Properly Calculate Relief Systems Steady and Dynamic Reaction Forces

An ioMosaic Corporation White Paper

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Properly Calculate Relief Systems Steady and Dynamic Reaction Forces

Pressure Relief Systems Practices

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July 20, 2020

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1 Introduction

Accurate estimates of fluid flow reaction forces are especially necessary for pressure relief systems. Substantial fluid flow reaction forces can be developed when relief systems actuate for both reactive and non-reactive systems. Specific relief systems scenarios where dynamic loads may be important include but are not limited to pressure relief caused by runaway reactions, loss of high-pressure/low-pressure interface, control valve failure, heat exchanger tube failure, etc.

Even if the relief requirements or required vent size are properly calculated, without adequate relief piping supports, the relief system will most likely fail when a demand is placed on it. The likelihood of failure of poorly supported relief systems increases as the systems are actuated more than once, i.e. reaction forces damage to supports can be cumulative.

Relief systems engineers are often challenged when developing reaction forces, especially for high pressure and/or emergency relief where dynamic loading becomes important. The recently published 2nd Edition of the CCPS Guidelines for Pressure Relief and Effluent Handling Systems [1] includes a detailed chapter on how to calculate reaction forces for different types of relief devices and relief configurations. This paper provides a quick reference/primer on how to properly calculate fluid flow reaction forces.

2 Reaction Forces

The rapid opening of a relief device or the sudden rupture of a heat exchanger tube can result in large and rapid changes in flow rate which can subject the relief piping systems to transient forces, transient impulses, and quasi-steady state forces. Depending on the piping layout, significant moments may be generated in the relief piping and associated equipment. The piping overall transient force can be represented in one dimension as a function of time:

$$F(t) = \underbrace{u_c\dot{M} + (P_c - P_a)\,A_c}_{\text{Thrust load},F_s} + \underbrace{\frac{\partial}{\partial t}\int \rho u A dx}_{\text{Wave load, }F_u} = F_s + F_u \tag{1}$$

where F_s is the steady state reaction force (steady state thrust load) applied to the piping supports, u_c is the velocity just inside the exit plane of the pipe, $\dot{M}$ is the mass flow rate, P_c is the absolute pressure just inside the exit plane of the pipe, A_c is the exit area of the pipe, and P_a is the ambient absolute pressure. The thrust load includes the momentum flux of the discharging fluid and the differential pressure at the exit. F_s is often considered for piping support design as the reaction force produced when a fluid is discharged from the end of a pipe to the atmosphere. Because closed piping systems under steady flow do not exert a reaction force onto their supports, F_s is the only steady state reaction force that is applied to the piping supports:

$$\boxed{F_s = u_c\dot{M} + P_cA_c - P_aA_c = u_c\dot{M} + (P_c - P_a)\,A_c} \tag{2}$$

F_u is the unsteady reaction force (wave load) applied to the piping supports and x is distance down the pipe segment. The wave load is the unsteady reaction force caused by the rate of fluid momentum change. The wave load approaches zero at steady state. The magnitude of F_u is proportional to the rate of change in the relief mass flow within the piping. Depending on the relief scenario and pipe segment length, the duration may be of the order of a few milliseconds to several seconds. The transient force is frequently neglected for gas phase relief but can be significant for liquid or two-phase relief:

$$F_u = \frac{\partial}{\partial t} \int \rho u A dx \tag{3}$$

This transient reaction force can also be approximated by:

$$F_u = \frac{d}{dt}\left(\rho u A L\right) = \frac{d}{dt}\left(\dot{M} L\right) \simeq L \frac{\dot{M}_2 - \dot{M}_1}{t_2 - t_1} \tag{4}$$

Where $\dot{M}$ is the mass flow rate, t is the arrival time, subscript 1 refers to the pipe inlet plane and subscript 2 refers to the pipe exit plane. Transient loads tend to be applied for short durations, tens of milliseconds to several seconds. In many cases the initial flow $\dot{M}_1$ is zero and t_1 is also equal to zero. It is recommended that the mass flow rate $\dot{M}$ be calculated without the influence of piping resistance (nozzle estimate) and at the maximum allowable pressure accumulation in the vessel. As a result,

$$\boxed{F_u = L \frac{\dot{M}}{\Delta t} = \frac{\dot{M}^2}{\rho A_c}} \tag{5}$$

When analyzing structures for short duration structural loads, it may be useful to include the impulse. The impulse is the product of the load application times the duration:

$$I = \int_{t_1}^{t_2} F_u dt \tag{6}$$

F_u typically decays over time after the sudden opening of a relief device or flow element. If we assume the decay occurs linearly (triangular shape), then:

$$I = \int_{t_1}^{t_2} F_u dt = \frac{F_u \Delta t}{2} \tag{7}$$

or

$$\boxed{\Delta t = 2 \frac{I}{F_u}} \tag{8}$$

Using the approximated expression of F_u above results in a simple equation for impulse:

$$I = L\left(\dot{M}_2 - \dot{M}_1\right) \tag{9}$$

If the flow changes between two steady state conditions, the net impulse that is applied to the piping supports is equal to the length of the pipe segment times the difference in the steady state flows. In many cases the initial flow $\dot{M}_1$ is zero and the impulse is equal to the segment length times the final mass flow rate. It is recommended that the mass flow rate $\dot{M}_2$ be calculated without the influence of piping resistance (nozzle estimate):

$$\boxed{I = L\dot{M}} \tag{10}$$

In addition to the overall transient force $F(t) = F_s + F_u$, relief piping can also experience an increase in the tension force within the piping, F_{PT} (see [1]). Usually, the tension force used in designing flanges and other joints is based upon the piping design pressure. The selected design pressure will typically have enough margin where any increased tension due to flow does not impact the design. High fluid velocities such as those encountered in relief systems applications can result in higher piping tension forces than those that are typically obtained from the operating pressure. The piping tension force can be calculated from:

$$\boxed{P_{PT} = \frac{F_{PT}}{A_p} = P_i + \frac{F_u}{A_p} - P_a} \tag{11}$$

where F_{PT} is the pipe tension force, P_i is the initial local pipe absolute pressure which is typically ambient pressure, P_a is the ambient absolute pressure and P_{PT} is the minimum design pressure for tension loads which is equal to F_{PT}/A_p or:

$$\boxed{\frac{P_{PT}}{P_i} = 1 - \frac{P_a}{P_i} + \frac{F_u}{P_i A_p}} \tag{12}$$

where F_u is the transient reaction force experienced upon opening of the pressure relief device defined earlier in equation 5. Usually other piping design considerations lead to higher pipe design pressure values than P_{PT}. The mass flow rate value used in equation 5 should be calculated at the maximum allowable pressure accumulation in the vessel.

