Investigation of the Flow-Induced Vibration in the E2 Test Facility

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1 Abstract

An investigation of flow induced vibration due to coupling between the fluid flow and the propellants lines (LOX and RP-1) was performed. Various flow rate conditions were studied to check whether flow induced vibration was possible due to vortex shedding in both valves and pipe lines. Resonance test was conducted for all segments of the LOX-feedline for the preburner under test. In addition, critical values of frequency and velocity are calculated using a mass damping model. A simple chart characterizing the relation between frequency and velocity is developed for each component; i.e. propellant lines, valves and flow meters. It was found that flow induced vibration occurs for various segments with flow rates of 113 lb/s, 275 lb/s and 40 lb/s. Even more interesting using critical conditions for buckling, it was found that the valve or pipe may collapse for a flow rate of 275 lb/s and valve height of 10% of pipe diameter. Furthermore, two models for the acoustic pressure acting on the segments particularly for the valve are proposed.

2 INTRODUCTION

Flow induced vibration occurs when the natural frequency of the line transporting the propellant and the fluid flow are the same. This matching of the two frequencies yields to a phenomenon known as resonance. This behavior in many cases yield to a collapse of the structure; such as in engine components and propellant lines. In order to understand whether re-design or additional testing is required critical values for the fluid frequency and flow rate will be calculated for each segment of the E2-LOX line.

There are various type of phenomena that may induce vibration on components; vortex shedding, turbulence, water hammer, acoustic among others. Figure 1 shows a flow chart with other cases. However, we will focus on vortex shedding studies and the water hammer problem. Vortex shedding occurs when the flow past over an obstacle such as cylinder, sphere or any other object is disturbed; resulting in vortices behind the cylinder. These vortices are moving downstream of the flow at a frequency, f_s and when this frequency is equal to the natural frequency of the structure, vibration is induced by the fluid flow. Water hammer occurs normally during the opening or closing of a valve, and it generates an acoustic wave that propagates upstream and downstream of the valve. This transient phenomenon shows up as a loud noise coming out of the pipe. This is what you have heard sometimes when you open the water faucet in the morning. Fluid flow through valves, bends and orifices generates turbulence as the flow passes through the obstacle. This in turns radiates acoustic waves (of velocity Ua and pressure Pa) upstream and downstream the valve (Blake 1996). So as the area of the valves and flow meters changes so the acoustic waves, and when the frequency of these waves matches the natural frequency of the pipe, vibration of the components is observed. This is because the waves have an acoustic pressure which acts against the surface of the pipe.

Consequently, the fluid flow and the solid surface are coupled through the forces exerted on the wall by the fluid flow. The fluid forces cause the structure to deform, and as the structure deforms it then produces changes in the flow. As a result, a feedback between the structure and flow occurs; action and reaction. Figure 3 shows a diagram of this action-reaction feedback loop between the fluid flow and structure. Because

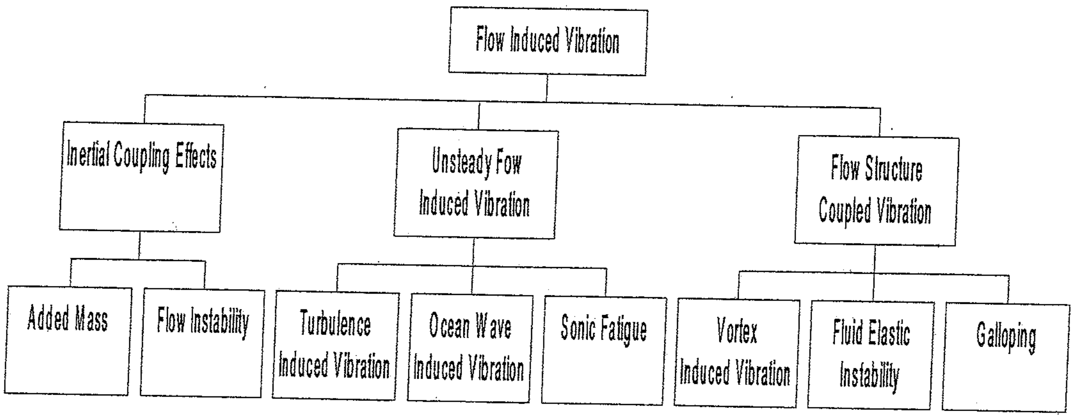

Figure 1: Flow Induced Vibration Cases.

of the interaction between the fluid flow and the solid surface the equations of motions describing the dynamics are coupled. This makes the problem more challenging, and even worse when the flow is turbulent. In addition, it means that we must solve the Navier-Stokes equation and the mass damping equation for the solid surface simultaneously with their corresponding boundary conditions.

3 The scope

As mentioned in previous section flow induced vibration is a result of the coupling of the frequency generated by the vortex shedding from fluid flow and the natural frequency of the feedlines of the propellants. The analysis of this investigation is divided into three phases:

- **Phase 1- Acoustic Model:**
  This phase consists of modeling the acoustic pressure acting on the structure. The Navier-Stokes equation, mass-spring model and structure equation will be used in order to couple the interaction between the fluid flow and the solid surface. Although the complete solution will not be given at this moment, future investigation will be carried out to study the accuracy of the model with actual experimental data.

