Transient Simulation of Pressure Oscillations in the Fuel Feedline of the Fastrac Engine Thrust Chamber

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Introduction

During mainstage testing of the 60,000 lbf thrust Fastrac thrust chamber at MSFC's Test Stand 116 (TS116), sustained, large amplitude oscillations near 530 Hz were observed in the pressure data. These oscillations were detected both in the RP-1 feedline, downstream of the cavitating venturi, and in the combustion chamber. The driver of the instability is believed to be feedline excitation driven by either periodic cavity collapse at the exit of the cavitating venturi or combustion instability. A portion of the TS116 system is shown in Figure 1.

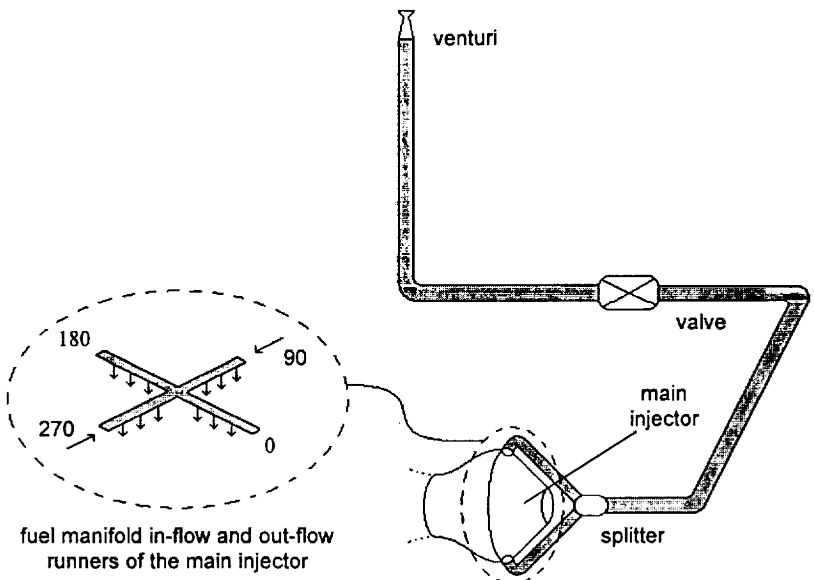

Figure 1. Test stand 116 fuel feedline and engine manifold layout.

In a cavitating venturi, static pressure drops as the flow passes through a constriction resembling a converging-diverging nozzle until the vapor pressure is reached. At the venturi throat, the flow is essentially choked, which is why these devices are typically used for mass flow rate control and disturbance isolation[1]. Typically, a total pressure drop of 15% or more across the venturi is required for cavitation. For much larger pressure differentials, unstable cavities can form and subsequently collapse downstream of the throat[2]. Although the disturbances generated by cavitating venturis is generally considered to be broad-band, this type of phenomena could generate periodic behavior capable of exciting the feedline. An excitation brought about by combustion instability would result from the coupling of a combustion chamber acoustic mode and a feedline resonance frequency. This type of coupling is referred to as "buzz" and is not uncommon for engines in this thrust range[3].

This analysis addresses the question of what natural frequencies are present in the feedline system and how they react to a driving frequency at either the venturi or combustion chamber.

Numerical Procedure

The transient response of the TS116 fluid system was modeled with a computer code based on a one-dimensional method of characteristics[1] (MOC) routine. This method continuously tracks unsteady pressure waves in a pipe network, resulting from a disturbance such as the closing of a valve or any other time-varying pressure or velocity boundary condition.

The basic principle of the method revolves around a network of sonic lines, called characteristics, that project in either direction at the local speed of sound from equally-spaced nodes throughout a system. As a disturbance propagates into the nodal network, it moves along one of these left- or right-running characteristics, as seen when projected onto an x-t plane, until it encounters a condition capable of producing a change in pressure or velocity such as a valve or reservoir. At this point, the wave may be partially or completely reflected, depending on the situation. If the wave is only partially transmitted, then a secondary wave is reflected back into the opposite direction. As reflections continue to occur, the system becomes filled with waves constantly interacting with one another. Figure 2 illustrates the complexity of the wave system produced for a single-pipe model with two in-line obstructions and a closed end.

