Measurement of fluctuation at the discharge of positive displacement compressors

C S Rohleder* and E A Groll
Ray W. Herrick Laboratories, Purdue University
177 S Russel Street, West Lafayette, IN, 47907-2099, USA
crohlede@purdue.edu

Abstract. All positive displacement compressors (PDC’s) experience pressure fluctuation at the discharge ports of the compression mechanism, and these fluctuations can adversely affect compressor performance. These pressure fluctuations are caused by the nature of PDC’s; when a pocket of working fluid is exposed to system piping at differing pressure levels, a wave of pressure equalization flows through the line, potentially causing the compressor and piping to vibrate and generate noise (Soedel 2007). Compressor manufacturers currently use serial muffler designs, discharge plenums, and valves to minimize this fluctuation, but pressure fluctuation may persist at the outlet of serial muffling methods. A parallel muffling design has been developed for use in Roots blowers, having significantly reduced noise and pressure fluctuation seen in these machines (Huang et al. 2014). To identify the potential for use of this parallel muffling design in refrigeration systems and PDC’s, the magnitude of fluctuation at the discharge of a scroll compressor and a dual rotary compressor have been measured at various operating conditions. The compressors were left unmodified, with any current internal muffling methods in place. The measurement of these unmodified compressors determined the effectiveness of current muffling methods, with the intent to determine if using a parallel muffling technique is warranted. Pressure measurements were taken at 1000 Hz, and Fast Fourier Transform implemented to isolate pressure fluctuation based on frequency. Measurements lead to magnitudes of up to 16 kPa in pressure fluctuation with current muffling methods, indicating that further fluctuation reduction is possible.

1. Introduction
Constant and repetitive noise can be an annoyance at low levels but becomes more hazardous as the noise level increases (Schneider, Paoli, & Brun, 2005). As compressors are found in virtually every building, it is important that noise level from these machines remains minimal. One of the largest sources of compressor noise comes from the periodic gas discharge of the compressor, which causes pressure fluctuations in the discharge line (Sano & Mitsui, 1984). These pressure fluctuations are caused by the discrepancies between pressure ratio in the evaporator to the condenser, and the fixed volume compression process that is inherent in most positive displacement compressors (Soedel, 2007). Fixed volume ratio compressors have an ideal operating pressure ratio, which yields highest performance levels. When the compressor operates at conditions that are not at the ideal pressure ratio, the compressed gas pocket inside the compression chamber must equalize with the condensing pressure, through either a backflow into the compression chamber known as under-compression, or a rapid expansion through the condensing line known as over-compression. In either the over-compression or under-compression cases, pressure waves are possible. Nieter and Gagne (1992)
simulated the compression process for a scroll compressor and found that cases of under-compression caused more mass flow fluctuation (and therefore pressure fluctuation) than in over-compression cases, although both cases saw this fluctuation.

Huang modelled these pressure waves through flash tube theory (2012). Using this theory, Huang developed a parallel muffling technique to reduce the pressure fluctuation at the discharge of a Roots blower (Huang, Yonkers, & Hokey, 2014). The investigations discussed in this paper were performed to determine the viability of implementing this parallel muffling design in compressors more commonly used in air conditioning settings. Two compressors were tested, a 5-ton scroll compressor and a 4-ton dual rotary compressor. The compressors were not modified, so all manufacturer muffling techniques were left in place.

While others have seen that pressure fluctuation correlates with the pressure difference between the discharge of the compressor and the pressure of the condensing line (Wu, Xing, Peng, & Shu, 2004), the data recorded in this paper illustrates that after manufacturer attempts to minimize fluctuation with discharge valves, mufflers, and dead volumes, the pressure fluctuation remains fairly constant across pressure ratio, while correlating more heavily with evaporating temperature. While noise level and increased compressor performance is the motivation for pressure fluctuation reduction, sampling noise level was not a possibility in this assembly due to the location of the calorimeter test stand, leading to high levels of ambient noise that would interfere with any noise measurements around the compressor.

2. Experimental Setup
For ease in compressor testing, a calorimeter was utilized. A schematic of the calorimeter is shown in Figure 1. Testing in the calorimeter allowed for the control of the evaporating and condensing refrigerant temperatures and pressures, the inlet refrigerant temperature, and the surrounding air temperature. The compressor is placed in a sealed chamber, with heat exchangers and expansion valves outside of the chamber. The evaporating pressure was controlled via two pneumatic expansion valves, where a preset air pressure leads to a corresponding equal evaporation pressure. The condensing pressure was controlled via manipulating the temperature of recirculating cooling water with a metering valve. The metering valve adjusts the amount of cooling water that is recirculated, where new cooling water is introduced when lower temperatures are desired. The suction superheat was controlled through a secondary two-phase R134a tank with electric heating elements. The evaporator capacity was determined by the amount of electric heat input to the secondary tank. This capacity was adjusted by modulating the electric heat input to attain the superheat required.

![Figure 1: Schematic of the compressor calorimeter used for testing (Mösch, 2015).](image-url)
Instrumentation involved the use of pressure transducers, thermocouples, and a mass flow meter in the refrigerant line. Outside of the refrigerant line, power monitors for the compressor electric power consumption and the electric heating elements power consumption were utilized. The surrounding air temperature and velocity inside the compressor housing chamber was also measured.

To measure the pressure fluctuation at the discharge of the compressor, a dynamic pressure transducer was installed at the discharge port of the compressor. The probe was placed flush with the pipe wall, and 15 inches of straight pipe was installed after the probe insert as shown in Figure 2.

