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TRANSIENT EVENTS IN LIQUID PIPING SYSTEMS

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Mr. Fierro has experience in the fields of mechanical vibrations, finite element analysis, acoustics, and compressor and piping system design. He performs mechanical system analyses with the aid of ANSYS and CAESAR II to predict vibrational mode shapes and frequencies, vibration amplitudes, and related dynamic and thermal stresses. He also performs acoustic analyses of complex piping systems to optimize the acoustical responses to minimize adverse pulsation effects on compressor performance, system pressure drop, and piping vibration and related stresses. These analyses have been performed on existing and planned compressor installations.

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ABSTRACT

Pump piping systems are frequently subject to transient events such as slugs, water hammer, and cavitation that can create high amplitude forces and pressure spikes. Liquid systems are more susceptible to damaging forces during transients than gas systems due to the high density and incompressibility of the operating fluid. These events often result in high vibrations--and sometimes failures--that can be prevented in the design process or resolved in the field using a multi-angle approach. This lecture will discuss three case studies showing the need for using transient analysis of the flow in combination with mechanical and structural analyses and/or a sensitivity analyses to resolve on-site problems and predict the likelihood of failures in the design stage.

INTRODUCTION

Pump systems are rarely analyzed for transient events as these can be dependent on factors that are not always known or finalized in the design stage or change during the lifetime of the system. Dependent variables include the characteristics of the fluid, the piping geometry, pressure reducing valves, operating conditions and scenarios and flow rates. Various modeling tools exist to provide reliable predictions and cost-effective solutions for transient vibration or pulsation problems, but often multiple types of analyses must be used in combination to evaluate the system as a whole or solve complex problems.

Three different case studies will be presented in this paper as problems existing on site that required a combination of analyses types to solve. The traditional method of analyzing transient events in liquid systems is to only use a commercially available software for hydraulic analysis (such as Synergi Pipeline Simulator or Aspen HYSYS); however, often these transient fluid models need to be combined with mechanical or thermal stress analyses; dynamic pressure versus time inputs, and sensitivity studies or field data evaluations to be effective in resolving problems and ensuring pipeline integrity.

CASE STUDY 1—Seawater Transient Pressure Wave

An offshore pump piping system experienced a backwards failure of a pressure safety rupture disk resulting in a pressure wave through the seawater piping. This caused a piping failure resulting in a loss of fluid as well as extensive downtime. After the first piping failure was repaired, it was determined that a much larger pressure wave could occur due to a heat exchanger tube rupture, and further analysis was needed to ensure the piping and support system was sufficiently robust to withstand additional pressure waves without failures. The two primary causes for failures in this system would be due to over-pressurization and/or overstress. Therefore, to evaluate the piping for all aspects of failure, three different modeling tools were combined to calculate the pressure wave, the transient pressures in the piping system, and the pipe stress.

To calculate the pressure wave resulting from a tube rupture, a finite difference code was developed to perform a transient analysis. The finite difference program models the change in pressure in a liquid filled container subject to a constant gas inflow and a time varying outflow. This code was used to produce the pressure change in the heat exchanger for a tube rupture case. The heat exchanger was modeled with the shell, tube, and relief devices varying accordingly. The liquid flow rate out of the heat exchanger ($v_{out,line}$) was dependent on the differential pressure between the heat exchanger and the return line. The program modeled liquid flow exiting the system through the seawater return line, as well as liquid exiting through the relief device, once opened ($v_{out,RD}$). The relief device opened once the pressure at the relief device location reached or exceeded the set relief pressure. Depending on the distance, the pressure difference between the heat exchanger and the relief device may be significant. The required time for the pressure wave to propagate from the heat exchanger to the rupture disk was found using a speed-of-sound calculation. This information was used by the code to burst the rupture disk at the correct time.

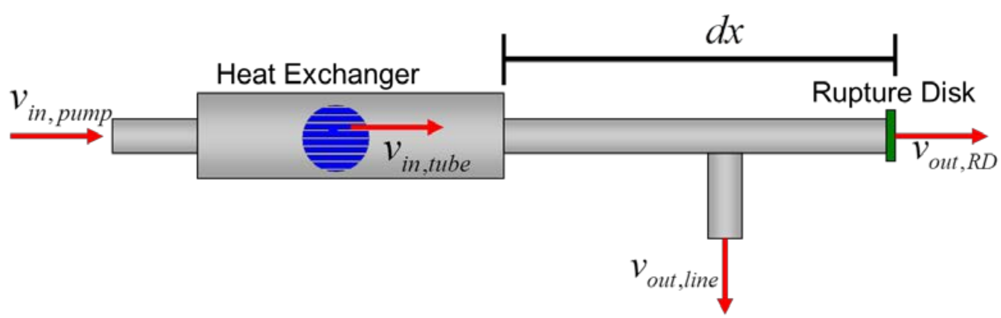

Figure 1: Visual Representation of the Heat Exchanger Pressure Wave Calculation for Case Study 1

A mass balance was performed of the mass entering and exiting the heat exchanger. The change in mass in the volume results in a change in pressure. The pressure wave in the heat exchanger is then calculated for the first few milliseconds until the pressure in the system is stable. The code outputs a pressure versus time plot at the heat exchanger that can be used as an input file in a transient fluid model. Sensitivity studies were performed to investigate the effect of the upstream pressure, opening time, distance of rupture disk to heat exchanger, and set rupture pressure.