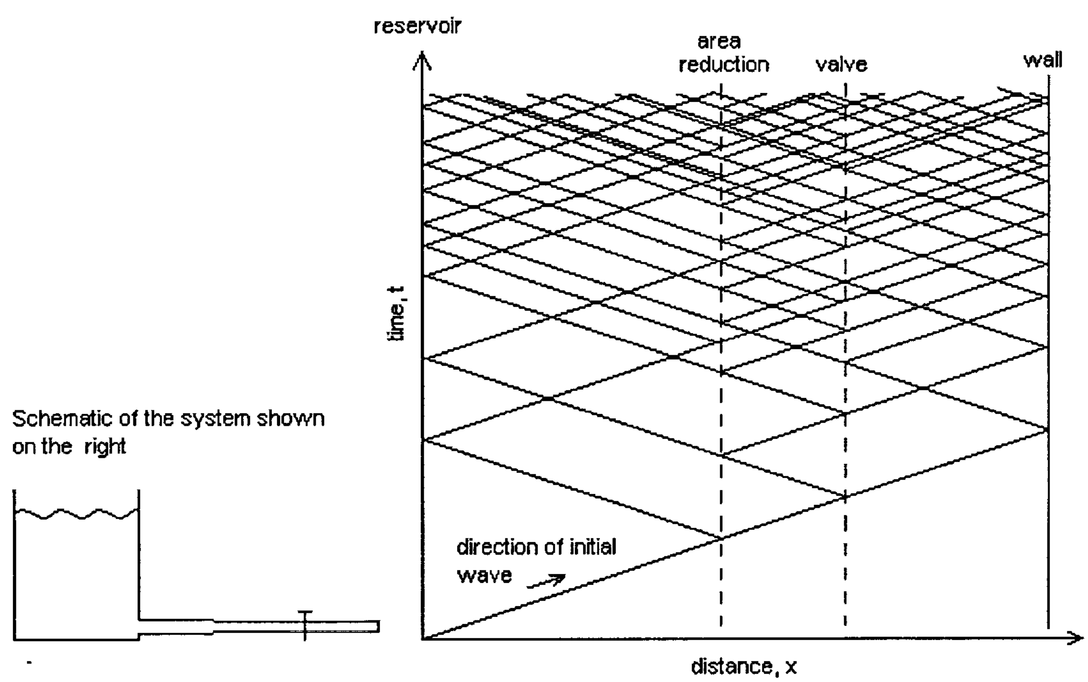

Figure 2. x-t diagram of a simple reservoir-closed end system.

The primary influences on system resonance frequencies are the speed of sound in the liquid, a, and pipe lengths, L. For a system with a single, uninterrupted span of pipe, the frequency of the first mode is simply the reciprocal of the wave round-trip transit time in the pipe, f_n=a/2L, for an "open-open" system such as a reservoir/open valve combination, or half this value, f_n=a/4L, for an "open-closed" system, as with a reservoir/closed valve combination.

A coupled speed of sound for the liquid/pipe combination is computed in the analysis because pipe expansion or contraction resulting from changes in pipe internal pressure can significantly affect the wave speed in the fluid. The coupled speed of sound is computed from Equation 1 and has the units of ft/sec. The terms in this equation are: ρ = fluid density, k = fluid bulk modulus, D = pipe ID, ϕ = restraint factor, t = pipe wall thickness, and E = pipe elastic modulus.

$$a = \left[\frac{\rho}{g_c}\left(\frac{1}{k} + \frac{D\phi}{tE}\right)\right]^{-1/2} \qquad \text{Eq. 1}$$

Fluid density in the system is held constant in order to maintain a constant speed of sound. This is necessary in order for characteristics to reach the nodes at predictable times. The assumption of an isothermal system is imposed as a consequence of the constant density requirement. Because the flow work imparted to the fluid during compression or expansion is typically small for liquid systems, this is justifiable.

