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Influence of Shell Volume on Pressure Pulsations in a Hermetic Reciprocating Compressor

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ABSTRACT

In a hermetic reciprocating compressor the suction pressure pulsations affect compressor performance and noise. The pressure pulsations are created due to the dynamic flow of refrigerant through the suction valve into the cylinder bore. One dimensional (1D) acoustic method, Finite Element Method (FEM) calculated impedance transfer matrix method, and Computational Fluid Dynamics (CFD) are three commonly used methods to calculate these pressure pulsations. With CFD being used most often to solve this type of analysis; however CFD is often time consuming and requires significant computer resources. In order to solve the pressure pulsation analyses faster, the suction plenum geometry is often simplified by reducing the model size. This simplification could lead to inaccuracies in pressure pulsation modeling. This paper will use the FEM calculated impedance transfer matrix method to analyze a hermetic HVAC reciprocating compressor to show that geometry simplifications could result in poor predictions of pressure pulsation. The FEM calculated impedance transfer matrix method was chosen due to its short solution time and the inherent ability to compare similarities and differences in the plenum's impedances between the full geometry model and a simplified geometry model. This paper will specifically look at the influence of the compressor shell volume on calculating suction pressure pulsations. In the compressor analyzed, the compressor shell volume influences the pressure pulsation at the compressor's first harmonic speed. A pressure pulsation at the first harmonic has the greatest influence on the compressor performance. This paper will compare measured pressure pulsations to simulated pressure pulsations. The pressure pulsations are in good agreement when the full model is analyzed but when simplifications are done to the model it fails to provide 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 elements 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 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.

Novak and Sauls (2012) replaced the four poles calculated from basic 1D acoustic elements with FEM calculated impedance transfer matrices and incorporated this into their 1D compressor thermodynamic software to simulate pressure pulsations. They modeled the actual suction and discharge flow path plenum geometry without adding short tubes. Then they averaged the response pressure over the inlet or outlet area to calculate average impedance. They validated this technique by comparing the simulated pressure pulsations to measured pressure pulsations in an HVAC reciprocating compressor with good agreement. They also showed the FEM calculated transfer impedance pressure pulsation method produced similar pressure pulsations as CFD in a test case of a two cylinder suction plenum.

CFD is commonly used to solve pressure pulsations in complex suction or discharge plenums and is generally accepted to model pressure pulsations. CFD models the transient flow of refrigerant through the three dimensional suction or discharge plenums in time domain. CFD can be modeled as simple as pulsating mass-flow applied at the valve with flow through the plenum geometry to a complex model with moving mesh to model valve dynamics and piston motion. CFD is generally a computationally expensive analysis. In order to reduce the computation expense, the analysis is often simplified by increasing time step intervals or simplifying the geometry. Both simplifications could influence the accuracy of the modeling.

This paper will expand on the FEM calculated impedance transfer matrix method used in Novak and Sauls (2012) and provide a method to help understand consequences in geometry simplification. The consequences can be seen as the difference between impedance transfer functions on different plenum geometries. This paper will specifically analyze the influence of shell volume versus not including the shell volume on suction pressure pulsations. The differences in the impedance transfer functions between these two models occur at a relatively low frequency. This is because the differences in plenum geometry are further away from the suction valve and influence the longer acoustic wavelength. A complementary paper written by the author (Novak, 2014) study the effects of cylinder bore volume on suction pressure pulsations.

