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INVESTIGATION, ANALYSIS AND MITIGATION OF COMBUSTION DRIVEN VIBRATION IN A SULFUR RECOVERY BURNER ASSEMBLY

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Abstract

During initial operation, one of three identical Claus furnace burners of a large Sulfur Recovery Complex was observed to develop a vibration at certain operational conditions that affected the reliability of some of the instruments attached to the burner front. The resonance was not sufficient to lead to mechanical damage of the burner or the instruments but led to spurious operational trips and corresponding plant shutdowns. The observed vibration, first considered to be a result of mechanical resonance within the burner assembly, was found to be the direct result of acoustic excitation of a burner pressure head by the natural acoustical frequencies present in the attached furnace during the combustion process. The investigation included gathering field operational conditions, field vibration measurements, and analytical computations using finite element methods. This paper reports the investigation process, results obtained, and the modifications that were determined necessary to sufficiently reduce the vibration of the instruments.

INTRODUCTION

Burner technology is a complex science involving many factors including flame stability, mixing and operating temperatures. The sulfur recovery Claus furnace burner combusts a feed gas composed of sulfur compounds and inerts in a sub stoichiometric firing condition. In this application the burner is attached to a large horizontal vessel that provides retention time for the Claus reactions to occur. The burner design and combustion process often interact to produce energy pulsations at the flame envelope that can resonate to audible levels within the assembly. One of three identical Claus furnace burners of a large sulfur recovery complex was observed to develop a vibration at certain operational conditions that affected the reliability of some of the flame scanner instrumentation. The instrumentation reliability was addressed with special supports and arrangements. As this sulfur complex was to be expanded by adding three additional identical units, it was desirable to avoid similar vibration conditions in the new units. Field observations of the original burners and the vibration of one burner did not establish the reason for the excitation. Field measurements of the frequency and amplitude of this burner were made and evaluated. It was concluded that resonance within the furnace chamber was the driving force for the burner excitation. As the original three burners were successfully performing from a process perspective, it was decided to not alter the burner in a manner that would affect the process performance. The investigation approach used was Finite Element Analysis (FEA) combined with field measurements to arrive at an altered structural design that would mitigate the vibration occurring in one of the three burners before the new burners were fabricated.

Nomenclature

Hz Vibration frequency - cycles per second

Modeling

The field observation of the vibration determined the principle frequency was in the mid 50 Hz range. Closed form calculations had indicated there would be an acoustic resonance in the system in this same range. The original FEA work was expected to focus on moving the burner structure mechanical resonances out of the 50 Hz range to reduce excitation from the acoustic resonance of the process gas. The burner assembly, including associated large air and process gas piping, was modeled using Algor FE modeling software [1] as shown in Figure 1. This model consisted of the complete burner assembly, the associated piping up to the first constrained support, and the vessel head to which the burner was attached by welding. The model consisted of approximaterly 80,000 nodes defining approximately 63,600 brick, plate and beam elements. Several additional models, representing the assembly with various modifications, were also constructed and analyzed.

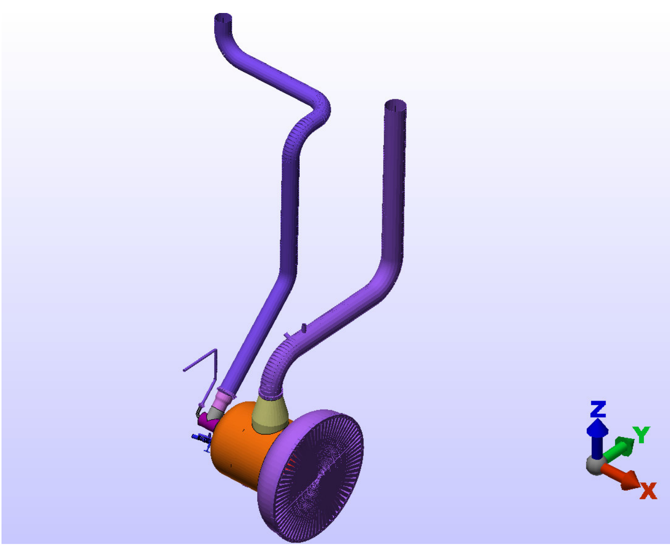

Figure 1 – Image of model of burner and piping

Analysis

A modal analysis was conducted for each of the models (as noted above). The purpose of this analysis was to identify the modal frequencies (eigenvalues) and mode shapes (eigenvectors) associated with the burner system. All modes with a frequency of less than 100 Hz were computed for each of the models.

