T1400698 - Vacuum Beam tube FEA notes:

Scope:
- Will the back scattering form the vacuum beam tube be a dominant noise at certain frequency?
- If yes, how to mitigate the problem?

To help answering these questions, a FEA model of the beam tube and its support structure has been made. Static analysis have been performed to see if the static compliance of the model is in agreement with expected values and measurements.
Modal analysis have been carried out to predict the shapes of the beam tube deformation.

Contents:

Robert’s analysis:

robert.schofield@LIGO.ORG - posted 17:43, Sunday 19 May 2013 (6418)

Beam Tube Motions

Scattering from beam tube baffles is more likely to be a noise source in aLIGO than it was in iLIGO. With this in mind, the motion of the beam tube was studied. Figure 1 shows beam-line, horizontal and vertical motion for accelerometers mounted midway up the side of the beam tube near one of the fixed supports. This spectrum was taken at 988m Y from the beam splitter, when the wind was less than 5 MPH. Figure 2 shows beam-line motion spectra for different locations along the beam tube and at different wind velocities. These results are similar to results at LLO here.

The beam tube consists of a series of modular segments, with the great majority being 39 m long and weighing about 4000 kg (A+B segments). The segments are supported close to the middle with a fixed support, and each end hangs from a guided support. The beam tube segments are connected at the guided supports to the next segment in line by bellows that have a measured spring constant of 8.2e5 N/m. Figures 3a and b show that, in the 5 – 20 Hz band, the segments act as rigid bodies, with beam-line motion at the middle and at the ¾ point of a segment being virtually identical, highly coherent and with zero phase difference. Figure 3b also shows that, in contrast, the motion of adjacent segments is not very coherent. This is likely better news than if adjacent sections moved coherently.

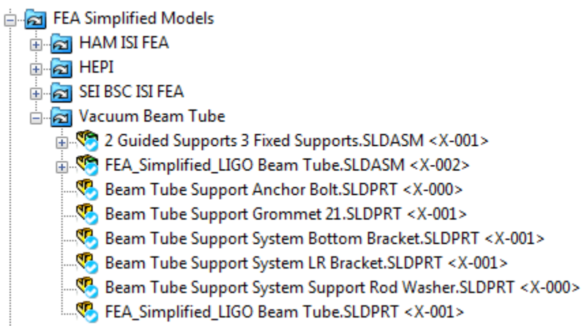
The calculated beam-line resonance of a beam tube segment, considering only the bellows springs, would be about 3 Hz, while the actual lowest resonance in the figures is at about 8 Hz. This demonstrates that the central support is the stiffest spring. Near the stations, some of the segments are shorter; Figure 3 shows that the lowest beam-line resonant frequency of a 33.5 m segment is about 9.3 Hz instead of 8 Hz. The frequency goes approximately as mass rather than the square root of mass, suggesting, as one might expect, that there is a torsional component about the central support. A modal explanation of the 8 and 14 Hz resonances awaits modeling.

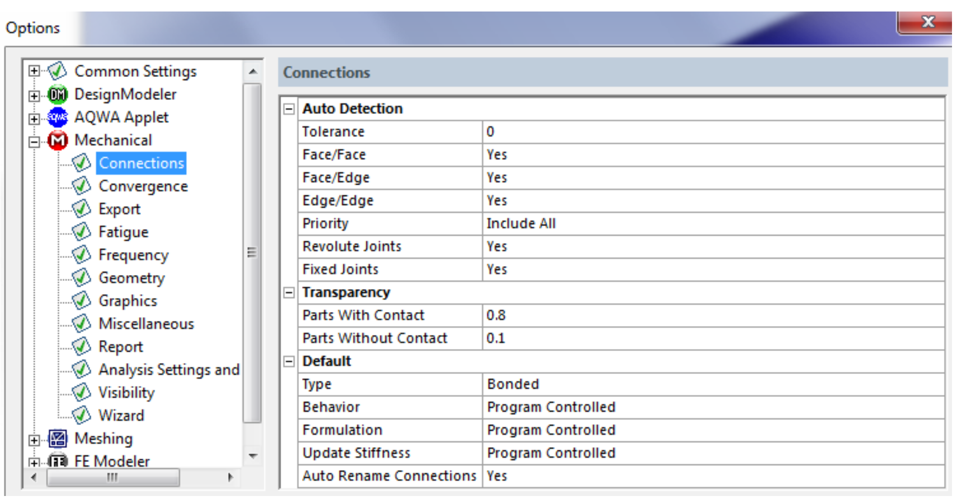
Since the beam tube segments are moving as rigid bodies in the 10 Hz region, the insulation may not damp the motion much at these frequencies. We may get a chance to test this as John is going to remove some insulation.

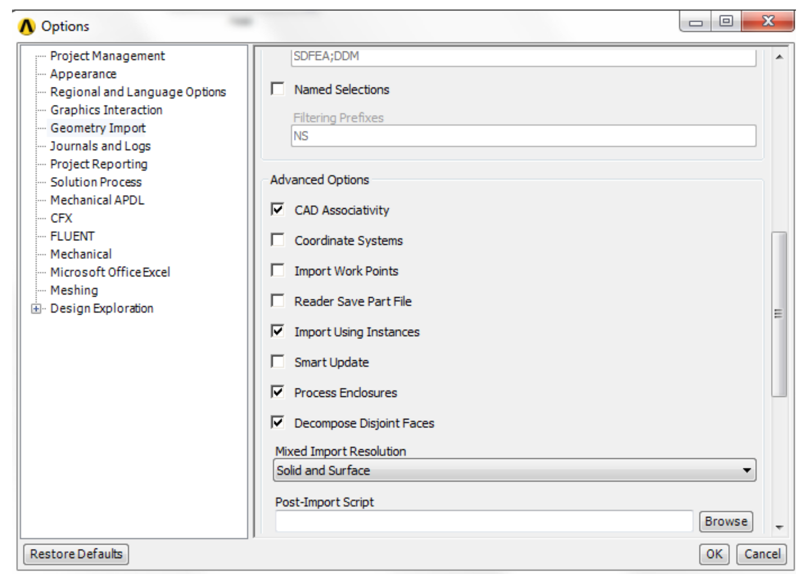
1. Model Overview

1.1 Model Description

CAD Model and import settings:

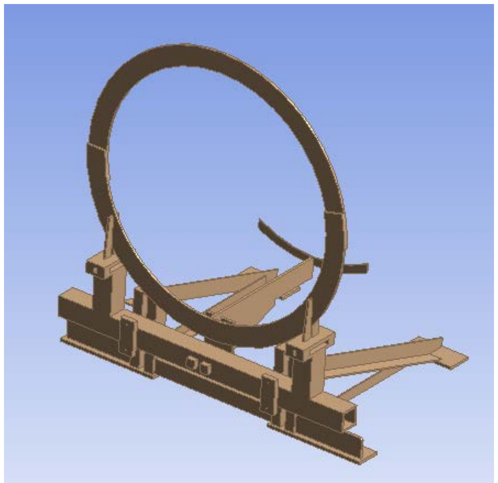


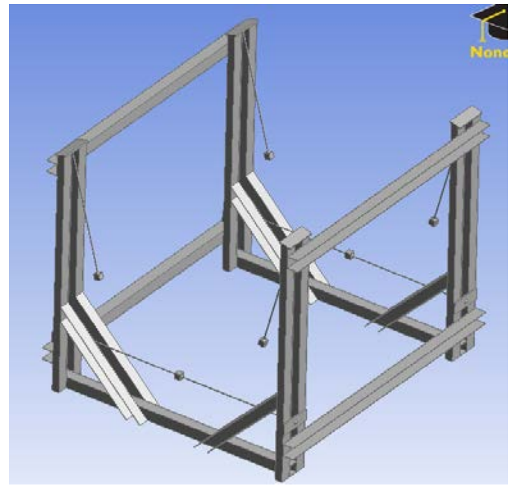


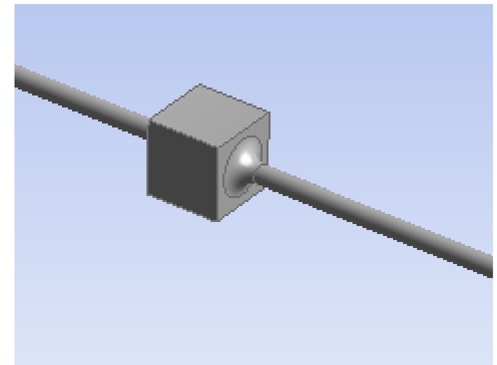


1.1 Model Description

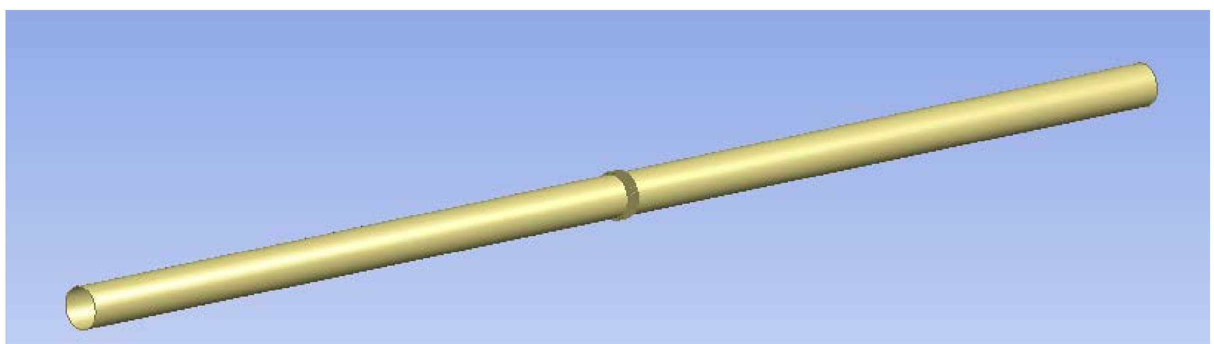
Sub-assemblies:







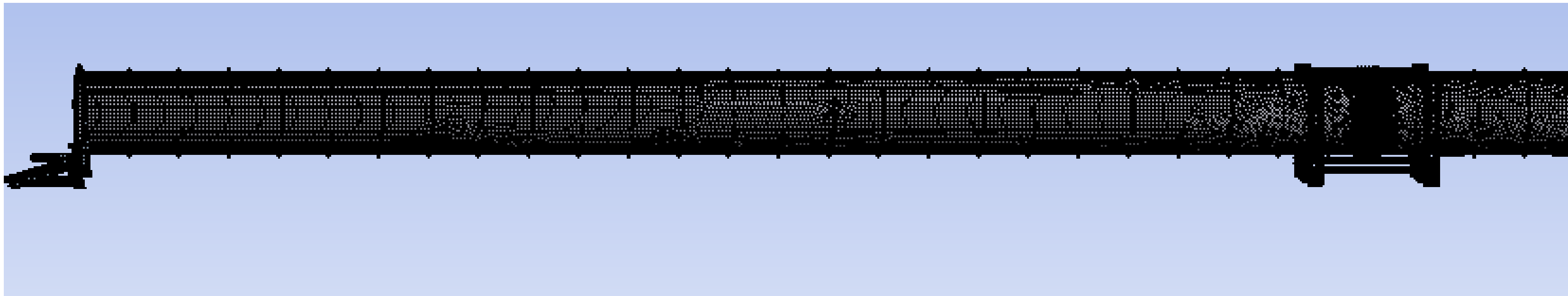
Fixed and guided support: 3D elements

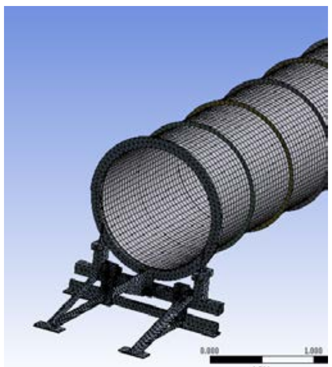


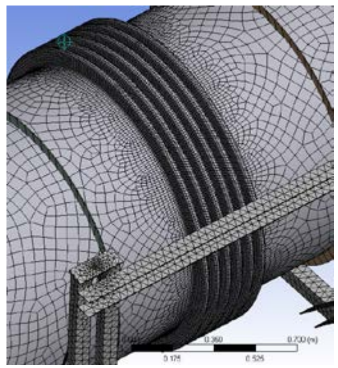
Beam tube, bellows and stiffening rings (not represented): shell elements

1.1 Model Description

Meshing:

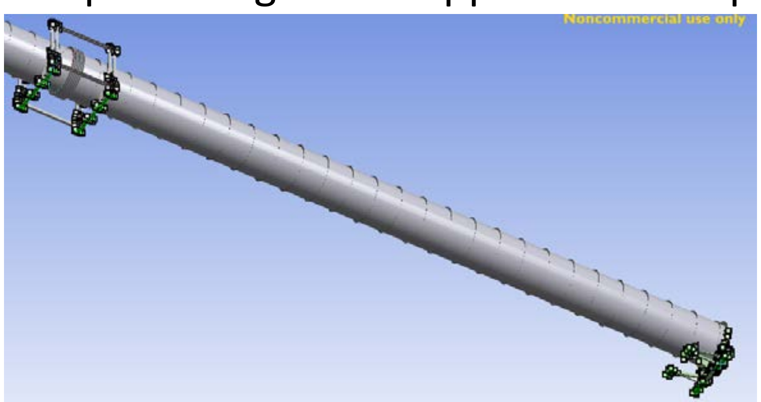






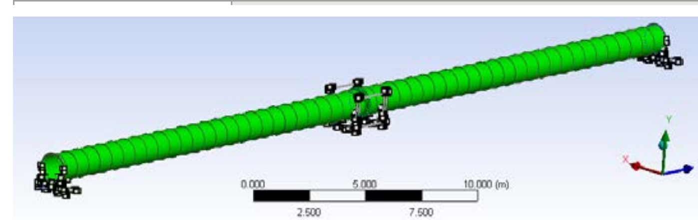
Statistics	
Nodes	326871
Elements	210477

Boundary conditions: bottom faces of clamped and guided support are clamped.



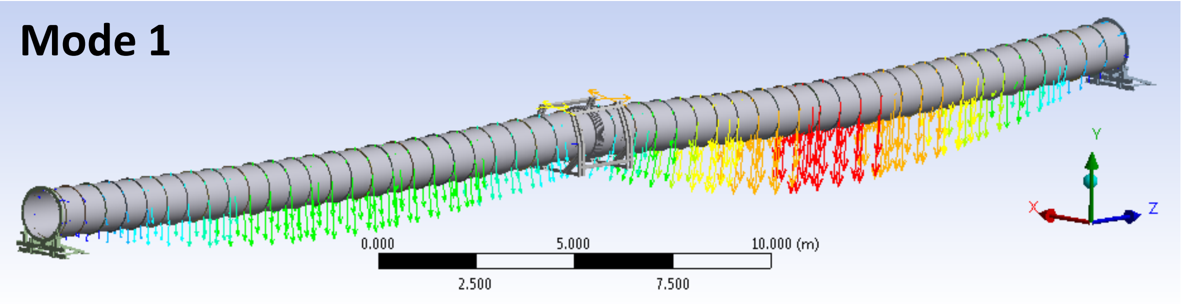
Properties check:

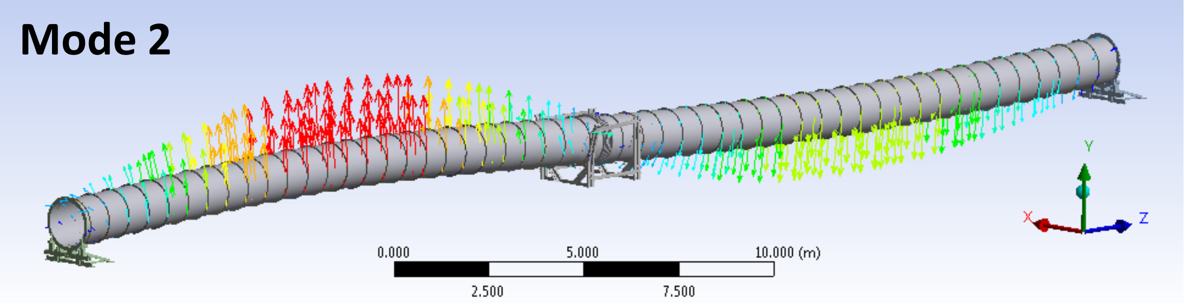
Properties	
Volume	0.50531 m^3
Mass	3966.7 kg

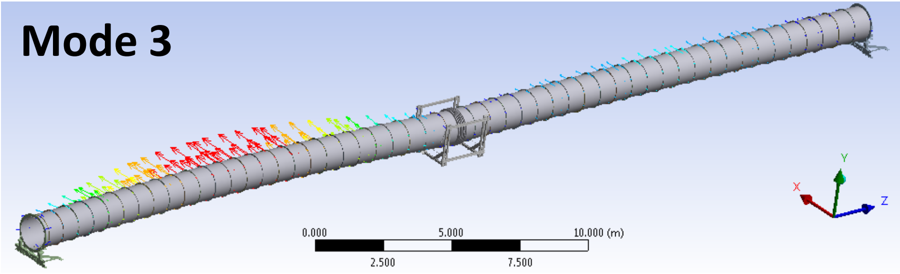


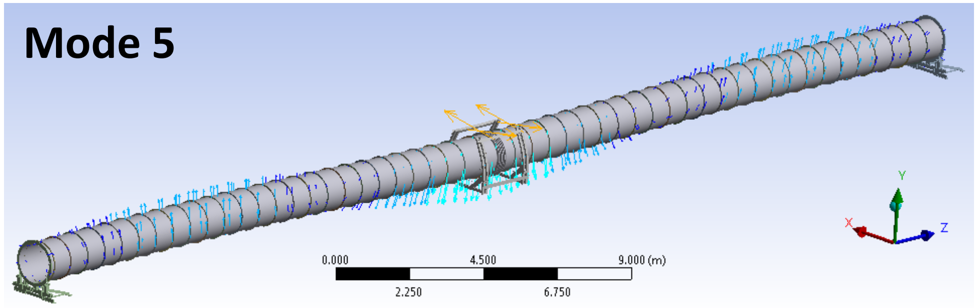
1.2 Modal Analysis Results

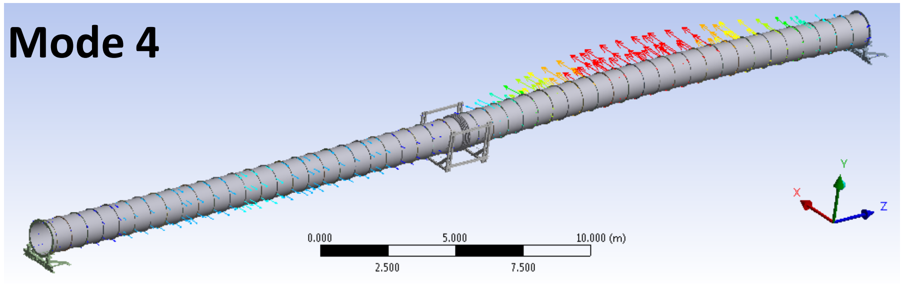
	Mode	Frequency [Hz]
1	1.	9.5391
2	2.	10.176
3	3.	10.587
4	4.	10.597
5	5.	19.169
6	6.	27.245

