Fluid-hammer induced pressure oscillations in a cryogenic feed line

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Abstract. A transient, thermodynamic flow model is developed to simulate pressure oscillations in cryogenic fluid occurring due to sudden closing of valves, a phenomenon commonly known as fluid-hammering. The effects of line dimensions and flow rate changes on amplitude and frequency of these oscillations are investigated using numerical analysis. The model is validated with in-house experimental data and literature based on MOC solution for fluid-hammer. Current study is significant for understanding pressure oscillations during valve operation in launch vehicle cryogenic engine. Very low pressures caused due to fluid-hammer could lead to reduction in pump inlet pressure below saturation level, resulting in pump cavitation. Pressure oscillations also cause fluctuations in propellant flow rate, resulting in undesirable variations in thrust output from the engine. Computational analysis shows that increase in line diameter and reduction in the rate of change of flow rate reduce the peak amplitude of pressure oscillations.

1. Introduction

Sudden change in fluid momentum in a pipe results in pressure spike inside a feed system which triggers pressure oscillations in the liquid column. It can occur in a pipe line system when sudden closing operation of feed valves is carried out. These pressure oscillations are prominent in liquids whose compressibility is very low and pressure peaks are significant. Pressure oscillation effect is common in household hydraulic systems and was primarily studied with water as working-fluid [1], known as water-hammer studies. Liquid pressure oscillations caused due to water-hammer can damage the pipe lines themselves due to large differential pressure across pipe wall during pressure peaks [2]. Buffer tanks with trapped volumes are commonly used in long distance water transportation pipelines to counter the adverse effects due to these oscillations [3].

In cryogenic launch vehicle systems, thermally insulated feed lines are used to transportsub-cooled propellants, stored in large cryogenic tanks, towards combustion chamber of the engine. These lines, if poorly chilled prior to engine operation, experiences sudden pressure surges due to propellant evaporation [4] at the start of ignition. During the start-up operation of cryogenic engine, sudden closing of chill valve creates typical fluid-hammer effect inside the feed lines, resulting in characteristic pressure oscillations. This adversely affects the engine start up by causing combustion instabilities which result in undesirable thrust output. Moreover, low pressures encountered during pressure oscillations could lead to liquid cavitation in the pump which again results in thrust variations.

Various experimental and numerical studies regarding fluid-hammer phenomenon and chilldown characteristics are reported in literature. Early experiments were conducted by Joukowsky [5] who developed a relation between fluid pressure amplitude and momentum change. Cavitation of fluid during water-hammer pressure oscillations were studied by Billings et al. [6] and showed the...
possible dangers of the phenomenon. Impact of water-hammer on fluid structure was experimentally studied by Fan and Tijsseling [7]. Numerical simulation of water-hammer transients using Method of Characteristics (MOC) was first reported by Streeter and Wylie [8] which became a benchmark for subsequent numerical approaches. Effect of variation in flow area on pressure waves generated by water-hammer was experimentally and numerically studied by Meniconi et al [9]. Their study showed that small cross-sectional area in a pipe would lead to increase in amplitude of pressure oscillation. Tian et al [10] carried out numerical analysis to optimize water-hammer characteristics to minimize the damage due to the phenomenon for pipeline containing pump element. Steelant [11][12] investigated the water-hammer effects of priming operation in spacecraft propulsion system using simplified experimental setup with water as working fluid and developed numerical model for the same.

Though plenty of literatures on water-hammer and chilldown studies are available, very few literatures are available on investigation of fluid-hammer phenomenon using cryogenic fluid as working medium. In present study, fluid-hammer phenomenon is investigated with liquid hydrogen as working fluid to simulate the characteristic pressure oscillations during valve closing operation. A one-dimensional model is developed for the study which is validated using literature based on MOC and in house experiments. Effects of line dimensions and valve closing time are studied. Transient pressure profile at valve inlet location is determined for each case.

