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Keith Adam Novak

Ingersoll Rand - Trane, United States of America, keith.novak@trane.com

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Influence of Cylinder Bore Volume on Pressure Pulsations in a Hermetic Reciprocating Compressor

Keith NOVAK

Trane – Ingersoll Rand
La Crosse WI, USA
Phone: (608)787-4105, Keith.Novak@trane.com

ABSTRACT

Suction pressure pulsations created when the suction valve opens are caused by unsteady mass flow through the valve exciting acoustic resonances in the suction plenum. These pressure pulsations influence valve dynamics, compressor performance and compressor noise. This paper will show the importance of including the cylinder bore volume in the flow path analysis in order to accurately calculate pressure pulsations. Pressure pulsations will be calculated using Finite Element Method (FEM) calculated impedance transfer matrix method in a quasi-static solution. The method models the impedance of the suction plenum flow path including cylinder bore volume using a static geometry condition just after the suction valve opens. The interaction between the suction plenum, suction valve port geometry, and cylinder bore volume influence acoustic resonances in this system. These resonances cause pressure pulsations that effect valve dynamics. This paper uncouples the dynamics of the suction valve from the pressure pulsation modeling. This was done to specifically look at pressure pulsation created due to refrigerant volume flow through the suction valve and suction plenum acoustic resonances without valve dynamics influencing pressure pulsation. The paper then compares simulated suction plenum pressure pulsations when the valve opens to measured pressure pulsation in a reciprocating residential HVAC compressor with good agreement.

1. INTRODUCTION

Suction pressure pulsations in a reciprocating compressor influences the compressor's performance and noise. Acoustic theory was first used by Elson and Soedel (1972) to predict pressure pulsations from dynamic volume flow of refrigerant into and out of the compressor's compression mechanism. This theory was refined in the early 1970's by J. Elson, R. Singh and W. Soedel in numerous publications. The theory was developed into a four pole method based on one dimensional acoustic wave theory and this method is found in compressor design textbooks. The four pole matrix is created by assembling a combination of basic 1D acoustic element consisting of mathematical equations for specific lengths and diameters of pipes, and small volumes to represent plenum geometry. Influences of suction and discharge plenum pressure pulsations on compressor performance is accomplished by incorporating the four pole method into the 1D compressor thermodynamic software and iterating on pressure pulsations and mass-flow until a converged solution (Zhou *et al*. 2001).

Wu and Zhang (1998) proposed an improved method to calculate the four pole parameters in a three dimensional (3D) plenum using Boundary Element Method (BEM) to calculate transmission loss in a muffler. Zhou (1998) used surface source excitation to calculate the four poles in a 3D plenum. Both papers added short tubes to the inlet and outlet plenum geometry. This was done to create planar waves, and reduce the pressure variations over the inlet and outlet surface. This allowed the pressure response functions to be taken at a point on the inlet and outlet tube end. Kadam's (2005) work calculated pressure response functions using FEM and then compared the FEM calculated four poles to analytical calculated and experimental measured four poles.

In pressure pulsation analyses additional geometry added to the inlet and outlet of the plenum will influence impedance calculations and produce incorrect impedance transfer functions. Short tubes should not be added to the inlet or outlet of the plenum in pressure pulsation calculations because these added tubes influence or create acoustic