System Model

The TS116 fuel feedline was modeled from the cavitating venturi to the main injector fuel manifold. None of the facility upstream of the venturi was modeled because the venturi is essentially choked during mainstage, inhibiting the passage of flow disturbances upstream of this point.

Referring to Figure 1, the facility section of the feedline from the venturi to the fuel splitter block consists of 3 in. stainless steel pipe and contains a control valve approximately half-way between the splitter and the venturi. The flow splits into two ducts upstream of the injector before it is directed into the fuel manifold. Fuel enters the manifold at the 90 and 270 degree locations (inset of Figure 1) and runs inward toward the center, where it is directed outward via the 0 and 180 degree runners. From these four passages, the flow exits into the combustion chamber through a set of channels that direct the fuel to the injector faceplate. In order to simulate the out-flow of fuel from the manifold to the combustion chamber, equal portions of the flow are tapped-off from three locations in each of the four runners in the manifold and linked to a boundary condition representing the combustion chamber.

Each pipe in the system was divided into an integer number of sections based on the shortest pipe in the network. In this model, the shortest pipe was 0.2 ft and was divided into four sections. Because pressure waves move at the speed of sound in the liquid, time steps for this method are defined as the time required for a wave to travel from one node to the next, $\delta t = \delta x/a$. Using a calculated RP-1 speed of sound of 3025 ft/sec, a time step of 1.653E-5 seconds was obtained for the analysis.

The oscillating pressure boundary condition at the venturi/combustion chamber was modeled as $P(t)=P_o(1+A\sin(2\pi Bt))$, where P_o is the steady state pressure, A is the amplitude of the oscillation, provided as a fraction of P_o, and *B* is the frequency in Hz. A surface roughness of 0.0018 in. was used throughout the system to compute frictional losses. Minor losses in the system included the control valve, elbows, and sudden contractions.

Results

During the analysis, the system was driven at discreet frequencies approaching the 530 Hz value seen in the test data at amplitudes of ±5% of the steady state pressure. Larger amplitudes resulted in localized regions within the pipe where pressure dropped below the fuel vapor pressure of 0.03 psia, in which case the liquid in the line flashes to a gas. Because the program could not account for the substantial changes in sonic velocity between liquid and gas, care was taken to insure that this did not occur.

Figures 3 and 4 show time- and frequency-domain plots, respectively, at the valve inlet when the venturi is driving the system at 450 Hz. This frequency was chosen in order to excite the system without obscuring the response near 530 Hz. Unfortunately, a means of driving the system with broadband noise was not available at the time that this analysis was run. The peak-to-peak pressure differential for this case is approximately 115 psi. The 1st mode resonance of 66.7 Hz can be seen easily in the PSD of Figure 4, as can the driving frequency of 450 Hz. This first mode prediction corresponds to a reflection between the venturi and a point within the fuel manifold. The peak at 533 Hz, seen in the PSD of Figure 4, represents an 8x multiple of the 1st feedline mode and matches the frequency observed in the test data.

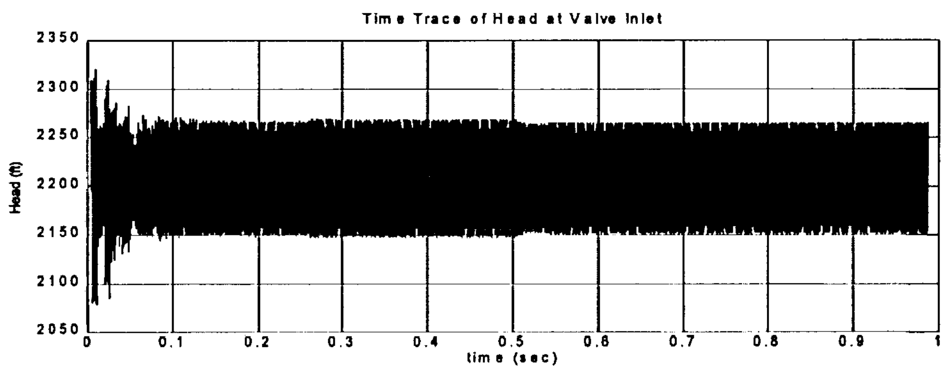

Figure 3. Time-domain response at the valve inlet to 450 Hz applied at the venturi.