2. PULSATION MODEL

Figure 1 represents the fluid path on the suction side of a HVAC hermetically sealed reciprocating compressor. Refrigerant enters the shell volume through the suction line at the shown inlet area to the suction plenum. The shell volume acts as a reservoir for oil and refrigerant and is shown as a semi-transparent volume. The refrigerant inside the shell volume then flows across the top cap of the electric motor and down a suction tube into the annular plenum in front of the suction valve and through the suction valve or outlet area. The basic design of the suction plenum side is a plenum-tube-plenum configuration. Figure 2 represents the simplified plenum model, where the shell volume is removed from the analysis. The FEM calculated transfer impedance method requires two FE analysis runs to determine the influence of refrigerant volume flow on pressure pulsations. The FEM excitation flow load is applied to the inlet area for run 1 and is applied at the outlet area for run 2, as shown in Figure 1 and 2. A full harmonic response analysis is solved over the frequency range of interest with small frequency steps. The impedance functions are calculated as the average area pressure response at the inlet or outlet divided by the FEM flow load as shown in equation 1 at each frequency. The impedance matrix is a reduced form of the impedance function at the compressor harmonic frequencies, equation 2. The impedance matrix was permutated to calculate the four pole coefficient, equation 3. Using the average pressure at the inlet and outlet area account for planar and transverse waves and can be best seen by calculating the determinant of the four pole matrix. When the determinant at each frequency of the four pole matrix is not equal to one, then the analysis or model is insufficient to proceed. However, with a uniform mesh on the surface area of the inlet and outlet and using the average pressure to calculate the four pole coefficient the determinant of the four pole matrix should be equal to one. By averaging the pressure over the inlet or outlet area this accounts for both planar and transverse waves and short tubes at the inlet and outlet to force planar waves are not required. The author's analyses have shown these short tubes produce errors in pressure pulsation modeling. These short tubes create additional resonances in the system corresponding to the length of the tube, and produce a time delay equal to the speed of sound multiplied by the length of tube.

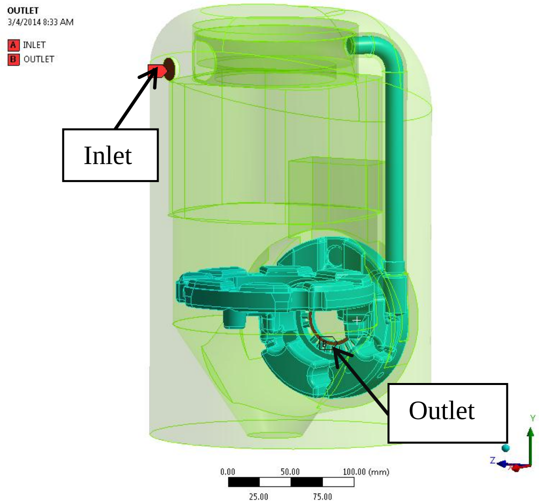

Figure 1 Suction plenum geometry from FEM model with-shell volume. Internal shell volume is semi-transparent.

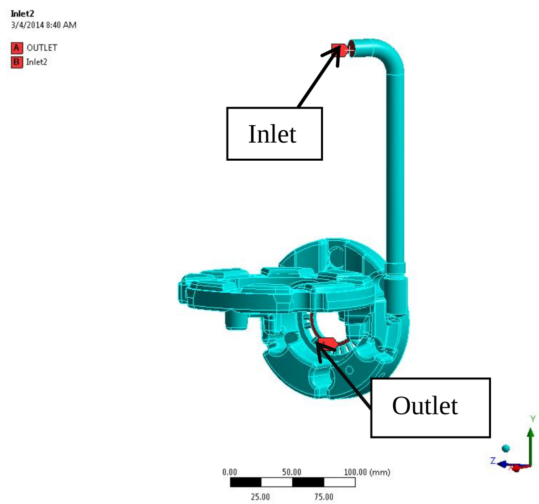

Figure 2. Suction plenum geometry from FEM model without-shell volume

$$f_{11} = \frac{Pressure\ (average)}{Flow\ Load(at\ valve)} \tag{1}$$

$$\begin{Bmatrix} P_1 \\ P_2 \end{Bmatrix} = \begin{bmatrix} f_{11} & f_{12} \\ f_{21} & f_{22} \end{bmatrix} \begin{Bmatrix} Q_1 \\ Q_2 \end{Bmatrix} \tag{2}$$

$$\begin{Bmatrix} Q_1 \\ P_1 \end{Bmatrix} = \begin{bmatrix} A & B \\ C & D \end{bmatrix} \begin{Bmatrix} Q_2 \\ P_2 \end{Bmatrix} \tag{3}$$

Pressure pulsations are the result of acoustic standing waves excited inside the suction plenum by the volume flow of refrigerant through the suction valve. In order to include low frequencies into the pulsation simulation the CAD model length should represent at least the half wavelength of the lowest frequency of interest. For a 60 Hz reciprocating compressor with the speed of sound of the refrigerant the 180 meters per second the wavelength is 3 meters. The first frequency of interest is the reflective wave based on this wavelength. In order to capture this wave the model plenum length should be longer than half the wavelength or 1.5 meters. This length is often deemed impractical and simplifications are done to the plenum geometry to reduce the length. These simplifications may truncate or add additional resonance frequencies resulting in poor pressure pulsation predictions.

