Numerical Prediction of Gas Pulsation in a Scroll Compressor Using 1-D Modeling: A Validation Study Based on AHRI Standard 530-2011

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ABSTRACT

Compressors are one of the core components in HVAC units in which efficiency and sound/vibration performance are imperative for optimal functionality. One of the factors that can impact both efficiency and sound/vibration performance of the compressor is gas pulsation. In a scroll compressor, the gas pulsation typically originates at the final scroll pocket, where the compressed gas discharges downstream. Along the flow path, the pulsation can be amplified or attenuated by various parameters such as boundary conditions, geometry changes, etc. This paper depicts a numerical model that predicts the discharge pulse of a scroll compressor. Commercially available software GT-SUITE was used to develop a 1-D model of the scroll compressor subassembly which includes scrolls and gas flow path components. AHRI Standard 530-2011 Rating of Sound and Vibration for Positive Displacement Refrigerant Compressors was used to verify the GT-SUITE model. The standard focuses on determining gas pulsation by spatially distributing pressure transducers along the discharge line. The standard emphasizes having an anechoic termination, with no variation in inside diameter or sharp bends in the discharge line. Emerson has developed a discharge pulse measurement procedure, in line with AHRI 530-2011, that is used to validate the GT-SUITE model. Since correlation with measurement data has been demonstrated, the GT-SUITE model is now used as the primary tool to determine the discharge pulse of scroll compressors, thereby reducing measurement time. Additionally, the model can be used to predict the discharge pulse across the compressor operating map and perform design optimization of the compressor and downstream flow path components.

1. INTRODUCTION

Pulsation in compressors is defined as pressure fluctuations in the discharge or suction line about some mean pressure. In scroll compressors, the gas is compressed in pockets that are formed by interaction between the orbiting and fixed scrolls. As the compression progresses, the gas is routed from the suction to the final scroll pocket where porting takes place. As the port opens, the compressed gas is discharged through an array of components as shown in Figure 1, including a shutdown valve, muffler plate, top cap plenum, and check valve, before reaching the discharge outlet. Figure 1 was originally developed for Emerson internal communication. Downstream of the discharge outlet, the compressed gas undergoes volume change and restrictions due to flow boundaries. As mentioned by Soedel (2006) in a scroll compressor, the highest-pressure fluctuation is caused when gas passes from the discharge pocket to the top cap plenum. This results in pulsation in the discharge line.

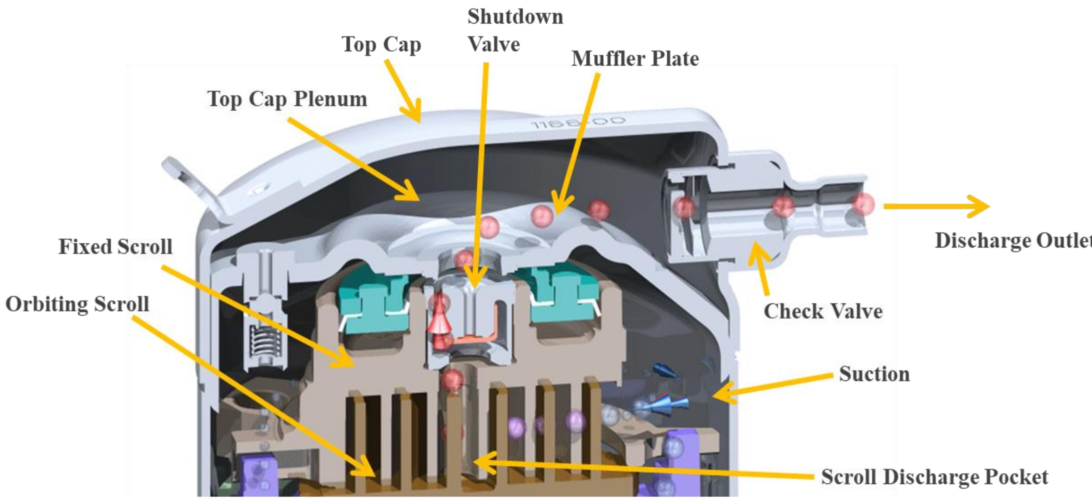

Figure 1: Compressor Components

The discharge pulse can be estimated as a function of two parameters: 1) the ratio of the scroll discharge pocket volume and the top cap volume and 2) the difference between the discharge plenum pressure and the scroll discharge pocket pressure. The scroll discharge pocket, shown in Figure 1, is the innermost compression volume, from where the porting is initiated. The top cap plenum is the volume between the muffler plate and the top cap where the downstream gas undergoes significant volume change before passing through the check valve to the compressor outlet. The general form of the equation is given below. The top cap plenum volume is determined by the top cap-muffler plate geometry. The other parameters are related to the pocket volume at the onset of porting which is a function of both the scroll and port geometry as indicated in Equation 1.

