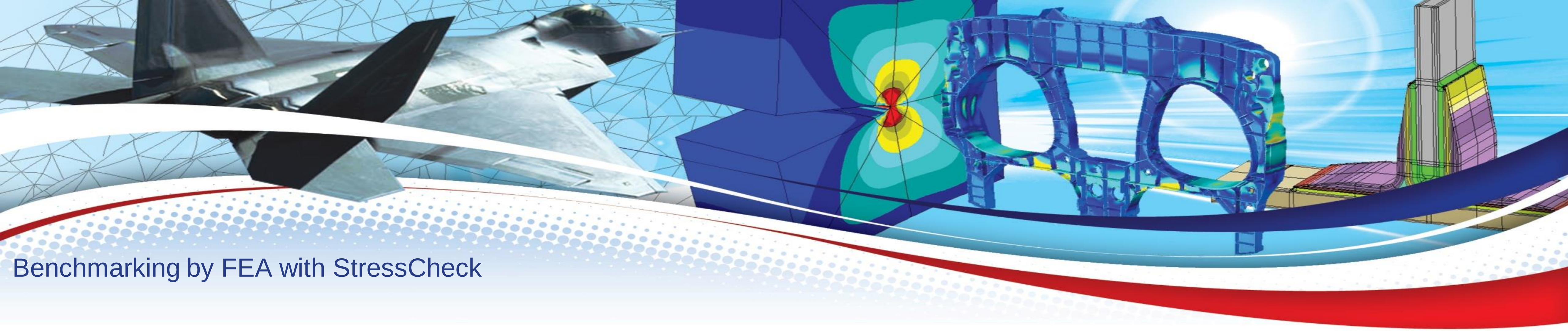
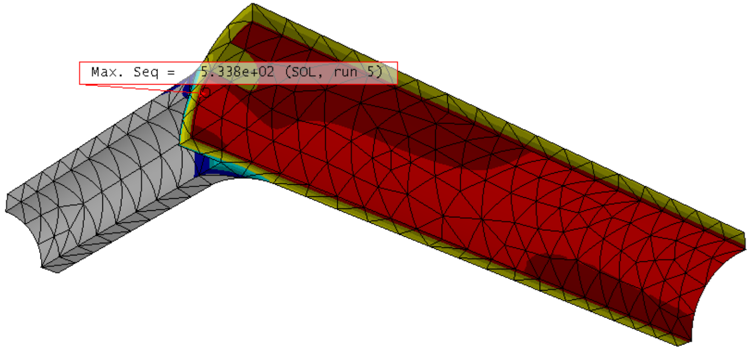
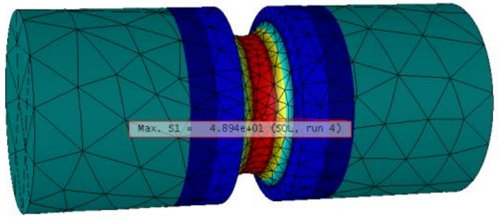


January 2020 NAFEMS Linear Benchmarks: **StressCheck Solutions**

ESRD, Inc.

2020





Outline

- In January 2020, NAFEMS Benchmark magazine published an article by Technical Officer Ian Symington titled “Designer Oriented Software - Is it Accurate?”
  - The goal of the article was to compare the results of several designer oriented software tools with the target solutions for a set of eight (8) linear benchmarks.
    - About the target solutions (via Symington): “The target solution used for most of the benchmarks in this study has been produced using the traditional Finite Element Analysis (FEA) approach. Confidence in the target solutions is gained using mesh convergence studies. A number of the benchmark examples reference an analytical solution, this information has been included to provide further confidence in the target solution.
  - The download of the article may be found here.
- StressCheck Professional was then used to solve the NAFEMS benchmarks via p-extension on automeshed tetrahedra, with the minimum discretization (i.e. mesh density and displacement polynomial orders) required to show convergence of the data of interest (e.g max von Mises stress, Seq, max 1st principal stress, S1, spring rate K, or natural frequencies) **within 1% estimated error**.


Designer Oriented Software - *Is it Accurate?*

Ian Symington, NAFEMS

When I started my first job in the CAE industry 18years ago there were already well-established products enabling designers to leverage simulation to improve their products. Over theyears the number of standalone designer oriented analysis tools has declined. Designer oriented analysis tools are now commonly includedwithin the CAD environment andwe have seen organisations successful democratise their CAE capability to design teams by providing template-based customisations/apps. This trend has changed recently and we have started hearing a lot of noise about two new designer oriented analysis tools, SimSolid from Altair and Discovery Live from ANSYS.

Before taking a dive into these products to seewhat they can do, it is useful to define what I mean by a “designer oriented” analysis tool.

- Firstly I'm looking for a tool that is easy to use and needs little specialist knowledge, I don't expect the user to have to worry about selecting an appropriate element type or spend hours conducting mesh refinement studies.
- The tool needs to be “quick to run”, an analysis tool that takes hours or days to run is of no use to the designer who needs near instant answers to guide the direction of development. And by “quick to run” I'm not just talking about how well the solver performs, I'll looking for a tool that handles thewhole simulation process quickly and a big part of this is how tolerant the tool is to poor geometry.
- The tool needs to be accurate enough to correctly inform the direction that a design should take. The software doesn't need the level of accuracy that can be achieved by a full blown general purpose analysis package but the designer, and everyone using the designer's input, must have confidence in the tool.

