THE STRUCTURAL DYNAMICS ANALYSIS OF THE MAIN INJECTOR LOX INLET TEE AND ITS REDESIGN

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ABSTRACT

A ground test of a specially instrumented Space Shuttle Main Engine (SSME) revealed excessively high strains in the high pressure Liquid Oxygen (LOX) Inlet line which carries LOX to the Main Injector. The inlet tee (the instrumented part) acts as a manifold which utilizes two splitter vanes to direct the flow of LOX into the main injector. When the high strains were discovered, it was thought that these splitter vanes, coupled with high energy flow was the source. This resulted not only in high strains in the LOX inlet tee, but excessive vibration at a frequency of 4000 Hz located at the gimbal bearing (attach point to orbiter), and at the main injector. This was later to be known as "The 4kHz Phenomenon".

A team was assembled to determine the extent, cause, and solution of the phenomenon. Parallel efforts were conducted through laboratory testing to simulate the phenomenon and theoretical analysis to predict response. This paper will deal with the dynamic analysis which was used to predict the response due to the 4kHz phenomenon and the analysis used to aid in the design modification required to the hardware to reduce or eliminate the phenomenon.

HISTORY

The 4kHz phenomenon was first observed during a ground test of an SSME which contained strain gages at the underside of the LOX inlet tee (see Figure 1). During the mainstage portion of the test, high strains were measured at the inlet tee at a frequency of about 4000 Hz. At this same time during the test, high accelerations were measured at the gimbal bearing. It was hypothesized that the splitter vanes within the inlet tee were the cause (refer to Figure 1). This was later confirmed when a downward shift in the 4kHz frequency was observed in a later test. More ground tests followed. Several tests even exceeded 104% rated power level with no change in the shifted frequency (now, at 3700 Hz). During this time, a detailed finite element model (FEM) of the splitter vanes was created to predict the location and size of a possible crack in the vanes (refer to Figure 2). It was thought that a crack in the vanes caused the downward shift in the 4kHz frequency. The model showed that a single crack of about .8 inches long, through the vane at its lower leading edge would produce a frequency shift seen on engine test results. When the engine was disassembled, and the elbow cut to expose the vanes, cracks were indeed found on one vane. Figure 3 illustrates the crack locations and sizes. The model predicted and the inspection confirmed that the lower leading edge of the vane was particularly sensitive to cracking. This was judged by the relative size of the cracks seen in Figure 3.

A second engine was also found to have exhibited the 4kHz phenomenon with a sudden downward shift in frequency. This engine was unique in that there was a double shift in the frequency. Inspections revealed that both vanes had cracks present in them with the largest located at the lower leading edges (refer to Figure 4).

THE PHENOMENON ITSELF

It was clear that a phenomenon existed at 4kHz and it originated near or at the splitter vanes of the LOX inlet tee. The question was; What was the cause? Hundreds of tests were reviewed and checked against any hardware changes or changes in test stand configurations. The study revealed that the phenomenon was independent of hardware changes and test stand configurations. The study did reveal, however, that the phenomenon existed on only 17% of the engines built and it existed since nearly the beginning of the program. The reason it was not detected early on was because the response was interpreted as the High Pressure LOX Pump (HPOTP) eight times synchronous frequency which occurs very near 4000 Hz. Three mechanisms were considered for investigation; 1) flow induced vibrations of the vanes coupled with a strong 4kHz frequency of the thrust cone which amplified the energy from the vanes to the gimbal bearing, 2) combustion driven oscillations, and 3) acoustics. The ladder two mechanisms were ruled out early due to lack of support by future measurements taken on subsequent tests. This left flow induced vibration as the main candidate. Parallel efforts were then conducted to, 1) understand the cause through extensive flow and vibration testing as well as analysis and 2) develop a design change, retrofitable to all affected and future engines that would reduce or eliminate the phenomenon.

Extensive flow tests were conducted on several different test articles using different flow media. Plastic models were used as well as actual hardware in order to obtain a proper database in which to get results. Measurements taken in the wake of the vanes identified a periodic response similar to vortex shedding which occurred near 4kHz.

DYNAMIC ANALYSIS

While extensive testing to determine the mechanism was in progress, structural analyses were being performed to determine the modes, frequencies, and response of the LOX system to various forcing functions and vibration environments. Two analyses were conducted in parallel using finite element models (FEM). A course system model ("Global Model") was used which included the high pressure LOX duct, inlet tee, and thrust cone with gimbal bearing (see Figure 5). This model was used to determine system response as well as response at the gimbal bearing due to various loading conditions. This model consisted of mainly plate (shell) elements and beam elements. The second model was a much more refined model ("Local Model"). It included the high pressure LOX duct and inlet tee only. The inlet tee was comprised of solid brick elements with membranes on the inner and out surfaces for surface response, and the duct was comprised of beam elements (see Figure 6). This model was used to obtain detail responses of the splitter vanes due to various loading conditions. The analysis described here will be concerned with the "local model," since the design modifications were studied using a portion of the local model.

MODEL DESCRIPTION

As mentioned earlier, the local model was mainly comprised of solid brick elements. The geometry of the inlet tee was taken from the computer generated drawing created by the designers. This geometry was transferred to a preprocessor known as Computer Aided Design System (CAEDS). A solid mesh was generated on the geometry in CAEDS. Due to the criticallity of the analysis, the high frequencies considered and the complexity of the part, a very detailed grid was created. This meant a large number of degrees of freedom (DOF) was required (about 30000 DOF). Figure 7 illustrates the complexity of the model. Once the FEM was created and checked out, it was transferred to STARDYNE finite element code for the dynamic analysis. A program was written to create membrane elements on the inner and outer surfaces of the brick elements in order to obtain response on the surfaces of the inlet tee and splitter vanes. A model of the high pressure LOX duct along with the Main Oxidizer Valve (MOV) was included in order to obtain the effects of duct coupling with the inlet tee. It was unknown at this time whether to include the thrust cone and gimbal bearing, therefore, case studies were run on the smaller, less refined model. Frequencies and mode shapes were calculated to 5000 Hz for the inlet tee/duct model with and without the thrust cone. Most of the modes and frequencies showed very little difference between the two boundary conditions, especially in the 4kHz range. TABLE A shows the frequency comparison in the 4kHz range of the two boundary conditions. The results of the case study indicated that the thrust was not necessary for determining the response of the splitter vanes. Rather than fix the area where the inlet tee is welded to the thrust cone, spring elements which matched the stiffness of the thrust cone were placed at the weld area. The other end of the duct, which is attached to the HPOTP was assumed to contain a fixed boundary since the stiffness of the pump far exceeds the stiffness of the duct and therefore would resemble a near rigid boundary.

