Gas Passage System Pulsation Analysis for Modern Reciprocating Compressors

by:

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Abstract:

Pulsation models of reciprocating compressor systems commonly utilize a one-dimensional (1-D) representation with acoustic length modifications to represent the three-dimensional system. Given the simplicity and cost-effectiveness of this approach, one-dimensional models are the preferred means of analyzing dynamic pressures in the compressor system and generally accurate for piping systems where the dominant physical length is in the flow direction. In the areas near the compressor cylinder, very close to the fluid force excitation from the piston, the 1-D assumptions break down since many of the high-frequency energy components have not diminished.

The following paper describes a new methodology and its benefits for pulsation analysis of the gas passage system (which includes the cylinder nozzle and primary volume bottle). The intent of the investigation by Southwest Research Institute was to determine the most effective method of predicting natural acoustic responses and expected pulsation levels within the gas passage system. The study resulted in an improved means of developing a 1-D representation using a 3-D acoustic modal analysis, which provides much improved predictions of nozzle and gas passage resonances. This work validated the new approach and refined the SwRI 1-D Navier Stokes fluid modeling method. SwRI also determined the potential benefit of a small Helmholtz resonator designed to attenuate problematic gas passage responses in the range of 100-300 Hz.

1. Introduction

Pulsation models of reciprocating compressor systems commonly utilize a one-dimensional (1-D) representation with acoustic length modifications to represent the three-dimensional system, given the simplicity and cost-effectiveness of this approach. One-dimensional models are generally accurate for piping systems where the dominant physical length is in the flow direction. In the areas near the compressor cylinder, very close to the fluid force excitation from the piston, the 1-D assumptions break down since many of the high-frequency energy components have not diminished. A new methodology and its benefits for advanced pulsation analysis of the cylinder gas passage system (which includes the cylinder nozzle and primary volume bottle) is studied in this work. The intent of the investigation by Southwest Research Institute was to determine the an improved method of predicting natural acoustic responses within the gas passage system and the potential value to operating companies in applying the new methodology.

Without accuracy in both frequency and amplitude predictions of the related gas passage resonances, it is unclear whether a significant acoustic resonance exists. For new compressor cylinder designs, certain 1-D representations of the gas passage may not be valid and could lead to incorrect predictions. Uncontrolled responses associated with the cylinder gas passage system are primarily evident as higher frequency pulsations and vibrations at the compressor valves, cylinder body and in the cylinder nozzles. Incorrect designs will lead to use of cylinder nozzle orifice plates, poor valve life and low compressor efficiency due to high dynamic pressure drop.

SwRI conducted an investigation of the acoustic responses related to typical high speed reciprocating compressor cylinder gas passageways. Many times, a 3-D transient CFD model is too time and cost intensive and cannot be justified for standard design studies (i.e., API 618 acoustic analyses). However, the combination of a 3-D acoustic response model and a 1-D fluid representation model can provide accurate predictions of all gas passage system responses in a cost effective manner. This work validated the new approach and refined the SwRI 1-D fluid modeling method.

2. Background

Three-dimensional (3-D) acoustic responses in the compressor cylinder gas passageways due to the dynamic flow through the compressor valves can have a significant influence on the performance of the compressor. In some cases, the high-frequency uncontrolled acoustic resonances could lead to valve failures, cylinder pulsations and vibrations, and compressor manifold system vibrations. Since the acoustic frequency content tends to be much higher in the gas passageways than in other areas of the manifold system, the cycles to metal fatigue failure are considerably reduced. Additionally, cylinder nozzle pulsations near the gas passage / nozzle resonant frequency can produce major nozzle vibrations and require the addition of power consuming orifice plates in the flow stream to adequately control pulsations.

Standard fluid models of reciprocating compressor systems use a one-dimensional representation with acoustic length modifications to represent the three-dimensional system. This is adequate except in two cases: (1) when the overall geometry is highly three-dimensional, such that one particular length does not dominate and it is difficult to collapse to a one-dimensional representation (this is the case in the internal gas passages (IGP's)); and (2) when dynamic pressure frequency content is high enough that the wavelength of the response frequency is not influenced by the corrections for one dimension.