resonances, and cause incorrect pressure pulsations. In the same respect, if short tubes influence or create acoustic resonances so will cylinder bore volume. So, by not including cylinder bore volume in the impedance analysis the cylinder bore volume will not influence acoustic resonances and cause incorrect pressure pulsation predictions. This paper analyzes the influence of cylinder bore volume on suction pressure pulsations, specifically the high frequency pressure pulsations developed at suction valve opening. Due to the motion of the piston and the impedance calculation requiring the cylinder bore volume to be fixed this problem is solved as a quasi-static problem. The analysis solves for impedance by fixing the cylinder bore volume at a crank angle just after the suction valve opens. Additional crank angle positions were also analyzed to determine the impedance function sensitivity due to cylinder bore volume. The pressure pulsation analysis used in this paper decouples the pressure pulsations from valve dynamics. This was done to determine only the influence of refrigerant volume flow and acoustic plenum resonances on pressure pulsations. The paper compares the calculated suction pressure pulsations at suction valve opening to measured pulsations with good agreement. Due to changes of refrigerant sonic velocity as the refrigerant travels through the suction plenum this analysis only focuses on pulsation when the valve opens. The refrigerant entering the suction plenum has a lower sonic velocity than the refrigerant inside the cylinder bore volume. The FEM model assumed constant sonic velocity and scaled impedance function. This causes slightly inaccurate pulsation predictions at lower frequency pulsation. Improvements could be made to the pulsation analysis by solving the FEM model by mapping the actual sonic velocity along the suction flow path.

2. MODEL

In order to accurately calculate pressure pulsations, the 3D CAD geometry needs to represent the actual flow path geometry. Simplification in the 3D geometry often results in poor pulsation simulations. Figure 1 shows the suction plenum geometry specifically looking at the suction plenum near the suction valve. The outlet area highlighted in red is the suction valve passage. The suction valve passage area is the location where the FEM flow load is applied for both geometry configurations analyzed. The four pole method calculates pressure pulsations at the inlet and outlet of the plenum; however pressure transducers cannot be mounted at these locations. This requires needing addition locations in the four pole method or impedance method. Expanding the 2x2 matrix into a 3x2 matrix allows for the inclusion of P3 location. The area B (PT_1) is the location of the pressure transducer in the suction plenum and the location where the transfer impedance P3 is simulated in equation 1.
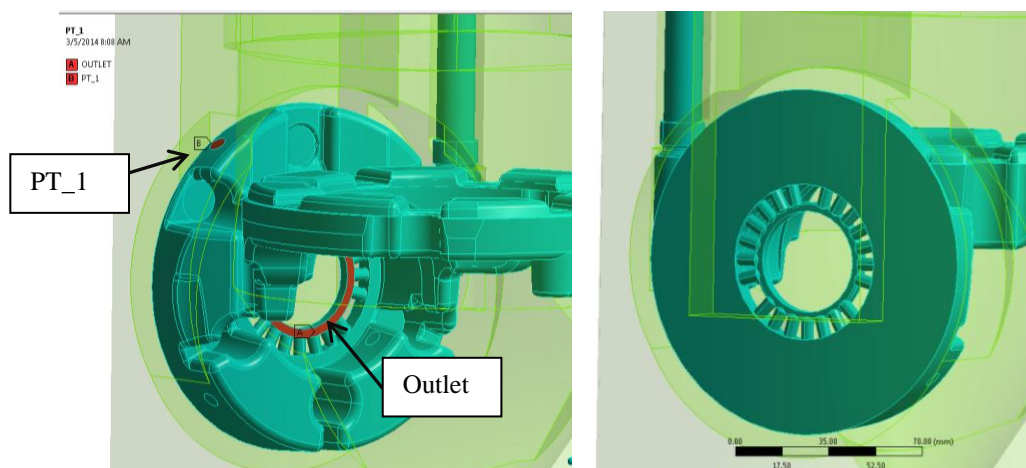

Figure 1. Suction plenum geometry CAD without cylinder volume.