$$\text{Discharge pulse} = f\left(\frac{V_{sdp}}{V_{tc}}, P_d\text{-}P_{sdp}\right) \tag{1}$$

V_{tc} = top cap plenum volume
V_{sdp}= scroll discharge pocket volume
P_d = discharge plenum pressure
P_{sdp} = scroll discharge pocket pressure = $\left(\frac{V_{ssp}}{V_{sdp}}\right)^{\gamma} P_s$
P_s = suction pressure
V_{ssp} = volume of suction pocket
γ = ratio of specific heat

The scroll discharge pocket pressure is directly influenced by the system operating condition and building volume ratio. The operating map of a typical residential fixed speed scroll compressor is shown in Figure 2. The envelope presented in the figure is generated to meet the market demands at various condensing and evaporating temperatures. Note that there are regions of over- and under-compression that are separated by the design line. The discharge gas pulsation tends to increase in amplitude as the compressor operates away from the design line. Further, the running speed of the compressor also impacts the pulsation. This is imperative in the case of a variable speed compressor.

This paper predicts the discharge pulse of a fixed speed Emerson scroll compressor using a 1-D model. The commercial tool GT-SUITE from Gamma Technologies, LLC. was chosen to develop the model. Previous correlation studies done on the prediction of performance and pressure pulsations of positive displacement machines including automotive air conditioning scroll (Harrison et al., 2018) and swashplate (Ramchandran and Harrison, 2018) compressors, oil free and oil injected twin-screw compressors (Ramchandran et al., 2021), and two-phase twin-screw expanders (Vimalakanthan et al., 2020) provided confidence in selecting GT-SUITE as the tool of choice for this study. The termination boundary condition in the model is built consistent with Air Conditioning, Heating, & Refrigeration Institute AHRI Standard 530-2011. The gas compression cycle at various operating conditions is simulated to estimate the discharge pulse. The numerical results are validated by testing, and the results are presented in the later sections.

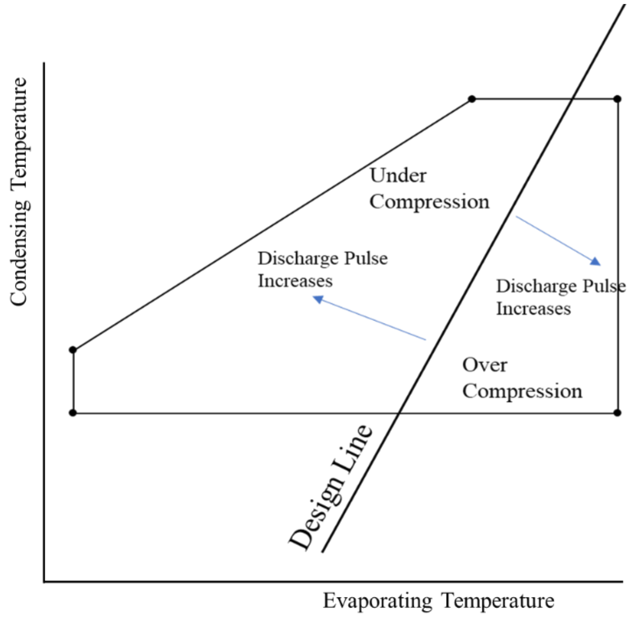

Figure 2: Typical Operating Map of a Residential Fixed Speed Scroll Compressor

2. 1-D MODEL – GT-SUITE

The schematic of the 1-D model is presented in Figure 3. A fixed speed 4-ton scroll compressor from Emerson is used in the mathematical model for analysis. The model is built similar to the 1-D scroll model developed by Harrison *et al*. (2018). Figure shows the model is divided into three segments.