There are several significant acoustic responses related to the gas passage and its connection to the pulsation filter bottles:

1. Valve-to-valve (VTV) acoustic responses are more analogous to an acoustic length response than a classic Helmoltz response. These types of responses are typically half-wave modes or higher orders of a length response – as indicated by responses 1 and 2 shown in Figure 1. These occur at higher frequencies in the range of 100-300Hz for most high speed compressors. This response depends on the length between valves and can be calculated by hand as a general approximation (as a closed-closed half wave response). Depending on the amount of high frequency excitation and the passage design, these can be strong sources of high frequency pulsation and vibration. These are internal to the gas passage and do not typically propagate beyond the compressor cylinder flange. If the reciprocating compressor valves have a mechanical natural response near the VTV pulsation wave frequency, this acoustic response can complicate the valve behavior and will adversely impact valve life.
2. The so-called "cylinder nozzle resonance" (CNR) is actually a combined effect of the gas passage, cylinder nozzle, and primary volume bottle. As such, it should be termed the passage-nozzle-bottle response – see

response 3 in Figure 1. These three component geometries in combination produce a small choke tube and open volume which produces a quasi Helmholtz acoustic response. Since the gas passage and nozzle dimensions often have similar diameters and lengths, the response is difficult to calculate by hand and must be simulated in a one-dimensional (or 3-D) fluid model. The nozzle resonance is often misrepresented as a quarter wave acoustic response (with open-end, closed-end boundaries) but it is in typically much closer in its behavior to a Helmholtz type acoustic response.

3. Other acoustic responses at even higher frequencies (above 250 Hz) are possible due to radial modes, side-to-side length responses, and higher order components of the above responses. These are less likely to be problematic for modern high speed reciprocating compressors but coincidence with cylinder stretch excitation frequencies or mechanical resonances in the manifold system can still cause significant vibrations at times.

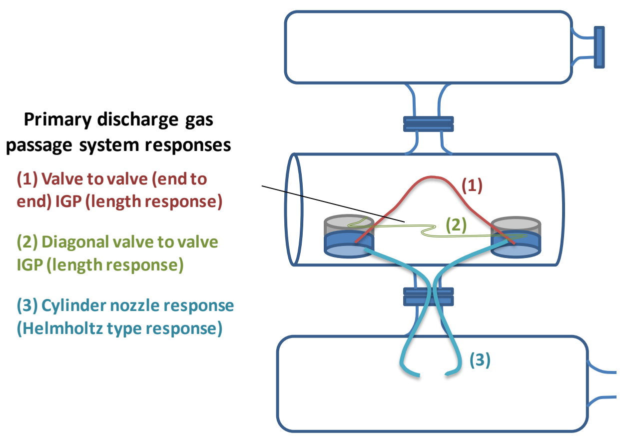

Figure 1. Conceptual View of Primary Gas Passage System Responses for a Typical Discharge Gas System on a Reciprocating Compressor

The various fluid dynamic models, which were used in this investigation, required either a fluid dynamics based approach or a finite element approach. The finite element solution was effectively an inverted solid model which allowed the fluid space to be created. The fluid space could then be analyzed for its natural resonances similar to how a solid body finite element model would predict mechanical natural frequencies. The 3-D response analysis using an acoustic modal tool provides only the resonant frequencies in the fluid domain (not their relative strengths or amplifications).

Both the 1-D and 3-D fluid dynamic models utilize the kinematics of the piston in the compressor cylinder to determine a transient boundary condition based on the generated pressure pulse. The models can determine acoustic responses of the fluid based on the natural acoustics and the excitation forces. However, the fluid models are more versatile than the finite element response modeling because the excitation forces modeled through the piston's compression process provide a "forcing function" to the fluid resonances, which allows the fluid models to predict the frequency and amplitude of the dynamic pressures. Frequency and amplitude determination are critical to an effective design process since many responses will arise in a given piping system and the design must focus on attenuating the high amplitude pulsations which can lead to component failures and poor performance.

After characterizing the forcing function in the fluid model, the boundary condition is then applied to the mass, momentum, and energy equations. The finite element approach utilizes the geometry of the gas passage to determine the natural acoustic resonances. The state of the gas in the compressor and manifold is determined by two factors: (1) the kinematics of the drive, which determines the volume inside the cylinder and the isentropic change of state in the gas, and (2) the inflow and outflow through the suction and discharge valves at the boundaries. The piston position, $z(\varphi)$, with respect to crank angle is given by:

$$z(\varphi) = \frac{V_s}{A_p} + r \times (1 - \cos\varphi) + \frac{r}{\lambda} \times \left(1 - \sqrt{1 - \lambda^2 \sin^2\varphi}\right) \qquad (1)$$

where: r = crank radius
λ = rod ratio
V_s = clearance volume
A_p = piston area

The piston position translates to the transient gas pressures during the compression process. With the pressure-volume card which is generated by the piston position versus time trace, the boundary condition for a 1-D or 3-D fluid model is provided. This is effectively the velocity or pressure of the gas as it enters the suction valve (for a suction side model) or exits the cylinder through the discharge valve (for the discharge side model). In order to calculate the velocity and the pressure at every time-step, the transient one-dimensional Navier-Stokes equations of fluid dynamics must be solved (Brun, 2007):