Figure 2 show the inclusion of the cylinder bore volume at a piston position when the suction valve is fully open. The green highlighted geometry is the cylinder bore volume. This volume interfaces with the suction plenum at the suction valve passage. The systems impedance calculation is a quasi-static solution because the cylinder bore volume is an ever changing volume due to the motion of the piston. The piston position and cylinder bore volume is fixed during FE analysis and subsequent pressure pulsation analyses. The translucent green volume is the volume inside the shell which is included suction plenum CAD geometry.
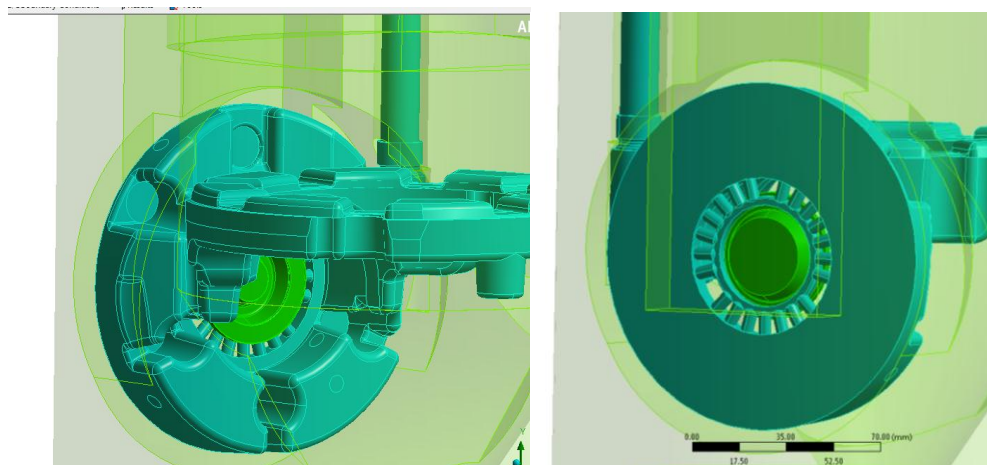
Figure 2. Suction plenum geometry CAD with cylinder volume when the valve is fully open

3. Pressure Pulsations Calculations

Suction pressure pulsations are calculated by multiplying the FEM calculated transfer impedance matrices by the volume flow of refrigerant through the suction valve, equation 1, at the compressor speed harmonics. The 3D impedance transfer function is solved using FEM harmonic acoustic analyses over the frequency range of interest. The method to calculate the impedance of the suction plenum is similar to the process defined in Wu and Zhang (1998). The FEM flow load applied over the valve area in Figure 2 is the same area as Figure 1 geometry. The impedance transfer functions are calculated by dividing the average pressure over the area by the FEM flow load, as shown in equation 2. Due to the harmonic nature of the refrigerant flow the frequencies of interest are at the harmonics of the compressor speed. The transfer impedance matrix is a reduced form of the impedance transfer function, where the matrix coefficients are at the compressor harmonic frequency. The first 20 compressor harmonics are used to calculate pressure pulsations in order to capture higher frequencies content. To determine pressure pulsations at a location other than inlet or outlet an extra row is added to the impedance matrix. This extra row is used to calculate the pressure pulsations at a pressure transducer location (PT_1), as shown in figure 1. The matrix can be expanded further by adding additional rows for additional pressure measurement locations. The refrigerant volume flow through the valve is determined through 1D thermodynamic compressor simulations and converted by Fast Fourier Transformation (FFT) into frequency domain. Normally, the impedance transfer matrices are incorporated inside a 1D thermodynamic compressor program to determine pressure pulsations influences on performance and valve dynamics. However, this paper decouples the pressure pulsations calculation from the valve dynamics to show that the pressure pulsations created when the suction valve opens are independent of valve dynamics. This was done by solving the 1D thermodynamic compressor program to determine refrigerant volume flow through the suction valve without including plenum pressure pulsation or higher order valve dynamics. Essentially the suction valve ramps open, stays open and then ramps closed. The FFT of volume flow produce volume flow only at the harmonics of the compressor speed. Pressure pulsations at the harmonic frequencies are calculated by multiplying the impedance matrices by the volume flow in frequency domain. An Inverse Fast Fourier Transformation (IFFT) on the pressure pulsations in the frequency domain converts the pressure pulsations into time domain.

$$\begin{Bmatrix} P_1 \\ P_2 \\ P_3 \end{Bmatrix} = \begin{bmatrix} f_{11} & f_{12} \\ f_{21} & f_{22} \\ f_{31} & f_{32} \end{bmatrix} \begin{Bmatrix} Q_1 \\ Q_2 \end{Bmatrix} \qquad (1)$$

$$f_{11} = \frac{Pressure\ (average)}{Flow\ Load\ (at\ valve)} \qquad (2)$$

4. PRESSURE MEASUREMENTS

Figure 3. Pressure transducers mounted to suction plenum.