Anechoic termination may also be applied to the inlet as shown in equation 4 (Zhou, *et al.* 2001) to relate pressure pulsations at the boundary area to volume flow at the boundary area. This boundary condition will not correct for the missing or additional resonance frequencies due to a reduced plenum length. As the plenum geometry length is reduced the anechoic termination has greater influence on the pressure pulsations. The author has used the term "anechoic ratio" (equation 5) to quantify a frequency range of acceptable impedance transfer functions. The author defines the term "anechoic ratio" as ratio of C/A : D/B using the four poles coefficients. When the anechoic ratio approaches the value one, over the frequency range of interest, the model produces pressure pulsations that are not influenced by the anechoic termination. Therefore, fundamentally the longer the geometry plenum models the better the result. The anechoic ratio is used to determine the length of the suction plenum geometry needed to provide good results. In a large geometry model the pressure pulsations near the valve are predominately influenced by refrigerant flow through the valve or outlet area. This allows the impedance transfer functions or impedance plots calculated by flow load applied to the outlet area be directly used to predict pressures in the system.

$$Z_0 = \frac{(S_a C_T + c_0 \rho_0 D_T)}{(S_a A_T + c_0 \rho_0 B_T)} \tag{3}$$

$$AnechoicRatio = \frac{C_T}{A_T} : \frac{D_T}{B_T} \tag{4}$$

3. PRESSURE MEASUREMENTS

Figure 3. Pressure transducer 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 47 mm and 11 mm, 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.

4. RESULTS

The impedance transfer functions with the shell volume and simplified model without-shell volume are shown in Figure 4. The functions are very similar over the frequency of interest; however at frequencies less than 100 Hz the two functions become dissimilar, as shown in Figure 5. The model with the shell volume calculates a resonance at 35 Hz whereas the simplified model does not calculate a resonance. This resonance is due to the annular plenum volume, the tube connecting the annular plenum to shell volume, and the shell volume, as shown in Figure 6. This figure is the FEM impedance plot excited at the outlet by an excitation flow load near 35 Hz. Without the shell volume the simplified model does not predict this resonance. This resonance is the most important resonance to influence compressor performance due to closeness to the primary frequency of the compressor speed.

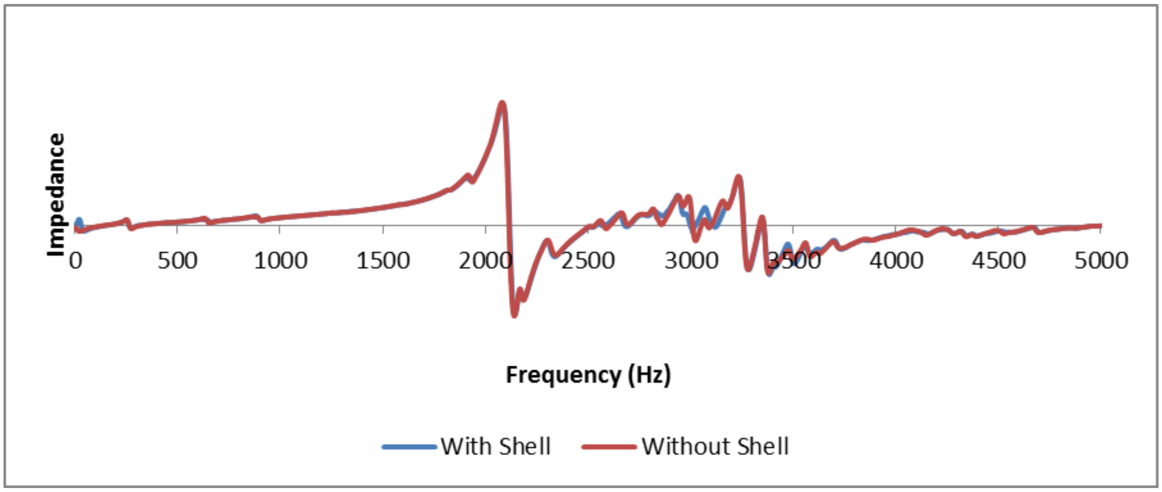

Figure 4. Imaginary Component of the Impedance Transfer Functions at Outlet (Valve)