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Benchmark 1: Pressure Component

- Symington: “The pipeline is to be analysed in order to determine if the component is appropriately designed to withstand a 100MPa pressure load. The goal of the benchmark is to predict the peak Von Mises stress in the component.”
- StressCheck solution (1/8th domain, symmetry boundary conditions):
  - 792 geometrically mapped tetrahedra, p=2 to 6, <1 minute run time
  - Max Seq converges to **534 MPa (with 0.45% estimated error)**

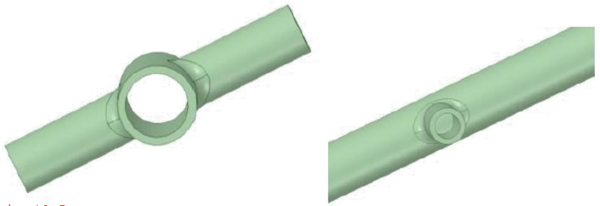
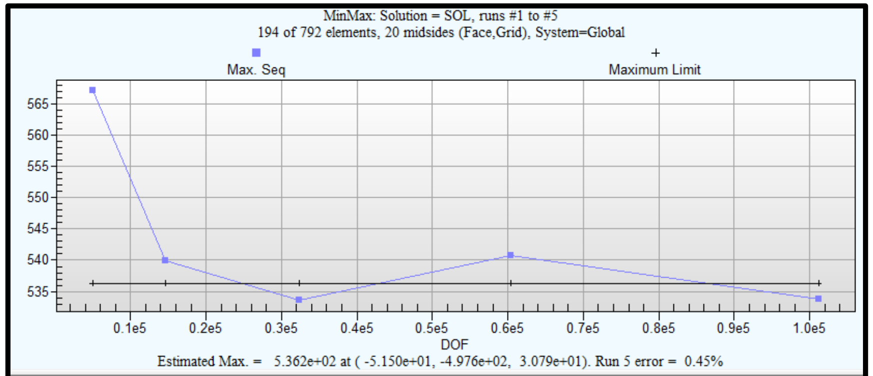
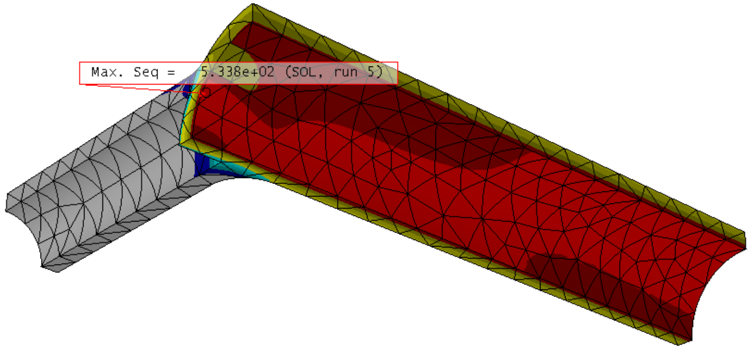


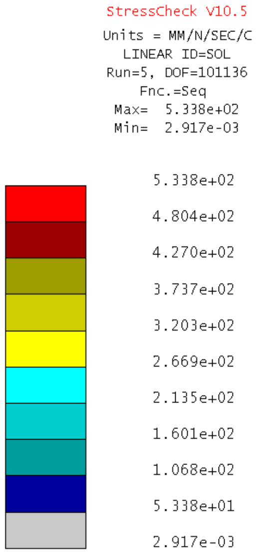
Image courtesy NAFEMS

Target max Seq: 534 MPa



Run	DOF	Max. Seq	Maximum Limit	X (Global)	Y (Global)	Z (Global)
1	4944	5.672e+02	5.362e+02	-2.620e+01	-8.407e+01	5.398e+01
2	14640	5.399e+02	5.362e+02	-5.271e+01	-4.951e+02	2.867e+01
3	32320	5.335e+02	5.362e+02	-5.196e+01	-5.000e+02	3.000e+01
4	60360	5.408e+02	5.362e+02	-5.220e+01	-4.943e+02	2.959e+01
5	101136	5.338e+02	5.362e+02	-5.150e+01	-4.976e+02	3.079e+01





Benchmark 2: Coil Spring

- Symington: “Benchmark 2 tests the ability of the package to predict the compliance of a coil spring. The coil spring is a challenging geometry to mesh using a traditional FEA approach. The challenge, target solution and results of the respondents are presented in the NAFEMS Benchmark Article “How Confident Are You?”
- StressCheck solution (same as the reference for the NAFEMS solution):
  - 9691 quadratically mapped tetrahedra, p=2 to 5, 2 minute run time
  - Spring rate converges to **20.83 N/mm (with 0.05% estimated error)**

