a location where the pressure remains zero throughout the wave's cycle.

### 9.35 Electric potential

Analysis types: Static 2D, Static 3D, DC Current Flow

Solvers: Internal

Electric potential applies a uniform potential over every item it is applied to. In DC current flow analysis, current can flow into or out of the model as needed to maintain that potential. This is a voltage source relative to a common ground.

### 9.36 Charge

Analysis types: Static 2D, Static 3D

Solvers: Internal

Charge defines an electric charge uniformly distributed across the volume or surface it is applied to. If it is applied to nodes then the specified charge is divided equally between each node.

Charge can be applied to nodes, faces or elements.

### 9.37 Current

Analysis types: DC Current Flow

Solvers: Internal

Current defines an electric current flowing into the model and uniformly distributed across the surface it is applied to. If it is applied to nodes then the specified current is divided equally between each node.

Current can be applied to faces or nodes.

A positive value on the face of an element represents current flowing into the element.

### 9.38 Robin boundary condition

Analysis types: Static 2D, Static 3D, DC Current Flow

Solvers: Internal

The Robin Boundary Condition imposes the following equation on a face

$$k \partial u / \partial n + c_0 u + c_1 = 0$$

where

- $u$  is electric potential
- $k$  is either the electrical conductivity or absolute permittivity of the material
- $\partial u / \partial n$  is the normal derivative of  $u$  at the face
- $c_0$  and  $c_1$  are user specified parameters. If  $c_0 = 0$  this is a Neumann boundary condition, constraining the field value's gradient normal to the face.

### 9.39 Cyclic symmetry

Analysis types: Static 3D, Nonlinear Static, Modal Vibration 3D, Thermal Steady State

Solvers: Internal, CCX

If geometry and loading are regularly repeated about an axis of rotation, then cyclic symmetry (rotational periodicity) can be used to reduce the model to only one segment, allowing a smaller mesh and improved solver performance.

You must specify each cut surface as either the master or slave using either node or face selections.

In thermal analysis, cyclic symmetry is equivalent to applying constraint equations coupling the corresponding pairs of nodes on the cut surfaces.

In static analysis, cyclic symmetry is equivalent to applying constraint equations which equate the radial, circumferential and axial displacements of corresponding nodes on the cut surfaces.

In modal vibration analysis, cyclic symmetry applies transformed coordinates and couples the degrees of freedom using special complex constraints described in (*D. L. Thomas, Dynamics of Rotationally Periodic Structures, 1979*). This method can find modes which are not cyclically symmetric. The number of modes specified for the solution is not the total number of modes, but the number of modes having the same number of nodal diameters. For example if 5 modes are specified in a model with 7 segments, the solver will find the lowest 5 modes with each of 0, 1, 2 and 3 modal diameters, leading to a total of 20 modes.

Differences between the internal and CCX solvers:

CCX	Internal
Master and slave meshes don't have to be compatible.	Master and slave node layouts must match each other.
The surface with the finer mesh should be the slave to ensure all master faces have corresponding slave	

nodes.	
Shells and beams are not allowed in any analysis type.	Shells and beams are not allowed in modal vibration analysis type.
No constraints allowed on the slave surface.	Constraints on the cut surfaces are limited to: - Displacement in the direction of the symmetry axis - Zero displacement normal to the symmetry axis - Temperature
Any single point constraints allowed on the master surface.	
Solution shows the number of segments requested.	Solution only shows one segment with static and thermal analysis types.

## 9.40 Constraint equation

Analysis types: Static, Modal Vibration, Dynamic Response, Buckling, Thermal Steady State, DC Current Flow, Acoustic Resonance

Solvers: Internal, CCX

A constraint equation (also called multipoint constraint or MPC) is a more general type of constraint which enforces a linear relationship between several DOFs (degrees of freedom). Examples of uses are:

- The X-displacement of one node can be set equal to the Y-displacement of another node so they act as if they are connected by a 90° bell crank such as  $0 = u_{x1} - u_{y2}$ .
- In thermal analysis, the temperatures of many nodes can be coupled together to form a perfectly conducting region.
- In DC current flow, a voltage source can be made by setting a constant difference in the potential of two nodes such as  $5V = V_2 - V_1$ .
- Simple dissimilar meshes can be connected together and shells can be connected to solids.

A constraint equation has the form

$$A = B \times \text{dof } 1 + \dots + C \times \text{dof } n$$

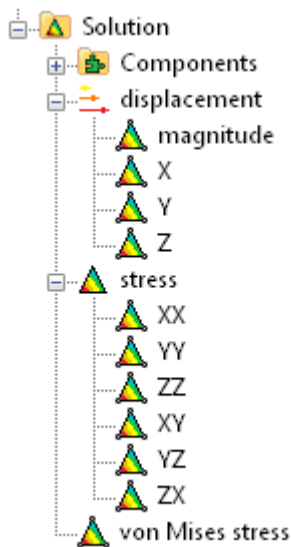
where A, B and C are constants. The first DOF, with coefficient B is special and is called the slave.

With some exceptions, there are restrictions that constraint equations must comply with:

- The slave DOF must not be a member of any other constraint equation or constraint.
- The slave DOF's coefficient must not be zero.
- Each DOF can only appear once in the same constraint equation.
- The constant (A) must be zero in modal vibration, dynamic response, buckling, acoustic resonance, and all analysis types with the CCX solver.
- With the CCX solver, they cannot refer to the rotational DOFs of beam or shell elements.

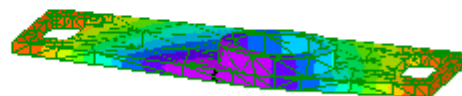
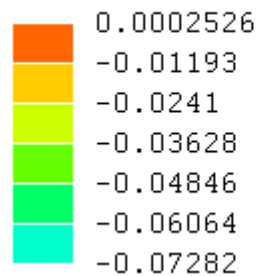
## Results

### 10.1 Display



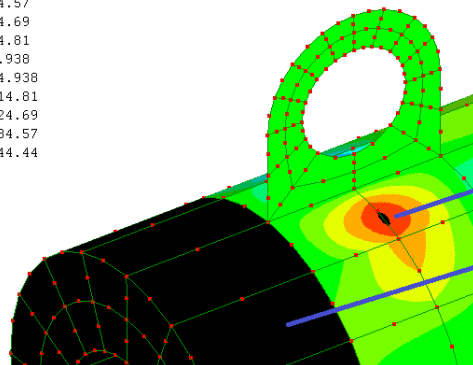
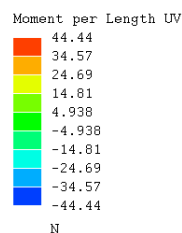
Select an item below **Solution** to display a field variable as a color coded contour plot. These are only available after solving the model. You can add additional field variables by right clicking the **Solution** branch.

Displacement in Z



The contour plot can be configured using by selecting **Options** then **Contour plot** from the **Tools** menu. Here you can change the color scheme, number of colors and resolution. A higher resolution is slower to render but gives smoother boundaries between colors so it can be useful for presentation. The **Gradient** option gives a smoother image which is also faster to generate with large meshes however it doesn't accurately show changes of color within a single element face.

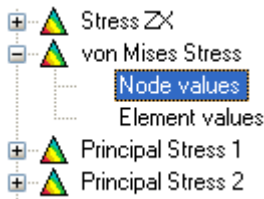
Sometimes black regions appear in the contour plot. This happens when there's no correct color that can be used. One reason is the value is outside the range of values that map to colors. It can also occur if the field value being displayed doesn't exist on that element. For example, solid elements have no bending moment so bending moments appear black. You can change this from black to



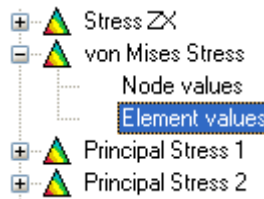
Value is greater than 44.44 N

Moment per Length UV does not exist on solid elements

another color using **Tools** → **Options** → **Contour plot** and **Above range color**, **Below range color**, and **No value color**.



**Node values** will display the results at each node as the average of the values from all its adjacent elements. This produces a continuous plot.

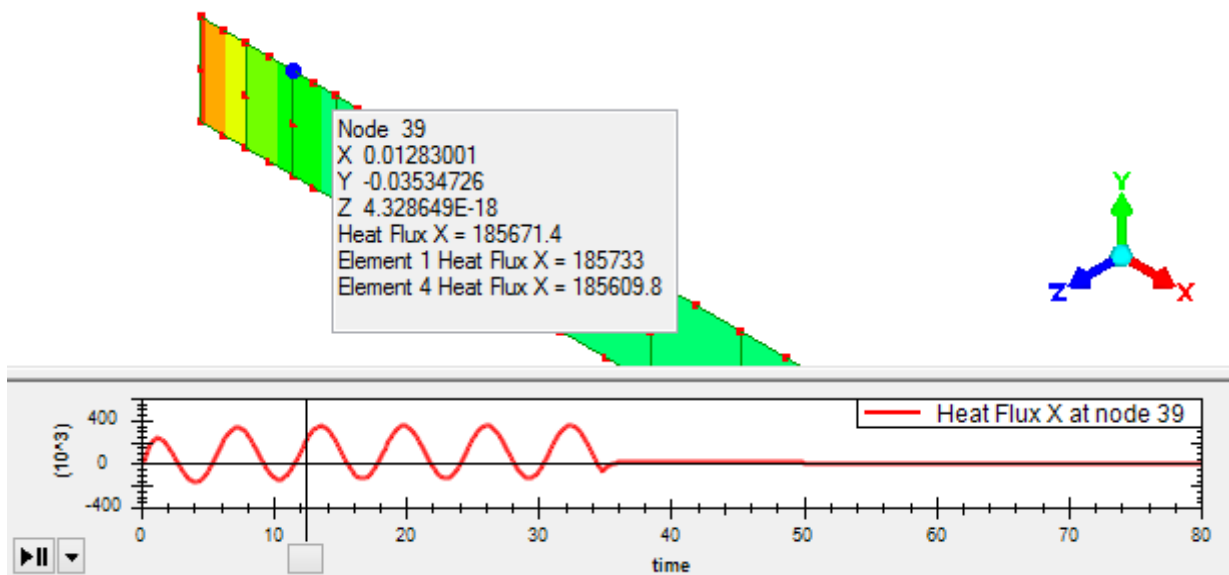


**Element values** will display the results without nodal averaging. Beam and shell values that are expressed in local element coordinates (U, V, W) should be displayed this way because the meaning of values on adjacent elements may be different. Not available with the CCX solver.

To turn off the contour plot, click the **Components** branch under **Solution** in the outline tree.

Click on a node to display both its averaged node value and the non-averaged values of all its adjacent elements. If there are time steps in the solution, the timeline will also show a graph of the node's value.

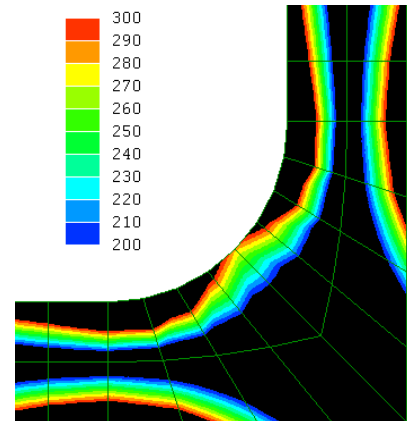
You can export the graph's data to a text file by right clicking on the graph and clicking **Export to CSV file...** or copy it to the clipboard with **Copy data**.



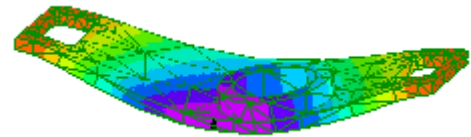
You can control the range of values displayed in the contour plot by moving the sliders to the left of the color key. Parts of the model outside the specified range will be displayed using the **Above range color** or **Below range color** which default to black.



An automatic way to set the range of values is to right click some selected elements and click **Fit contour plot max/min to selection**. This adjusts the scale to match the selected elements and ignore everything else.



**Deformed view** shows exaggerated deformation. A default scale factor is chosen automatically to give a clearly visible change. The minimum automatic scale factor is 1 except for mode shapes.



**Undeformed shape** superimposes the undeformed profile on the exaggerated deformation. The dropdown menu allow you to display it as either element edges or model edges.



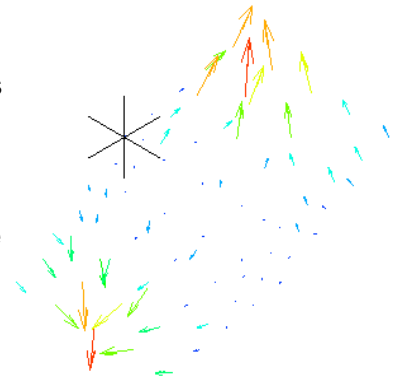
Vector fields are displayed as arrows if you select the group name in the outline tree, such as **displacement** or **velocity**.



**Animation** animates the deformation. This only works for results without time steps such as static and modal vibration. For results with time steps such as dynamic response, use the timeline described below.

Time steps in transient problems are displayed on the timeline, the length of which is the total time period of the analysis. The result in the graphic display will correspond to the slider's position.

To animate the display, click the play/pause button. The drop-down menu can be used to control the speed. You can use this together with **Deformed view** to show movement.



To move the slider in single steps, click its handle then press the left and right arrow keys.

To save the animation as an MPEG file, click the down arrow next to the animation button and choose **Save video to file**.



**Table** displays the results in a table which can be copied and pasted into a spreadsheet. The options you set here are saved with the model so you can easily generate the same table after solving again. If you enter a file name under **Write to csv file after solving** Mecway will save the table to the specified text file every time it solves the model. This file can be opened with a spreadsheet application or other software for further processing. If you omit the extension, no extension will be added to the name and the file will be saved without an extension. You can add multiple tables to

output different parts of the solution to different files and you can suppress different tables in different configurations.

**Tools** → **Find max/min** shows lines pointing to the maximum and minimum values in whole mesh if nothing is selected, or the selected nodes or elements if any are selected.

## 10.2 File Output

Various results can be written to files when the solver runs. Most of these options are available by right click on the **Analysis** item at the top of the outline tree. However, to write the solution data in tabulated form to a csv file, use the **Table** item under the **Solution** branch of the outline tree. This is described in the previous section.

**New save after solve** saves the model and solution to the specified file.

**New export stiffness matrix** saves the stiffness matrix to a text file. It can be stored in either a sparse or dense format. The **Dense** format contains the entire square matrix with one row on each line of the file. The **Sparse** format doesn't contain most of the zero elements and has only the upper triangle of the symmetric matrix. Each line of the file contains 3 numbers: the row number, the column number and the value in that location of the matrix.

**New export mass matrix** saves the mass matrix with the same two format options as the stiffness matrix.

**New export load vector** saves the global load vector to a file with the value at each DOF on a separate line.

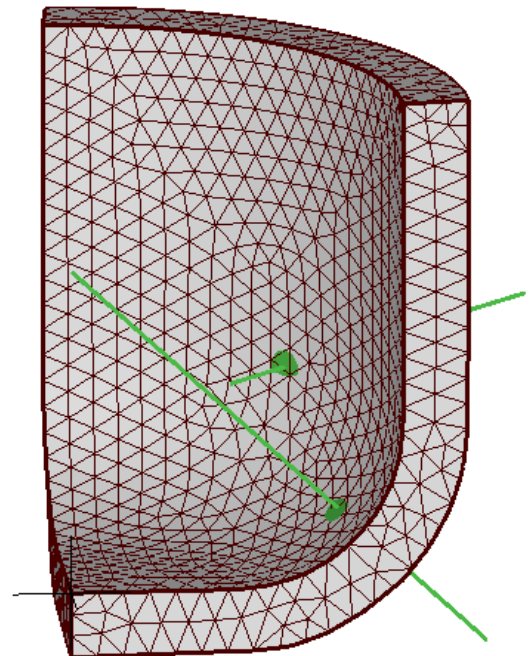
**New export DOF ordering** saves the relationship between node numbers and DOFs and the row numbers of the matrix. Each line of the file contains the node number, DOF name and matrix row number separated by commas. The matrix row number is shown as 0 where the global DOF has been condensed out of the matrix such as when it has a constraint or is a constraint equation slave DOF.

## 10.3 Stress Linearization

This tool finds linearized stresses along a straight line (stress classification line) in a solid model. It can be used for pressure vessel design with ASME or other codes. It interpolates 101 uniformly spaced points between the two surfaces of the mesh along a Stress Classification Line (SCL) defined by two endpoints. It then produces a graph and table showing linearized von Mises, maximum shear stress  $\times 2$  (stress intensity) or principal stresses at each point.

There is no need to have mesh nodes on the SCL. Mecway interpolates stress values from the elements that the line passes through.

To use Stress Linearization, first make sure the solution is displayed, then click **Tools** → **Stress linearization**. You can specify the SCL endpoints either by entering 2 sets of coordinates or by selecting two nodes before opening the Stress linearization tool. The SCL endpoints can be outside the mesh but should not be inside it.



You can add multiple **stress linearization** items to create multiple SCL lines which are saved with the model.

### Membrane stress

Membrane stress ( $\sigma^m$ ) is constant along the line and is determined by evaluating the following integral using the trapezoidal method:

$$\sigma^m = \frac{1}{t} \int_{-t/2}^{t/2} \sigma dT$$

t is the length of the line

T is the coordinate in the direction of the line

$\sigma$  is the 6-component total stress tensor produced by the solver.

### Bending stress (only used internally)

Each component of bending stress ( $\sigma^b$ ) varies linearly along the length of the line, passing through zero at the center. The bending stress at the beginning of the line (the end closest to the origin) is determined by evaluating this integral using the trapezoidal method:

$$\sigma^b(0) = \frac{-6}{t^2} \int_{-t/2}^{t/2} \sigma T dT$$

### Membrane plus bending stress

If **Ignore through-thickness bending stress** is checked then the components of bending stress in the direction of the line (T) are ignored. Here M and H are the meridional and hoop directions.

$$\begin{aligned}\sigma_{TT}^{m+b} &= \sigma_{TT}^m \\ \sigma_{MM}^{m+b} &= \sigma_{MM}^m + \sigma_{MM}^b \\ \sigma_{HH}^{m+b} &= \sigma_{HH}^m + \sigma_{HH}^b \\ \sigma_{TM}^{m+b} &= \sigma_{TM}^m \\ \sigma_{MH}^{m+b} &= \sigma_{MH}^m + \sigma_{MH}^b \\ \sigma_{HT}^{m+b} &= \sigma_{HT}^m\end{aligned}$$

If **Ignore through-thickness bending stress** is unchecked, then all bending stress components are included:

$$\sigma^{m+b} = \sigma^m + \sigma^b$$

### Peak Stress

Peak stress is the remaining stress besides membrane and bending:

$$\sigma^{\text{peak}} = \sigma - \sigma^{m+b}$$

### Total Stress

$\sigma$  is the non-linearized stress produced by the solver.