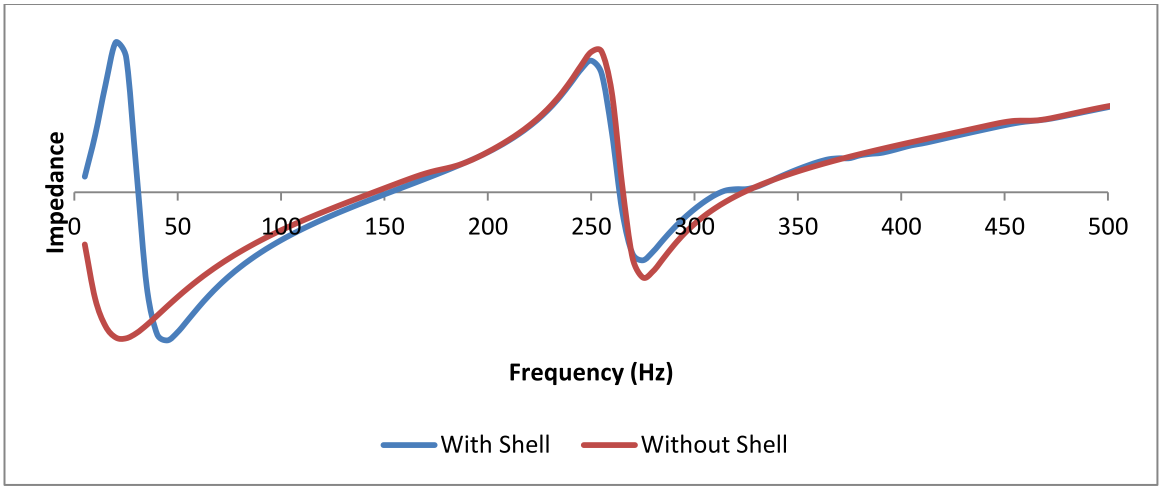

Figure 5. Imaginary Component of the Impedance Transfer Functions at Outlet (Valve)

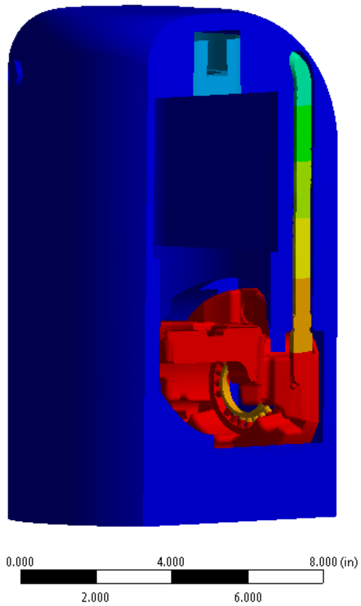

Figure 6. Impedance Plot near Resonance Frequency (35 Hz) due to Flow Load at Outlet (FEM Run 2)

Figure 7 shows the anechoic ratio as a function of frequencies for the two models analyzed and Figure 8 is over a reduce frequency range. When the anechoic ratio is not near the value one, these frequencies are influenced by the anechoic termination and the FEM model may not calculate pulsations accurately. Such accuracy is dependent upon assumed boundary conditions. In the analysis with-shell volume, the anechoic ratio shows only slight influence of the anechoic termination at 35 Hz.