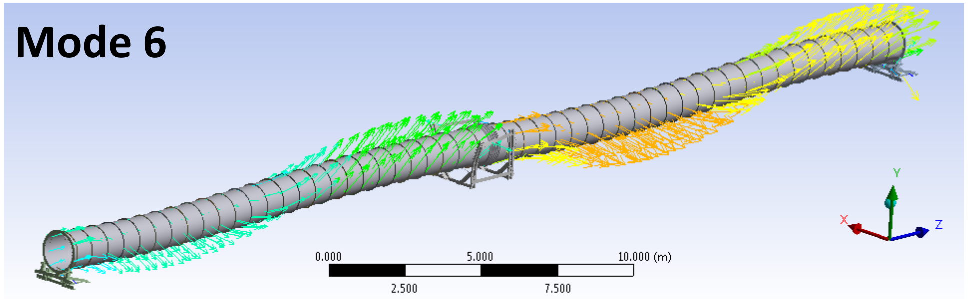






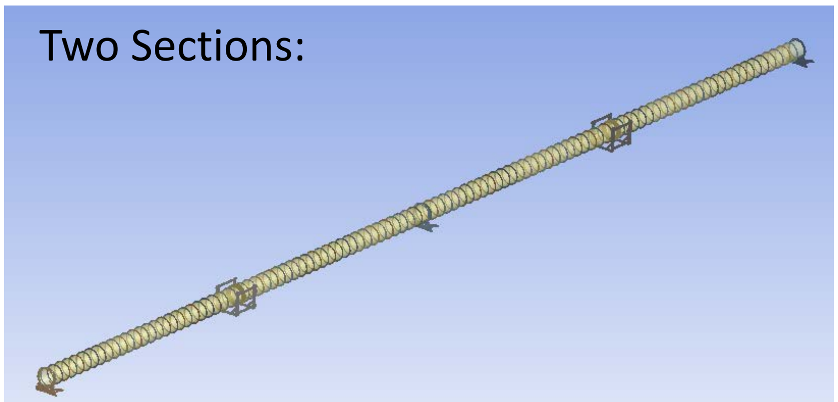


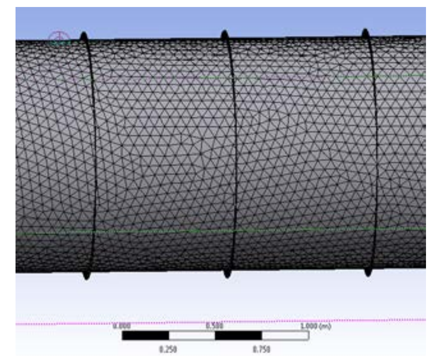


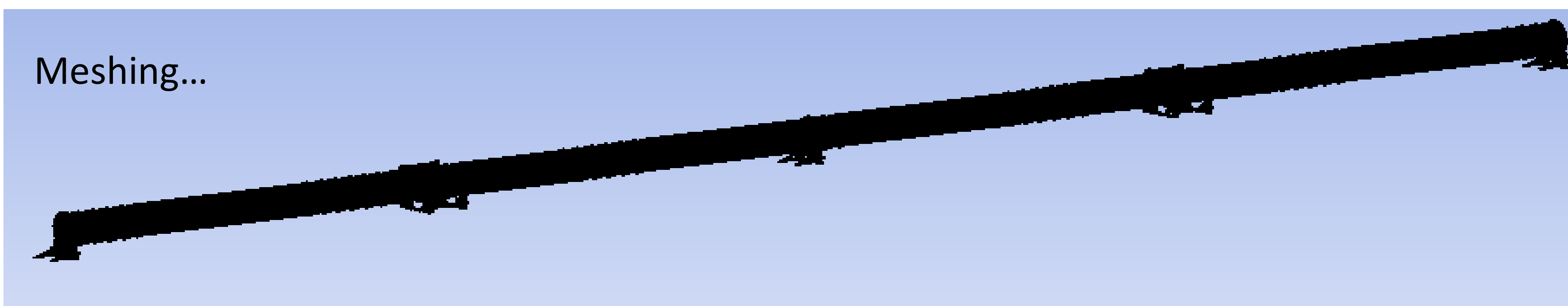


2. Refined Model

Questions about the first model / objectives for the refined model:
- Stiffening effect from contacts elements?
- Can we mesh it all solid?
- Gravity/pre-stress effects in cables?
- Longitudinal modes, bellows stiffness?

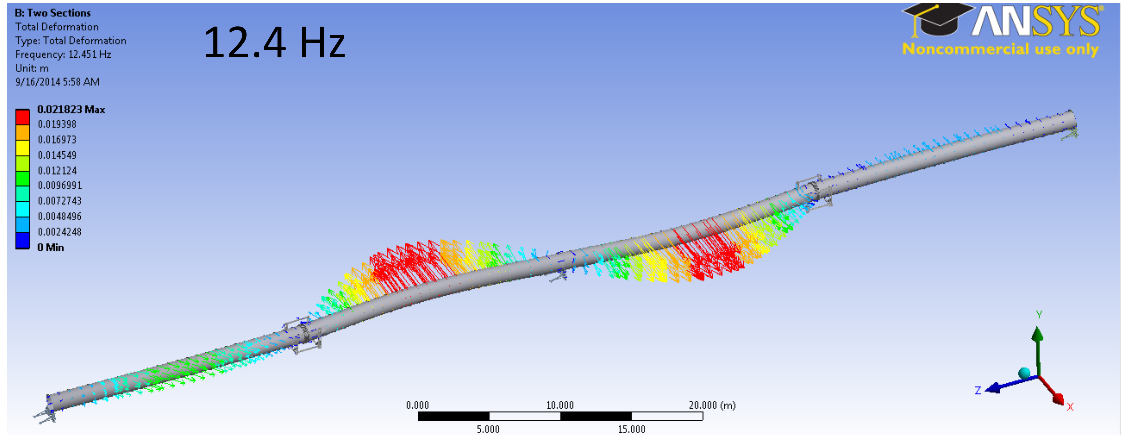


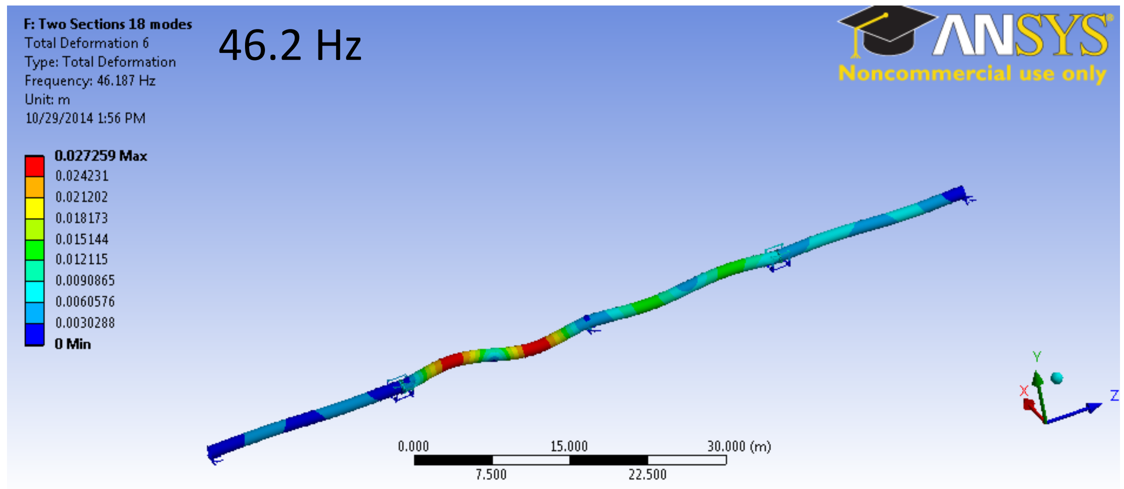


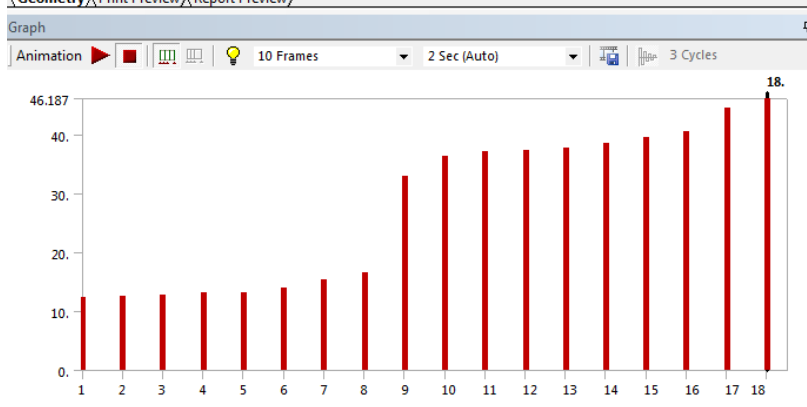


Way longer to mesh/solve than a single section (had not completed after several hours)

2. Refined model results







Tabular Data

	Mode	Frequency [Hz]
1	1.	12.451
2	2.	12.574
3	3.	12.788
4	4.	13.125
5	5.	13.247
6	6.	13.898
7	7.	15.364
8	8.	16.671
9	9.	32.962
10	10.	36.375
11	11.	37.106
12	12.	37.399
13	13.	37.75
14	14.	38.499
15	15.	39.512
16	16.	40.58
17	17.	44.684
18	18.	46.187

3. Tubes Modeling

3.1 Bending and breathing modes

Tube bending mode: analytical eigen frequency (Blevins)

```
f1 =

   21.6377
```

```
%% Eigen frequency for the tube section

ro=7850*1000;
b=0.01;
m=ro*b^3;

L=19.1;
m=1885/L;

E=2e11;
r=1.245/2;
t=0.0032;
I=pi*r^3*t;

lambda=4.730;
f1 = lambda^2/(2*pi*L^2)*(E*I/m)^0.5
```