3 The Use of Dynamic Load Factors

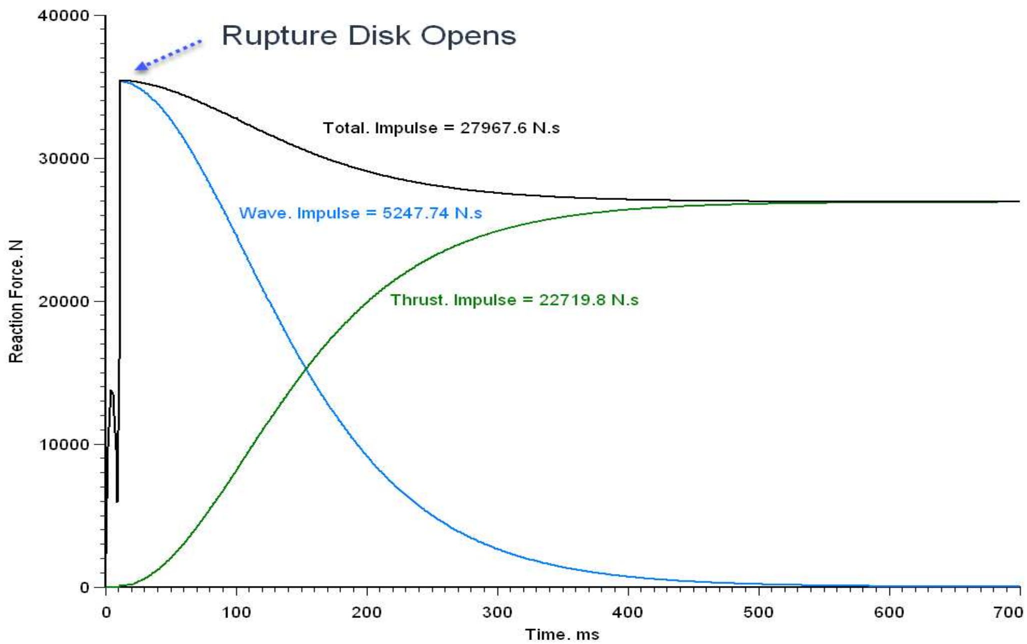
When a load is quickly applied to a structure, the structure vibrates similar to a mass being supported by a spring. This dynamic response results in instantaneous loads within the structure that are greater than the applied load. Because structures are usually analyzed using static models instead of using dynamic structural models, a dynamic load factor is typically used to relate the equivalent load to the applied load [1].

$$F_{eq} = DLF \times F_s \tag{13}$$

where F_{eq} is the he equivalent static load and DLF is the dynamic load factor. For loads that are applied quickly and that are of long duration the dynamic load factor varies between 1 and 2. A value of 2 is recommended. Please refer to the CCPS Guideline [1] for more specific information about recommended values for the dynamic load factors for pressure relief valves and rupture disks in gas, liquid, and two-phase service.

A load is considered to be applied quickly if the rate of application is short compared to the natural frequency of the structure. The application rate is considered to be slow if the time is long compared to the structural natural frequency. The same principles hold for the load duration. If the duration of the load is short, then the dynamic load factor can vary from 0 to 2. The use of the dynamic load factor is only appropriate when a static analysis is performed on the structure. If a dynamic structural analysis is to be done, then the dynamic load factor should not be used because

Figure 1: Reaction force components and associated impulse values as calculated by SuperChems Expert™



the analysis will include the effect of dynamic loading. When a time history pipe stress model is used, the DLF is captured by the participating elements and inertial response.

When using a dynamic load factor, the analysis is only valid for the single application of a load to a structure. If a load is applied repeatedly to a structure, then resonance may occur. A term from vibration theory that is used to relate the applied load to the equivalent load when resonance may occur is the magnification factor. Depending on the applied frequency, structural natural frequency, duration, and dampening, the magnification factor may be as large as 10 times or more.

4 Case Study - Rupture Disk in Liquid Service

We consider a 6 inch NPS (0.0186 m^2 flow area), 9 m long inlet relief line that contains water and is at an initial pressure of 1 bara (100,000 Pa) and room temperature. A 6 inch rupture disk with a Kr value of 1.5 and an opening time of 2 ms separates the inlet line from the discharge point. The stagnation pressure of the source increases from 1 bara to 20 bara over 10 ms. The discharge backpressure is constant at 1 bara. An equivalent discharge coefficient $C_d \simeq 0.58$ is calculated to account for the rupture disk Kr value and the inlet line entrance effects and frictional pressure loss.

Figure 1 shows the calculated reaction force components as a function of time using the detailed 1D dynamics model of SuperChems Expert . Impulse loads are also reported for a total duration of 1 second. The dynamics show that the mass flow rate reaches 583 kg/s at steady state (> 300 ms). We note from Figure 1 that the wave component decreases and tends to zero at steady state while the thrust component increases and remains constant at steady state.

It is very interesting to note that because the piping is short the estimate of wave impulse value using the simple form in equation 10 yields almost exactly the same answer as the detailed dynamics:

$$I = \dot{M}L = 583 \times 9 = 5,247 N.s \tag{14}$$

We can also approximate the duration of the wave loading by calculating the maximum value of F_u:

$$F_u = \frac{\dot{M}^2}{\rho A_c} = \frac{583^2}{1000 \times 0.58 \times 0.0186} = 31,506 \ \mathrm{N} \tag{15}$$

The duration of wave for a triangular shape equals:

$$\Delta t = 2\frac{I}{F_u} = 2\frac{5247}{31,506} = 0.333 \ \mathrm{s} \text{ or } 333 \ \mathrm{ms} \tag{16}$$

The value of 333 ms is very close to the duration of the wave load as shown in Figure 1. The minimum design pressure for tension loads P_{PT} for the downstream discharge piping (6 inch NPS) is given by:

$$\frac{P_{PT}}{P_i} = 1 - \frac{P_a}{P_i} + \frac{F_u}{P_i A_p} = 1 - 1 + \frac{31,506}{100,000 \times 0.0186} = 16.93 \tag{17}$$

5 Reaction Forces Following Heat Exchanger Tube Failure

Transient reaction forces on a liquid full shell side of a heat exchanger and its associated relief systems piping can be caused by the initial acceleration of the liquid in the shell following a sudden tube rupture containing high pressure gas. This is a form of the Joukowsky pressure which can be used to determine the value of the initial pressure surge. Piping and piping components upstream and downstream of the relief device are exposed to reaction forces due to the surge pressure upstream of the relief device, and also reaction forces due to the movement of the liquid slug entering the downstream piping once the relief device opens.

The duration of the transient reaction force on the relief systems piping caused by the acceleration of the initial liquid slug will depend on the relative locations of elbows and fittings and the shape of the pressure wave, i.e. whether it includes any reflections or not. Once the slug enters the piping downstream of the relief device, the duration of a reaction force is a function of the slug velocity.

Structural loads for three distinct phases of flow following a sudden tube rupture are normally considered:

a) Forces applied to the shell due to the initial acceleration of the liquid in the shell (upstream of the relief device),

b) Transient reaction forces applied to the shell side relief piping during initial liquid flow or pipe filling (downstream of the relief device), and

c) Quasi-steady reaction forces applied to the shell side relief piping during established initial liquid flow, followed by two-phase flow, and finally all gas flow.