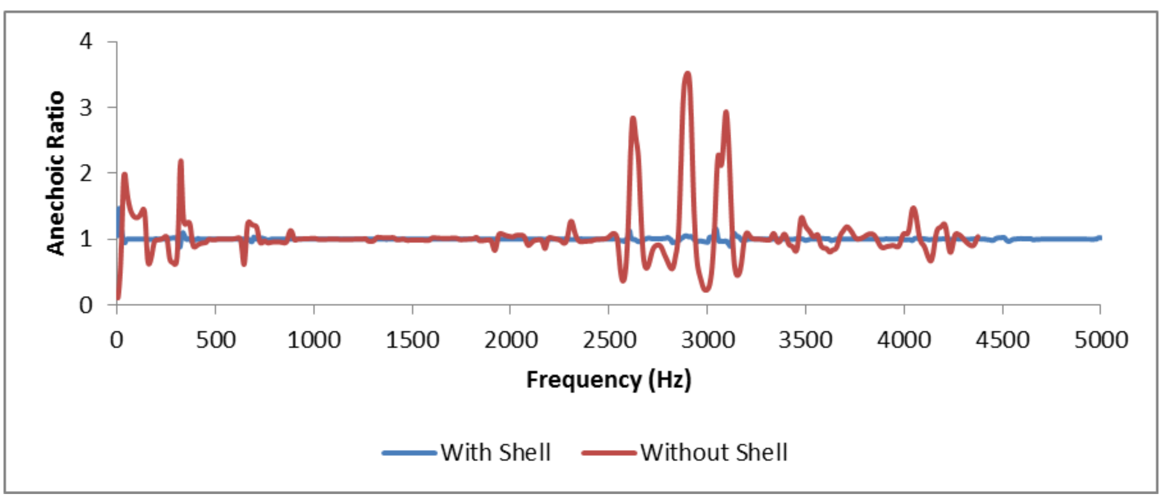

Figure 7. Comparing Anechoic Ratio's over the Frequency Range of Interest

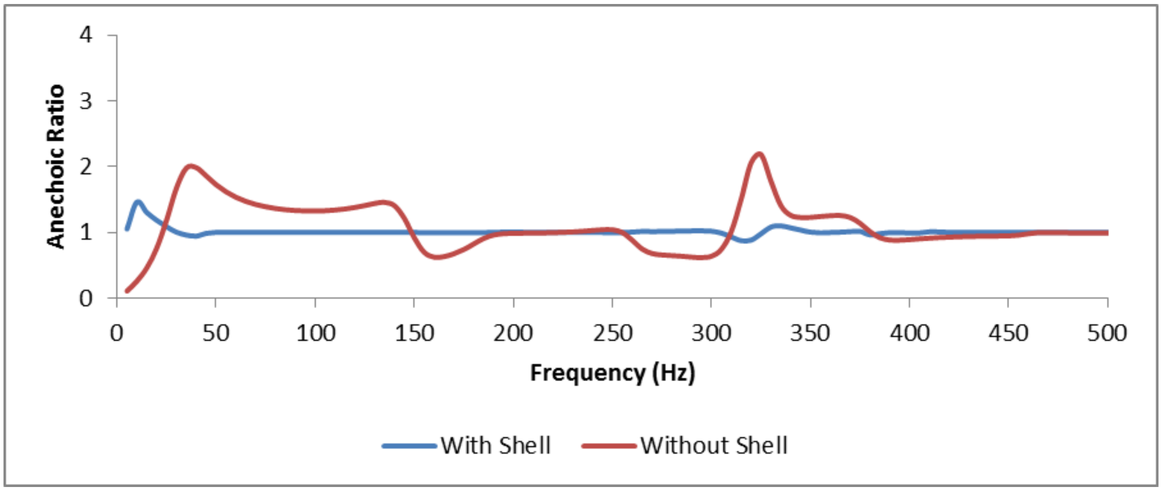

Figure 8. Comparing Anechoic Ratio's at Low Frequencies

Figure 9 shows the location of the calculated suction pressure in the FEM calculated transfer impedance analysis which is at the same location as the pressure transducer in Figure 3. Pressure pulsations at this location are simulated using the method defined in Novak and Sauls (2012) and limited to the first 10 harmonics of compressor speed. This limit was used to exclude cylinder bore volume pressure pulsation resonances which occur at higher harmonic frequencies. Figure 10 A, B, C, D, and E show comparisons between with-shell calculated pressure pulsations and experimentally measured pressure pulsations at various compressor speeds. The with-shell volume calculated pressure pulsation agrees with the measured pressure. Figure 11 A, B, C, D, and E show comparisons between without-shell calculated pressure pulsations and experimentally measured pressure pulsations. When the shell volume is not included in the analysis, the calculated pressure pulsation did not match measured pressure pulsation.

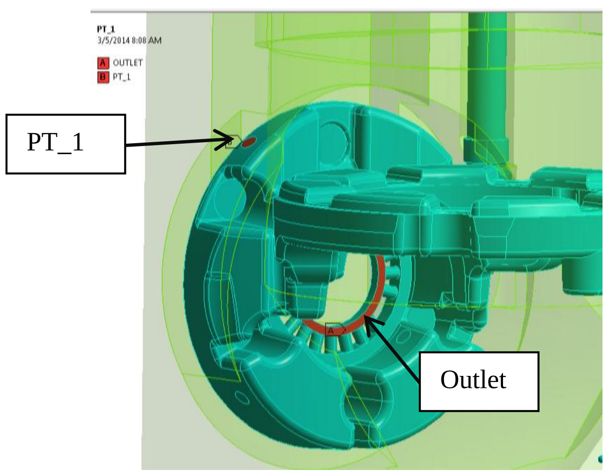

Figure 9. Location of Pressure Transducer (PT_1).