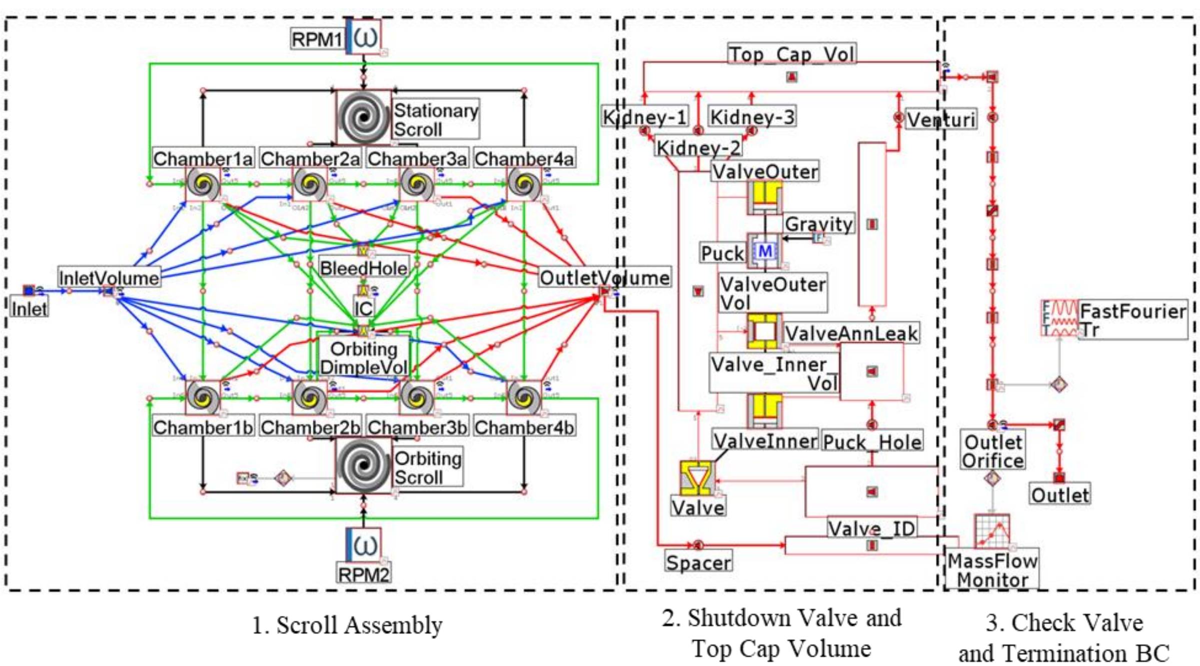

Figure 3: GT-SUITE 1-D Model of the Scroll Assembly and Downstream Components

The first segment contains the scroll assembly, where scroll pockets are modelled as variable-volume chambers. The suction volume, intermediate cavity, bleed hole, and discharge port volume are modelled as flow splits. This model assumes no tip or flank leakage during compression but does include leakage around the discharge end of the vane. The discharge port areas vs. crank angle are modelled from the CAD file (Harrison *et al*., 2018). The input is given to the model such that the pressures are estimated for 4 crank cycles and laid over a span of 1440°. Bleed hole and intermediate cavity effects during the compression cycle are included in the model. The model also considers how operating speed varies with condensing temperature. The pressure at the discharge pocket, predicted by the 1-D model at the cooling operating condition 50°F /115°F /70°F, is presented in Figure 4. Typically, the compressor operating condition is represented as Evaporating Temperature (Te)/ Condensing Temperature (Tc)/ Return Gas Temperature (Trg). 50°F /115°F /70°F is the rating condition for Air Conditioning (AC) as specified by AHRI 530-2011.

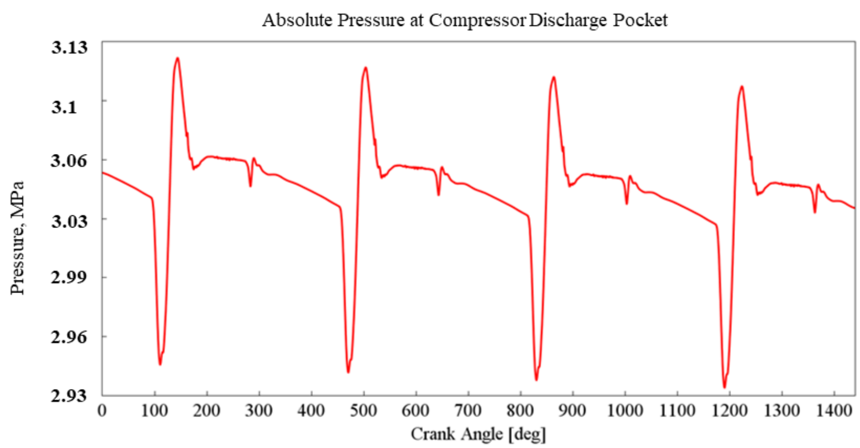

Figure 4: Discharge Pocket Pressure Predicted by the 1-D Model at 50°F /115°F /70°F