2. Model Description

The study aims to bring out the effect of water hammering due to sudden valve closing in a straight pipeline carrying cryogenic fluid. A comprehensive one-dimensional thermal and flow model is developed for this purpose as shown in figure 1. Super-insulated stainless steel lines of thickness 1mm are considered in the model, with source tank pressure and temperature of 5bar and 20K, respectively and ambient pressure of 1bar.

![Figure 1. Schematic of numerical model](image1)

The study is carried out with initial fluid flow rate of 100g/s as boundary condition, to capture the pressure oscillations due to fluid hammering. The scenario is typical in propellant feedlines during cryogenic tank filling operation. The model consists of 100m, 40mm line which is chilled to 20K. The valve closes in 10ms and the time history for valve operation is linear. Schematic of initial conditions for the studies are shown in figure 2.

![Figure 2. Schematic of initial conditions for valve closing case studies shown in Table 1](image2)

Matrix of initial condition is shown in Table 1. Pipe dimensions of various lengths (50m, 100m, 200m) and diameters (20mm, 40mm, 80mm), impact of different initial fluid flow rates (50g/s, 100g/s, 200g/s) and effect of different valve closure timings (10ms, 100ms, 200ms, 400ms) with linear valve operation profile on pressure oscillation characteristics are investigated. Source tank pressure of 5bar and temperature of 20K are considered and ambient pressure of 0.2bar is taken. Matrix of initial conditions, for various valve opening case studies, is shown in Table 1.
Table 1. Matrix of initial condition

| Case                      | Varying Parameter | Values            |
|---------------------------|-------------------|-------------------|
| (Base case: L=100m,      | Length, L         | 50m, 100m, 200m   |
| d=40mm, valve closing     | Diameter, d       | 20mm, 40mm, 80mm  |
| time=10ms, flow rate=100g/s) | Flow rate        | 50g/s, 100g/s, 200g/s |
|                           | Valve closing time| 10ms, 100ms, 200ms, 400ms |

3. Mathematical Formulations

A transient one-dimensional fluid dynamic formulations are developed to study the fluid hammer phenomenon using lumped-parameter network flow simulator called SINDA/FLUINT. The simulator solves mass, momentum and energy conservation equations based on finite-difference method. Following are the mathematical formulations used in this study:

3.1 Mass Conservation

The net mass flow from the fluid lump must equate to the rate of change of mass in the control volume of the lump as shown below:

\[ m_u - m_d = \frac{dm}{dt} \]  \hspace{1cm} (1)

3.2 Momentum Conservation

The governing equation for flow connectors is simply a complex form of Newton’s second law which can be expressed as:

\[ \frac{dm}{dt} = A \frac{L}{L'} [(P_u - P_d) + f_r \dot{m}^2 - K_f \dot{m}^2] \]  \hspace{1cm} (2)

A recoverable loss is the pressure drop that becomes a pressure gain if the flow is reversed. This is most often caused by area and density changes between the inlet and the outlet of the flow line. Recoverable loss coefficient [13] is shown as below:

\[ f_r = \frac{2 \cdot (p_d A_d - p_u A_u)}{\rho_d A_d \cdot \rho_u A_u \cdot (A_u + A_d)} \]  \hspace{1cm} (3)

Single-phase frictional pressure drop is calculated using Darcy friction factor [14] as shown below

\[ f = 8 \left( \frac{8}{Re} \right)^{12} + \frac{1}{(a + b)^{3/2}}^{1/12} \]  \hspace{1cm} (4)

where,

\[ a = -2.457 \ln \left( \frac{7}{Re} \right)^{0.9} + \frac{0.27 \varepsilon}{D} \]  \hspace{1cm} (5)

\[ b = \left( \frac{37530}{Re} \right)^{16} \]  \hspace{1cm} (6)

Two-phase pressure drop is calculated using Lockhart-Martinelli correlations [15].

3.3 Energy Conservation

According to first law of thermodynamics, the energy conservation equation based on enthalpy can be written as below:

\[ \frac{dU}{dt} = (H_u m_u - H_d m_d) + Q \]  \hspace{1cm} (7)

where,

\[ Q = h_c A_{w'} (T_s - T_f) \]  \hspace{1cm} (8)
The rate of increase of internal energy in the control volume is equal to the rate of energy transport into the control volume minus the rate of energy transport from the control volume.