## 10.4 Mean and Volume Integral

These tools integrate any continuous field value (F) over the volume of a domain (D) defined by a set of 3D solid or 2D plane elements. The **Mean** tool also divides this integral by the volume of the domain to obtain the mean of the field value. Example uses are to find the total power dissipated in a solid conductor by integrating power density or to find the average temperature of a non-uniformly heated object.

$$\text{Mean: } \frac{\int_D F dV}{V} \qquad \text{Volume integral: } \int_D F dV$$

To use it, first make sure the solution is displayed, then click **Tools** → **Mean** or **Tools** → **Volume integral**. If you want to use a domain which isn't the complete model, you must define it as an element selection or a component before solving.

If the field value is defined by both element values and node averaged values, the element values are used for greater accuracy. If only node averaged values are present, they are used instead.

## 10.5 Surface integral

This is similar to **Mean** but integrates over a surface defined by face selections. You can use it to calculate heat flow rate from heat flux or current from current density. It can integrate both scalar and vector fields.

## 10.6 Sum

**Tools** → **Sum** sums the values of solution variables on the nodes of predefined named selections. It can only be used for reaction force components, external force components, formulas, and unknown field variables imported from external solvers.

## 10.7 Formula

**Tools** → **Formula** generates a new field variable that's a function of existing field variables, position, time, and some material properties. The available variables and operators are listed in the formula window. Some examples:

### 10.7.1 Reaction moment

Reaction moments composed of force couples are not included in the default reaction moment output. However, you can calculate the moment of each reaction force about the origin using the following formulas and then use **Solution** → **Sum** to sum them to a single value.

$$\text{Reaction moment about X: } FR.z * y - FR.y * z + MR.x$$

$$\text{Reaction moment about Y: } FR.x * z - FR.z * x + MR.y$$

$$\text{Reaction moment about Z: } FR.y * x - FR.x * y + MR.z$$

The  $MR.x$ ,  $MR.y$ , and  $MR.z$  terms must not be included if there are no elements with rotational DOFs.

### 10.7.2 Cylindrical coordinate system

The files **DisplacementCylindrical.liml** and **StressCylindrical.liml** in the samples folder contain formulas which transform the displacement and stress fields of a solution into cylindrical coordinates

with Z as the axis of symmetry. You can use **File** → **Import...** to add these formulas to an existing model.

## Samples and Verification

### 11.1 BeamBendingAndTwisting.liml

Analysis type: Static 3D

Elements: Beam (line2)

Loads and constraints: Fixed support, Force, Moment

This sample shows Mecway finding the longitudinal stresses due to bending of a beam using a predefined cross-section. It also shows torsional stiffness and that torsion does not influence longitudinal stress. The beam is made from a hollow rectangular tube with properties:

Width:  $a=23\text{mm}$

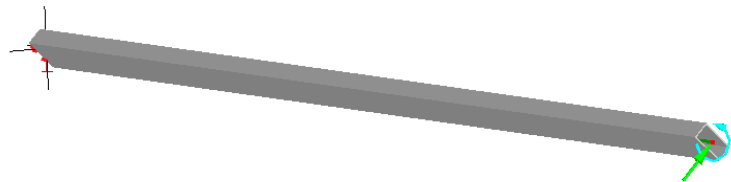
Height:  $b=50\text{mm}$

Wall thickness:  $t=2\text{mm}$

Length:  $l=2\text{m}$

Young's modulus:  $E=200\text{GPa}$

Poisson's ratio: 0.3

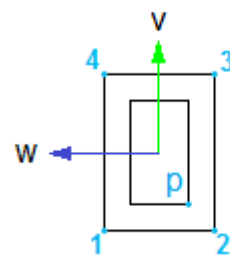


The beam is oriented at 45 degrees to the Y axis and is fully constrained at one end and loaded at the other end with a lateral force of 1N and a twisting moment of 2Nm.

Mecway reports stress at 5 locations on the cross-section: each of the four outer corners and a user defined point p on an inside corner. These results are compared to hand calculations below.

The longitudinal stress at the base of the beam at point 1 ( $W_1 = 12.5\text{mm}$ ) should be

$$\sigma_1 = \frac{M_v W_1}{I_{vv}} = 0.844490 \text{ MPa}$$



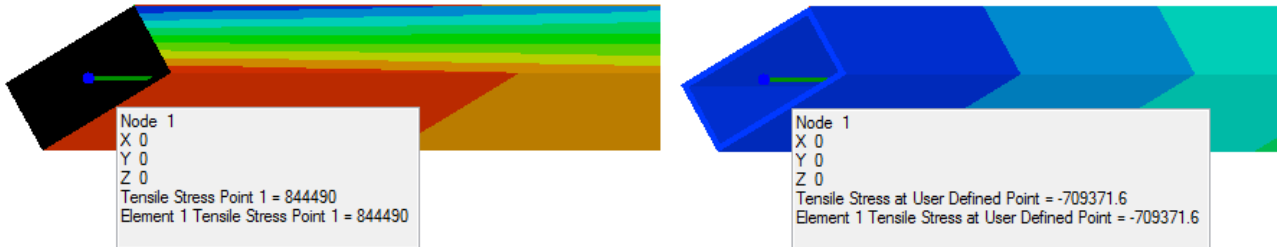
The longitudinal stress at the user defined point p ( $W_p = -10.5\text{mm}$ ) should be

$$\sigma_p = \frac{M_v W_p}{I_{vv}} = -0.709372 \text{ MPa}$$

The bending moment about the V axis ( $M_V$ ) is  $2\text{m} \times 1\text{N} = 2\text{Nm}$ . The 2nd moment of area about the V axis ( $I_{VV}$ ) can be found using a readily available formula and the parallel axis theorem:

$$I_{VV} = 29603 \frac{2}{3} \text{mm}^4$$

Mecway shows the same stress values:



The twisting angle can be found using

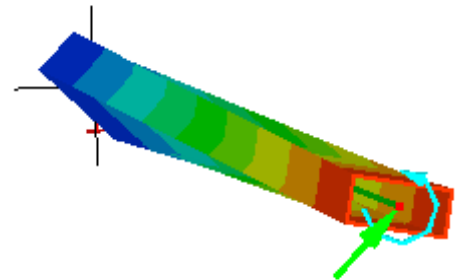
$$\theta = \frac{LT}{GJ}$$

where  $L$  is the beam length (2m),  $T$  is the applied torque (2Nm),  $G$  is the modulus of rigidity ( $\frac{200}{2.6} \text{GPa}$ ), and  $J$  is the torsion constant which is approximated by

$$J \approx \frac{2t(a-t)^2(b-t)^2}{a+b-2t} = 6.86657 \times 10^{-8} \text{m}^4$$

(Warren C. Young, *Roark's Formulas for Stress & Strain*)

This gives  $\theta = 7.5729 \times 10^{-4}$  radians which is the same as the twist angle found by Mecway because Mecway uses the same approximate formula for  $J$ .



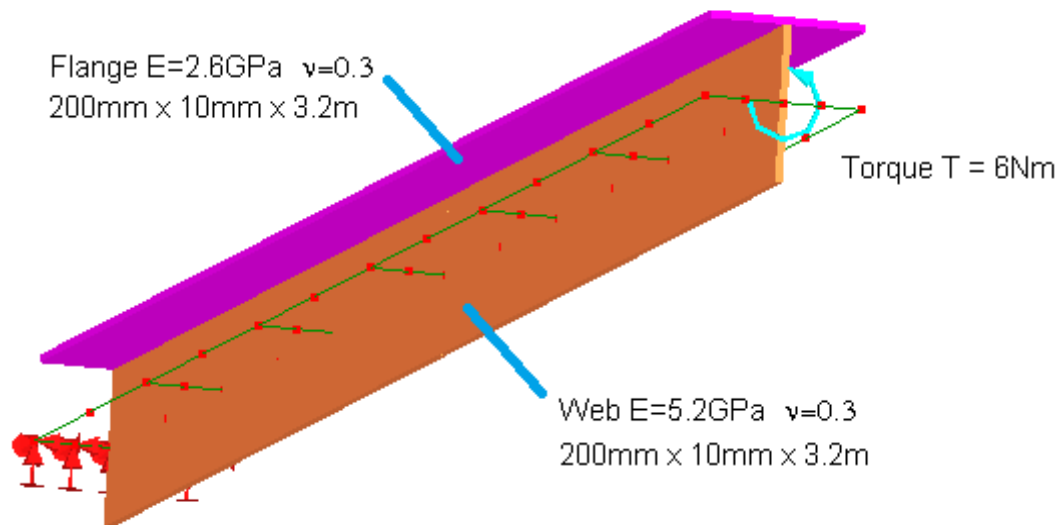
## 11.2 CompositeBeam.liml

Analysis type: Static 3D

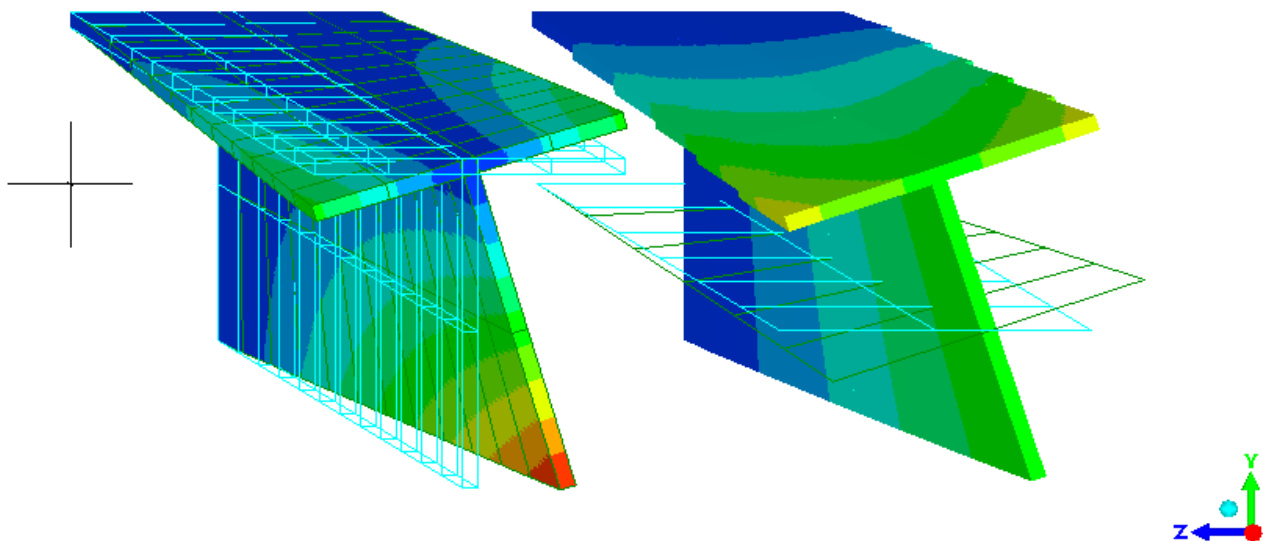
Elements: Beam (line2), Shell (quad8), Solid continuum (hex20)

Loads and constraints: Fixed support, Force, Moment

This beam is made from two different materials by using a combination of beam and shell elements. The web is made from a line of beam elements whereas the flange is made from shells connected to the beams elements. A 0.105m **Shell offset** is specified in **Element properties** to locate the shell elements at the edge of the web rather than at the center.



An equivalent solid model is included in the same file for comparison. The solutions to both models show the beam rotating about a point at the intersection of the web and flange because this is the location of the shear center.



We will now calculate the angle of twist by treating the web and flange stiffnesses independently:

Torsion constant of either the flange or the web

$$J = \frac{Bt^3}{3} = \frac{0.2 \times 0.01^3}{3} = 6.6667 \times 10^{-8} m^4$$

Shear modulus of flange



$$G_{flange} = \frac{E_{flange}}{2(1+0.3)} = 1 \times 10^9 Pa$$

Shear modulus of web

$$G_{web} = \frac{E_{web}}{2(1+0.3)} = 2 \times 10^9 Pa$$

Angle of twist

$$\theta = \frac{IT}{J(G_{flange} + G_{web})} = \frac{3.2m \times 6Nm}{6.6667 \times 10^{-8} m^4 (1 \times 10^9 Pa + 2 \times 10^9 Pa)} = 0.108 radians$$

This closely matches the angle of 0.109 radians given by Mecway for rotation about X at the tip of the beam.

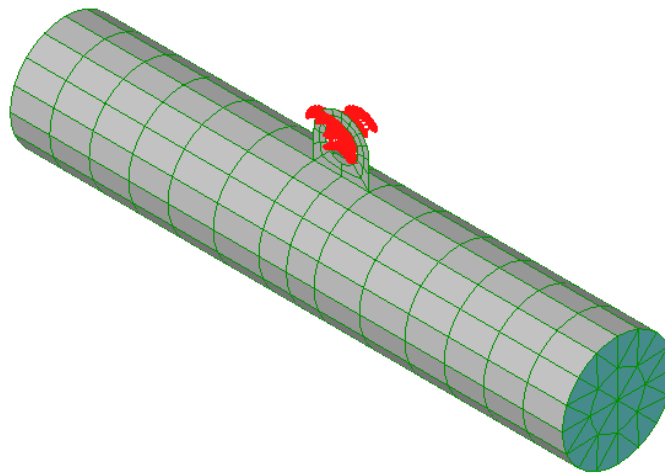
### 11.3 CylinderLifting.liml

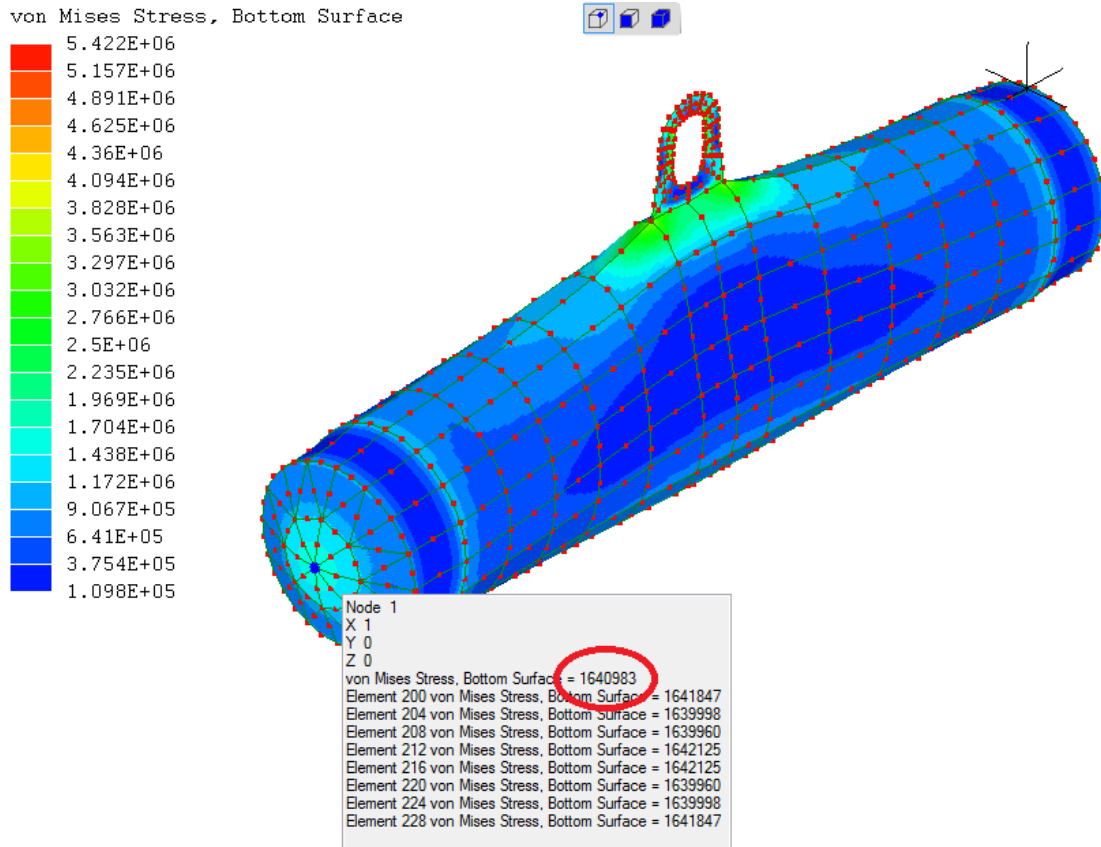
Analysis type: Static 3D

Elements: Shell (tri6, quad8)

Loads and constraints: Fixed support, Gravity, Pressure

This sample shows the use of shell elements for a pressure vessel in two different configurations – self weight under gravity, and the same with an additional internal pressure load. It also uses material of different thickness for the end caps.





In the With\_pressure configuration, the von Mises equivalent stress on the outside surface at the center of each end cap is  $1.64 \times 10^6 \text{ Pa}$ . Mesh refinement to 16 times the number of elements shows this stress to be in error by about 2%. The outside surface is called the bottom surface in the solution. You can see this by turning on **Show element axes** to show the direction of the blue W axis of each element.

## 11.4 PressureVesselAxisymmetric.liml

Analysis type: Static axisymmetric

Elements: Axisymmetric continuum (quad8)

Loads and constraints: Displacement, Pressure

This is a cylindrical pressure vessel with semi-ellipsoidal ends. It is modeled as axi-symmetric because the geometry and loads all have axial symmetry about the longitudinal axis. Only one half is modeled due to mirror symmetry and a displacement constraint in the Y direction is applied to enforce this boundary condition.

It is made from steel ( $E=200\text{GPa}$ ,  $\nu=0.3$ ) with wall thickness  $t=20\text{mm}$ , inside radius  $r=500\text{mm}$  and internal pressure  $P=1\text{MPa}$ .