A hermetically sealed 3 ton R410a residential HVAC variable speed reciprocating compressor was used to study pressure pulsations in the suction plenum as shown in Figure 3. Pressure pulsations were measured at compressor speeds of 1800 rpm to 4500 rpm at 120 rpm increments. The piston diameter and stroke is approximately 47mm and 11mm, respectively. Pressure measurements inside the suction plenum were acquired with B&K Pulse data acquisition system using a Kistler pressure transducer. The pressure transducer was mounted flush to the wall of the suction plenum. Sampling frequency of the data acquisition was 25,600 Hz. Data was filtered with a low pass filter to remove high frequency noise.

5. RESULTS

Figure 4 shows the impedance transfer functions at the suction plenum outlet or suction valve for the two geometries, when excited by a flow load at the outlet. The blue line is the input impedance function for the geometry without the cylinder bore volume and the lime-green line is for the geometry with the cylinder bore volume. The geometry without the cylinder bore volume calculates an acoustic resonance in the system at approximately 2100 Hz. However, when the volume of the cylinder bore is included in the analysis the resonance is approximately 1100 Hz. The cylinder bore volume geometry reduces the stiffness of the acoustic impedance at the suction valve. The resonance is due to the volume and geometries of the cylinder bore, the suction valve passage, and suction plenum. Figure 5 shows the sensitivity of the impedance transfer function to multiple cylinder bore volume. The larger the cylinder bore volume the lower the resonance frequency. Figure 5 also shows the impedance transfer function below 800 Hz are similar for all geometries analyzed. This shows that the first 10 pressure pulsations harmonics in a 60 Hz compressor are not influenced by the cylinder bore volume. In compressor performance modeling generally only the first few harmonics have the greatest effect on performance and cylinder bore volume can be excluded from the analysis. However, in acoustic noise modeling the cylinder bore volume is important to include in the geometry model to capture all excitable frequencies.
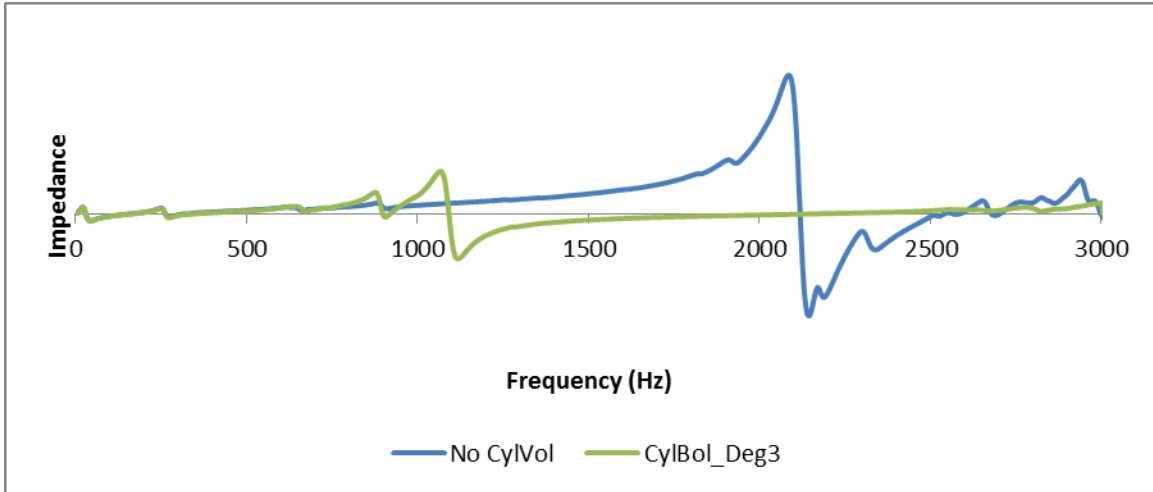
Figure 4. Imaginary Component Impedance Transfer Functions at Outlet (Valve)
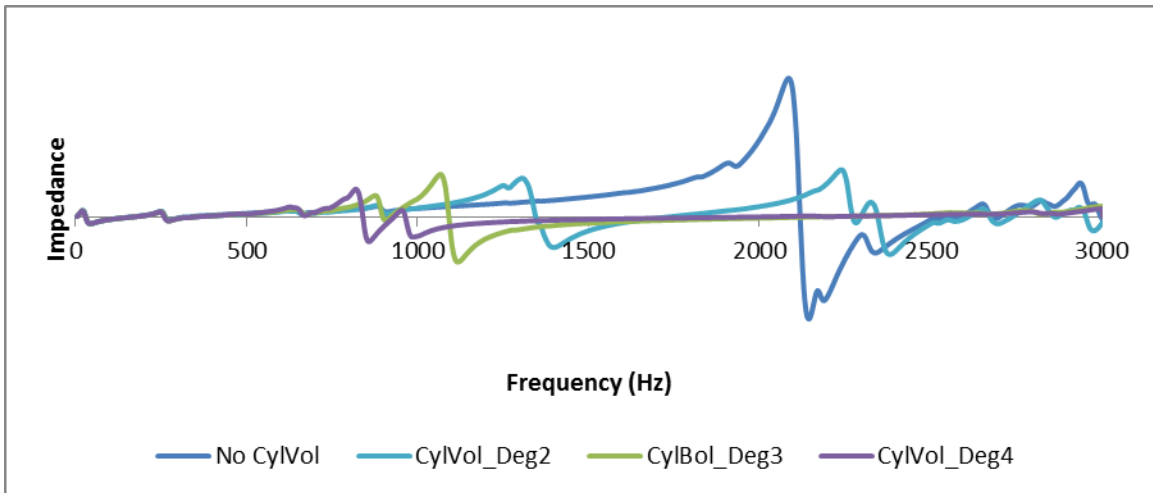