5.1 Shell Side Liquid Acceleration Forces

Immediately after the tube failure, the shell experiences a short duration transient force associated with the initial acceleration of the liquid in the shell. This force is caused by the pressure discontinuity where the the passing of the pressure wavefront produces a force:

$$F_u = \rho_l \Delta u_{is} A_s c_{l_S} = \Delta P_{is} A_s \tag{18}$$

where Δu_{is} is the imposed initial liquid velocity, $\Delta P_{is} = \rho_l \Delta u_{is} c_{l_S}$ is the initial fluid impact induced step increase in pressure (commonly known as water hammer) from the Joukowsky equation, $P_{is} = P_r + \Delta P_{is}$ is the fluid induced initial pressure, P_r is the low pressure medium operating pressure, ρ_l is the low pressure medium liquid density, and c_{l_S} is the isentropic speed of sound [2] in the liquid adjusted for the presence of dissolved gas, the shell material of construction elasticity, and structural support conditions.

If the shell length is L_s and time required for the tube to completely rupture is t_{open}, the initial pressure wave will be completely contained within the shell for a period of:

$$t_u = \frac{L_s}{c_{l_S}} - t_{open} \tag{19}$$

The trailing edge of the pressure wave in the shell will be at $t_{open} \times c_{l_S}$. It is not always possible to locate a relief device next to where the tube rupture will happen and as a result the initial pressure can be reflected at closed ends and the reflection adds another ΔP_{is} to the load:

$$F_u = 2\rho_l \Delta u_{is} A_s c_{l_S} = 2\Delta P_{is} A_s \tag{20}$$

The Institute of Petroleum [3, 4] conducted several tests to better understand the impact of short duration high amplitude structural loads exhibited by the shell (liquid) following instantaneous high pressure (gas) tube ruptures. Figure 2 illustrates the pressure transient experienced on the shell side following a sudden tube rupture at a location close to the tube rupture point and a location that further from the tube rupture point. Note the time shift associated with the transit time of the pressure wave from near the failure point to the location that is further from it.

The heat exchanger used by the Institute of Petroleum was a 3.75 m long steel shell with an internal diameter of 0.74 m containing a standard tube bundle with internal tube diameters of 15 mm each. Although the measured peak pressures are high compared to the pressure rating of the shell, their durations were short on the order of 5 ms. These short durations are typically less than the natural

Figure 2: Typical shell side pressure transients following a sudden tube rupture [3, 4]

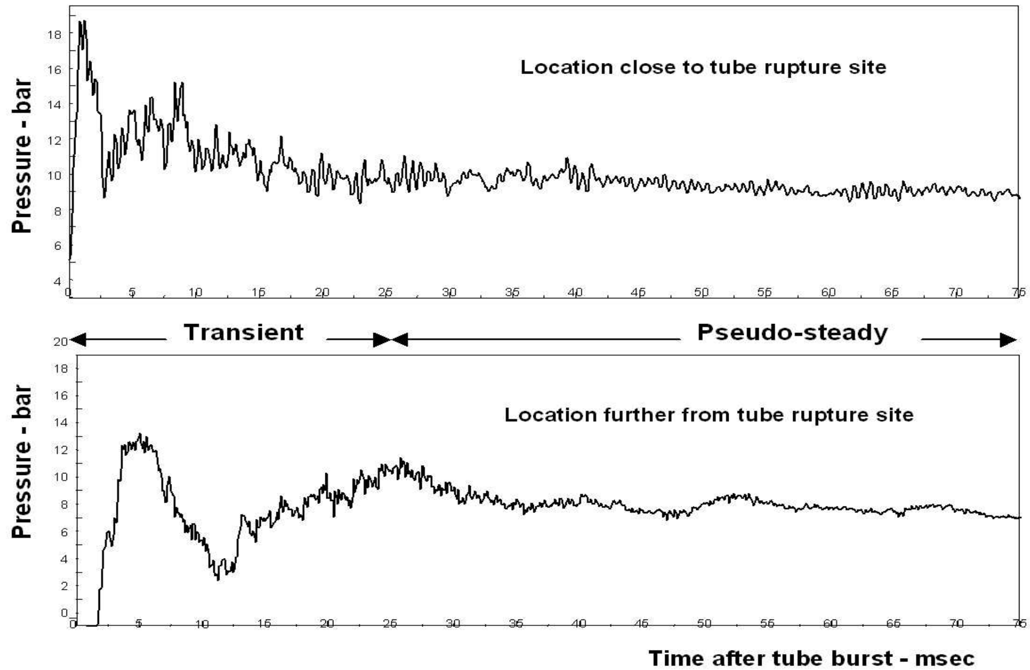
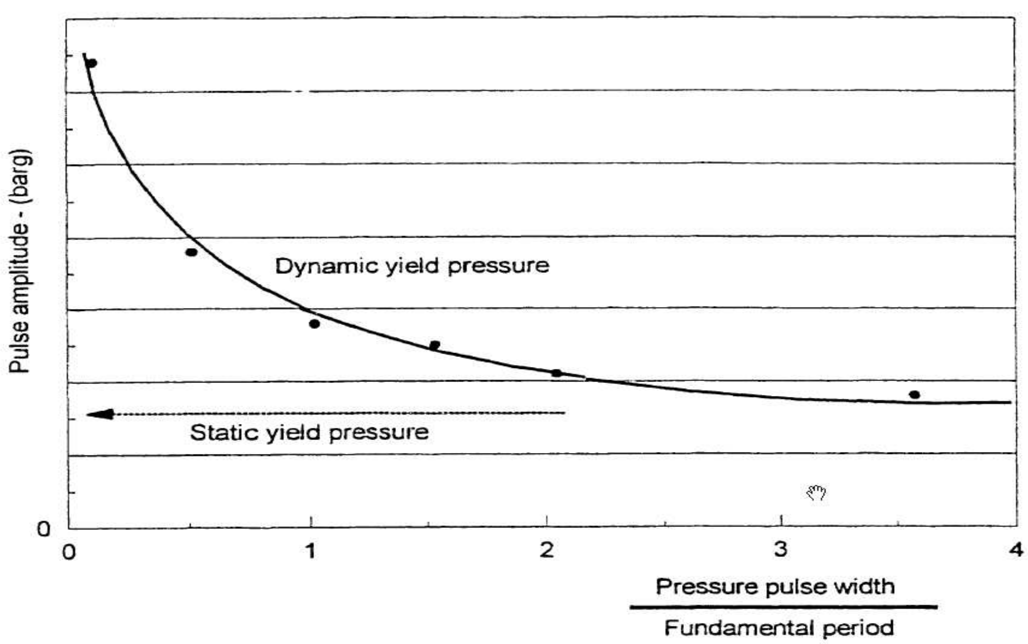


Figure 3: Dynamic shell failure limits as a function of pressure pulse durations [3, 4]



frequency of the shell, in this case 9.8 ms. Under such short duration dynamic loads, the shell can possibly tolerate more than the static yield limit as shown in Figure 3.