Three tests are used to validate the results:

The longitudinal reaction force in the cylindrical wall at the mirror symmetry boundary is

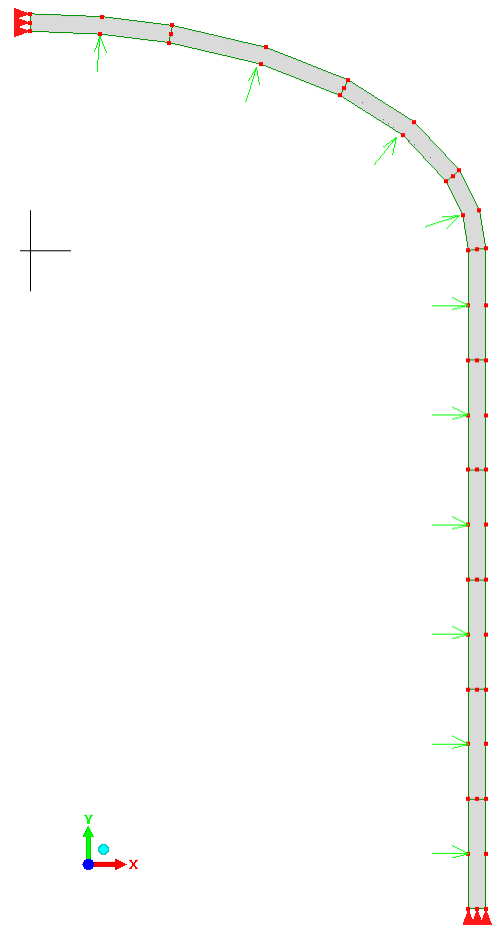
$$F_Y = -P \times \pi r^2 = -785\text{kN} \text{ compared to Mecway's } -798\text{ kN}$$

The axial stress in the cylindrical wall is

$$\sigma_{YY} = \frac{F_Y}{\pi((r+t)^2 - r^2)} = 12.25\text{MPa} \text{ compared to Mecway's } 12.44\text{ MPa}$$

The hoop stress in the middle of the cylindrical wall is

$$\sigma_{ZZ} = \frac{Pr}{t} = 25.00\text{MPa} \text{ compared to Mecway's } 25.00\text{ Mpa}$$



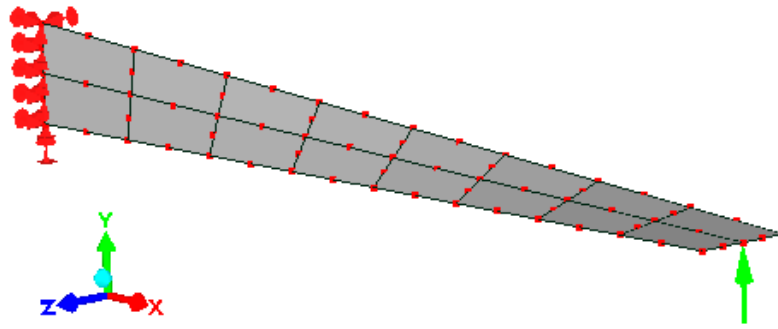
## 11.5 TwistedBeam.liml

Analysis type: Static 3D

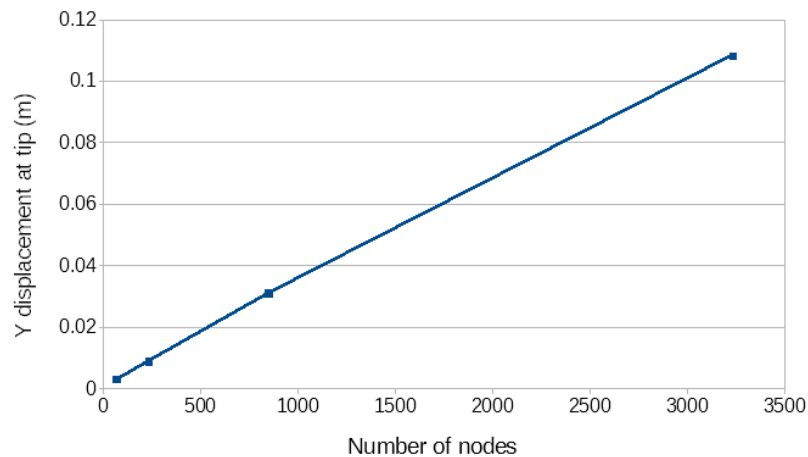
Elements: Shell (quad8)

Loads and constraints: Fixed support, Force

This sample demonstrates a popular test case where Mecway's shell elements fail to produce results because they are warped. It is a cantilever beam with a  $90^\circ$  twist along its length and a point load at the free end.



The graph below shows that as the mesh is refined, the displacement at the tip increases in proportion to the number of nodes. It does not converge to a solution. The first data point with 69 nodes is the solution from the sample pictured above without any mesh refinement.



*Failure to converge with mesh refinement*

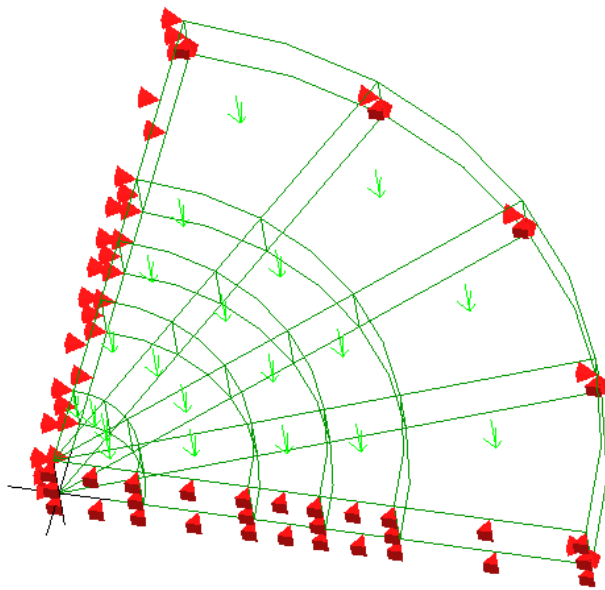
## 11.6 MembraneActionPlate.liml

Analysis type: Nonlinear Static

Elements: Solid continuum (hex20)

Loads and constraints: Frictionless support, Displacement, Pressure

This sample demonstrates the nonlinear effect of membrane action to stiffen a circular plate supported at the edges. The supports allow rotation about the edge but not displacement in any direction. Because radial displacement is constrained, when a pressure load is applied to the surface and the plate deflects, a tensile membrane stress is developed within it and further deflection is resisted.



Only one quarter of the plate is modeled due to symmetry. Frictionless supports are used to enforce mirror symmetry on two surfaces.

The solution shows the Z-displacement at the center of the circle to be -1.5mm which is significant compared to the 1mm thickness of the plate.

Changing the analysis type from Nonlinear Static 3D to Static 3D then solving it again shows a much greater center displacement of -7.8mm. This linear solution is wrong because it is made with the assumption of small displacements so it ignores the dominant membrane stiffness which only appears after the plate begins to deflect.

## 11.7 SaggingCable.liml

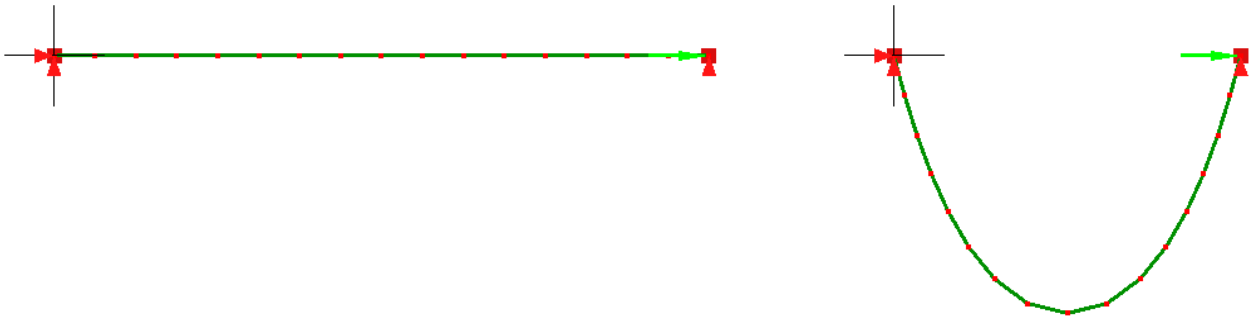
Analysis type: Nonlinear Static

Elements: Truss (line2)

Loads and constraints: Displacement, Force, Gravity

A flexible cable or rope is suspended between a fixed point and a horizontally sliding point. A force is applied at the sliding point to keep the cable in tension. We will find the vertical displacement at the center.

Because the untensioned cable has no bending stiffness and cannot support any lateral load, the tensile force is applied first. This gives it stiffness to support the gravity load which is applied in the 2<sup>nd</sup> time step.



At time=1s the cable is tensioned (left). at time=2s gravity is applied causing it to sag (right).  
The deformed shape is a catenary, which, with the vertex at the origin has the formula

$$y = a \cosh(x/a) - a$$

$$a = F / (\rho A g)$$

Tensile force  $F = 2000\text{N}$

Density  $\rho = 8000\text{kg/m}^3$

Cross sectional area  $A = 0.01\text{m}^2$

Gravity  $g = 9.8\text{N/kg}$

The solution shows that half the horizontal length of the deformed cable is  $x = 3.63792\text{m}$ . Using this value, the formula gives  $y = 3.0645\text{m}$  which is within 0.1% of distance in the solution (3.0680m).

## 11.8 BucklingBeam.liml

Analysis type: Buckling 2D Beam

Elements: Beam (line2)

Loads and constraints: Fixed Support, Force

Consider a column fixed at the base and free at the top, subject to an axial compressive force. We will find the value of the force at which it buckles, the critical load.

The column is of length  $L=100\text{mm}$ , the cross section is a  $10\text{mm}\times 10\text{mm}$  square and the Young's modulus is  $E=0.2\text{ MPa}$ .

The model is made with line2 elements. The bottom face has a fixed support to constrain displacement and rotation. A point load of  $1\text{N}$  has been applied at the top.

The Euler buckling equation is used to calculate the critical load:

$$P_{cr} = \pi^2 E I / (K L)^2$$

$K = 2$  for these boundary conditions

$E = 2 \times 10^5 \text{N/m}^2$

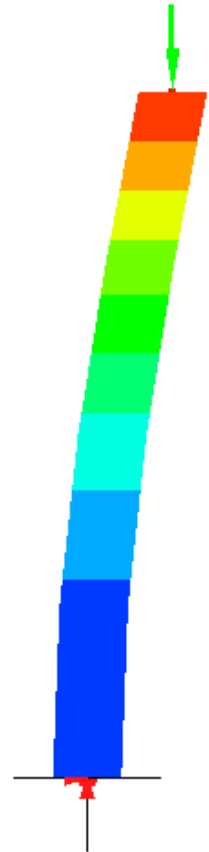
$I = 833.333 \text{mm}^4$

$L = 100\text{mm}$

Critical load,  $P_{cr} = 41123\text{N}$

The buckling factor for mode 1 found by Mecway is 41125 and approaches the theoretical value with mesh refinement. This is equal to the buckling load in newtons because a force of  $1\text{N}$  was applied in the model. The mode shape shows that this is indeed the first buckling mode.

The shift point is set to 1000 which is typically reasonable being an order of magnitude below the lowest buckling factor.



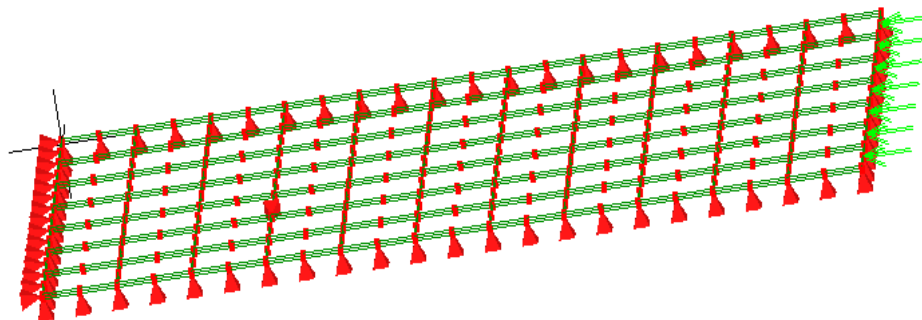
## 11.9 BucklingPlate.liml

Analysis type: Buckling 3D

Elements: Solid continuum (hex20)

Loads and constraints: Displacement, Pressure

Consider a simply supported plate with compressive edge loading on two opposite edges. We will determine the critical loads of lowest few buckling modes.



The translational motions along the  $Z$  axis are constrained for all the edges, but they are free to rotate about the edge directions. Translational motion in the  $X$ - $Y$  plane is allowed, except the bottom edge is

restrained to provide a reaction to the applied force. A single arbitrarily chosen node is constrained in the x direction to prevent rigid body motion.

The total applied load is equal to 1, so the buckling factors produced by the solver are equal to the critical loads.

For thin plates such as the one used above, the following analytical equation can be used to calculate the first four critical loads:

$$F_{cr} = b\pi^2 a^2 D ([m/a]^2 + [n/b]^2) / m^2$$

$F_{cr}$ : The critical load applied to one edge

$a=2$ : Length in direction of load

$b=1$ : Width

$D=E t^3 / [12 (1-\nu^2)]$  : Plate rigidity

$t=0.0125$  : Thickness

$\nu=0.3$ : Poisson's ratio

$E=1e6$ : Young's modulus

$m$ : Number of half-waves of the buckle in the direction of the load

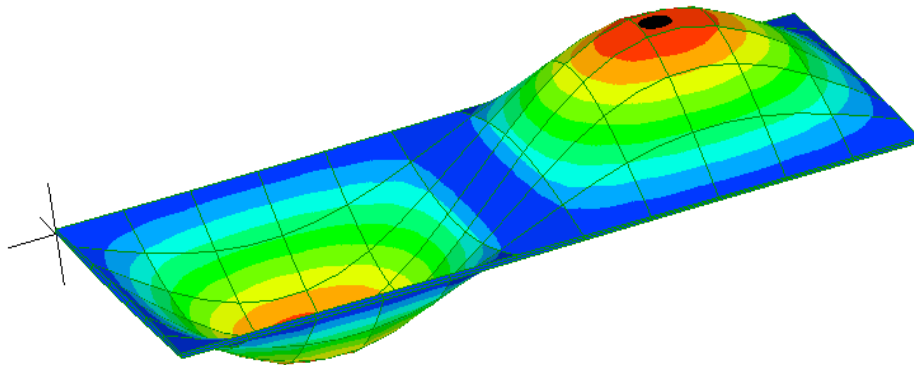
$n$ : Number of half-waves of the buckle perpendicular to the load

For  $m=1$  and  $n=1$ ,  $F_{cr}=11.0$ . The Mecway model gives 11.1

For  $m=2$  and  $n=1$ ,  $F_{cr}=7.06$ . The Mecway model gives 7.14 (pictured below)

For  $m=3$  and  $n=1$ ,  $F_{cr}=8.29$ . The Mecway model gives 8.60

For  $m=4$  and  $n=1$ ,  $F_{cr}=11.0$ . The Mecway model gives 12.0



Notice that the lowest critical load is for the mode with 2 half-waves so this mode will develop first under increasing load.

## 11.10 BucklingPlateNonlinear.liml

Analysis type: Nonlinear Static

Elements: Solid continuum (hex20)

Loads and constraints: Displacement, Force

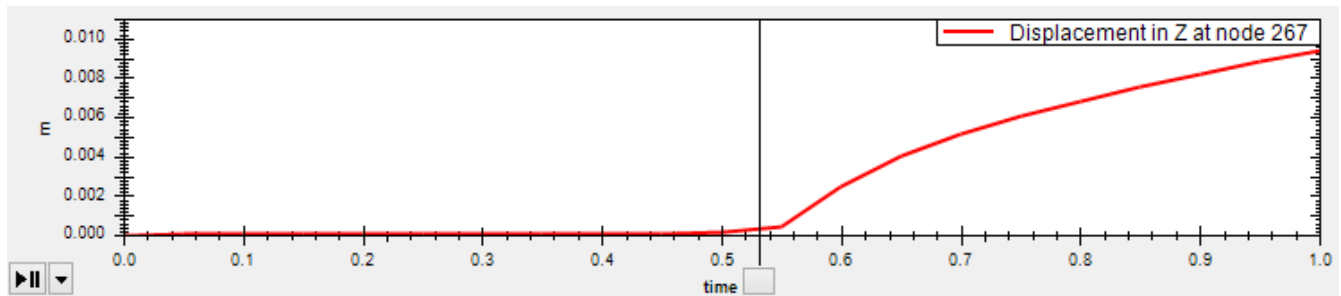
This is the same case as BucklingPlate.liml above but is solved using nonlinear static analysis instead of linear buckling analysis.



The applied load is ramped up from 6N to 8N which includes the expected critical load for the 1<sup>st</sup> buckling mode. If you didn't know the critical load in advance, you would ramp up the load from zero.

A perfectly symmetric model won't buckle in nonlinear analysis so we must apply an imperfection to initiate it. A small out of plane force of 0.0001N is applied at an arbitrary location that's not on a nodal line of the mode shape.

The solution for out-of-plane displacement shows that buckling is initiated at a load of 7.0-7.1N which contains the theoretical critical load of 7.06N. In this graph, time=0 corresponds to a load of 6N and time=1 corresponds to 8N.