The second segment in figure 3 is comprised of the shutdown valve and the top cap plenum. The shutdown valve has a puck in the flow cavity that moves with the direction of the flow allowing the valve to range from fully open to fully closed during compression. The dynamic nature of the puck during the compression cycle is included in the model. The top cap plenum is modeled as a flow split based on the dimensions derived from the CAD model. From the plenum, the gas is routed to the third segment which includes the check valve, discharge line, termination boundary condition, and outlet.

The third segment of the 1-D model is shown in Figure 5 as a flow chart representation. The compressed gas passes through the check valve and enters the discharge line where the gas pulsation is measured. In GT-SUITE, the discharge line is modelled as a 0.0127m (0.5-inch) diameter pipe in which 4 pressure points, each 0.1524m (6 inches) apart are spatially positioned. Following the discharge line, a 15.24m (50-ft) long outlet pipe is modelled similarly to the measurement setup presented in the next section. The long outlet pipe is assumed to create an anechoic termination, so that standing waves are avoided in the discharge line due to reflection.

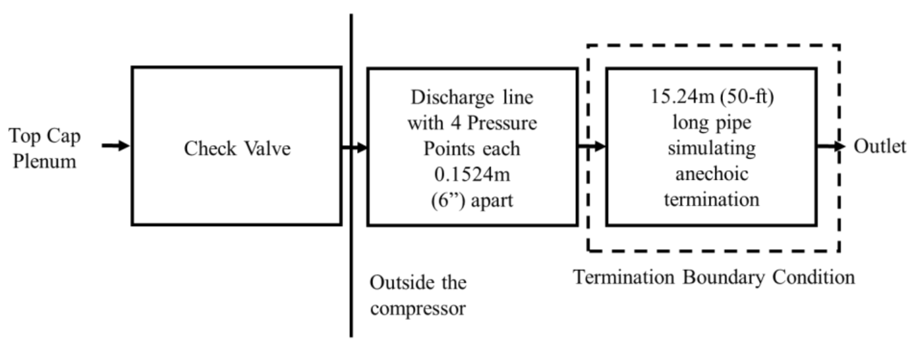

Figure 5: Flow Chart of Segment 3 in the 1-D Model

3. DISCHARGE PULSE MEASUREMENT SETUP

3.1 AHRI Standard 530-2011

According to the standard, a flexible refrigerant line is used for the discharge pulse measurement. The refrigerant line should be a minimum of 12.2m (40-ft) long without any change in inner diameter or sharp bends. At least 2 pressure transducers are required to be used in the discharge line, spaced ¼ wavelength apart on the fundamental pulsation frequency. Proper insulation must be added to the refrigerant line to avoid heat transfer to the test setup. Insulation can be achieved by burying it in sand or in another heat resistant material. If the test setup is packaged in a box, the refrigerant line can be coiled with a minimum radius of 30 tube diameters before it is routed to the load stand.

3.2 Test Setup

The measurement setup is shown in Figure 6. The compressor is connected to the discharge pulse box through a flexible line that is 0.9144m (3-ft) long. Poysat and Liegeois (2006) evaluated discharge pulse using a measurement setup with 5 pressure transducers. Similarly, in this paper, the discharge pulse is measured with 4 pressure transducers. Figure 6 shows the placement of the 4 sensors in the pulse box, each positioned 0.1524m (6 inches) apart. Beyond the transducers, the discharge line is connected to the test stand using a refrigerant line with a length of 15.24m (50-ft). The long hose enables the compressor outlet to simulate an anechoic termination. The pipe is coiled inside the box without any sharp bends or turns, so that standing waves are minimized near the pressure transducers. Standing waves are a result of constructive and destructive interference of the incident and reflected waves which can overestimate/ underestimate the discharge pulse.

