High Luminosity LHC

Crab Cavity support system

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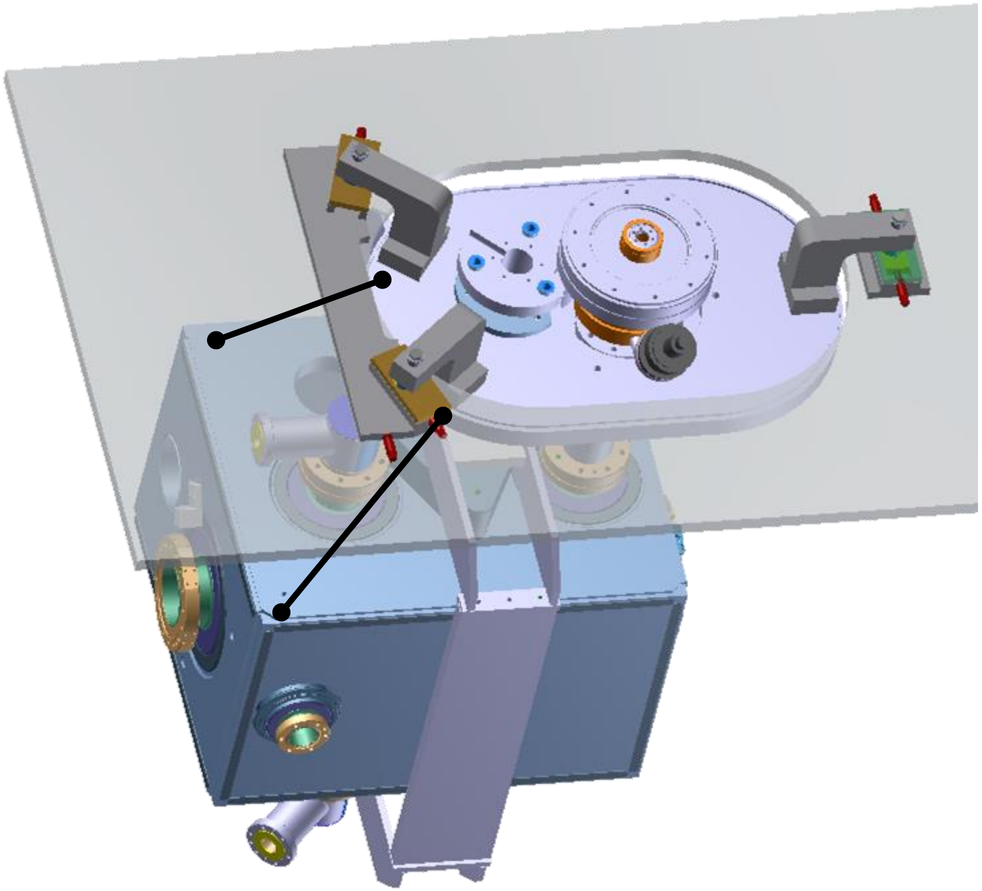
Science & Technology Facilities Council

The HiLumi LHC Design Study is included in the High Luminosity LHC project and is partly funded by the European Commission within the Framework Programme 7 Capacities Specific Programme, Grant Agreement 284404.

Introduction

- The cavities will be rigidly supported by the input coupler.
- This eliminates the need for an unsupported bellows between the FPC and the outer vacuum chamber, which would put the load of the RF input mass onto the cavity.
- However, additional supports are required to reduce stress in the FPC and ensure maximum rigidity of the cavity.
- Initially several options were studied using a simplified model to determine the best method of support.



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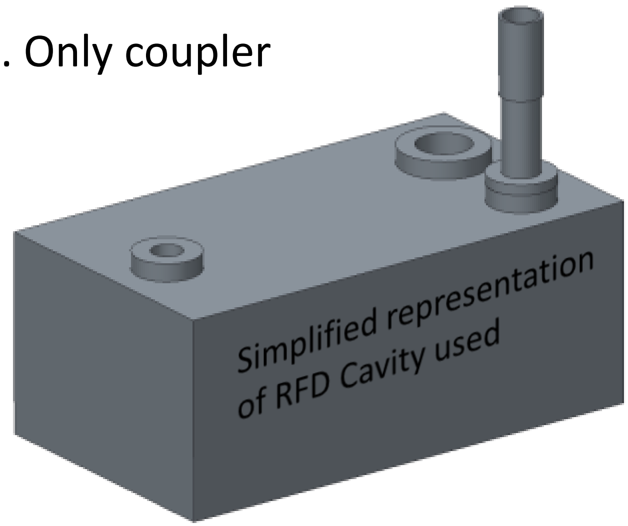
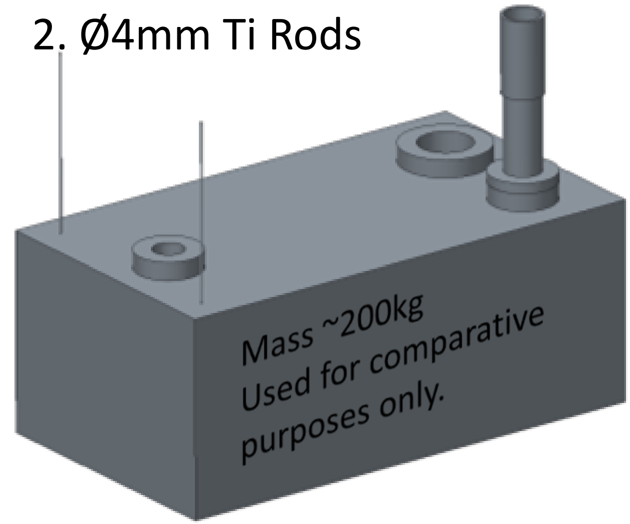
Analysis models

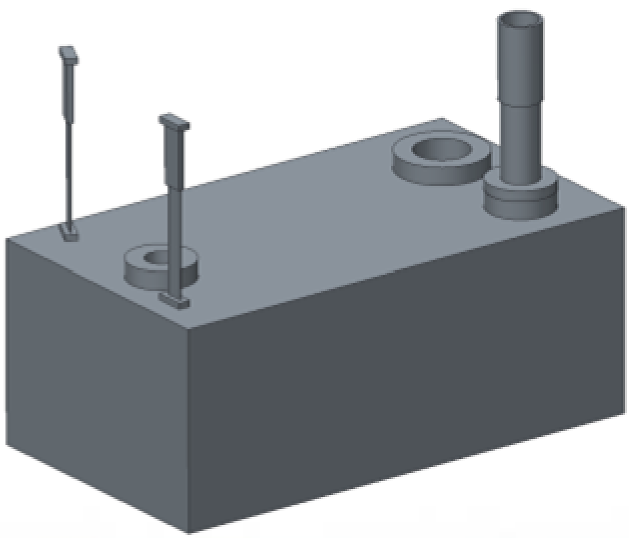
1. Only coupler



2. Ø4mm Ti Rods

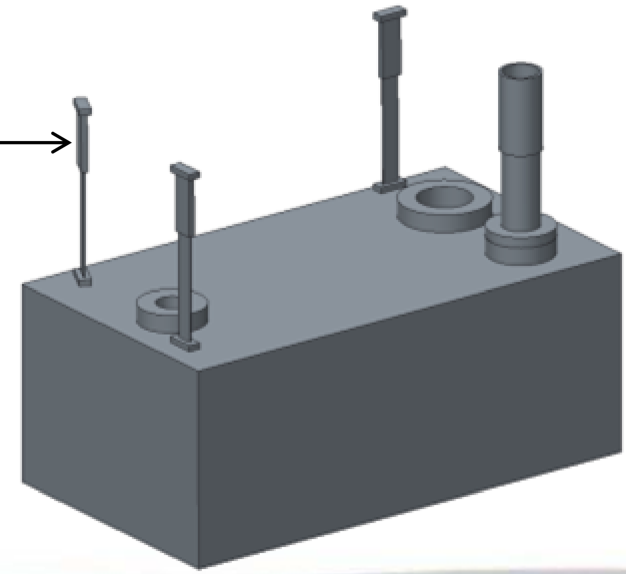


3. Stainless Steel flexures (2mm thick)



Thickness to give similar thermal conduction and therefore similar temperature profile and contraction as the input coupler

4. Additional flexure



Analysis setup

- RFD cavity used as it represents the 'worst case' due to being the most cantilevered support.
- The mass is approximate, but is valid for these comparative purposes.
- Standard earth gravity applied.
- Coupler, rods and flexures fixed at common support plate.
- Static total deformation, Max von-Mises Stress and first 4 modes were found.
- No mesh convergence check performed, but same meshing used for each analysis (see next).
- Material properties measured by FermiLab from 300K to 2K used.

Analysis Result

Analysis	Max Deformation (mm)	Max von-Mises stress (MPa)	Mode 1 Frequency (Hz)	Mode 2 Frequency (Hz)	Mode 3 Frequency (Hz)	Mode 4 Frequency (Hz)
1	3.9	183	7.7	8.3	16.1	61.1
2	0.24	65.2	8.5	25.3	38.3	70.9
3	0.025	15.3	25.1	48.3	56.5	122
4	0.01	10.5	27.2	50.15	66.9	174

- Performance is significantly improved using blade type flexures.
- The 3 flexure design is best, however, gives increased heat load and difficulty of integration.
- Therefore a 2 blade solution was chosen for the SPS cryomodule.

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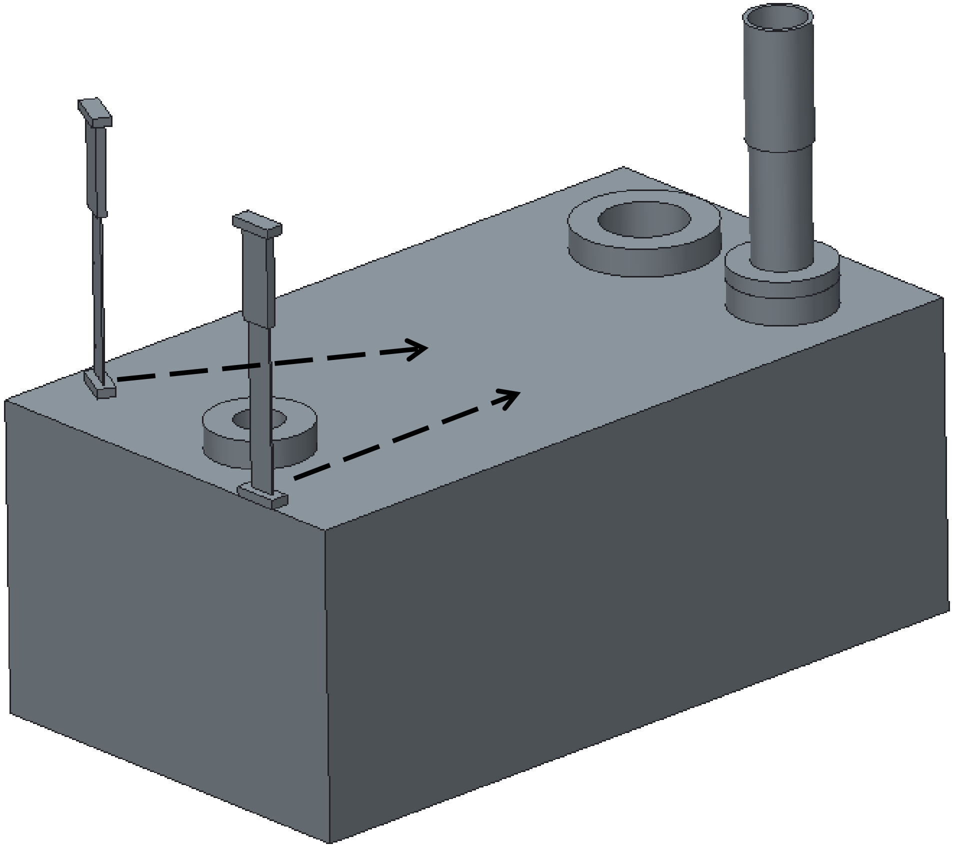
Support distance sweep

Several parameter sweeps were performed to optimise the blade performance.

In this analysis the supports were moved closer to input coupler in steps of 50mm from 700mm to 400mm.



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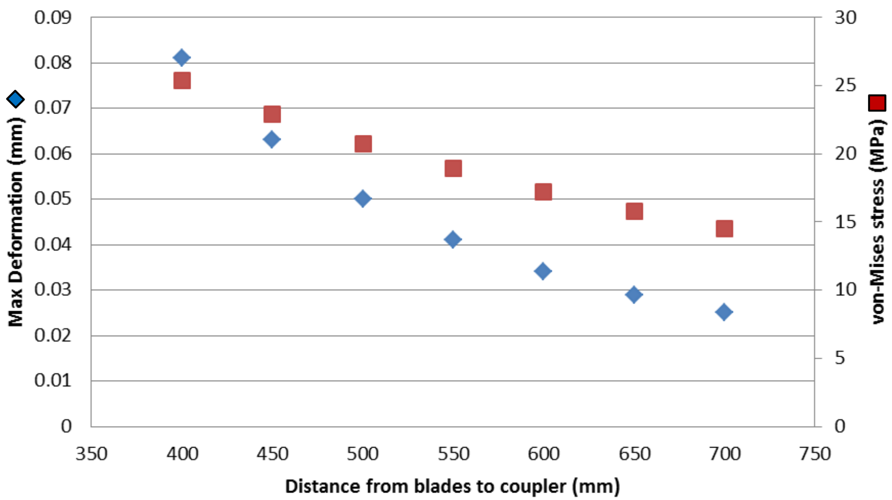
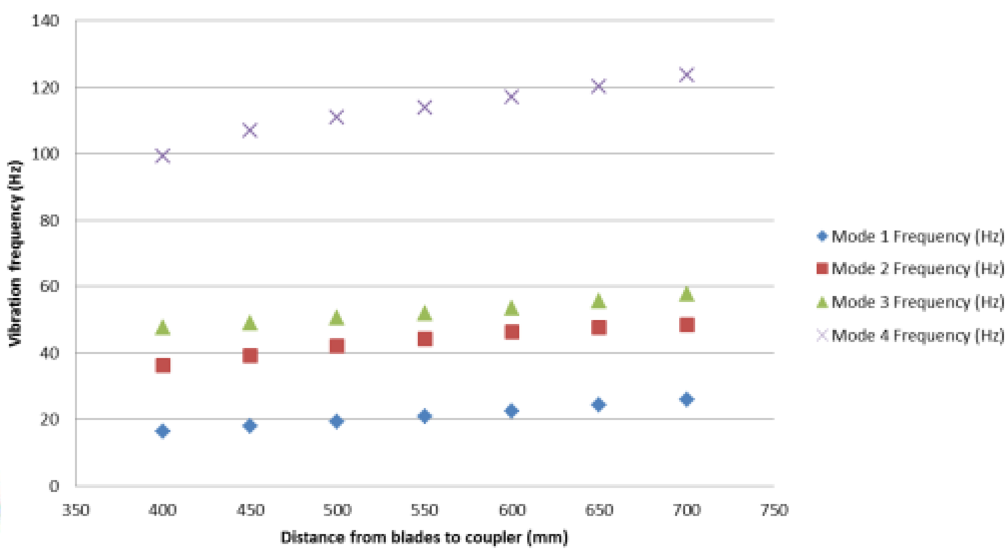
Result comment

Deformation and Stress vs. blade position



Modes vs. blade position



- Deformation and stress reduce the further the blades are from the coupler.
- Due to the increased stiffness of the system, the modes of vibration increase the further the blades are positioned from the coupler.
- It is recommended that the blades be as far into the corners of the vessel as practicable.

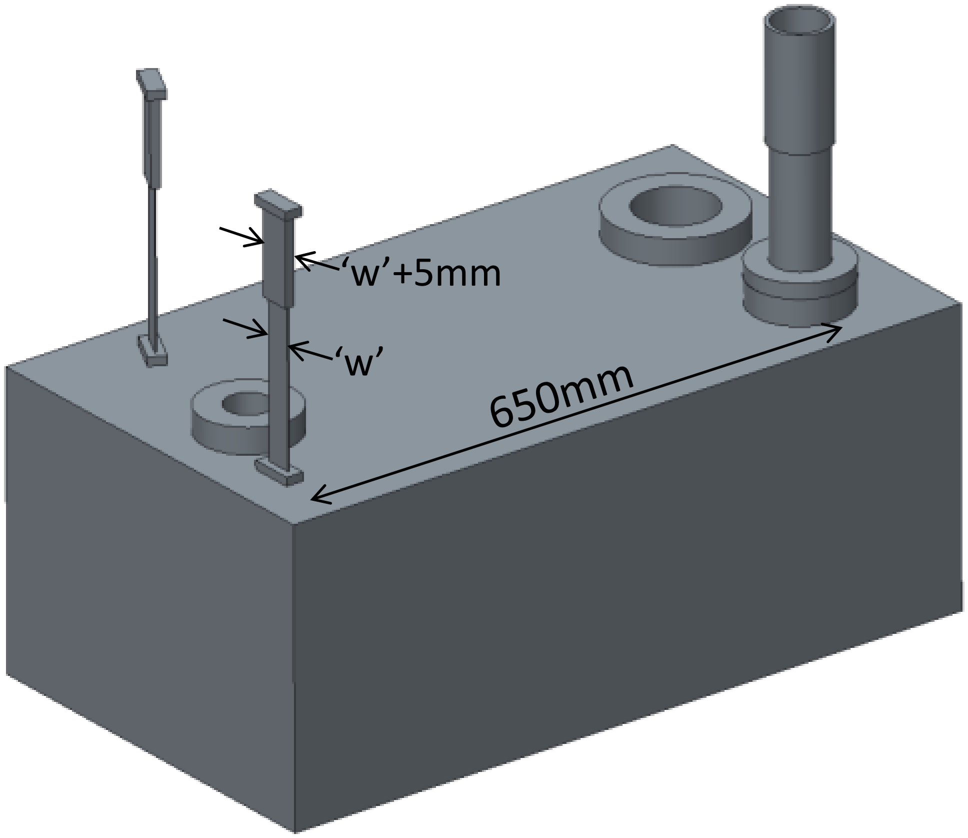
Blade width sweep

Width ‘w’ varied from 20mm to 60mm.