## 11.11 PipeClip.liml

Analysis type: Nonlinear Static 3D

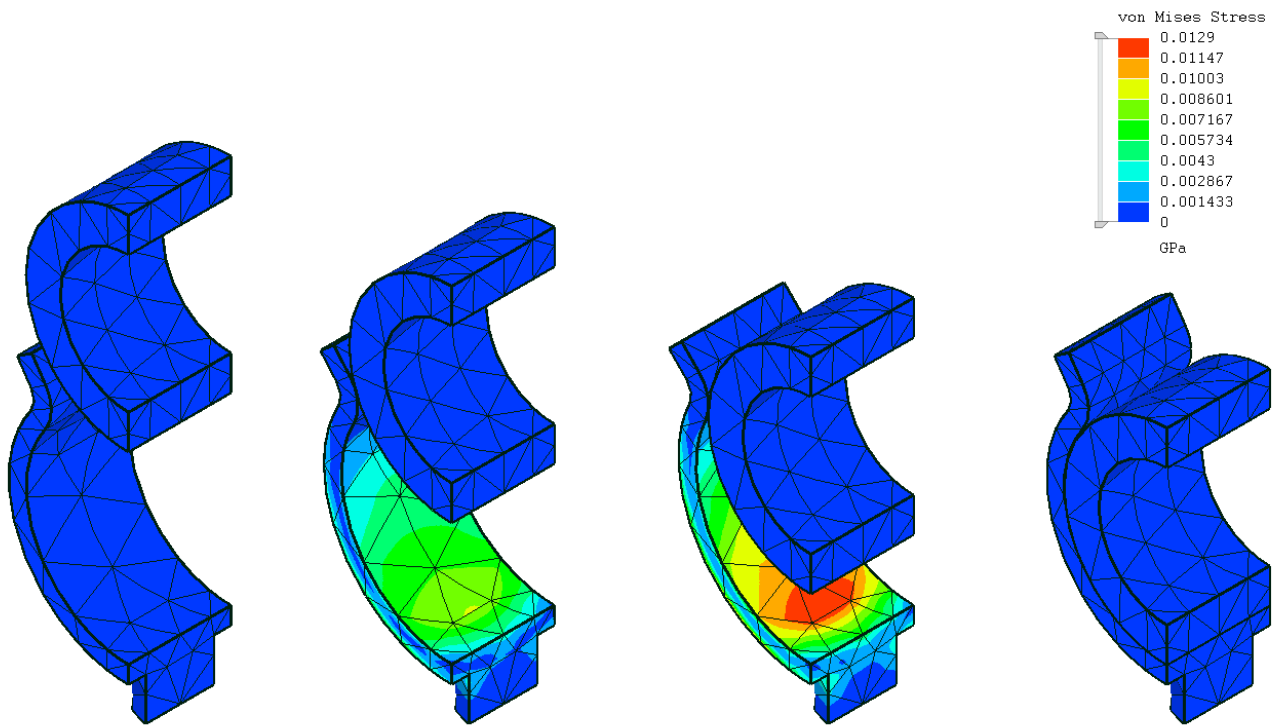
Elements: Solid continuum (tet10)

Loads and constraints: Fixed Support, Frictionless Support, Displacement, Contact

The sample shows non-linear contact between two different parts - a pipe and a clip using the CCX solver.

The two parts are defined by CAD geometry in two .step files clip.step and pipe.step which are only needed to rebuild the mesh.

Boundary conditions include frictionless support to enforce mirror symmetry, fixed support at the base of the clip where it would be mounted to another part, and contact between the outside of the pipe and the inside of the clip. The pipe is moved into the clip using a time-dependent constraint defined by the displacement function  $u_y = -12t$  mm/s.



The solution shows maximum stress when the clip is fully opened by the pipe, as expected.

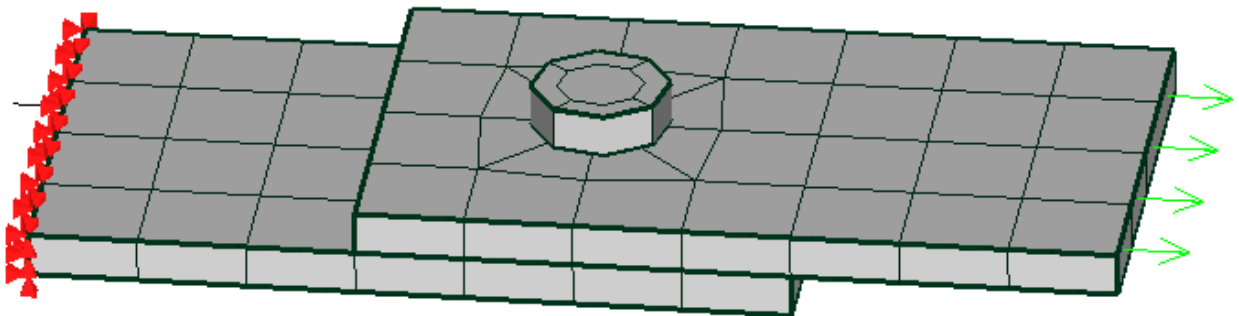
## 11.12 BoltedJoint.liml

Analysis type: Nonlinear Static 3D

Elements: Solid continuum (hex20)

Loads and constraints: Fixed Support, Bonded Contact, Contact, Pre-tension Section, Force

This sample shows two plates fixed together by a preloaded bolt with friction between the plates and an applied force that is increased until the connection fails.



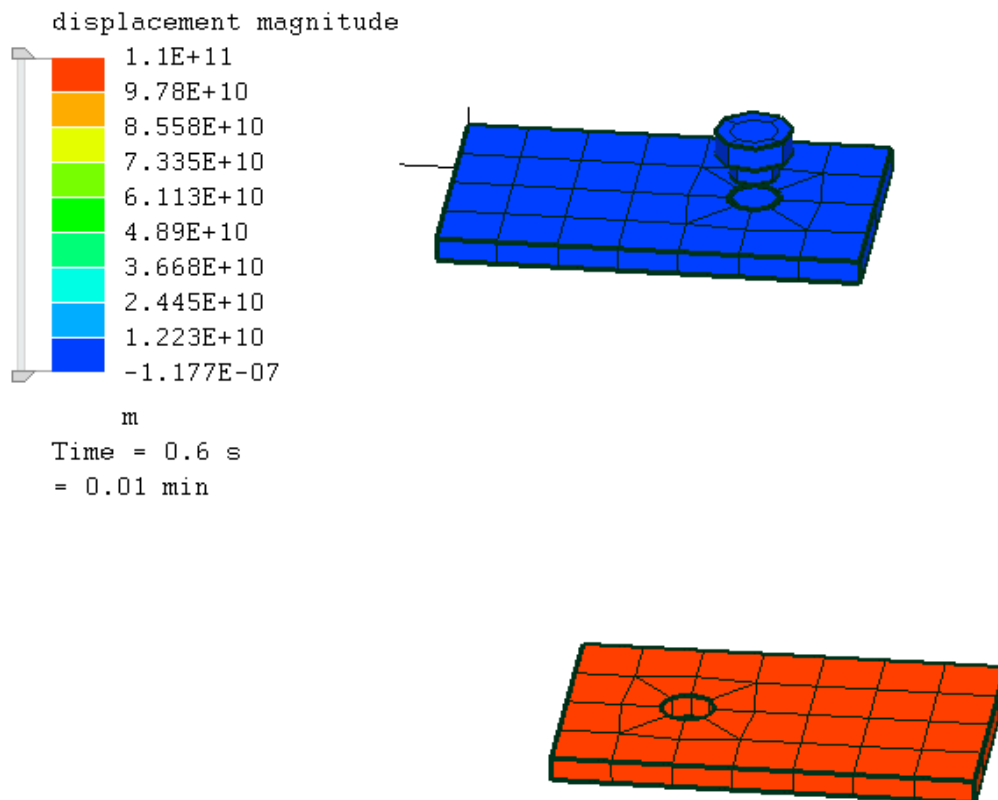
The bolt head is attached to one plate with bonded contact to make a robust connection. You may be able to use contact instead to allow it to slide or separate. The other surfaces are connected by contact so they can separate.

The preload force on the bolt is a constant 10 kN and the friction coefficient between the plates is 0.5 so we expect the maximum load the joint can carry is  $0.5 \times 10 \text{ kN} = 5 \text{ kN}$ .

The solver fails after time = 0.5 s with the message

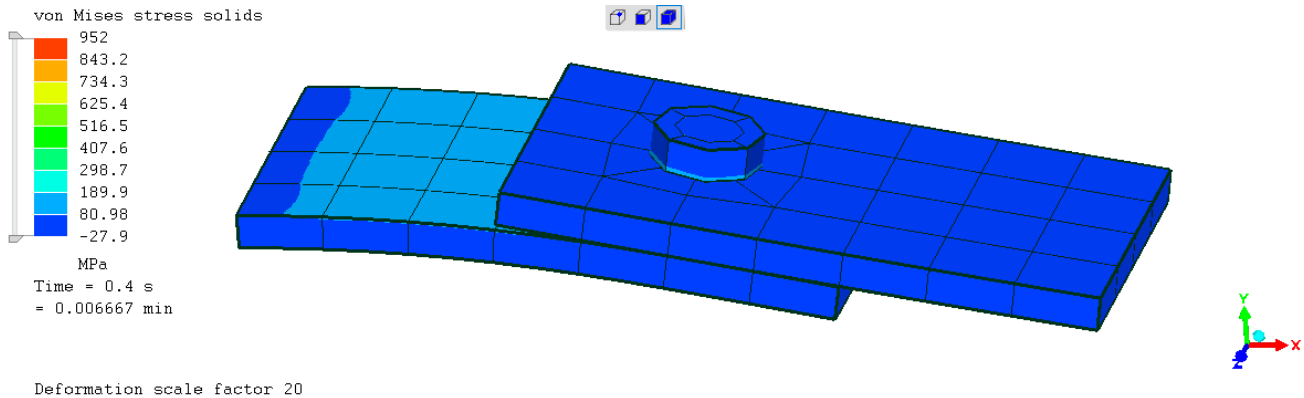
```
*ERROR: solution seems to diverge; please try
automatic incrementation; program stops
best solution and residuals are in the frd file
```

0.5 s corresponds to an applied force of 5 kN as expected. We can get some insight into the reason for the failure by looking at the final unconverged deformed shape. Set the timeline slider to 0.6 s and notice that displacement magnitude of the loaded plate is very large, in this case, on the order of  $10^{11}$  m. Set **Solution** → **Deformed view settings** → **Scale factor** to a reasonable scale of about 10% of the inverse of that, which is  $1\text{e-}12$  in this case. We can now see that there was rigid body motion of the loaded plate, indicating that it broke free of the friction as expected.



Deformation scale factor 1E-12

We can also see the deflection at lower loads such as  $t = 0.4$  s with a deformation scale factor of 20.



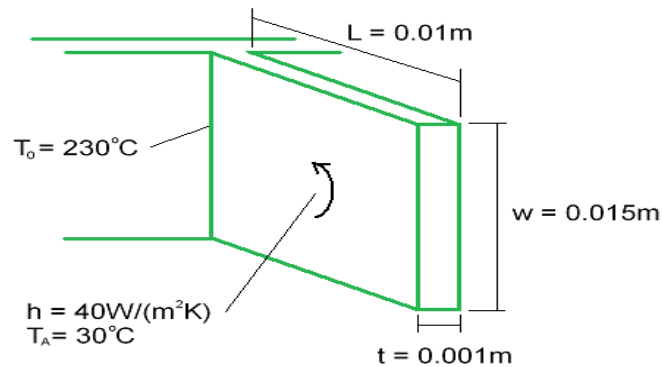
## 11.13 FinConvection.liml

Analysis type: Thermal Steady State

Elements: Fin (line2, line3)

Loads and constraints: Temperature, Convection

Mecway will find the temperature distribution along the length of the thermal fin shown below. Heat enters the base of the fin which is held at a fixed temperature of  $230^{\circ}\text{C}$  and is lost through the 4 sides by convection. The material has thermal conductivity of  $k=380\text{W}/(\text{mK})$ . The model contains two alternative meshes for the same fin, one using a line3 element and the other using two line2 elements.



The temperature ( $T_x$ ) a distance  $x$  from the base of the fin is given by

$$\frac{T_x - T_a}{T_0 - T_a} = \frac{\cosh(mL - mx)}{\cosh(mL)}$$

where  $m = \sqrt{\frac{hP}{kA}}$

$P$  is the perimeter of the cross-section,  $0.032\text{m}$

$A$  is the area of the cross-section,  $0.000015\text{m}^2$

The table below shows close agreement between Mecway's results and the temperatures given by the above formula.

Location	$T_x$ from the formula	2×line2 elements	1×line3 element
Midpoint ( $x=0.005\text{m}$ )	228.3306°C	228.3299°C	228.3306°C
End ( $x=0.01\text{m}$ )	227.7752°C	227.7742°C	228.7752°C

## 11.14 ConductionConvectionRadiation.liml

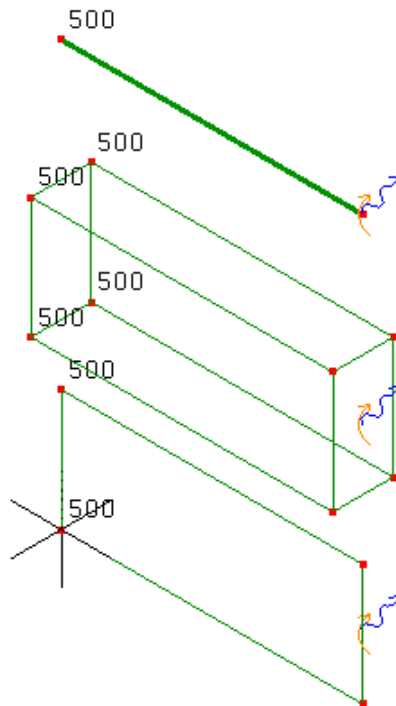
Analysis type: Thermal Steady State

Elements: Shell (quad4), Solid continuum (hex8), fin (line2)

Loads and constraints: Temperature, Convection, Radiation

This model shows one-dimensional steady state heat transfer by conduction, convection and radiation through a slab of thickness 0.05 m. Heat flows in through the left hand surface which is held at 500 K and heat flows out through the right hand surface by both convection (heat transfer coefficient  $h = 37.5 \text{ W/(m}^2\text{K)}$ ) and radiation (emissivity  $\epsilon = 0.8$ ) to an ambient temperature of 300 K.

Three versions of the model are included in the same file, using shell, solid and fin elements. All three give the same results except heat flux is not available for the fin element.



To verify the analysis we will perform a simple heat balance:

$$q_{\text{conduction}} = q_{\text{convection}} + q_{\text{radiation}}$$

Heat flux by conduction ( $q_{\text{conduction}}$ ) is given by Mecway as the x component of heat flux which is the same over the entire model

$$q_{\text{conduction}} = 3591.65 \text{ W/m}^2$$

Heat flux by convection is found using the temperature Mecway calculated for the right hand face ( $T_{\text{surface}}=380.2783\text{K}$ )

$$q_{\text{convection}} = h(T_{\text{surface}} - T_{\text{ambient}}) = 3010.44 \text{ W/m}^2$$

Heat flux by radiation is found in a similar way but using the Stefan-Boltzmann law

$$q_{\text{radiation}} = \epsilon \sigma (T_{\text{surface}}^4 - T_{\text{ambient}}^4) = 581.21 \text{ W/m}^2$$

where  $\sigma$  is the Stefan-Boltzmann constant,  $5.67037 \times 10^{-8} \text{ Wm}^{-2}\text{K}^{-4}$

The heat balance equation is satisfied:

$$3591.65 \text{ W/m}^2 = 3010.44 \text{ W/m}^2 + 581.21 \text{ W/m}^2$$

## 11.15 OscillatingHeatFlow.liml

Analysis type: Thermal Transient

Elements: Shell (quad8)

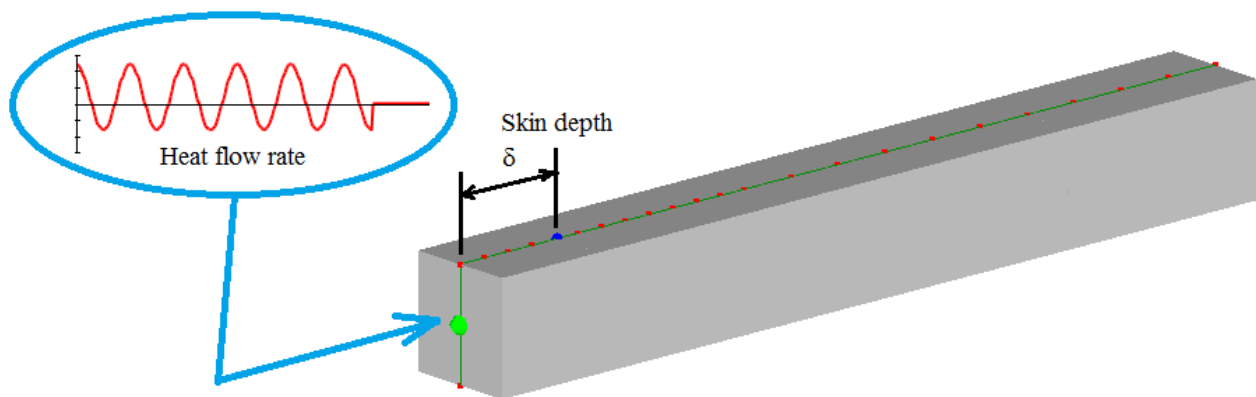
Loads and constraints: Heat flow rate

An insulated aluminum bar is heated by an oscillating thermal load applied to one end then removed. Mecway will find the temperature in the bar as a function of both time and distance along the bar. The material properties are:

Specific heat capacity,  $c_p = 900 \text{ K/(kg.K)}$

Density,  $\rho = 2700 \text{ kg/m}^3$

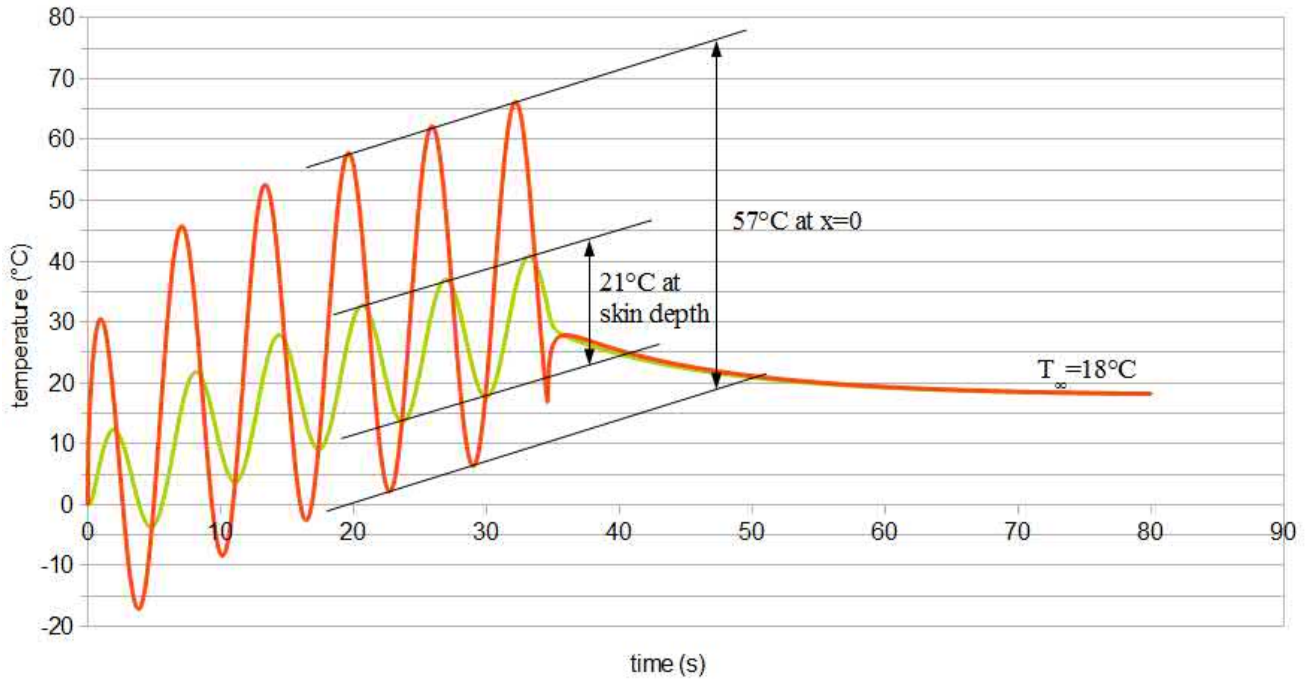
Thermal conductivity,  $k = 200 \text{ W/(m.K)}$



The heat load  $Q(t)$  is input into Mecway as a table of heat flow rate values every 0.2s according to this function:

$$Q(t) = \begin{cases} (100\cos(t) + 20) W & t \leq 34.6s \\ 0 & t > 34.6s \end{cases}$$

The graph below shows the temperature variation with time at two points. The red curve is measured at the end of the bar ( $x=0$ ) and the green curve is temperature measured at the skin depth ( $x=\delta$ ).