$$\frac{\partial \rho}{\partial t} + \frac{\partial(\rho v_x)}{\partial x} = 0 \qquad (2)$$

$$\rho\left(\frac{\partial u}{\partial t} + u\frac{\partial u}{\partial x}\right) = -\frac{\partial P}{\partial x} + \mu_S \frac{\partial^2 u}{\partial x^2} \qquad (3)$$

$$\Delta h_t + e = 0 \qquad (4)$$

This equation can also be solved for all three directions with more terms as in a CFD code. However, the discretization method is important to the solution of this type of highly dynamic flow from a reciprocating compressor. The third equation of the Navier-Stokes equations (4) will provide for heat transfer effects, important to the performance calculations. However, the mass and momentum equations are the governing solution for the prediction of pulsation levels from the reciprocating compressor cylinder.

In the past with a linearized wave equation solution to the one-dimensional fluid problem, all physical lengths were adjusted to "acoustic dimensions" which also compensated for 3-D effects. One of the classic end effect equations for a choke tube within a larger pulsation bottle is provided below:

$$L_e = L + D_c\left(0.85 - 1.06\left(\frac{D_c}{D_b}\right)\right) \qquad (5)$$

where:
L_e = new length with the end-effect included
L = length without the end-effect
D_c = smaller diameter (typically the choke tube)
D_b = larger diameter (typically the bottle diameter)

Another effect that becomes more important in smaller dimensioned bodies is the effect of compressibility on a small gas passageway. Typically, to adjust for a smaller cavity volume and the compressibility effect, a damping coefficient and modified frequency may be calculated that includes the density and viscosity terms of the fluid (Persico, et al., 2005):

$$\omega_n = \frac{c}{L\sqrt{\frac{1}{2} + \frac{V}{V_t}}} \quad \text{and} \quad \zeta = \frac{16\mu L\sqrt{\frac{1}{2} + \frac{V}{V_t}}}{d_t^2 c\rho}, \qquad (6)$$

However, these correction terms are only general methods of adjusting the classic linear wave equation models. It is now possible using advanced models with viscous effects, fluid compressibility and non-linear dynamic velocity terms to accurately predict these effects.

3. Approach

The SwRI study compared the fluid and finite element model results to data obtained on two high speed compressors. To focus the investigation, only the discharge system was studied from the discharge valves through the gas passage in the cylinder, through the cylinder nozzle to the first volume bottle. The discharge system is typically where higher pulsation amplitudes are observed, which makes this system easier to characterize through field measurements. Two data sets were available to verify the models:

- Case A: A fixed speed machine operating at 890 RPM. Data was recorded at the discharge valve at various load steps including fully double acting and other unloading steps using volume pockets. The discharge nozzle response was measured after the addition of orifice plates at the cylinder nozzle flange, which dampened the nozzle response.
- Case B: A variable speed machine operating from 825-1,000 RPM. Data was recorded on both the head and crank end in the discharge valves for the double acting (fully loaded) cylinder. Data was recorded prior to installing orifice plates at the nozzle flange.

The discharge valve cap on each of the machine cylinders was used as the dynamic pressure measurement point for the field pulsation data. The following advanced models were developed to compare to the gas passage and cylinder nozzle responses seen in the field data:

1) A 1-D representation model in the advanced SwRI pulsation modeling tool (TAPS). This model includes all terms of the Navier-Stokes equations in one-dimension.
2) A 3-D finite element type model using acoustic modal analysis tools, used to predict frequency responses and modes (using ANSYS FEA).
3) A full 3-D computational fluid dynamics (CFD) model solved with a transient boundary condition at the discharge valves, used to predict magnitude of pressure variation (%) and primary frequency responses.

The goal of the study was to determine if a combination of these advanced models could be used to better identify and predict both the cylinder nozzle response and the higher frequency gas passage responses for modern low and high speed compressors.

As part of the investigation, SwRI also examined possible design solutions for the higher frequency gas passage resonances using advanced pulsation control concepts from the Gas Machinery Research Council compressor research program. The use of a miniature virtual orifice (modified Helmholtz resonator) to absorb the valve-to-valve response showed particular benefit in mitigating high frequency pulsations.