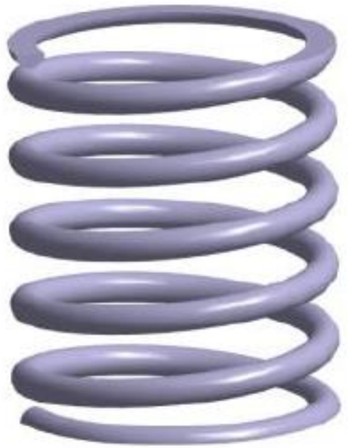


Image courtesy NAFEMS

p	N	U (Nmm)	% error
2	58,67	6558.910	0.78
3	173,989	6515.587	0.12
4	385,200	6512.737	0.07
5	721,473	6511.252	0.05
∞	∞	6507.872	–

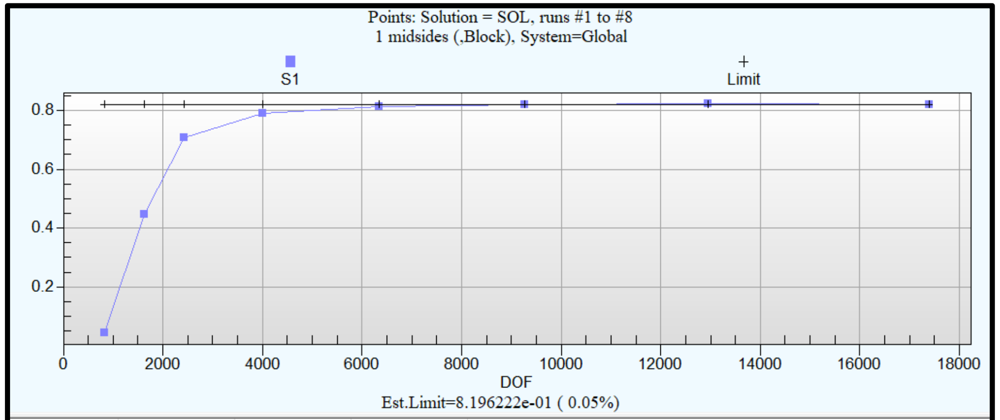
Image courtesy NAFEMS/Dr. Barna Szabó

U = 6507.87 N-mm, Δ = -25 mm
K = $2 * U/\Delta^2$ = 20.83 N/mm

Target spring rate:
K = 20.83 N/mm

Benchmark 3: Skew Plate – Bending Dominated

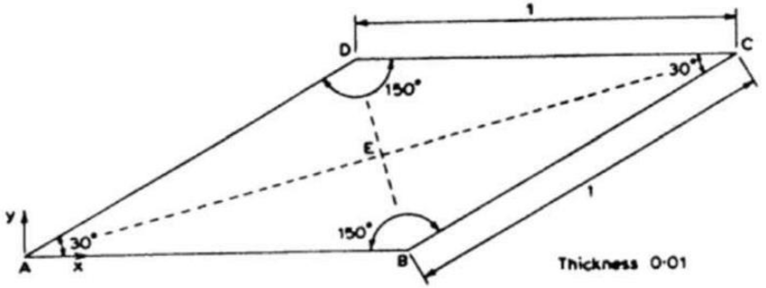
- Symington: “The third benchmark tackles a thin skewed plate. The skewed plate geometry is designed to introduce complexities in the discretisation process and may introduce distorted elements in an automatic meshing process. The thin skewed plate is simply supported and loaded with a uniform pressure. This benchmark is taken from the NAFEMS Linear Static Benchmarks Volume 1, test number IC 13.”
- StressCheck solution:
  - 64 geometrically mapped quads, p=1 to 8, <10 seconds run time
  - Max S1 converges to **0.820 MPa (with 0.05% estimated error) at point E on lower surface**



Run	DOF	S1	Limit
1	823	4.460e-02	8.196e-01
2	1623	4.457e-01	8.196e-01
3	2423	7.077e-01	8.196e-01
4	3997	7.914e-01	8.196e-01
5	6345	8.141e-01	8.196e-01
6	9266	8.191e-01	8.196e-01
7	12949	8.221e-01	8.196e-01
8	17394	8.201e-01	8.196e-01

Target max S1: 0.802 MPa

Image courtesy NAFEMS



More details on the StressCheck solution can be found here: https://www.esrd.com/product/standard-nafems-benchmarks-linear-elastic-tests/