- **Phase 2- Vortex Shedding:**
  In this phase we will determine the range of frequencies between the vortices and the natural frequencies of the feedlines. Then we will determined if flow induced vibration will occur due to coupling between the fluid flow vortices and the feedlines. The calculations of this phase will then tell us the critical frequency and flow rate at which such phenomena occurs.

- **Phase 3- Water hammer**
  In this phase flow induced vibration by water hammer phenomena will be studied. Because this

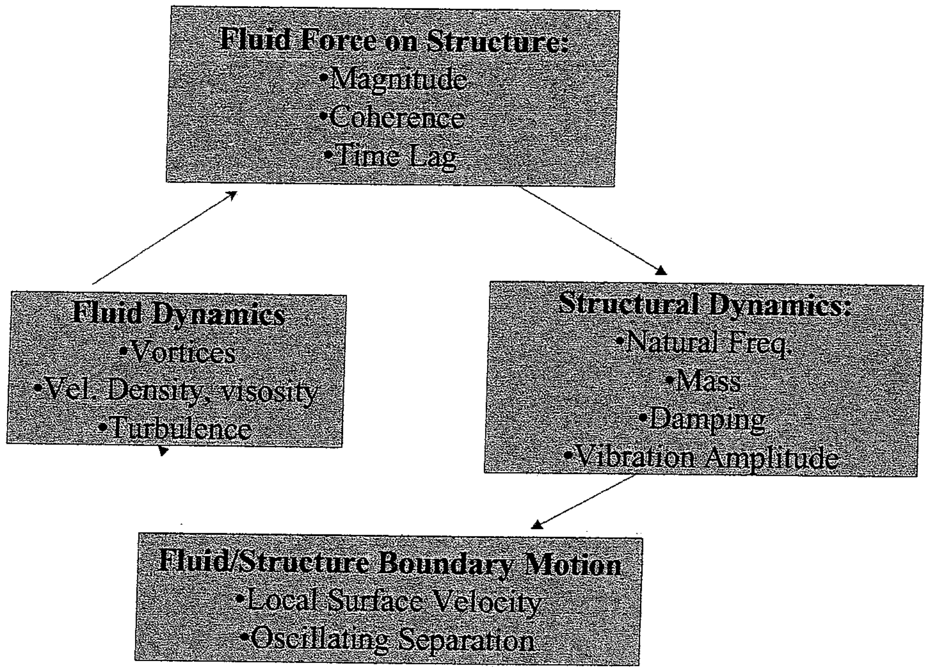

Figure 2: Feedbacl loop in Flow Induced Vibration.

phenomena occurs during the opening or closing of valve, special attention will be placed on valves. Determining the critical velocity and frequencies at which resonance occurs is imperative in this phase. In addition, buckling of the pipe or collapse of the valve will be carried out.

4 The Equations of Motion

It is clear now that the equations of motion for an incompressible flow can be written as,

$$\frac{\partial U_i}{\partial x} = 0 \tag{1}$$

and the Navier-Stokes equation,

$$\frac{\partial U_i}{\partial t} + U_j \frac{\partial U_i}{\partial x_j} = -\frac{1}{\rho}\frac{\partial P}{\partial x_i} + \frac{\partial}{\partial x_j}[\nu \frac{\partial U_i}{\partial x_j}] \tag{2}$$

where i and j vary from 1 to 3, thus yield the three components in the x,y and z plane of the velocity field. The first term in the left hand side represents the local change of velocity and the second term the transport of momentum. The first term from the right hand side is the pressure gradient and the second term is the viscous stress term. This is the well known Navier-Stokes equation for incompressible flow.

The equation describing the deformation of the solid material is obtained from Newton's second law and is given by,

$$EI\frac{\partial^4 y}{\partial x^4} + \rho A U^2 \frac{\partial^2 y}{\partial x^2} + 2\rho A U \frac{\partial^2 y}{\partial x \partial t} + M\frac{\partial^2 y}{\partial t^2} = f(x,t) \tag{3}$$

The first term from the left represents the elastic force experienced by the material as it deforms. Because now the material has been deformed, as the fluid flow travels inside the pipe it will experience two new forces arising from the centrifugal acceleration (second term) and coriolis forces (third term). The very last term from the left hand side is the acceleration or inertia term, details about this equation is given by Au-Yang (2001). Finally but not least, the first term from the right hand side represents the external forcing term due to the pressure fluctuations acting on the walls arising from the flow induced vibration. The advantage of using this equation is that as the pipe bends or in cases of elbows failure of the material due to buckling can be studied.

It is very clear that these equations are complicated to solve and must be solved simultaneously. Moreover, it is important to point out that this equation must be written with their proper boundary conditions for each element of the pipe including the valves, elbows, and flow meters. However, a simple mass-spring model for the system may allow us to provide a simple description of the system.