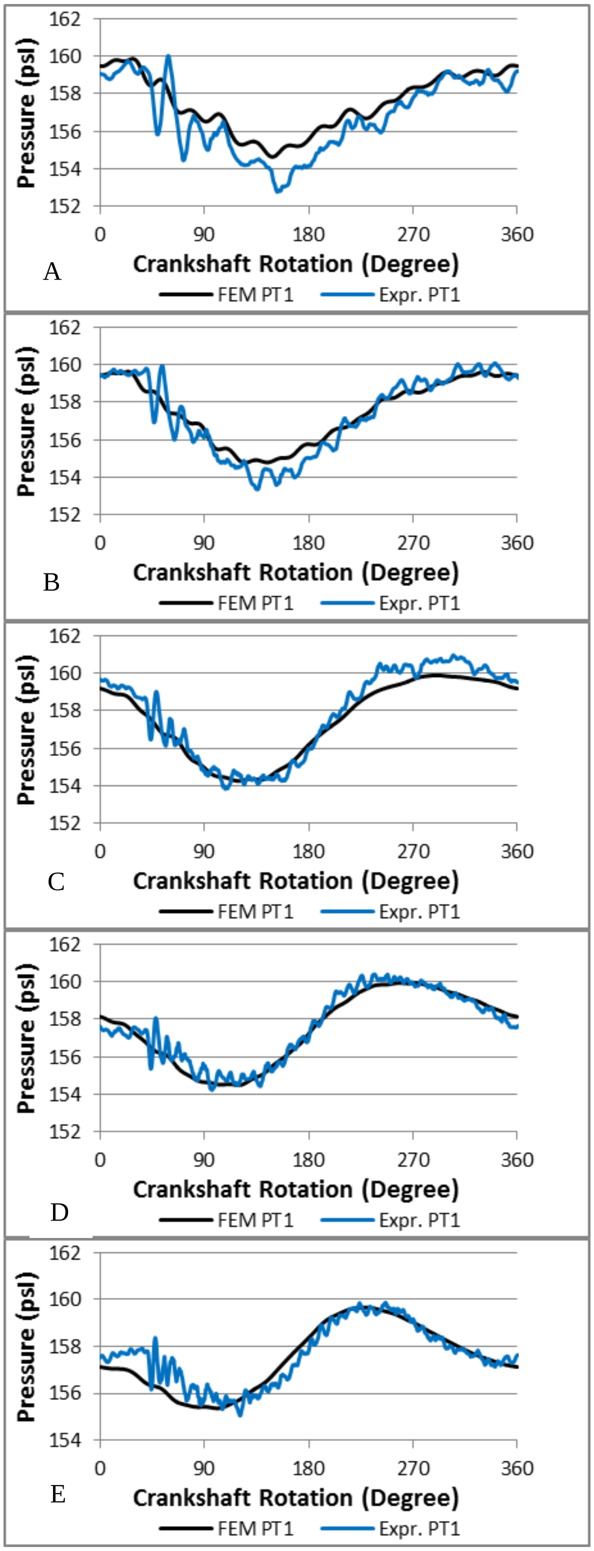

Figure 10. FEM calculated pressure pulsations with compressor shell volume and measured pressure pulsations at various compressor speeds