These results are verified in two ways, first by comparing the ratio of the temperature wave amplitudes at two points to their theoretical ratio, and second by comparing the net heat input to the total temperature rise after it has reached equilibrium.

The skin depth  $\delta$  is the distance at which the amplitude of the temperature variation is  $1/e$  of the amplitude at  $x=0$ . It is given by

$$\delta = \sqrt{\frac{kP}{\rho c_p \pi}}$$

where  $P$  is the period of the temperature wave.  $P=2\pi$

$$\delta = 0.01283001m$$

A mesh node is located at  $x=\delta$  for convenient measurement. The graph above shows the peak-peak amplitude of temperature at this node is  $21^\circ\text{C}$ . It also shows the peak-peak amplitude of temperature at  $x=0$  is  $57^\circ\text{C}$ .  $57/20 = 2.85 \approx e$  as required.

The expected temperature rise from  $t=0$  to  $t=\infty$  is

$$\Delta T = \frac{Q_{net}}{\rho c_p V}$$

$V$  = volume of the bar =  $1.5625 \times 10^{-5} \text{m}^3$

$$\begin{aligned}
 Q_{net} &= \int_0^{34.6} Q(t) dt \\
 &= \int_0^{34.6} (100\cos(t) + 20) W dt \\
 &= [100\sin(t) + 20t]_a^b J \\
 &= 687.75 J
 \end{aligned}$$

$$\Delta T = 18^\circ C$$

This agrees with the final value of temperature given by Mecway across the whole bar which is also  $18^\circ C$ .

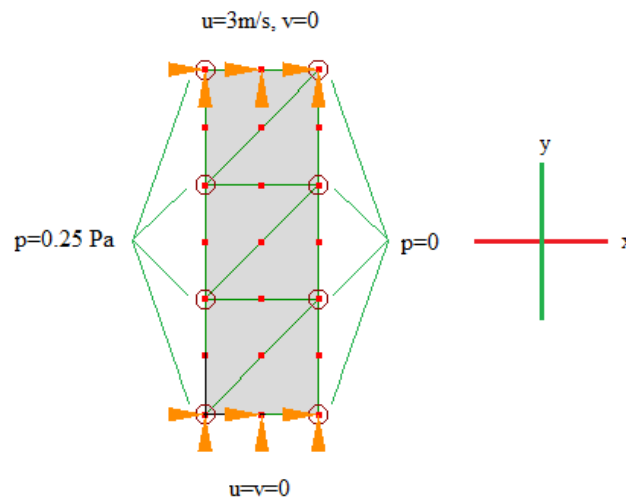
## 11.16 FluidCouette.liml

Analysis type: Fluid 2D

Elements: Plane continuum (tri6)

Loads and constraints: Velocity, Fluid Pressure

Here an analysis is done for fully developed flow between parallel plates with the lower plate fixed and the upper plate moving parallel to it with a constant velocity  $U=3\text{m/s}$ . The flow is driven both by the plate movement and an applied pressure gradient of  $-0.25\text{Pa/m}$ . This kind of flow is commonly known as Couette flow.



The theoretical velocity profile is parabolic, which can be represented exactly by the quadratic tri6 elements so the x-components of velocity from Mecway are identical to the theoretical values:

$$u(y) = \frac{U}{h} \left( y + \frac{h}{2} \right) - \frac{1}{2\mu} \frac{\partial p}{\partial x} \left( \frac{h^2}{4} - y^2 \right)$$

where



$$U = u(1.5) = 3\text{m/s}$$

$$h = \text{height} = 3\text{m}$$

$$\mu = \text{dynamic viscosity} = 0.1 \text{ Pa.s}$$

$$\frac{\partial p}{\partial x} = \text{pressure gradient} = -0.25\text{Pa/m}$$

y (m)	u(y) from formula (m/s)	u(y) from Mecway (m/s)
-1.5	0	0
-0.5	3.5	3.5
0	4.3125	4.3125
0.5	4.5	4.5
1.5	3	3

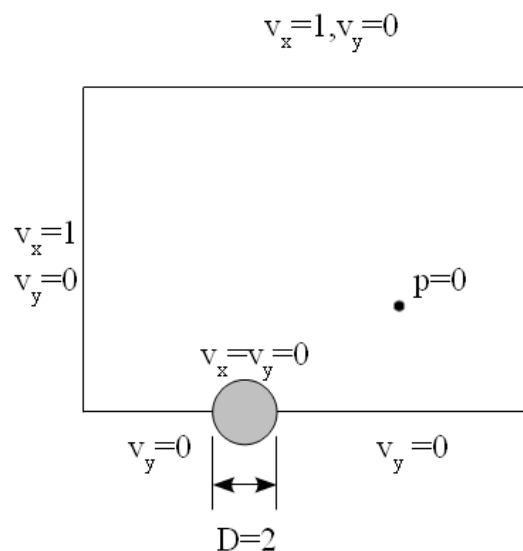
## 11.17 FluidViscousCylinder.liml

Analysis type: Fluid 2D

Elements: Plane continuum (quad8)

Loads and constraints: Velocity, Fluid Pressure

Rotational laminar flow around a cylinder is analyzed. This example has been taken from (C. Taylor & T. G. Hughes, *Finite Element Programming of The Navier-Stokes Equations*). Dynamic viscosity = 0.1 and density = 1. The Reynolds number is 20 which is expected to lead to the boundary layer separating into two steady, symmetrical eddies downstream of the cylinder. Only one will appear in the solution because only one half of the cylinder is modeled.



At the upper and left boundary, the fluid velocity in x is set to 1 and the fluid velocity in y is zero to represent uniform flow a large distance from the cylinder. At the lower boundary, the y component of velocity is set to zero to enforce symmetry. On the surface of the cylinder, both components of velocity are zero because it has a no-slip boundary condition.

The table below shows the close agreement between Mecway's results and those published in Taylor and Hughes:

Node number	Source	velocity x	velocity y	pressure
236	Taylor & Hughes	0.404	0.000	0.00173
	Mecway	0.404	0.000	0.00173
237	Taylor & Hughes	0.414	-0.0101	-
	Mecway	0.414	-0.0101	-
238	Taylor & Hughes	0.442	-0.0187	0.00211
	Mecway	0.442	-0.0187	0.00211
243	Taylor & Hughes	1.12	-0.0215	-
	Mecway	1.12	-0.0215	-
244	Taylor & Hughes	1.08	-0.0159	-0.0113
	Mecway	1.08	-0.0159	-0.0113

## 11.18 VibratingFreePlate.liml

Analysis type: Modal Vibration 3D

Elements: Shell (quad8)

Loads and constraints: none

This sample shows the undamped natural vibration modes of a square plate without any supports. Its properties are:

Side length,  $a = 0.1\text{m}$

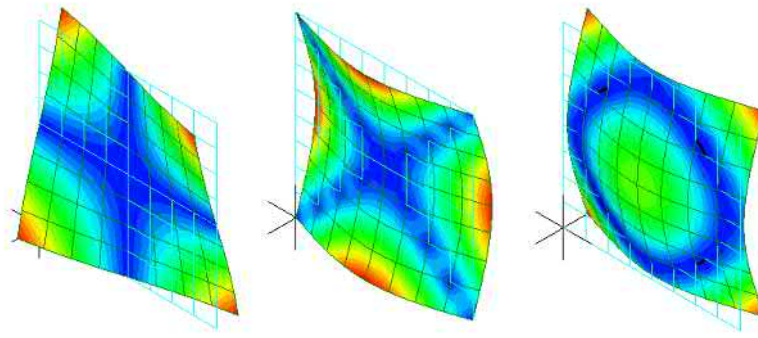
Thickness,  $h = 1\text{mm}$

Young's modulus  $E = 200\text{ GPa}$

Poisson's ratio,  $\nu = 0.225$

Density,  $\rho = 8000\text{ kg/m}^3$

The first three deformation modes found by Mecway are shown below along with their angular frequencies. Mecway also produces 6 rigid body modes because the plate is unconstrained.



$$\omega_1 = 2114 \text{ rad/s}$$

$$\omega_2 = 3122 \text{ rad/s}$$

$$\omega_3 = 3662 \text{ rad/s}$$

These results are verified by comparison with analytical formulas from (*William Weaver, Jr., Stephen P. Timoshenko, Donovan H. Young, Vibration Problems in Engineering, 1990*):

$$\omega = \frac{\alpha}{a^2} \sqrt{\frac{D}{\rho h}}$$

$$D = \frac{Eh^3}{12(1-\nu^2)}$$

$$\alpha_1 = 14.10, \omega_1 = 2089 \text{ rad/s}$$

$$\alpha_2 = 20.56, \omega_2 = 3046 \text{ rad/s}$$

$$\alpha_3 = 23.91, \omega_3 = 3542 \text{ rad/s}$$

For all three modes the error is less than 4% and can be reduced to less than 1% with mesh refinement.

## 11.19 VibratingCantileverBeam.liml

Analysis type: Modal Vibration 3D

Elements: Beam (line2)

Loads and constraints: Fixed support

Beam elements are used to model a vibrating cantilever fixed at one end and free at the other. At least four elements are needed to capture the higher bending modes. The beam's properties are:

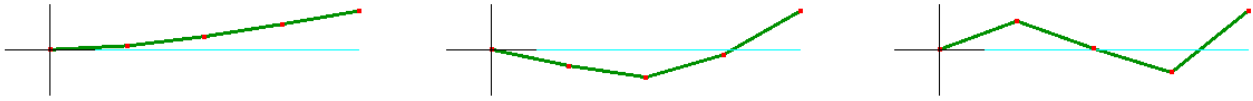
Length,  $L = 2.4\text{m}$

Width,  $b = 0.1\text{m}$

Height,  $h = 0.05\text{m}$

Young's modulus,  $E = 200 \text{ GPa}$

Density,  $\rho = 7860 \text{ kg/m}^3$



The frequencies of the first 3 modes for bending in the XY plane are verified by the formula

$$f_n = \frac{\alpha_n^2}{2\pi} \sqrt{\frac{EI}{\rho b h L^4}}$$

$$\alpha_1 = 1.875$$

$$\alpha_2 = 4.694$$

$$\alpha_3 = 7.885$$

$$I = \frac{bh^3}{12} = 2.0833 \times 10^{-6} m^4$$

which gives these frequencies compared to Mecway

Mode	Frequency from formula	Frequency from Mecway
n=1	7.073 Hz	7.074 Hz
n=2	44.33 Hz	44.38 Hz
n=3	125.1 Hz	125.1 Hz

Mecway's solution is 3D so it also includes modes for bending in the Z direction which are neglected in this 2D calculation.

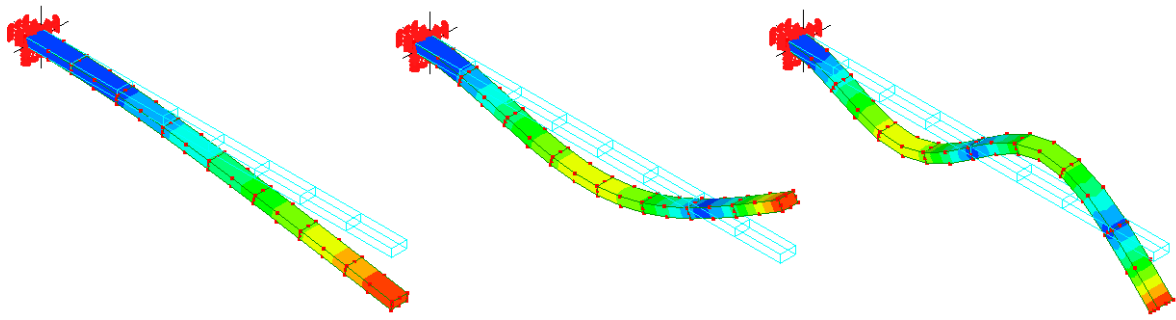
## 11.20 VibratingCantileverSolid.liml

Analysis type: Modal Vibration 3D

Elements: Solid continuum (hex20)

Loads and constraints: Fixed support

This is a model of the same beam described in VibratingCantileverBeam.liml but it is made using hex20 solid elements. The picture below shows the same mode shapes that are shown for VibratingCantileverBeam.liml.



Mode	Frequency from formula	Frequency from Mecway
n=1	7.073 Hz	7.089 Hz
n=2	44.33 Hz	44.99 Hz
n=3	125.1 Hz	128.7 Hz

## 11.21 VibratingTrussTower.liml

Analysis type: Modal Vibration 3D, Static 3D

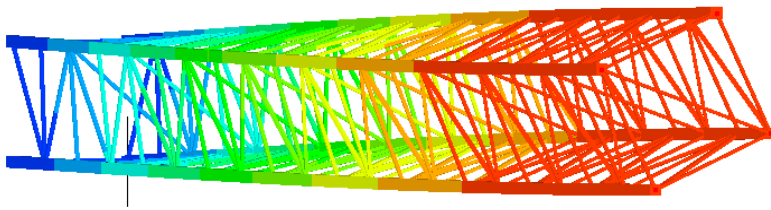
Elements: Truss (line2)

Loads and constraints: Displacement, Force

A truss tower will be analyzed to find the fundamental torsional vibration mode. It is 20m high and made from 10 identical 1m×1m×2m sections connected end to end. Each section uses two types of steel beam – the chords at the corners have 0.01m<sup>2</sup> cross sectional area and the bracing members have 0.001m<sup>2</sup> cross sections. All joints are free to rotate in any direction. The steel's Young's modulus is 200GPa and density is 7800kg/m<sup>3</sup>. The 4 nodes at the base are constrained against displacement in all directions.

The first two modes found by Mecway are bending modes and the 3rd is the fundamental torsion mode shown below with frequency

$$f_{\text{Mecway}} = 7.64 \text{ Hz.}$$



This result is validated using the formula for the fundamental torsion mode of a uniform shaft with distributed mass

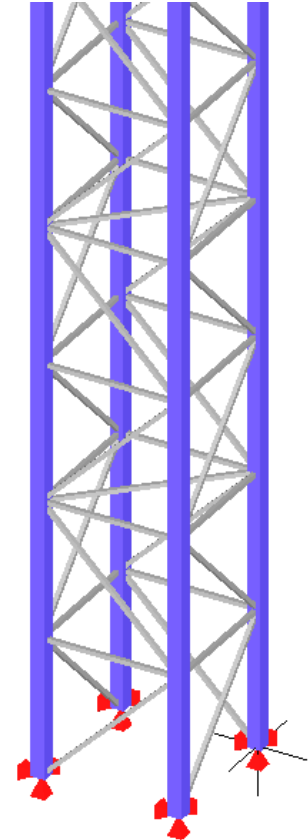
$$f = \frac{1}{4} \sqrt{\frac{K}{I}}$$

$I = 3640.54 \text{ kgm}^2$  is the rotational inertia of the whole structure about its longitudinal axis. It can be found by summing the rotational inertia of all 250 truss elements.

$K$  is the torsional stiffness of the structure. This is found by using Mecway to perform a static analysis with an applied moment of 1 Nm then measuring the resulting rotation at the end. The moment is applied using 4 forces on the top 4 nodes. If you unsuppress the group named *Moment* and change the analysis type to Static 3D then solve it, it will show the top nodes each rotate by  $\theta = 2.92843 \times 10^{-7}$  radians about the longitudinal axis of the tower. This gives  $K = 1 \text{ Nm} / \theta = 3.41480 \times 10^6 \text{ Nm/radian}$ .

The frequency given by this formula matches Mecway's result to within 0.3%.

$$f = 7.66 \text{ Hz}$$



## 11.22 VibratingMembrane.liml

Analysis type: Modal Vibration 3D

Elements: Shell (tri6)

Loads and constraints: Fixed Support, Displacement, Thermal Stress, Temperature

A hexagonal sheet of fabric is fixed around its outer edge and held in a state of uniform isotropic bi-axial tension. It can vibrate like a drum head due to the stiffness caused by its tension. We will find its first 3 natural frequencies and mode shapes.