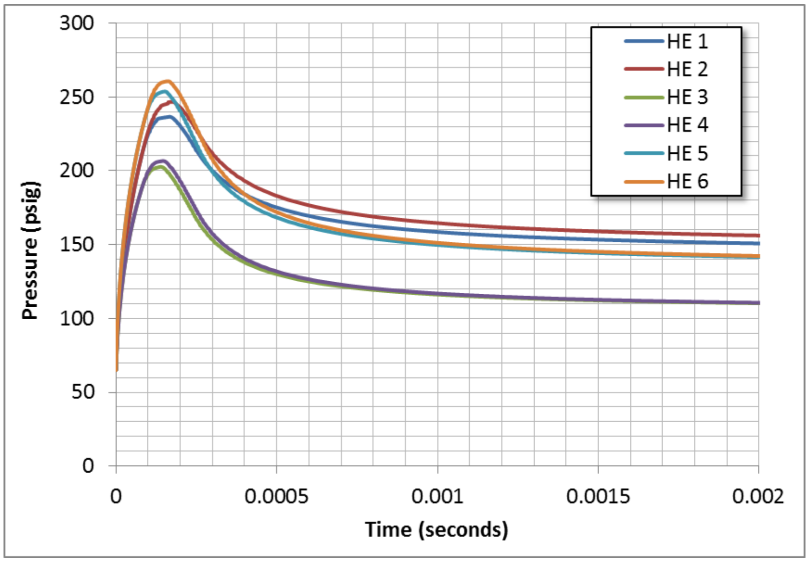

Figure 2: Predicted Pressure Waves in All Six Heat Exchangers

A pipe model of the system was developed using commercially available transient and steady-state hydraulic modeling software, Synergi Pipeline Simulator (SPS), to evaluate the effects of the pressure wave throughout the piping system. SPS reads text files containing detailed data that represents the pipeline. It can model operating characteristics of proposed pipeline configurations and predict the outcome of various control strategies for operating scenarios such as pipe rupture, equipment failure, or other upset conditions using a simulator based on the method-of-characteristics.

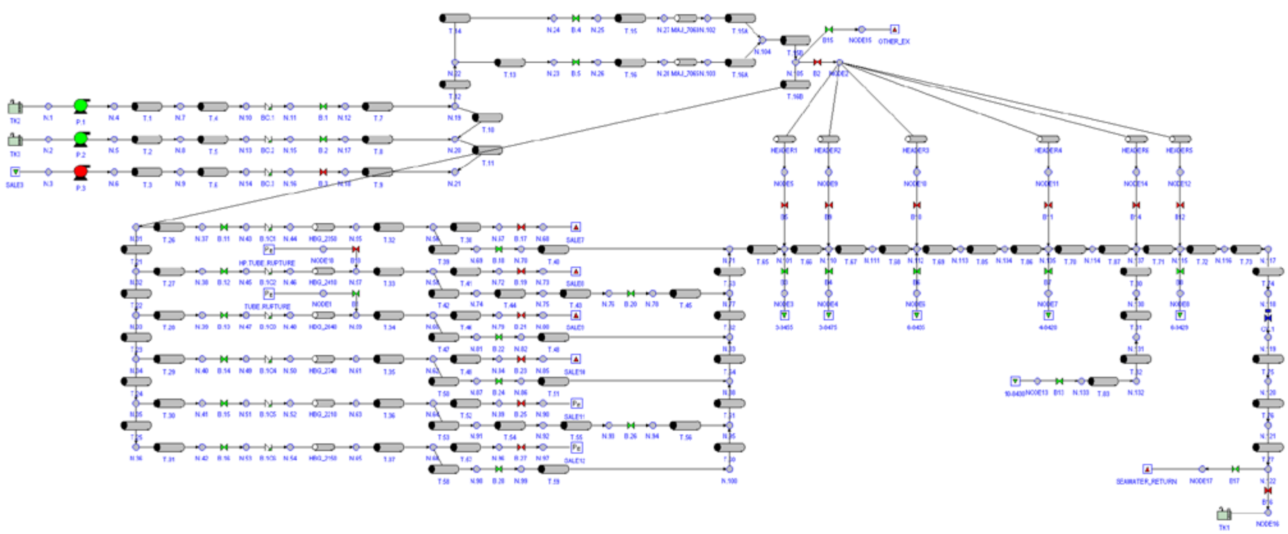

Figure 3: SPS Model of the Seawater Piping

The pressure wave from the finite difference code was used to induce a transient wave throughout the piping system. Pressure and flow transients resulting from each case were collected and analyzed to determine if the piping was safe from over-pressurization. The model included the piping from the suction side of the seawater pumps to the seawater return line caisson. A pressure boundary condition was established on the suction side of the pumps and a flow boundary condition at the seawater caisson. The three seawater pumps were also modeled in detail.

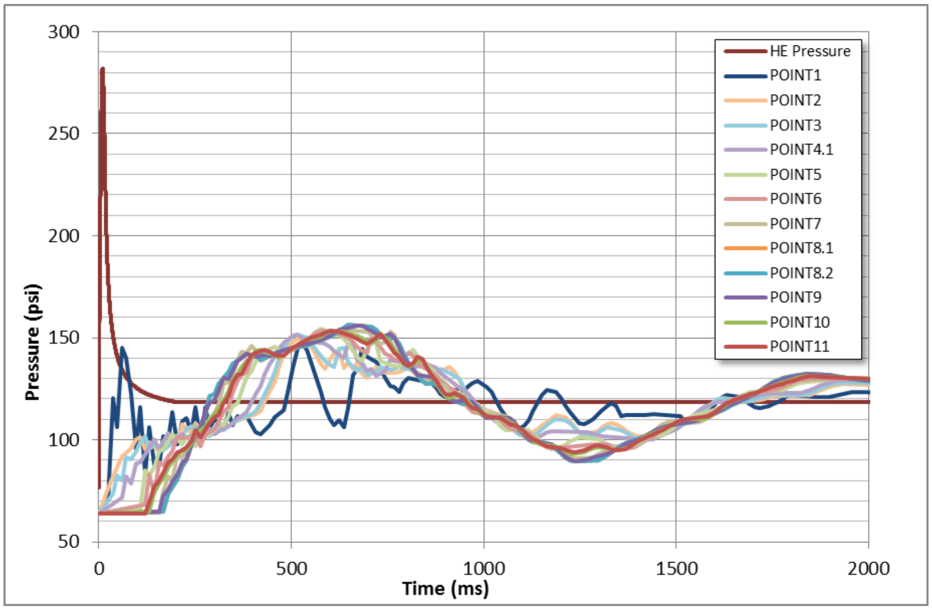

Figure 4: Pressure Waves at Different Points on the Seawater Return Line

To evaluate the potential for overstress, a PC-based pipe stress analysis software program (Caesar II) was used. This software package is an engineering tool used in the mechanical design and analysis of piping systems. A model of the piping system was created using simple beam elements and defines the loading conditions imposed on the system. With this input, the stress analysis program produces results in the form of displacements, loads, and stresses throughout the system. Additionally, the results are compared to limits specified by recognized codes and standards.