Benchmark 4: Plate with Hole - Kt

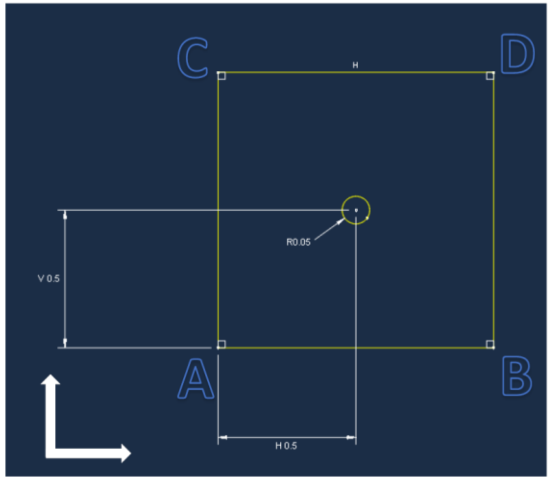
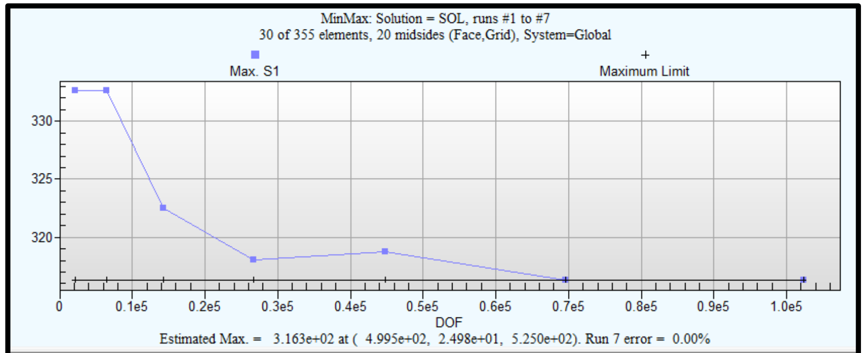


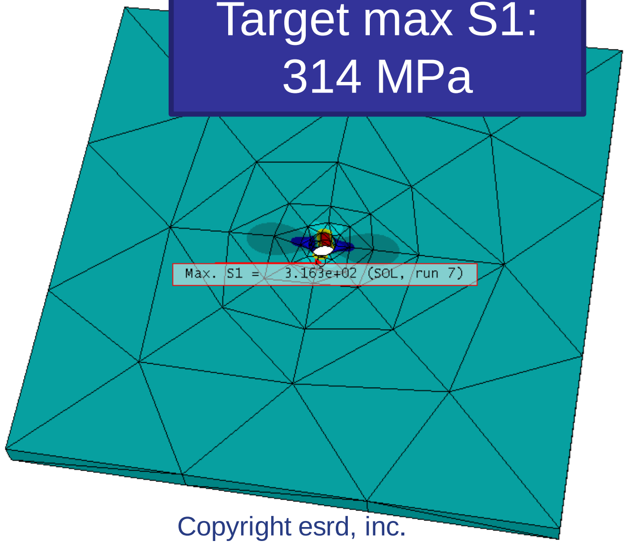
Image courtesy NAFEMS

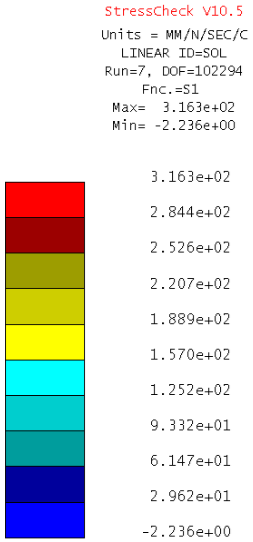
- Symington: “This benchmark tests the ability of the code to capture a stress concentration (Kt) in a plate containing a small hole. The benchmark has been designed so that the extent of the plate is large in comparison to the size of hole so as to pose a challenge when sizing the mesh in the vicinity of the stress concentration.”
- StressCheck solution:
  - 355 geometrically mapped tetrahedra, p=2 to 8, 1 minute run time
  - Max S1 converges to **316 Mpa (with 0.00% estimated error)**

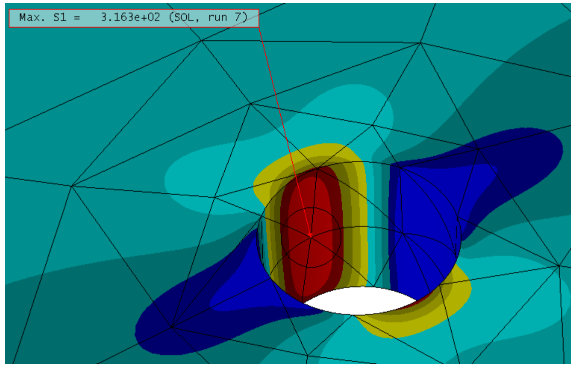


Run	DOF	Max. S1	Maximum Limit	X (Global)	Y (Global)	Z (Global)
1	2181	3.326e+02	3.163e+02	4.995e+02	2.498e+01	5.250e+02
2	6459	3.326e+02	3.163e+02	4.995e+02	2.498e+01	5.250e+02
3	14280	3.225e+02	3.163e+02	4.995e+02	2.498e+01	5.250e+02
4	26709	3.180e+02	3.163e+02	4.995e+02	2.498e+01	5.250e+02
5	44811	3.188e+02	3.163e+02	4.995e+02	2.498e+01	5.250e+02
6	69651	3.163e+02	3.163e+02	4.997e+02	2.379e+01	5.250e+02
7	102294	3.163e+02	3.163e+02	4.995e+02	2.498e+01	5.250e+02