There is a limit of 0.3W per flexure heat load at 2 Kelvin.

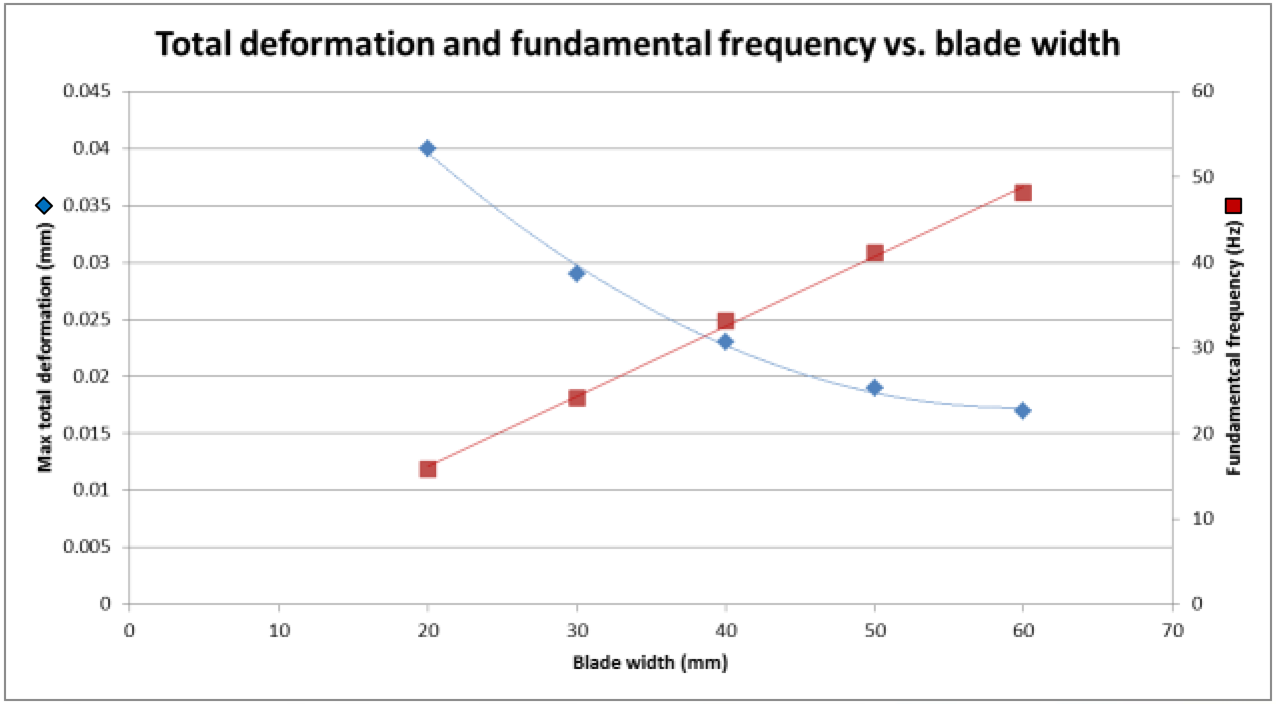
Width of thicker part was kept 5mm wider.

Thickness remained constant at 2mm.

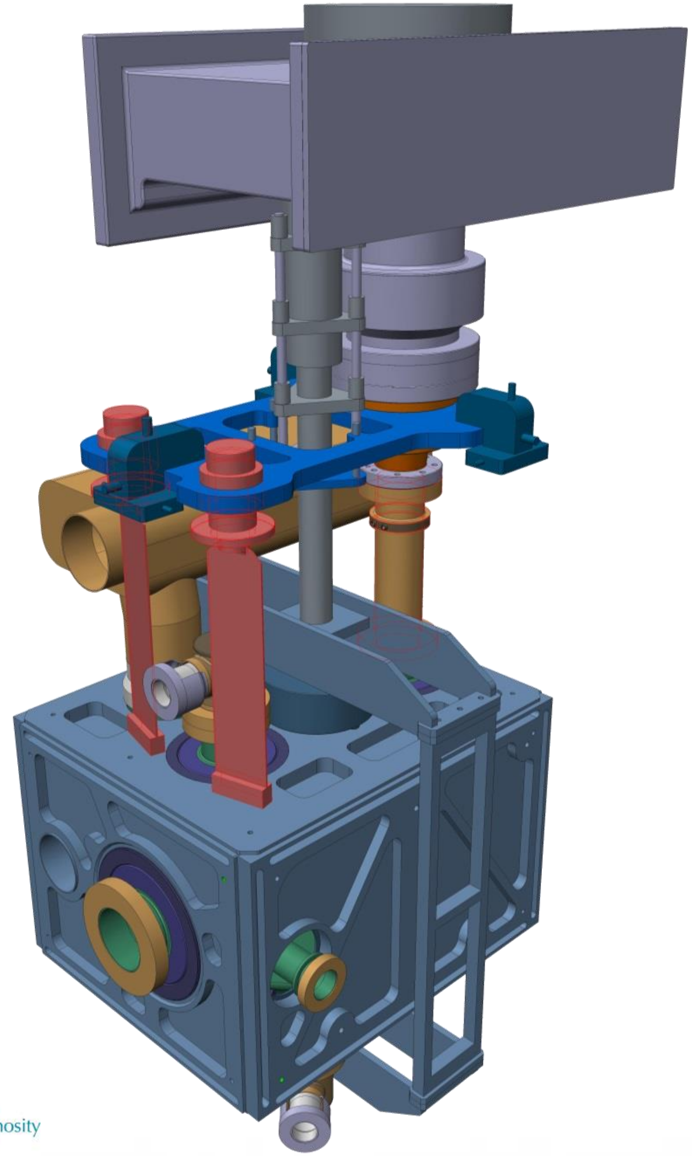


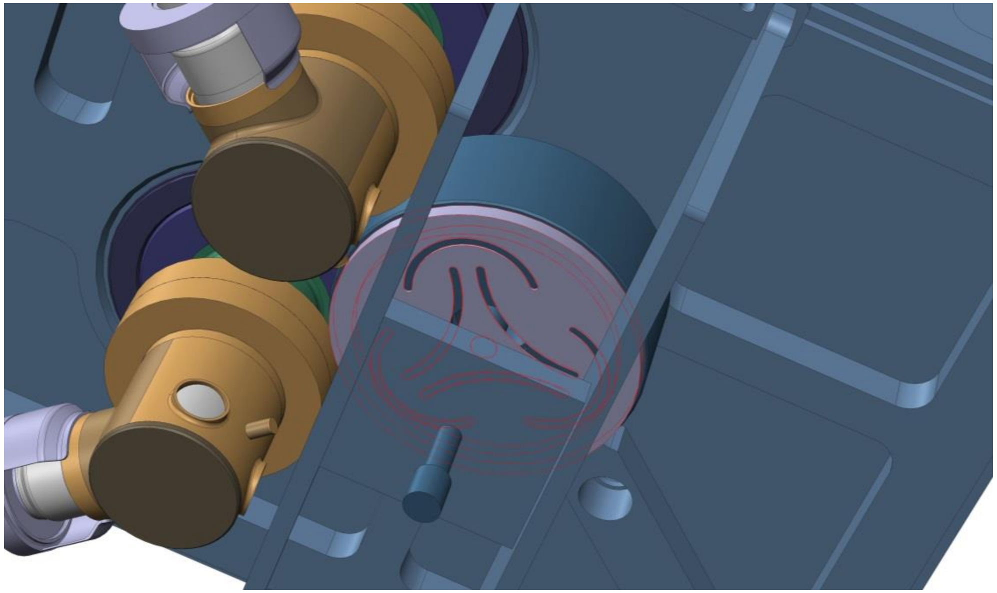
Result comment

- As expected maximum deformation and stress is reduced by increasing blade width.
- The frequency of vibration modes increases significantly with increasing blade width.
- However, increasing the blades does carry the risk of coupling to spikes in ground vibration.
- Therefore it was determined that detailed models should be studied for each cavity type to determine the most accurate modal results possible and then to assess transmission from the ground.



Final design

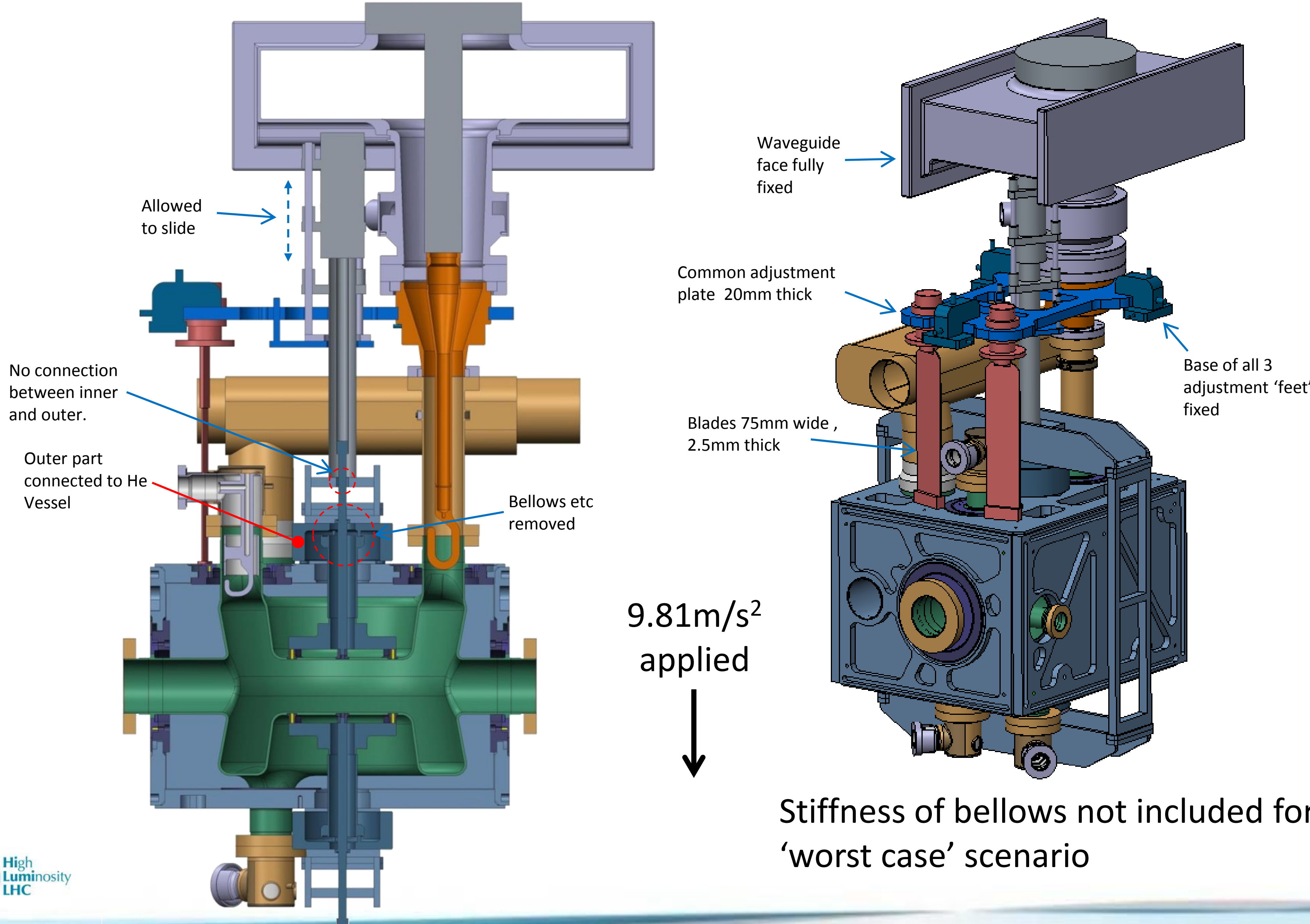




Above: Tuner light-weighted and flexure added to base of tuner assembly to link it to the helium vessel. This will eliminate previous low modes of tuner and unwanted stresses in the cavity under transport loads.

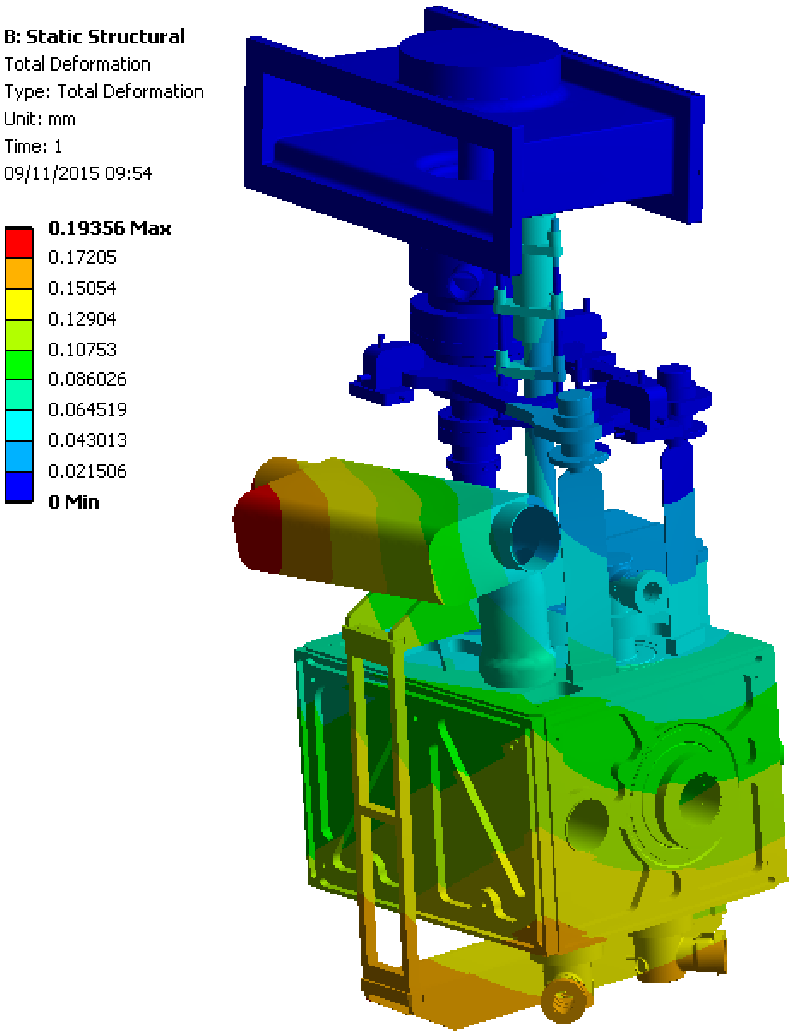
Left: Cavity support flexures increased in width to 75mm. This will increase stiffness in X direction and increase natural frequencies of the system.