Following the modal analysis of each model, a frequency response analysis was conducted. In this analysis, the model was subject to a sinusoidal base acceleration of 1 G in the global X (vessel axial) direction that was swept over the range of 1-100 HZ. This excitation was a *qualitative* representation of the presumed internal acoustic excitation. The response of the system to this excitation was then computed. A viscous damping value of 2% critical was used for the analysis. Since all results were considered qualitative for this analysis, knowing the actual damping value was not necessary.

Four reporting locations were used to compare the results for the various models. The locations are illustrated schematically on Figure 2. The points labeled FS-top and FS-bottom were located at the end of the flame scanners. The points labeled Shell-top and Shell-bottom were located on the burner shell adjacent to the flame scanner nozzle connection points

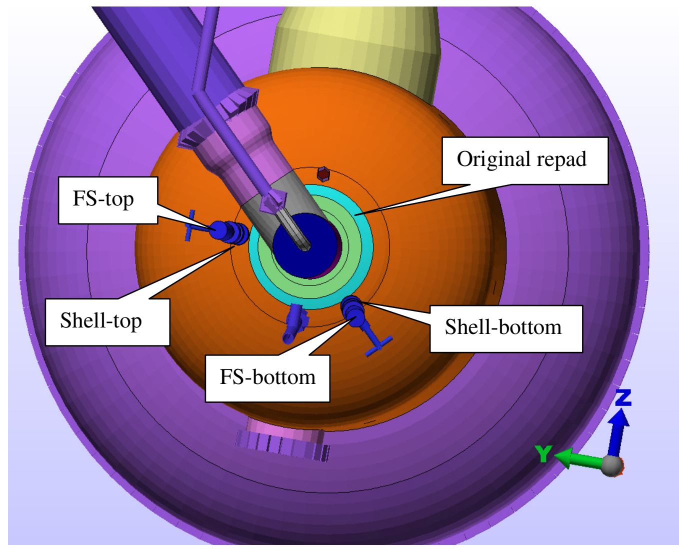

Figure 2 – Reporting locations

INITIAL RESULTS

Figure 3 (in Appendix A) illustrates the magnitude (the SRS sum of the vector responses) of the responses at the four locations on the model representing the as-designed condition of the burner assembly. A very prominent peak in the response of both the flame scanners and the shell is observed in the region of 38 Hz. Above 50 Hz, the response is seen to drop rather significantly.

There is a sliding expansion joint between the burner volute and the head of the vessel. Since it is possible that this joint remains effectively rigid for the small motion associated with vibration, a model was constructed with this joint modeled as rigid or “locked up”. The results of this analysis are illustrated in Figure 4. (Appendix A) The magnitude of the vibrations on the flame scanners is changed, but the overall pattern and frequencies are not significantly affected.

Reports from the field indicated that the lower flame scanner had been isolated from the burner shell by a flexible coupling. In order to check the effect of such a coupling, a model was constructed and analyzed without the lower flame scanner. The results derived from this analysis are illustrated in Figure 5. (Appendix A) This is seen to have little effect on the vibration patterns.

Field TEST INPUT

Since the model exhibited little or no response in the mid-50 Hz frequency range, a request was made for new vibration data on the operating unit. Figure 6 (in Appendix A) illustrates the vibration levels monitored at 8 locations, as shown on Figure 7 (in Appendix A), under various load conditions. These velocity magnitudes have been color coded to indicate the associated frequency. The black numbers represent vibrations with a dominant frequency of 57.5 Hz. The red numbers represent vibrations at higher frequencies and the green numbersare at lower frequencies. As may be seen, there is a significant scatter in the data. This scatter is further illustrated in Figure 8 (in Appendix A), where all measurements have been plotted as a function of level and frequency.

We found no clear pattern in these data that would indicate a mechanical resonance in the system. In fact, the data suggest that the vibrations are associated with the internal acoustic load acting on the system and that the response is a direct function of this load. Changing or eliminating the acoustical resonance would require modifications to the internal burner system. The process is currently functioning as designed. Since there is a risk of unintended consequences resulting from such a change, it would seem prudent to not take such an approach. Reducing the response to such a non-resonant loading condition requires stiffening the system.

FOLLOWUP ANALYSIS

In order to stiffen the system and reduce the vibration levels, several modifications were examined.

First, it was noted that the repad on the burner head in the original design ended just inside the nozzle connections for the flame scanners. If the head deflects due to the acoustic loading, this would cause localized bending in the head adjacent to the repad. This deflection would tend to cause the flame scanners to move in a manner similar to holding a stick at one end and waving it. To remedy this potential difficulty, the repad diameter was increased to encompass the flame scanner nozzles, as illustrated in Figure 9 (in Appendix A). At the same time, it was decided to double the thickness of the repad from 0.75” to 1.5”.