Target max S1:
314 MPa







Benchmark 5: Shaft with U-Shaped Notch- Kt

- Symington: “Benchmark 5 is again intended to test if the software package can appropriately capture a stress concentration (Kt). This benchmark uses a circular shaft loaded in uniaxial tension with a U-shaped notch running around the entire circumference of the shaft at midspan.”
- StressCheck solution:
  - 3833 geometrically mapped tetrahedra, p=2 to 5, 1.5 minute run time
  - Max S1 converges to **48.94 MPa (with 0.24% estimated error)**

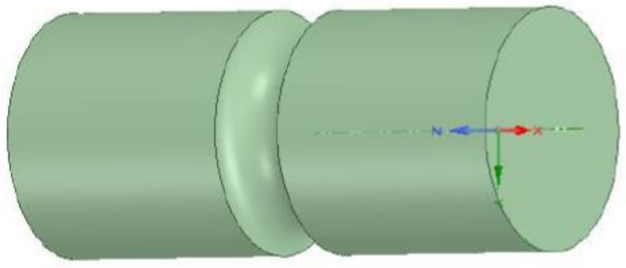
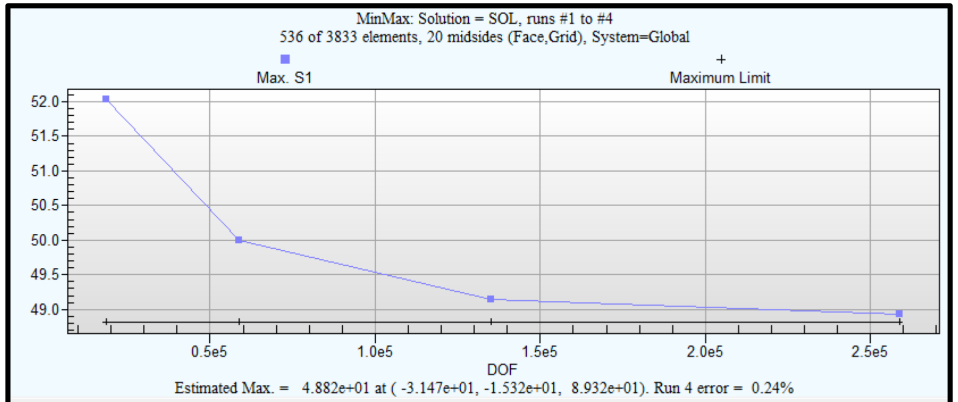


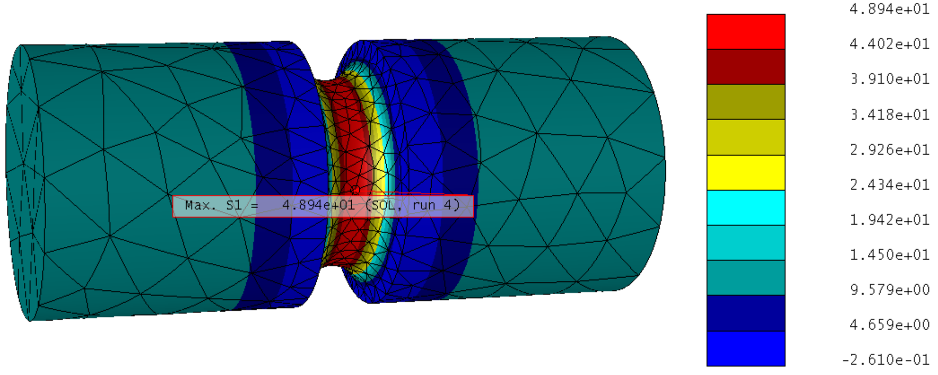
Image courtesy NAFEMS



Run	DOF	Max. S1	Maximum Limit	X (Global)	Y (Global)	Z (Global)
1	18687	5.203e+01	4.882e+01	2.400e+01	2.553e+01	8.788e+01
2	58935	4.999e+01	4.882e+01	-1.520e+00	-3.502e+01	8.776e+01
3	135117	4.914e+01	4.882e+01	-2.691e+00	3.490e+01	8.866e+01
4	258732	4.894e+01	4.882e+01	-3.147e+01	-1.532e+01	8.932e+01

Target max S1: 48.2 MPa

StressCheck V10.5
Units = MM/N/SEC/C
LINEAR ID=SOL
Run=4, DOF=258732
Fnc.=S1
Max= 4.894e+01
Min= -2.610e-01



Benchmark 6: Modal – Cantilevered Thin Plate

- Symington: “This benchmark explores the ability of the software package to accurately predict the first five modes of vibration of thin square plate constrained to act as a cantilever. The plate measures 10 x 10 x 0.05m. An analytical reference solution to this problem is provided in the NAFEMS Publication “Selected Benchmarks of Natural Frequency Analysis”.
- StressCheck solution:
  - 254 geometrically mapped tetrahedra, p=4 to 6, <20 seconds run time
  - Converged natural frequency modes and estimated errors are below:

1st

Run	DOF	Natural Freque...	% Error
1	10470	4.187830e-01	0.20
2	19515	4.179751e-01	0.01
3	32655	4.179502e-01	0.00