3.1 Data Review of Primary Responses

In Case A, SwRI recorded field data at the fixed speed of 890 RPM for one cylinder of a six cylinder unit. The field data was taken at the discharge valve cap at various load steps. The field data showed somewhat high frequencies in the nozzle resonance region at 60 and 90 Hz – see Figure 2. However it was difficult to pinpoint the location of the true cylinder nozzle resonance due to the fixed speed operation. In comparing the double acting case when the unit was fully loaded to a more unloaded cylinder condition, SwRI found that the 5x excitation at near 75 Hz increased to 36 psi pk-pk (see unloaded case – Figure 3). This excitation is likely due to the primary nozzle resonance peak lying close to this frequency. Pulsation data for the two load steps is shown below:

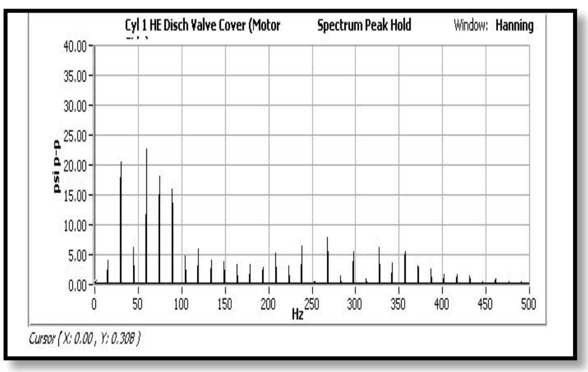

Figure 2. Case A Pulsation Levels at the Discharge Valve for the Fully Loaded Case

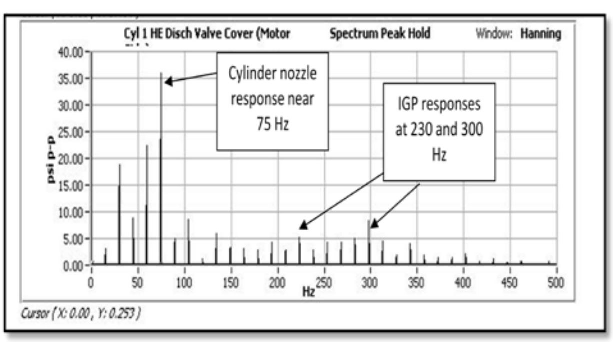

Figure 3. Pulsation Levels Measured at Discharge Valve in Unloaded Cylinder Operation, for Case A field site

As shown in Figure 1-2, the high frequency content of the dynamic pressure also showed significant responses in the gas passage at 200-250 and 300-350 Hz range, although it was difficult to determine the exact response due to the fixed speed test. These higher frequency responses were significant because the pulsation amplitudes were all measured to be between 5-10 psi pk-pk and had corresponding vibrations at the same frequencies. The gas passage responses were due to excitation of the various acoustic lengths across the cylinder chamber.

For Case B, field data was recorded during a speed sweep of the unit such that the responses coinciding with excitation from 825-1000 RPM (primary first order = 13.75-16.6 Hz). This case was added to the SwRI investigation of gas passage responses in order to further verify the 3-D response model and the fluid models. The speed sweep in the field allowed the responses in the gas passage system to be fully traced out, showing the exact location of pulsation maximums. The field data showed a fairly broad cylinder nozzle resonance at 65-75 Hz. The pulsation levels at the discharge valve cap due to the nozzle resonance were as high as 85psi pk-pk. Crank end pulsation levels recorded at the discharge valve cap are shown in Figure 3 for Case B.

Lower amplitude responses were evident at 250 Hz and around 270 Hz on both ends of the cylinder. These responses were attributed to valve-to-valve length responses across the cylinder in the piston direction. (The nozzle resonant pulsations were causing high vibrations and were the primary cause of concern. The vibration issue was later rectified with orifice plate installation at the nozzle cylinder flange.)

Both Case A and Case B were relatively high horsepower units which tended to cause higher frequency excitation in the gas passage area due to the added energy at higher orders in the system. Both cases exhibited acoustic valve-to-valve type responses in this higher frequency range.

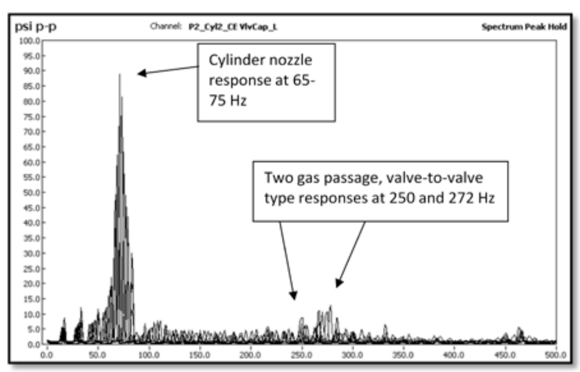

Figure 4. Pulsation Levels at the Discharge Valve Measured for Case B Field Site

3.2 3-D Response Model

The 3-D response analysis was performed using a combination of Solidworks and ANSYS finite element models to analyze the fluid space of the cylinder gas passage, nozzle, and filter bottle system. Acoustic modal analysis aims at determining the acoustic natural frequencies of a volumetric model. While all responses in a desired frequency range are determined using this type of analysis, not all responses are necessarily excited and are dependent on operating conditions. The outputs from the acoustic modal analysis are the natural frequencies and their corresponding mode shapes.