4.1 The Mass-Spring Model

Instead of trying to solve the Navier-Stokes equation and the equation for the solid surface it is possible to write down a simple set of equations for the acoustic pressure, Pa, and a mass spring model which will essentially provide information about the external forcing term due to flow induced vibration and the amplitudes of the wave for various modes. Thus,

$$(m + m_a)\frac{\partial^2 y}{\partial t^2} + (C_s + C_v)\frac{\partial y}{\partial t} + ky = f(t,x) \tag{4}$$

The first term is the inertia term or acceleration and it includes the mass added term, m_a, which the pipe experienced in bends. The second term is similar to the shear stress terms of the Navier-Stokes equation. Basically, this term is the damping term due to friction forces at the wall. The last term is the stiffens of the material, spring. Finally the only term in the right hand side is the forcing term and comes from the external forces due to flow induced vibration. This term needs a close form, and will be related to the acoustic velocity, u_a, of the wave traveling upstream and downstream of a given point.

As the fluid flow travels through valves, bends and orifices, it generates turbulence and radiates energy upstream and downstream of the area reduced. This acoustic waves reflected by changes in area creates a series of standing waves which in turns exert forces on the bends, valves and other surfaces and result in vibration. If the acoustics source possesses sufficient energy and the acoustic natural frequencies of the fluid flow (i.e. vortex shedding) coincide with the structural frequency then large acoustic motion can occur. In fact, I think that this wave propagates upstream and changes the thermodynamics properties of the system, resulting in changes of thermodynamic equilibrium points. Figure 3 shows a diagram of the waves generated at the valve and propagates upstream toward the tank, causing the conditions at the tank to vary from its equilibrium state.

4.2 The Acoustic Velocity Model

Based on the above information it makes sense to have a model for the forcing term, $f(t,x)$, which is a function of the acoustic wave speed, u_a, and the sound speed, c. Thus,

$$f(x,t) = P_a/A = (Ua_j \dot{n}_j)\frac{\rho C}{A} \tag{5}$$

where n_j is the unit vector in the direction of the acoustic velocity, Ua_j, and the dot product yields to a scalar. Notice that an equation for the acoustic wave velocity is required in order to know the forcing term, $f(t,x)$, or the acoustic pressure. Using the Navier-Stokes equations and the continuity equation it is possible to construct an equation for the acoustic wave or for the pressure wave. Assuming that the actual flow has the contribution of the mean flow, Uo_j, and the acoustic wave, Ua_j. This way of decomposing the flow is similar to the Reynolds decomposition or linear instability analysis. Thus,

$$U_j = Uo_j + Ua_j \tag{6}$$

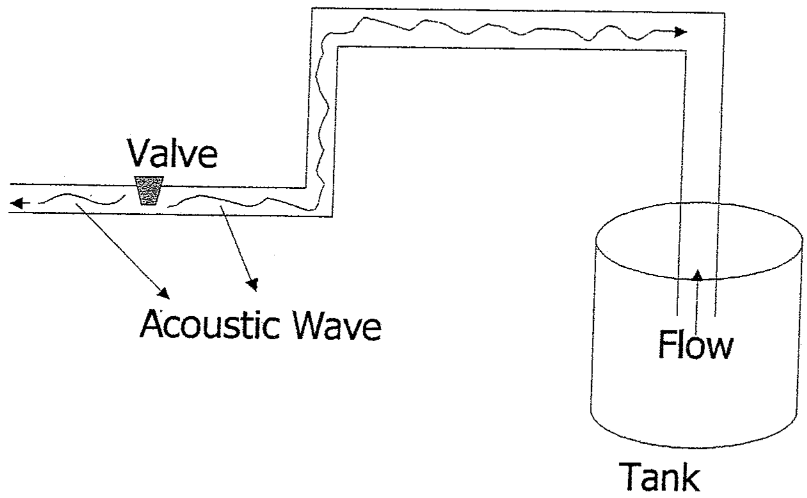

Figure 3: Acoustic Wave in Pipes

where $Uo_j >> Ua_j$. Thus, substituting equation 6 into the Navier-Stokes equation and using the continuity equation we get,

$$\frac{\partial Ua}{\partial t} + \frac{\partial U_j Ua}{\partial x_j} = -\frac{1}{\rho}\frac{\partial Pa}{\partial x_i} + \frac{\partial}{\partial x_j}[\nu \frac{\partial Ua}{\partial x_j}] \tag{7}$$

but, the waves are nearly inviscid so the last term can be dropped. Furthermore, this equation can be written in terms of the pressure waves and simplified since the mach number, M_o is less than 1 for incompressible flow. Thus,

$$\frac{\partial Pa}{\partial t} + c\frac{\partial Pa}{\partial x_i} = 0 \tag{8}$$

where c is the sound speed and since we are only interested in one direction (i.e. x_1 = x) we then re-write the above equation as,

$$\frac{\partial Pa}{\partial t} + c\frac{\partial Pa}{\partial x} = 0 \tag{9}$$

This equation can be solved by using separation of variables and its solution is the particular solution of equation 4 and equation 3. It is then necessary to construct a set of equations with its boundary conditions for each segment of the feedlines.