2nd

Run	DOF	Natural Freque...	% Error
1	10470	1.030907e+00	0.71
2	19515	1.024018e+00	0.04
3	32655	1.023656e+00	0.00

3rd

Run	DOF	Natural Freque...	% Error
1	10470	2.591137e+00	1.12
2	19515	2.562989e+00	0.02
3	32655	2.562435e+00	0.00

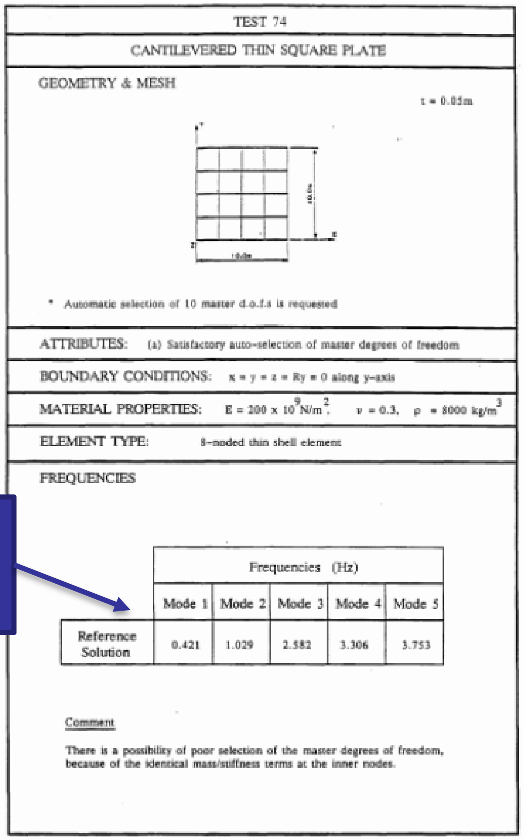
4th

Run	DOF	Natural Freque...	% Error
1	10470	3.341952e+00	2.09
2	19515	3.275812e+00	0.07
3	32655	3.273620e+00	0.00

5th

Run	DOF	Natural Freque...	% Error
1	10470	3.793778e+00	1.86
2	19515	3.727300e+00	0.07
3	32655	3.724749e+00	0.01

Target natural frequencies


TEST 74

CANTILEVERED THIN SQUARE PLATE

GEOMETRY & MESH

t = 0.05m

* Automatic selection of 10 master d.o.f.s is requested

ATTRIBUTES: (a) Satisfactory auto-selection of master degrees of freedom

BOUNDARY CONDITIONS: x = y = z = Ry = 0 along y-axis

MATERIAL PROPERTIES: $E = 200 \times 10^9 N/m^2$, $\nu = 0.3$, $\rho = 8000 kg/m^3$

ELEMENT TYPE: 8-noded thin shell element

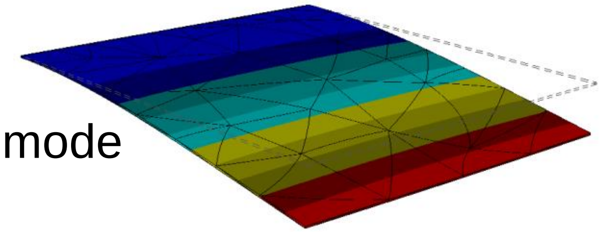
FREQUENCIES

	Frequencies (Hz)				
	Mode 1	Mode 2	Mode 3	Mode 4	Mode 5
Reference Solution	0.421	1.029	2.582	3.306	3.753

Comment

There is a possibility of poor selection of the master degrees of freedom, because of the identical mass/stiffness terms at the inner nodes.

Image courtesy NAFEMS


1st mode

Benchmark 7: Cantilever Under End Load

- Symington: “On first inspection, the simple cantilever beam bending problem described in this benchmark appears trivial. The reason for including a pure bending problem in this study is because it can often highlight deficiencies in both element formulation and the refinement of the automatically generated mesh. The cantilever in question has a square cross section and is loaded with a distributed force acting on the end face. The target solution can be obtained from an engineering handbook or using general purpose FEA code. Both the peak deflection and bending stress are defined as the targets of this benchmark.”
- StressCheck solution:
  - 125 geometrically mapped tetrahedra, p=2 to 8, <10 seconds run time
  - Max bending stress converges to **226.5 MPa (with 0.14% estimated error)**
  - Max deflection converges to **24.71 mm (with 0.00% estimated error)**