Breathing mode
Cylindrical shells of infinite length
(Blevins)

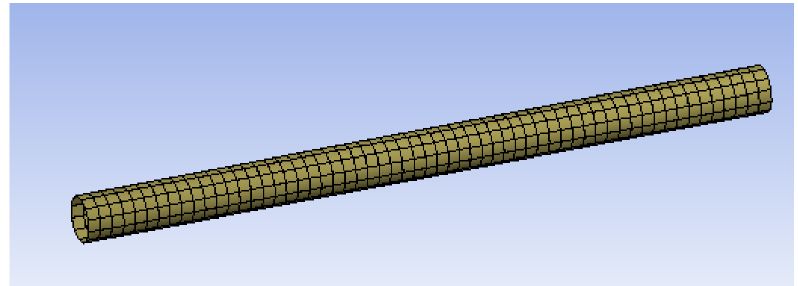
```
f1 =

     5.4040


f2 =

   15.2848


f3 =

   29.3072
```

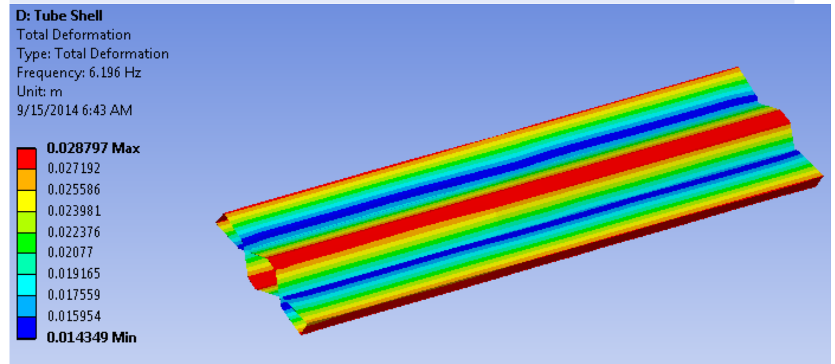
```
%% Shell (Tube, breathing) Mode
clc
mu=7800;
nu=0.3;
k=2;
lambda1 = (1/12)^0.5 *t/r *k *(k^2-1)/(1+k^2)^0.5;
f1= lambda1/(2*pi*r)*(E/(mu*(1-nu^2)))^0.5

k=3;
lambda2 = (1/12)^0.5 *t/r *k *(k^2-1)/(1+k^2)^0.5;
f2= lambda2/(2*pi*r)*(E/(mu*(1-nu^2)))^0.5

k=4;
lambda3 = (1/12)^0.5 *t/r *k *(k^2-1)/(1+k^2)^0.5;
f3= lambda3/(2*pi*r)*(E/(mu*(1-nu^2)))^0.5
```

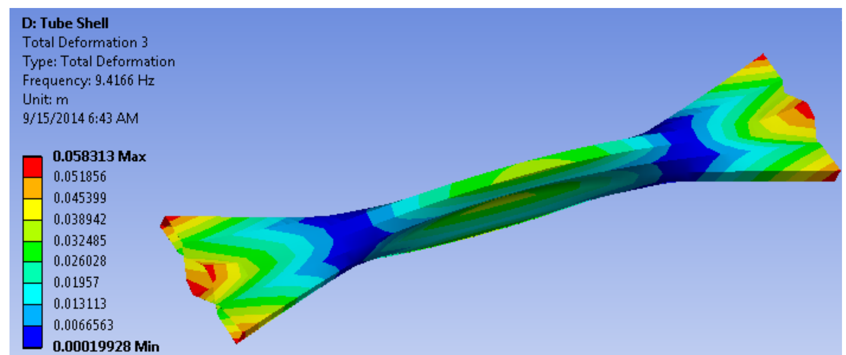
3.1 Bending and breathing modes

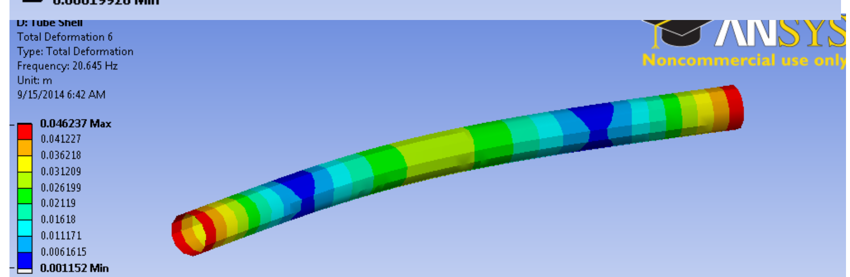
Using shell Elements

	Mode	Frequency [Hz]
1	1.	0.
2	2.	0.
3	3.	0.
4	4.	7.1346e-006
5	5.	1.4559e-005
6	6.	1.6228e-005
7	7.	6.196
8	8.	6.1967
9	9.	6.2047
10	10.	6.2065
11	11.	9.4166
12	12.	9.4187
13	13.	20.273









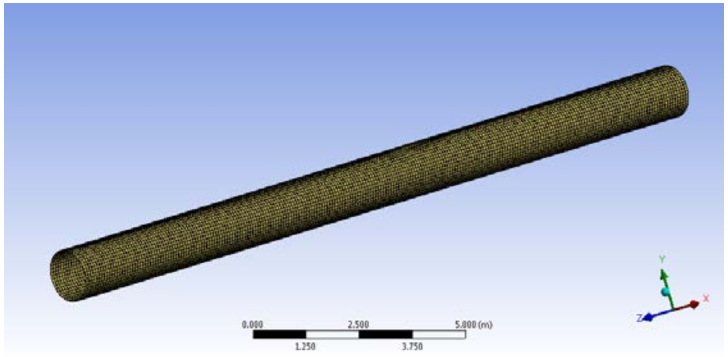
3.1 Bending and breathing modes

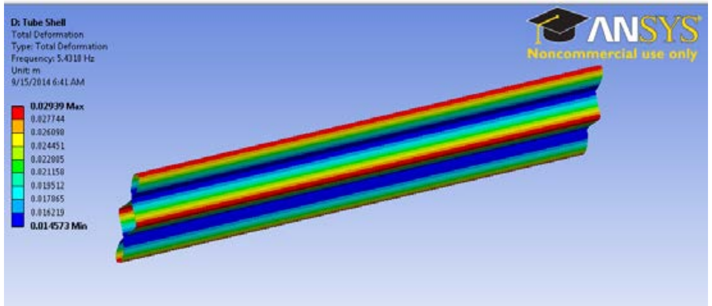
Thinner Meshing (Shell)

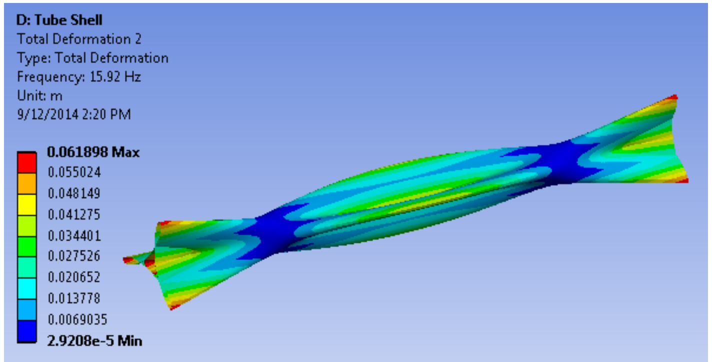
Statistics	
Nodes	11376
Elements	11328

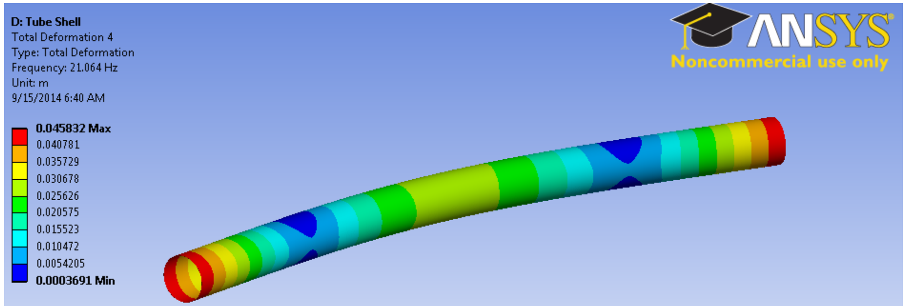
Tabular Data

	Mode	Frequency [Hz]
1	1.	0.
2	2.	0.
3	3.	0.
4	4.	4.5056e-006
5	5.	1.9943e-005
6	6.	2.3188e-005
7	7.	5.4318
8	8.	5.4318
9	9.	5.4409
10	10.	5.4409
11	11.	8.7579
12	12.	8.7579
13	13.	15.525