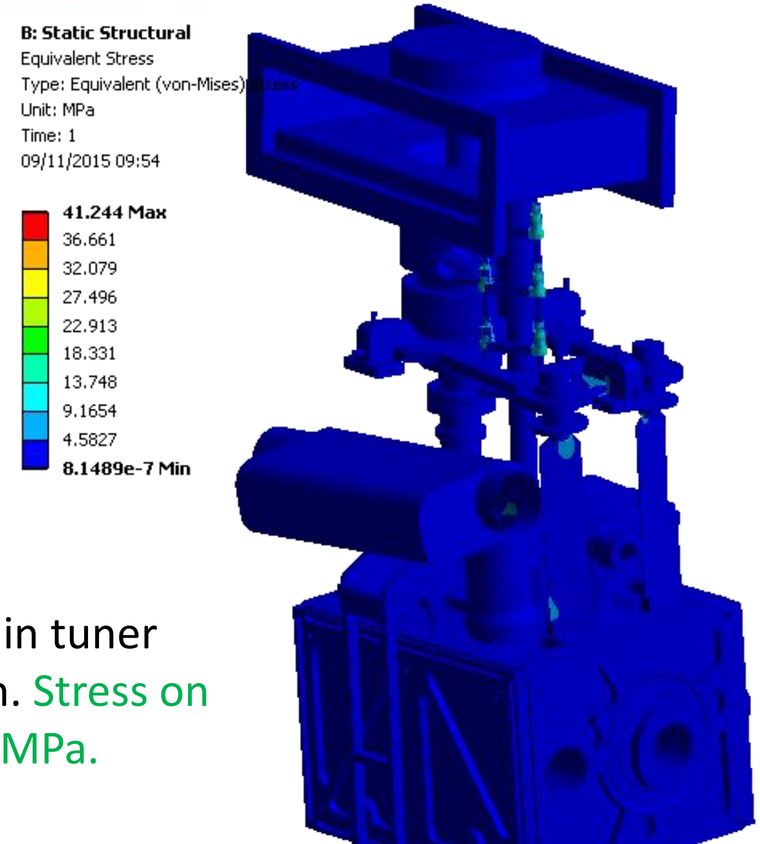
Structural and Modal FEA setup



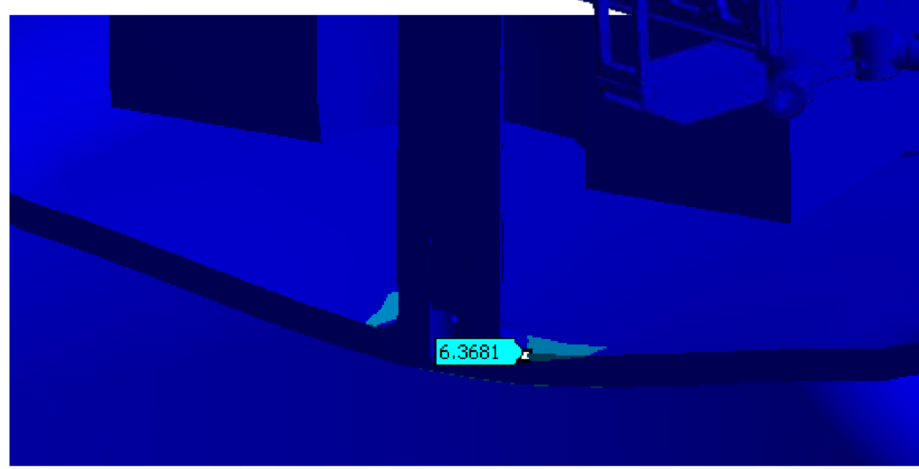
Stiffness of bellows not included for 'worst case' scenario

Structural result





Max stress in tuner mechanism. Stress on Cavity ~6.4MPa.



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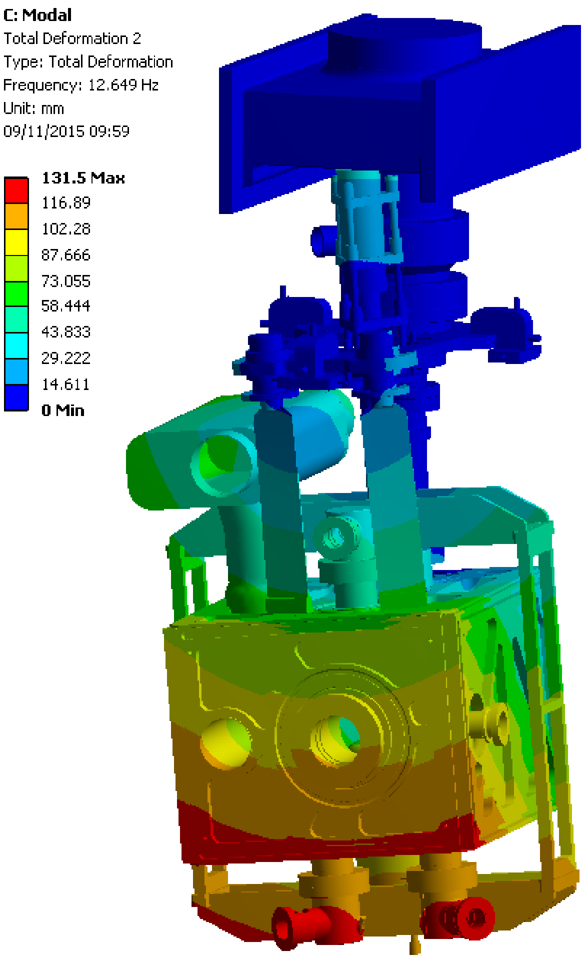
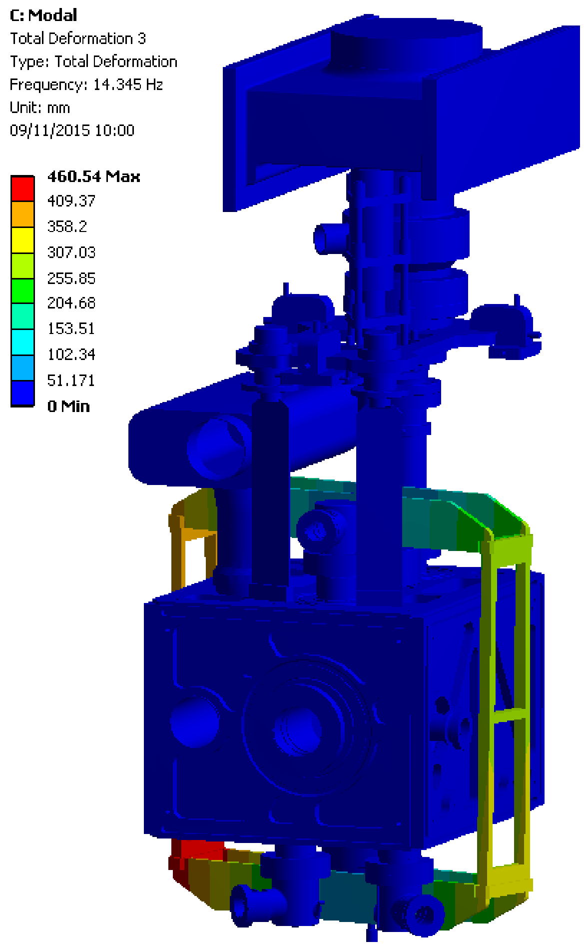
Modal result

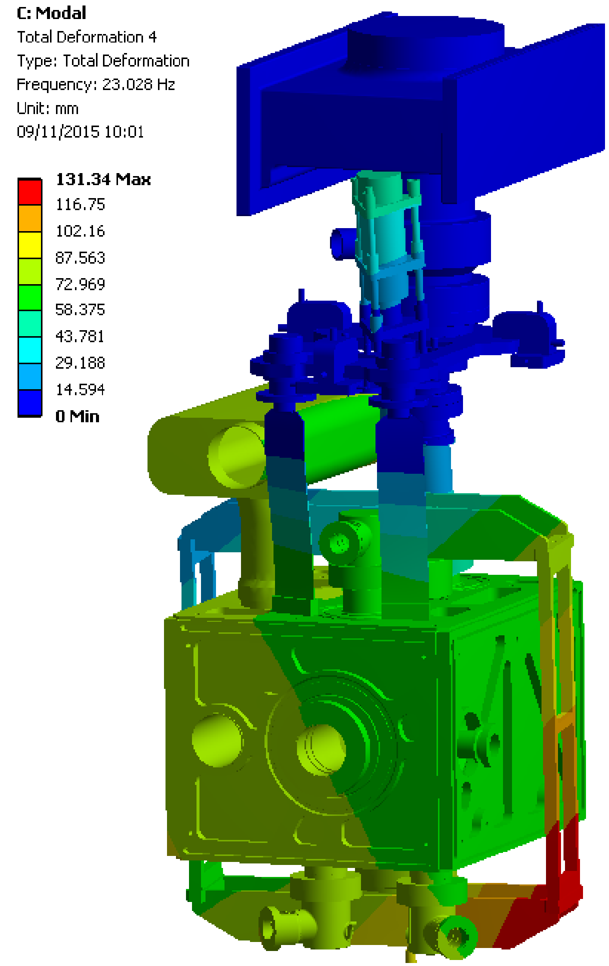
Mode 1



Mode 2



Mode 3



Modal summary

Mode	Frequency (Hz)	Description
1	12.649	Cavity and tuner swinging laterally
2	14.345	Tuner twist about central axis
3	23.028	Cavity and tuner swinging longitudinally
4	31.464	Cavity and tuner rotation about coupler axis
5	34.991	Tuner swinging longitudinally
6	35.683	Tuner swinging laterally
7	50.3	Tuner twist about central axis with frame twist
8	54.346	Helium reservoir tank motion
9	55.994	Helium reservoir tank motion
10	56.748	Tuner bending at corners

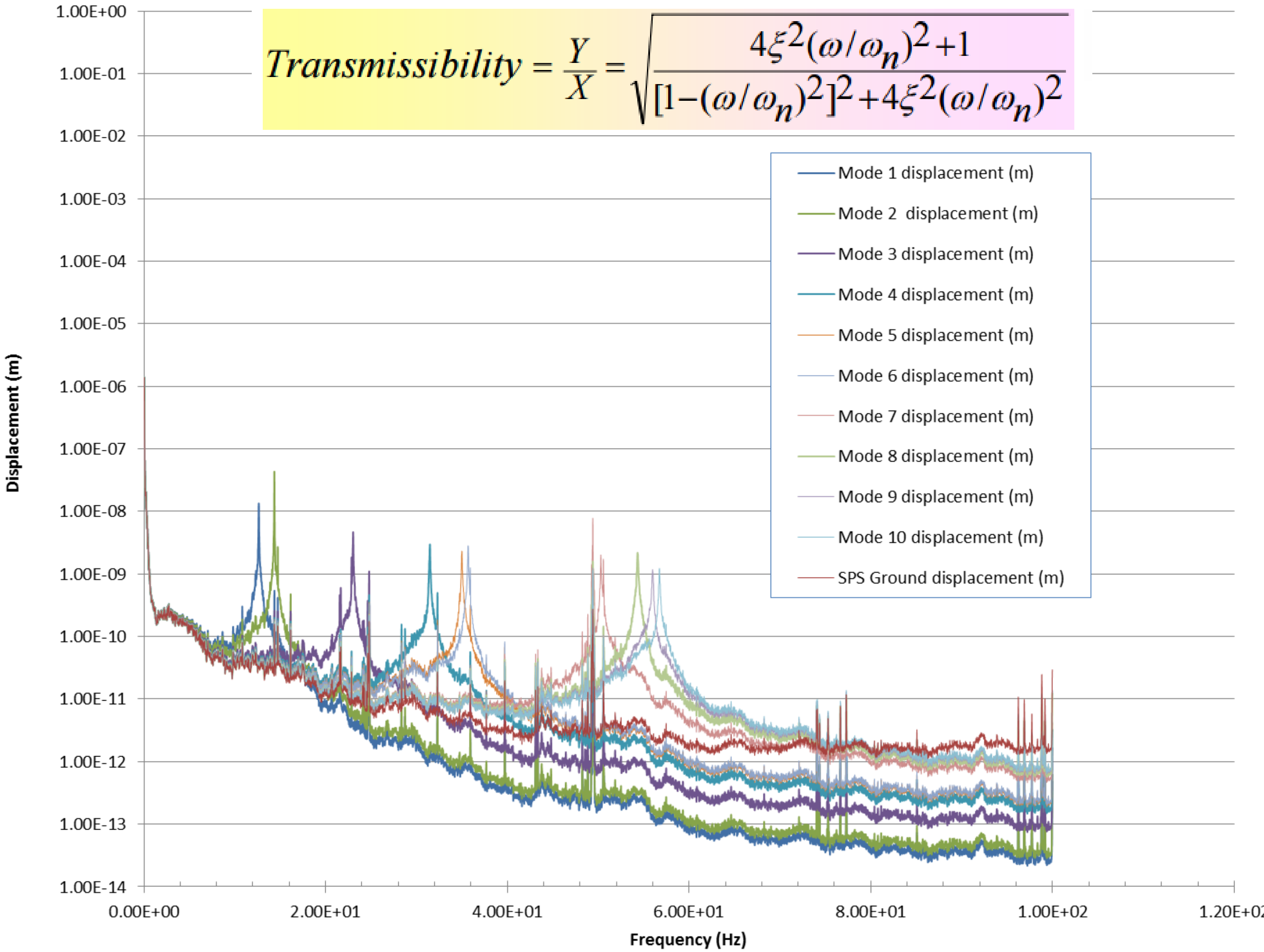
Response spectrum - SPS

Ground amplitudes are larger at lower frequencies, therefore we aim to increase the values of fundamental modes.

SPS has spike at 49.4Hz, close to the 50.3Hz mode of the tuner. There is some amplification but below 10Nm. This is at extremities of the tuner, actual cavity deflection is <1Nm.

This data is for the relatively quiet SPS area which was measured with <u>Cryo OFF.</u>

The next slide shows the results for using the Diamond PSD data, which can in this instance be taken as a worst case. Although, Diamond is still a relatively quiet site.



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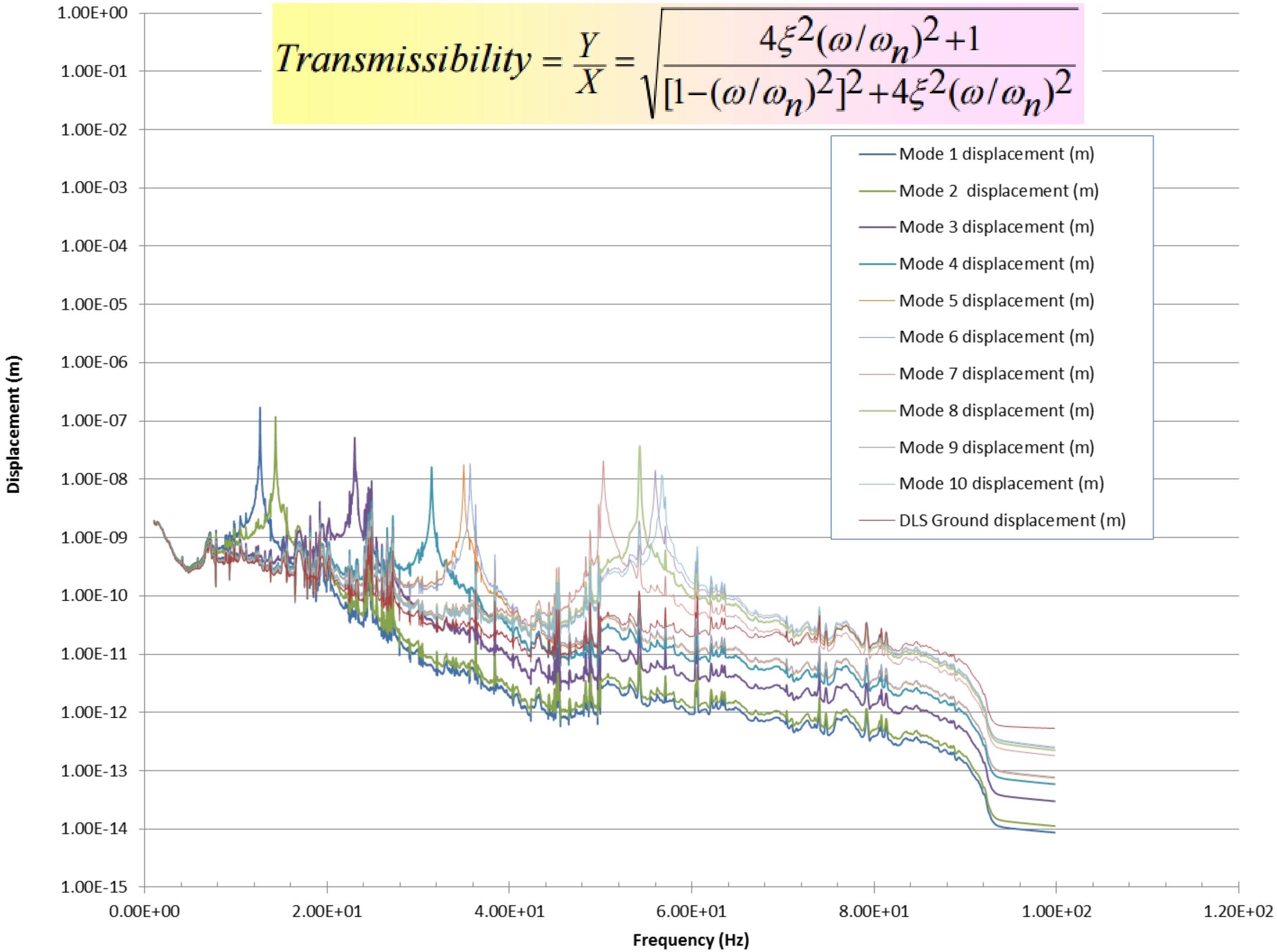
Response spectrum - DLS