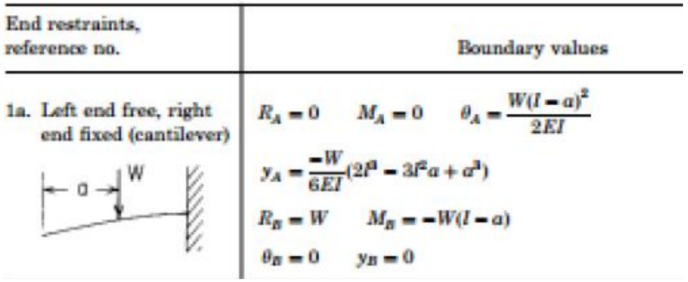
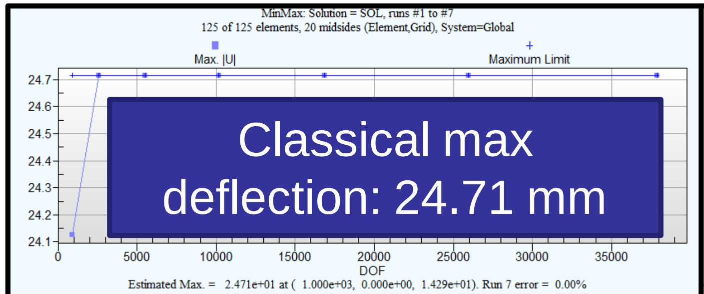
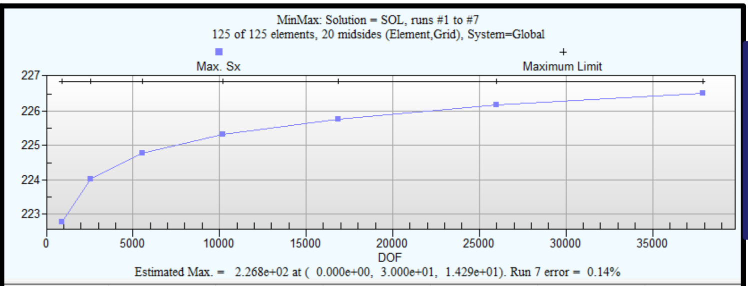


Image courtesy Roark, Table 8.1



Estimated Max. = 2.471e+01 at (1.000e+03, 0.000e+00, 1.429e+01). Run 7 error = 0.00%

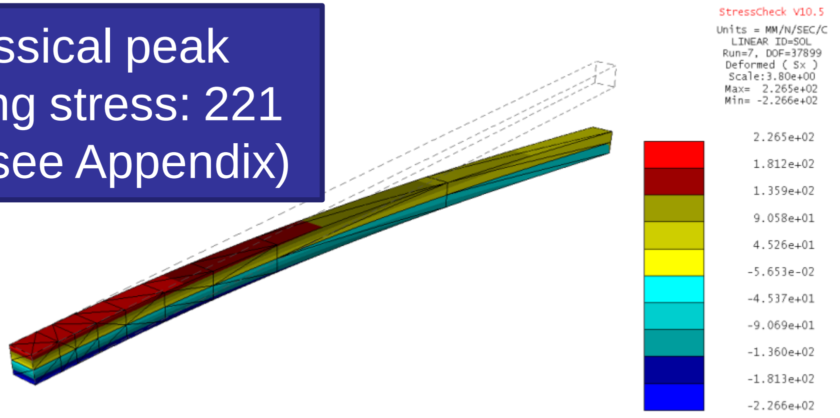
Run	DOF	Max. |U|	Maximum Limit	X (Global)	Y (Global)	Z (Global)
1	918	2.413e+01	2.471e+01	1.000e+03	0.000e+00	2.000e+01
2	2579	2.471e+01	2.471e+01	1.000e+03	3.000e+01	1.714e+01
3	5541	2.471e+01	2.471e+01	1.000e+03	0.000e+00	1.571e+01
4	10179	2.471e+01	2.471e+01	1.000e+03	0.000e+00	1.571e+01
5	16868	2.471e+01	2.471e+01	1.000e+03	0.000e+00	1.571e+01
6	25983	2.471e+01	2.471e+01	1.000e+03	0.000e+00	1.429e+01
7	37899	2.471e+01	2.471e+01	1.000e+03	0.000e+00	1.429e+01



Estimated Max. = 2.268e+02 at (0.000e+00, 3.000e+01, 1.429e+01). Run 7 error = 0.14%

Run	DOF	Max. Sx	Maximum Limit	X (Global)	Y (Global)	Z (Global)
1	918	2.228e+02	2.268e+02	0.000e+00	3.000e+01	0.000e+00
2	2579	2.240e+02	2.268e+02	0.000e+00	3.000e+01	1.143e+01
3	5541	2.248e+02	2.268e+02	0.000e+00	3.000e+01	1.286e+01
4	10179	2.253e+02	2.268e+02	0.000e+00	3.000e+01	1.286e+01
5	16868	2.258e+02	2.268e+02	0.000e+00	3.000e+01	1.000e+01
6	25983	2.262e+02	2.268e+02	0.000e+00	3.000e+01	1.714e+01
7	37899	2.265e+02	2.268e+02	0.000e+00	3.000e+01	1.429e+01

Classical peak bending stress: 221 MPa (see Appendix)



Benchmark 8: Cantilever with Realistic Support

- Symington: “The geometry used in Benchmark 7 was extended and built into a larger structure with the intention of exploring the stress concentration at the point of connection. A 5mm fillet radius is used to smooth the transition between cantilever and supporting structure. Due to the small size of the fillet radius in regard to the height and length of the component’s major dimensions, accurately capturing the stress concentration is not a trivial task.”
- StressCheck solution:
  - 317 geometrically mapped tetrahedra, p=2 to 8, 1 minute run time
  - Max Seq converges to **356.9 MPa (with 0.58% estimated error)**