While the FEM of the LOX system was being created and checked, modal testing was being conducted on various test articles using different flow media such as water, freon, LN_2, and air. These "rap" tests showed the effects due to mass loading, pressure, and temperature. The modal tests indicated the following results;

1) Two dominant vane modes were found near 4kHz, a first bending mode of the vanes and a more dominant out-of-phase bending of the leading edge and trailing edge of the vanes (refer to Figure 8).

2) The thrust cone fifth diametral mode, which occurred at 4kHz actively participated in the phenomenon by amplifying the energy from the splitter vanes. This increased the validity of a fluid-structure interaction (refer to Figure 9).

3) The effect due to pressure temperature, and mass loading lowered the splitter vane frequencies (near 4kHz) by about 18-25%. Other modes exhibited different frequency shifts.

4) The duct itself exhibited a dominant mode (3rd diametral) at 4kHz which could be coupled with the vane response.

Once the boundary conditions of the FEM were established, the natural frequencies and mode shapes were calculated. At this time, the dominant vane modes had been determined through lab testing, which offered a direction for the analysis. A careful review of all modes and frequencies was conducted and two dominant vane modes were discovered. A first bending mode of the vanes was found at 3895 Hz and a more dominant twisting mode was found at 5360 Hz. It should be noted that this is a linear analysis, therefore, the effects of pressure (stress stiffening) and mass loading were not taken into account. Since lab tests revealed that these conditions caused an 18-25% downward shift in frequency in the 4kHz modes, the actual model frequencies for the dominant vane twisting mode would be between 4020 and 4360 Hz. A comparison of the model 4kHz modes and several of the modal test results can be found on TABLE B. A point to note here is that there is a difference between the model modes and "Engine 0008" modes due to loading effects in 0008, but the ratio of the twisting mode-to-the first bending mode is the same for both cases which indicates that there is an "apples-to-apples" comparison.

After the natural frequencies and mode shapes were calculated and their comparison to lab test results were favorable, a random vibration response analysis was initiated. A multi-base response analysis was required using the influence coefficient method of excitation. The input included POWER SPECTRAL DENSITY's (PSD's) taken from actual engine ground test data. The results showed that the highest stressed areas in the vanes were located at the lower leading and lower trailing edges. This correlates with visible inspection of the cracked areas of the vanes. A program was written which converted modal stresses to modal strains. The modal strains were used in a parallel random response analysis in order to obtain strains which were compared to measured strains from the ground tests performed earlier. Results from the comparison can be seen on TABLE C. Strain gage two (2) is the gage located directly below the splitter vanes (see Figure 1).

The next step was to determine the vane response to flow loads. The results would give an indication if the vanes would respond to the various flow loads at 4kHz. The first loading condition was turbulant buffeting along the vanes. The input was formulated from a water flow test through a plastic test article. The PSD and load profile can be seen on Figure 10. The results indicated that the twisting mode of the vanes (at 5360 Hz) did respond to the turbulent buffeting, but the response was not sufficient to cause high accelerations at the gimbal bearing, nor were the stresses high enough to crack the vanes. It was concluded that turbulence alone would not be the mechanism for the 4kHz response, but might have had some contribution. This conclusion was based on the fact that the high stressed areas were located in the regions where the vanes have cracked.

The second loading condition was a sinusoidal load applied at the trailing edge of the vanes. The steady state loads were taken from a CFD solution and the time dependent forcing function was developed using a vortex shedding formula. The load was applied at the trailing edge of the splitter vanes as a sinusoidal forcing function. Since the amplitude of the forcing function was unknown, a unit load was applied. The results were very similar to the turbulence results. The areas of highest stresses in the vanes were in regions where cracking occurred. The unit load itself only produced a stress of 310 psi at the lower leading edge of the vane. The stresses required tocrack the vane were 21000 psi. This meant the input load would have to be increased by a factor of 68. This results in a 68 pound load at the trailing edge occurring at 4000 Hz. This amplitude of loading is well within reasonable limits.

While analyses were continuing in determining a possible mechanism, parallel efforts were initiated to determine a retrofittable "fix" to reduce or eliminate the phenomenon. The initial focus was to increase the frequency of the vanes (about 15%) enough to decouple the effects of the flow at 4kHz from a 4kHz mode. When the sinusoid load at the trailing edge was identified, an additional change to the vanes was necessary in order to reduce the forcing function itself. Many design changes were proposed. One of the first design changes was an 'S' shaped scallop of the leading and trailing edge (see Figure 11). The 'S' shape was derived by detailed flow calculations. Unfortunately, this design only increased the twisting mode by 1.4%. Similar modifications were considered but yielded similar results. Finally, a scallop of the full length of the vane at the leading edge was proposed. Several depths of scalloping were considered. All yielded favorable results. Scalloping the trailing edge was also considered with good results but was later replaced with the more favorable beveled trailing edge design which would theoretically reduce the flow loads derived from a sinusoidal excitation. The final design modification was scalloping the leading edge about .400 - .525 inches and beveling the trailing edge (refer to Figure 12). This design increased the frequency about 13% which would decouple the twisting mode from the flow load excitation frequency and it would also reduce the flow loads themselves due to the geometry of the trailing edge (refer to TABLE D for frequency comparison of vane modifications). Once the final design modification was agreed on, the engine that exhibited a high 4kHz response was modified and a modal test was conducted. The results can be found on Figure 14. The modal test showed an increase of 14% in the twisting mode compared to the model prediction of 13%