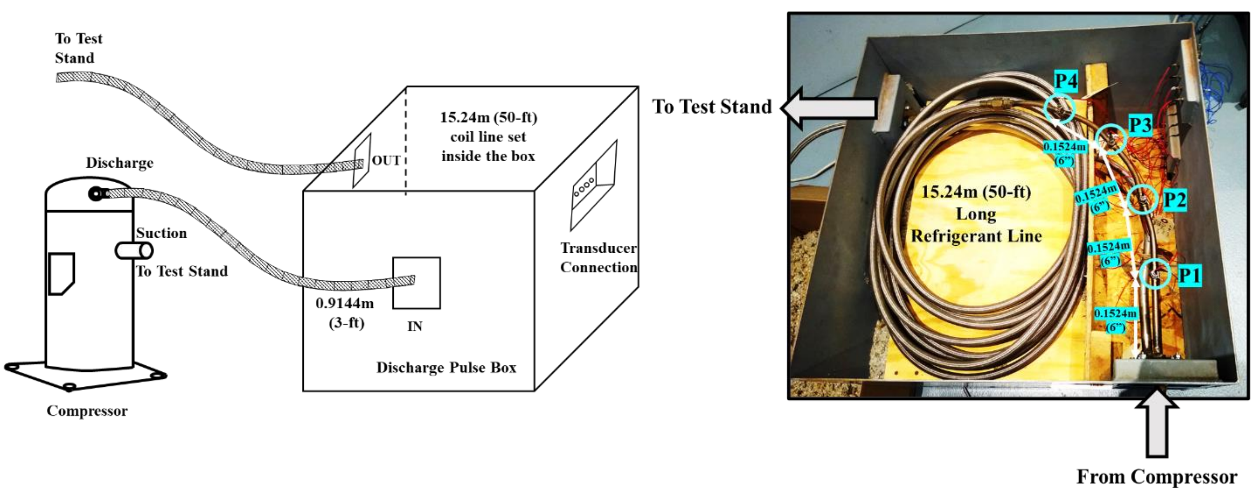

Figure 6: Discharge Pulse Box Setup

Figure 7 shows the pressure fluctuations recorded by all 4 transducers at 50°F /115°F /70°F. A time span of 0.0625 second which encompasses 3 full compression cycles is shown. Note that there is a slight phase shift between the pressure recordings due to the spatial positioning of the transducers. For example, P4 is placed the farthest from the compressor and measures the highest phase shift. According to AHRI 530-2011, the value of the highest single cycle peak-to-peak pressure recorded by each transducer is averaged and the pressure plot of the sensor that best fits the averaged value is considered. Then, the highest single cycle peak-to-peak pressure of the selected transducer is reported as the discharge pulse of the compressor. In Figure 7, P3 is closest to the average, so the discharge pulse of the compressor is recorded as 20KPa (2.9 psi) pk-pk based on sensor P3.