The piping system model began at the heat exchanger discharge flange for each of the six heat exchangers being studied in detail, and ended upstream of the seawater caisson. All piping attached to the seawater return line main header was modeled at least to the first restraint, to take into account its resistance on the system.

Stiffness values for each restraint location were chosen based on the type of restraint present at that location. In order to simulate the transient event, all locations where mechanical coupling was likely (e.g., pipe elbows, tees, blind flanges, etc.) were identified. A free body diagram was then created to show the X, Y, and Z components of the forces that would be acting upon the system during a transient event.

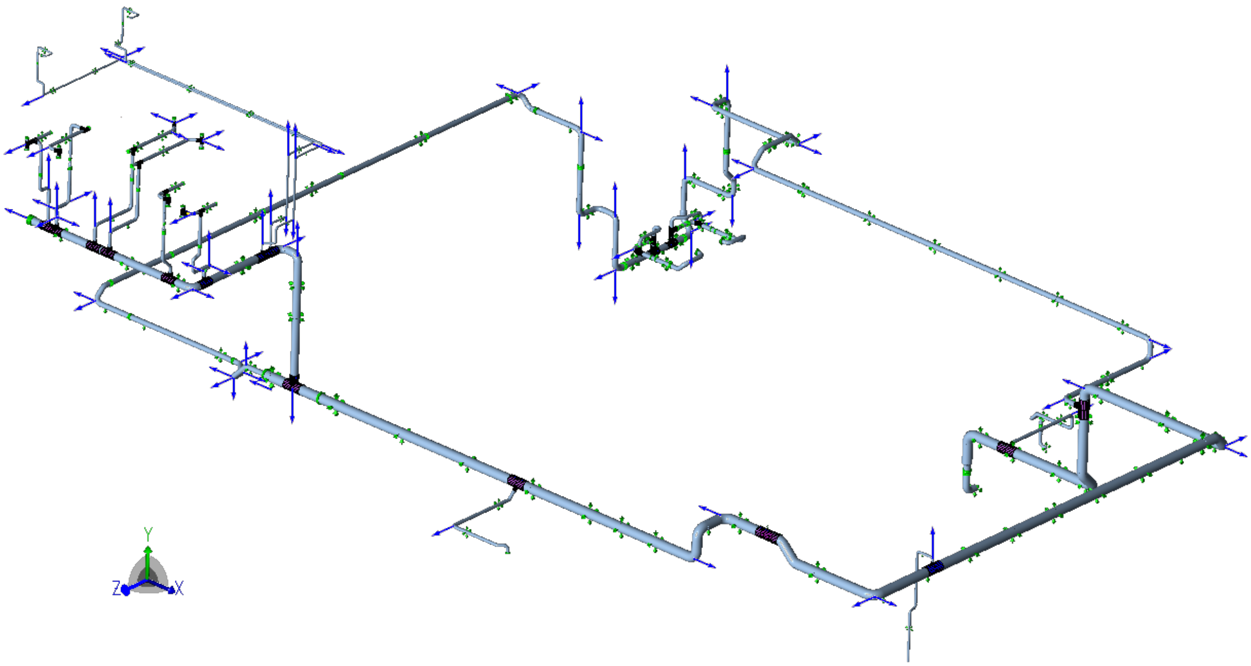

Figure 5: Forces on Piping Due to Pressure Wave

After time versus force loading data were collected from the SPS hydraulic model, the free body diagram was used to input the loading data into the pipe stress model Dynamic Field in all appropriate locations and directions. This is known as a Time-History-based dynamic analysis. This method uses numeric integration of the dynamic equations of motion to simulate the system response throughout the load duration.

The output processor provided piping stresses, forces, moments, and displacements caused by the transient event. It also contained an Animation Module that allowed viewing of a simulation of the system response to the force-time profile.

A series of sensitivity studies were performed to develop modifications to reduce predicted stress in the piping including varying the following:

- rupture disk sizes
- rupture disk set burst pressure
- rupture disk distance from heat exchanger
- heat exchanger operating pressures
- number of pipe restraints

These parameters were varied to help identify the most cost effective and reasonable modifications to ensure safe and reliable operation during a tube rupture event. For each case where the rupture disk or heat exchanger parameters were modified, all three analyses were repeated.

CASE STUDY 2—Valve Internal Corruption

A seawater injection line, Line A, experienced excessive vibration leading to the development of fatigue cracking on small-bore connections as well as producing a distinct noise. A vibration survey of the piping was performed prior to the analyses with excessively high vibration recorded at 158 Hz and significant vibration recorded at 250 Hz. The first steps taken to resolve the problem was to add supports to increase the mechanical natural frequencies of the piping; however this did not resolve the noise and vibration problem. Another similar line, Line B, operating in similar service with an identical control valve did not have any of these problems. The most likely source of these problems was the control valve; however, before incurring the cost and downtime of replacing the valve, analysis was needed to evaluate if there were other sources of the vibration and noise problem as well as to determine if there was a clear indication of valve damage.

Given the piping geometry and support system as well as the fluid operating conditions, it was concluded that the likely sources of the noise and vibration problems were either vortex-shedding of the flow, cavitation, or damage to the control valve trim. This requires combining multiple types of analyses to analyze the system for a range of different types of problems. Therefore, the scope of work included a fluid flow analysis using SPS, a flow-induced vibration (FIV or Strouhal) analysis, and a flow-induced turbulence analysis with the purpose of evaluating the acoustic excitation and natural frequencies of the piping system, the mechanical natural frequencies, the fluid phase throughout the system, and the performance of the control valves. An acoustic schematic of the layout showing the control valve and high vibration areas is shown in Figure 6.