Figure 5. Imaginary Component Impedance Transfer Functions at Outlet (Valve)

Figure 6 shows an FEM calculated impedance plot near the 1100 Hz resonance frequency for the model with the cylinder bore volume. The resonance is an interaction between the cylinder bore volume, suction valve port geometry, and the suction plenum, as seen on the figure. This plot shows the pressure on the piston top will have the opposite sign or be out of phase by 180 degrees from the pressure transducer at this frequency. Since compressor performance is calculated by the work done by the piston; calculating and using this pressure at the piston top in the 1D compressor simulation software to calculate performance will result in a better modeling scheme than using pressure at the valve location.
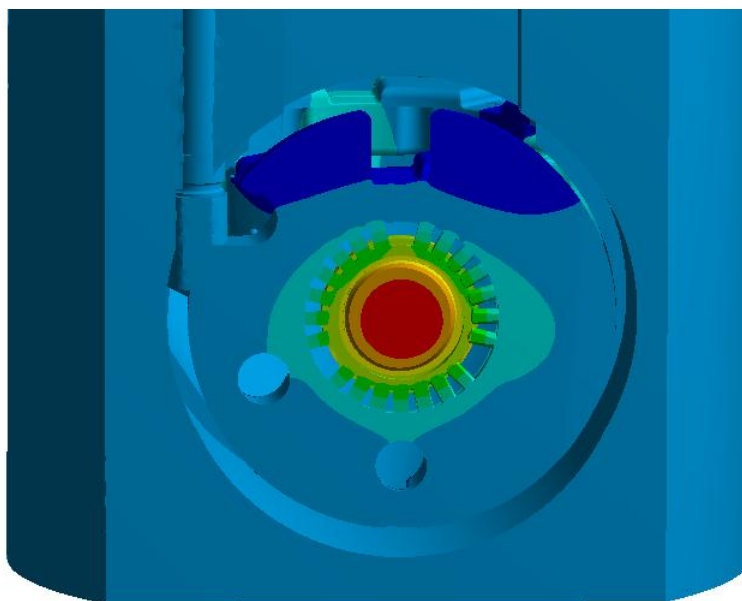

Figure 6. Impedance Distribution Plot near 1100 Hz Resonance Frequency when excited by volume flow load at outlet of suction plenum.