The acoustic modal analysis tool is an effective means of determining the natural responses of a solid or fluid domain but when used by itself, this type of analysis will not determine the amplification of the various responses. It is a true three-dimensional model and provides a means of visualizing the acoustic response, which can be beneficial in determining means of attenuating, altering or shifting the acoustic response.

In determining the significance of the various responses, the designer must review the dimensions and acoustic node and anti-node points in the response plots.

For Case A, the 3-D acoustic response model predicted four responses in the gas passage system – see Figure 5. The primary responses of significance to the manifold system were the cylinder nozzle response predicted at 80.5 Hz and the valve-to-valve response at 302 Hz, in the piston direction. Other internal gas passage responses predicted for Case A included a side-to-side response opposite the piston direction at 236 Hz and a diagonal valve-to-valve response at 333 Hz.

Although the field data for Case A was only available at the fixed speed of 890 RPM, the cylinder nozzle response at 80 Hz was close to the field data high pulsations at 35 psi pk-pk observed at 75 Hz during the unloaded operation (when off order excitation for 1x, 3x, 5x etc. would have been higher). The primary length response within the gas passage was predicted at 302 Hz. This response had good correspondence in the field data at the same frequency because it was the dominant acoustic response to the gas passage.

For Case B, the 3-D acoustic response model predicted four responses in the gas passage system which matched very closely to the field data – see Figure 6. The primary responses of significance to the manifold system were the cylinder nozzle response predicted at 67 Hz and the valve-to-valve response at 258 Hz and 277 Hz.

The cylinder nozzle resonance predicted to be at 67 Hz in the 3-D response model, actually peaked at approximately 70 Hz in the field data. The small frequency shift from the field data was likely due to damping effects and the magnitude of excitation frequencies in the actual field system. This was also a fairly broad cylinder nozzle response which could be excited by multiple compressor orders, as shown by the field data speed sweep.

The gas passage responses at 258 and 277 Hz also matched the field data closely (shown in Figure 3). Both of these responses were aligned with the piston direction, which served as the forcing excitation for the fluid system. The side-to-side gas passage response at 180 Hz was not as amplified in the actual system. One reason for the diminished amplification of this response is due to its direction conflicting with the piston direction. As such, this response was not as evident in the field data.

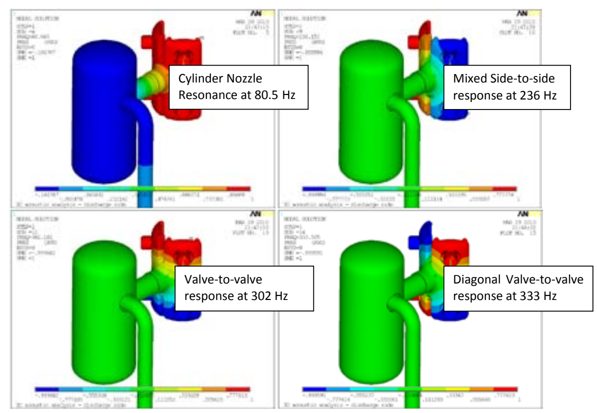

Figure 5. 3-D Response Analysis for Case A – Primary Responses Shown at 80.5 Hz, 236 Hz, 302 Hz and 333 Hz

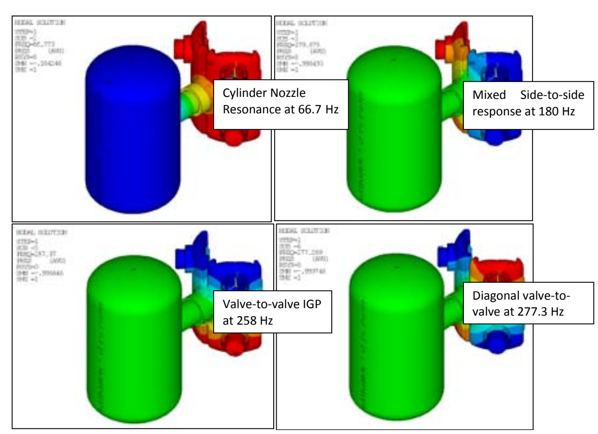

Figure 6. 3-D Response Analysis for Case B – Primary Responses Shown at 66.7 Hz, 180 Hz, 258 Hz and 277.3 Hz

3.3 3-D Transient Fluid Model

The next option considered in the gas passage pulsation analysis was to utilize a full 3-D fluid dynamic model run for a transient condition. Any CFD analysis depends on the solution of the Navier-Stokes equations, which define any single-phase fluid flow. In this case, the fluid equations were solved using the ANSYS CFX tool after the computational mesh of the fluid space was generated. Then the boundary conditions were applied and the simulation was started, where the equations were solved iteratively as a transient problem. The simulation was appropriate because the pulsations associated with a reciprocating compressor are highly transient and the acoustic mode excitations must be studied using a transient model.