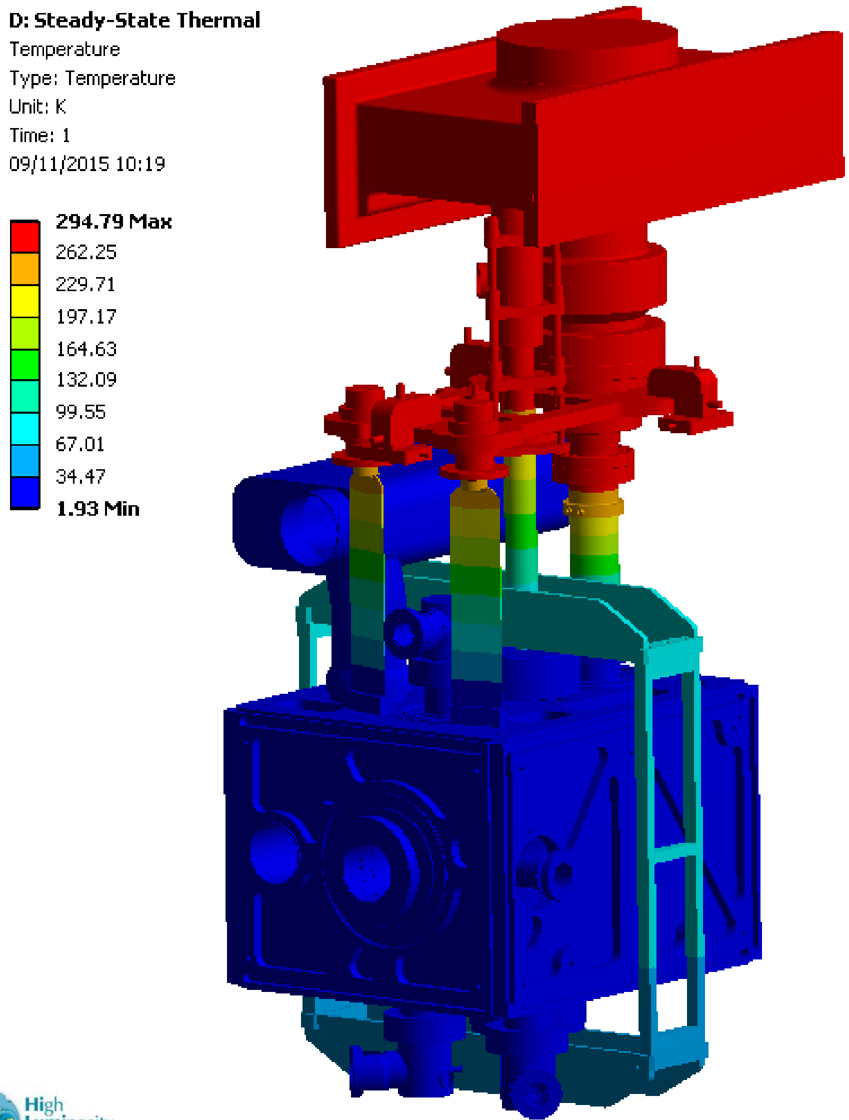
Vibration amplitudes now as high as 170nm.

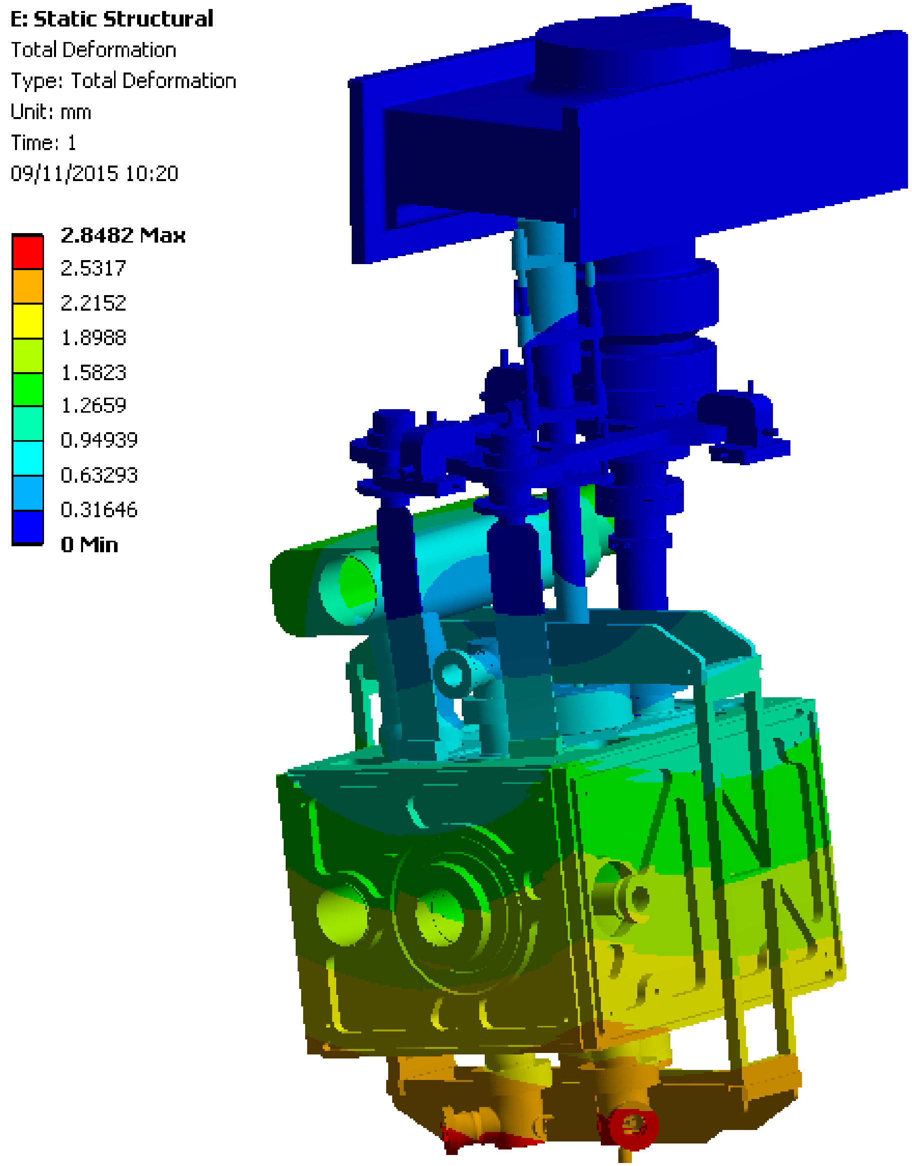
This highest amplitude will be at the extreme edges of the helium vessel/tuner.

At this mode there is no deformation of the cavity, it acts as one rigid body. Therefore detuning will be <1Hz.



Thermal analysis





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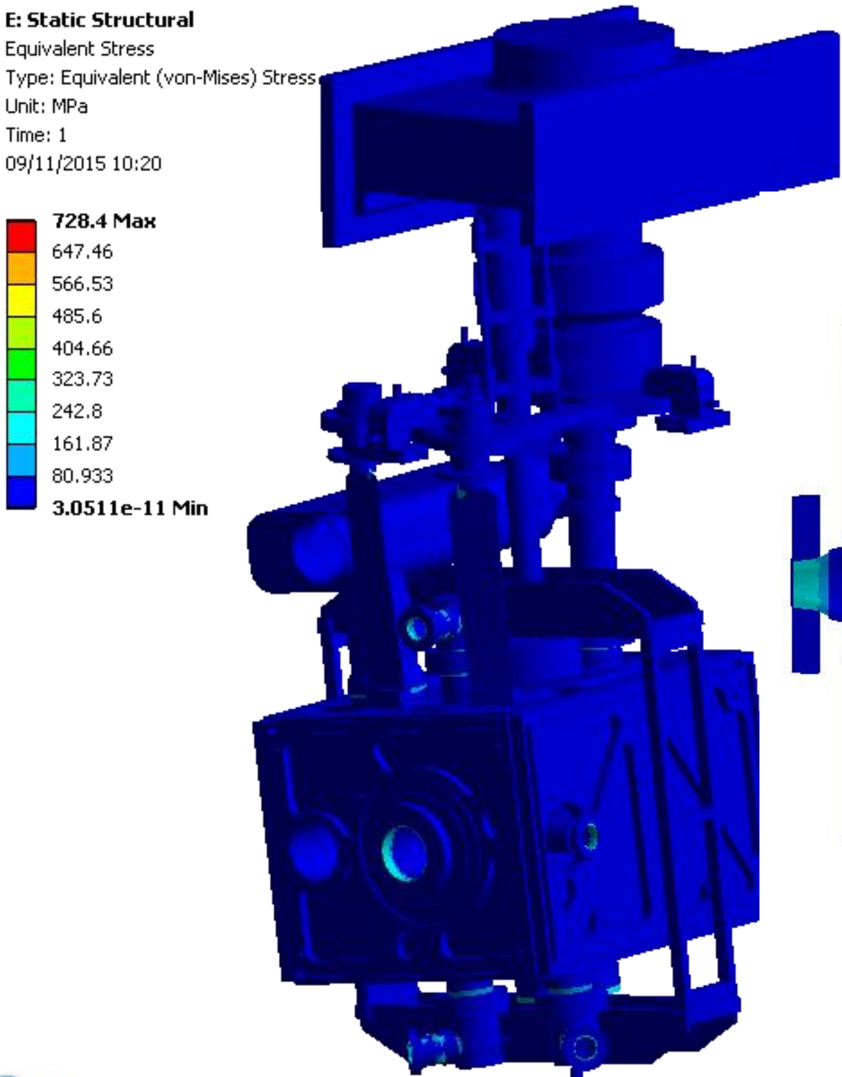
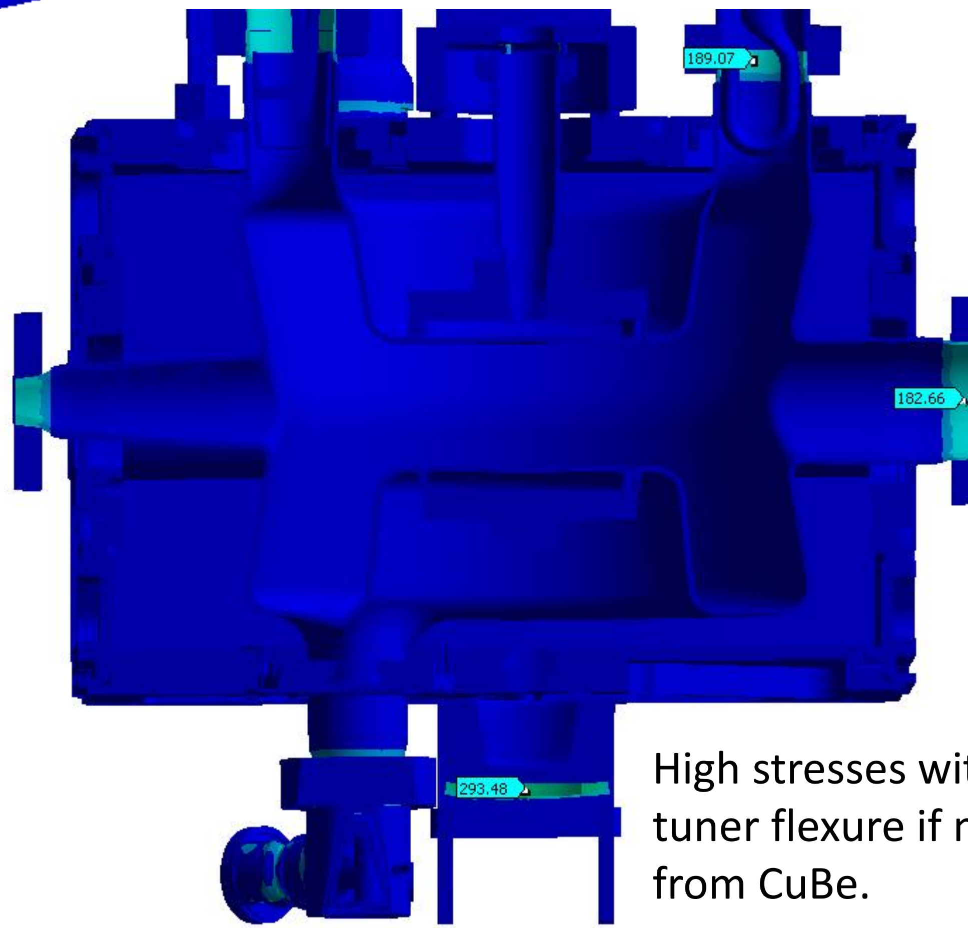
Thermal analysis

Higher stresses on interfaces between differing materials.

Stresses within acceptable limits in cavity.



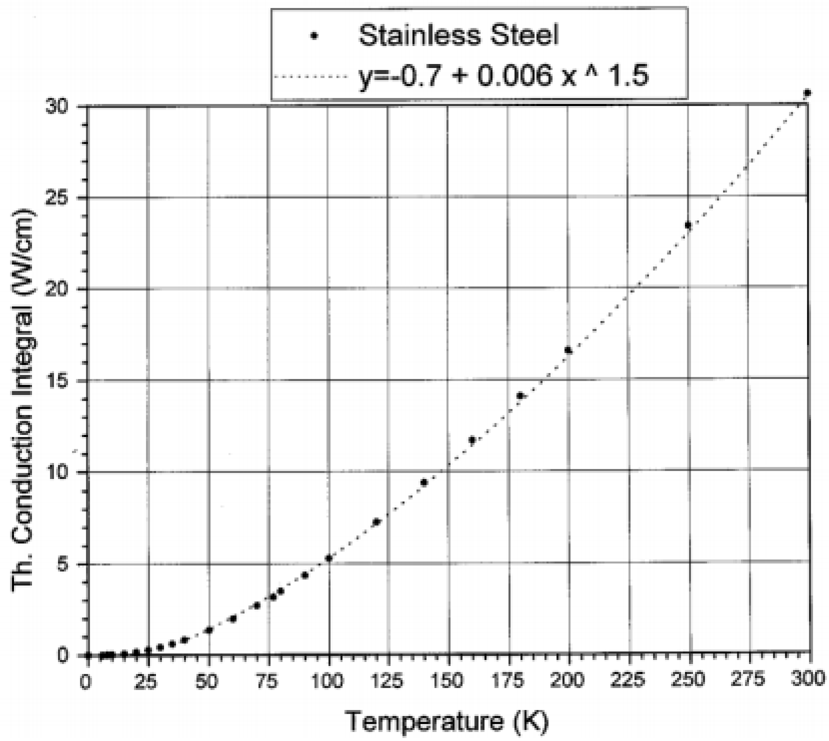


High stresses within tuner flexure if made from CuBe.

75mm wide support heat leak

$Q = (K_{dt}A)/X$ i.e. integral value already contains ΔT	
K_{dt} 300k to 80k (W/m)=	2743
K_{dt} 80k to 2K (W/m)=	317
Width of blade 300k to 80k (mm) =	75
Width of blade 80k to 2k (mm) =	75
Thickness of blade 300k to 80k (mm) =	2.5
Thickness of blade 80k to 2k (mm) =	2.5
Length 300k to 80k (mm) =	80
Length 80k to 2k (mm) =	220
Cross sectional area 300k to 80k (mm^2) =	187.5
Cross sectional area 80k to 2K (mm^2) =	187.5
$Q_{300k\ to\ 80k}$ (W)=	6.43
$Q_{80k\ to\ 2k}$ (W)=	0.27

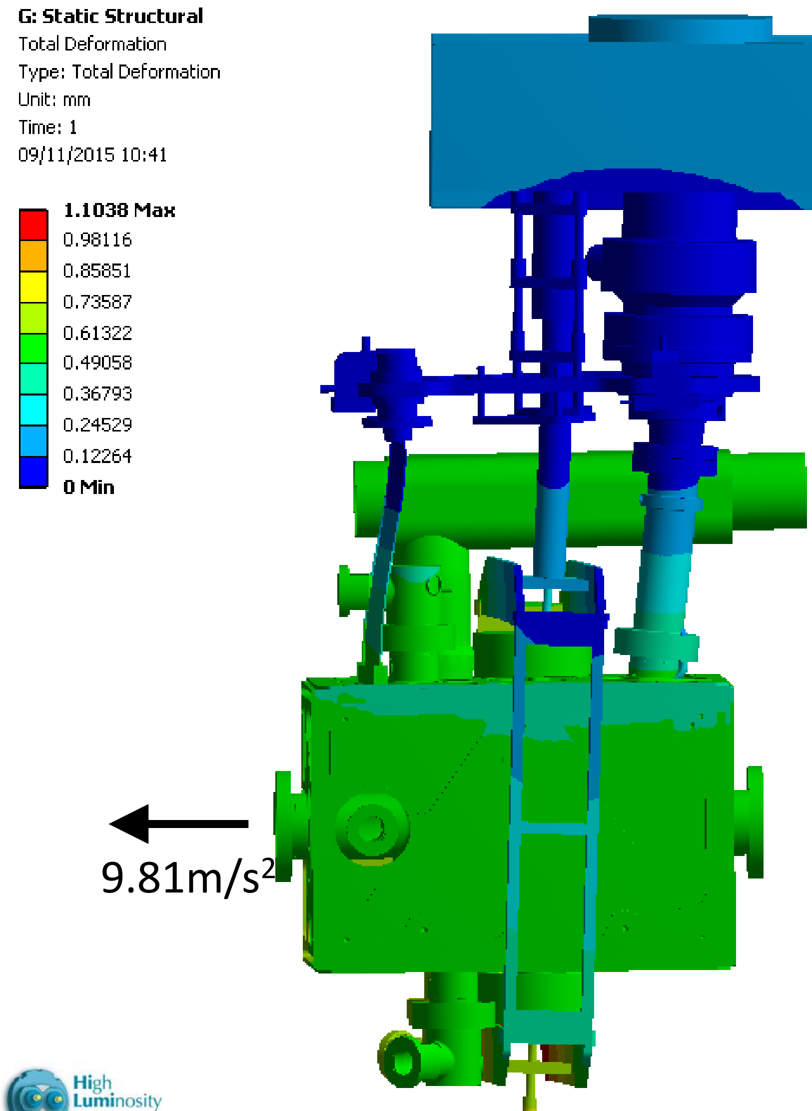
Thermal Conductivity Integral for Stainless Steel
Jacob W. Kooi