Figure 7 and 8 shows the measured pressure pulsations versus calculated pressure pulsations on geometry without and with cylinder bore volume, respectively, at different compressor speeds. When the suction valve opens the dynamics of refrigerant volume flow excite the acoustic resonance of the system. This resonance causes pressure fluctuations near the valve location and inside the cylinder. Due to the motion of the piston and changes in cylinder bore volume the impedance transfer function changes with crank angle. A quasi-static solution is required to solve this problem, however this results in loss of accuracy in pressure pulsation modeling as the cylinder bore volume expands past the quasi-static geometry. Figure 8 shows the pressure pulsations have good agreement when the valve first opens but as the cylinder volume increases the impedance of the system changes and lowers the resonance of the system the pulsations don't match as well.
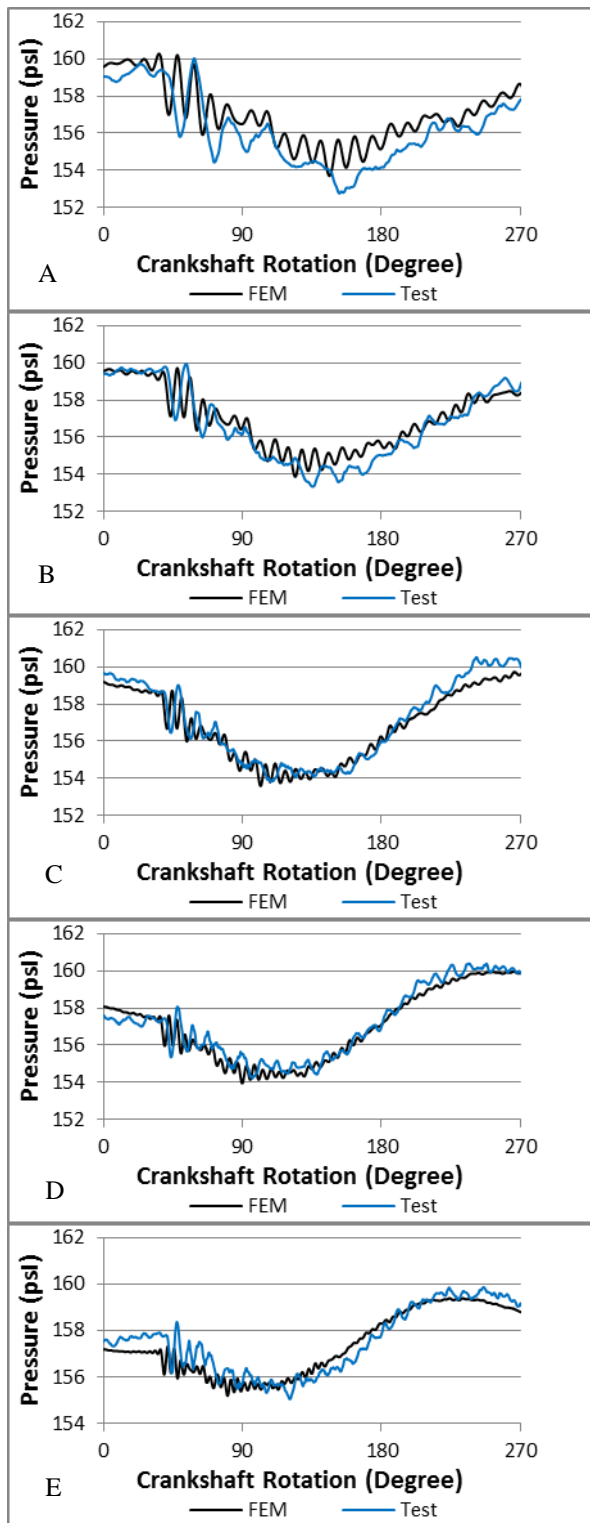
Figure 7. FEM calculated pressure pulsation without cylinder bore volume and measured pressure pulsations at various speeds
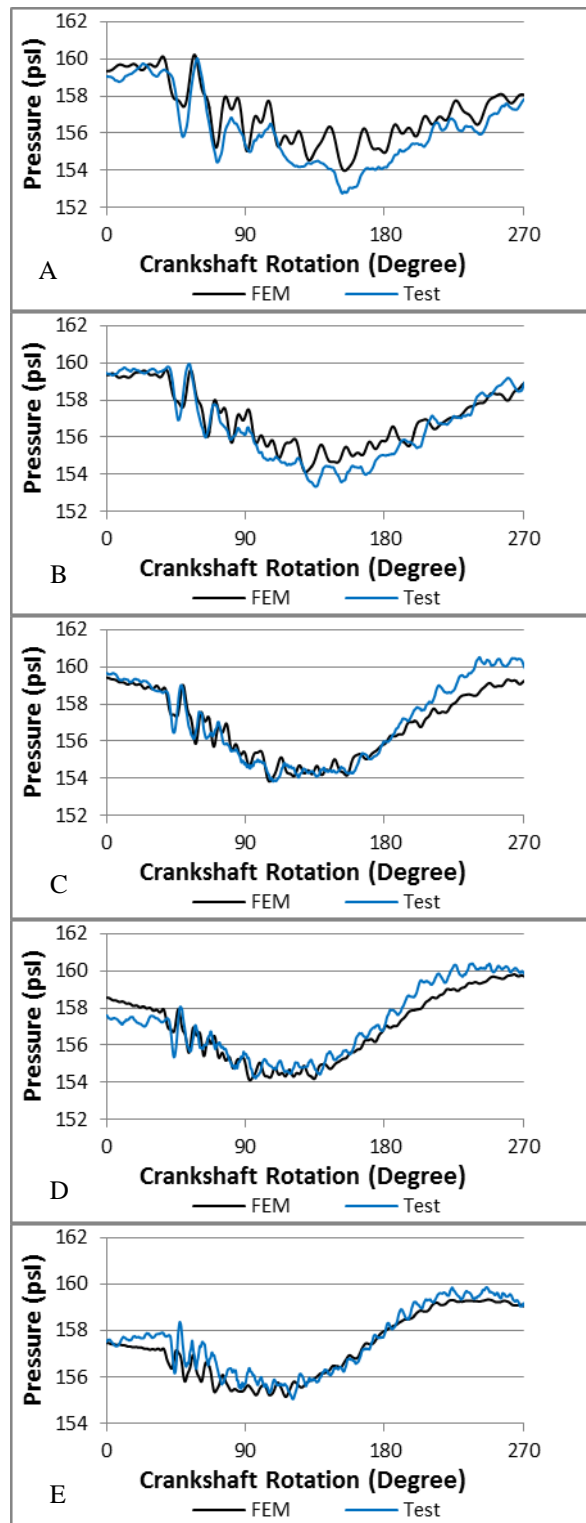
Figure 8. FEM calculated pressure pulsations with cylinder bore volume included in the model, and measured pressure pulsations at corresponding speeds to Figure 7