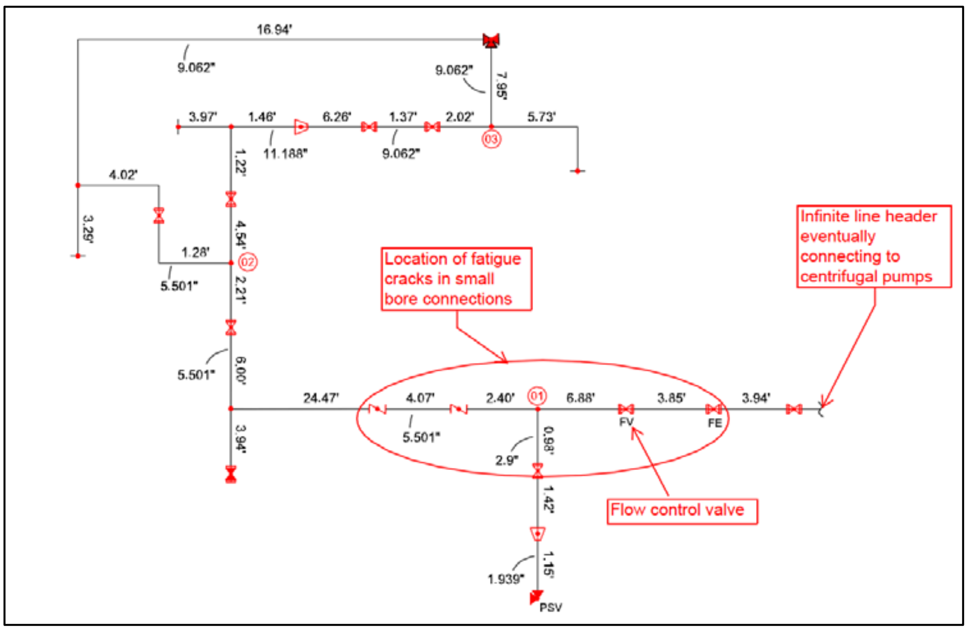

Figure 6: Acoustic Schematic of the Piping System for Case Study 2

The first analysis performed was a vortex-shedding (Strouhal) response/FIV screening to determine if vortex-shedding excitation of any acoustic natural frequencies (ANF) in the piping may be causing high pulsations or vibrations in the system. In piping systems where fluid flow is not machine driven, vortex-shedding of the flow is the primary source of low frequency pulsations (under 200 Hz). Vortex shedding of the flow can occur at any significant change in piping geometry. This can excite the acoustic natural frequencies of the pipe in which the vortices are being generated.

The frequencies of vortex shedding are calculated using the equation below:

$$f_{st} = \frac{St(U)}{D},$$

Where, fst is the vortex-shedding frequency (Hz), St is the dimensionless Strouhal number, U is the fluid flow velocity (ft/s), and D is the characteristic geometry dimension (ft). The acoustical response frequencies of each perpendicular side-branch configuration were determined. The side branches included in the analysis are labeled with circled numbers in Figure 8. Passive response data for complex branch configurations were obtained, if necessary, from an acoustical simulation of the system. This included branch piping with varying diameters or multiple branches. The acoustical response frequencies of simple branch configurations were determined mathematically. However, no acoustic resonances were found in the system between vortex-shedding and acoustic natural frequencies.

The second analysis performed was a flow induced turbulence analysis. Low frequency vibrations in piping systems can be caused by broadband excitation forces that occur due to flow-induced turbulence (typically at 10-15 Hz and less, but higher frequencies can also be an issue). The amplitude of the excitation is a function of the fluid density and the fluid flow rate as well as a function of frequency. The Energy Institute Guidelines[1] were used to evaluate the potential for vibration problems from flow-induced turbulence through the use of a “Likelihood of Failure” (LOF) screening value.

Additionally, this analysis included a mechanical piping restraint review. The mechanical natural frequencies (MNF) of the individual pipe spans were estimated and compared to frequencies of excitation energy potentially in the system. Ideally, modifications should be made, where necessary, in order to avoid significant coincidence between such frequencies. In general, a stiffer system with more supports will have higher mechanical natural frequencies and will be less susceptible to vibration.

The calculated LOF values are summarized in Table 2 with the corresponding recommended actions summarized in Table 3. None of the piping areas evaluated has a high risk for turbulence induced vibration; therefore poor supports or operating with a fluid at high densities, viscosities, or velocities was not a cause for concern in this case.

Table 1. Summary of Calculated Likelihood of Failure Values

Nominal Pipe Diameter (in)	Minimum Mechanical Natural Frequency (Hz) *	Fluid Velocity (ft/sec)	LOF
2.375	15 Hz	9.61	0.181
3.500	15 Hz	9.61	0.169
6.625	10 Hz	9.61	0.255
10.75	10 Hz	9.61	0.215
14.00	15 Hz	9.61	0.075

* Estimated

Table 2. Energy Institute LOF Screening Values

LOF	Action
LOF ≥ 1	Main line **shall** be re-supported or redesigned for relevant piping. Small bore connections actions **shall** be undertaken
1 > LOF ≥ 0.5	Main line **should** be re-supported or redesigned for relevant piping as far as practicable, **or** vibration monitoring of the main line **should** be undertaken after commissioning to ensure mitigation has been successful or to identify if further modification (e.g. bracing) is required. Small bore connections actions **shall** be undertaken
0.5 > LOF ≥ 0.3	Modifications to the main line are not required however; small bore connections actions **should** be undertaken.
LOF < 0.3	Acceptable LOF value. No action required.