4.3 The Lift and Drag Model

The goal is to have another expression for the forcing term, f(x,t), that will enable us to study how far the acoustic wave will propagate upstream and downstream from a given point. The rationale for the following analysis is based on the fact that as the fluid flow passes through a valve or any other obstacle vortex structures are form. They in turn produce a lift force and drag force on the object. Thus, the equation for the lift force acting on the object is then given by,

$$F_L = \frac{\rho U_a^2}{2} DC_L sin(2\pi f_L t) \tag{10}$$

where U_a is the acoustic velocity to be determined from the Navier-Stokes equations, D is the diameter of the object, C_L is the lift coefficient, and f_L is the frequency at which the lift force acts on the object and is equal to the frequency of the vortex shedding, f_s. Moreover, the lift force acts perpendicular to the flow direction. Similarly, the object will experience a drag force, F_D, given as,

$$F_D = \frac{\rho U_a^2}{2} D C_D sin(2\pi f_D t) \tag{11}$$

where f_D is the drag frequency of this component given as $2f_s$, and C_D is the lift coefficient. Thus, having these two forces acting on the object we can then say that total force acting on the object by the acoustic wave is,

$$f(x,t) = F_L + F_D \tag{12}$$

now we only need to have a close form for the acoustic velocity. In previous section we relate the acoustic velocity with the acoustic pressure (i.e. $U_a = P_a/\rho c$), and assuming that the actual flow is composed of an undisturbed base flow and the wave disturbance due to vortex shedding. Moreover, it is important to assume in the Navier-Stokes equations that the acoustic wave is inviscid and that the fluid flow is incompressible. Therefore, after some simplification and manipulating the equations, the equation for the acoustic velocity is given by,

$$\frac{\partial Ua}{\partial t} + U_o \frac{\partial Ua}{\partial x} = 0 \tag{13}$$

where U_o is the bulk velocity. Using this equation into any of the structure equation (i.e. eqn. 3 or eqn. 4) with corresponding boundary conditions will give us the response of the structure due to the acoustic wave acting on the surface.

5 Test Facility and Input Data

The study of flow induced vibration will be carried out on the E2 facility for the LOX feedline. The schematic of this facility is shown in figure 4. Figure 5 shows the diagram for the E2 RP-1 facility. The study will evaluate each component shown in the diagrams (i.e. pipes and valves) for various flow rates. Input data for the E2 LOX is shown in figure 6 and figure 7. For the LOX studies, flow rates of 40 lbm/s, 113 lbm/s and a maximum of 275 lbm/s were analyzed and for each flow rate studies of resonance for the valve at various percentage of valve opening are considered (water-hammer). Some of the empirical calculations and approximations used for the model are given in this subsequent sections.

6 Results and Discussion

The results of this investigation consists of determining the natural frequency of each segment of the LOX-line and the frequency of the vortex sheddings. Studies using the frequency of the simple mass-spring equation will be reported in this section. In addition, the frequencies using the material deformation equation, particularly buckling of the pipe due to high fluid velocity are computed. The studies considered three cases of flow rates: 113 lb/s, 275 lb/s and 40 lb/s.

The frequency from the mass spring model of equation 4 is given by,

$$f_n = \frac{1}{2\pi}\sqrt{\frac{K}{m}} \tag{14}$$

where K represents the stiffness of the pipe and m the mass of the pipe and fluid. The values for K where obtained from the finite element analysis using Algor. The values for the frequency of the vortex shedding

E2 – LOX LINE

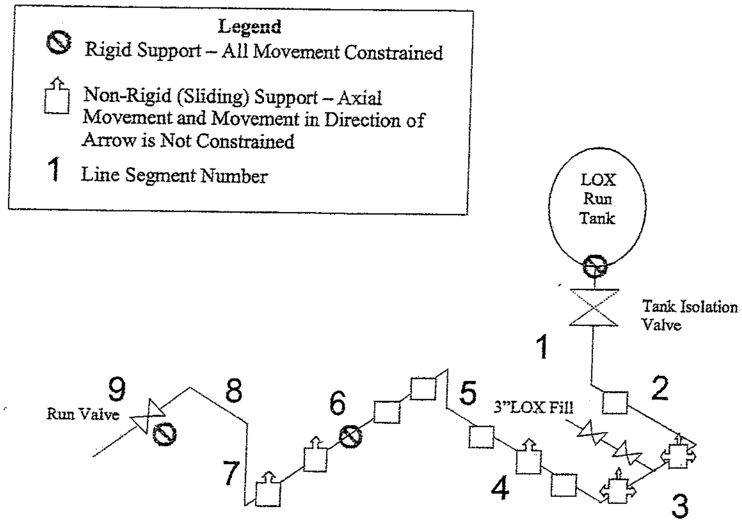

Figure 4: E2-LOX feedline diagram.

were determined from the following equation,

$$f_s = \frac{SU}{d_i} \tag{15}$$

where S is the Strouhal number, U is the flow velocity and d_i the inner pipe diameter. For the high Reynolds number range, $5.43 \times 10^5 \leq Re_D \leq 6.06 \times 10^6$, of this research a Strouhal number of about 0.41 was appropriate (Blevins, 1994).