To limit the computational time, only the Case A system was studied. The computational domain used for the analysis discharge side of the compressor is shown in Figure 7. In order to monitor the pressure pulsations inside the domain, twelve (12) monitoring points were setup and the time history of the pressure at each point was recorded. A square wave type flow pulse was used at the inlet to simulate the alternate crank-end and head-end valve openings, with a volumetric efficiency matched to a typical high speed compressor. Boundary conditions used for the simulation were based on the compressor operating conditions from the field data for Case A.

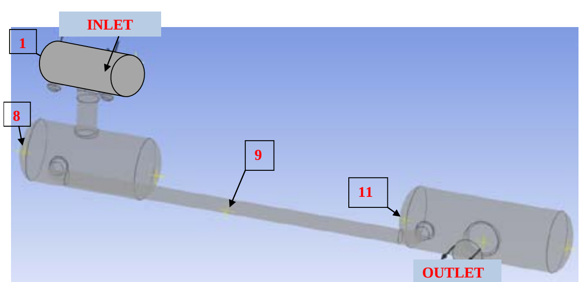

Figure 7. Overall Computational Domain for the 3-D Transient Model (Cross marks represent the monitoring / sample points inside the domain)

In order to obtain the frequency content of the dynamic pressure at each test point, a fast Fourier transformation (FFT) was performed on the time history of the pressure at the various monitoring points. An appropriate transient simulation time step was used such that frequency content up to 400 Hz was captured. The FFT program determined the spectral content in terms of maximum pulsation amplitudes obtained during the sampling time period. Figure 8 and Figure 9 show the FFT results for the pressure at test points 1 and 8, in the gas passage and at the end of the first volume bottle, respectively.

The transient fluid model predicted a dominant 5x response at 75 Hz, which corresponds well with the cylinder nozzle resonance seen in the field data (also at approximately 75 Hz) and the 3-D acoustic response model and 1-D representation model (predicted to be at 80Hz). Also evident was the Helmholtz response of the two-bottle system which fell below the 1x excitation at approximately 10-11 Hz. The amplitude predicted was less than the field data showed for the nozzle resonance (35 psi pk-pk compared to 25 psi pk-pk in the transient model). However, if the damping factors can be adjusted, the transient 3-D fluid model may be able to determine accurately the amplitude and frequency of cylinder nozzle resonances. Higher frequency gas passage responses are not predicted as well with the current transient fluid model and need to be further investigated in the representative one-dimensional fluid model. Since the transient fluid model was not able to delineate the higher frequency internal gas passage responses, this task was discontinued until the method could be better refined.

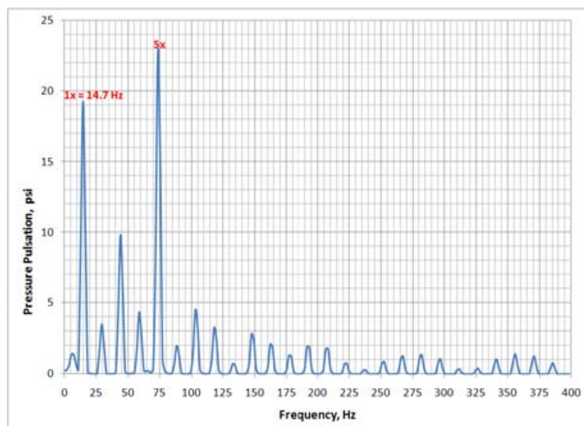

Figure 8. 3-D Transient Fluid Model Results for Dynamic Pressure Predicted Internal to Cylinder (Test Point 1)

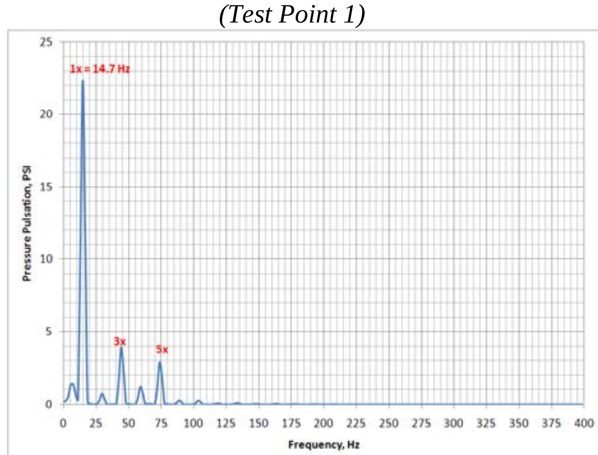

Figure 9. 3-D Transient Fluid Model Prediction of Pulsations within the Primary Volume (Test Point 8)

3.4 1-D representation fluid model

Construction of the one-dimension (1-D) acoustic model required conversion of the 3-D system into a 1-D representation. The 1-D representation was comprised of specific lengths and diameters of circular pipe. The boundary conditions for the Case A suction and discharge models included an infinite line upstream (suction) or downstream (discharge) of the compressor bottle filter (secondary) volume, the filter volume, the choke tube, the primary volume, the cylinder nozzle, and the internal gas passage to the compressor cylinder valves. Two different variations of the 1-D representation were analyzed for Case A, including a branched style H-pattern and central volume / distributed X-pattern for the gas passage area – see Figure 10 for conceptual depictions.