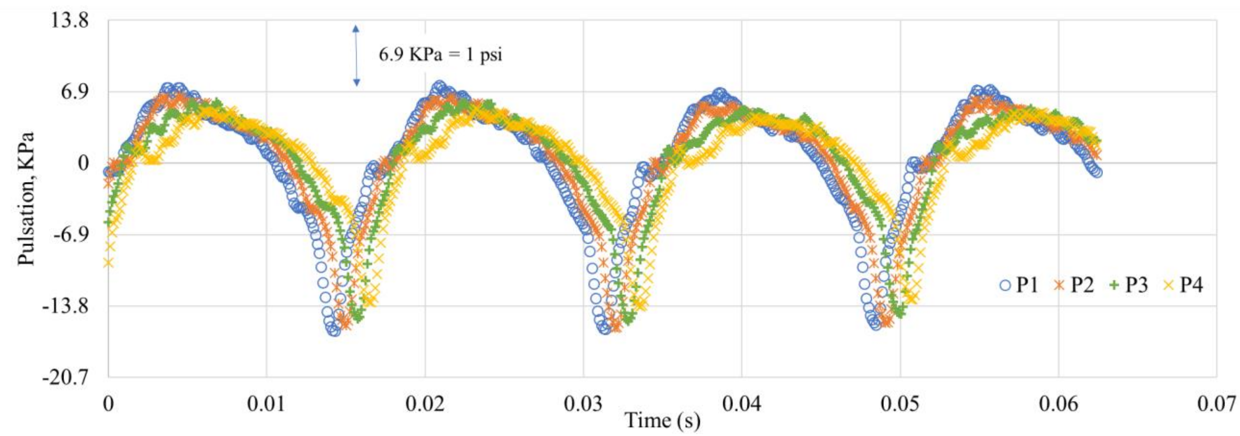

Figure 7: Discharge Pulse Recorded at 50°F /115°F /70°F

4. RESULTS AND VALIDATION

This section focuses on the validation between the 1-D model and the test results. As mentioned earlier, a 4-ton fixed speed residential scroll compressor is used for testing. Table 1 shows the operating conditions used for the validation study. The heat pump (HP) and air conditioning (AC) rating conditions are used as per AHRI 530-2011 referenced to AHRI 540-2020 and the measurement is performed based on the approach explained in the previous section. The table also contains the operating condition with the suction and discharge temperatures very close to the Seasonal Energy Efficiency Ratio (SEER2) B rating condition for single speed system operating in cooling mode, per AHRI Standard 210-240-2023. The suction/discharge temperatures tabulated at SEER2 B are close to the pressures recorded at the outdoor unit when the ambient temperature is 82°F. The SEER2 B rating is used by the OEMs to rate the energy efficiency of the unitary air-conditioning outdoor units. Finally, a high mass flow rate operating condition is also tabulated and is used to evaluate discharge pulse.

Table 1: Operating Conditions

Compression Cycle			Low Side		High Side - Subcritical	
Application	Rating Testing Point	Cycle Type	Suction Dew Point Temperature, °F	Suction Superheat, R °F	Discharge Dew Point Temperature, °F	Condenser Exit Sub Cooling, °F
AC and HP **AHRI 530-2011 Rating**	Heating	Subcritical	5	20	95	0
	Cooling	Subcritical	50	20	115	0
SEER2 B	Cooling		50	20	100	0
High Mass Flow Rate	Cooling		50	20	80	0

Figure 8 shows the discharge pulse of the scroll compressor estimated from the 1-D model and compared to AHRI 530-2011. The peak-to-peak amplitude of the pressure fluctuation is recorded and presented in the figure. The first impression from the figure is that the discharge pulse of the scroll compressor is influenced by the operating condition. The pulsation recorded in the cooling part of the operating map is lower compared to HP rated condition. This is because the HP rated condition falls in the under-compressed region, far from the design line, and the cooling operating conditions presented in the figure are relatively closer to the design line.

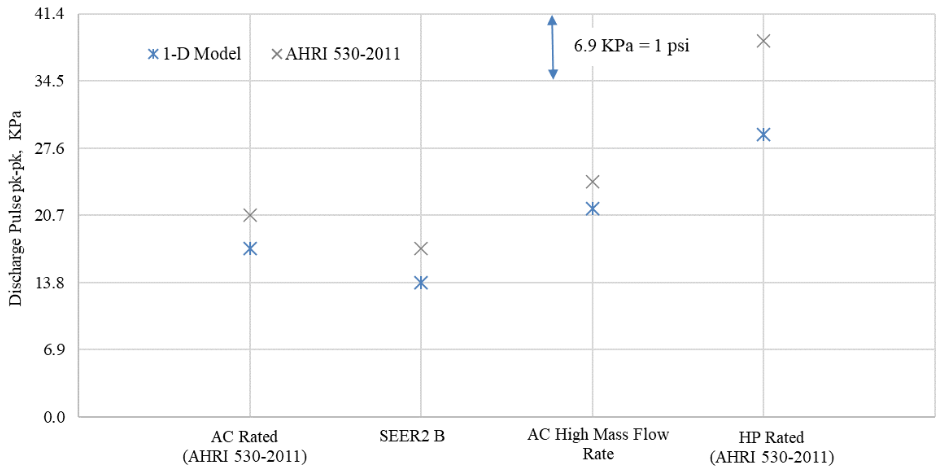

Figure 8: Discharge Pulse Peak-to-Peak at Various Operating Conditions

During the rated HP compression cycle, when the gas is approaching porting, the discharge pocket and port are open at the same time. At that instant, the discharge pocket pressure is lower than the top cap plenum pressure. This enables the high-pressure gas from the plenum to flow back into the discharge pocket, resulting in recompression of the gas. This mechanism is referred to as under-compression in Figure 2. As a result, a pressure pulse with a min-max differential can be created in the scroll discharge pocket as the port opens. Due to under compression, the min-max amplitude of the pressure pulse at the scroll pocket is higher at HP rated condition compared to the cooling conditions presented in Table 1. Further, the top cap plenum acts as a transfer path between the scroll discharge pocket and the compressor outlet to the load stand where the discharge pulse is measured. As a result, the scroll discharge pocket and the top cap plenum creates a transfer function, as per Equation 1, to determine the discharge pulse at the compressor outlet. Since the plenum volume remains constant, the pressures at the scroll discharge pocket at the onset of porting acts as the key parameter influencing the discharge pulse. Hence, under compression causes the compressor to have higher discharge pulse at HP rated condition compared to the cooling operating conditions presented in Table 1.