These measurements highlight the importance of the relief device opening times and relief device location relative to the failure point. The magnitude and duration of the pressure transients can be reduced with the use of multiple and optimally located relief devices with fast opening times.

5.1.1 Initially Liquid Full Relief Discharge Piping

It is undesirable to have relief discharge piping that is initially liquid filled because the sudden opening of a relief device or the failure of a tube under high pressure can create large transient reaction forces on the piping. If the opening of the relief device or tube rupture is short compared to the time required for the pressure wave to travel the pipe segment, then the maximum applied reaction force for a straight pipe segment can be approximated as the burst pressure times the flow area of the pipe segment. In the case of a tube rupture, the maximum applied reaction force for a straight pipe segment is given by Equation 18. If the opening of the relief device or tube rupture is long compared to the time required for the pressure wave to travel the pipe segment, then the applied reaction force to segment i is given by:

$$F_{u,i} = \Delta P_{is} A_i \frac{L_i}{L_T} \tag{21}$$

where $F_{u,i}$ is the reaction force applied to segment i, A_i is the pipe segment flow area, L_i is the length of segment i, and L_T is the total length of the relief discharge pipe. The reaction force can be significantly higher if resonance occurs between the structural natural frequency and the acoustic natural fequency of the liquid filled line.

The pressure wave will reflect back and forth in the line until steady state flow is achieved. The impulse can be calculated more easily using the steady state mass flow rate rather than the actual dynamic reaction force and often serves as a better design criterion. Frequently, the duration of the dynamic reaction force is so short that the structural support for the piping does not have enough time to respond to the loading. The impulse considers both the load and duration of the load. It is the integral of the applied reaction force over time. The impulse for pipe segment i is calculated as:

$$I_i = L_i \dot{M} \tag{22}$$

where $\dot{M}$ is the steady state flow rate.

5.2 Relief Piping Transient Liquid Reaction Forces

The shell side can either be protected with a fast acting rupture disk or a PRV. It has been shown through actual field tests that the opening time for PRVs can be as low as 5 to 7 milliseconds. This response time is enhanced by the high pressure water hammer wave as it passes the device. Initially, the relief discharge line is empty of liquid and full of air or methane or nitrogen. The transient reaction force is primarily due to the liquid filling the discharge pipe and the duration of this transient load is approximately equal to the time it takes the liquid to fill the discharge pipe. These transient reaction forces are particularly important for liquid and two phase discharges. Since the relief discharge piping is initially empty, the liquid or two-phase mass flow rate should be calculated assuming no piping resistance to flow.

Due to the uncertainty of the liquid-gas interface in the relief discharge piping, a dynamic load factor of 2 should be applied to the reaction forces calculated below for liquid flow. The dynamic load factor should not be applied if a dynamic structural analysis is being performed. The factor of 2 should only be used if a static analysis is being performed with the transient loads calculated below.

5.2.1 Rupture Disk

The transient reaction force for a rupture disk is proportional to the pressure difference of the rupture disk opening pressure and the relief piping backpressure:

$$F_u = \frac{\dot{M}^2}{\rho_l A_p} = \frac{2\Delta P A_p}{1 + K_{ent} + K_R} = \frac{2\left(P_{open} - P_a\right) A_p}{1 + K_{ent} + K_R} = 2\left(P_{open} - P_a\right) A_p C_d^2 \tag{23}$$

where A_p is the flow area of the pipe, $\dot{M}$ is the liquid flow rate through the rupture disk, K_{ent} in the number of velocity heads lost in the inlet piping between the shell and the rupture disk, K_R is the loss through the rupture disk assembly, P_{open} is the maximum pressure at which the rupture disk is expected to open which is not necessarily equal to the set point depending on the manufacturing range of the rupture disk, and P_a is the backpressure. Convervative estimates can be obtained by assuming K_{ent} and K_R are zero.

A relief discharge piping segment of length L_i is subjected to an impulse load equal to:

$$I = \dot{M} L_i = A_p L_i \sqrt{\frac{2\rho_l\left(P_{open} - P_a\right)}{1 + K_{ent} + K_R}} = C_d A_p L_i \sqrt{2\rho_l\left(P_{open} - P_a\right)} \tag{24}$$

The duration of this transient reaction force is calculated by:

$$t_u = 2\frac{I}{F_u} = 2\frac{\dot{M} L_i}{F_u} \tag{25}$$

5.2.2 Pressure Relief Valve

Transient reaction forces from liquid and two-phase flow from a PRV are calculated similarly to those from a rupture disk. However, in the case of a PRV the transient force may be significantly reduced if the time required to reach full flow from the PRV (from 0 % to 10 % overpressure for example) is longer than the time it takes to fill the pipe.

$$F_u = \frac{\dot{M}^2}{\rho_l A_p} \tag{26}$$

$$I = \dot{M} L_i \tag{27}$$

$$t_u = 2\frac{I}{F_u} = 2\frac{\dot{M} L_i}{F_u} \tag{28}$$

5.3 Relief Piping Steady State Liquid Reaction Forces

A steady state reaction force is exerted on the piping when liquid is discharged to the atmosphere. The reaction force is equivalent to liquid mass flow times liquid velocity because liquid flows are almost always subsonic:

$$F_s = u_l \dot{M} + \underbrace{(P_c - P_a)\, A_p}_{0} = u_l \dot{M} \tag{29}$$

A dynamic load factor of 2 is also applied to this reaction force when used for structural analysis without the dynamic components:

$$F_{eq} = DLF \times F_s = 2F_s \tag{30}$$

6 Case Study - High Pressure Ethylene Heat Exchanger Tube Failure

The concepts developed in this paper are illustrated using a Shell and Tube Heat Exchanger (STHE) of high pressure ethylene gas on the tube side and cooling water on the low pressure shell side. The high-pressure side maximum operating pressure of 2,500 psig exceeds the low pressure side design pressure of 285 psig. The gas in the tubes enters at 110 F and exits at 95 F. The cooling water on the shell side is at ambient temperature, assume 80 F. The shell side was hydro tested at 428 psig. If we correct that hydro test pressure to a maximum operating shell fluid temperature of 130 F, the hydro test pressure can be derated to 410 psig.

If tube failure occurs, it is expected at the backside of the tubesheet. As a result, the flow from the high-pressure side to the low-pressure side is modeled as (a) the flow through the U-tube and (b) the short tube stub remaining in the tubesheet. For use in the dynamic simulations, the flow through these paths is calculated as a function of built-up pressure in the low-pressure side. Other normal process flow in and out of the exchanger is assumed to stop. The tubes are 0.75 in outside diameter with a 0.109 in wall thickness leading to a tube internal diameter of 0.532 in. The tube stub remaining in the tubesheet is 4 inches long and the tube length remaining in the shell is 356 inches long. There are a total of 250 tubes.