and the test showed a 3% decrease in the vane bending mode compared to the model prediction of a 4.5% decrease. After management approval, the modification was made on the first engine discovered to exhibit the 4kHz phenomenon. This engine was placed in a test stand and hot fired several times. Figure 15 illustrates the change in response from the baseline engine to the modified engine (both results are from the same engine). The composite levels from 20 Hz to 10000 Hz had been reduced by a factor of about 6. The actual amplitude at 4kHz had decreased by more than two orders of magnitude.

WHY ONLY CERTAIN ENGINES?

In the beginning of the program, detailed reviews of material and manufacturing processes were initiated which showed no unusual problems. Modal tests of many inlet tees revealed differences between each tee. The investigation discovered that the 4kHz tees may have had minor geometry variations significant to the 4kHz problem at the splitter vane area. Further investigating is continuing.

SUMMARY

This paper has demonstrated a perfect example of how a problem was identified through testing, and a program established to identify the extent, cause and solution to the problem. The program incorporated the close relationship between analysis, laboratory testing and design in order to understand the problem and find a solution. Extensive testing was performed to identify the cause and to verify analysis results, while analysis, through the use of the finite element method, was used to determine response and to determine effects of different design modifications which could reduce or eliminate the problem. The program proved to be a large success due to the success of the different disciplines involved.