Figure 10 (in Appendix A) illustrates the response of the system with the larger and thicker repad. The maximum response has dropped from something over 0.7 to approximately 0.25. [Note: Since the 1G excitation is only a qualitative representation of the acoustic force, all reported response values are relative to one another and do not represent the actual field vibration levels.] Thus, increasing the pad diameter and thickness results in reduction of the vibration levels by nearly two-thirds.

Increasing the thickness of the head from 0.625” to 1.25” resulted in an even greater reduction in the vibration levels. The response of the system to both increased head and repad thickness is illustrated in Figure 11 (in Appendix A). The reduction due to increasing the thickness of both the head and the repad is nearly 80%.

In order to check the effect of a “locked” expansion joint, a final model was constructed and analyzed. The results for this model are illustrated in Figure 12 (in Appendix A). While there is some slight increase in the indicated vibration levels, the increase is not significant.

In addition to the burner head and pad, the effect of increasing the pad/head thickness on the furnace vessel was examined. No significant improvement was observed due to changes in the larger vessel.

Conclusions

The vibration of the flame scanners on the front of the burner assembly were first thought to be due to excitation as the result of a mechanical resonance in the burner assembly. Initial analysis did not confirm mechanical resonance at the frequency of the observed acoustic resonance [2]. Additional field measurements confirmed significant displacements in the attached instruments and much smaller displacements on the burner assembly. Additional analysis lead to the conclusion the excitation was the direct result of an internal acoustic resonance that excited the burner elliptical head. In order to change the response levels, stiffening of the burner assembly was required.

In order to reduce the system response and lower the vibration levels, the thickness of the repad was doubled and the pad diameter was increased to encompass the flame scanner nozzles. Thickness of the burner head was also doubled. These increased thicknesses stiffened the burner structure sufficiently to reduce the relative displacement of approximately 0.7 from the initial analysis of the existing structure to less than 0.2 for the stiffened structure.

REFERENCES

1. Algor Version 19.2 Finite Element Software, Algor, Inc., Pittsburgh, PA.
2. Beranek, L.L, ed, Noise and Vibration Control, 1971, McGraw-Hill Book Company, NY, NY.