The shell operates at 110 psig and has an inside diameter of 24.375 inches and a wall thickness of 0.25 in. The length of the shell is 360 inches. The portion of the flow area of the shell that is not occupied by tubes is 76.3 % or 356 in^2. The speed of sound in water at ambient conditions is

Table 1: Impact of gas molecular weight on shell side liquid displacement loads. $P_o = 2500$ psig, $T_o = 100$ F

	Ethylene	Methane	Hydrogen
Molecular Weight	28	16	2
P_{is}, reflected, psig	420	530	1016
$F_{u,r}$, reflected, kN	442	597	1290
Δu_{is}, m/s	1.05	1.41	3.05

specified at 1,480 m/s, uncorrected for the shell flexibility. Assume the shell steel properties are well represented by a density of 7,800 $\mathrm{kg/m^3}$ and a modulus of elasticity of 200 GPa.

The shell side is equipped with a relief system. The relief system is a 2 inch NPS rupture disk ($K_R = 0.24$) set at 255 psig. The relief discharge piping consists of one foot 2 inch NPS horizontal inlet line, followed by a 4 inch NPS vertical 10 ft discharge segment which is followed by by a 6 inch NPS horizontal 60 ft discharge segment. It is assumed that the rupture disk maximum opening pressure is 5 % more than the actual or 268 psig.

6.1 Shell Side Liquid Acceleration Reaction Force

We first calculate the initial incident pressure upon tube failure so that we can assess if the shell is likely to survive this initial pressure pulse or surge. First, we derate the shell side flow area by 10 % to add a reasonable safety margin to the incident pressure estimates as recommended by the Energy Institute, leading to a flow area on the shell side of 320 in^2.

The calculated reflected peak incident pressure value is 420 psig which is approximately equal to the hydrostatic test pressure of 410 psig for the shell adjusted for maximum operating temperature. The reflected pressure value has to be used because it is difficult to know the location of the tube rupture relative to the location of the pressure relief device. We also note that the duration of this initial pressure pulse will be close to 9 ms because the shell diameter to thickness ratio is approximately 100 leading to a 30 % reduction in the pipe/fluid speed of sound for the shell side. The associated force for the reflected pressure associated with the liquid displacement in the shell is very large at 442 kN (99.23 kips). It would also be useful to establish the natural frequency of the shell as well as the ratio of the duration of the calculated reflected pressure pulse at 8.9 ms to that as shown in Figure 3 for typical dynamic shell failure limits. If the ratio is less than 1, the shell can tolerate a higher dynamic short duration pressure pulse.

The incident pressure would be higher if the fluid on the tube side has a lower molecular weight and/or if the initial gas temperature is higher. For example, if the fluid in the tube was hydrogen at 2500 psig, and 100 F, the peak incident pressures would increase from 420 psig to 1016 psig as shown in Table 1. As a result, worst case liquid displacement conditions should be expected to occur with high pressure hydrogen at elevated temperatures striking low pressure water.

Additional reaction forces analysis is required in order to conduct a proper structural evaluation of

Figure 4: SuperChems Expert calculated initial liquid displacement incident pressure pulse on the shell side

Status	Normal return		
Solver	Use Expanding Fluid Method		
** INITIAL HIGH PRESSURE CONDITIONS			
Stream	TUBE - COMBINED FLOW		
Vapor mole fraction. %	79.938		
Pressure. psig	680.000		
Temperature. F	44.705		
Density. kg/m3	155.244		
Flow velocity. m/s	253.249		
Speed of sound. m/s	253.249		
Speed of sound piping correction factor	1.00		
** INITIAL LOW PRESSURE CONDITIONS			
Stream	SHELL SIDE		
Vapor mole fraction. %	0.000		
Pressure. psig	110.000		
Temperature. F	80.000		
Density. kg/m3	994.089		
Flow velocity. m/s	0.000		
Speed of sound. m/s	1028.746		
Speed of sound piping correction factor	0.69		
** FINAL SHOCKED FLUID CONDITIONS	ACTUAL	/LOW P	/HIGH P
Reflected shock pressure. psig	420.106	3.487	0.626
Shock pressure. psig	265.053	2.243	0.403
Shock temperature. F	-22.262	0.811	0.867
Shock density. kg/m3	52.246	0.053	0.337
Shock velocity. m/s	214.535	0.209	0.929
Shocked fluid velocity. m/s	1.045	0.001	0.005
** SHOCK LOADS	ACTUAL	REFLECTED	DURATION. ms
Upstream shock load. kN	0.0	0.0	42.15
Downstream shock load. kN	220.7	441.4	8.89

the shell and the relief systems piping:

A All liquid flow from the existing rupture disk without any discharge piping at the maximum opening pressure of the rupture disk, 5 % in this case. Use 10 % if this value is unknown or uncertain. This design case will be used to establish the initial dynamic reaction force the relief piping will be subjected to when the rupture disk first opens. This analysis can be performed using simple Bernoulli flow if detailed piping flow models are not available. The rupture disk resistance to flow, K_R, should be selected for the fluid phase that initially bursts the rupture disk (liquid in this case) even if during the transient (after bursting) the fluid phase changes to two-phase and then vapor. The flow geometry of the bursted disk causing the resistance to flow is different when opened by liquid from when it is opened by vapor or gas.

B Transient two-phase flow from the shell to establish the maximum transient pressure level in the shell and the associated quasi-steady state reaction forces that the relief piping will be subjected to. In general, this simulation is the only one needed since the initial flow is almost always 100 % liquid as the incoming high pressure gas pushes the liquid out of the shell through the relief piping.

C If the pressure in the shell exceeds a tolerable value, two-phase flow should be used to establish a relief device size such that the maximum pressure reached during quasi-dynamic flow is tolerable. The initial dynamic reaction forces for all liquid flow should be re-computed at this new size and so should the steady state reaction forces.

6.2 Relief Piping Transient Reaction Forces from Initial Liquid Flow

After the shell is exposed to the short duration liquid acceleration forces and pressure increases sufficiently to cause the opening of the relief device, the relief piping will be subjected to transient reaction forces. These forces are primarily caused by the discharge piping being filled as the liquid makes its way to the end of the discharge piping. One can use the simple equations provided earlier for either a rupture disk or pressure relief valve, or simply remove the discharge line from the actual relief line and estimate the peak flow at the maximum opening pressure of the relief device, 5 or 10 % for a rupture disk, and 10 % for a PRV.

If we do that for all liquid flow for the ethylene STHE example, we calculate a maximum liquid flow rate of 83.5 kg/s at a stagnation pressure of 268 psig. The discharge piping consists of two segments, a 10 ft, 4 inch NPS (12.730 in^2 flow area) vertical segment and a 60 ft, 6 inch NPS (28.890 in^2 flow area) horizontal segment. The transient liquid force applied to each segment is calculated below.