Finally, a model of the piping system was constructed in the hydraulic modeling software to evaluate the pressure profiles in the piping system at the provided valve flow coefficients, Cv, for the normal, minimum and maximum flow rates. The boundary conditions of the model were a fixed inlet pressure and a fixed discharge flow rate provided for the system. Various control valve positions were modeled, and it was found that at no point in the system was there a risk of cavitation due to variations in operating points.

Table 3: Simulated Valve Operating Parameters

Case	Inlet Velocity (ft/sec)	Inlet Pressure (barg)	Outlet Pressure (barg)	Pressure Drop (bar)	Valve Coefficient (GPM/[psi]^0.5)
Min Flow Provided Cv	9.61	280.00	122.7	157.30	15.10
Max Flow Calculated Cv	9.61	280.00	40.92	239.08	12.26
Normal Flow Provided Cv	9.59	308.00	127.75	180.25	14.10
Normal Flow Calculated Cv	9.59	308.00	32.98	275.02	11.43
Max Flow Provided Cv	9.58	322.00	122.48	199.52	13.40
Max Flow Calculated Cv	9.58	322.00	16.77	305.23	10.85

With eliminating cavitation as the source of the problem, the control valve was further evaluated by comparing the calculated Cv based on the system conditions to the provided Cv from the valve data sheets. The results show a discrepancy between the provided Cv and the resulting pressure drop across the valve for each operating case. To find a flow coefficient that would provide the required pressure drop across the control valve, the Cv was calculated to a lower Cv value as shown in **Figure 7.**. Hand calculations were also performed using the normal flow conditions provided by the valve data sheet and the equation for the flow coefficient, shown below. The hand calculation showed very similar results to the tuned model with a flow coefficient of 11.48 for normal flow using Equation 1.

$$C_v = \frac{Q}{\sqrt{\frac{\Delta P}{SG}}} \qquad \text{Equation 1}$$

To further investigate the possible discrepancy between the valve design Cv versus the actual Cv, detailed analysis was performed of site data. Pressures and flows were provided for lines A and B while operating. A comparison was performed of the flow coefficients of the identical control valves for those lines as shown in Figure 2. The valve for the line studied has a higher Cv for the same operating flow rate as shown with the vertical green line. By Equation 1, if the flow is fixed and the Cv of the valve is increased, then the pressure drop imposed on the fluid is reduced. Therefore, it is likely that the valve trim or seat is damaged or significantly worn allowing more flow at the same pressure drop. This provided sufficient evidence indicating the valve is likely the source of the problem that the valve was replaced which resolved the vibration problems.

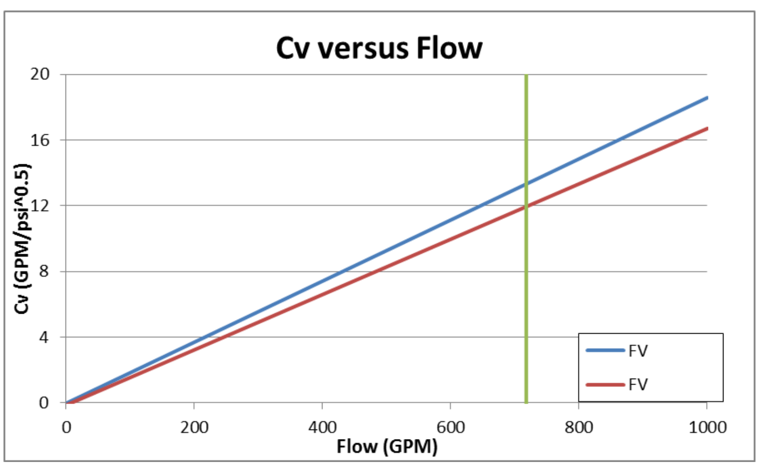

Figure 7: Valve Flow Coefficient Comparison, Blue Line A and Red Line B.

CASE STUDY 3—Slugging Forces in a Pipeline System

An increase in capacity from a production system originated flow pulsation issues in its main pipeline and receiving facility. An existing pipeline that transport multiphase streams was used to manage an additional production from new fields. The system hydrodynamic is very complex and chaotic since it involves phase mixing, different flow patterns, mass transfer and, phase changes. The observed flow pulsations were not critical; however, they were not present prior the increase in capacity. Therefore, an analysis to ensure that they would not affect the integrity of the system was required.

This case study presents an assessment of a 30-inch trunkline from an offshore production separation plant to a separation processing plant on-land. The studied pipeline operates with three phases: oil, water, and gas; thus, flow patterns vary from stratified to severe slugging for some operating conditions and locations. A flow assurance study was conducted to improve the design and reliability of the off-shore transporting multiphase system. The study included a production development from an offshore production-separation platform which is located at approximately 55 miles offshore. The production of different wellhead platforms combine resulting in a multiphase stream. A detailed hydrothermal model of the entire system was developed and different operating conditions were evaluated considering the worst case scenarios for slugging and liquid accumulation. Different gas oil ratios were evaluated to determine the most optimum and safe operating conditions for the system based on a slugging analysis and severity scale. Slug severity statistics and system hydrodynamics were used narrow down critical conditions that should be avoided as well. This case study presents the methodology used in the assessments as well as the results obtained for the steady state and transient cases. A schematic of the studied system is presented in Figure 8.

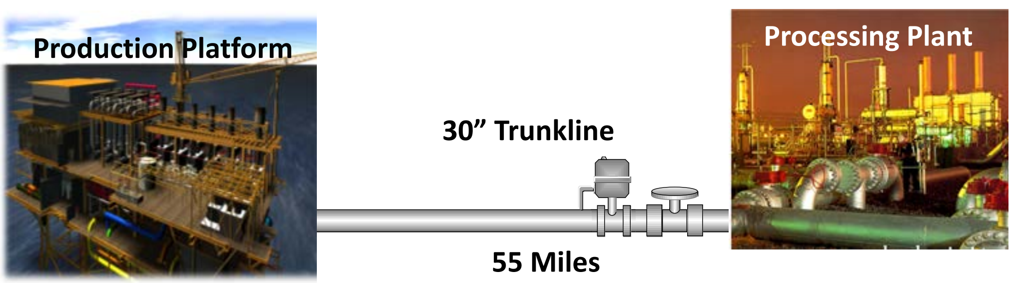

Figure 8. Schematic of the Studied Production System

Analysis Approach

The conducted assessments included a flow assurance analysis to determine the nature of the flow pulsation (slugging) and to hydraulic quantify the forces generated by the slugging. The resulting forces were calculated on the critical locations using the change in momentum and pressure predicted by the hydrodynamic model. The last part of this analysis was to incorporate the slugging forces in a mechanical model to determine the level of stresses generated by the slugging in the piping and determine if there was a possibility of failure. A schematic of the approach used in this analysis is presented in Figure 9.