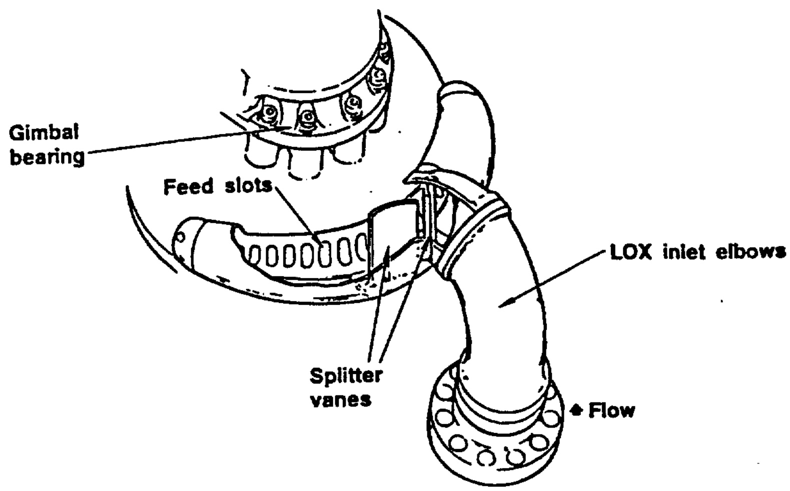

Gimbal bearing
Feed slots
LOX inlet elbows
Splitter vanes
Flow

Figure 1

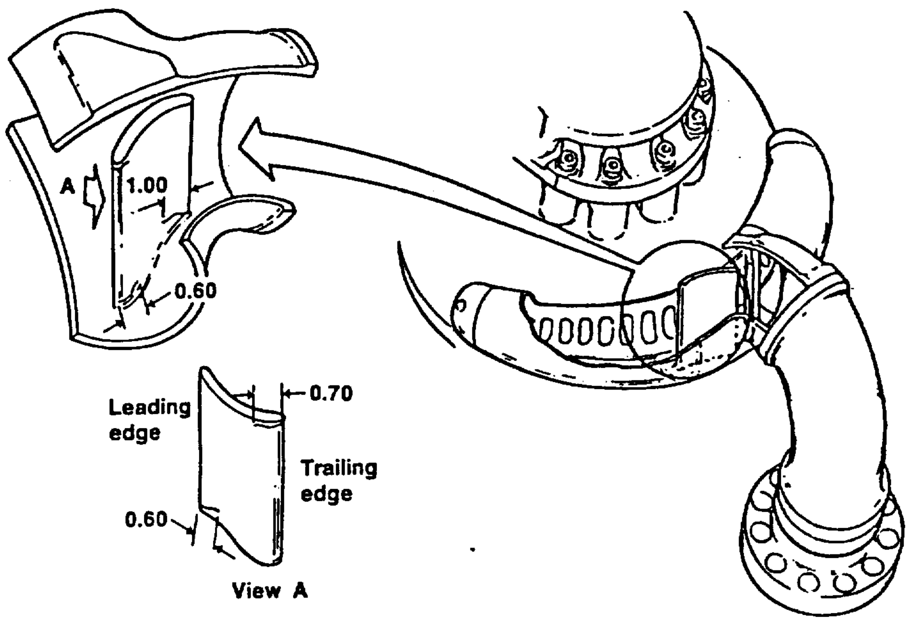


Figure 2

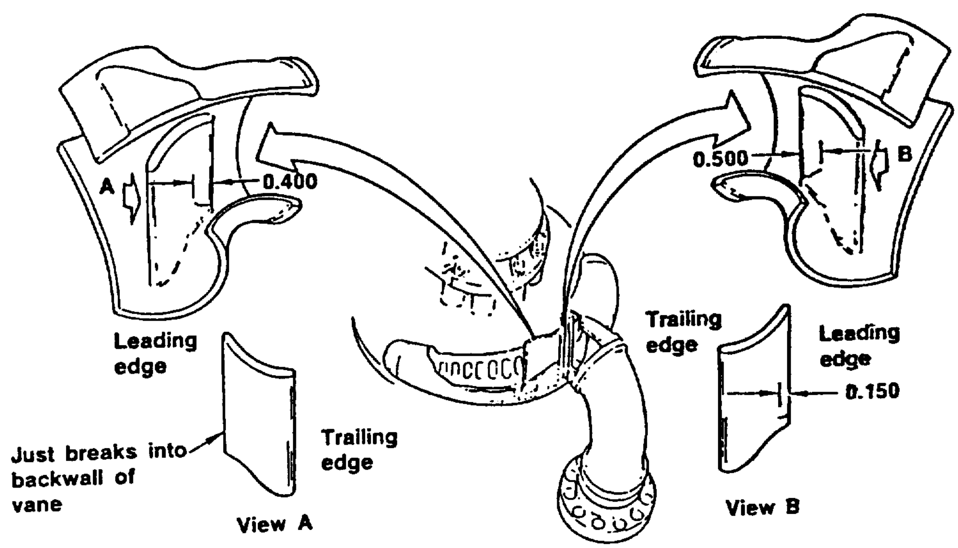
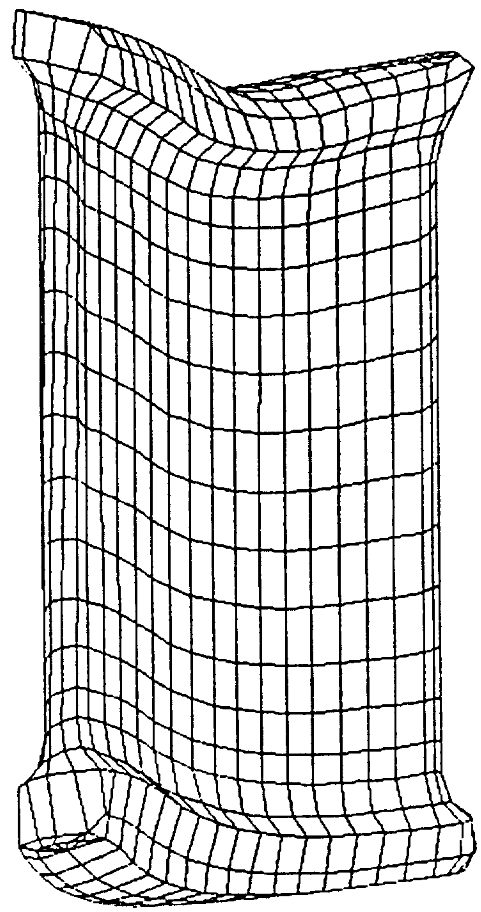

A
0.400
Leading
edge
Just breaks into
backwall of
vane
Trailing
edge
View A
0.500
B
Trailing
edge
Leading
edge
0.150
View B

Figure 3



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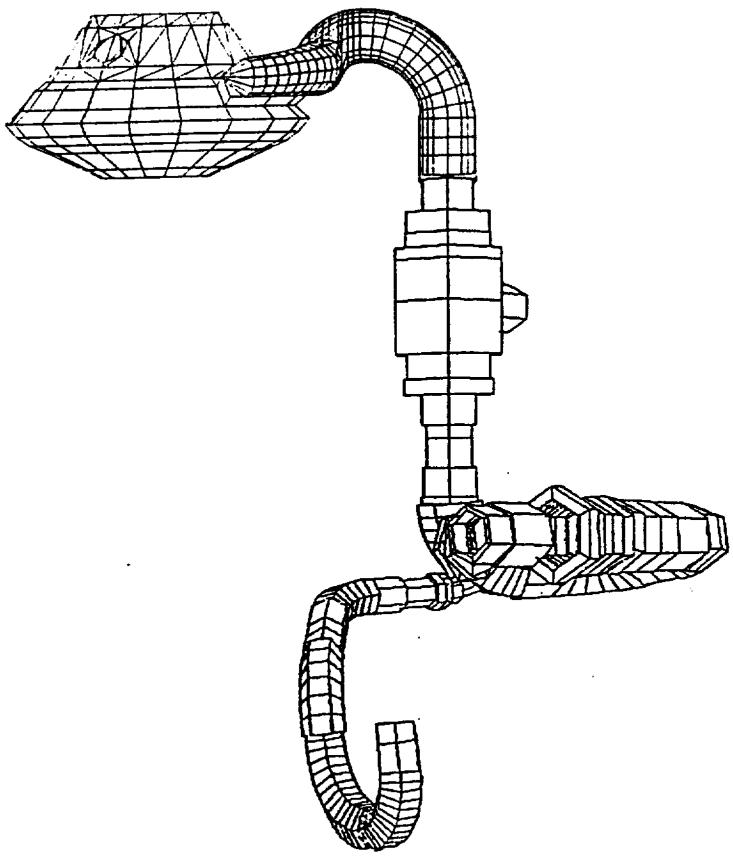

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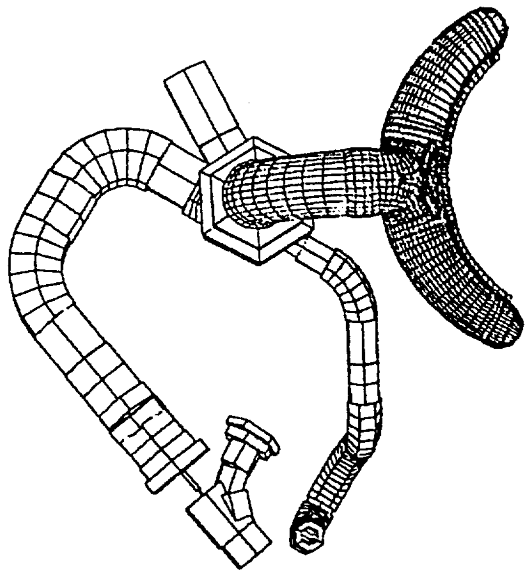