The calculated values for I and F_u below should be multiplied by the DLF of 2 if a static analysis is being performed. The use of the dynamic load factor is only appropriate when a static analysis is performed on the structure. If a dynamic structural analysis is to be done, then the dynamic load factor should not be used because the analysis will include the effect of dynamic loading.

6.2.1 10 ft Segment

$$
\begin{aligned}
F_u &= \frac{\dot{M}^2}{\rho_l A_p} = \frac{83.5^2}{1000 \times 12.73 \times 0.0254^2} = 849 \text{ N} \\
I &= \dot{M} L = 83.5 \times 10 \times 0.3048 = 254.5 \text{ N.s} \\
t_u &= 2\frac{I}{F_u} = 2\frac{254.5}{849} = 0.6 \text{ s}
\end{aligned}
$$

6.2.2 60 ft segment

$$
\begin{aligned}
F_u &= \frac{\dot{M}^2}{\rho_l A_p} = \frac{83.5^2}{1000 \times 28.890 \times 0.0254^2} = 374 \text{ N} \\
I &= \dot{M} L = 83.5 \times 60 \times 0.3048 = 1527 \text{ N.s} \\
t_u &= 2\frac{I}{F_u} = 2\frac{1527}{374} = 8.16 \text{ s}
\end{aligned}
$$

6.3 Reaction Forces from Quasi-Dynamic Two-Phase Flow

An important part of the evaluation deals with the time dependent increase of pressure in the shell as high pressure fluid is expanding into the shell and causing relief. Initially, all liquid flow occurs. This is followed by two phase flow and ultimately as the liquid is mostly depleted, single phase gas flow. This behavior is discussed by Melhem [5] and also verified using actual test data by the Energy Institute [6]. Simpson [7] assumed that the all liquid-venting assumption is conservative and that two-phase mixture behavior is intermediate between the gas-only and liquid-only venting. As shown in Figure 7, the all-liquid assumption is not conservative and the two-phase flow assumption leads to a larger relief requirement.

In general, the mechanical integrity of the shell is likely to be determined by this short quasi-dynamic period of time. Most existing STHE relief systems have been historically designed for all vapor venting. The relief areas will be undersized for two-phase venting. The purpose of this quasi-dynamic analysis is to determine if the stress in the shell will be high enough to cause deformation of the shell considering the low frequency of the initiating scenario.

A first step in this analysis is to confirm the pressure rating of the shell and to provide a failure stress criteria as a function of metal temperature. For this example, the shell pressure rating is confirmed by SuperChems Expert as shown in Figure 5 for the material of construction selected. The ultimate tensile strength is scaled by 2/3 to allow for uncertainties in material properties, corrosion, etc. If the built up internal shell stress exceeds this criteria at the temperature of interest, then the relief device/system is not adequate for this low frequency scenario and should not be tolerated. If on the other hand, the internal stress reached under two-phase venting is below this limit, an operator can

Figure 5: SuperChems Expert verification of shell design rating

Verify Vessel Design Ratings

	A	B	C	D	E	F	G
1	Vessel Name	HEAT EXCHANGER					
2	Vessel Type	Vertical Cylindrical; User defined heads					
3	Vessel Top Head Type	Flat					
4	Vessel Bottom Head Type	Flat					
5							
6	** METERIAL OF CONSTRUCTION SPECIFICATIONS. OBTAIN THIS INFORMATION FROM ASME TABLES						
7	Actual Material of Construction Description	STEEL					
8	Databank material of construction used for thermal	STEEL	STEEL				
9							
10	Specification	A 312					
11	P-No, G-No, or S-No	P-8					
12	Product Form	Seamless Pipe					
13	Grade	TP304L					
14							
15	Note 1	Enter note 1 here					
16	Note 2	Get allowable stress data from ASME tables. this data is for seamless pipe and is used as					
17							
18	Minimum Temperature. F	-425.00					
19	Minimum yield strength. psia	25000.00					
20	Minimum tensile strength. psia	70000.00					
21							
22	Allowable Stress Basis	ASME					
23	Allowable Stress Temperature. F	-425.00	100.00	200.00	300.00	400.00	500.00
24	Allowable Stress. psia	16700.00	16700.00	16700.00	16700.00	15800.00	14800.00
25							
26	Failure Stress Basis	2/3 UTS					
27	Failure Stress Temperature. F	-425.00	100.00	200.00	300.00	400.00	500.00
28	Failure Stress. psia	46666.67	46666.67	46666.67	46666.67	44151.70	41357.29
29							
30	** CALCULATED VESSEL PRESSURE RATING DATA BASED ON USER SPECIFIED WALL THICKNESS						
31	Shell Wall Thickness with Corrosion Allowance. in	0.2500					
32	Inside Radius. in	12.1875					
33	Corrosion Allowance. in	0.0000					
34	Joint Efficiency. 0 to 1	0.85					
35							
36	Maximum Allowable Temperature. F	-20.00	100.00	150.00	200.00	250.00	300.00
37	Maximum Allowable Shell Internal Pressure. psig	287.64	287.64	287.64	287.64	287.64	287.64
38							

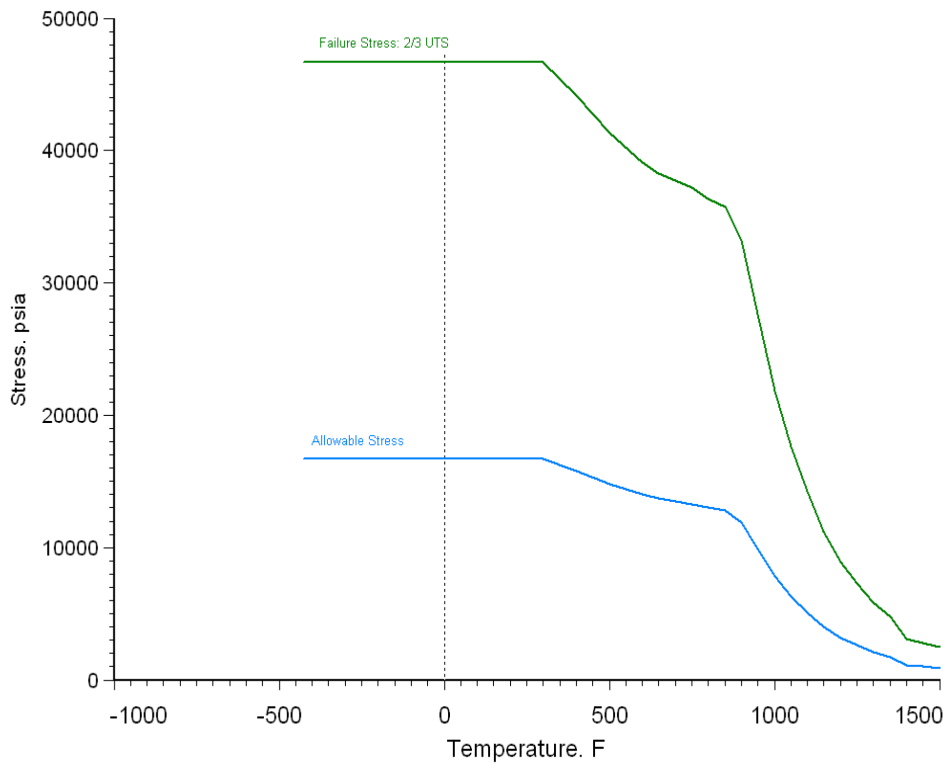
elect to tolerate the risk of deforming the shell but not failing it at the frequency of the tube failure because it is low enough.