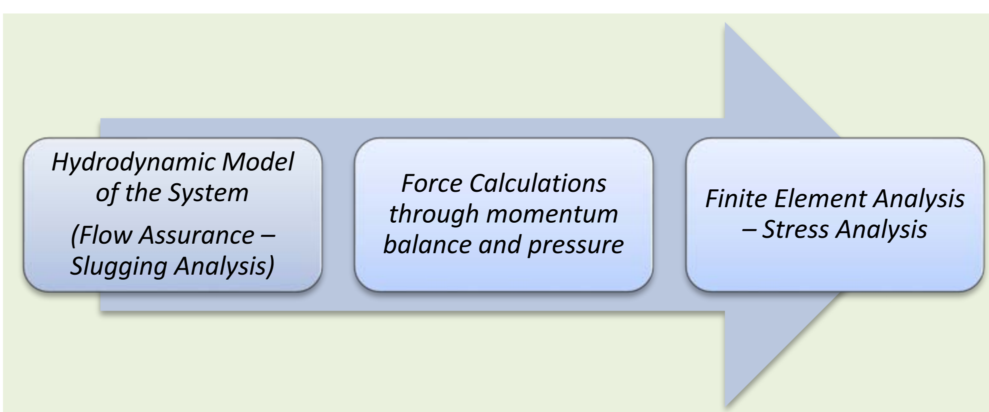

Figure 9. Analysis Approach

Slugging Analysis

Slugging is a multiphase flow pattern which can be considered as a hydrodynamic instability in a pipeline system. Slug flow can be formed due to terrain changes, hydrodynamic conditions, and severe pipe turns such as risers. Slugging can generate harsh transient pressures and flows which cause several operational issues and can damage equipment. It can usually be identified by measurements of pressure, flow, and composition; however, its formation may not follow a cyclical pattern or trend. There are, in general, several means to mitigate slugging in pipelines such slug catchers; oversized separators, which can be very large and expensive; and automatic control of the topside choke or back pressure regulation based on measurements of pressure and flow along the pipeline.

From the flow assurance point of view it is important to understand which operational conditions are more prone to generate slugging and how those conditions can be mitigated. Different steady state and transient scenarios such as production ramp-up and turndown conditions are evaluated to determine the slugging severity, amount of produced liquid, slug volume, and transient pressures. Those parameters are usually compared against the system design conditions and equipment specification such as separation capacity and maximum allowable surge pressure. In addition, pigging operations can lead to slugging; thus, special consideration should be taken when defining the pigging philosophy [2].

Different approaches are taken to determine the severity of pipeline slugging aiming to identify its effect and consequences in the system operation and mitigate any possible risk. Over the years several theoretical and experimental studies [3, 4, 5, 6, and 7] have focused on determining the adverse consequences of slugging to minimize the wear and tear of process equipment that usually leads to unplanned shutdowns.

Multiphase flow pattern maps and correlations have been used to determine the presence of slugging conditions and its severity [8, 9, 10, and 11]. However, the general consensus is that harsh cycling pressures are considered high severity slugging conditions. There will be hydrodynamic interaction between the liquid and the gas phases that will play a big role in defining the intensity of the slugging. In addition, transitional regions can also lead to more unstable conditions that could increase the severity of the slugging for even short or long periods which creates a potential risk for the process and installation. Figure 10 presents a diagram of the flow model developed for the flow assurance assessment. It includes the well flow sources, trunkline, receiving manifold and flow supply from other pipeline as well as the on-land facility boundary condition. The slugging analysis presented in this case study is used to determine the expected reaction forces in the receiving header. For the conducted slugging analysis different conditions were evaluated to determine the worst case scenarios based on the highest forecasted production phase.

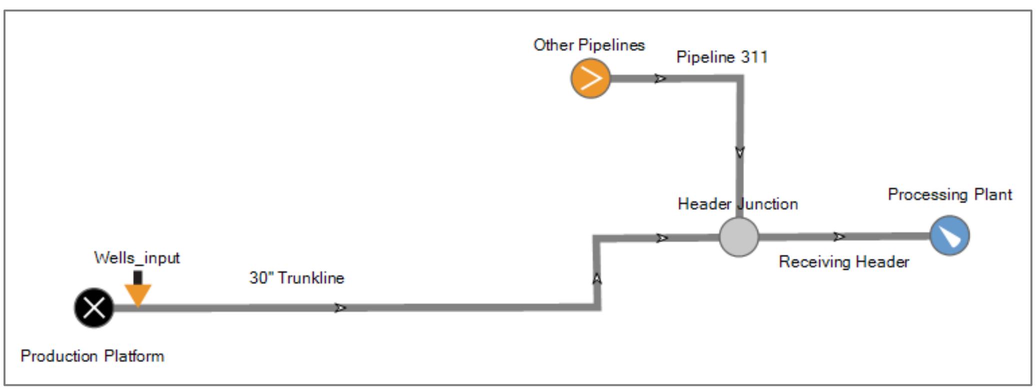

Figure 10. Diagram of the Production System Fluid Model

Pressure fluctuations were monitored closely at the critical locations where slugging is more likely to occur to ensure that the surge pressure does not exceed the maximum limit established for the pipeline. The maximum design pressure for the pipeline is 900 psig and the maximum slug dynamic pressures calculated for the system are about 14-24 psi which does not pose an overpressure risk since the maximum operating pressures are in the range of 330-350 psig. Thus, the levels of surge pressures are in compliance with the ASME standard [11] and other engineering criteria.