6. CONCLUSIONS

This paper shows the significance to include the cylinder bore volume in the analysis in order to calculate the pressure pulsations when the valve opens. A quasi-static FEM calculated impedance transfer method is used to calculate pressure pulsation by fixing the cylinder bore volume at a position when the suction valve is fully open. The suction pressure pulsation were decoupled from the analysis to determine the influence of the refrigerant volume and acoustic resonances on pressure pulsations. The combination of suction plenum, suction valve port geometry, and cylinder bore volume create an acoustic resonance in the system. This resonance creates high frequency pressure pulsations when the valve opens. The simulated pressure pulsations are in good agreement with the measured pulsations at the pressure transducer location when the valve first opens. However, as the cylinder volume expands during suction process the resonance frequency changes. This causes the simulated pressure pulsations and measured pressure pulsations to not match later in the suction process.

Whether the analysis is 1D acoustic wave model, FEM impedance transfer matrix modeling or CFD; all three approaches solve the wave equation in order to predict pressure pulsations. When a geometry model is simplified it is important to verify the simplification does not change the acoustics in the model over the frequency of interest. High frequency resonances are generally created by geometry near the valve. The further the geometry is from the valve the lower the frequency it effects. This is due to the acoustic frequency wavelength interacting with the plenum geometry. The lower the acoustic frequency the longer the wavelength is and its interaction with the plenum geometry. A complementary paper written by the author (Novak, 2014) study the effects of the compressor shell volume on suction pressure pulsations.

NOMENCLATURE

P   Pressure
Q   Volume Flow
f   Impedance Function
Z   Impedance

Subscript
1   Inlet
2   Outlet

REFERENCES

Bilal, N., Adams, D., Novak, K., Sauls, J., 2010, A Hybrid Approach of Calculating Gas Pulsations in the Suction Manifold of a Reciprocating Compressor, *Proceedings of the 2010 International Engineering Conference at Purdue, Purdue University*; *July, 2010.*

Elson, J., Soedel, W., 1972, A Review of Discharge and Suction Line Oscillation Research; *International Compressor Engineer Conference, Purdue University*, Paper 49, http://docs.lib.purdue.edu/icec/49

Kadam, P. H., 2005, *Development and Comparison of Analytic, Numerical and Experimental Techniques to formulate Four-Poles Matrices of Three Dimensional Acoustic Systems,* MSc. Thesis, University of Cincinnati

Novak, K., Sauls, J., Comparing FEM Transfer Matrix Simulated Compressor Plenum Pressure Pulsation to Measured Pressure Pulsations and to CFD Results; *Proceedings of the 2012 International Engineering Conference at Purdue, Purdue University*, *July, 2012.*

Sauls, J., Novak, K., Incorporating 3D Suction and Discharge Plenum Geometry into a 1D Compressor Simulation Code to Calculate Compressor Pulsations, *Proceedings of the 2012 International Engineering Conference at Purdue, Purdue University*; *July, 2012.*

Singh, R., Soedel, W., 1979, Mathematical Modeling of Multi-cylinder Compressor Discharge System Interactions, Journal of Sound and Vibration, vol. 63, no. 1: p. 125-143.

Singh, R., Soedel, W., 1974, A Review of Compressor Lines Pulsation Analysis and Muffler Design Research -Part I: Pulsation Effects and Muffler Criteria; *International Compressor Engineer Conference, Purdue University*, Paper 106, http://docs.lib.purdue.edu/icec/106

Singh, R., Soedel, W., 1974, A Review of Compressor Lines Pulsation Analysis and Muffler Design Research -Part II: Pulsation Effects and Muffler Criteria; *International Compressor Engineer Conference, Purdue University*, Paper 107, http://docs.lib.purdue.edu/icec/107

Wu, T. W., Zhang, P., 1998, Boundary Element Analysis of Mufflers with an Improved Methods for Deriving the Four Pole Parameters, *Journal of Sound and Vibration,* vol. *217, no. 4: p. 767-779.*

Zhou, W., Kim, J., Soedel, W., 2001, New Iterative Scheme in Computer Simulation of Positive Displacement Compressors Considering the Effects of Gas Pulsations, *Transactions of the ASME,* Vol. *123, June 2001: p. 282-288.*

Zhou, W., Kim, J., 1998, Formulation of Four Poles of Three-Dimensional Acoustic Cavities Using Pressure Response Functions with Special Attention to Source Modeling, *International Compressor Engineering Conference.* Paper 1297. http://docs.lib.purdue.edu/icec/1297