Appendix A

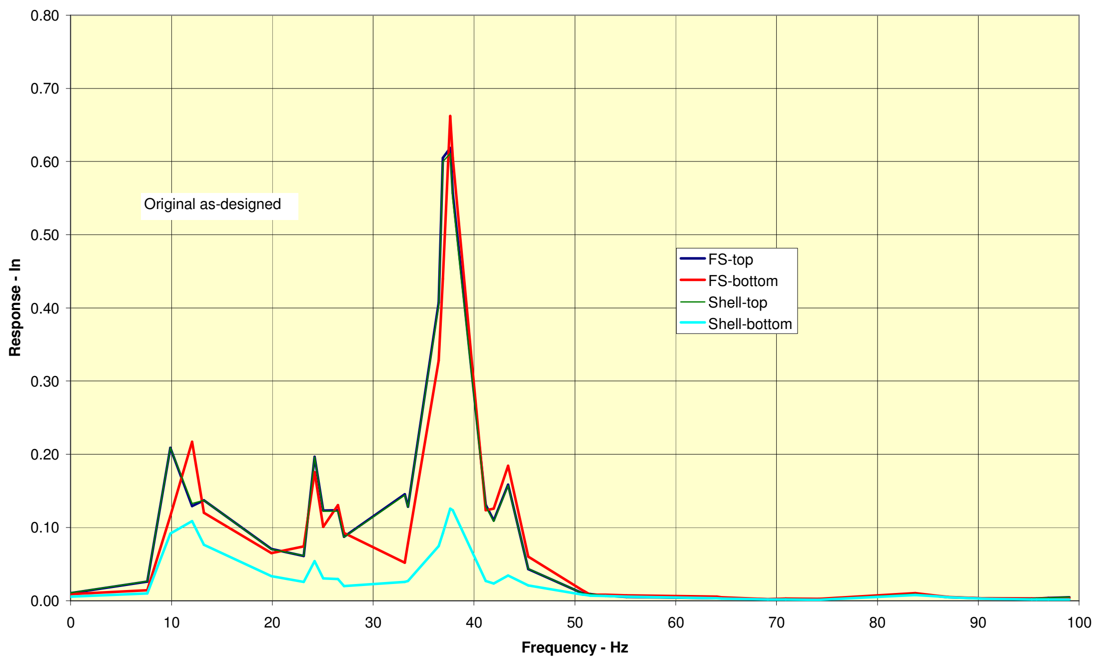

Figure 3 – Displacement for original as-designed system

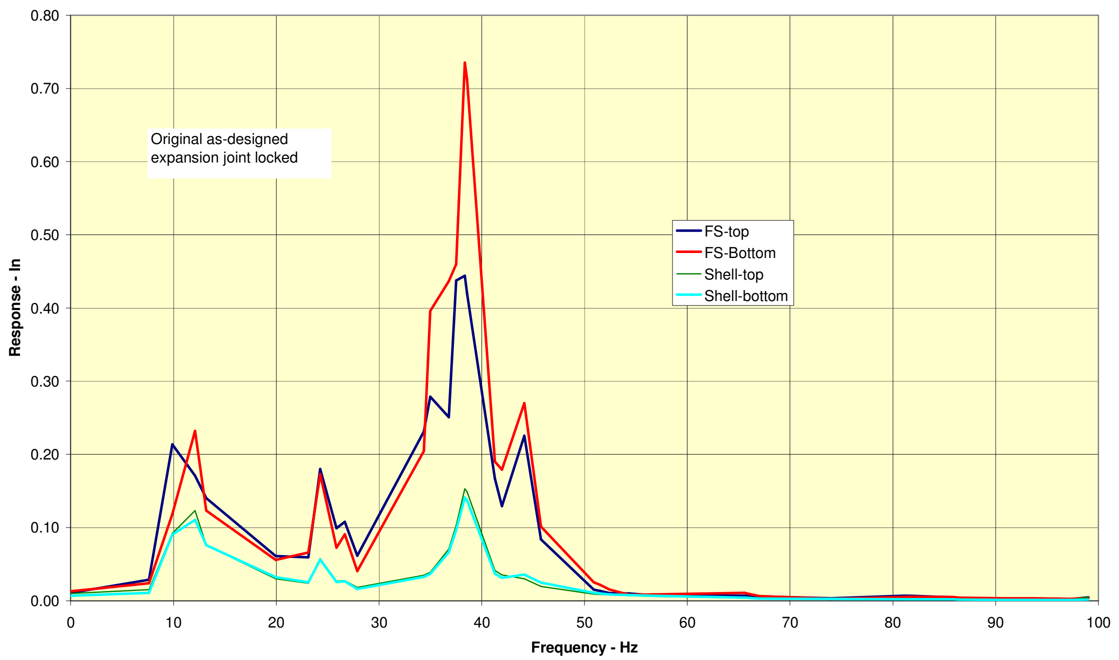

Figure 4 – Displacement for original as-designed system – expansion joint locked

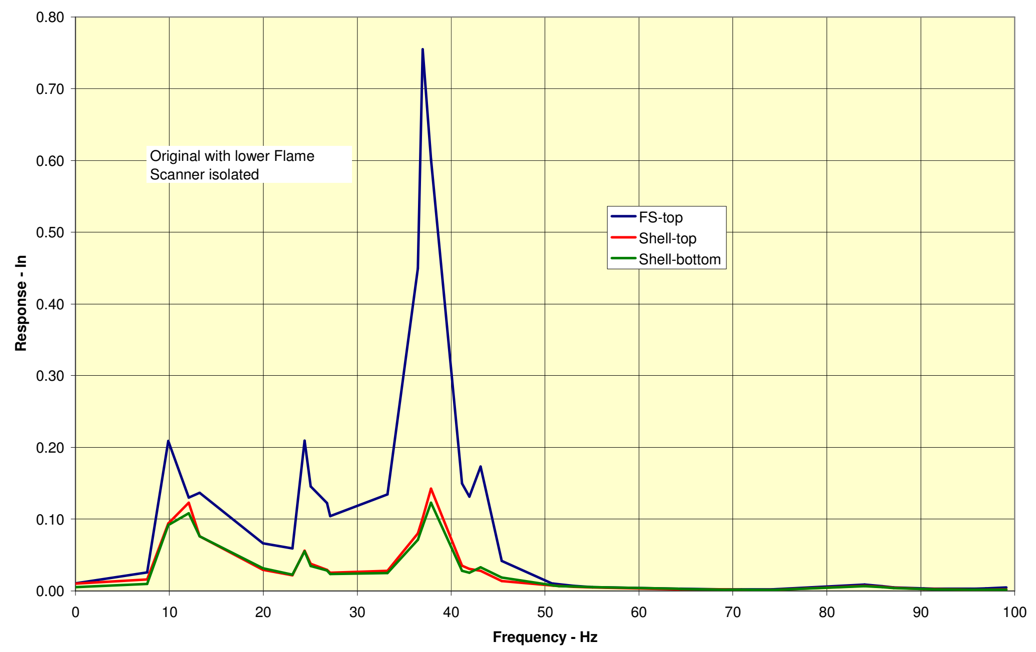

Figure 5 – Displacement for original as-designed system with bottom FS removed

Load 15K				Load 14K			
Measure ment location	Velocity (mm/s)			Measure ment location	Velocity (mm/s)		
	Horizontal	Vertical	Axial		Horizontal	Vertical	Axial
1	**15.64**	**12.7**	5.577	1	**14.66**	**11.78**	**5.401**
2	5.542	0.469	**1**	2	5.472	0.269	**0.982**
3	**14.66**	**3.402**	**8.277**	3	**16.7**	**3.086**	**8.067**
4		**1.35**	**6.173**	4		1.42	**6.89**
5	2.788		**7.962**	5	2.069		**8.348**
6		1.517	**7.892**	6		1.456	**8.102**
7	**3.7**	1.806	**4.367**	7	**6.316**	**2.297**	**5.577**
8	2.35	2.069		8	2.365	1.491	

Load 18K				Load 17K				Load 16K			
Measure ment location	Velocity (mm/s)			Measure ment location	Velocity (mm/s)			Measure ment location	Velocity (mm/s)		
	Horizontal	Vertical	Axial		Horizontal	Vertical	Axial		Horizontal	Vertical	Axial
1	**21.18**	**18.66**	**6.13**	1	**19.5**	**12.21**	**5.998**	1	**14.59**	**10.03**	**5.682**
2	6.419	0.728	1.14	2	5.927	**0.3**	**0.636**	2	6.348	1.324	**0.8**
3	**21.75**	5.682	**11.78**	3	**17.82**	**3.437**	**10.03**	3	**11.5**	**1.254**	7.506
4		2.026	3.437	4		1.447	**7.927**	4		1.534	**6.594**
5	3.385		8.277	5	2.297		**7.085**	5	2.525		**7.716**
6		2.052	6.138	6		1.955	**8.418**	6		1.289	**8.628**
7	**4.805**	2.157	3.946	7	**4.875**	**2.473**	**7.05**	7	**3.928**	2.131	4.56