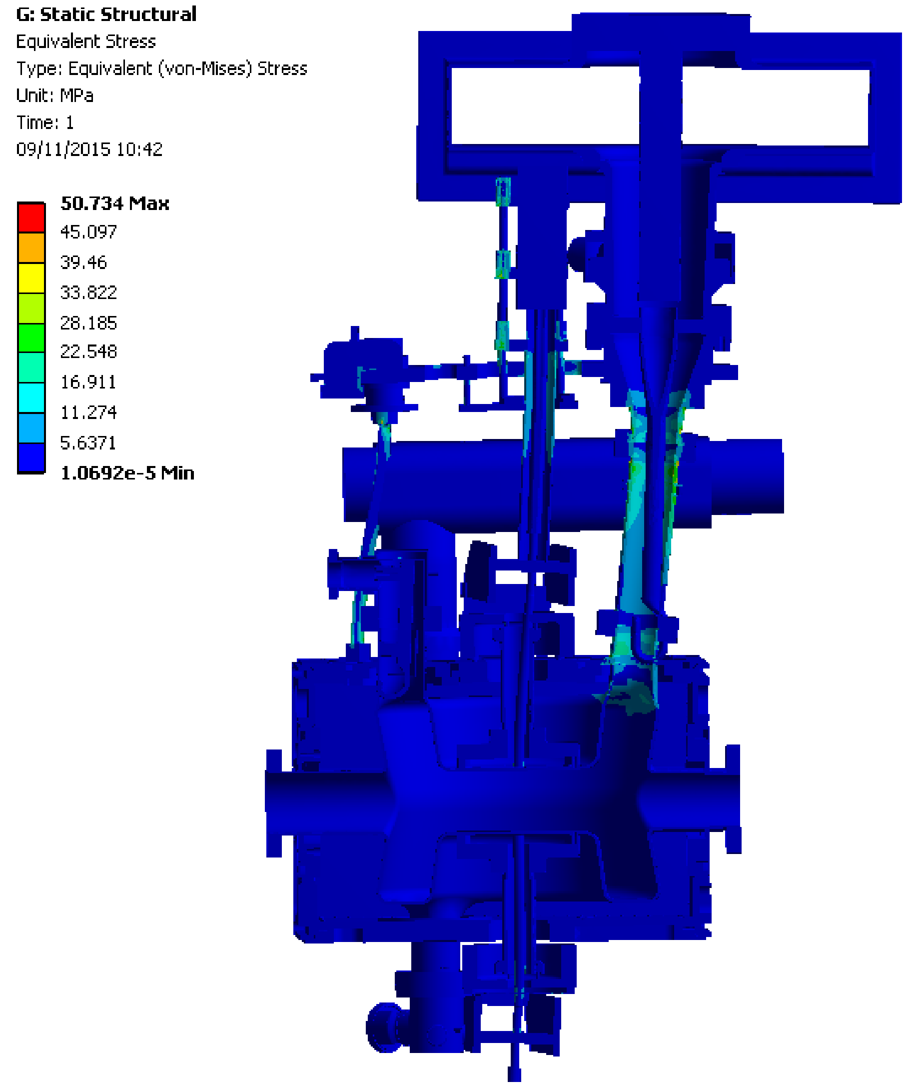


6.43W per support to thermal shield – Total 25.72W

0.27W per support to 2K – Total 1.08W

1G longitudinal acceleration – Transportation load

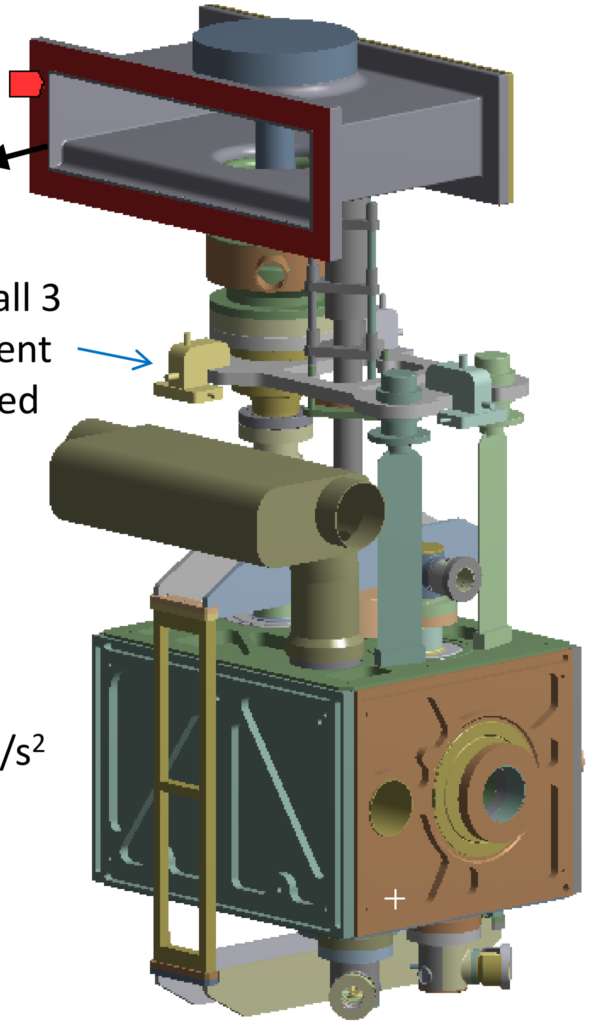




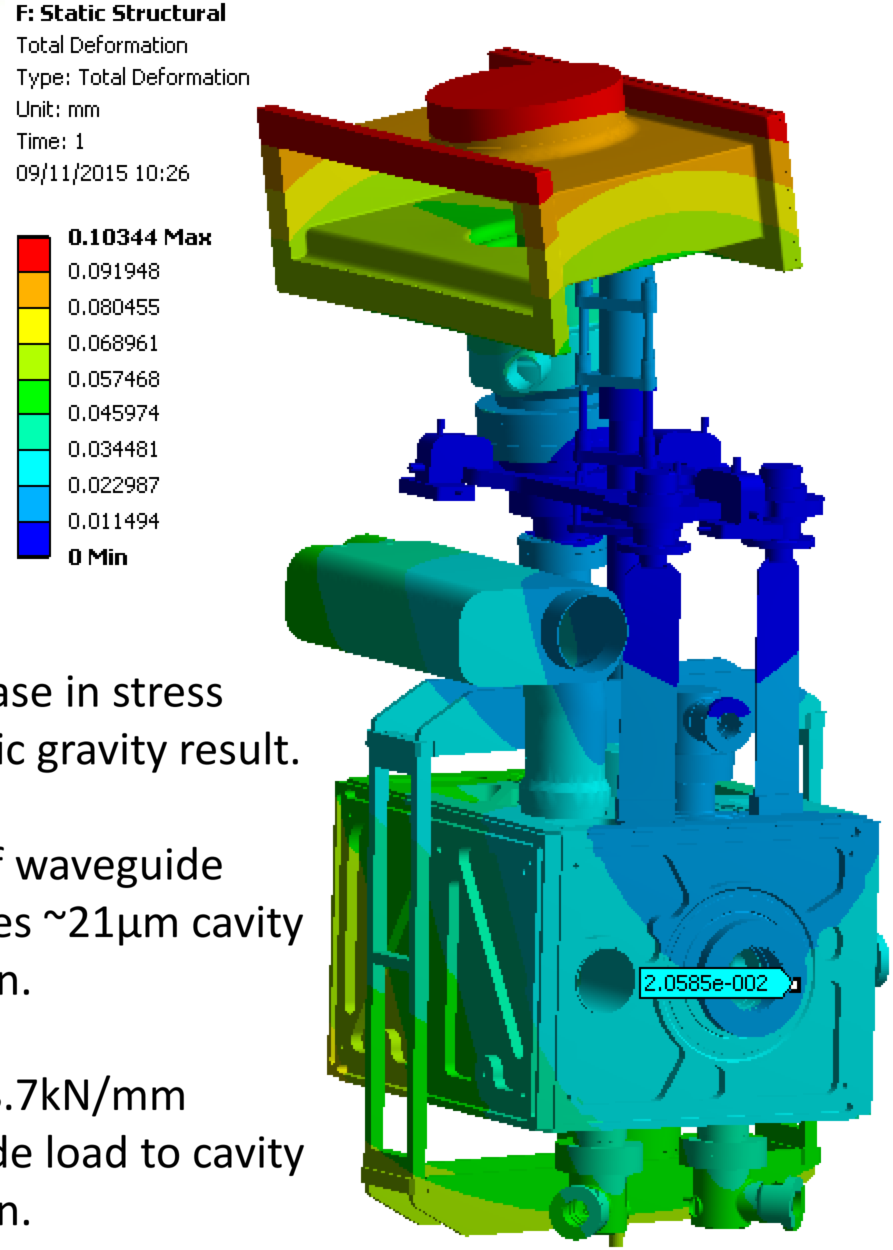
Waveguide bellows

1000N

Base of all 3 adjustment 'feet' fixed



9.81m/s^2



No increase in stress over static gravity result.

1000N of waveguide force gives ~21µm cavity deflection.

This is 48.7kN/mm waveguide load to cavity deflection.

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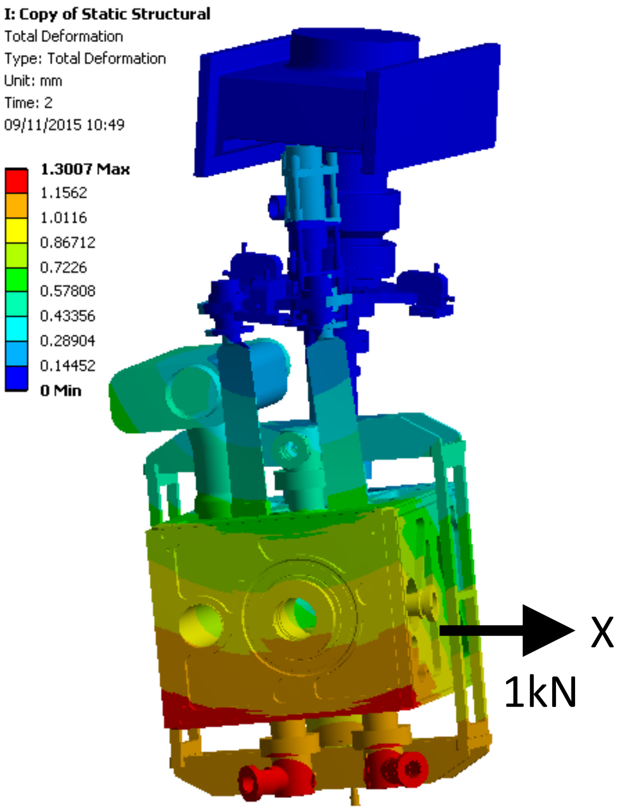
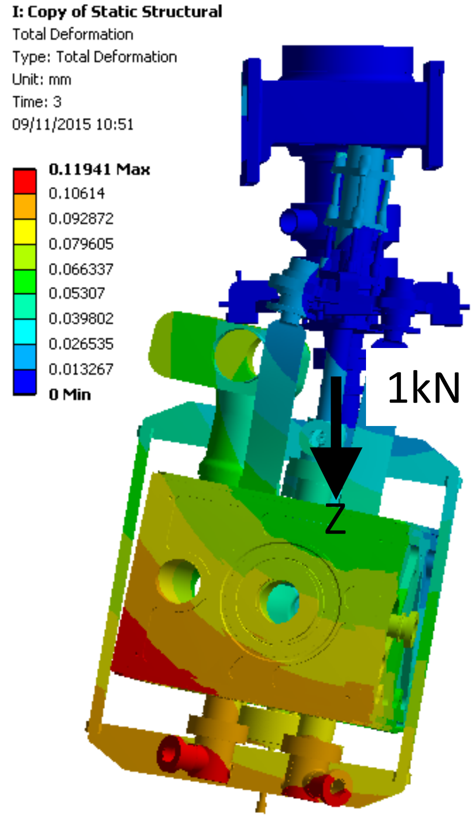
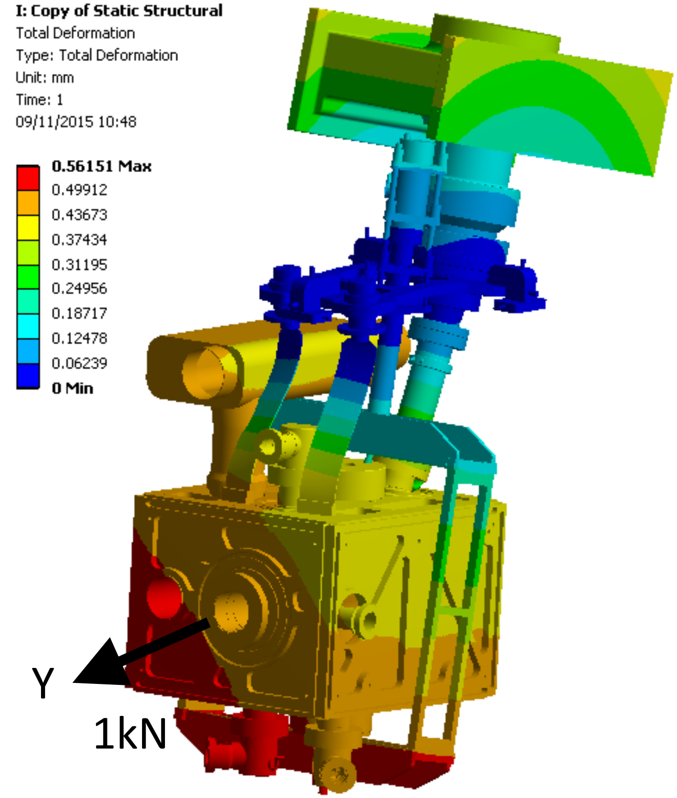
Directional Stiffness







Forces applied normal to appropriate helium vessel face

Direction	Stiffness
X	0.77kN/mm
Y	1.78kN/mm
Z	8.37kN/mm

Resistive stiffness of all bellows to be determined

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Conclusions and further work

- The support system for the HiLumi Crab cavities has been studied and then optimised.
- The current configuration gives acceptable stress and deformation results for the following load conditions;
    - 1G vertical load
    - SPS and DLS vibration conditions
    - Thermal loads
    - 1G longitudinal load (transport condition)
- The stiffness of the system has also been calculated ;
    - X - 0.77kN/mm
    - Y - 1.78kN/mm
    - Z - 8.37kN/mm
    - Waveguide bellows affect – 48.7kN/mm
- Further work
    - Perform ANSYS random vibration response analysis of complete system
    - Assess rotational stiffness if required
    - Assess stiffness of bellows around FPC and blade supports and determine their affect
    - Repeat detailed analyses for RFD cavity