The failure limit decreases with increasing metal temperature as shown in Figure 6 although this is not an issue for this example. This type of criteria is often used in conjunction with dynamic simulation to assess the mechanical integrity of vessels under fire exposure as shown by Melhem [8].

The SuperChems Expert dynamic simulation considers the flow from both ends of the ruptured tube as a function of shell backpressure and performs detailed time dependent mass and energy balances as well as physical and chemical equilibria to determine pressure, temperature, composition, reaction forces, etc. for the shell as a function of time. For illustration purposes, we calculate the time dependent pressure in the shell and the reaction forces at the end of the relief discharge piping for both all liquid flow and two-phase flow. We note that two-phase flow will be homogeneous and there will be no vapor/liquid disengagement because the geometry of most heat exchangers is not suitable for vapor/liquid disengagement. Initially the two-phase flow will be liquid rich as the shell is essentially liquid full. Figure 7 illustrates the pressure history in the shell. The venting is performed through the actual relief piping isometric consisting of the inlet line, rupture disk, and relief discharge piping at every time step. We note that the shell is emptied much faster with all liquid flow, and the pressure is higher with two-phase flow.

The quasi-steady state reaction forces will be higher for two-phase flow in this case because the pressure reached in the shell is higher. Note that SuperChems Expert calculates a time dependent flow impulse ($mu + PA$) at every piping axial location. In order to get the actual reaction force

Figure 6: Shell failure stress as a function of metal temperature



imparted to the entire relief piping, $P_a A$ must be subtracted from the flow impulse at the exit plane of the discharge piping. For liquid and/or sub-sonic flow the net reaction force will be equal to mu_e and for sonic flow where choking occurs it will be equal to $mu_e + (P_e - P_a)A_e$. This is illustrated in Figure 8.

The relief piping will be subjected to a peak flow impulse of approximately 11.6 kN as shown in Figure 8. However, the exit pressure is choked at 30.2 psig and the exit flow area of the 6 inch pipe is 28.890 in^2. A value of 5.75 kN would need to be subtracted from the 11.6 kN flow impulse to yield a peak quasi steady reaction force of 5.83 kN. There are utility scripts available to extract this data automatically in SuperChems Expert . By comparison,

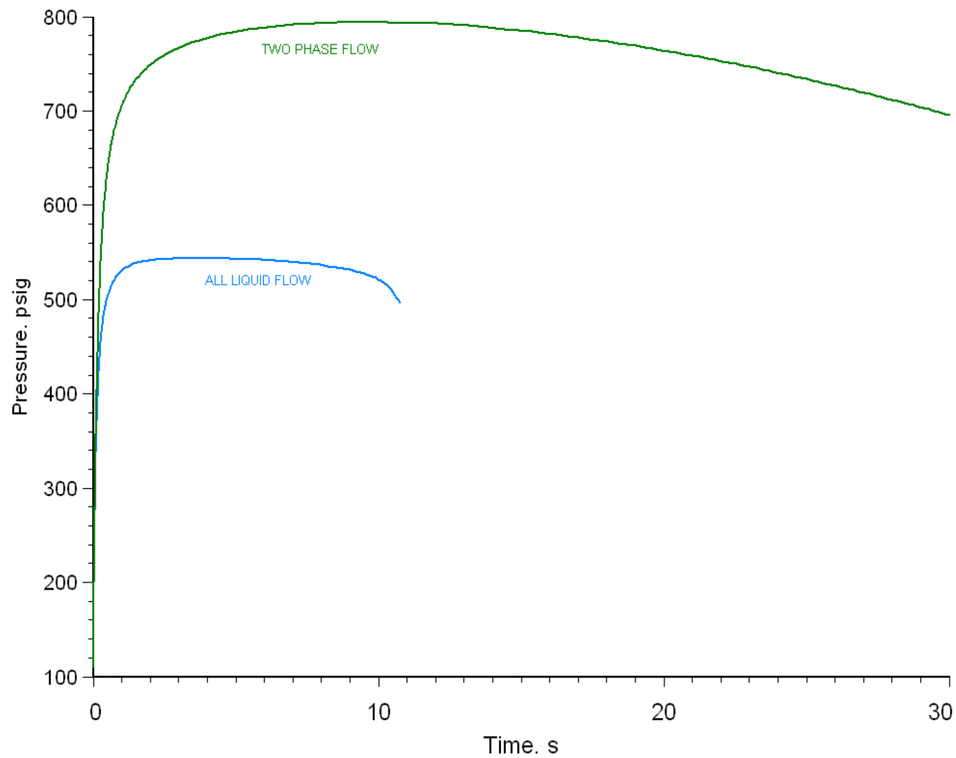
A The reflected liquid displacement shell force was calculated at 441.5 kN for 8.9 milliseconds, or as an impulse of 3925 N.s

B The transient liquid reaction force (without the dynamic load factor) applied to the 6 inch discharge segment of the relief piping was calculated to be 0.37 kN for 4.08 seconds or as an impulse of 1525 N.s

C The quasi-steady reaction force (without the dynamic load factor) during two-phase venting was calculated at 5.83 kN for 30+ seconds or as an impulse of 174,900 N.s

The relief piping must be analyzed for structural integrity of the supports using the right tools for structural analysis.

Figure 7: Shell pressure history following tube rupture for all liquid and two-phase flow