3.4 Equation of State
The resident mass in fluid lump can be expressed from the equation of state for a real fluid as:

\[
m = \frac{P \cdot V}{R \cdot T \cdot z}
\]

(9)

3.5 Two-Phase Properties
The vapor quality of a saturated liquid-vapor mixture is calculated based on enthalpy ratio as shown in Eq. (10).

\[
x = \frac{(H - H_l)}{(H_v - H_l)}
\]

(10)

For homogeneous mixture of liquid and vapor, the density, specific heat, and viscosity are computed from the following relation:

\[
\phi = (1 - x) \cdot \phi_l + x \cdot \phi_v
\]

(11)

where, \( \phi \) represents specific volume, specific heat or viscosity.

3.6 Heat Transfer
Each fluid lump is connected with a surface node as shown in Fig. 2. The energy conservation equation for the solid node is solved on the basis of thermal mass of surface node and convective heat transfer between solid to inside fluid. The energy conservation equation for the solid can be expressed as shown below:

\[
-Q_{s-f} = m \cdot C_p \cdot \frac{dT}{dt}
\]

(12)

where Qs-f is heat transfer from the wall to the inside fluid by convection as shown below:

\[
Q_{s-f} = h_c A_w \cdot (T_s - T_l)
\]

(13)

Convective heat transfer coefficient (hc) is calculated by different correlations according to the fluid state inside the flow line. Dittus-Boelter formulation [16] is used to calculate Qs-f for single phase fluid in turbulent flow as shown in Eq. (14) and (15).

\[
h_c = Nu \cdot \frac{k}{D}
\]

(14)

\[
Nu = 0.023 \cdot (Re)^{0.8} \cdot (Pr)^{0.4}
\]

(15)

Miropolskii’s correlations [17] are used to calculate heat transfer coefficient for two-phase homogeneous mixture as shown below.

\[
h_c = Nu \cdot \frac{k_v}{D}
\]

(16)

where

\[
Nu = 0.023 \cdot (Re_{mix})^{0.8} \cdot (Pr_v)^{0.4} \cdot (Y)
\]

(17)

\[
Re_{mix} = \left( \frac{\rho \cdot u \cdot D}{\mu_v} \right) \cdot \left[ x + (1 - x) \cdot \frac{\rho_v}{\rho_l} \right]
\]

(18)

\[
Pr_v = \frac{C_p \cdot \mu_v}{k_v}
\]

(19)

\[
Y = 1 - 0.1 \left( \frac{\rho_v}{\rho_l} - 1 \right)^{0.4} (1 - x)^{0.4}
\]

(20)

Chen’s correlation [18] is used to calculate heat transfer coefficient for nucleate boiling regime. Groeneveld’s correlation [14] is used for convective heat transfer coefficient in film boiling regime as shown in Eq. (21).

\[
h_c = 0.052 \frac{k_v}{D} \left\{ \bar{m} \cdot x \cdot \frac{D}{\mu_v} \cdot \left[ x + \frac{\rho_v}{\rho_l} (1 - x) \right] \right\}^{0.686} \cdot Pr_v^{1.26} \cdot Z^{-1.06}
\]

(21)
where

$$Z = 1 - 0.1 \left( \frac{p_1}{p_v} - 1 \right)^{0.4} (1 - x)^{0.4}$$

(22)

The pressure, mass flow rate, enthalpy and resident mass in internal fluid nodes are iteratively calculated by solving Eqs. (1), (2), (7), (9) and (11). A combination of the Newton-Raphson method and the successive substitution method are used to solve the set of equations.

4. Results

Detailed two-phase fluid dynamic formulations, described above, are used to simulate pressure surges in a liquid hydrogen cryogenic pipe occurring due to fluid hammer as well as liquid evaporation. One-dimensional lumped network model of the cryogenic line is developed. In this section, the numerical formulations are validated with well-known MOC procedure and subsequently, with cryogenic feedline flight component ground test data. Later, the section discusses the impact of various feedline parameters on pressure oscillation characteristics.