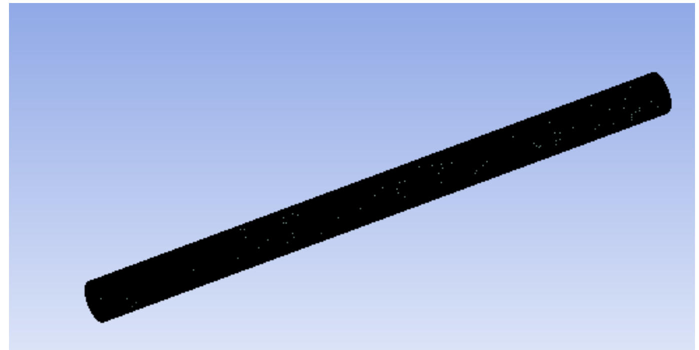


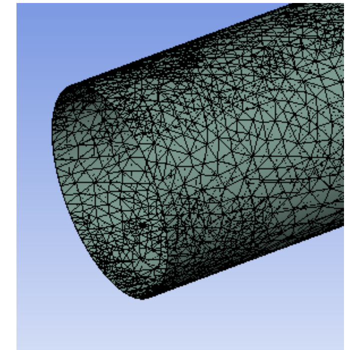
3.1 Bending and breathing modes

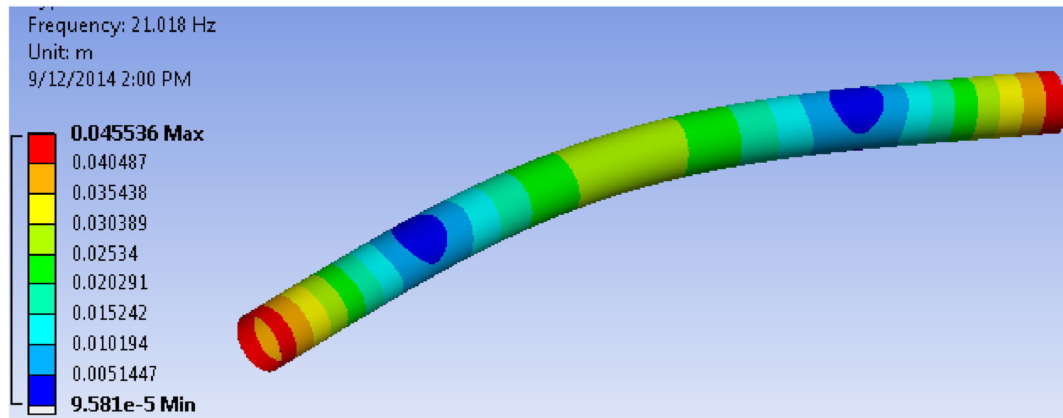
Solid Elements

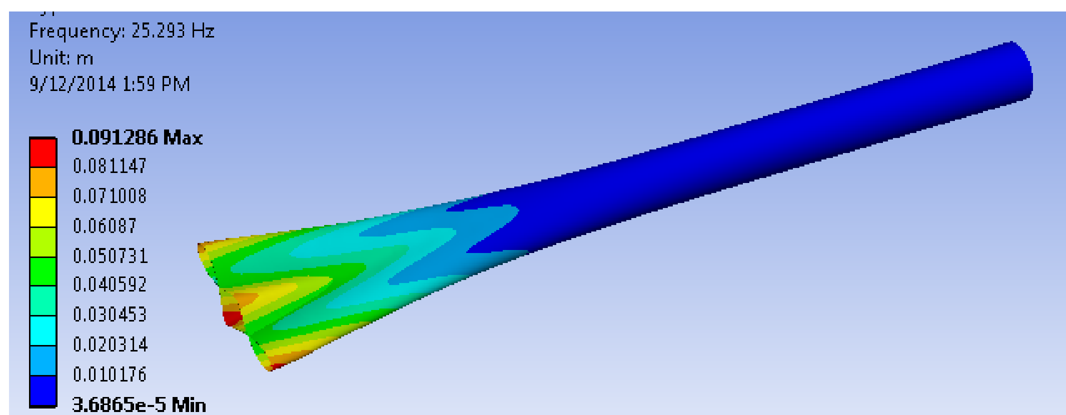
Tabular Data

	Mode	Frequency [Hz]
1	1.	8.3653e-004
2	2.	1.4116e-003
3	3.	1.5707e-003
4	4.	1.9305e-003
5	5.	2.444e-003
6	6.	2.4583e-003
7	7.	21.018
8	8.	21.044
9	9.	25.293
10	10.	25.75
11	11.	27.421
12	12.	27.674
13	13.	28.224
14	14.	28.426
15	15.	32.885
16	16.	33.177
17	17.	45.545
18	18.	45.671
19	19.	55.084
20	20.	55.173









Breathing mode not Predicted by the solid meshing

3.1 Bending and breathing modes

```
*** SELECTION OF ELEMENT TECHNOLOGIES FOR APPLICABLE ELEMENTS ***
     --- GIVE SUGGESTIONS AND RESET THE KEY OPTIONS ---

ELEMENT TYPE    1 IS SHELL181. IT IS ASSOCIATED WITH ELASTOPLASTIC
MATERIALS ONLY. KEYOPT(8)=2 IS SUGGESTED AND KEYOPT(3)=2 IS SUGGESTED FOR
HIGHER ACCURACY OF MEMBRANE STRESSES; OTHERWISE, KEYOPT(3)=0 IS SUGGESTED.
KEYOPT(8) HAS BEEN RESET BUT KEYOPT(3) CAN NOT BE RESET HERE. PLEASE RESET
IT MANUALLY IF NECESSARY.
 KEYOPT(1-12)=    0    0    2    0    0    0    0    2    0    0    0    0

ELEMENT TYPE    1 HAS KEYOPT(3)=2. FOR THE SPECIFIED ANALYSIS TYPE, LUMPED MASS
MATRIX OPTION (LUMPM, ON) IS SUGGESTED.
```

SHELL181 Input Summary

Nodes
I, J, K, L

Degrees of Freedom
UX, UY, UZ, ROTX, ROTY, ROTZ if KEYOPT(1) = 0
UX, UY, UZ if KEYOPT(1) = 1

ANSYS Structural Analysis Guide:

http://orange.engr.ucdavis.edu/Documentation12.1/121/ans_str.pdf

KEYOPT(1)
Element stiffness:

0 --
Bending and membrane stiffness (default)

1 --
Membrane stiffness only

2 --
Stress/strain evaluation only

KEYOPT(3)
Integration option:

0 --
Reduced integration with hourglass control (default)

2 --
Full integration with incompatible modes

KEYOPT(8)
Specify layer data storage:

0 --
Store data for bottom of bottom layer and top of top layer (multi-layer elements) (default)

1 --
Store data for TOP and BOTTOM, for all layers (multi-layer elements)

Note: Volume of data may be excessive.

2 --
Store data for TOP, BOTTOM, and MID for all layers; applies to single- and multi-layer elements

KEYOPT(9)
User thickness option:

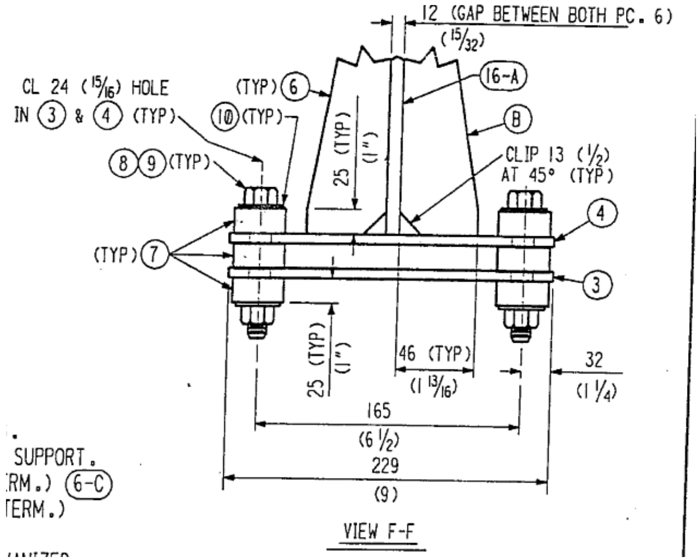
0 --
No user subroutine to provide initial thickness (default)

1 --
Read initial thickness data from user subroutine UTHICK

4. Including the "Micarta Block"

12	6-7		MICARTA BLOCK 44 X 44 X 25 W/ 22		GRADE G11	C
			(7/8) DIA HOLES (1 3/4 X 1 3/4 X 1)			

LIGO-D950035-08-B



http://www.professionalplastics.com/professionalplastics/MicartaGrades-DataSheet.pdf

Micarta ® Laminates – Various Grades Technical Information

Phenolic sheet is a hard, dense material made by applying heat and pressure to layers of paper or glass cloth impregnated with synthetic resin. These layers of laminations are usually of cellulose paper, cotton fabrics, synthetic yarn fabrics, glass fabrics or unwoven fabrics. When heat and pressure are applied to the layers, a chemical reaction (polymerization) transforms the layers into a high-pressure thermosetting industrial laminated plastic.