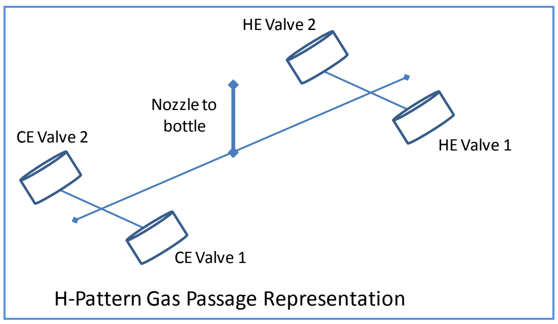

Figure 10A. Gas Passage H-Pattern 1-D Representation Style

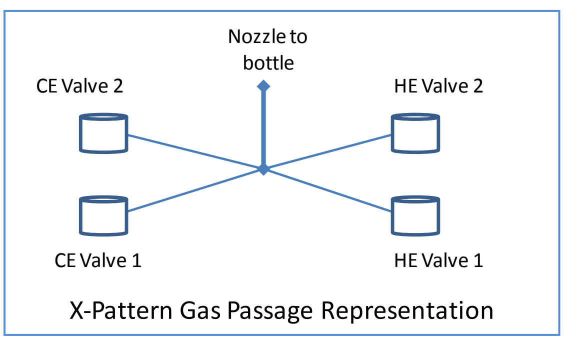

Figure 11B. Gas Passage X-Pattern 1-D Representation Style

The boundary conditions for the Case B discharge model included an infinite line downstream of the compressor bottle filter (secondary) volume, the entire filter bottle (the two choke tubes, the two primary volumes, and the filter chamber), the two cylinder nozzles, and the two internal gas passages to the compressor cylinder valves. Four variations of the 1-D representation were developed for Case B, including both the branching type and central-volume / distributed type system.

Once the 1-D representation was developed and the operating conditions matched to the field data, the fluid system was modeled using the SwRI full Navier-Stokes solution in one-dimension (termed the Transient Analysis Pulsation Solver or TAPS). This solver is applicable to any complex manifold and piping system and was developed internally by Southwest Research Institute (Brun, 2008). This is a more physically realistic model than the acoustic wave equation and includes effects of compressibility and non-linear pressure and velocity effects in time and space.

Boundary conditions were modeled as pressure varying boundary conditions at the discharge valves at the speeds and operating conditions specific to the system being analyzed. Output data from the 1-D pulsation analysis can be directly compared with the field data plots. For the first case, SwRI developed two internal gas passage (IGP) representations of the 3-D system, termed IGP1 and IGP2. IGP1 was an X-pattern type model while IGP2 was a distributed H-pattern gas passage. After comparing to the field data and 3-D response analysis, SwRI determined the H-pattern provided much improved frequency matches and better correlation with the other two data sets for this particular gas passage design.

Case A results using IGP2 for the discharge valve test point is shown in Figure 11. The model matched the predictions of the response model for responses at 75-77 Hz and the broad gas passage valve-to-valve response shown at 235-240 Hz. Other system responses predicted at 160 and 320 Hz. Damping factors are needed to reduce these responses which are believed to be less Figure 11 significant as they are not primary length responses or valve-to-valve type responses. This additional refinement to the 1-D fluid model was completed using Case B.

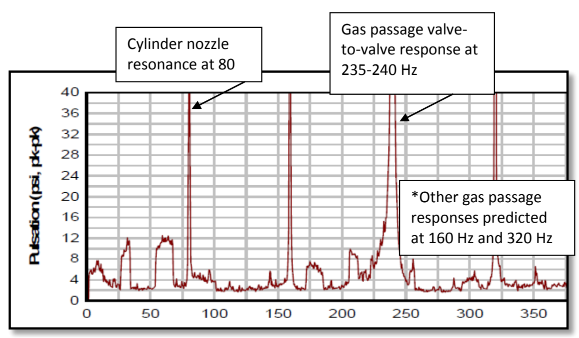

Figure 12. Case A Results using IGP2 – Discharge Valve Cap Pulsations Predicted at 76 Hz, 160 Hz, 230 Hz and 320 Hz

After determining the one-dimensional representation to use, SwRI further developed the 1-D representation model using Case B data by investigating the use of physical lengths versus acoustic lengths. Damping factors were also adjusted, which affected the amplitude of some responses and the frequency prediction of the cylinder nozzle resonance. This investigation was completed for the Case B system where more useful field data was available because of the variable speed operation. Damping was added to the small passageways near the valves. A preferred 1-D representation was also utilized. In addition, two models were run, where the first model used an acoustically modified length for the nozzle and the second model used the actual physical length of the nozzle.