4.1. Validation with literature

The current formulations are validated with fluid-hammer studies carried out by Majumdar [19] using MOC technique. The studies were based on 121.9m long, 6.35mm diameter chilled pipe, carrying liquid oxygen at 34.47bar and 111.1K, and a flow rate of 45.35g/s being stopped using valve at downstream end of the pipe, whose closure time is 100ms. Pipe outlet pressure is taken as 31bar. Comparison of transient pressure profile between current model and MOC at valve inlet location is shown in figure 3.

![Figure 3](image_url)

Figure 3. Validation with literature based on MOC

It is evident from the above figure that both the amplitude and frequency of the pressure oscillation, simulated by the current model, are very closely matched with MOC procedure discussed in literature [19].

4.2. Validation with test data

Cryogenic feedlines and components used in flight are tested for leak checks and fault detections, prior to their installation in launch vehicle. One such test on liquid hydrogen feedline showed significant pressure oscillations immediately after closing the chill valve, which is characteristic to fluid hammer. The configuration for the test consists of source tank at 5bar pressure filled with 20.8K liquid hydrogen connected to cryogenic engine via 60m long 160mm diameter line followed by 1m long 90mm diameter line, 20mm orifice and 5m long 80mm diameter line. Prior to the test, chilldown of feedlines are carried out after which the vent valve located at engine inlet is closed linearly in 20ms resulting in pressure oscillations.

Analysis of the test using current formulation is carried out and pressure oscillations at various locations of the feedline are simulated. Comparison of simulated pressure profiles and test measurement data at orifice inlet and outlet locations are shown in figures 4(a) and 4(b) respectively.
It can be seen from the above figures that simulated liquid hydrogen pressures are in trend with measured test data. Frequency of pressure oscillation is well captured whereas a decent match for pressure amplitude is obtained.

4.3. Valve closure study
The computational model is validated using MOC procedure and test measurement data. Subsequently, effects of various parameters such as feedline geometry, flow rate and valve closure time on transient pressure surge pattern during valve closing operation are studied. The case study for 100m long 40mm diameter feedline carrying liquid hydrogen at a mass flow rate of 100g/s when valve closure time is 10ms, is considered as base case. Case studies are carried out with variations in any one parameter assuming other parameters remain same as the base case. Following section discusses the outcomes of the study.

4.3.1. Effect of line dimensions
Analyses are carried out to determine the effect of different line lengths (50m, 100m, 200m) and diameters (20mm, 40mm, 80mm) on transient pressure oscillation profile in cryogenic feedline carrying liquid hydrogen. Figure 5 shows pressure profile at valve inlet for different line lengths and figure 6 shows pressure profile at line mid-point for different line diameters.

It can be seen that for the base case (dashed lines in figures 5 and 6) the initial amplitude of oscillation is 0.91bar with frequency of 2.83Hz. Analysis shows that increasing the length of feedline would reduce the frequency of oscillation by the same factor but has no effect on initial oscillation amplitude (figure 5). Analysis also shows that increase in diameter of feedline decreases the initial amplitude of oscillation by the square of factor, without affecting its frequency (figure 6).
Characteristic time \( t_c \) of a line is defined as the time taken for pressure or sound waves to travel upstream and reflect back downstream of the line with length, \( L \).

\[
t_c = \frac{2L}{s}
\]  

(23)

The characteristic time of feedline in the current model is 177ms and the valve closure time \( t_v \) is 10ms. During valve closure, the change in fluid momentum exerts force at valve inlet, creating a high pressure peak in the time \( t_v \) (figure 5). This pressure information takes \( t_c \) time to reflect back, at which point there exist a maximum negative pressure difference between line inlet and outlet. Consequently, the fluid flows towards inlet side creating a low pressure peak at valve inlet in time \( t_c \). This pressure information again takes \( t_c \) time to reflect back, after which maximum pressure difference exist between inlet and outlet of the line. The cycle continues until the wave is dampened by dissipative forces such as viscous and frictional forces. Hence the frequency of oscillation is twice the characteristic time.