6.1 Vortex Shedding Studies Using Simple Mass-Spring Analysis

Table 8 shows the flow induced vibration results due to vortex shedding. The first column from the left refers to the segment number of the LOX line, and the second column is the natural frequency of the pipe using equation 14. The third column from the left is the frequency of the vortex shedding (eqn. 15) for a mass flow rate of 113lb/s. The fourth column is the frequency for a maximum flow rate of 275 lb/s, and the last column is the frequency of the vortex shedding (eqn. 15) for a flow rate of 40 lb/s. The cells in greens indicates that flow induced vibration will occur for a given segment. Clearly most of the flow induced vibration due to vortex shedding will occur for a flow rate of 275 lb/s. Notice the vibration for 113 lb/s and 40 lb/s will be expected for segments 3 to 6, while for a flow rate of 275 lb/s (the maximum for this test, RS76) segments 2, 5 and 7 to 9 flow induced vibration will occur. The results from this table are plotted in figure 9.

6.2 The Effect of Valve on Flow Induced Vibration: Water-Hammer

The focus of this section was to investigate how closing or opening the valve induced vibration on the segments of the LOX-lines. Due to the lack of time only the valve located at segment 1 was analyzed. Table 10 shows the frequencies where flow induced vibration occurs. The first column is the segment for the LOX system. The second one refers to the % of the valve open. The third column is the diameter of the

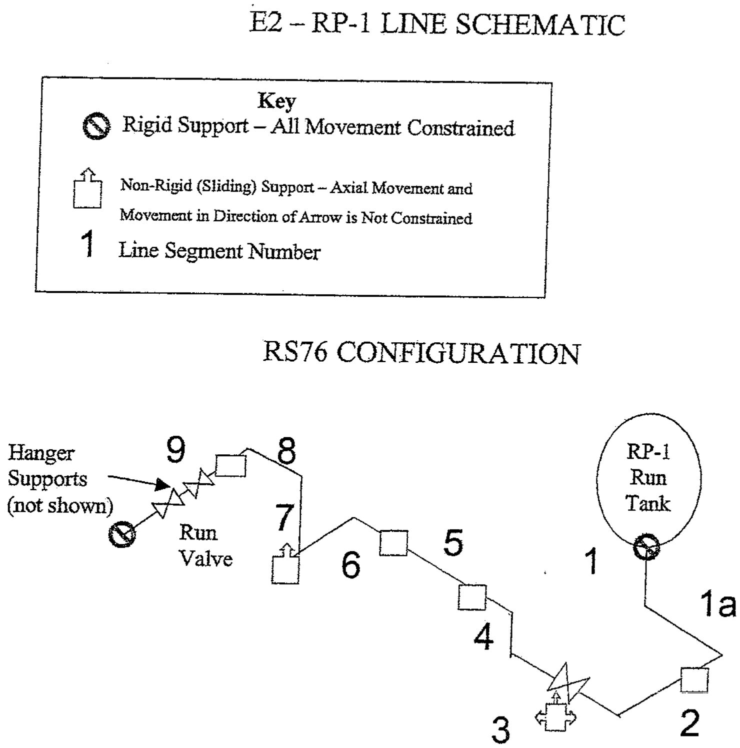

Figure 5: E2-RP1 feedline diagram.

valve in inches. The fourth column is the natural frequency of the pipe for each segment. The last three columns then refer to the vortex shedding frequency of the flow for mass flow rate of 113 lb/s, 275 lb/s and 40 lb/s consecutively as it passes through the valve.

Figure 11 shows the results of these same effects. It is clear that the flow rate of 40 lb/s indeed induced vibration for the first few segments (i.e. 1 to 6). For higher flow rate such as 113 lb/s and 275 lb/s vibration occurs at the segments near the end (i.e. 7 to 9). Notice that the frequencies are not exactly the same as the natural frequency of the pipes but close enough that resonance will take place. The fact that resonance occurs at 40 lbm/s may be due to the fact that as the flow is restrained in the valves its local Reynolds number increases to a new flow regime, which could be in transition to turbulence. As a result, other phenomena maybe superimpose on acoustic wave, resulting into a larger disturbance propagating upstream and downstream in the LOX-feedlines.

6.3 Buckling of Pipe due to High Velocity

In this section we will investigate buckling effects of the pipe as the speed through the valve is increased. For this study we will use equation 3 (i.e. the deformation of pipe due to high speeds). Thus, the natural

Segment	Description	O.D. (D_o)($in.$)	I (in^4)	I.D. (D_i)(in)	Length (in)	Pipe Area (in^2)
1	6" pipe-ACK4	6.625	79.155	4.209	101.0	20.558
2	6" pipe-ACK4	6.625	79.155	4.209	177.0	20.558
3	6" pipe-ACK4	6.625	79.155	4.209	115.5	20.558
4	6" pipe-ACK4	6.625	79.155	4.209	207.0	20.558
5	6" pipe-ACK4	6.625	79.155	4.209	29.50	20.558
6	6" pipe-ACK4	6.625	79.155	4.209	278.3	20.558
7	6" pipe-ACK4	6.625	79.155	4.209	89.00	20.558
8	6" pipe-ACK4	6.625	79.155	4.209	56.50	20.558
9a	6" pipe-ACK4	6.625	79.155	4.209	10.50	20.558
9	4" pipe-ACK4	4.500	17.893	2.598	67.40	10.603

Figure 6: E2-LOX Pipe Data: The letter I in the table represents the moment of inertia of the pipe.