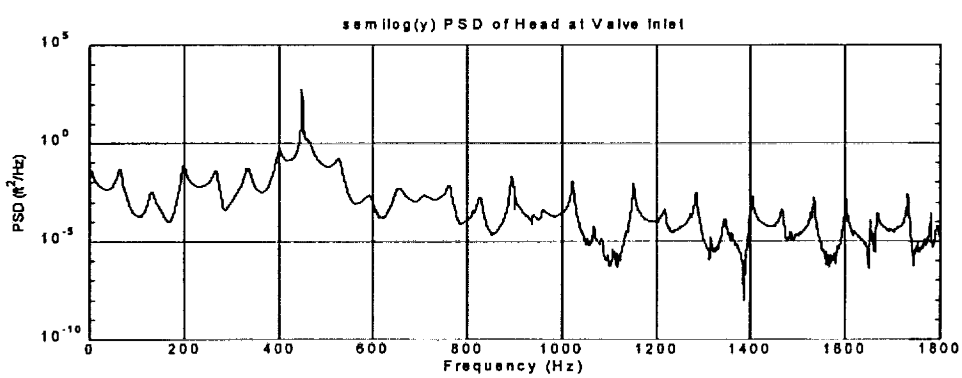

Figure 4. Frequency-domain response at the valve inlet to 450 Hz applied at the venturi.

When the driving frequency at the venturi is increased to 510 Hz, the 533 Hz mode becomes partially excited and couples with the driving excitation, producing peak-to-peak pressures of 260 psi. Figures 5 and 6 show the time- and frequency-domain plots for this case, respectively. When the driving frequency at the venturi was increased to 530 Hz, the solution diverged due to the strong coupling of the driving and feedline frequencies.

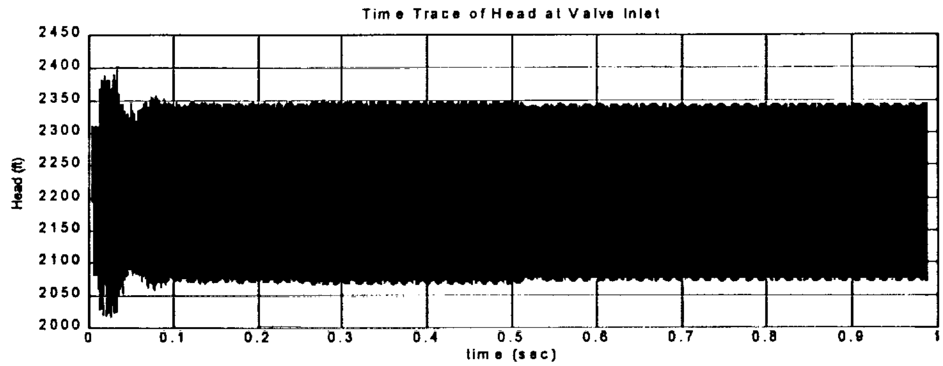

Figure 5. Time-domain response at the valve inlet to 510 Hz applied at the venturi.

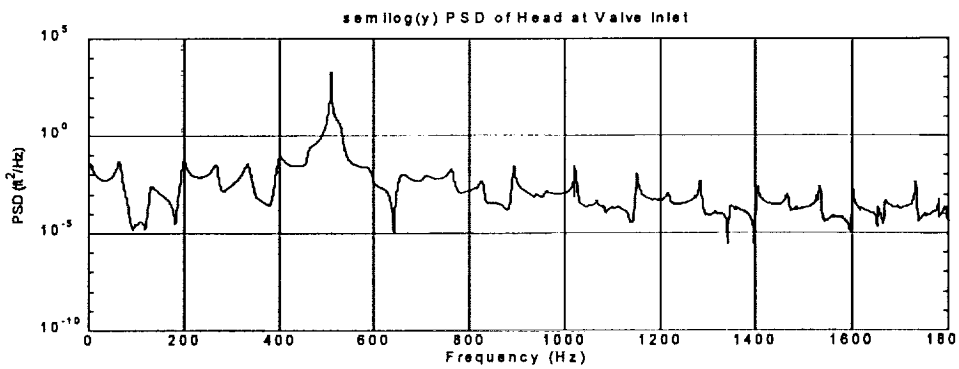

Figure 6. Frequency-domain response at the valve inlet to 510 Hz applied at the venturi.