4. Including the "Micarta Block"

Standard stock grades include:

G1O/FR4 Glass Reinforced Epoxy - natural (yellowish to light green) The most versatile all around laminate, this grade is a continuous glass woven fabric base impregnated with an epoxy resin binder. It has extremely high mechanical strength, good dielectric loss properties, and good electric strength properties, both wet and dry. Certifies to Mil-I-24768/27 GEE-F

G11/FRS Glass Reinforced Epoxy - natural (yellow green to amber) This grade is similar to G10/FR4, with the addition of a higher operating temperature and some improved mechanical strength at elevated temperatures. Certifies to Mil-I-24768/28 GEB -F

Typical properties of phenolics

ASTM or UL test	Property	General purpose	Impact	Non bleeding	Electrical	Heat resistant	Glass reinforced	Chemical resistant compound
				PHYSICAL				
D792	Specific gravity	1.35- 1.46	1.36- 1.41	1.37- 1.38	1.36- 1.75	1.41- 1.84	1.7- 2.0	1.37- 1.75
D570	Water absorption, 24 hours, 1/8 inch thick (%)	0.6-0.7	0.6-0.9	0.8-0.9	0.05- 0.20	0.30- 0.35	0.05- 0.2	0.20- 0.40
				MECHANICAL				
D651	Tensile strength (psi)	6,500- 7,000	6,000- 7,000	6,000- 7,000	5,000- 7,000	5,000- 6,000	6,000- 12,000	7,000- 9,000
D790	Flexural modulus (105 psi)	11-13	12	10	17-25	14	20-30	10
D790	Flexural strength (psi)	9,000- 11,000	10,000	10,000	9,000- 11,000	10,000	12,000- 24,000	9,500

TYPICAL PROPERTIES of GLASS LAMINATES (SHEET FORM)

(mechanical properties of rod and tube forms may differ)

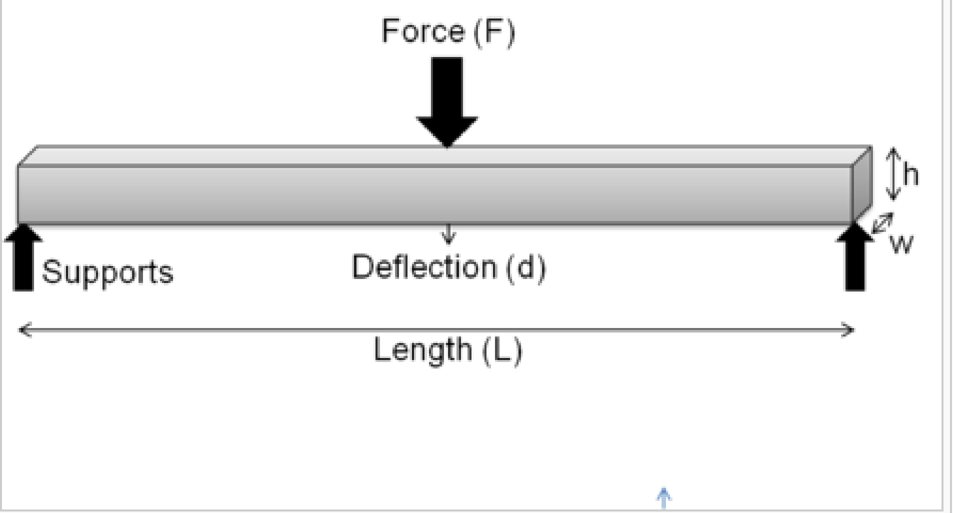
ASTM or UL test	Property	G-3	G-5/G-9	G-7	G-10	G-11
	PHYSICAL					
D792	Density (lb/in³) (g/cm³)	0.065 1.80	0.067 1.85	0.065 1.80	0.065 1.80	0.065 1.80
D570	Water Absorption, 24 hrs (%)	2.65	0.60	0.10	0.10	0.20
	MECHANICAL					
D638	Tensile Strength (psi) -lengthwise -crosswise	 42,000 34,000	 61,600 51,100	 20,000 -	 45,000 38,000	 43,000 37,000
D790	Flexural Strength (psi) -lengthwise -crosswise	 40,500 34,000	 61,600 51,100	 30,000 -	 75,000 65,000	 80,000 70,000
D790	Flexural Modulus (Kpsi) -lengthwise -crosswise	 1,800 1,400	 2,000 1,700	 1,600 -	 2,700 2,400	 3,000 2,700
D256	IZOD Notched Impact (ft-lb/in) -lengthwise -crosswise	 12.0 11.0	 12.5 8.5	 13.0 -	 14.0 12.0	 12.0 9.0
D695	Compressive Strength (psi)	55,000	65,000	50,000	65,000	63,000
D785	Hardness, Rockwell M	M110	M115	M105	M110	M112

$E = 3000$ kpsi ~ 20,000 Mpa

Flexural modulus

From Wikipedia, the free encyclopedia

In mechanics (mechanics is a branch of physics), the **flexural modulus** or **bending modulus**[1] is the ratio of stress to strain in flexural deformation, or the tendency for a material to bend. It is determined from the slope of a stress-strain curve produced by a flexural test (such as the ASTM D 790), and uses units of force per area.[2] It is an intensive property.



Flexural modulus measurement

For a 3-point test of a rectangular beam behaving as an isotropic linear material, where *w* and *h* are the width and height of the beam, *L* is the distance between the two outer supports and *d* is the deflection due to the load *F* applied at the middle of the beam, the flexural modulus:

$$E_{\mathrm{bend}} = \frac{L^3F}{4wh^3d}$$

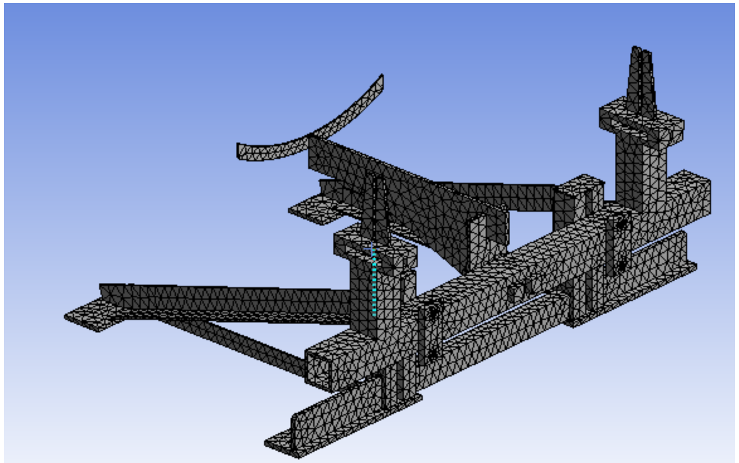
From elastic beam theory $d = \frac{L^3F}{48IE}$ and

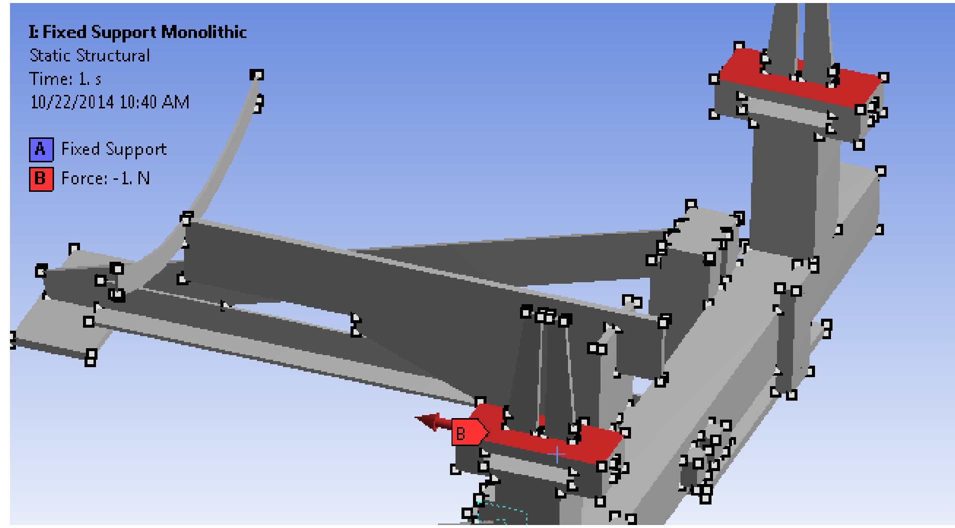
for rectangular beam $I = \frac{1}{12}wh^3$

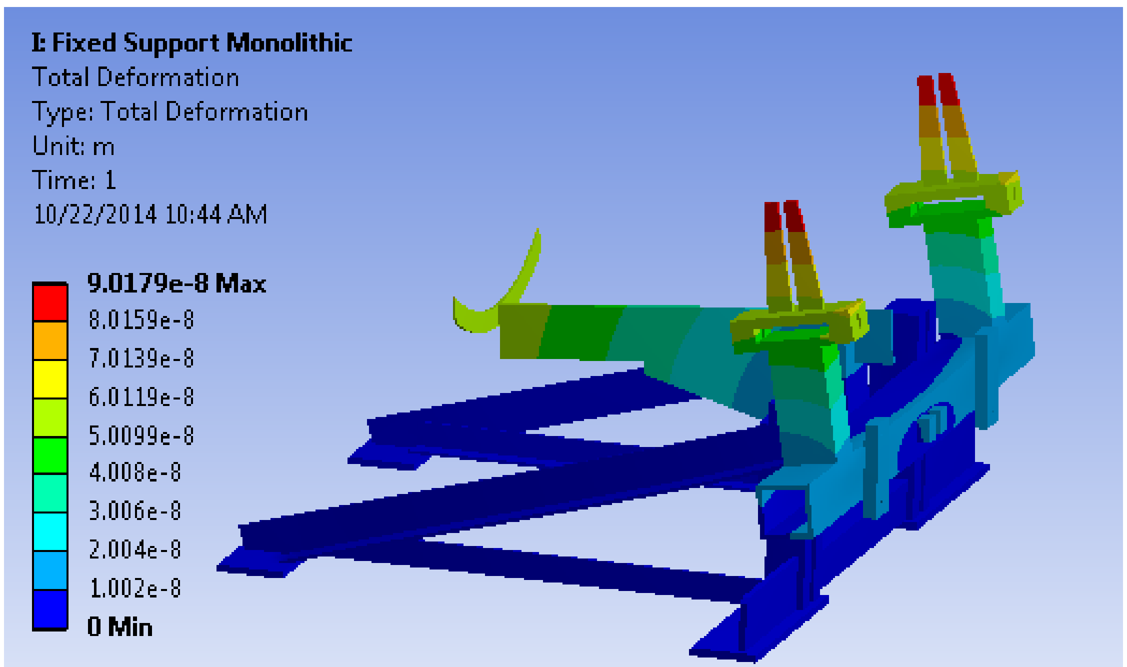
thus $E_{\mathrm{bend}} = E$ (Elastic modulus)

Ideally, flexural or bending modulus of elasticity is equivalent to the tensile or compressive modulus of elasticity. In reality, these values may be different, especially for plastic materials.