Segment	Fluid Weight (lb.)	Pipe Weight (lb.)	Other Weight (lb)	Total Weight (lb)	Fluid Area (in^2)
1	57.8	596	–	653.8	13.914
2	101.2	144.4	–	1146	13.914
3	66.0	681.5	300[a]	1048	13.914
4	118	1221.5	–	1340	13.914
5	16.9	174.1	–	191	13.914
6	159	1642.2	400[b]	2201	13.914
7	50.9	525.2	–	576	13.914
8	32.3	333.4	–	366	13.914
9a	6.0	62	–	366	13.914
9	14.7	205	800[c]	8288[d]	5.301

Figure 7: E2-LOX Fluid Data: a-weight of fill line, b-weight of flow meter and filter, c-weight of run valves and rigid frame, d-weight includes segment 9a.

frequency of the structure is given by,

$$f_n = \frac{\pi}{2L^2}\sqrt{\frac{EI}{m_p + m_f}} \tag{16}$$

where E is the modulus of elasticity, I the moment of inertia, m_p the mass of the pipe and m_f the mass of the fluid. Similarly, Au-Yang (2001) determined a critical velocity where buckling of the pipe will take place and is given as,

$$V_{cr} = \frac{\pi}{L}\sqrt{\frac{EI}{\rho A}} \tag{17}$$

A fluid flow reaching this critical velocity, buckling of the pipe should be expected (AU-Yang, 2001). Table 12 shows the tabulated values for the natural frequency of the pipe using the equation 16. These results represent the vortex shedding of the fluid flow for fully open valve case.

As before, the first column from the left side refers to the segment of the E2-LOX feedline. The second column is the natural frequency of the pipe using the equation 16 for material deformation. The next three columns are the vortex shedding frequency (eqn 15) for mass flow rate of 113 lb/s, 275 lb/s, and 40 lb/s. The last column from the left is the length of the pipe. The entries in blue mean that flow induced vibration occurs for this length. The important point about this table is that only lengths greater than 4 meters experienced flow induced vibration. Also, notice that essentially flow induced vibration due to vortex shedding occurs at flow rate of 275 lb/s which is the maximum expected for this test at the E2 (RS76).

Table 13 shows the critical velocity of equation 17 for the fully open valve. Notice that no buckling due to high speed will ever occur for any of the flow rates considered. The velocities shown in this table have units of m/s.

However, the results for various valve height are shown in table 14. Notice that when the valve is open 10% a velocity of 37m/s is achieved by the flow passing through the valve, and its value exceed the critical speed of 21 m/s. Therefore, for a mass flow rate of 275lb/s the segment 1 or the valve will collapse due to high flow velocity. The other items mark in purple color does not exceed the critical velocity but, it is close enough (i.e. 15 m/s for segment 1 at 113 lb/s and 9.47 m/s for segment 2 at 275 lb/s) so that close monitoring of the system should be considered. Notice that for segment 2 the critical velocity is 12.07 m/s and the velocity of the flow through the valve is 9.47 m/s. For the other segments values as close as these ones are not observed.

It is very important to understand that the solutions shown here are assuming that a valve of spherical shape is used to constrain the flow. Moreover, the nature of the vortex shedding and resonance will depend on the geometrical and other variables of the valve. For instance, figure 15 shows some profiles from Castillo at el. (2000, 2001) investigating the effect of initial conditions on the downstream flow. Notice that various curves for the mean and Reynolds stresses profiles exist depending on the shape, location and wind tunnels speed of the tripping device; which can be viewed as different shape and velocities through the valve. The top figure is the mean velocity deficit profiles for adverse pressure gradient, favorable and zero pressure gradient flow. The interesting observation is that even tough some of the profiles have the same range of local Reynolds number and same strength of pressure gradient the profiles are different. This difference observed in all of the profiles is due to the fact that different initial conditions were used in each experiment, and as a result the profiles of the turbulent flow develop differently depending on those conditions. The profiles at the center show the growth of the boundary layer for the adverse pressure gradient profiles of the first figure. Again notice the remarkable difference among the profiles. This may explain why the turbulence models used under-predict the growth rate of plumes. Finally, the profiles at the bottom represent the Reynolds stress profiles in the y-direction for zero pressure gradient data of Castillo and Johansson (2000). Notice that this is one of the terms that arise due to turbulence, and a similar effect will be expected in the acoustic velocity as the flow is block by the valve or flow meters.

7 Conclusion

It was found that flow induced vibration due to vortex shedding do occur for the LOX line of the E2 test facility. The analysis for various valve height (water-hammer) and for various flow rate (i.e. 113 lb/s, 275 lb/s and 40 lb/s) was performed. Furthermore, for a maximum flow rate of 275 lb/s bending of the segment 1 and possibly collapse of the valve may occur. The resonance effect occurring between the pipe and fluid flow produced an acoustic wave which propagates upstream and downstream of the test valve. More interesting this phenomena may explain the drastic change of the thermodynamic properties of the LOX tank. As the flow is restrained by the valve the fluid flow speed increases, and the vortex shedding behind the valve increases also.