Dynamic pressure and surge volume were monitored at different locations along the pipeline to identify the critical points. Figure 11 and Figure 12 present the pressure and surge volume trends obtained at mile 5.95 and at the receiving facility; low frequency and low amplitude pressure were observed at all the critical locations. No distant pattern was observed for either the pressure or surge volume; however, a direct relation was obtained for both variables as expected. An average pressure of approximately 333 psig with a variation of 14.1 psi was calculated at mile 5.95. At the receiving plant the average holding pressure is about 80 psi and the pressure fluctuation were approximately 16.8-22 psi. At this location three critical points were identified in the connecting pipe between the trunkline and the facility receiving header. Those three points were 45º and 90º turns where the high flow rate fluid stream makes drastic change in direction and pressure fluctuations occur.

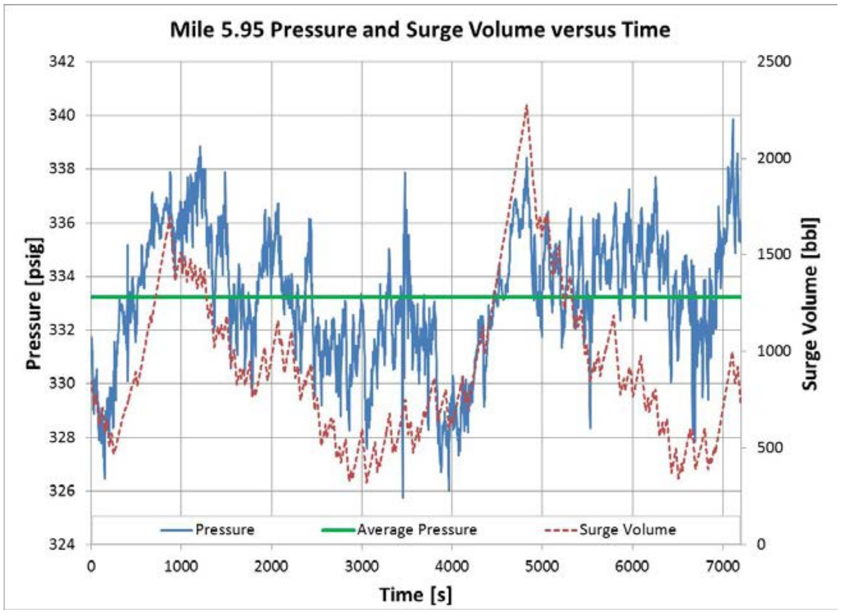

Figure 11. Pressure and Surge volume at Mile 5.95 versus Time

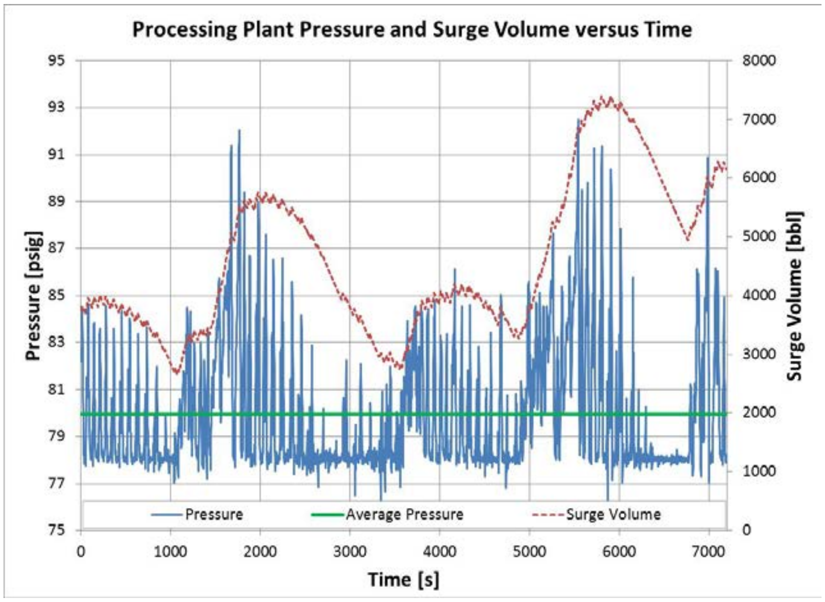

Figure 12. Pressure and Surge volume at the Receiving Facility versus Time

Stress Analysis

A mechanical and thermal analysis was performed on the lines between the trunkline tie-in and gathering header lines of the processing plant. These analyses were performed using piping stress analysis software (Caesar II). The analyses were performed to show compliance with the API 618 specification (minimum mechanical natural frequencies) and ANSI B31.8 piping code (thermal stress).

A basic diagram of the finite element model used in the analyses is shown in Figure 14. The thermal analysis considered weight, temperature, pressure, and pipe excitation loads. The load cases analyzed are presented in Figure 13. The pipe excitation loads on the indicated elbows upstream of gathering header line were calculated as previously discussed. The total force components were obtained from the momentum balance and pressure on each direction and then multiplied by a dynamic amplification factor, resulting in the FX and FY direction force components that were applied to the model.