Figure 6 – Vibration data from reliance

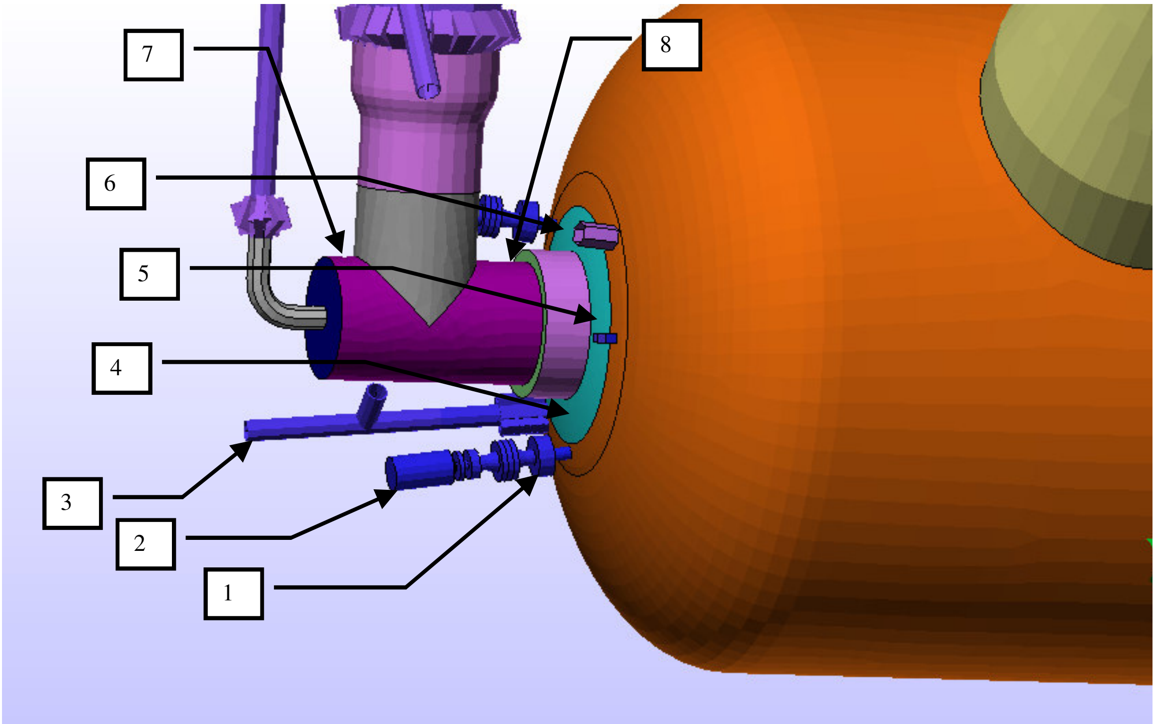

Figure 7 – Reliance measurements point locations

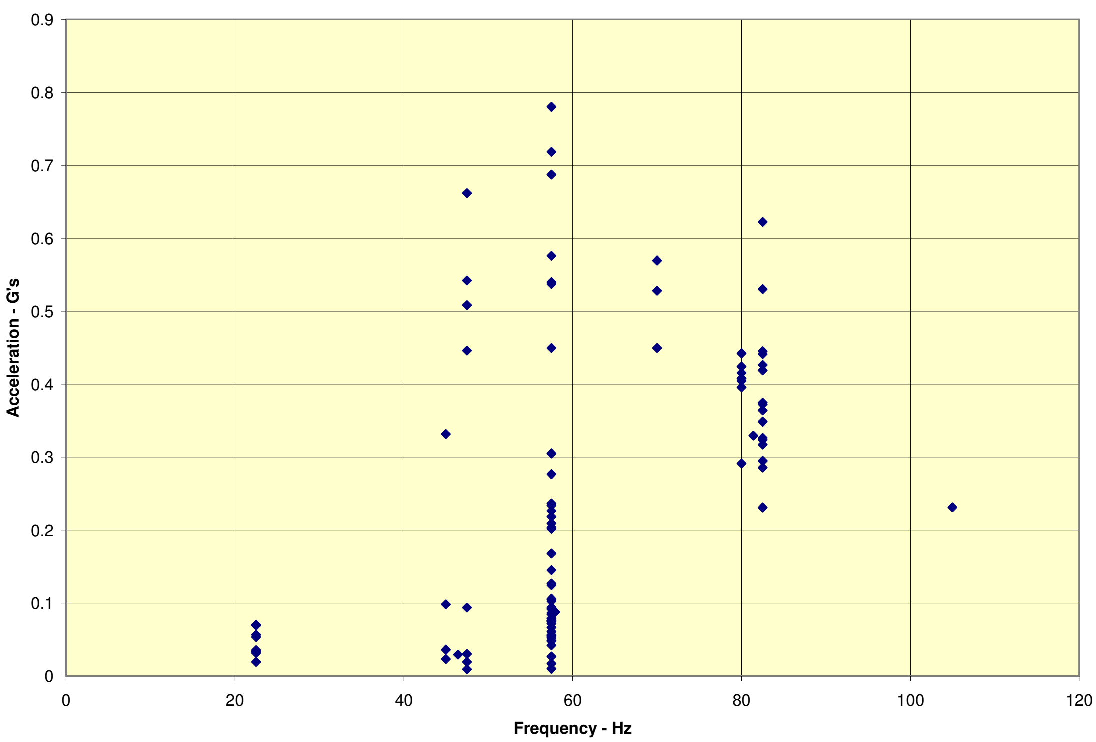

Figure 8 – Reliance measurements in acceleration units