Moreover, simple models for the acoustic pressure acting on the pipe surface was proposed. One of the approach relates the acoustic pressure with the acoustics velocity. This velocity is obtained from the Navier-Stokes equation. The second approach involved the lift force and drag force experienced by the valve due to vortex shedding. They both enter the mass-spring model as a forcing term and its solution describes the interaction between the solid surface and fluid flow.

In summary, further research is required to model the other valve, elbows and flow meter. Testing of the models for each segment and component of the LOX line is required. Also, similar analysis is necessary for the RP1 feedline, particularly of the valves. Verification of this results will be carried usingexperimental techniques developed by NASA Stennis. Using a software, FREDOM, it will be possible to test the theoretical models and results presented in this study. Although some approximations were necessary in order to get some understanding of the problem, the present analysis lead to better understanding of the current situation.

8 Acknowledgments

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Flow Induced Vibration: Vortex Shedding Using Simple Model: Mass-Spring

Segment	fn	f_f(113)	f_fmax(275)	f_f(40)
1	2.9491697	0.59737901	1.45370771	0.21140178
2	2.22795143	0.59737901	1.45370771	0.21140178
3	0.34812856	0.59737901	1.45370771	0.21140178
4	0.2962007	0.59737901	1.45370771	0.21140178
5	0.78440572	0.59737901	1.45370771	0.21140178
6	0.67367987	0.59737901	1.45370771	0.21140178
7	1.19120252	0.59737901	1.45370771	0.21140178
8	2.61684451	0.59737901	1.45370771	0.21140178
9	4.68549791	2.54020808	6.18153639	0.89893434

Figure 8: Flow Induced Vibration by Vortex Shedding: Fully Open Valves

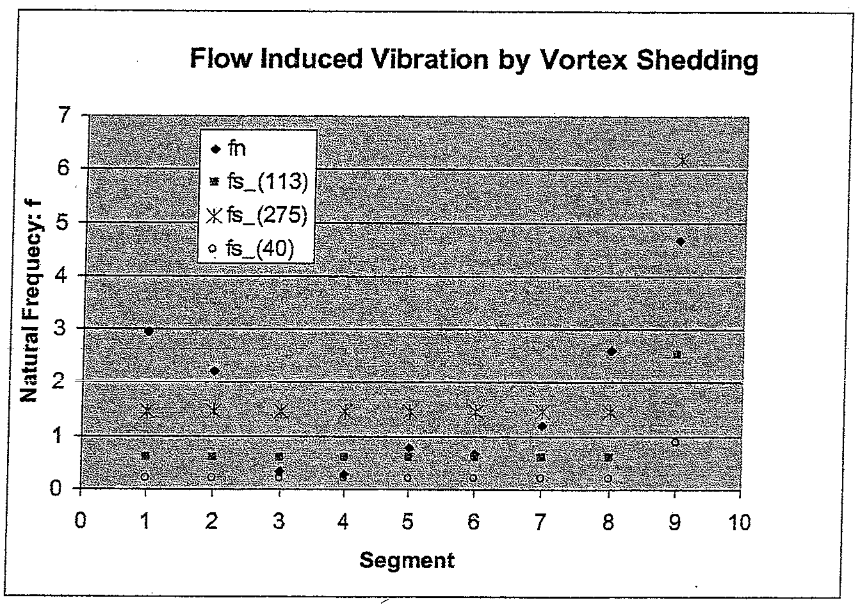

Figure 9: Flow Induced Vibration by Vortex Shedding: Fully Open Valves

VORTEX SHEDDINGS CALCULATIONS

Segment	% Pipe Di	di_valve(in)	fn	fs_(113)	fs_(275)	fs_(40)
1	10	0.4209	2.9491697	597.379005	1453.70771	2.11401778
2	20	0.8418	2.22795143	74.6723757	181.713464	1.05700889
3	30	1.2627	0.34812856	22.1251483	53.8410262	0.70467259
4	40	1.6836	0.2962007	9.33404696	22.7141829	0.52850445
5	50	2.1045	0.78440572	4.77903204	11.6296617	0.42280356
6	60	2.5254	0.67367987	2.76564354	6.73012828	0.3523363
7	70	2.9463	1.19120252	1.74162975	4.23821489	0.30200254
8	80	3.3672	2.61684451	1.16675587	2.83927287	0.26425222
9	90	2.3382	4.68549791	3.48451039	8.47947379	0.99881594

Figure 10: Effect of Closing or Opening the Valve in Flow Induced Vibration by Vortex Shedding