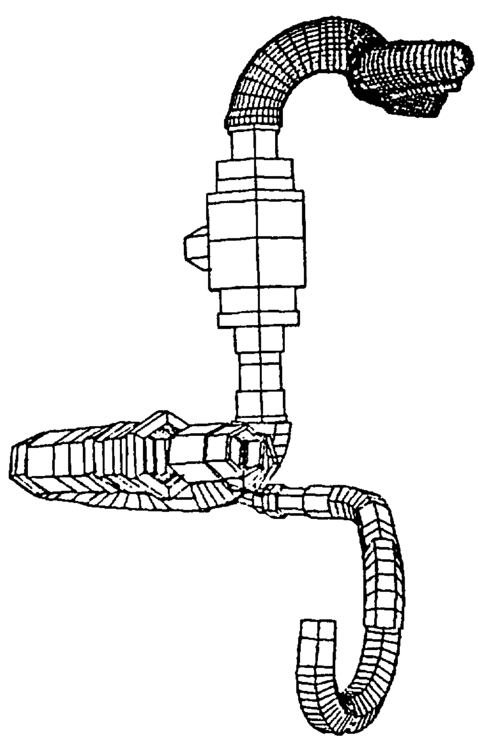

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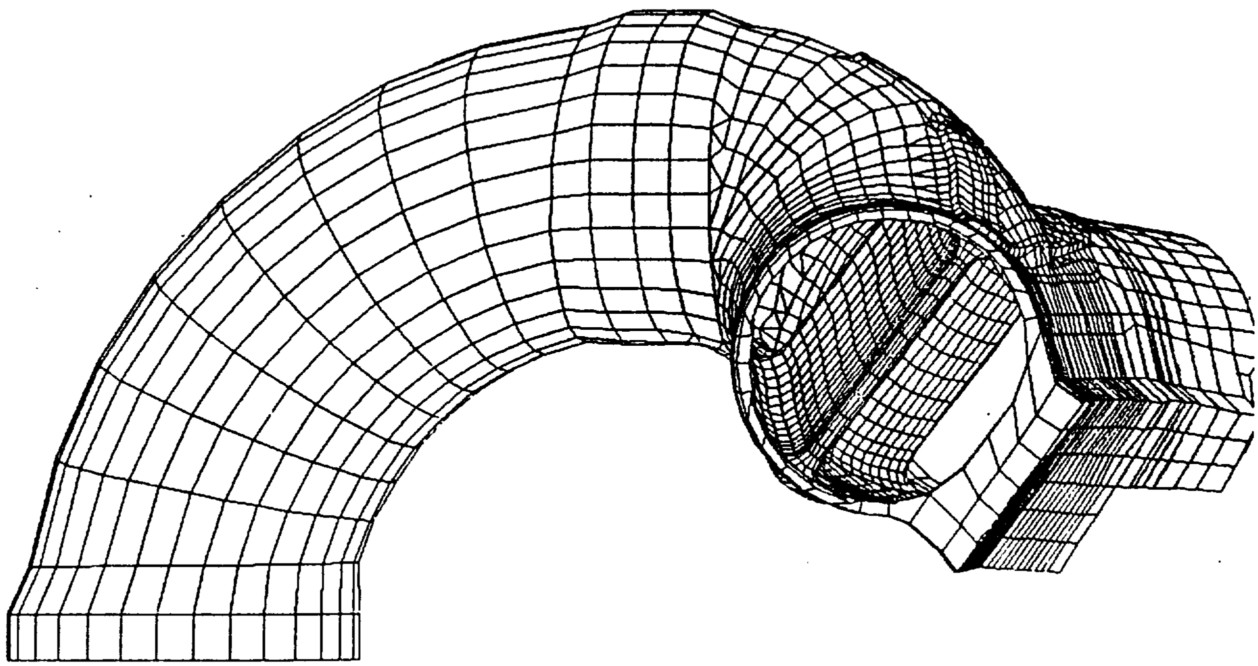

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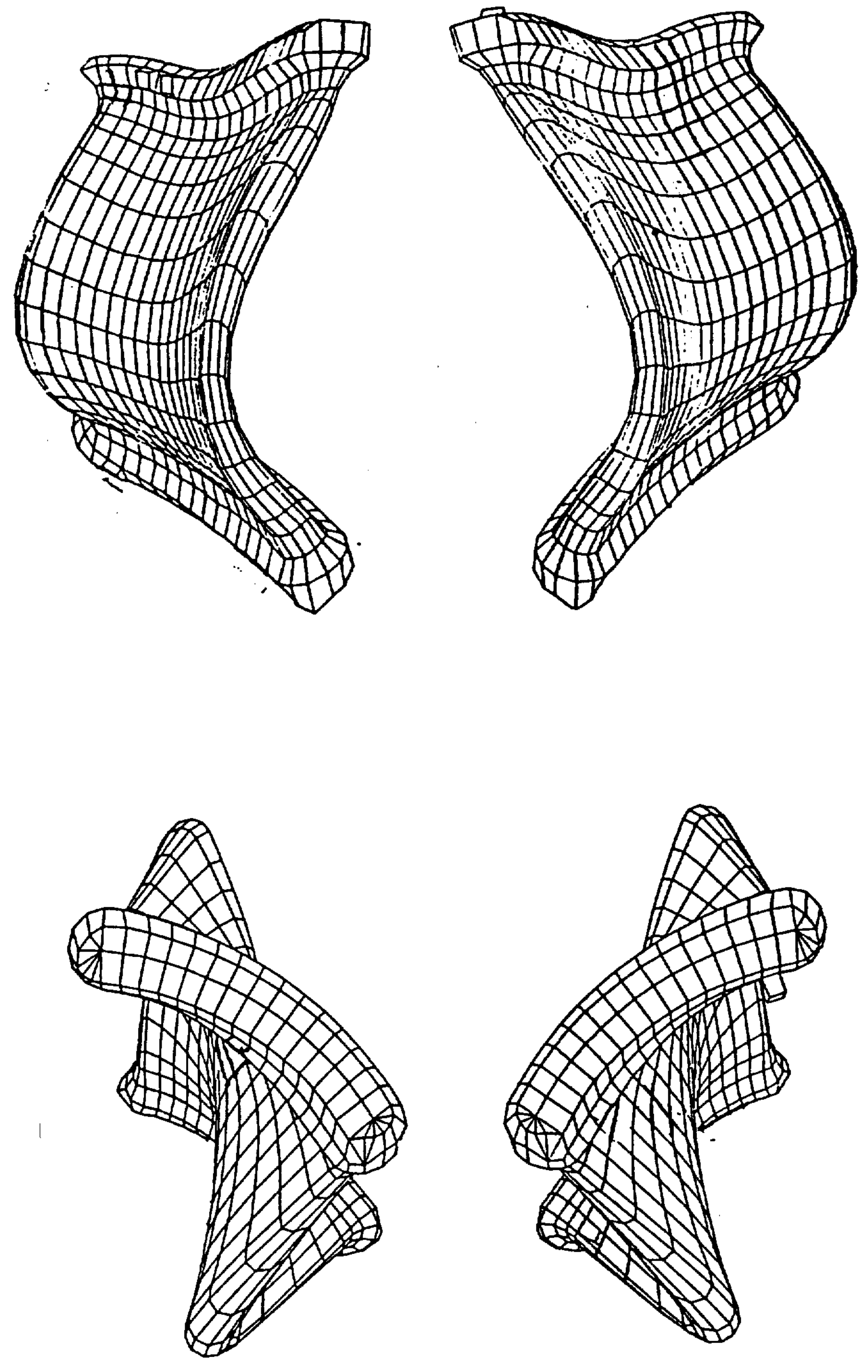