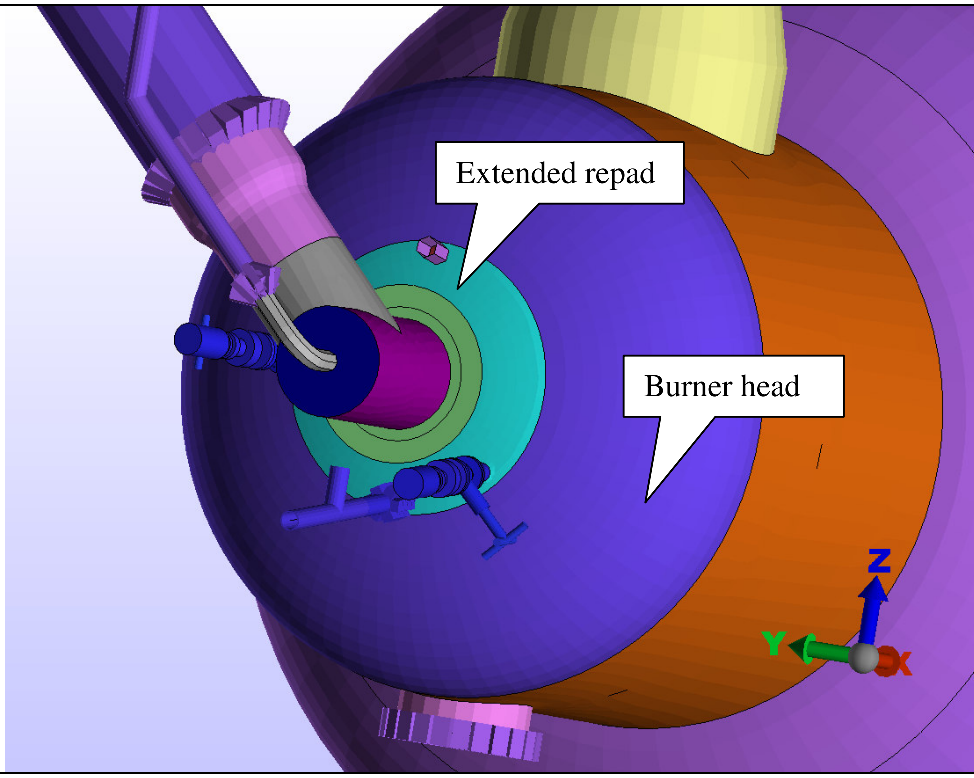

Figure 9 – Extended repad and burner head locations

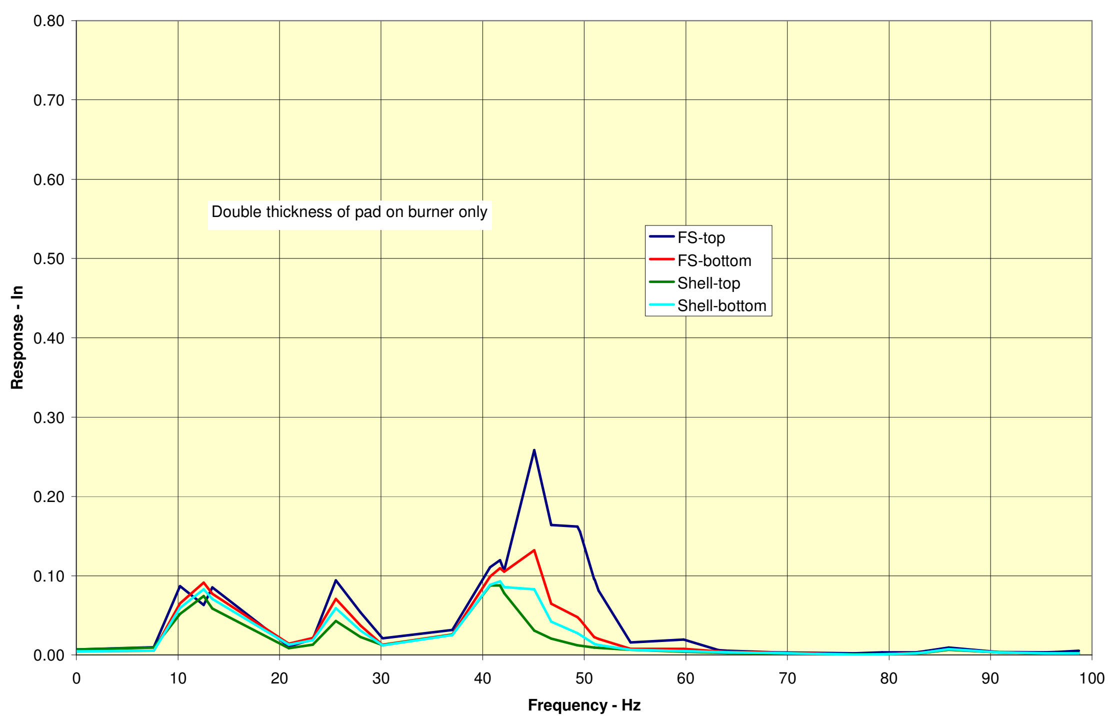

Figure 10 – Displacement with extended and thickened pad

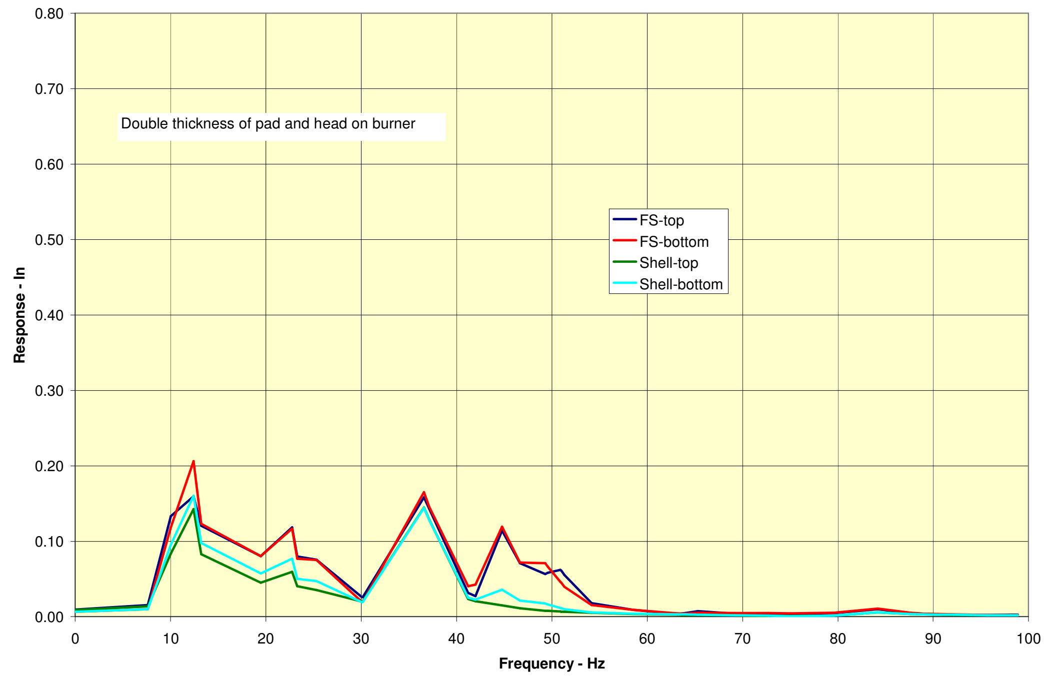