Using the physical length of the nozzle resulted in a frequency shift closer to the actual nozzle resonance at 72 Hz (see blue line in Figure 12.) The physical length model also showed a better match with the 180-185 Hz mixed passage response and the 255-258 valve-to-valve response. **Both gas passage responses matched very closely to the field data and the predictions of the 3-D response model. These results were to be expected since the new SwRI pulsation tool is a physical model without the acoustic wave equation simplifications.**

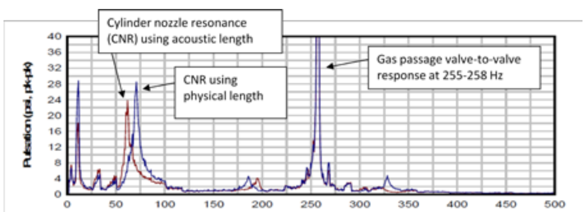

Figure 13. Case B Pulsation Predicted at Discharge Valve Cap. Blue Line Shows Shift in Nozzle Response to 70-72 Hz, using Cylinder Nozzle Physical Length Compared to Acoustic Length Model (Red Line)

The final investigative task was to determine if a small Helmholtz resonator could be effective at attenuating high frequency internal gas passage valve-to-valve type responses in the cylinder. The design of the Helmholtz resonator is based on the GMRC Virtual Orifice device developed for nozzle resonance control (Broerman, 2008). A small volume with an internal orifice and tube projection was sized for an absorption of the pressure pulsation at 258 Hz. This volume and choke tube was added to the 1-D representation model to represent installation near a valve cap or mounted on the compressor cylinder and connected to the gas passage internals (either head end or crank end can be used, but only one volume is required per side of the cylinder.) The results shown in Figure 13 suggest a reduction of about 20:1 in the pulsation amplitudes within the gas passage. This amount of attenuation would be effective in reducing cylinder vibrations and extending valve life.

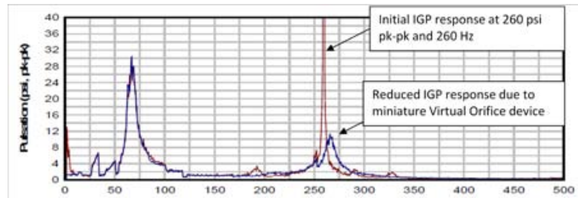

Figure 14. Reduction in Gas Passage Valve-to-valve Response at 260-270 Hz using the Miniature Virtual Orifice in the Gas Passage System

4. Conclusions

The SwRI investigation resulted in an improved methodology for accurate pulsation modeling of new compressor cylinders. This advanced multi-faceted design approach can be used in combination with OEM / bottle design options to place the gas passage and cylinder nozzle responses at optimal frequencies for lower pressure drop compressor manifold systems. Other specific conclusions are as follows:

1. One dimensional representations of the gas passage can influence the nozzle and gas passage responses. The current SwRI investigation showed that the representation affects the model predictions for the gas passage system responses. The methodology developed is a reliable means of validating the 1-D fluid representation model, which can then be used to run the more extensive compressor manifold and piping system. It is important to utilize a rigorous full Navier-Stokes fluid solution for high horsepower systems with larger pulsation amplitudes.

2. The 3-D finite element response model of the fluid system can be used to calibrate the one-dimensional fluid model for correct nozzle frequency response and gas passage response frequencies. The response model will predict less important responses as well and must be reviewed by an experienced engineer. Verification of the one-dimensional model must still be performed to verify gas passage volume similarity in the 1-D representation and correct damping due to the gas passage diameters and valve boundaries.

3. The miniature "virtual orifice" developed in the GMRC research program warrants further development. The model predictions showed a reduction in pulsation amplitudes of approximately 20:1 for the high frequency valve-to-valve response at 270 Hz, evident in Case B. The mini VO would likely be designed to fit on the head or crank end of the cylinder and resemble a small pocket unloader in its overall dimensions. The mini-VO would be much smaller than the previously

developed Virtual Orifice. It would be designed to tune out problematic higher frequencies in the gas passage to potentially reduce cylinder vibrations and increase valve performance.

5. References

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