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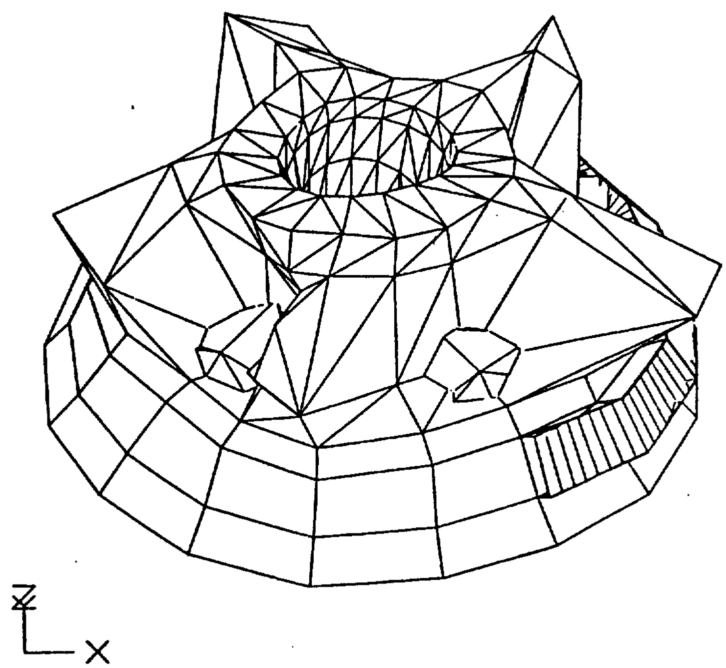

Figure 9

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Rocketdyne Division
Rockwell International

PREPARED BY:

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DATE:

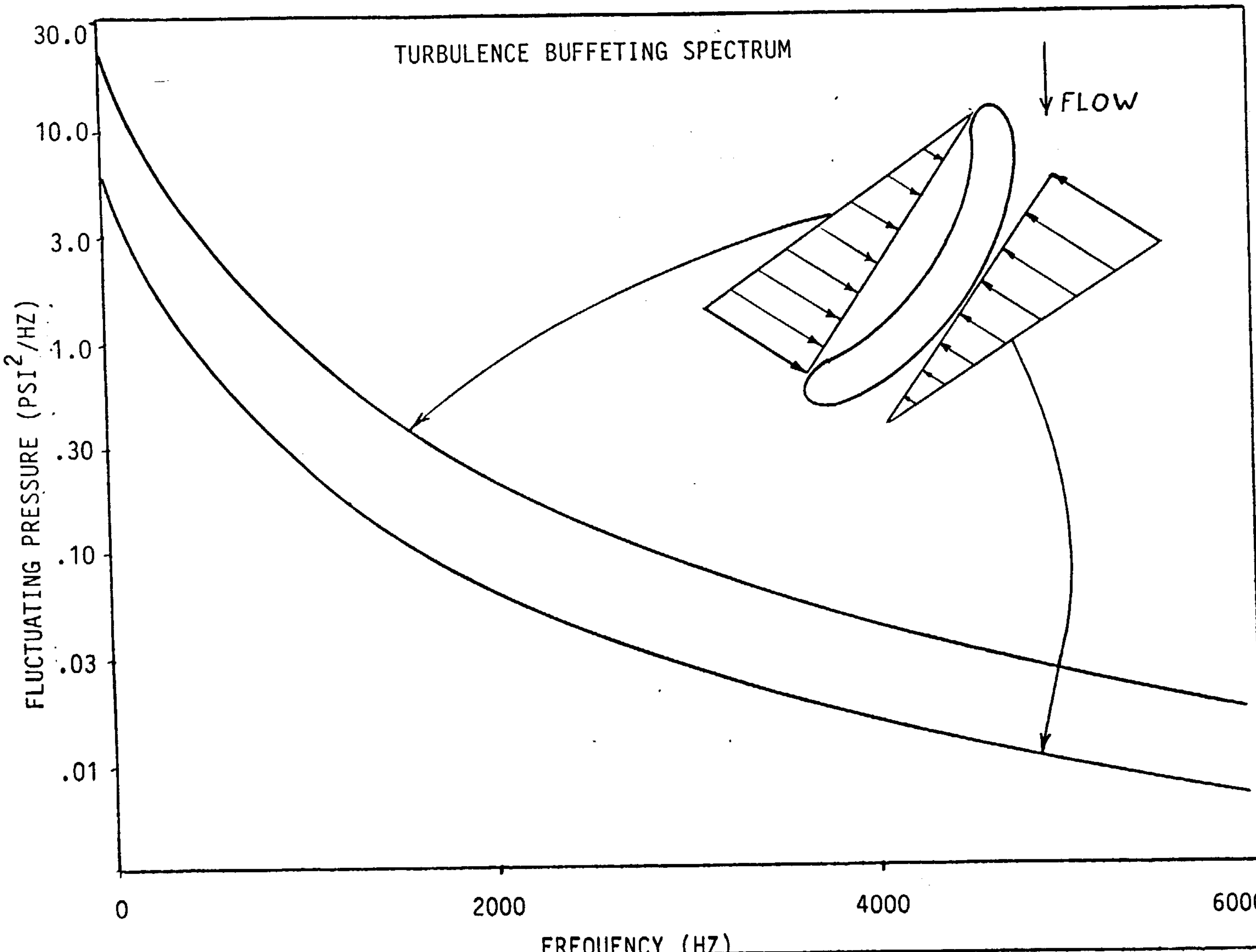


Figure 10

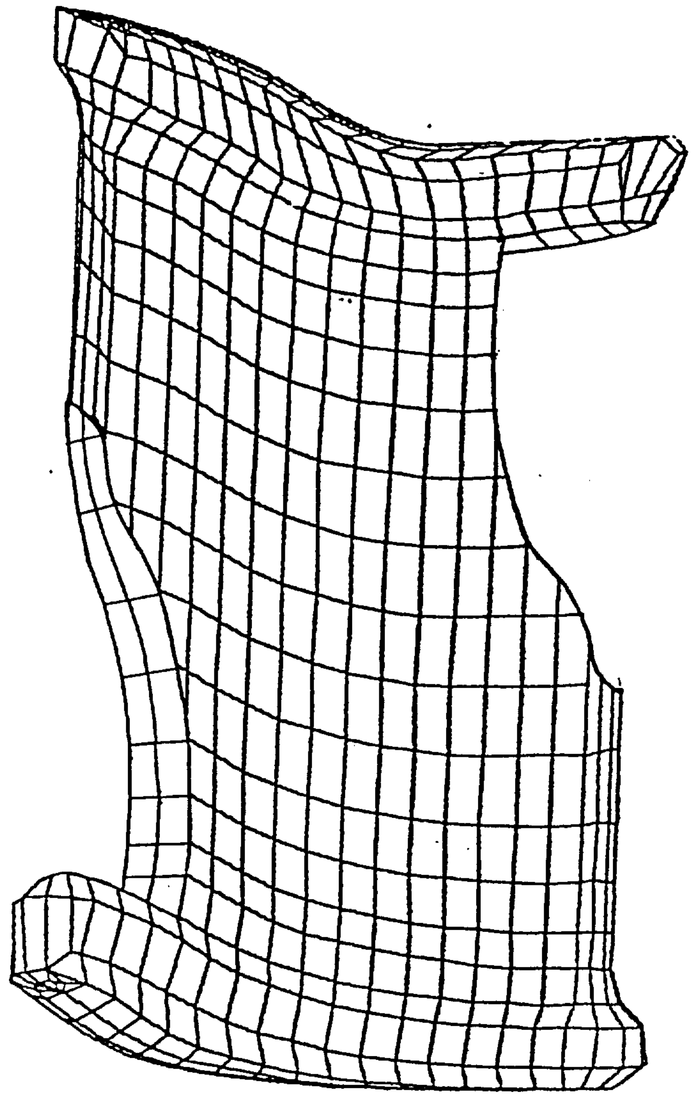
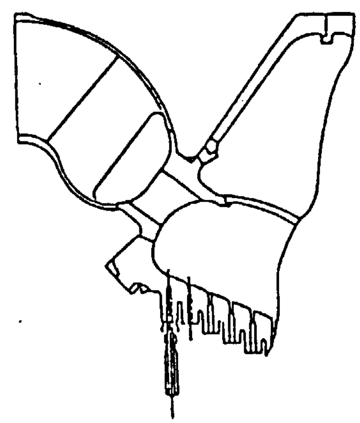
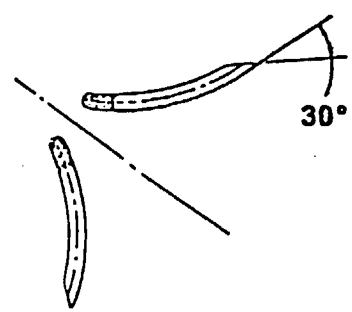