Figure 11 – Displacement with extended and thickened pad and thickened head

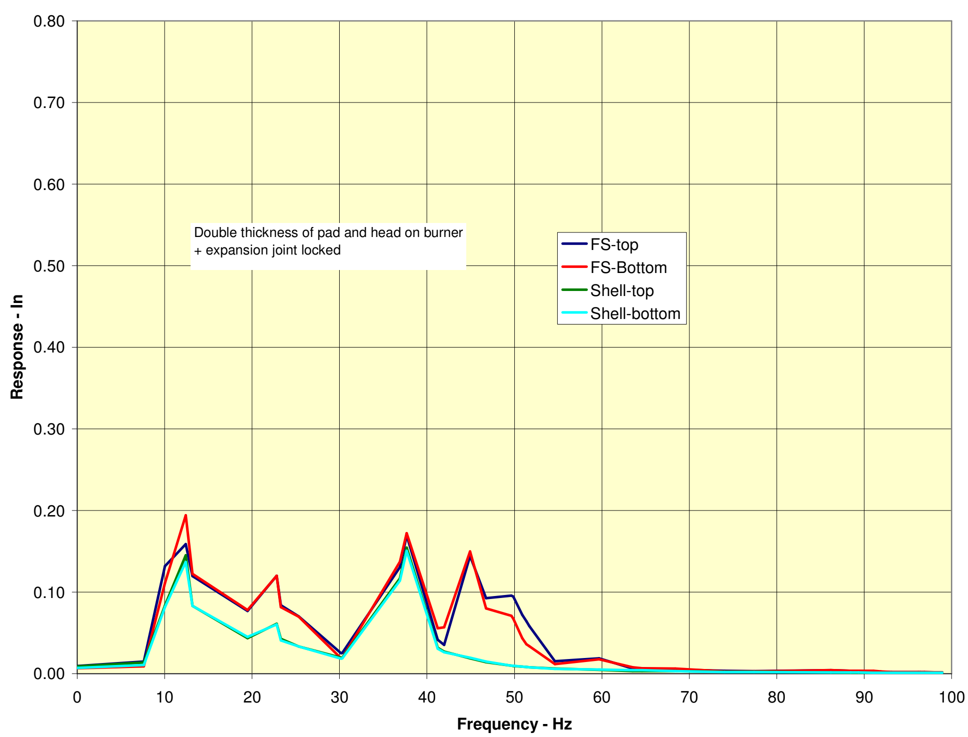

Figure 12 – Displacement with extended and thickened pad and thickened head – joint locked