6.4 Required Rupture Disk Size

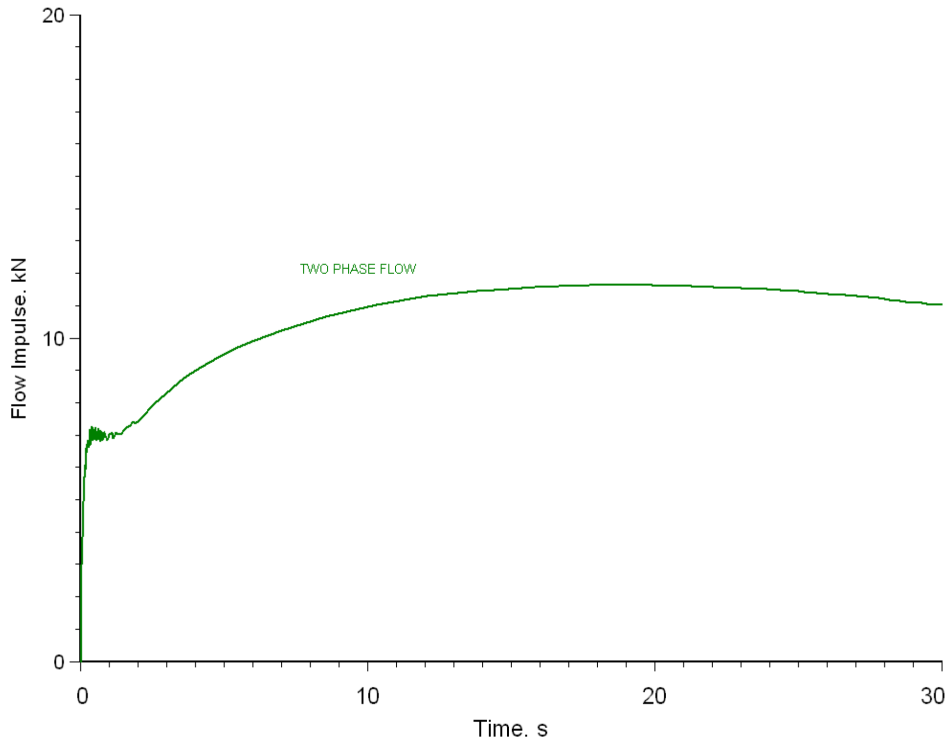
Although the pressure exceeds the corrected hydrostatic pressure in the shell, this does not mean that the shell is likely to fail. As shown in Figure 9 the built up internal stress in the shell is below the established failure stress based on 2/3 UTS. Although the shell is likely to get deformed, the shell is not likely to fail. As indicated earlier, this might be an outcome that can be tolerated if the tube failure frequency is low enough or the risk is low enough.

7 Conclusions

A pressure relief systems evaluation is deficient [1] or at best incomplete without the proper assessment of dynamic and steady state reaction forces. SuperChems Expert includes detailed tools for the estimation of reaction forces associated with fluid acceleration, quasi-dynamic single and multiphase flow from vessels, and one-dimensional transient analysis for single and multiphase flow from complex piping configurations. These tools are useful for the proper design of relief and vent containment systems, piping and piping supports, as well as the analysis of pressure relief valve stability.

[1] This information is required process safety information (PSI) for PSM regulated facilities in the United States

Figure 8: Relief piping flow impulse at the exit plane for two-phase flow



8 How can we help?

In addition to our deep experience in the conduct of large-scale site wide relief systems evaluations by both static and dynamic methods, we understand the many non-technical and subtle aspects of compliance and legal requirements. When you work with ioMosaic you have a trusted partner that you can rely on for assistance and support with the lifecycle costs of relief systems to achieve optimal risk reduction and compliance that you can evergreen. We invite you to connect the dots with ioMosaic.

Figure 9: Shell internal stress vs. failure stress at 2/3 UTS

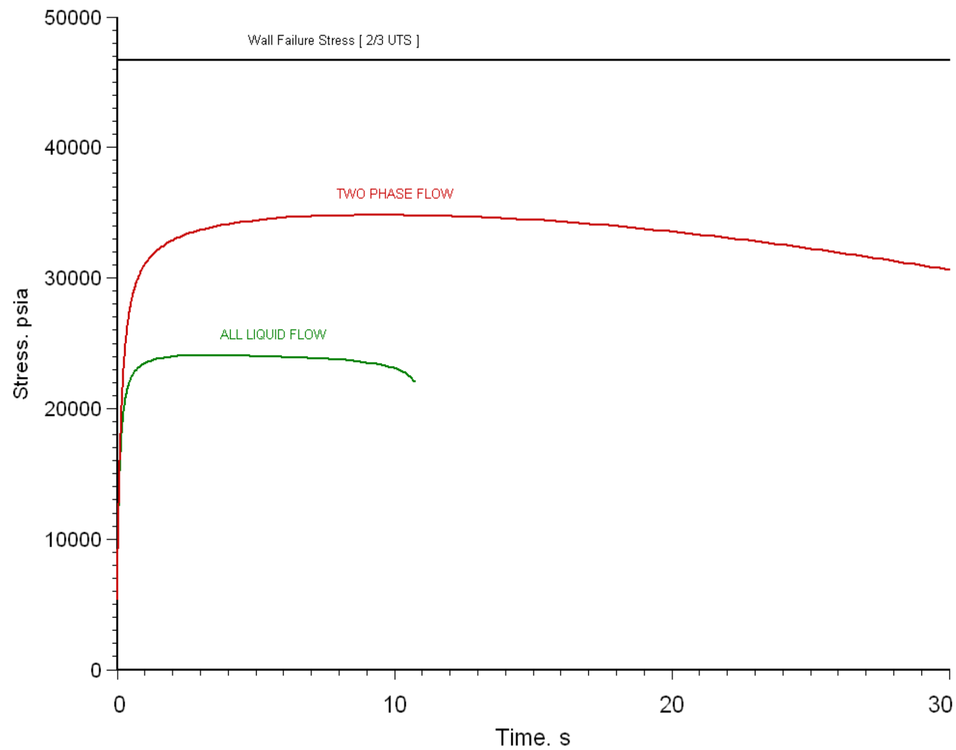


Figure 10: Connect the dots with ioMosaic

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About the Author

Dr. Melhem is an internationally known pressure relief and flare systems, chemical reaction systems, process safety, and risk analysis expert. In this regard he has provided consulting, design services, expert testimony, incident investigation, and incident reconstruction for a large number of clients. Since 1988, he has conducted and participated in numerous studies focused on the risks associated with process industries fixed facilities, facility siting, business interruption, and transportation.

Prior to founding ioMosaic Corporation, Dr. Melhem was president of Pyxsys Corporation; a technology subsidiary of Arthur D. Little Inc. Prior to Pyxsys and during his twelve years tenure at Arthur D. Little, Dr. Melhem was a vice president of Arthur D. Little and managing director of its Global Safety and Risk Management Practice and Process Safety and Reaction Engineering Laboratories.

Dr. Melhem holds a Ph.D. and an M.S. in Chemical Engineering, as well as a B.S. in Chemical Engineering with a minor in Industrial Engineering, all from Northeastern University. In addition, he has completed executive training in the areas of Finance and Strategic Sales Management at the Harvard Business School. Dr. Melhem is a Fellow of the American Institute of Chemical Engineers (AIChE) and Vice Chair of the AIChE Design Institute for Emergency Relief Systems (DiERS).

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Through innovation and dedication to continual improvement, ioMosaic has become a leading provider of integrated process safety and risk management solutions. ioMosaic has expertise in a wide variety of areas, including pressure relief systems design, process safety management, expert litigation support, laboratory services, training and software development.

As a certified ISO 9001:2015 Quality Management System (QMS) company, ioMosaic offers integrated process safety and risk management services to help you manage and reduce episodic risk. Because when safety, efficiency, and compliance are improved, you can sleep better at night. Our extensive expertise allows us the flexibility, resources, and capabilities to determine what you need to reduce and manage episodic risk, maintain compliance, and prevent injuries and catastrophic incidents.

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