Fixed Support Monolithic







$k = F/x \sim 1e7$ N/m

Shear Stiffness of a Micarta block (no holes first, to check with shear stress formula)

Analytical shear behavior:

Gamma=tau/G

G=E/2/(1+nu)

Tau =F/A

x= gamma * h

-> F = (A*G / h) * x

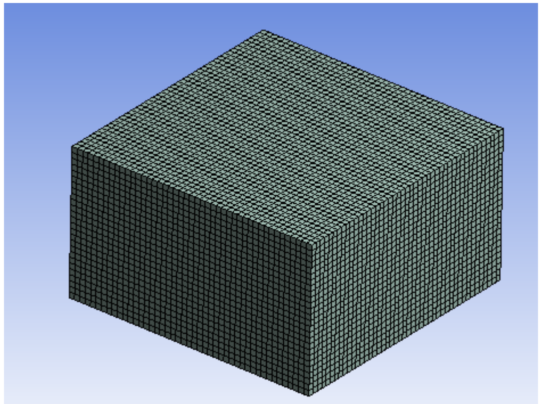
E = 20,000 Mpa

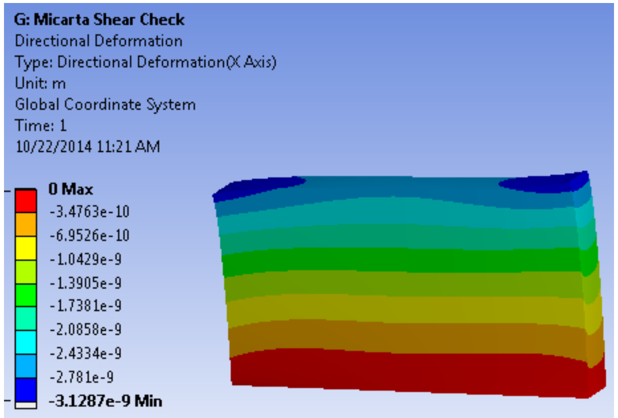
Nu=0.3

A=44mm*44mm

H=25 mm

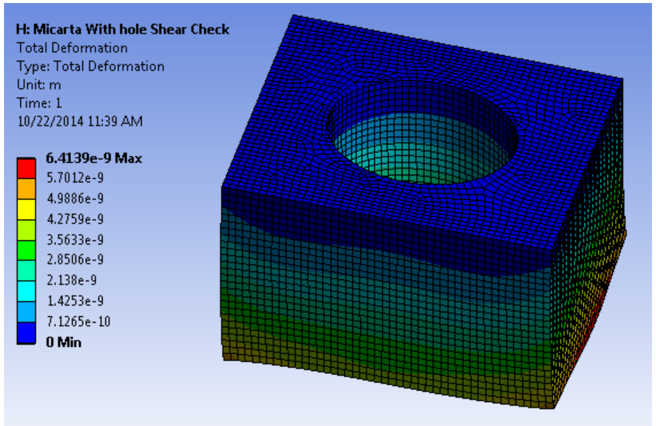
k=5.96e8 N/m





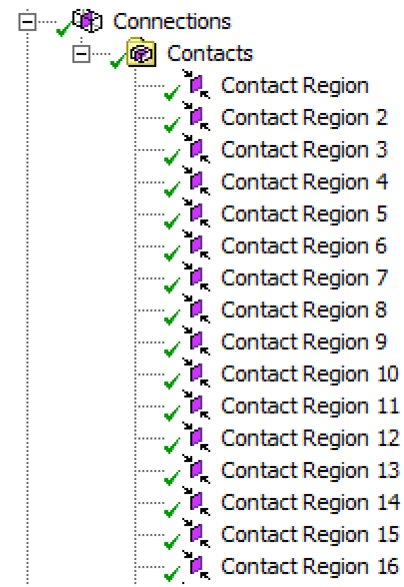
k = F/x ~ 3.2e8 N/m

With hole

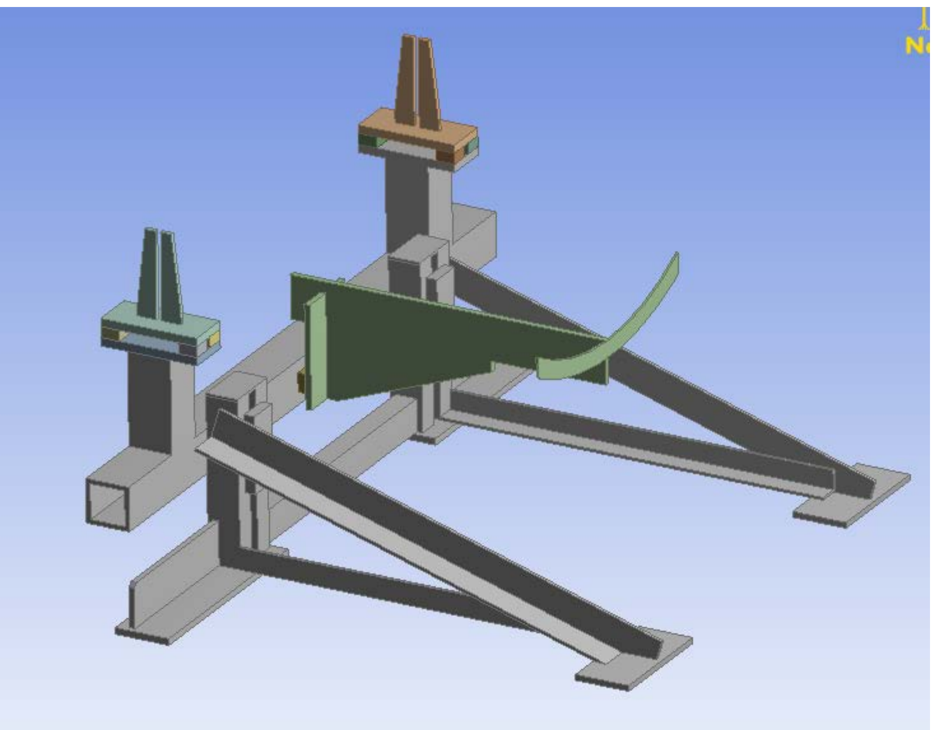


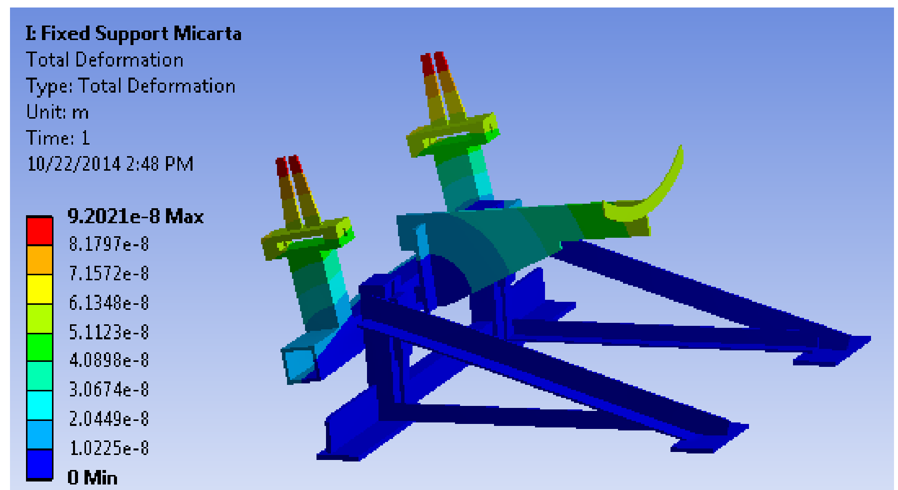
k = F/x ~ 1.56e8 N/m

Fixed Support with insulator still modeled with steels, but as independent parts. Contact elements between the insulators and the brackets. Check we get the same results as with the monolithic structure.



76 contacts…

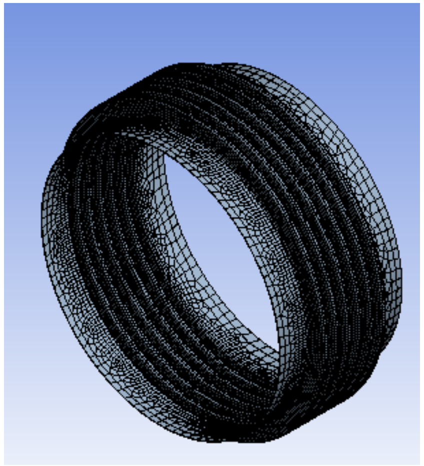


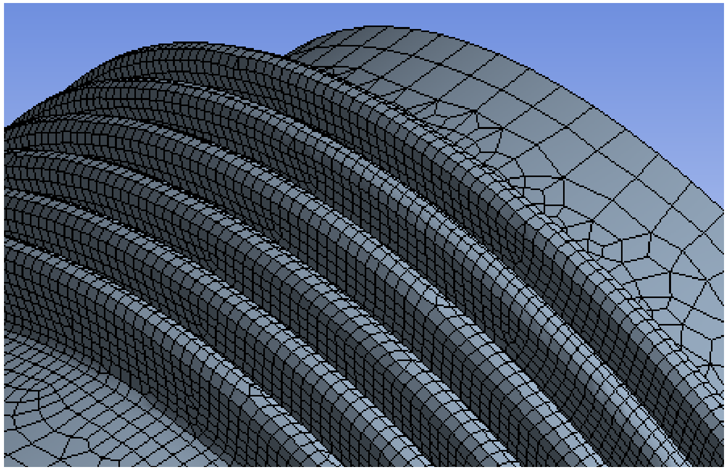


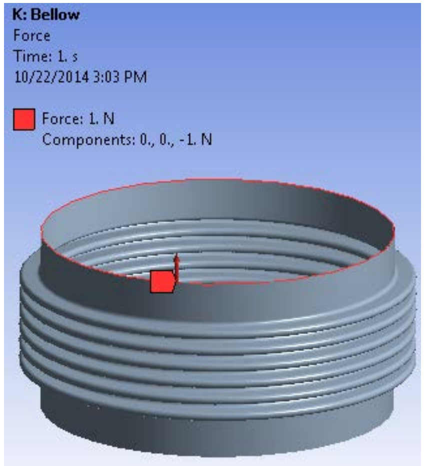
9.02e-8m compared to 9.02e-8 for the monolithic steel structure -> similar result