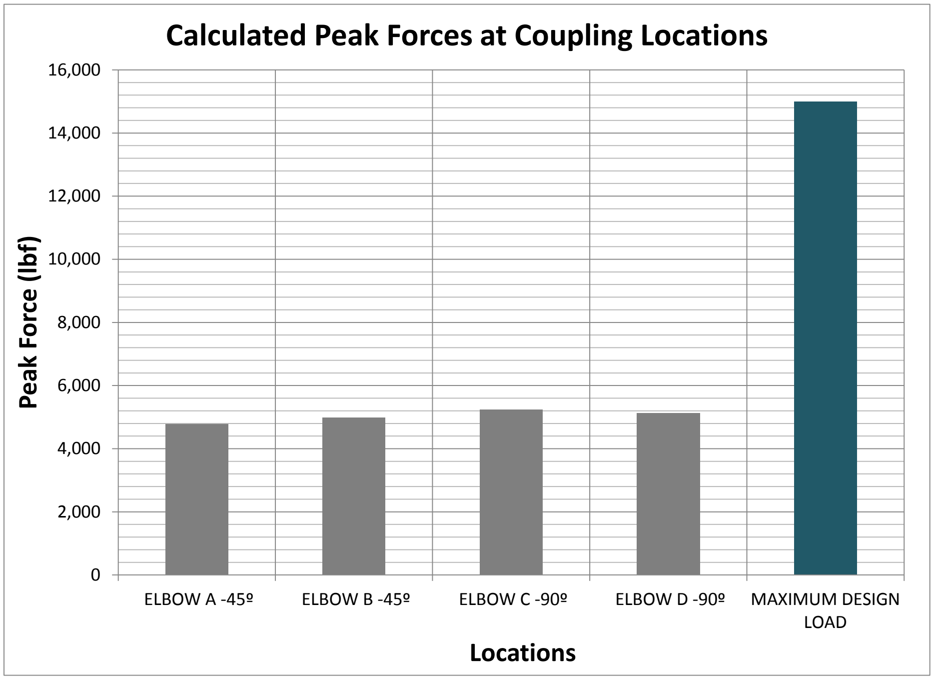

Figure 13. Calculated Peak Forces for the Stress Analysis

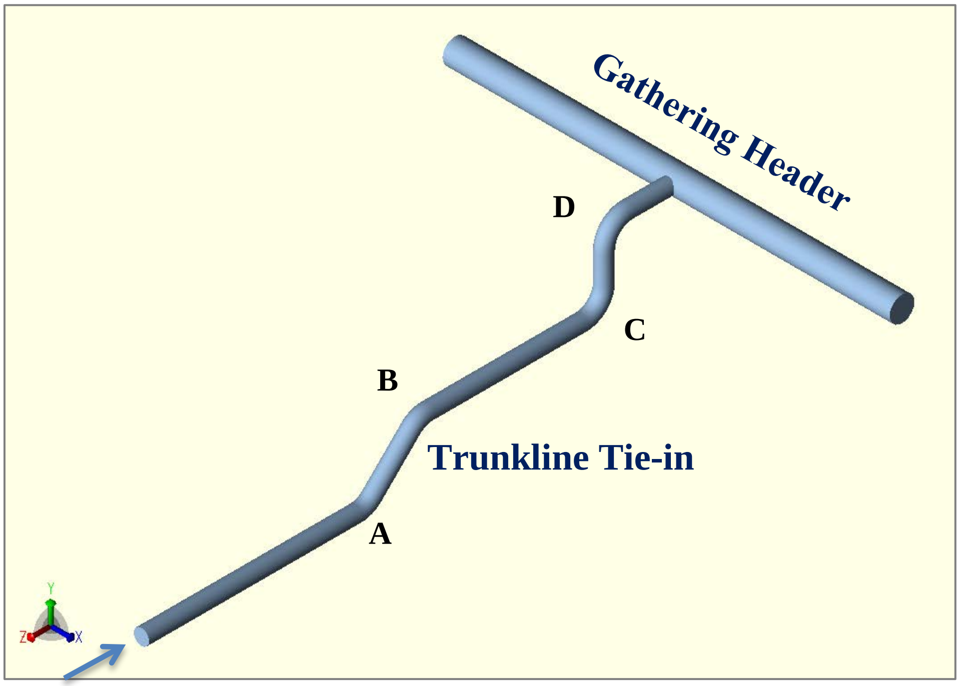

Figure 14. Identified Critical Locations for the Stress Analysis

The multiphase assessment indicated hydrodynamic and terrain slugging within the pipeline system for all evaluated operating conditions. A slug statistical analysis tracked the frequency and duration of the hydrodynamic slugs along the pipeline which provided critical locations for its detail evaluation which are located just prior to entering the production facility as presented in Figure 14.

An increase in the number of slugs for the higher production rates, when compared with the current design, was observed. In addition, the slugging at the processing plant shifts from high frequency, short duration to lower frequency, high duration. However, the magnitude of this shift in the slugging behavior is not significant when compared to the total production of trunkline and presents minimal slugging risk. Nevertheless, the existing separating capacity at the receiving plant is more than sufficient to accommodate the slugging behavior of the system and production liquid rates safely.

Pressure fluctuations and surge volume at the critical locations do not exceed the maximum limit established for the pipeline. The maximum slug dynamic pressures calculated for the system are about 14-24 psi which does not pose an overpressure risk since the maximum operating pressures are in the range of 330-350 psig. The calculated stresses and reaction forces were in compliance with the current piping design; therefore, no additional integrity risk was considered for the operation of the system for these higher production rates.

CONCLUSIONS

Pump piping systems are frequently subject to transient events such as slugs, water hammer, and cavitation that can create high amplitude forces and pressure spikes. Liquid systems are more susceptible to damaging forces during transients than gas systems due to the high density and incompressibility of the operating fluid. These events often result in high vibrations--and sometimes failures--that can be prevented in the design process or resolved in the field using a multi-angle approach.

Three different case studies were presented in this paper as problems existing on site that required a combination of analyses types to solve. The traditional method of analyzing transient events in liquid systems is to use a software such as SPS or HYSYS; however, often these transient fluid models need to be combined with mechanical or thermal stress analyses; dynamic pressure versus time inputs, and sensitivity studies or field data evaluations to be effective in resolving problems and ensuring pipeline integrity.

NOMENCLATURE

D	characteristic geometry dimension (ft)
F_{st}	frequency of vortex-shedding (Hz)
ΔP	pressure drop , [psi]
Q	flow rate [GPM]
SG	specific gravity
St	Strouhal number
U	fluid flow velocity (ft/s)

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