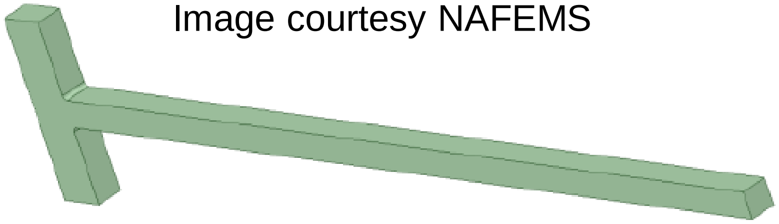
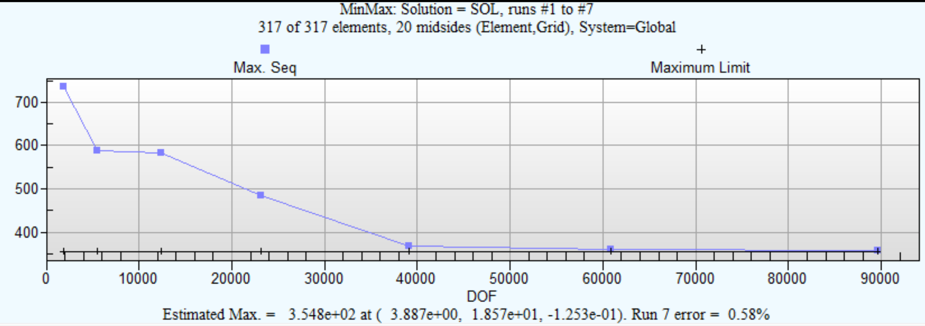
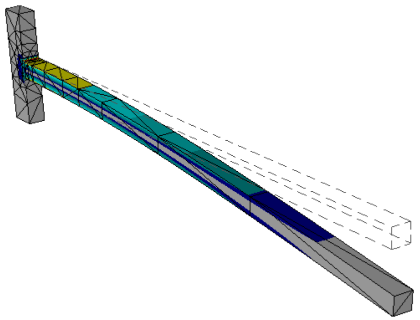


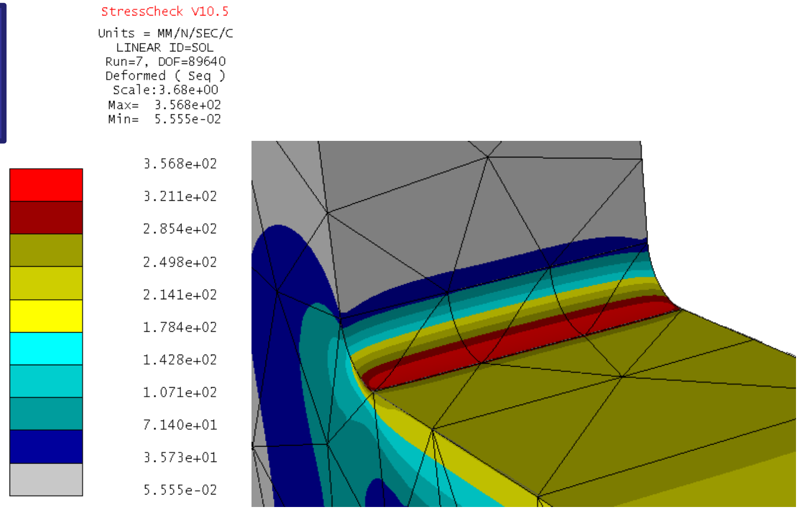
Image courtesy NAFEMS



Run	DOF	Max. Seq	Maximum Limit	X (Global)	Y (Global)	Z (Global)
1	1860	7.362e+02	3.548e+02	5.000e+00	2.000e+01	0.000e+00
2	5550	5.869e+02	3.548e+02	5.000e+00	2.000e+01	0.000e+00
3	12348	5.839e+02	3.548e+02	5.000e+00	2.000e+01	0.000e+00
4	23205	4.838e+02	3.548e+02	5.000e+00	2.000e+01	0.000e+00
5	39072	3.689e+02	3.548e+02	4.255e+00	1.905e+01	-5.587e-02
6	60900	3.598e+02	3.548e+02	3.887e+00	1.857e+01	-1.253e-01
7	89640	3.569e+02	3.548e+02	3.887e+00	1.857e+01	-1.253e-01

Target max Seq:
356.5 MPa





APPENDIX

A Note on Benchmark 7: Cantilever Under End Load

Benchmark 7 Notes

- In the classical beam model (quoted from Roark) it is assumed that plane sections remain plane.
  - This is not a valid assumption for the constrained end when solving a problem of elasticity.
- The StressCheck 3D model assumes self-equilibrated shear loading at the ends, with symmetry at one end (to restrict two rotations) and rigid body constraints on the lower edge (to prevent in plane translations and one rotation).
  - Therefore, the 3D model does not assume a “fixed” end condition as fixing the end would cause stress singularities (stresses at the fixed end would never converge).
  - The 3D model still targets the classical end displacement, and well represents the bending stresses.