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LHC

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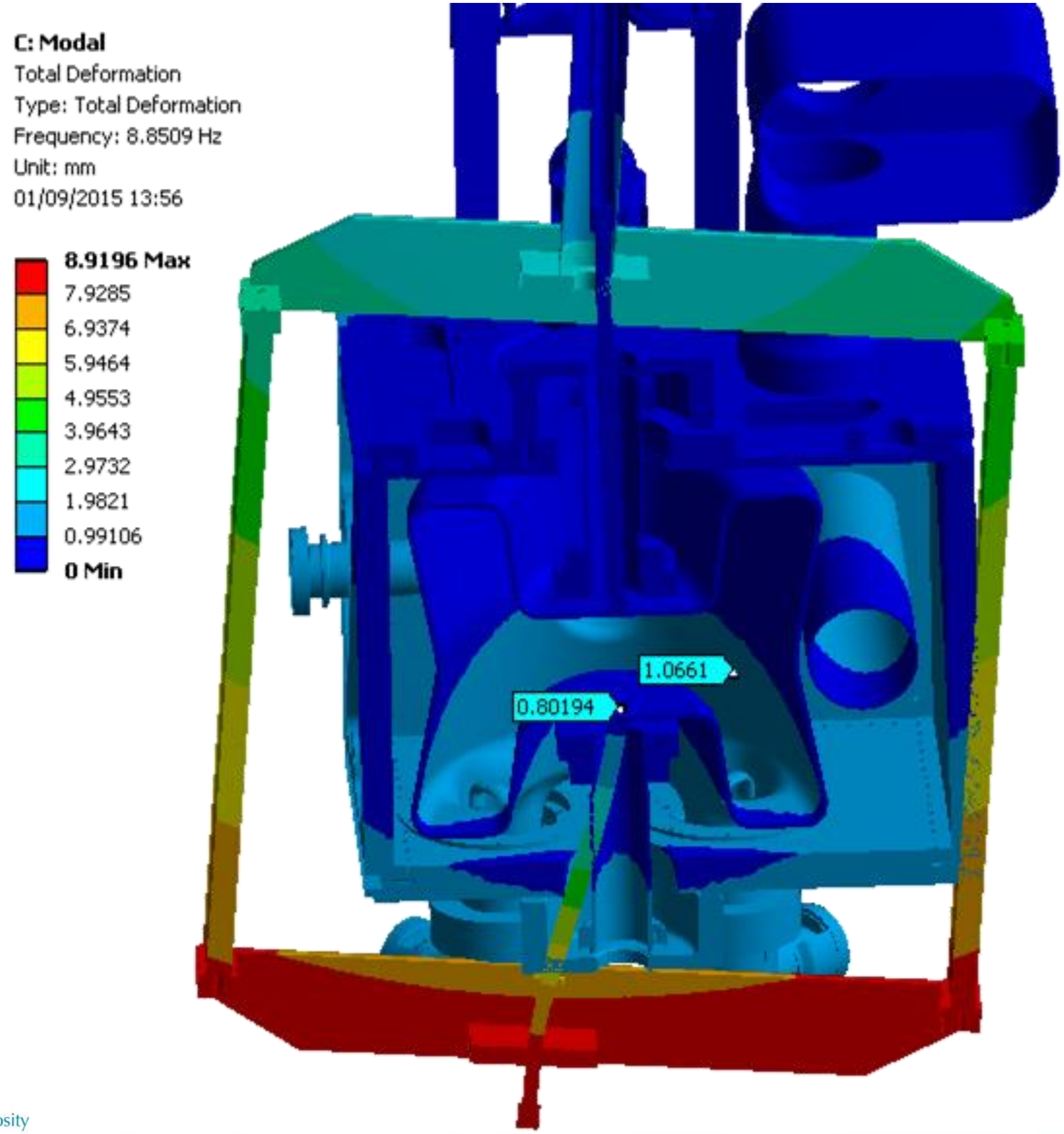
Additional Slides

Previous Mode 1 Cavity deflection



Relative cavity deflection due to 1st mode tuner movement ~10% of maximum deflection.

Ignore actual value, for modal results these are purely relative movements. There is no activation or damping included.

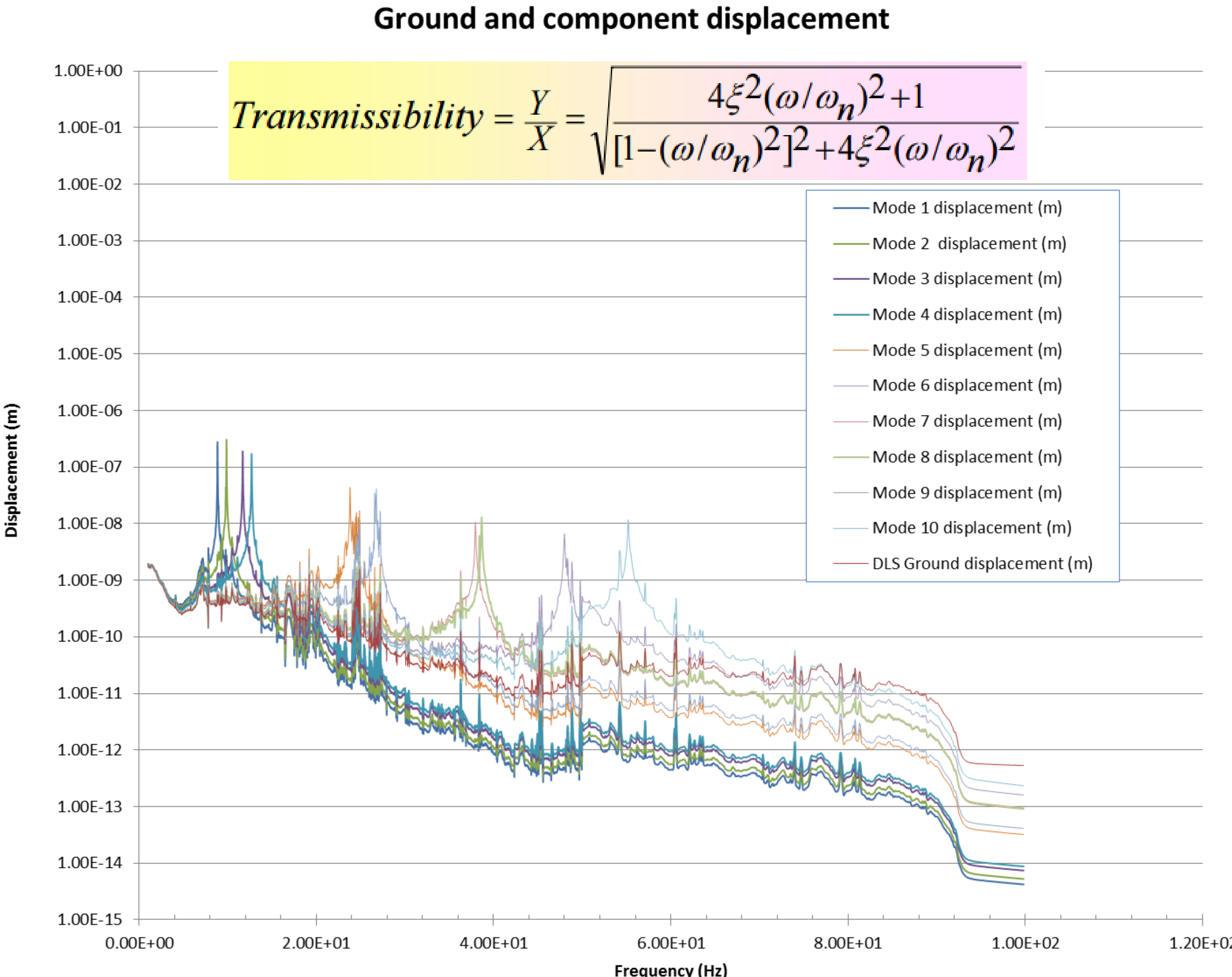
Previous Response spectrum - DLS

Vibration amplitudes now as high as 300nm.

This is above the specification limit of 100nm.

However, this deflection will be at the extremities of the tuner. The actual cavity deflection is ~10% of this as can be seen on slide 16.

Therefore cavity deflection due to vibration is ~30nm.



$$Transmissibility = \frac{Y}{X} = \sqrt{\frac{4\xi^2(\omega/\omega_n)^2+1}{[1-(\omega/\omega_n)^2]^2+4\xi^2(\omega/\omega_n)^2}}$$

Previous Frequency shift

- Paper at IPAC '15 – MOBD2 by Silvia.
- Tuning sensitivity of 372kHz/mm in this region
- If deformation is 30nm this is a frequency shift of 11.2Hz.
- This is significantly below the tuning resolution of 100Hz.
- Should tuning resolution be reduced these microphonics may become more of an issue.
- It is also worth noting that vibration studies should be taken with a high safety factor, as a small shift in resonant frequency can lead to larger changes in amplitude.

Validation

1. One Degree of Freedom Torsional system

Consider the one degree of freedom systems shown in figures 1, and 2. Figure 1 represents a torsional system and figure 2 represents a translational system.

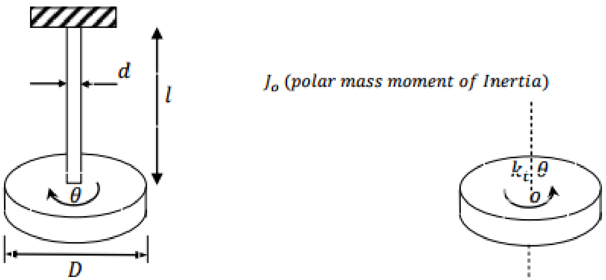


Figure 1

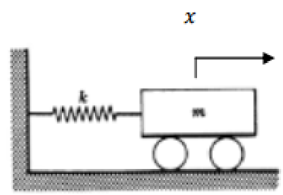


Figure 2

Both of these systems are represented by similar equations of motion. Newton's second law is used to determine these equations.

$$\sum F_x = m\ddot{x} \Rightarrow m\ddot{x} + kx = 0 \quad \Rightarrow \omega_n = \sqrt{\frac{k}{m}}\ rad/sec \quad (1)$$

$$\sum M_o = J_o\ddot{\theta} \Rightarrow J_o\ddot{\theta} + k_t\theta = 0 \quad \omega_n = \sqrt{\frac{k_t}{J_o}}\ rad/sec \quad (2)$$

J_0 = Mass x r (distance to pivot)^2

In equation 2, k_t is the torsional spring constant of the shaft and J_0 is the polar mass moment of inertia for the disk. The torsional spring constant k_t is determined from the relationship between moment (M) and angular displacement (θ) of the shaft.

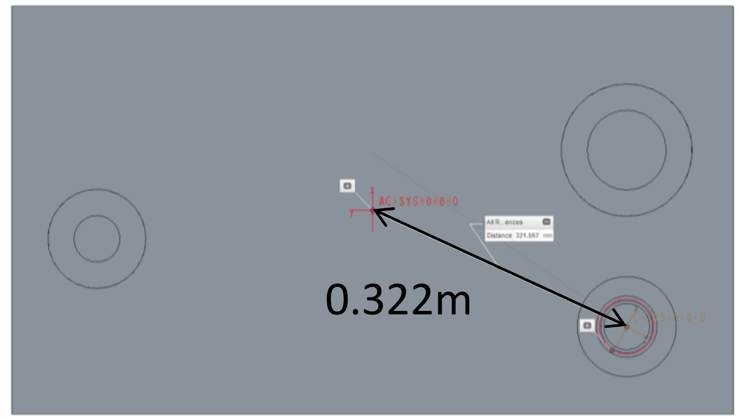
$$M_o = k_t\theta \quad also \quad M_o = \frac{GJ_p\theta}{l} \quad \Rightarrow \quad k_t = \frac{GJ_p}{l} \qquad (3)$$

Where G, J_p , l are the shear modulus, polar area moment of inertia, and the length of the shaf respectively. For a circular shaft J_p is given by $\frac{\pi d^4}{32}$ (in^4). Therefore

$$\Rightarrow \quad k_t = \frac{\pi G d^4}{32 l} \qquad (4)$$

Equation 4 is used to determine the natural frequency$(\omega_n, or\ f_n)$ of the system shown in figure 1.

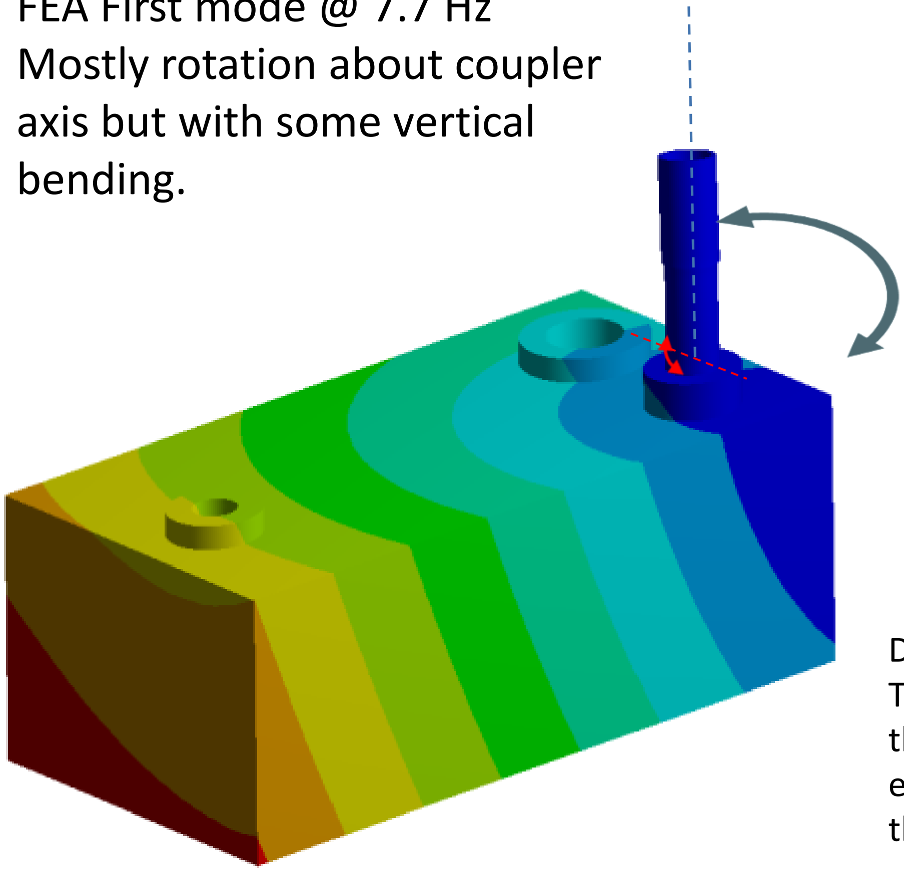
$$\omega_n = \sqrt{\frac{\frac{\pi G d^4}{32 l}}{J_0}} \left(\frac{rad}{sec}\right) \quad OR \quad f_n = \frac{1}{2\pi}\sqrt{\frac{\frac{\pi G d^4}{32 l}}{J_0}} \left(\frac{cycle}{sec}\right) \qquad (5)$$



r	0.322	m	distance to centre of mass
m	202	kg	supported mass
G	7.50E+10	Pa	shear modulus
d	6.30E-02	m	coupler diameter
l	3.00E-01	m	coupler length
t	1.50E-03	m	coupler wall thickness
j_p	2.7E-07	mm^4	polar area moment of inertia
j_0	20.7	$kg\ m^2$	polar mass moment of inertia
k_t	6.85E+04		torsional spring constant of coupler
f_n	**9.1**	**Hz**	**Natural frequency due to torsion**

Validation

FEA First mode @ 7.7 Hz
Mostly rotation about coupler axis but with some vertical bending.



r	0.322 m	distance to centre of mass
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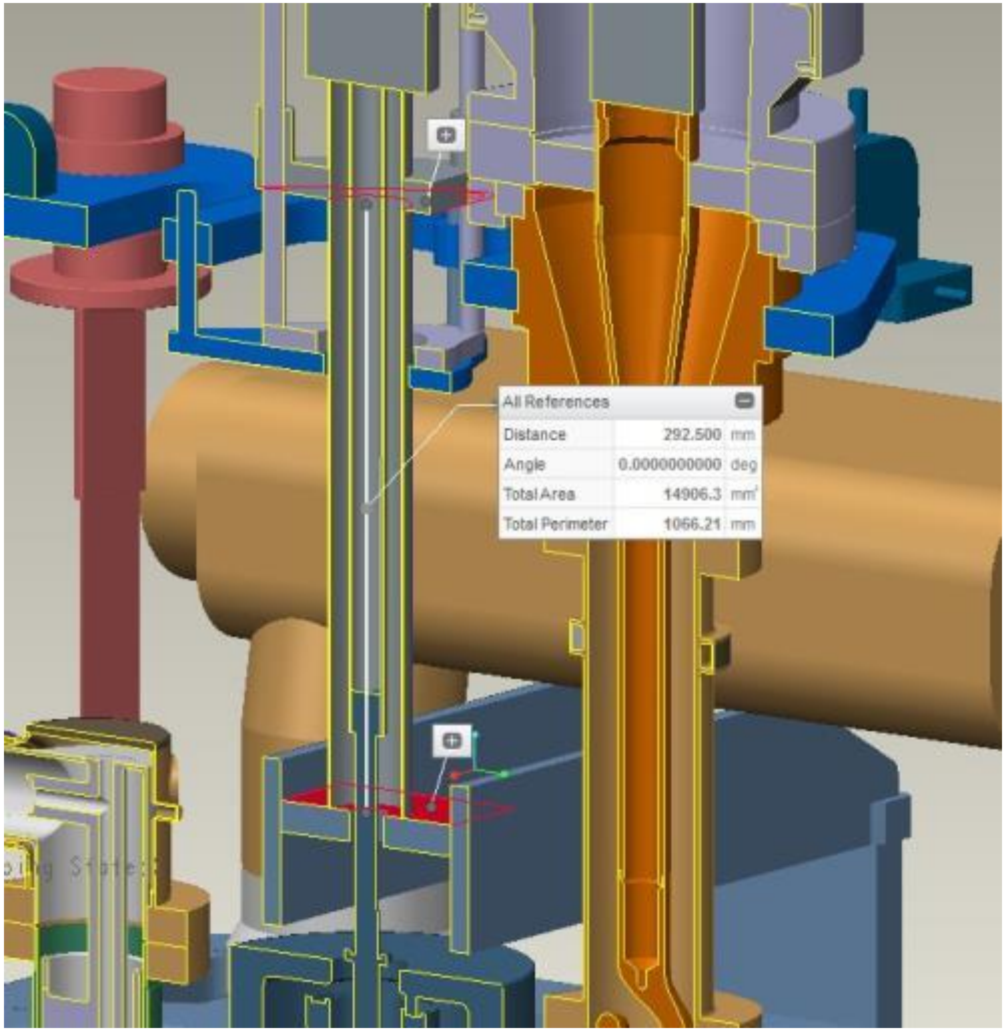
j_0	20.7 kg m²	polar mass moment of inertia
k_t	6.85E+04	torsional spring constant of coupler
f_n	**9.1 Hz**	**Natural frequency due to torsion**

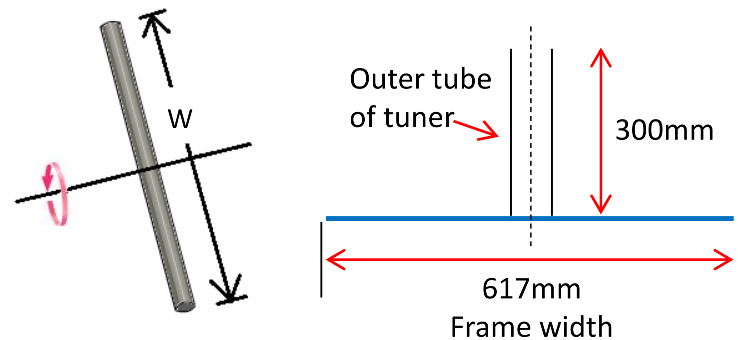
Difference of 1.4Hz between calculation and FEA.
This is due to the FEA model calculating some bending of the shaft in the vertical orientation, whereas the empirical model is based purely upon rotation/torsion of the shaft.