Density: 1000kg/m<sup>3</sup>

Thickness: 1mm

Distance between opposite corners: 600mm

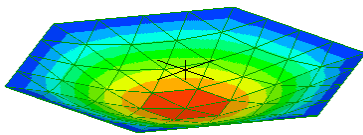
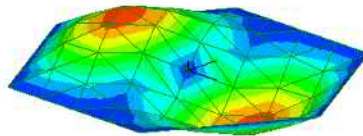
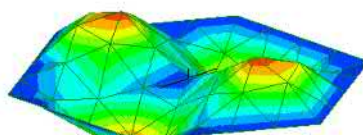
Tension per unit length: 2N/m

The uniform stress is applied using thermal strain. An artificial thermal expansion coefficient of 1K<sup>-1</sup> is used with an arbitrary small Young's modulus of 0.001N/m<sup>2</sup> and a temperature change of -2×10<sup>6</sup>K. This generates a stress of 2000N/m<sup>2</sup> which corresponds to a tension of

$$2000\text{N/m}^2 \times 1\text{mm} = 2\text{N/m}.$$

You can confirm the 2000N/m<sup>2</sup> stress by changing the analysis type to Static 3D and solving.

All elements are constrained against movement in the in-plane directions to speed up solving and prevent spurious in-plane modes caused by the very low stiffness of the material.

Mode	Mode shape	Frequency
Mode 1 0 nodal diameters		2.007 Hz
Modes 2 and 3 1 nodal diameter		3.197 Hz
Modes 4 and 5 2 nodal diameters		4.282 Hz

## 11.23 Impeller.liml

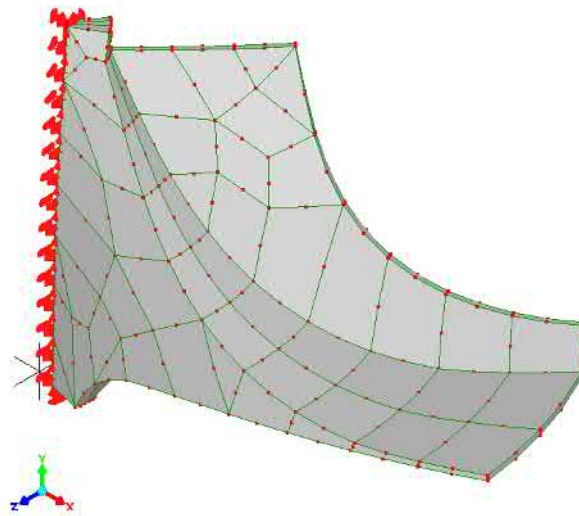
Analysis type: Modal Vibration 3D

Elements: Solid continuum (wedge15, hex20)

Loads and constraints: Fixed Support, Centrifugal Force, Cyclic Symmetry

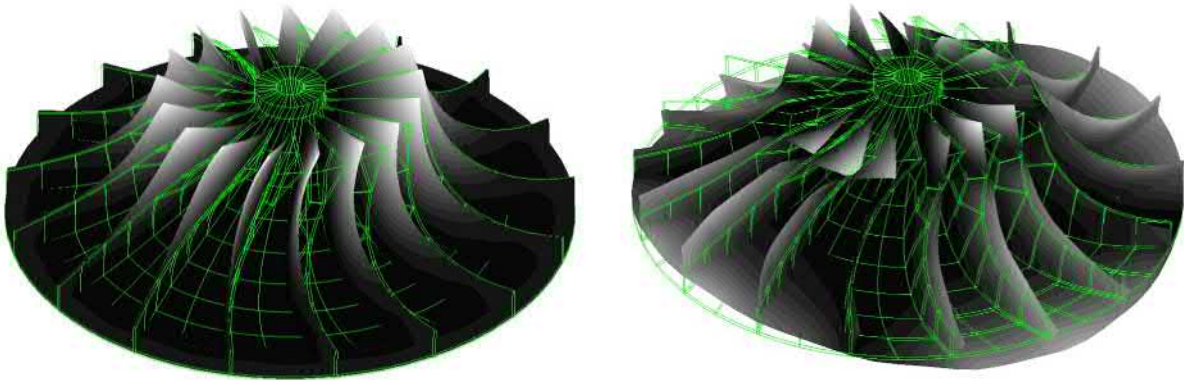
Vibration modes of a 17-blade, 100,000 rpm impeller are found. The frequencies of the modes depend on the speed of rotation because centrifugal force alters the stiffness through the effects of stress stiffening and spin softening.

The structure is composed of 17 identical segments so only one segment is modeled by taking advantage of cyclic symmetry. Since it uses the internal solver, the node patterns on both surfaces of the segment are the same. The CCX solver doesn't have this limitation.



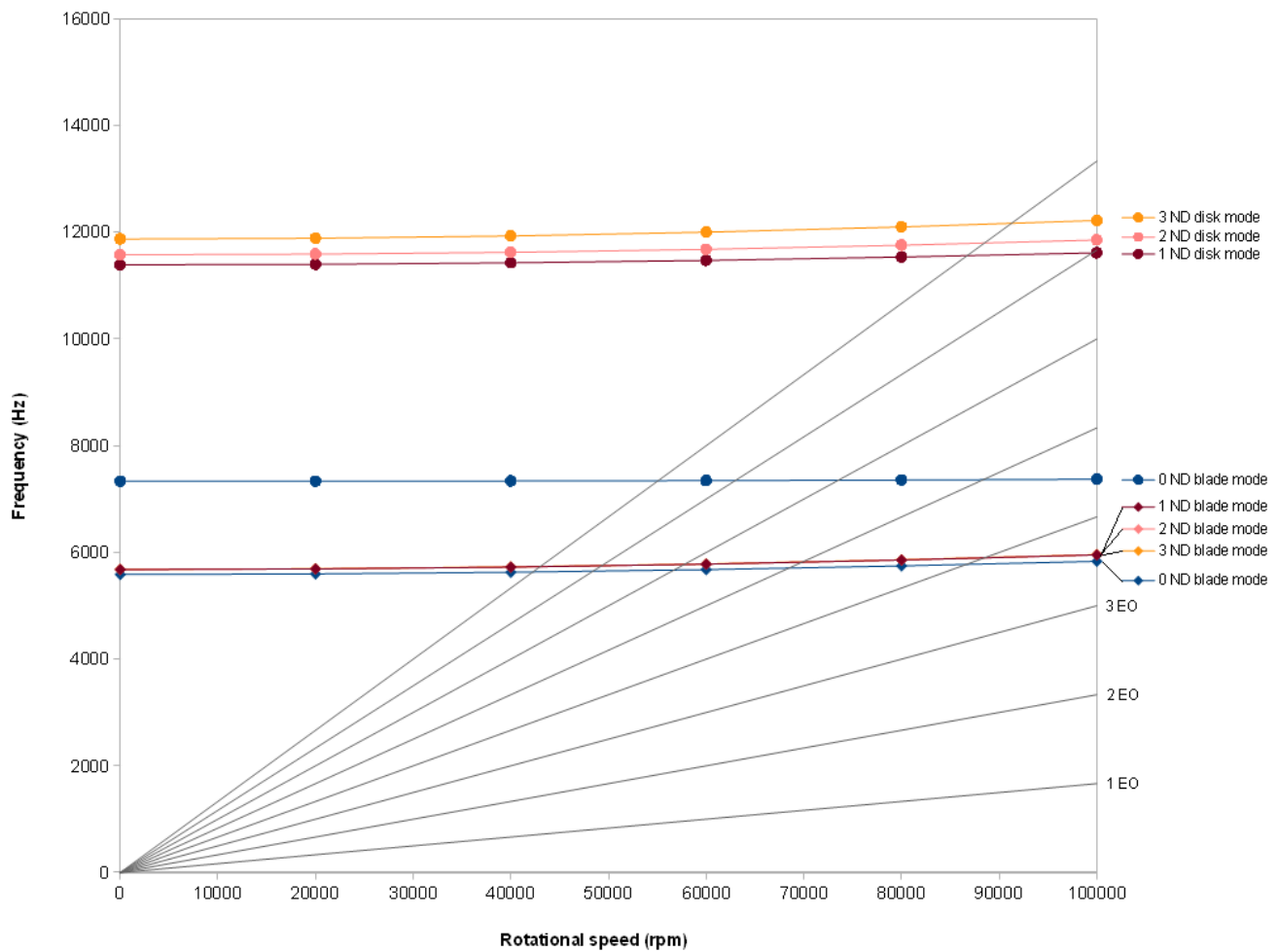
The material is steel ( $E=210\text{GPa}$ ,  $\nu=0.3$ ,  $\rho=7800\text{kg/m}^3$ ) and the inside surface is constrained in the rotating reference frame to represent a rigid shaft. Results may be inaccurate for a flexible shaft due to the Coriolis effect. Two mode shapes are shown below.





*The 1<sup>st</sup> blade mode. Deformation of each segment is identical with 0 nodal diameters.* *A disk mode with 2 nodal diameters*

The Campbell diagram below shows the first 2 modes for each of 0, 1, 2 and 3 nodal diameters (ND). It was made by solving the model 6 times, at 6 different rotational speeds.



## 11.24 VibratingString.liml

Analysis type: Modal Vibration 3D

Elements: Solid continuum (hex8)

Loads and constraints: Displacement, Force

We will model the transverse vibration of a straight flexible cable held in tension. The cable has the following properties:

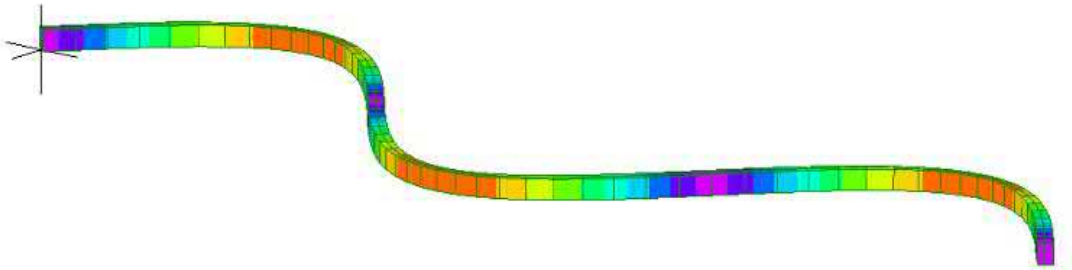
- Length:  $L=60\text{m}$
- Linear density:  $\rho=3\text{kg/m}$
- Tensile force:  $F=40\text{N}$
- Bending stiffness: Negligible

The solution is compared to hand calculation using the formula

$$f = n/2 (F/\rho)^{1/2}/L$$

where  $f$  is the frequency and  $n$  is the mode number of the standing wave. The Mecway model includes axial modes which are ignored here because they are a consequence of the arbitrary material stiffness.

Mode	$f$ (hand calculation)	$f$ (Mecway)
$n=1$	0.03043 Hz	0.03042 Hz
$n=2$	0.06086 Hz	0.06088 Hz
$n=3$	0.09129 Hz	0.09141 Hz
$n=4$	0.1217 Hz	0.1221 Hz



## 11.25 PandSWaves.liml

Analysis type: Dynamic Response 2D

Elements: Plane continuum (quad4)

Loads and constraints: Displacement, Force

Dynamic response analysis is used to model wave propagation through an elastic medium (rock). A sinusoidal force is applied at the center of an infinite domain. This generates pressure waves (P-waves) traveling outward in the direction of the force, and shear waves (S-waves) traveling outward perpendicular to the force.

Only one quadrant of the domain is modeled due to symmetry. There is mirror symmetry about the X axis which is enforced by constraining displacement in Y along the axes. There is also anti-symmetry in load and displacement about the Y axis which is enforced by constraining displacement in X along the Y axis too. Mecway has no infinite boundary condition so a finite boundary far from the source of the waves is used instead. The analysis is only run until the first waves reach the boundary so any reflections from the boundary are excluded.

The material's properties are:

Young's modulus  $E = 7 \text{ GPa}$

Poisson's ratio  $\nu = 0.2$

Density  $\rho = 3000 \text{ kg/m}^3$

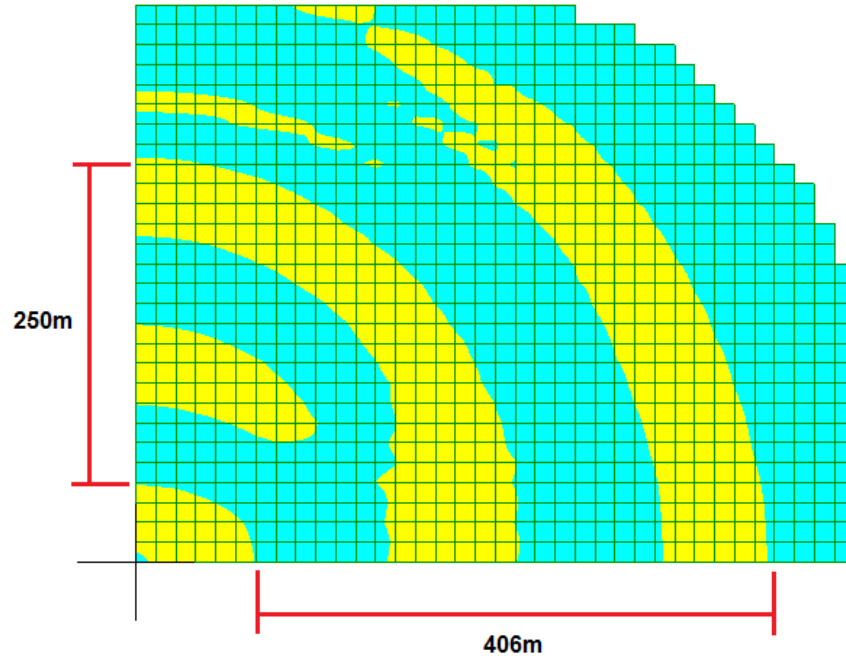
Shear modulus  $G = \frac{E}{2(1+\nu)} = 2.917 \text{ GPa}$

Bulk modulus  $K = \frac{E}{3(1-2\nu)} = 3.889 \text{ GPa}$

Displacement in X

2.221E-10  
1.527E-12  
-2.19E-10

Time = 0.3  
= 0.005 × 60



The contour plot above at time = 0.3s shows displacement in the X direction using only 2 colors (**Tools** → **Options** → **Contour plot**) so the wavelengths can be easily read off it. The tape measure tool is used to measure the average wavelength of two P-waves along the X axis as  $406\text{m}/2 = 203\text{m}$ . In the same way, the average wavelength of 2 S-waves measured along the Y axis is  $250\text{m}/2 = 125\text{m}$ .

The solution is verified by comparing the wavelengths of the P-waves and S-waves to their theoretical values.

The theoretical wavelengths are

$$\lambda_p = \frac{c_p}{f} \quad \text{and} \quad \lambda_s = \frac{c_s}{f}$$

where

$$f = \frac{50}{2\pi} \text{ Hz} \quad \text{is the frequency of the applied force}$$

$$c_p = \sqrt{\frac{\frac{4}{3}G + K}{\rho}} = 1610 \text{ m/s} \quad \text{is the speed of P-waves.}$$

$$c_s = \sqrt{\frac{G}{\rho}} = 986.0 \text{ m/s} \quad \text{is the speed of S-waves.}$$

This gives

$$\lambda_p = 202 \text{ m} \quad \text{and} \quad \lambda_s = 124 \text{ m}$$

which is within 5% of Mecway's results of 203m and 125m respectively.

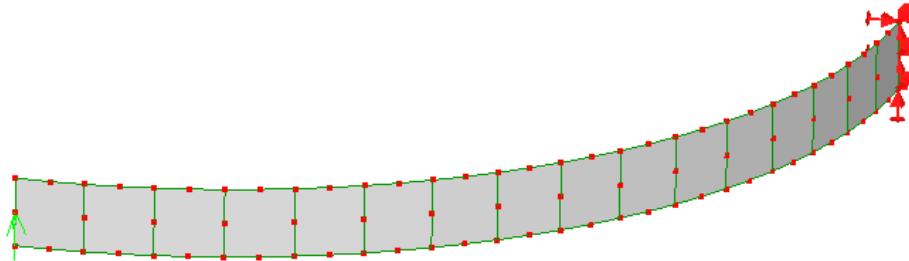
## 11.26 DampedVibratingStrip.liml

Analysis type: Dynamic Response 3D

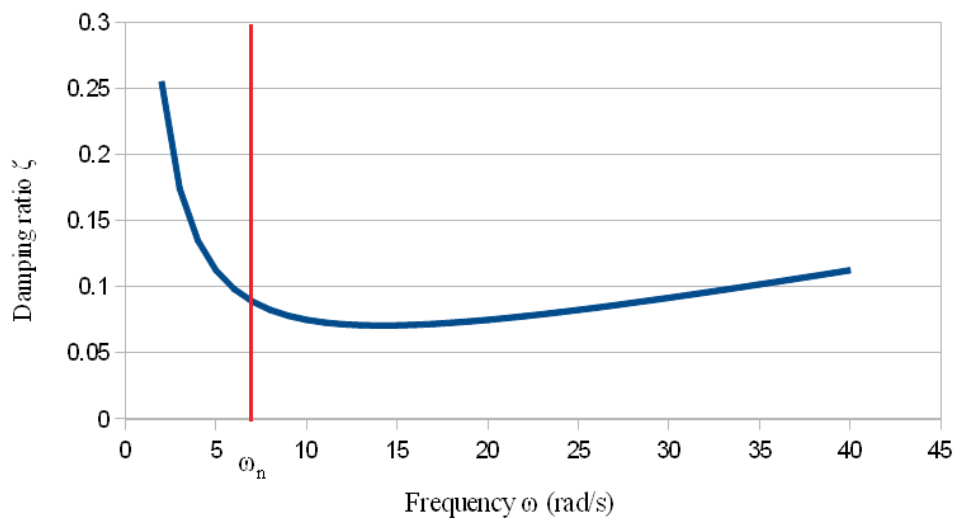
Elements: Shell (quad8)

Loads and constraints: Fixed Support, Force, Rayleigh damping

This sample shows Rayleigh damping applied to the vibration of a curved strip.