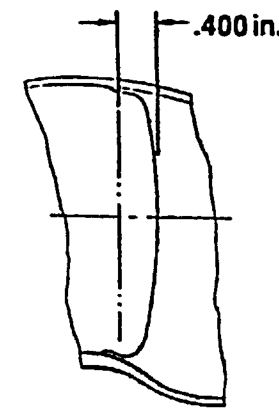
Figure 11



Baseline Configuration



Trailing Edge



Leading Edge

Figure 12

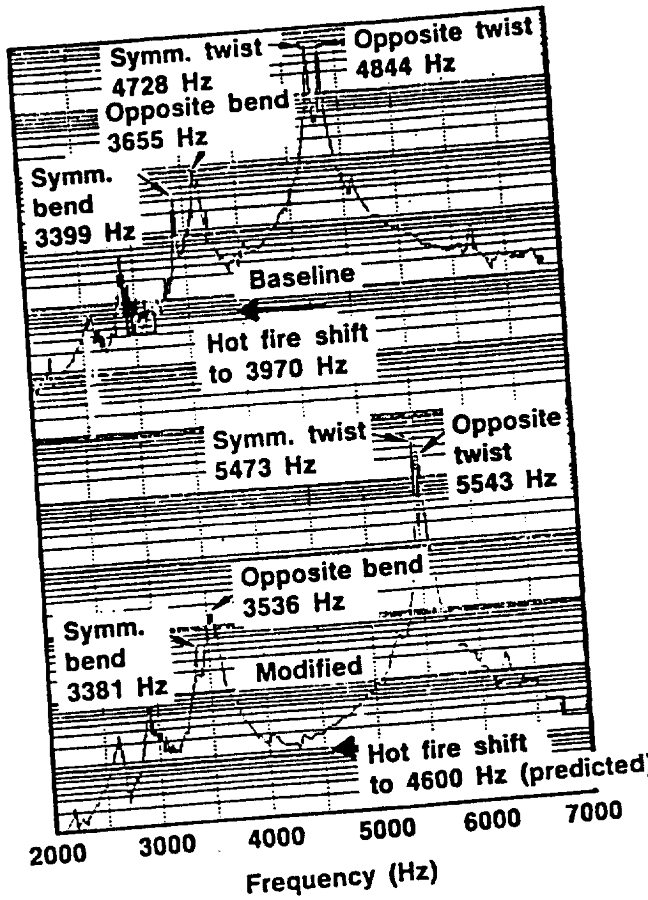


Figure 13

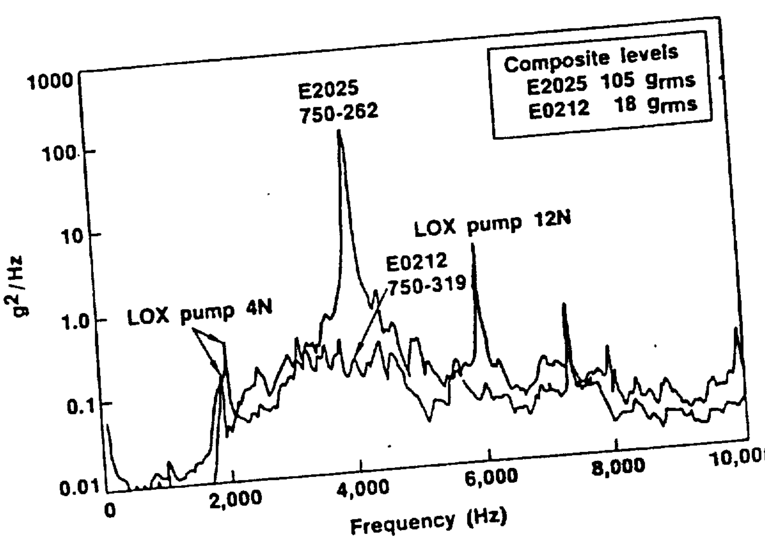


Figure 14

- EVALUATION OF BOUNDARY CONDITIONS AT THRUST CONE:

LOX INLET TEE W/O THRUST CONE (FIXED @ SEAM WELDS)	LOX INLET TEE W/O THRUST CONE (FIXED @ GIMBAL BRG)
3857 Hz	3870 Hz
3916 Hz	3916 Hz
4055 Hz	4060 Hz
4139 Hz	4080 Hz
	4533 Hz
4546 Hz	4545 Hz
4561 Hz	4594 Hz

TABLE A

MODAL TEST COMPARISON WITH MODEL (BASELINE)

ELBOW MODE FLANGE/ELBOW	SPLITTER VANE MODE	UNSYMMETRIC (OPP) OR SYMMETRIC (SYM)	MODEL FREQ. (HZ)	TEST ARTICLE POWERHEAD W/MOV	2116 POWERHEAD W/O MOV	2027 POWERHEAD ASSEMBLED ENGINE	ENGINE 0008 FREQ. (HZ)
2ND/2ND			2806	2933	2744	3099	2976
			2893	3048	3079	3245	3343
			3020	3155	3203		3389
			3212	3258	3314		
			3256	3620			
2ND/3RD			3496	4191	4011	4286	
				4299	4231	4852	
				4340	4758		
				5170			
3RD/3RD	1ST BEND	OPP	3895*				3713*
	1ST BEND	SYM	4041				
	2ND BEND	OPP	4197				
	1ST BEND	OPP	4337				
	TWIST	SYM	4830				4868**
	TWIST	OPP	4843				
	1ST BEND	OPP	4872				
	TWIST	SYM	4935				5165
			4977				
	TWIST	OPP	5198				
	TWIST	OPP	5444				
	TWIST	OPP	5360**				5347

* 1ST BENDING MODE BEING CONSIDERE
** TWISTING MODE BEING CONSIDERED

TABLE B

STRAIN GAGE	STARDYNE	HOT FIRE
2	99μIN./IN.	100μIN./IN.
4	58	43
5	52	27
6	33	35
7	27	20
8	34	23

TABLE C

MODAL TEST COMPARISON WITH MODEL (MODIFIED VANE)

MODEL FREQ (HZ)	ENGINE 0008 MODAL FREQ (HZ)	MODE SHAPE (SPLITTER VANE)	UNSYMMETRIC (OPP) OR SYMMETRIC (SYM)
	3372	1ST BENDING	OPP
	3390		OPP
	3434		OPP
	3452		OPP
3699*	3486		OPP
3619	3525		OPP
3814	3760	1ST BENDING	SYM
3918	4074		SYM
	4224		SYM
	5588	TWISTING	SYM
	5633		SYM
	5682		OPP
	5875		OPP
	5900		OPP
6061**	?		SYM
6245			SYM

* 1ST BENDING MODE BEING CONSIDERED

** TWISTING MODE BEING CONSIDERED

TABLE D