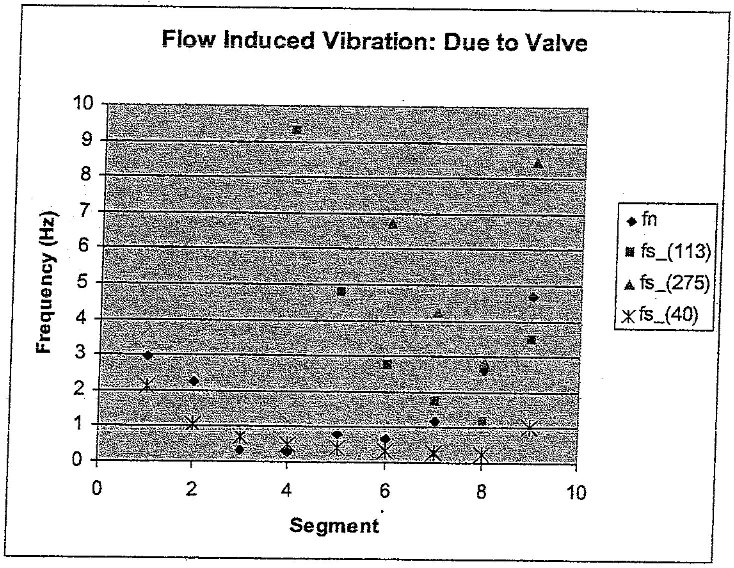

Figure 11: Flow Induce Vibration due to Valve Opening or Closing: Water-Hammer

Critical Velocity at Buckling for Various Valve Location						
Segment	% Pipe Di	di_valve(in)	Vc	Vel_val_113	Vel_val_275	Vel_val_40
1.000	10.000	0.421	21.077	15.577	37.906	5.512
2.000	20.000	0.842	12.027	3.894	9.476	1.378
3.000	30.000	1.263	18.431	1.731	4.212	0.612
4.000	40.000	1.684	10.284	0.974	2.369	0.345
5.000	50.000	2.105	72.160	0.623	1.516	0.220
6.000	60.000	2.525	7.649	0.433	1.053	0.153
7.000	70.000	2.946	23.918	0.318	0.774	0.112
8.000	80.000	3.367	37.677	0.243	0.592	0.086
9.000	90.000	2.338	24.328	0.505	1.228	0.179

Figure 14: Critical Velocity for Buckling of Pipe: Valve at Various Height.

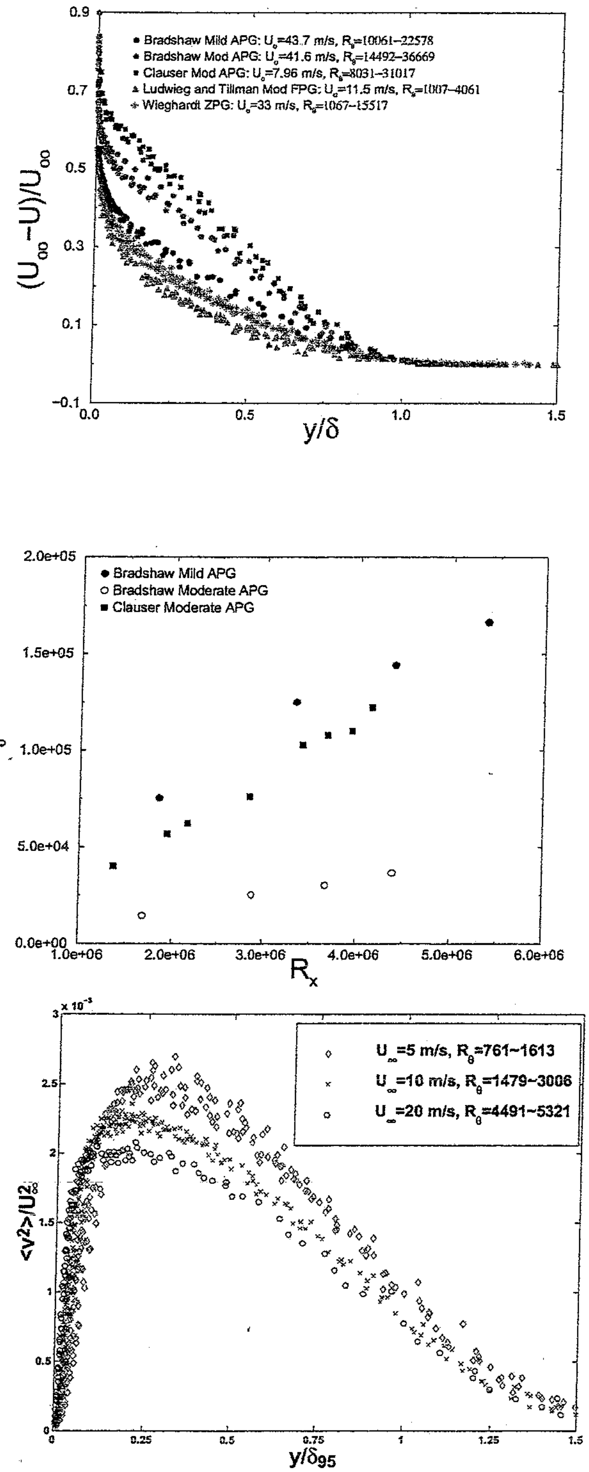

Figure 15: The effect of the initial conditions on the flow.

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