The feedline response of the system when driven at the combustion chamber is similar to that generated when the system is driven at the venturi, with the exceptions that 1) the 1st mode is 180 degrees out of phase with venturi-driven cases and 2) all major peaks seen in the PSDs of the venturi-driven system above 1020 Hz are no longer present. Figures 7 and 8 show the feedline response when driven at 510 Hz from the combustion chamber.

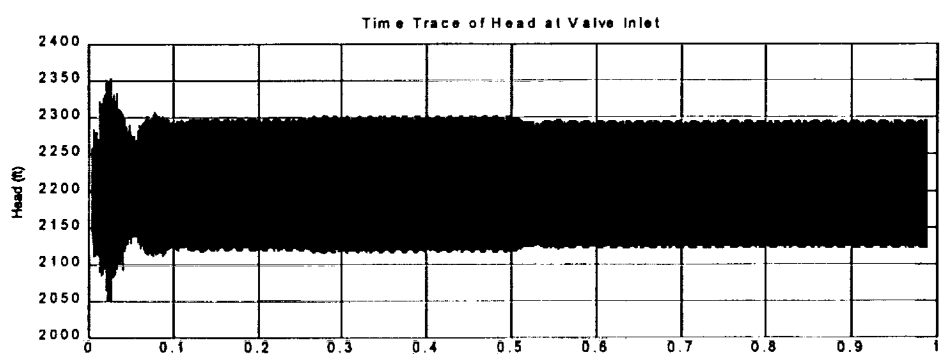

Figure 7. Time-domain response at the valve inlet to 510 Hz applied at the combustion chamber.

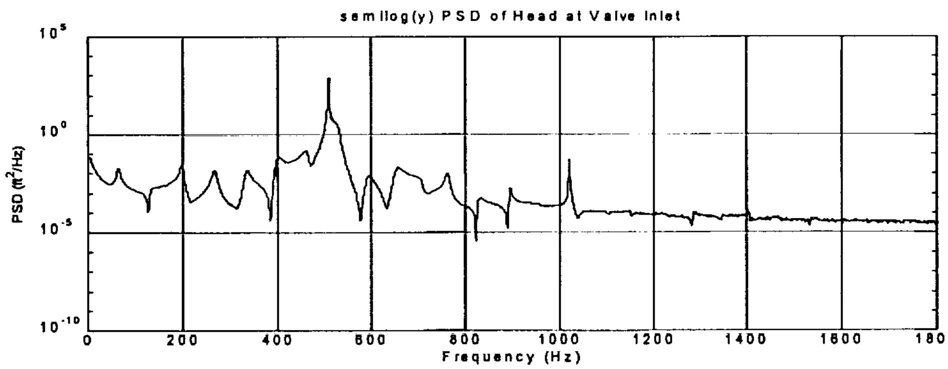

Figure 8. Frequency-domain response at the valve inlet to 510 Hz applied at the combustion chamber.

Based on the analysis results, it can be seen that a feedline resonance exists at the frequency observed in the test data and could be excited through either venturi- or combustion chamber-driven pressure oscillations. Tests are currently being planned in which the pressure differential across the venturi will be reduced. If the venturi is indeed the driving source of the vibrations, this should eliminate the discreet frequency component generated by cavity collapse. If the driving oscillations are generated in the chamber, then the addition of an accumulator to the feedline may be necessary.

References

1. paper on cavitating venturis
2. paper mentioned in tom nesman's presentation
3. Sutton, Rocket Propulsion Elements...