\[
\omega = \frac{1}{2t_c}
\]  

(24)

Thus, from equations 23, 24 and figure 5, it is evident that frequency of oscillation depends inversely on line length. It is noted from the dashed lines in figures 5 and 6 that the shapes of pressure oscillation pattern at valve inlet and line midpoint locations are different. At valve inlet, there is an instant rise in pressure since this location is the source of hammering effect. This pressure information travels upstream of the line at the speed of sound through the liquid hydrogen. Thus, the pressure peaks at midpoint of the line after one-fourth of \( t_c \) upto which fluid is still in motion at that location, resulting in staggered square pattern of pressure oscillations seen in figure 6.

From continuity equation shown in equation 25, assuming the flow is incompressible; velocity of flow \( (u) \) through the line is inversely proportional to line area or square of line diameter for same mass flow rate \( (\dot{m}) \). The initial amplitude of oscillation \( (\alpha) \) depends on rate of change of momentum of the fluid \( (mu) \) and valve closing time \( (\Delta t) \) as shown in equation 26.

\[
\dot{m} = \rho \cdot u \cdot \frac{\pi d^2}{4}
\]  

(25)

\[
\alpha \propto \frac{\Delta (mu)}{\Delta t} \propto \frac{\Delta (u)}{\Delta t}
\]  

(26)

Higher the line diameter, lower the flow velocity for same mass flow rate, resulting in lower initial pressure amplitude. Thus from equations 25, 26 and figure 6, it can be said that initial amplitude of pressure oscillation depends inversely on square of line diameter.

4.3.2. Effect of flow rate

Analysis is carried out with 40mm diameter 100m long line for various flow rates of liquid hydrogen to study the effect of change in flow rates through the line on pressure oscillation due to valve closure. Figure 7 shows the outcome of the study.

![Figure 7](image.png)

Figure 7. Simulated pressure at valve inlet for various initial fluid mass flow rates after valve closure

It can be seen from the figure 7 that pressure amplitude is directly proportional to initial fluid flow rate. This is expected from the discussion in section 4.3.1 that pressure amplitude depends on rate of change of fluid velocity and consequently on fluid mass flow rate as seen in equations 25 and 26.
4.3.3. **Effect of valve closure time**

Study is carried out to determine the effect of different closing times of valve \( t_v \) on pressure oscillation pattern. In the studies regarding line dimension effects, it is seen that the characteristic time of the feedline in the model is 177ms. Comparisons of closing timings were made between 10ms, 100ms, 200ms and 400ms. The results of the analysis are shown in figure 8 and figure 9.

![Figure 8](image1.png)  
**Figure 8.** Pressure pattern simulated after valve closure at valve inlet for valve closure time less than characteristic time

![Figure 9](image2.png)  
**Figure 9.** Pressure pattern simulated after valve closure at valve inlet for valve closure time greater than characteristic time

Figure 8 shows two valve closure timings \( t_v \) less than characteristic time \( t_c = 177\text{ms} \) and figure 9 shows two valve closure timings \( t_v \) greater than characteristic time \( t_c \). It can be seen from figure 8 that the amplitude of pressure oscillation remains same when \( t_v < t_c \). As discussed earlier, the force exerted by the fluid at valve inlet represents a pressure peak. Since pressure wave travels at speed of sound of the fluid, if \( t_v < t_c \), the valve comes to fully closed condition by the time the wave reflects back to its source. This condition represents maximum pressure peak as \( \Delta t \) in equation 26 equals \( t_c \). The time taken to reach the peak pressure is same as \( t_v \) which can be seen in figure 8.