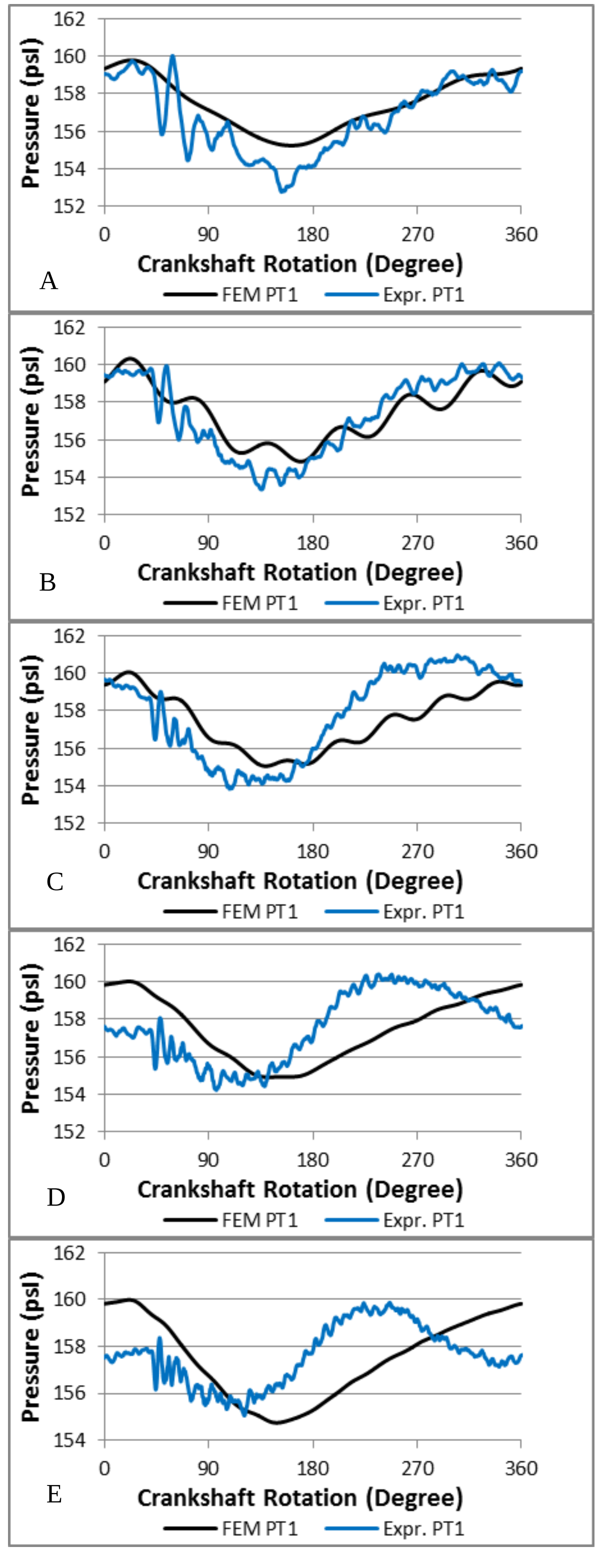

Figure 11. FEM calculated pressure pulsations without compressor shell volume and measured pressure pulsations at speeds corresponding to Figure 10

5. CONCLUSIONS

Simplifying the plenum geometry to reduce model size can produce erroneous pressure pulsation calculations. This paper has shown the volume inside the HVAC reciprocating compressor shell influences pressure pulsations due to acoustic resonance. The acoustic resonance is due to geometries of the suction plenum, the suction tube and the volume inside the compressor shell. Without these three geometries the resonance would not have been predicted. In the case analyzed, the resonance frequency is near the first fundamental of the compressor speed. This pressure pulsation frequency has the greatest effect on compressor performance. The impedance transfer functions inherently show the similarities and differences between the two models. At first glance both impedances seem very similar however the differences near the first fundamental frequency of compressor speed makes significant difference to the pressure pulsation calculations. The with-shell calculated pressure pulsations have good agreement with the measured pressure pulsations at the various compressor speeds measured. When the speed of the compressor approaches the resonance frequency near 35 Hz; the corresponding pressure waveform undergoes a phase shift in the pressure pulsations relative to crankshaft rotation. This pressure wave phase shift will influence the performance of the reciprocating compressor. By understanding the acoustic standing waves inside the suction plenum through FEM and their interaction with the refrigerant harmonic volume flow to create pressure pulsations; this will improve suction plenum design for a reciprocating compressor. The anechoic ratio helps evaluate the effects of simplification on the geometry model. When the anechoic ratio is near the value one the anechoic termination does not influence pressure pulsations. By evaluating the anechoic ratio and determining geometry model size based on this criterion the calculated pressure pulsations will have better agreement to measured pressure pulsations. The impedance transfer functions and anechoic ratio provide critical information in the plenum design and further analyses of suction or discharge plenums. These calculations should be done prior to CFD to determine acoustic resonances and plenum geometry size for the CFD analysis.

NOMENCLATURE

The nomenclature should be located at the end of the text using the following format:

P	Pressure
Q	Volume Flow
f	Impedance Function
Z	Impedance

Subscript

1	Inlet
2	Outlet

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