Validation of Tuner rotation mode



- Empirical model is less stiff as does not account for rods connected to cavity. Therefore mode is lower.
- However, this result is within 20% and shows that the tuner will exhibit this low frequency mode.

The moment of inertia of a thin rod with midpoint as the axis of rotation is, $I = \frac{1}{2} Mw^2$



w	6.17E-01 m	Width of coupler frame
m	22 kg	supported mass
G	7.50E+10 Pa	shear modulus
d	4.40E-02 m	outer diameter
l	2.90E-01 m	length
t	1.00E-03 m	wall thickness
j_p	6.25E-08 mm^4	polar area moment of inertia
j_0	4.19E+00 kg m^2	polar mass moment of inertia
k_t	1.56E+04	torsional spring constant of tuner
f_n	**9.9 Hz**	**Natural frequency due to torsion**

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FEA value – 11.7Hz

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Consider the one degree of freedom systems shown in figures 1, and 2. Figure 1 represents a torsional system and figure 2 represents a translational system.

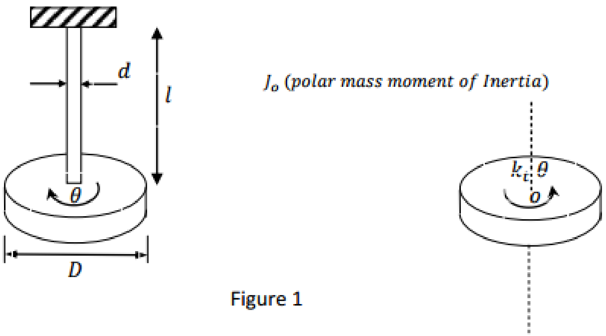


Figure 1

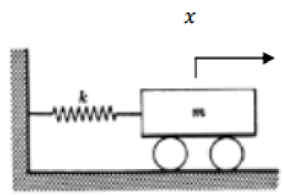


Figure 2

Both of these systems are represented by similar equations of motion. Newton's second law is used to determine these equations.

$$\sum F_x = m\ddot{x} \Rightarrow m\ddot{x} + kx = 0 \quad \Rightarrow \omega_n = \sqrt{\frac{k}{m}}\ rad/sec \quad (1)$$

$$\sum M_o = J_o\ddot{\theta} \Rightarrow J_o\ddot{\theta} + k_t\theta = 0 \qquad \omega_n = \sqrt{\frac{k_t}{J_o}}rad/sec \quad (2)$$

In equation 2, k_t is the torsional spring constant of the shaft and J_0 is the polar mass moment of inertia for the disk. The torsional spring constant k_t is determined from the relationship between moment (M) and angular displacement (θ) of the shaft.

$$M_o = k_t\theta \quad also \quad M_o = \frac{GJ_p\theta}{l} \quad \Rightarrow \quad k_t = \frac{GJ_p}{l} \qquad (3)$$

Where G, J_p , l are the shear modulus, polar area moment of inertia, and the length of the shaf respectively. For a circular shaft J_p is given by $\frac{\pi d^4}{32}$ (in^4). Therefore

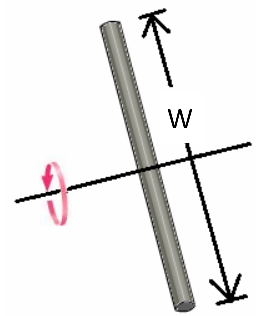
$$\Rightarrow \quad k_t = \frac{\pi G d^4}{32l} \qquad (4)$$

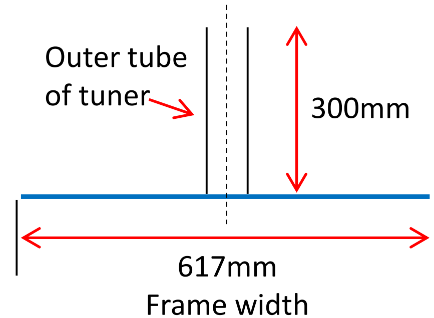
Equation 4 is used to determine the natural frequency$(\omega_n, or\ f_n)$ of the system shown in figure 1.

$$\omega_n = \sqrt{\frac{\frac{\pi G d^4}{32l}}{J_o}}\ \left(\frac{rad}{sec}\right) \quad OR \quad f_n = \frac{1}{2\pi}\sqrt{\frac{\frac{\pi G d^4}{32l}}{J_o}}\ \left(\frac{cycle}{sec}\right) \qquad (5)$$

- Empirical model less stiff as does not account for bottom rod connected to cavity. Therefore mode is lower.
- However, this result is within 20% and shows that the tuner will exhibit this low frequency mode.

The moment of inertia of a thin rod with midpoint as the axis of rotation is, $I = \frac{1}{2}Mw^2$

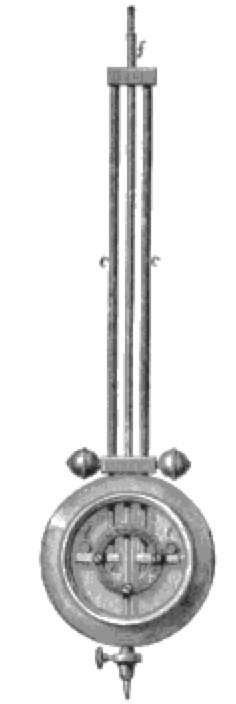




w	6.17E-01 m	Width of coupler frame
m	22 kg	supported mass
G	7.50E+10 Pa	shear modulus
d	4.40E-02 m	outer diameter
l	2.90E-01 m	length
t	1.00E-03 m	wall thickness
j_p	6.25E-08 mm^4	polar area moment of inertia
j_0	4.19E+00 kg m^2	polar mass moment of inertia
k_t	1.56E+04	torsional spring constant of tuner
f_n	**9.9 Hz**	**Natural frequency due to torsion**

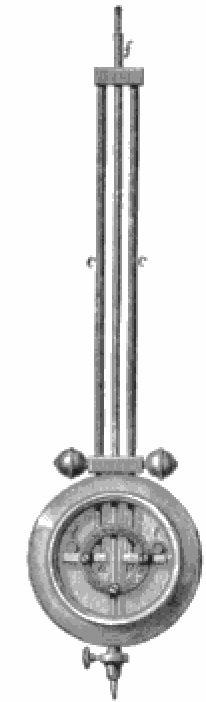
Inter-cavity supports

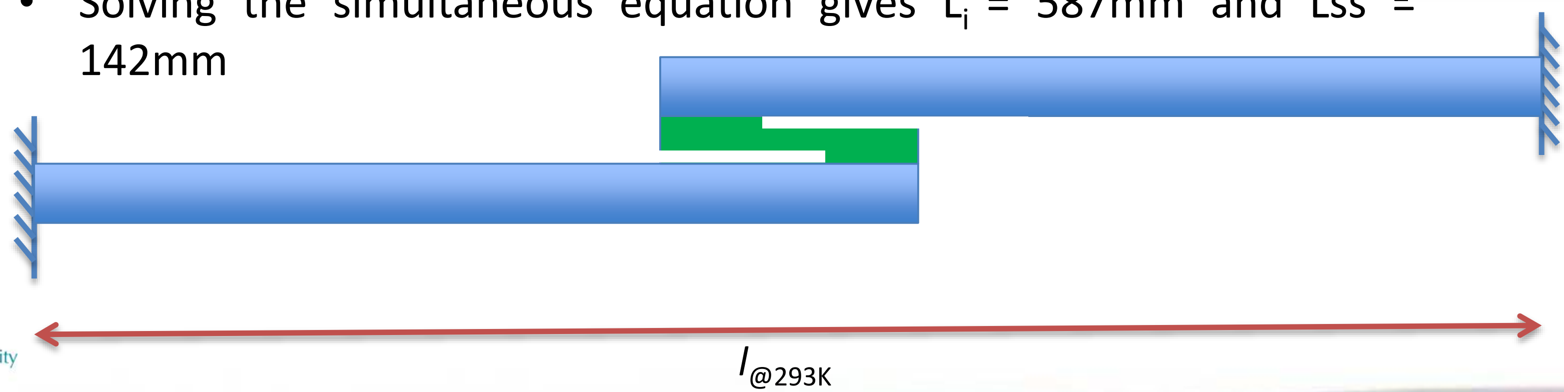
- A thermally neutral support system was developed for the UK4ROD.
- This relied on differences in thermal contraction of materials to maintain the same length after cooled down to 2K.
- It was shown to increase the stiffness and low frequency modes of the system.
- However, this was compared to a ‘coupler only’ type support.
- Also the UK4ROD had much more balanced forces, i.e. the coupler was in the centre of the cavity. This was by design, to improve the support system of the cavity.



Inter-cavity supports

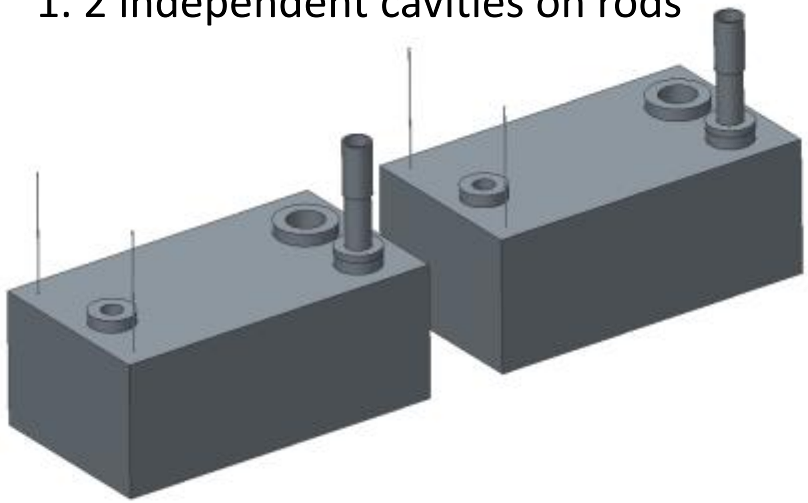
- Invar integrated contraction from room temperature to 2K is L_i x (0.037/100)
- 304 Stainless steel contraction from room temperature to 2K is L_{ss} x (0.306/100)
- Therefore;
  $2L_i$ x (0.037/100) = L_{ss} x (0.306/100)
  AND
  $2L_i - L_{ss}$ = 1032mm (distance between fixed points)
- So, 2Li x (0.037/100) = (2Li – 1032) x (0.306/100)
- Solving the simultaneous equation gives L_i = 587mm and Lss = 142mm