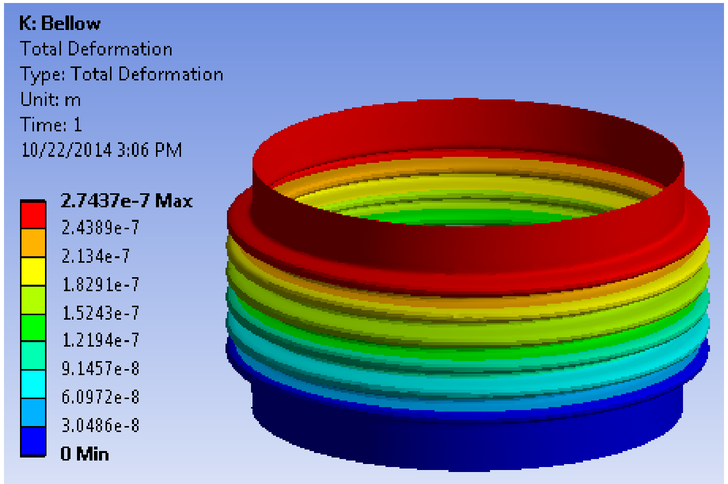
5. Bellows stiffness

5.1 Using shell elements









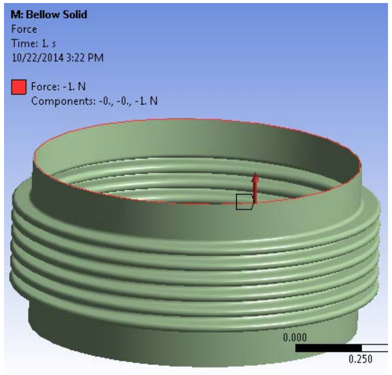
k = F/x ~ 3.7e6 N/m

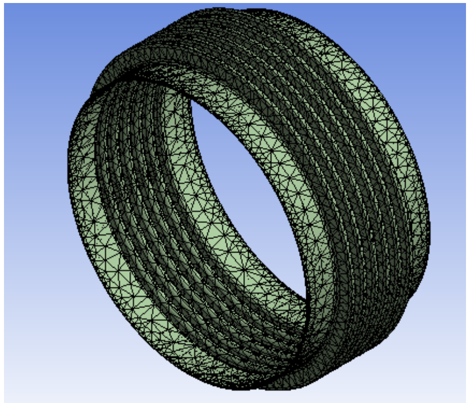
Got same results with twice many elements
(40,000 elements to 80,000 elements)

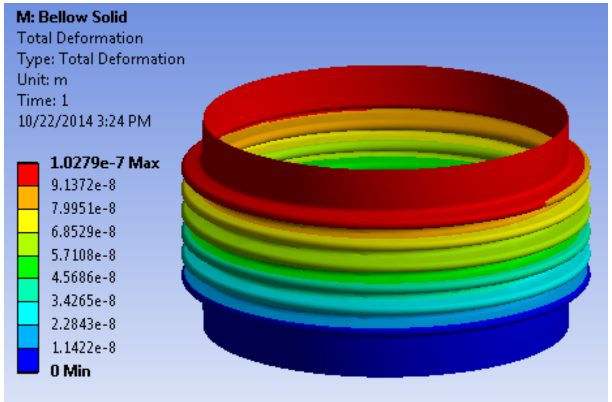
5. Bellows stiffness

5.2 Using solid elements

First Model:



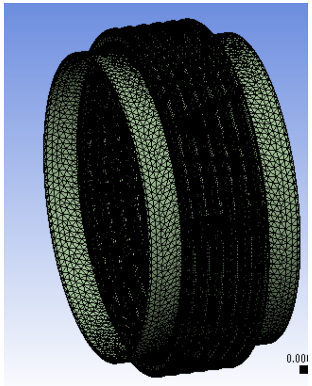


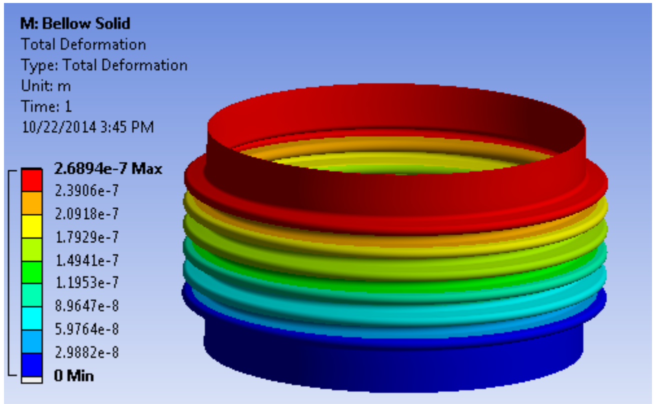


k = F/x ~ 9.7e6 N/m

Meshing too coarse?

Thinner mesh: 300,000 elements





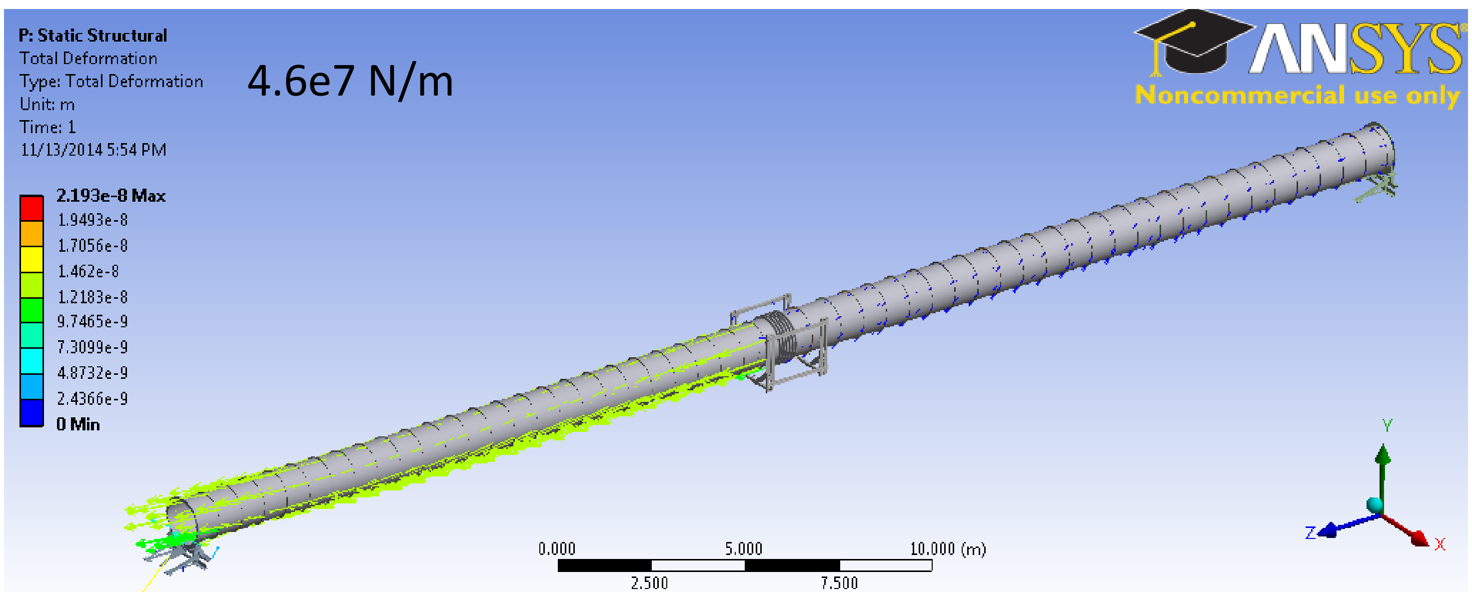
k = F/x ~ 3.7e6 N/m

Same results as with the shell meshing.

6. Static Model

Robert: “the pusher rod is attached to the bottom center of the 3rd stiffening ring from the fixed support. I had no load cells, only accelerometers, so no force measurements.”

Static FEA:



Conclusion / summary:

- The finite element models predict the **bellow's stiffness** to be approximately **3.7e6 N/m.**
- Some stiffness measurements of the bellows are reported in LIGO document "**E960123 Beam Tube Bakeout Design Requirements Document**". The document reports a mean value of 4677 lb/in (page 14). This translates into **8.2e5 N/m**, which is about four times lower than the finite element results.

-Robert had a look at the **spare section of bellow** that is seating on the LHO parking lot (mostly to check the thickness). He found an **engraved value indicating a stiffness approximately twice higher** than what is reported in E960123. That is still lower than what the FEA predicts.

- The report **E960123** states that "these **spring constants have been corrected** for the Hyspan calibration error in reporting the original data", but it does not provide further details.

- The finite element model predict the longitudinal stiffness of the assembly to be approximately **5e7 N/m**. The analysis shows that **electrical insulators (Micarta G10) should not contribute much to the overall compliance, assuming that they are perfectly bounded** (enough preload in the joints).

- The **finite element model** (several sections of bean tubes, guided and fixed supports) **does not predict any pure longitudinal mode**. All the modes predicted involve a great amount of bending of the tubes. So **either the full model over constrains the longitudinal motion, or the measurements underestimate the bending contribution of the tube**.