Figure 8 also illustrates the 1-D model slightly underestimating the measured discharge pulse within the acceptable range. The model assumes no tip or flank leakage, which could be the reason for predicting lower discharge pulse than the measurement. Additionally, the temperature of the gas entering the suction is modeled based on an engineering assumption. The assumption helps to predict the discharge pulse very accurately when the compressor is operated close to the design line. However, at the HP rated condition, the gap between the 1-D model and measurement is widened (> 6.9KPa or 1psi) since the compressor operates far from the design line.

Figure 9 represents the time domain comparison of the discharge pulse between the 1-D model and AHRI 530-2011. The time data spanning for a 1000° crank angle is displayed at various operating conditions. As the figure suggests, the mathematical model closely captures the shape of the discharge pulse trace, as well as the peak-to-peak amplitude recorded by the measurement.

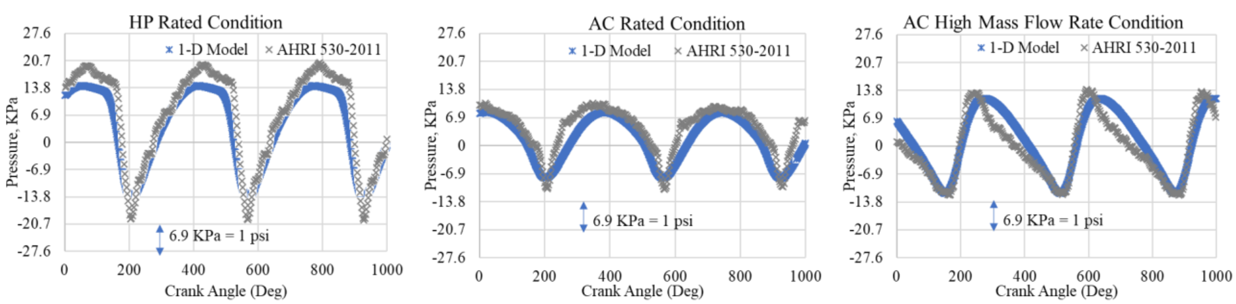

Figure 9: Discharge Pulse Comparison of the 1-D Model and AHRI 530-2011 in Time Domain

Figure 10 shows the Fast Fourier Transform (FFT) of the discharge pulse at the HP and AC rated condition. The time data from testing and simulation is converted to frequency data based on the AHRI 530-2011 guidelines. A flat top window is used for the FFT, so that the amplitude accuracy is maintained in the frequency domain. A narrowband pressure pulsation is presented up to 500 Hz. The energy peaks flatten beyond 300 Hz for the measurement and 200 Hz for the 1-D model. Hence, the FFT can be validated for the first 3 harmonics where most of the pulsation energy is concentrated. The peak amplitude of the harmonics in the narrowband frequency spectrum is used to estimate the discharge pulse of the compressor. It is evident that the 1-D model and the measurement have good agreement with the peaks up to 200 Hz, where the spectral energy concentration is at a maximum.