If \( t_v > t_c \) as seen in figure 9, pressure peak value reduces as the outlet valve is still open by the time the pressure wave reflects back to its source, which reduces the rate of change of momentum imparted at the valve. In this case, \( \Delta t \) in equation 26 equals \( t_v \). Nonetheless, the peak pressure occurs at \( t_c \) as seen in figure 9. Thus, having valve closure time higher than the characteristic time of the fluid would significantly reduce the pressure amplitude due to fluid hammering effect.

5. **Conclusion**

Severe pressure oscillations are observed during sudden closing of a valve in a cryogenic feed system and prior knowledge of the magnitude of oscillation characteristics can help in prevention of major damages to the feed systems and respective components. A transient fluid dynamic model is developed to understand the fluid-hammer induced pressure oscillation characteristics of liquid hydrogen in cryogenic feedline, occurring when valves in the feedline are subject to sudden closing. Initially, the model is well validated with literature based on MOC simulations and with pressure measurement data from in-house experiment.

Subsequently, effects of various line dimensions, initial flow rates and valve closing durations on pressure oscillations during valve closing operation were analysed. Following are the salient deductions from the study.

- Transient analysis using a feedline of 100m length and 40mm diameter, carrying liquid hydrogen at 5bar pressure and 20K temperature shows that when a mass flow rate of 100g/s is stopped in 10ms, initial pressure amplitude of 0.91bar and oscillation frequency of 2.83Hz is generated.
- It is understood that the pressure amplitude depends largely on rate of change of fluid momentum, ie, change in velocity due to diameter variation or due to different initial mass flow rates. If valve...
Closing time is greater than characteristic time, the initial pressure amplitude also depends on valve closing time. Frequency of oscillation depends on length of the line carrying liquid hydrogen.

- The study concludes that higher line diameter, lower initial mass flow rate and slow valve closing would lower hammering effect resulting in lower pressure amplitude. It is also seen that longer lines would reduce the frequency of pressure oscillation.

6. Nomenclature

\[ A \] Line cross-sectional area (m\(^2\))

\[ A_w \] Line surface area connected to fluid (m\(^2\))

\[ C_p \] Specific heat (J/Kg-K)

\[ D \] Inner diameter (m)

\[ f \] Friction factor

\[ f_r \] Recoverable loss coefficient (1/kg-m)

\[ g \] Gravitational acceleration (m/s\(^2\))

\[ H \] Enthalpy (J/kg)

\[ h_c \] Heat transfer coefficient (W/m\(^2\)-K)

\[ h_{lg} \] Latent heat of vaporization (J/K)

\[ k \] Thermal conductivity (W/m-K)

\[ K_f \] Frictional resistance coefficient (1/kg-m)

\[ L \] Line length between two fluid control volumes (m)

\[ L_e \] Entrance length (m)

\[ m \] Resident mass (kg)

\[ m_\dot{m} \] Mass flow rate (kg/s)

\[ h_c \] Heat transfer coefficient (W/m\(^2\)-K)

\[ Nu \] Nusselt number

\[ Pr \] Prandtl number

\[ P \] Pressure (Pa)

\[ Q \] Heat transfer (W)

\[ R \] Gas constant (N-m/kg-K)

\[ Re \] Reynolds number

\[ s \] Sound speed (m/s)

\[ t \] Time (s)

\[ T \] Temperature (K)

\[ u \] Velocity (m/s)

\[ U \] Fluid node internal energy (J)

\[ V \] Volume (m\(^3\))

\[ x \] Vapour quality

\[ Y \] Liquid-vapour mixture correlation factor

\[ z \] Compressibility factor

Greek Symbols

\[ \alpha \] Initial amplitude of pressure oscillation

\[ \mu \] Dynamic viscosity (Pa-s)

\[ \varepsilon \] Surface roughness of pipe (m)

\[ \rho \] Density (kg/m\(^3\))

\[ \sigma \] Surface tension (Kg/s\(^2\))

Subscripts

\[ l \] Liquid

\[ v \] Vapor

\[ u \] Upstream

\[ d \] Downstream

\[ mix \] Liquid-vapour mixture

\[ s \] Solid

\[ f \] Fluid

7. References

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