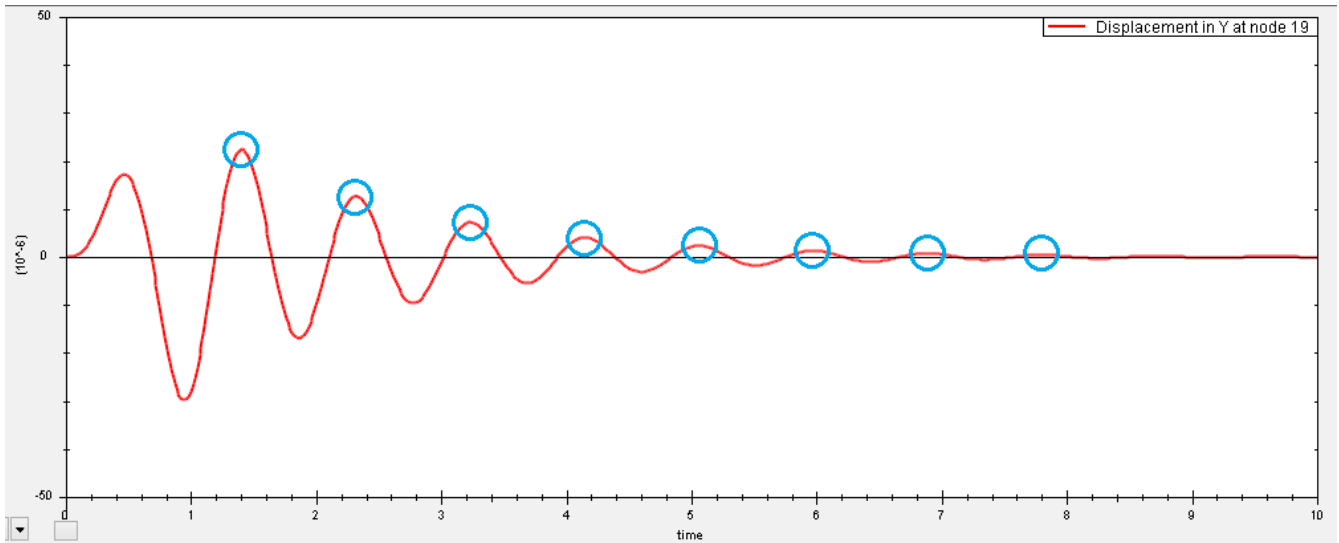
The load excites the lateral bending and twisting mode which has a natural frequency of  $\omega_n=6.91\text{rad/s}$ . The vibration is then allowed to decay by viscous damping. The Rayleigh damping constants are set to  $\alpha=1$  and  $\beta=0.005$  which give a damping ratio of  $\zeta=0.08978$  at  $\omega=\omega_n$ . Alternatively, this can be expressed as a Q factor of  $Q=\frac{1}{2\zeta}=5.569$ .



This damping ratio tells us that the amplitude of oscillation should decay with a time constant of

$$\tau = \frac{1}{\omega \zeta} = \frac{1}{6.91 \text{ rad/s} \times 0.08978} = 1.61 \text{ s}$$

Measurement of the time constant from the peaks of the displacement curve generated by Mecway also gives 1.61s.



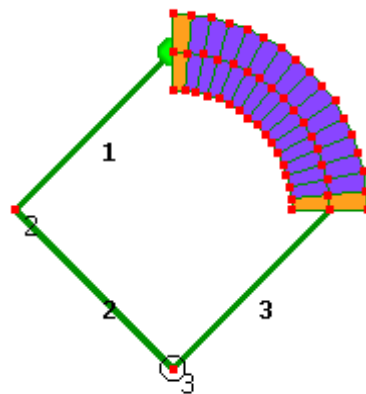
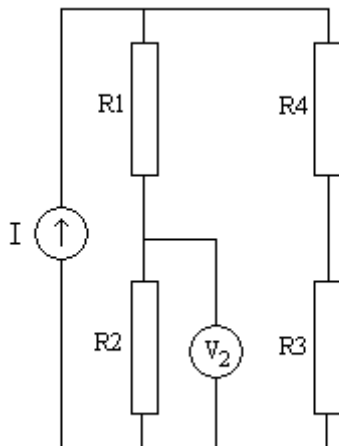
## 11.27 WheatstoneBridge.liml

Analysis type: DC Current Flow

Elements: Resistor (line2), Solid continuum (hex8)

Loads and constraints: Electric potential, Current

The Wheatstone bridge circuit shown below is modeled in Mecway to find the voltage  $V_2$  across  $R_2$  with an applied current of  $I=100\text{mA}$ . Three of the resistors are line2 resistor elements and  $R_4$  is modeled using solid elements having high-conductivity terminals to connect to the resistor elements. Node 3 is constrained to 0V and is the reference point for voltages at the other nodes.



$$R1=750\Omega$$

$$R2=1000\Omega$$

$$R3=500\Omega$$

$R4=1169.826\Omega$  which was found by integration of the curved shape.

Hand calculation shows that the voltage at node 2 is

$$V_2 = I \frac{(R1 + R2)(R3 + R4)}{R1 + R2 + R3 + R4} \frac{R2}{R1 + R2} = 48.828V$$

which matches Mecway's result of 48.823V. Most of the error is due to coarseness of the solid element mesh of R4 and can be eliminated by replacing it with a line2 resistor element having the resistance listed above.

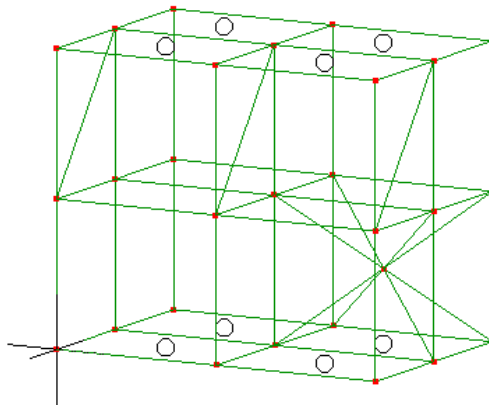
## 11.28 Capacitor.liml

Analysis type: Static 3D

Elements: Solid continuum (pyr5, wedge6, hex8)

Loads and constraints: Electric potential

The electric field between two plates of a capacitor will be modeled. The plates are  $d=0.25m$  apart with a dielectric ( $\epsilon_r=2.5$ ) in-between and a potential difference of  $V=1.5V$  across them. Each plate has an area of  $A=0.2m \times 0.3m$ .



The expected electric field is

$$E = V / d \\ = 1.5V / 0.25m = 6 \text{ V/m}$$

which is consistent with the results from Mecway of 6.000000 V/m on each node.

The expected energy stored in the capacitor is

$$U = 1/2 C V^2 \\ = 5.9766 \times 10^{-12} \text{ J}$$

where

$$C = A \epsilon_r \epsilon_0 / d$$

$\epsilon_0 = 8.854187818 \times 10^{-12} \text{ F/m}$ , the permittivity of free space

This corresponds to an energy density of  $U \times A \times d = 3.9844 \times 10^{-10} \text{ J/m}^3$ . The energy density given by Mecway is also  $3.9844 \times 10^{-10} \text{ J/m}^3$ .

Edge effects do not appear in this model because the mesh has no elements outside the area between the plates.

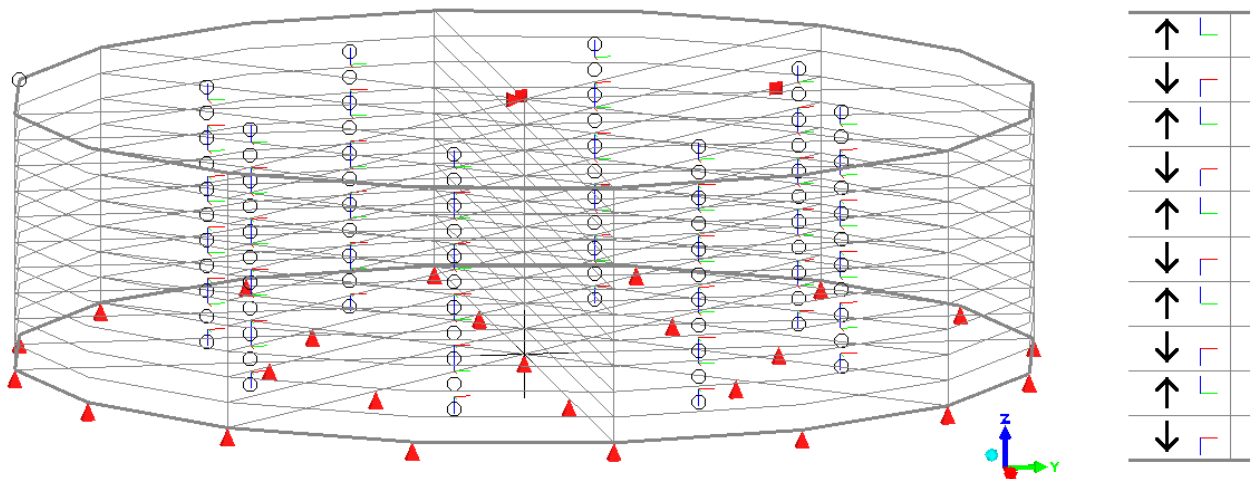
## 11.29 PiezoelectricStack.liml

Analysis type: Static 3D

Elements: Solid continuum (wedge15)

Loads and constraints: Frictionless support, Displacement, Pressure, Electric potential

A stack of 10 layers of piezoelectric ceramic disks is analyzed as both an actuator and a force sensor. The layers are oriented with alternating polarization directions of +Z and -Z. These are defined by the direction of the local W axis of each element. The base of the stack has a frictionless support to support the load on the sensor and keep the behavior 1-dimensional so it can be easily compared to hand calculations.



*Mesh. The black arrows indicate the polarization directions.*

Material properties for PZT-5A are:

Compliance matrix at constant electric field (short circuit)

$$s^E = \begin{bmatrix} 16.4 & -5.74 & -7.22 & 0 & 0 & 0 \\ -5.74 & 16.4 & -7.22 & 0 & 0 & 0 \\ -7.22 & -7.22 & 18.8 & 0 & 0 & 0 \\ 0 & 0 & 0 & 47.5 & 0 & 0 \\ 0 & 0 & 0 & 0 & 47.5 & 0 \\ 0 & 0 & 0 & 0 & 0 & 44.3 \end{bmatrix} \times 10^{-12} \frac{\text{m}^2}{\text{N}}$$



This shows that the material is transversely isotropic with the 3 axis being the axis of symmetry. In Mecway, this corresponds to the element's local W axis.

#### Coupling coefficient matrix d

$$\mathbf{d} = \begin{bmatrix} 0 & 0 & 0 & 0 & 584 & 0 \\ 0 & 0 & 0 & 584 & 0 & 0 \\ -171 & -171 & 374 & 0 & 0 & 0 \end{bmatrix} \times 10^{-12} \frac{\text{m}}{\text{V}}$$

Due to the transverse isotropy,  $d_{32}=d_{31}$  and  $d_{24}=d_{15}$  so some sources of material data may only include the 3 unique values  $d_{31}$ ,  $d_{33}$ , and  $d_{15}$ .

#### Permittivity matrix at constant stress (clamped)

$$\boldsymbol{\epsilon}^T = \begin{bmatrix} 1730 & 0 & 0 \\ 0 & 1730 & 0 \\ 0 & 0 & 1700 \end{bmatrix} \times \epsilon_0$$

#### **Actuator**

The sample file is configured to act as an actuator driven by a voltage source by having two sets of electric potential constraints on the faces between the layers. The strain is given by the actuator equation:

$$\mathbf{S} = \mathbf{s}^E \mathbf{T} + \mathbf{d} \mathbf{E}$$

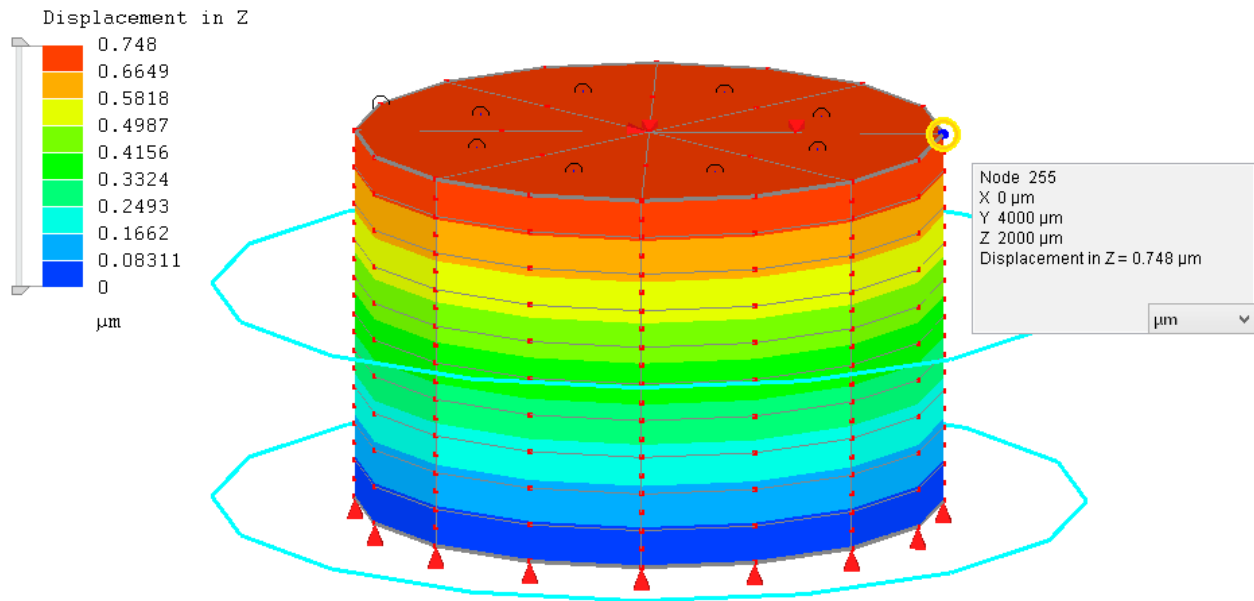
For 1D behavior, this becomes a scalar equation with

$$\text{Electric field } \mathbf{E} = E_W = 200 \text{ V} / 200 \text{ } \mu\text{m} = 10^6 \text{ V/m}$$

$$\text{Coupling coefficient } \mathbf{d} = d_{33} = 3.74 \times 10^{-10} \text{ m/V}$$

Stress  $\mathbf{T} = 0$  because there are no mechanical loads or constraints.

This gives strain  $S = 0.000374$ . The total length of the stack is 2 mm so its change in length should be  $2 \text{ mm} \times 0.000374 = 0.748 \text{ } \mu\text{m}$ . Mecway's solution agrees with this with Displacement in Z = 0.748  $\mu\text{m}$  at the top surface.



## Sensor

If you suppress the first two electric potential constraints and unsuppress the pressure load, the model becomes a sensor that produces an electric voltage across each layer in response to the applied pressure. The electric field is given by the sensor equation:

$$\mathbf{D} = \mathbf{d} \mathbf{T} + \boldsymbol{\varepsilon}^T \mathbf{E}$$

For 1D behavior, this also becomes a scalar equation with

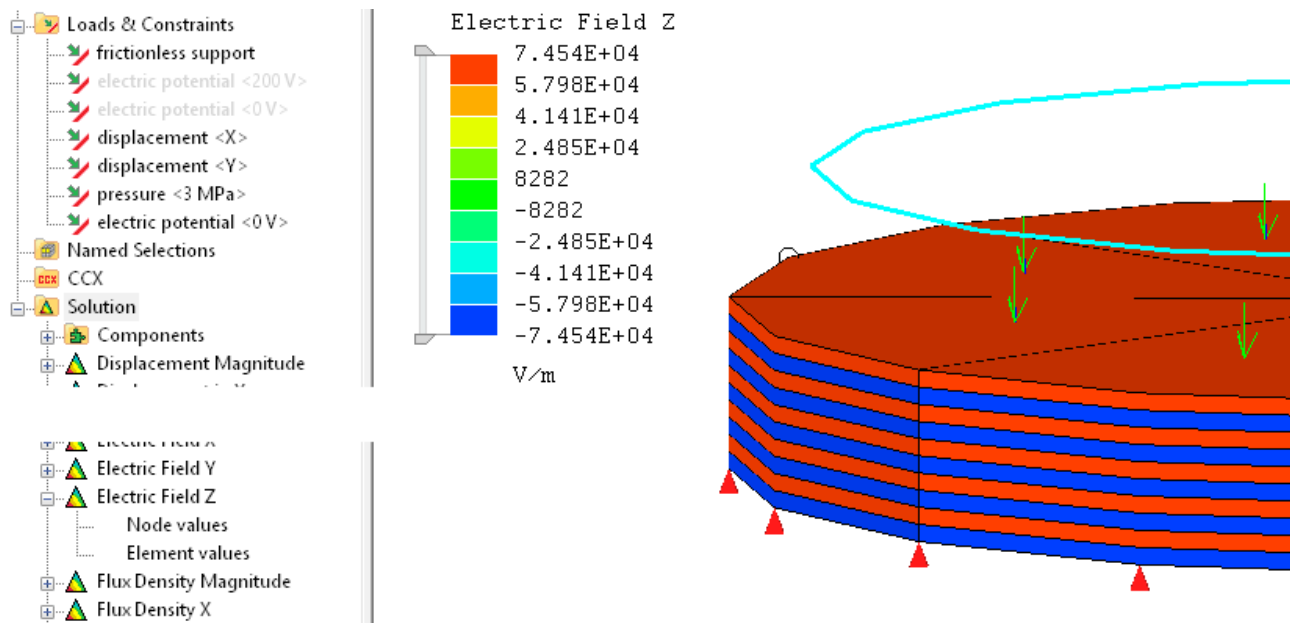
Electric displacement  $\mathbf{D} = 0$  because it's open circuit.

Coupling coefficient  $\mathbf{d} = d_{33} = 3.74 \times 10^{-10} \text{ m/V}$

Stress  $\mathbf{T} = -3 \text{ MPa}$

Permittivity at constant stress  $\boldsymbol{\varepsilon}^T = \boldsymbol{\varepsilon}_{33}^T = 1700 \times \varepsilon_0 = 1.505 \times 10^{-8} \text{ C/(Vm)}$

This gives electric field  $\mathbf{E} = 74500 \text{ N/C}$  (or  $\text{V/m}$ ) in the 3 direction. Mecway's solution agrees with this with Electric Field  $Z = 74541 \text{ V/m}$  on elements where the 3 direction is the same as the global Z axis and  $-74541 \text{ V/m}$  on reversed elements.



From the Z component of the electric field, we can find the potential difference across one layer as

$$V = -E d$$

On the top layer with thickness  $d = 0.2$  mm and positive  $E$ ,  $V = -14.9$  V which agrees with Mecway's result of 14.91 V at the bottom of the layer and 0 V at the top.

## File Formats

Mecway can open and save a range of file formats to inter-operate with other software. Some of these (dxf, msh, stl, vol and xyz) do not contain physical units so you will be asked to choose the length unit when opening them.

To export the deformed mesh from a solution, use **Mesh tools -> Transfer displacements from solution...** to deform the modeler's mesh, then export it.