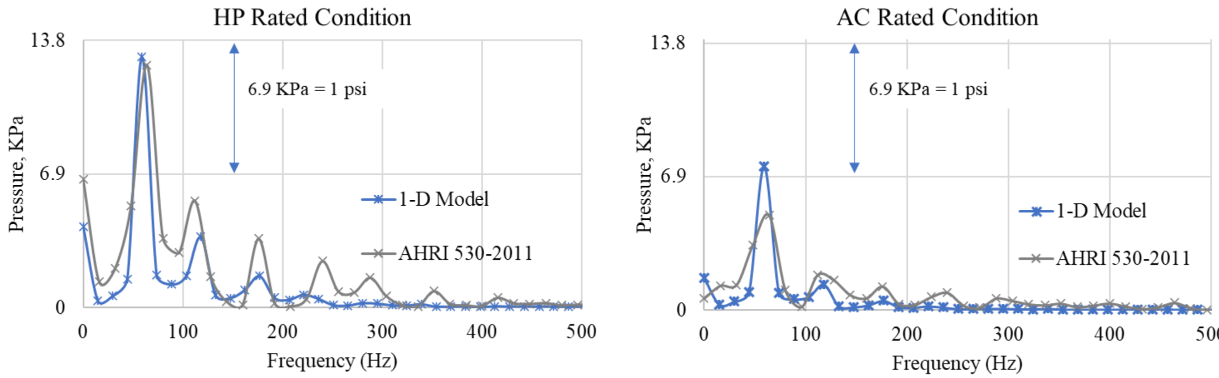

Figure 10: FFT of the Discharge Pulse at HP and AC Rated Condition

Figure 11 shows the effect of RPM on the discharge pulse at the SEER2 B rating condition. The fixed speed scroll compressor used for the validation study is setup with a special drive terminal, which is used to change the running speed. A slight decrease in discharge pulse is noted when the RPM is increased from 2400 to 5400. The maxima and minima of the discharge pulse trace gets flatter with the increase in RPM. However, this trend needs to be evaluated at other operating conditions to state, “as the RPM increases, discharge pulse decreases”. Nevertheless, the figure presents a good correlation between the measurement and the 1-D model at all the RPMs recorded.

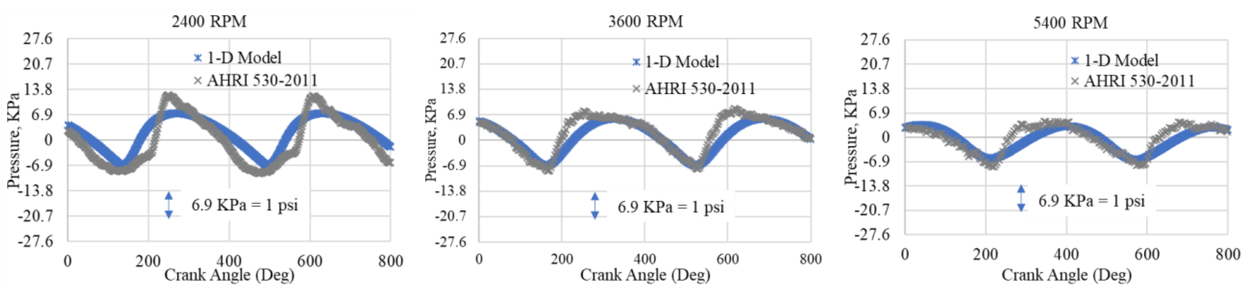

Figure 11: RPM Effect on the Discharge Pulse at SEER2 B

5. CONCLUSIONS

This paper highlights the prediction of discharge pulse through 1-D modelling. GT-SUITE is used to develop the model. The model consists of the scroll assembly of a fixed speed compressor along with the downstream components. The AHRI 530-2011 based measurement is used to validate the 1-D model. A discharge pulse box was built at Emerson to conduct the measurements. The discharge outlet inside the box is extended using a 15.24m (50-ft) long flexible line to simulate anechoic termination minimizing standing waves caused by reflection. In this study, the effects of operating conditions and RPM on discharge pulse are evaluated and validated. The validation is performed in both frequency and time domains. In the frequency domain, the peak amplitude between the model and test data is compared up to the third running harmonic. In the time domain, the peak-to-peak amplitude and the shape of the pulsation curve is compared for a span of 1000° of crank rotation. The maximum difference between the 1-D model and AHRI 530-2011-based test data is observed to be less than 6.9 KPa (1 psi) across the AC operating conditions compared. Further, this study can be expanded by performing a similar validation on a variable speed compressor. The assumptions used to develop the 1-D model can be improved to further minimize the gap between the numerical model and the measurement at HP operating conditions. The validated 1-D model can be emphasized as the primary tool to evaluate discharge pulse of the compressor thereby reducing the testing time and costs significantly.

NOMENCLATURE

AHRI	Air Conditioning, Heating, & Refrigeration Institute	
AC	Air Conditioning	
HP	Heat Pump	
Te	Evaporating Temperature	(˚F)
Tc	Condensing Temperature	(˚F)
Trg	Return Gas Temperature	(˚F)

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