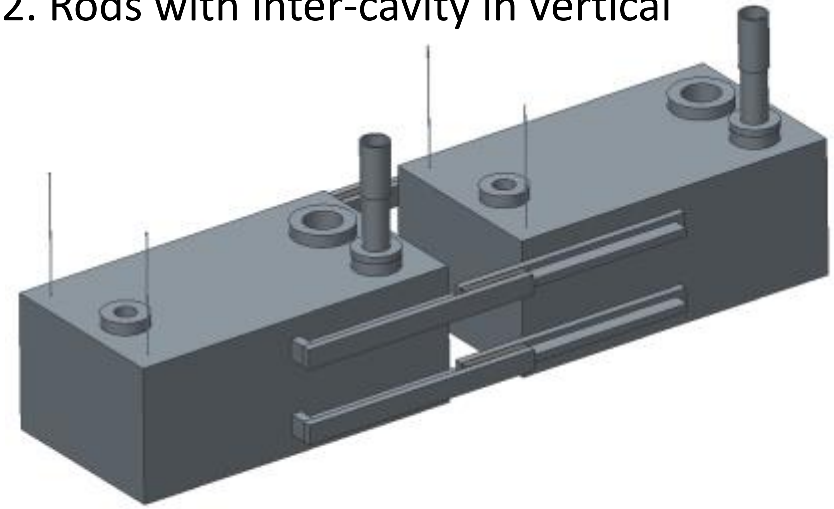


Analysis Models

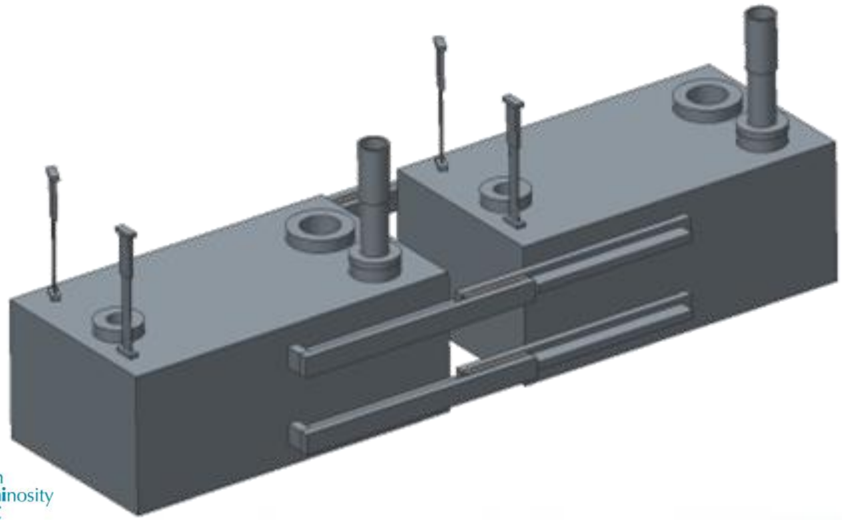
1. 2 independent cavities on rods



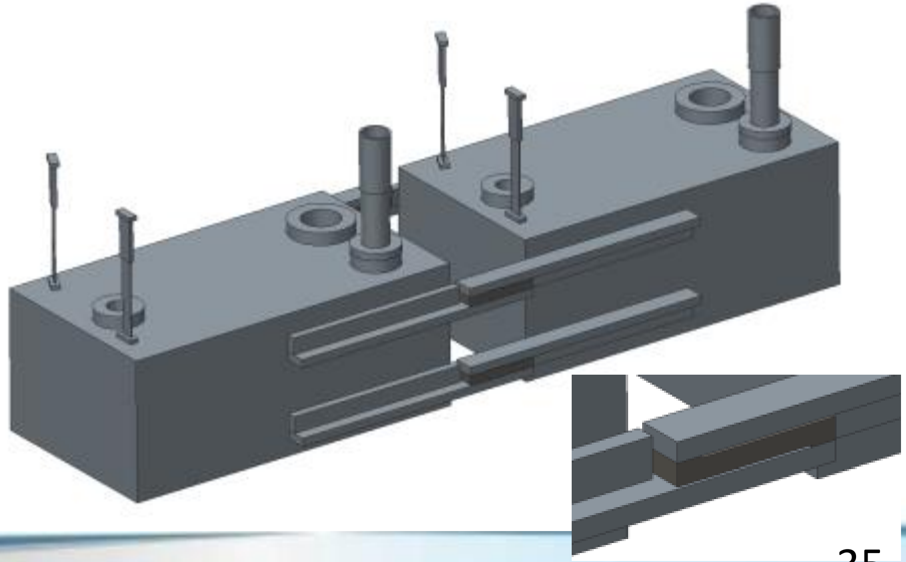
2. Rods with inter-cavity in vertical



3. Blades with inter-cavity in vertical



4. Blades with inter-cavity in horizontal



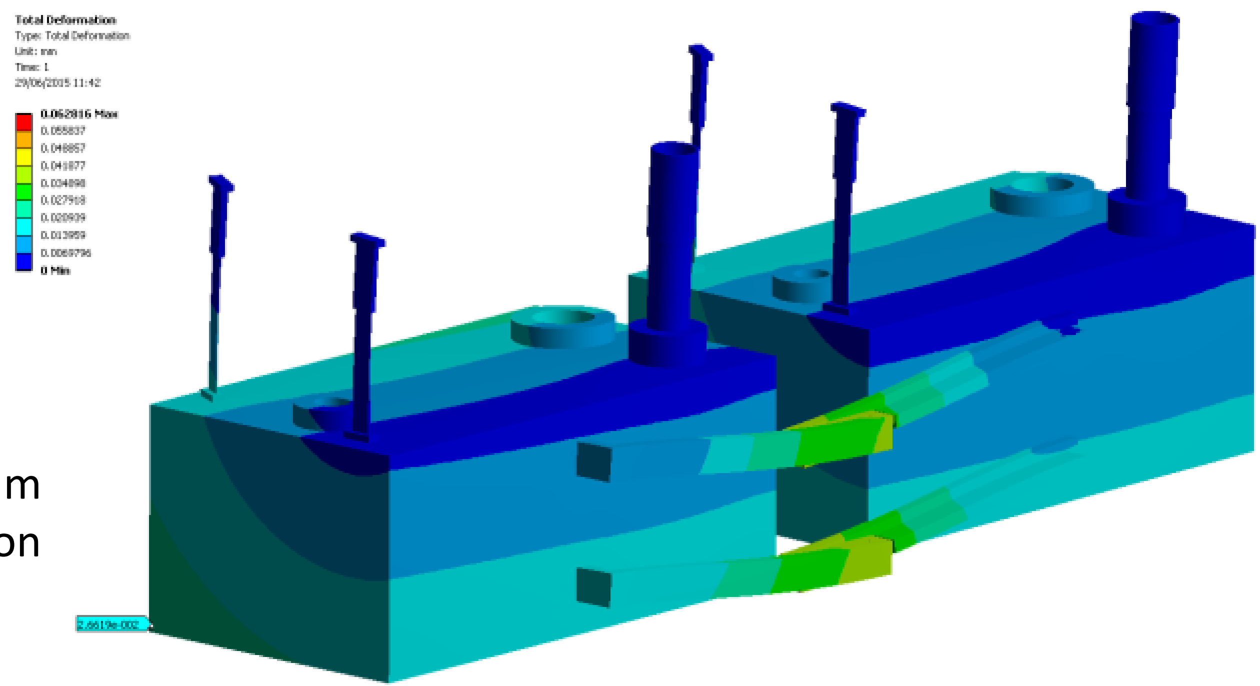
High Luminosity LHC

Result summary

Analysis	Max Deformation (mm)	Max von-Mises stress (MPa)	Mode 1 Frequency (Hz)	Mode 2 Frequency (Hz)	Mode 3 Frequency (Hz)	Mode 4 Frequency (Hz)
1	0.24	65.2	8.5	25.3	38.3	70.9
2	0.43	61.8	12.6	20	27.5	34.3
3	0.063*	18.2	26.4	40.5	44.7	55
4	0.068*	24	26.1	40.9	44.9	55.1

*deformation of the inter-cavity supports.

*0.026mm Cavity deflection



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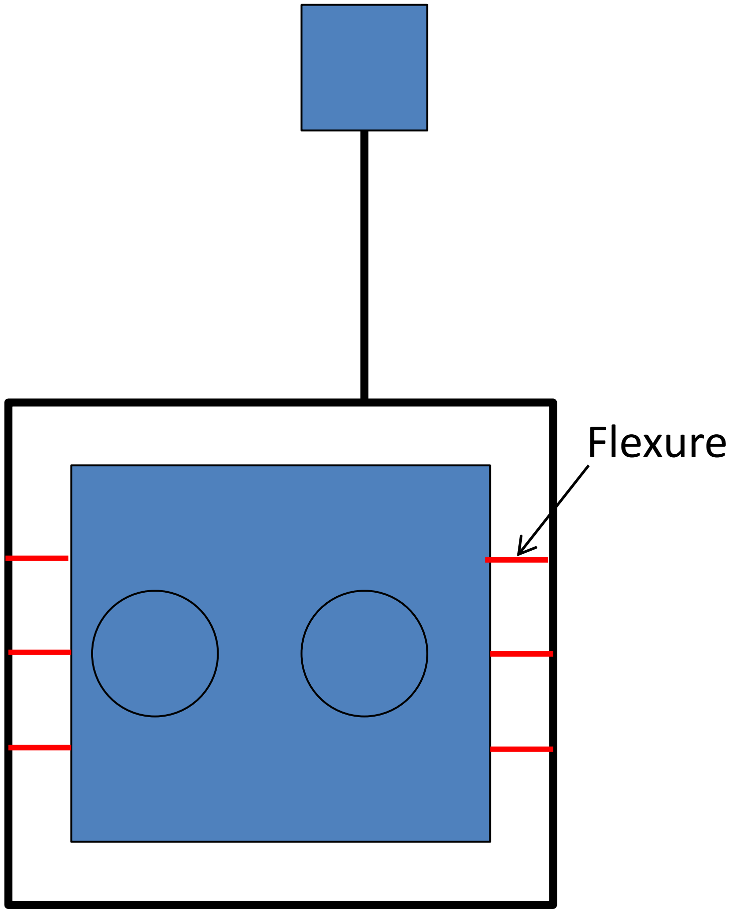
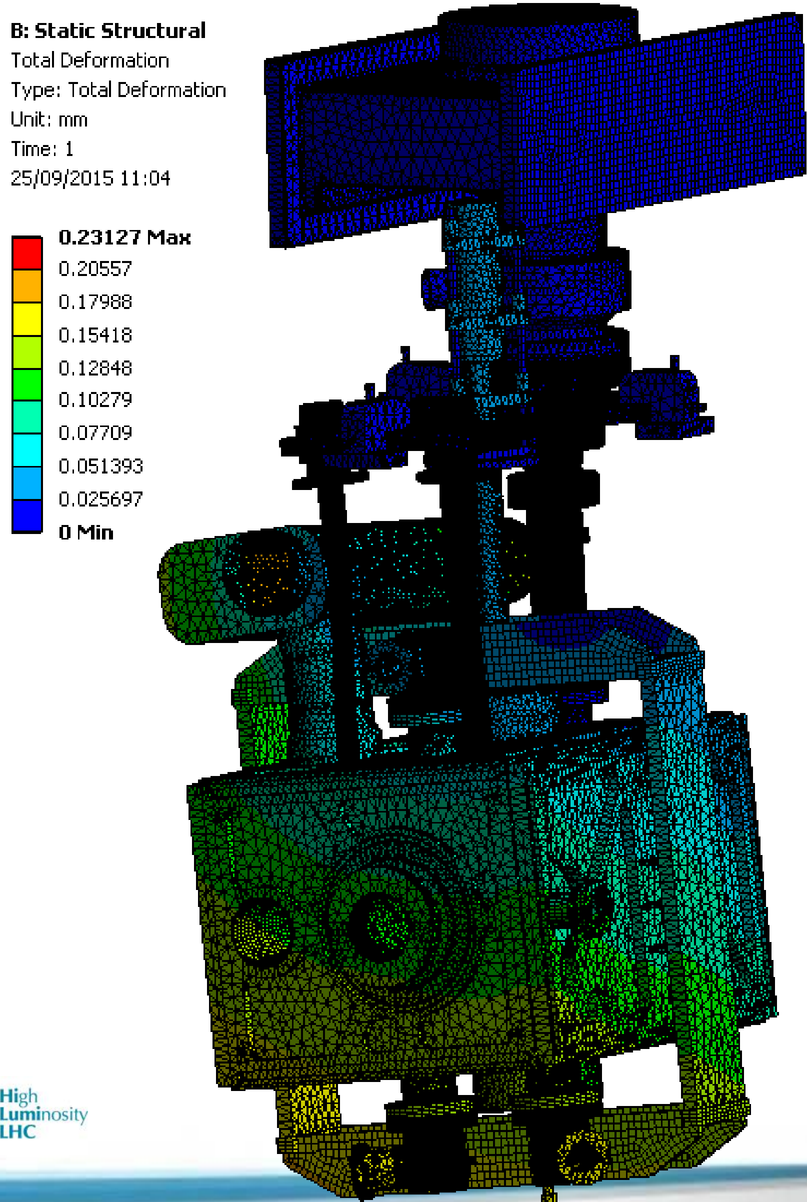
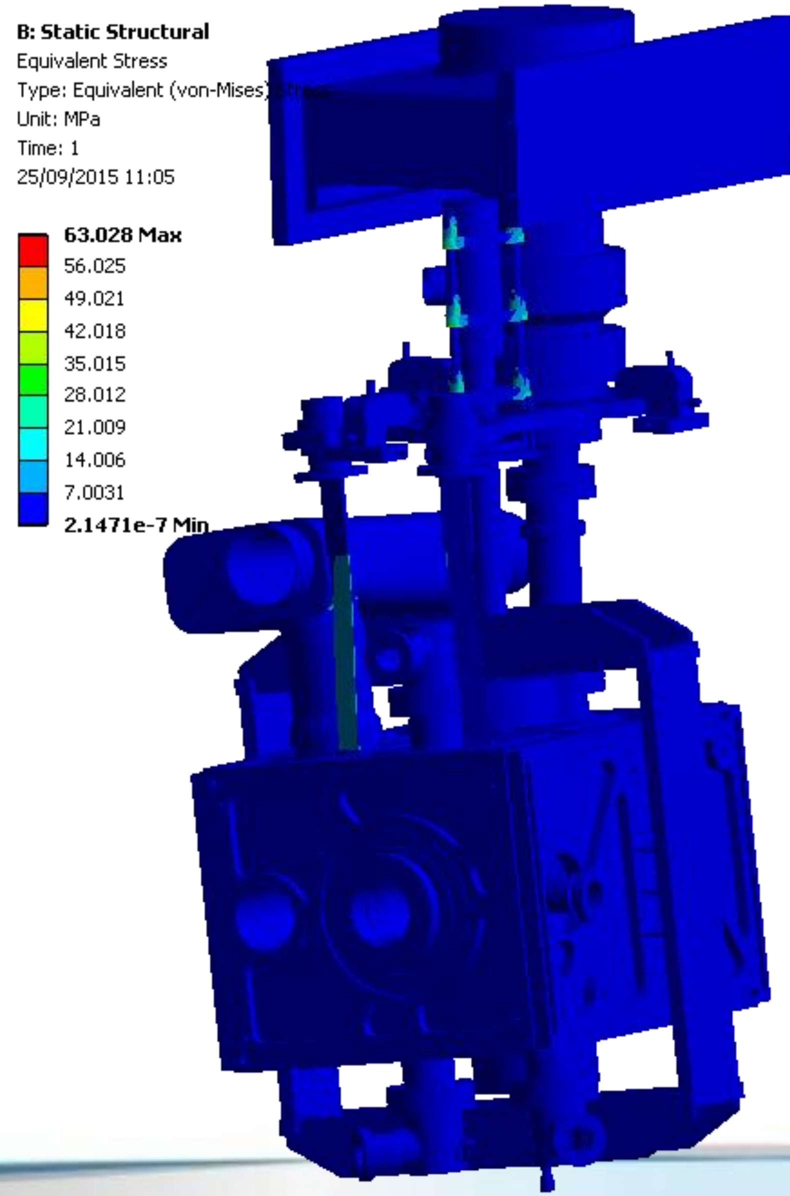
Suggestion



- Can we add flexures between the helium vessel and tuning frame?
- These could be bolted into the vessel, then the frame would bolt to this after tuner installation.
- This would help reduce the stresses under transportation and should also stiffen the tuner in several directions to raise the fundamental modes.
- This flexure will take some/all of the mass of the tuner off the cavity and onto the helium vessel.

Static result 4mm thick Grade 2 Ti flexures





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Modal result 4mm thick Grade 2 Ti flexures

Mode	Frequency (Hz)	Description
1	11.2	Cavity swinging laterally
2	22.2	Cavity swing and rotation
3	25.0	Cavity rotation
4	41.3	Top of tuner rocking forward and back
5	42.0	Top of tuner rocking side to side
6	49.1	Helium reservoir swinging vertically
7	56.3	HOM swinging side to side*
8	58.8	HOM swinging back and forth*

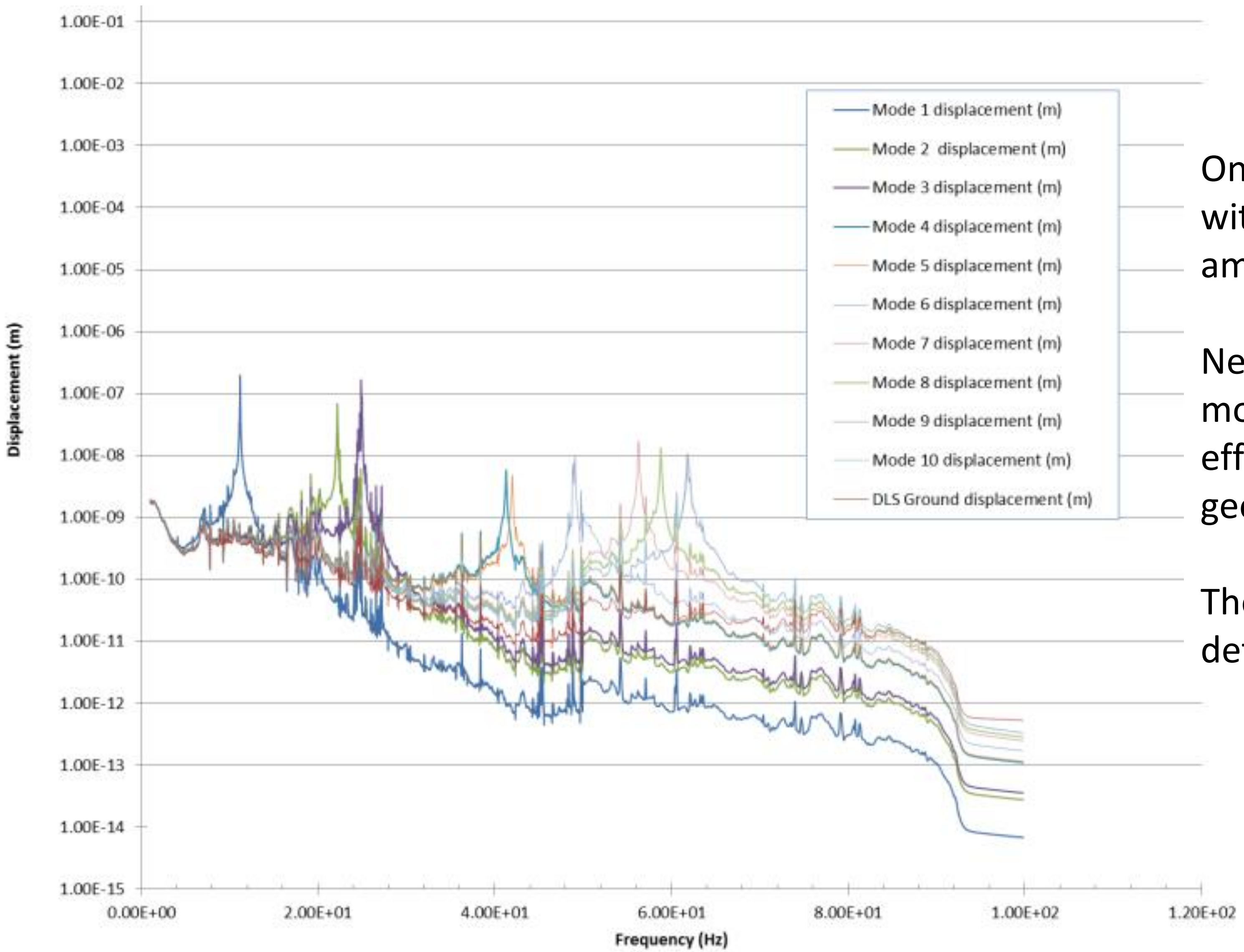
	Added flexures	Additional Bearing	Original model
Mode 1 =	11.2	9.5	8.9
Mode 2 =	22.2	10.3	9.9
Mode 3 =	25.0	12.0	11.7
Mode 4 =	41.3	13.1	12.7
Mode 5 =	42.0	23.9	23.8
Mode 6 =	49.1	26.8	26.7
Mode 7 =	56.3	36.5	38.0
Mode 8 =	58.8	37.6	38.7

* = HOM modes have changed. I think one of the contact faces has failed. I will investigate in next model.

Addition of flexures shows a significant improvement in the modal performance.
Lower modes do not directly effect the cavity.

High Luminosity LHC

Transmission 4mm thick Grade 2 Ti flexures



Only 2 modes now with 200nm amplitude.

Neither of these modes significantly effect the cavity geometry.

Therefore RF detuning <<1Hz.