### 12.1 Liml

This is the native file format for Mecway. It uses XML which is a human readable text format. A simple example of a file containing two nodes and one beam element is

```
<liml version="3">
  <node nid="1" x="0" y="0" z="0" />
  <node nid="2" x="1" y="1" z="0" />
  <elset>
    <elem eid="1" shape="line2" nodes="1 2" />
  </elset>
</liml>
```

Numbers without units are either dimensionless, such as node numbers shown above, or are in SI units, such as the node coordinates shown above which are in meters regardless of the unit used to display them in Mecway. The unit symbols are either included with after the number or, in the case of formulas, appear as separate XML attributes.

### 12.2 STEP (.step/.stp)

Mecway can open a STEP file containing a single solid body. When you open a STEP file, it appears in the **Geometry** branch of the outline tree. You can right click this item for the menu to generate a mesh. See Chapter 5 - *CAD models* for more details. Mecway cannot save to this format.

### 12.3 DXF

Mecway can read some features from the ASCII version of DXF R14 files. Nodes at the same locations are automatically merged together. Curves are converted to line3 elements which are parabolic curves so they cannot exactly represent the original geometry but are close enough for many purposes. If you refine these curved line3 elements or use **Automesh 2D** with them, they retain their curvature. Layers are converted to components with the 0 layer becoming the default component. Mecway cannot save to DXF format.

POINT – becomes a node

LINE – becomes a line2 element

LWPOLYLINE – straight segments become line2 elements and arc segments become strings of line3 elements in the same way as ARC.

CIRCLE – becomes 8 line3 elements.

ELLIPSE – becomes a string of line3 elements with each element spanning approximately 45 degrees.

ARC – becomes a string of uniformly sized line3 elements with each element representing an arc of approximately 45 degrees.

3DFACE – becomes a tri3 or quad4 element.

INSERT – imports the contents of a BLOCK. Rows and columns are ignored.

BLOCK – the contents are imported into Mecway with each associated INSERT entity but the block definition is discarded.

## 12.4 STL

An STL file contains a list of triangles in 3D space. Each triangle is imported to Mecway as a tri3 element. Coincident triangle vertices are automatically merged into the same node to form a continuous mesh. You can also convert these surface triangles into a volume mesh using **Mesh tools** → **Automesh 3D**.

The STL files generated by CAD applications often contain triangles with very large aspect ratios (long and thin). These are usually unsuitable for use as finite elements so you should use **Automesh 3D** to improve their shape.

## 12.5 Gmsh (.msh)

Gmsh is an open source automesh. You can use it to generate meshes then import them into Mecway. Mecway can read the ASCII file format versions 2.0, 2.1, 2.2, 4.0, and 4.1. If the file contains no solution data then Mecway assumes it is a mesh generated by gmsh and imports only the elements with the highest dimension – for example if there are both volume and surface elements, only the 3D elements appear in Mecway. This is because Gmsh uses surface and line elements to label the surfaces and edges of solids.

Geometric entities of surface elements are used to create named selections for surfaces in Mecway. If a volume element has a surface element on its face, then the geometric entity ID of that surface element is used to assign the volume element's face to a corresponding named selection. This preserves the distinct surfaces from the original geometry.

Geometric entities of volume elements are converted to components in Mecway. This allows you to use multiple materials in the same mesh with clearly defined boundaries between them.

Mecway can also write a mesh without solution data to Gmsh files. Components are stored as physical and geometric entities. 2D faces in face selections are stored as surface elements with physical and geometric entities to identify them.

## 12.6 UNV

You can open .unv mesh files generated by software such as Salome, Gmsh, and NX. Mecway reads nodes (dataset 2411), elements (dataset 2412), permanent groups (datasets 2435, 2452, 2467, 2477), units (dataset 164), and physical property table names (dataset 2470).

## 12.7 Netgen (.vol)

Netgen is an open source automesh. You can use it to generate meshes then import them into Mecway. Most of the functionality of Netgen is already included with Mecway so you typically don't need to use this. Mecway only reads the elements with the highest dimension – either volume or surface elements. This is because Netgen uses surface elements to label the surfaces of solid parts.

## 12.8 Polygon File Format (.ply)

Mecway can open surface meshes consisting of tri3 and quad4 elements in both the ASCII and binary versions of this file format.

## 12.9 JPEG, PNG, BMP

You can import images to display alongside the model. See the section *Tracing an Image* for more details. You can also save screenshots as PNG files.

## 12.10 XYZ

An XYZ file is a text file containing a list of points in 3D space. Each line has the 3 coordinates of a point. The coordinates can be delimited by whitespace or commas. Lines which contain text or not exactly 3 numbers are usually ignored. An exception is a single word followed by 3 numbers, which allows molecule geometry XYZ files to be opened. Mecway can open XYZ files but not save them.

## 12.11 ANSYS command file (.txt)

Mecway can save some parts of the model as ANSYS APDL commands. These are:

- Nodes: N
- 3D solid mechanical elements: SOLID185 and SOLID186
- 3D solid piezoelectric elements: SOLID5 and SOLID226
- Displacement constraints: D and NANG
- Electric potential constraints: D
- Isotropic material: MP,EX and MP,PRXY
- Anisotropic stiffness matrix: TB,ANEL and TB,DATA
- Anisotropic compliance matrix: TB,ANEL and TB,DATA
- Anisotropic permittivity matrix: TB,DPER and TB,DATA
- Piezoelectric stress matrix (e): TB,PIEZ and TB,DATA
- Piezoelectric strain matrix (d): TB,PIEZ and TB,DATA
- Pressure (normal, uniform and constant): SFE

## 12.12 CalculiX (.inp)

### 12.12.1 Opening .inp files

The following keywords and parameters are supported.

Keyword	Parameters	Notes
---------	------------	-------

*BEAM SECTION	GENERAL ELSET MATERIAL SECTION=PIPE	
*BEAM SECTION	ELSET MATERIAL SECTION=BOX / CIRC / I / L / PIPE / THICKPIPE / RECT	
*BOUNDARY		Displacement, rotation, fluid pressure, electric potential and temperature
*CFLUX		
*CLOAD		
*CONDUCTIVITY	TYPE=ISO / ORTHO	Can be temperature dependent with TYPE=ISO
*DASHPOT	ELSET	
*DENSITY		
*DLOAD		P, Pn load types for 2D/3D solids and shells. Not beams.
*DYNAMIC	DIRECT DIRECT USER CONTROL	
*ELASTIC	TYPE=ENGINEERING CONSTANTS TYPE=ISOTROPIC	
*ELEMENT	TYPE ELSET INPUT	
*ELGEN	ELSET	
*ELSET	ELSET GENERATE	
*END STEP		
*FILM		Face numbers on shells can use the Abaqus (FPOS, FNEG) or CalculiX (F1-



		F6) forms.
*FREQUENCY		Number of eigenvalues on data line
*HEAT TRANSFER	STEADY STATE	
*INCLUDE	INPUT	It cannot appear within another keyword block.
*MATERIAL	NAME	
*MEMBRANE SECTION	MATERIAL ELSET	
*NCOPY	OLDSET CHANGENUMBER NEWSET SHIFT MULTIPLE	
*NFILL	NSET BIAS TWO STEP	
*NGEN	NSET LINE=P / C	
*NODE	NSET SYSTEM=C / S / R INPUT	
*NSET	NSET ELSET GENERATE UNSORTED	
*ORIENTATION	NAME	Only a rectangular system defined by two or three points' coordinates and on 3D solid and shell elements. For shell elements, Abaqus and CalculiX assign different meanings to the coordinates. If either a 3rd point or a 2nd data line is specified then Mecway will interpret it as Abaqus format, otherwise it will interpret it

		as CalculiX format.
*PHYSICAL CONSTANTS	ABSOLUTE ZERO	
*RADIATE		Face numbers on shells can use the Abaqus (FPOS, FNEG) or CalculiX (F1-F6) forms.
*SHELL SECTION	MATERIAL ELSET	
*SOLID SECTION	MATERIAL ELSET ORIENTATION	
*SPECIFIC HEAT		Can be constant or temperature dependent
*SPRING	ELSET	
*STATIC	DIRECT	
*STEP	NLGEOM	
*SURFACE	NAME TYPE=NODE / ELEMENT	
*TRANSFORM	NSET TYPE=C / S / R	

Element types are converted to various element shapes, material properties and analysis types according to the table below. If there are conflicts such as a mixture of 2D and 3D elements or no compatible analysis type then only one will be chosen. Some element types are not supported by Mecway's solver but will still be imported. These include 3-node beams, 3-node shells, hexahedra collapsed into prisms, and axisymmetric shells. Other element shapes don't exist in Mecway, such as some of the cohesive (COH), gasket (GK) and cylindrical (CCL) element types. These will be converted to nearest available element shape without adding any extra nodes.

Element type in inp file	How it appears in Mecway
T2D2,...	2D truss
T3D2,...	3D truss, 2D beam, 3D beam or fin depending on the analysis procedure

SPRINGA	3D beam with spring material property
DASHPOTA	3D beam with damper material property
CPS3,...	2D continuum element with plane stress material property
CPE3,...	2D continuum element with plane strain material property
CAX3,...	Axisymmetric
S8R,...	3D shell. The local element axes are determined by an associated *ORIENTATION block if present. Otherwise the default is U is parallel to X unless the element's normal direction is within 0.1 degrees of the X axis, in which case U is parallel to Z.
C3D8,...	3D solid

### General Limitations

- All text is case insensitive, including labels in quotation marks.
- All spaces are ignored. This means filenames and labels in quotes may not be read correctly.
- Keywords and most parameter values must be complete although parameter names can be shortened, for example ISO instead of ISOTROPIC.
- If node or element numbers are not contiguous, they'll be renumbered so that they are.
- All physical quantities are assumed to be in SI units (m, kg, s, K) except angle which may be in degrees or radians according to the file format's specification. An exception is that you can choose the units of node coordinates when opening the file.

### **12.12.2 Saving .inp files**

This file is intended for use with the CCX solver and while it is also mostly compatible with Abaqus, some data is interpreted differently.

All quantities are saved in SI units (kg, m, s, K) and radians. So if a length is 1 inch in Mecway, it will appear as 0.0254 in the .inp file.

### **12.13 CalculiX results (.frd)**

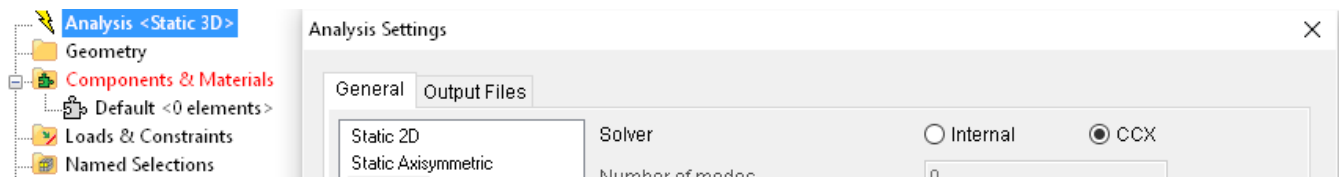
Mecway can read most forms of this file as generated by CCX versions 2.8-2.17. It assumes all quantities are in SI units (m, kg, s). This requirement will be met if the .frd file is the solution to a .inp file generated by Mecway regardless of the units used to define the model in Mecway.

## CalculiX Solver

In addition to Mecway's built in solver, you can also use the open source CalculiX solver (CCX). This gives access to more advanced features like frictional contact and plastic material behavior.

CCX 2.17 is installed along with Mecway and ready to use. If you prefer to install another version then install it and go to **Tools** → **Options** → **CalculiX** and **Browse** to the location of the solver's executable **ccx.exe**. In either case, if you prefer to keep the automatically generated files in your own working directory, check **Use the same directory as the LIML file**, otherwise temporary files will be created in a temporary folder and deleted when you close Mecway.

Change the model's solver option from **Internal** to **CCX** under **Analysis settings**. When solving, Mecway will generate a .inp file, run the CCX solver with this filename as a parameter then read the .frd file it generates and display the solution.



If the solve fails, click the **CCX output** button to show CCX's standard output and standard error messages. The **Errors** button also shows error messages generated by Mecway while it's writing the .inp file. You can also inspect the .inp file's contents with the **Inp file** button.

Units are converted to a consistent system (m, kg, s) as required by CCX. This means you can use any combination of units while building the model in Mecway and the solution will also have correct values which can be displayed in any of the available units.

Mecway sets the `MKL_PARDISO_OOC_MAX_CORE_SIZE` and `MKL_PARDISO_OOC_PATH` environment variables for the CCX process. These enable the out-of-core feature of MKL PARDISO with the version of CCX that is included as source code.

A branch called **CCX** appears in the outline tree when the **Solver** is set to **CCX**. By right clicking on this, you can control how the .inp file is generated. The options are:

### 13.1 Custom model definition

Here you can enter text that will be inserted into the .inp file at the end of the model definition section immediately before the first step section (identified by a \*STEP keyword). You can use it to manually add different elements, materials, homogeneous (zero) boundary conditions, etc. that are not available in Mecway. See the CCX documentation for details of the keywords.

For example, to define a compression-only material with Youngs modulus of 30 GPa and maximum tensile stress of 3 MPa, enter the following text in the **custom model definition**:

```
*SOLID SECTION, ELSET=Default, MATERIAL=COMPRESSION_ONLY_1
*MATERIAL, NAME=COMPRESSION_ONLY_1
*USER MATERIAL, CONSTANTS=2
30e9, 3e6
```

The elements must belong to a component named *Default* with no material assigned to it in Mecway.

## 13.2 Custom step contents

**Custom step contents** is similar to **custom model definition** but the text is placed at the end of a step block (identified by \*STEP). You can use it to define loads, output variables, non-homogeneous boundary conditions or additional step sections.

For analysis types that generate two step blocks, the text is placed in the step block containing loads.

## 13.3 Modify keyword

Sometimes the .inp file cards generated by Mecway may not have the same options that you want. If this happens, you can use **modify keyword** to alter all instances where that keyword would normally be generated by Mecway. Available modifications are:

**Omit key word block.** Prevents the keyword card and its associated data lines from being generated. This was called **Don't generate keyword** in earlier versions of Mecway.

**Set parameter.** Sets a parameter to a value. For example, to use mortar contact, set the parameter with **Name TYPE** to the **Value MORTAR** on the **\*CONTACT PAIR** keyword.

**Omit parameter.** Prevents the specified parameter being generated.

## 13.4 Don't generate STEP

Add this item to prevent the \*STEP section(s) being generated by Mecway. It's useful if you're replacing it with your own \*STEP section defined in **custom model definition**.

## 13.5 No extra contact step

Using this compatibility item may cause incorrect results for quasi-static contact. For more details, open it in Mecway.

## 13.6 Custom element type

This allows you to change the element type used for the CCX solver to something which isn't otherwise supported by Mecway. The specified element type is applied to all elements that match the specified shape and are in the selection that **custom element type** is applied to.

For example, to use reduced integration for all hex8 elements, add a **custom element type** with:

<b>Apply to</b>	<Whole model>
<b>Element shape</b>	Hex8
<b>CCX element type</b>	C3D8R

## Automation

### 14.1 Command Line Parameters

Command line parameters can be used to run Mecway from another application, a batch file or a shell script.

```
mecway [<filename> [solve | clearsolutions | script <script filename>]]
```

The optional <filename> is a file of any of the mesh or geometry types that Mecway can open. It causes Mecway to start with that file open. The file name must include the extension such as .liml, .stp, etc. For example:

```
mecway mymodel.liml
```

The optional **solve** parameter causes Mecway to solve the file then exit without displaying the main window. To obtain the solution, set an output file in the model, such as **Write solution to liml file** or a table with **Write to csv file after solving**. When you use the **solve** parameter, <filename> can contain wildcards. For example to solve all liml files in the current folder, run:

```
mecway *.liml solve
```

**clearsolutions** is useful to reduce the size of a large file by removing its solution data.

**script** opens the file <filename>, runs the script <script filename>, then exits. If <filename> defines multiple files by using wildcards then it runs the script after open each file.

### 14.2 Scripts

You can write scripts that appear as menu items and are executed by a built-in implementation of Python 2.7 (IronPython). The approximate scope of the functionality is currently:

- Simple operations on nodes and elements
- Material properties
- Numerical values in most loads and constraints
- Read data from the solution
- Measurement tools that integrate over elements or faces
- Read the current selection
- Read and modify the elements in components
- Solve (with no progress indication)
- Save

- Python standard library
- Importing modules written in pure Python

It does not include:

- Meshing tools
- Named selections
- Importing modules that require CPython, such as NumPy

To create a script, click **Tools** → **Scripts...** and press the **New** button. Enter a non-existent file name to be created or choose an existing one. The editor (default is Notepad) then opens where you can write and save the script. Here is a simple example that creates a node and displays a message in a pop-up box:

```
node_id = mw.new_node(Vector(1.2, 3.4, 5.6))
mw.message("Node " + str(node_id) + " created.")
```

To run a script, click its menu item in the **Tools** menu. You can also activate it from the keyboard using **Alt+T** followed by the first letter of the script's name or run it with **File** → **Open...**, **File** → **Import...** or **File** → **Import from library**.

The API to Mecway consists of the following objects and classes which are automatically available in every script:

- mw** Object containing functions that operate on the application and the model in the modeler
- solution** Object containing functions that operate on the solution in the active configuration
- FaceId** Class for identifying a face of an element
- Vector** Class for representing a 3D vector

Details are listed in a separate document accessible through **Help** → **API reference**.

Two sample scripts are provided in the **Samples** folder where Mecway is installed. They are **generate\_mesh.py** which creates an element, and **transfer\_displacements\_from\_solution.py** which reads displacement data from the solution and uses it to deform the mesh.

All physical quantities are expressed in the SI unit system (m, kg, s, K, A, radian) regardless of how they were specified in the model. For example, if your model has a node with a coordinate of 1 inch, then reading that using a script will return the value 0.0254.

Cautions:

- Scripts can contain malware so avoid running unknown scripts from untrusted sources.
- The application will be unresponsive while a script is running so avoid infinite loops.
- The API, Python version, and Python implementation may change in future versions of Mecway, breaking compatibility with existing scripts. However, you can continue to use old major versions of Mecway alongside newer ones if you don't want to update your scripts.

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That's all there is to it!

## 15.7 OCC CAD Kernel

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