![Figure 2: Location of pressure transducer in both the scroll (left) and dual rotary (right) compressors.](image)

Placing the pressure transducer in this manner reduced the possibility of interference in pressure readings at high frequency. The absolute pressure was sampled at 1000 Hz, giving a resolution of 16 data points per discharge at 60 Hz (8 points at 120 Hz).

Two different styles of compressors were tested; a scroll compressor with a volumetric flow rate of $14.09 \, m^3/h$ using the working fluid R407C, and a dual rotary compressor with a volumetric flowrate of $8.8128 \, m^3/h$ using the working fluid R410A. Tests were performed at a range of pressure ratios, with varying evaporating and condensing temperatures. The suction superheat was kept at a constant 11.1 K.

3. Results and Discussion

The two compressors were tested at varying pressure ratios and temperature distributions. Figure 3 presents the operating conditions of the tests conducted with the scroll compressor.

Using the highest overall isentropic efficiency as seen in Figure 1 to identify the ideal pressure ratio for the compressor, the ideal pressure ratio was found to be close to 4. Pressure ratios below 4 were considered to cause over-compression. In this case, the discharge from the compressor was at a higher pressure than the pressure in the condensing line, causing potential fluctuation of a high-pressure gas through the condensing line. Pressure ratios above 4 were considered to cause under-compression. In this case, the discharge pressure from the compressor was at a lower pressure than in the condensing line, causing potential back flow into the compression chamber. Due to these two potential disturbances, pressure ratios above and below the ideal pressure ratio were targeted for possible increases in pressure fluctuation.
The dual rotary compressor was tested more extensively, but in similar operating range as the scroll compressor. The operating conditions of the tests conducted with the rotary compressor are shown in Figure 5. For the dual rotary compressor, it was determined that the ideal pressure ratio was close to 3.5.

As stated earlier, the absolute pressure in the discharge line of the compressor was sampled at 1000 Hz immediately after entering the condensing line. With a constant sampling rate at steady state conditions, Fast Fourier Transform of the data was performed to isolate the magnitude of pressure fluctuation by frequency.
3.1. Gas Fluctuation Results

Figure 7, Figure 8, and Figure 9 display the pressure fluctuation measurements in both the time domain and their respective frequency domains for the scroll compressor. From Figure 7, Figure 8, and Figure 9, it is possible to see that the pressure fluctuations are isolated to 60 Hz, which coincides with the discharge rate of the scroll compressor. Based on the trends seen in the frequency domain plots, it can be concluded that the pressure fluctuations are minimal at ideal operating conditions, slightly higher in over-compression conditions, and the largest in under-compression conditions.

The same measuring technique was used for the dual rotary compressor. Figure 10, Figure 11, and Figure 12 display the pressure fluctuation measurements in both the time domain and their respective frequency domains for the dual rotary compressor. From Figure 10, Figure 11, and Figure 12 it is possible to see that the pressure fluctuations are isolated to 120 Hz, which coincides with the discharge rate of the dual rotary compressor. Although the motor rotates at 60 Hz, there are two rolling pistons with separate discharge chambers in this rotary compressor that rotate 180° out of phase of one another, causing the discharge rate seen in the condensing line to be 120 Hz.
Figure 7: Time (left) and frequency (right) domain of the scroll compressor discharge at an evaporating temperature of -13.3°C and condensing temperature of 54.7°C (pressure ratio of 7.7). This is a condition of under compression.

Figure 8: Time (left) and frequency (right) domain of the scroll compressor discharge at an evaporating temperature of -8.1°C and condensing temperature of 36.1°C (pressure ratio of 4.1). This condition operates near the ideal pressure ratio of the compressor.

Figure 9: Time (left) and frequency (right) domain of the scroll compressor discharge at an evaporating temperature of 4.3°C and condensing temperature of 37.9°C (pressure ratio of 2.7). This is a condition of over compression.
Figure 10: Time (left) and frequency (right) domain of the dual rotary compressor discharge at an evaporating temperature of -21.8 °C and condensing temperature of 53.6 °C (pressure ratio of 8.9). This is a condition of under compression.

Figure 11: Time (left) and frequency (right) domain of the dual rotary compressor discharge at an evaporating temperature of -1.1 °C and condensing temperature of 44.8 °C (pressure ratio of 3.5). This condition operates near the ideal pressure ratio of the compressor.

Figure 12: Time (left) and frequency (right) domain of the dual rotary compressor discharge at an evaporating temperature of -1.2 °C and condensing temperature of 28.4 °C (pressure ratio of 2.6). This is a condition of over compression.

While it was initially assumed that the pressure fluctuations will increase as the operating conditions move away from the ideal pressure ratio, the data obtained for the dual rotary compressor indicates that this is not the case. Figure 13 presents the pressure fluctuation at relative discharge frequencies with respect to the pressure ratio for the dual rotary compressor. It can be seen that pressure ratio has
minimal relation to the magnitude of pressure fluctuation. The evaporating temperature has a positive correlation with the magnitude of pressure fluctuation however. In addition, As seen in Figure 14, the magnitude of pressure fluctuation increases with evaporating temperature.

![Figure 13: Pressure fluctuation at various operating conditions compared across pressure ratio.]

![Figure 14: Pressure fluctuation at various operating conditions compared across evaporating temperature.]

3.2. Explanation of Minimal Fluctuations
As shown in Figure 13, the pressure fluctuations for the dual rotary compressor do not correlate clearly with pressure ratio, which disagrees with previous experiments and analytical models. The cause of this discrepancy is most likely due to the location of the pressure transducer, as it is set in the condensing line after the compressor manufacturer gas fluctuation reducing methods are implemented. In most commercially available compressors, the manufacturers have implemented serial muffling techniques in attempt to reduce pressure fluctuation. In the case of the scroll compressor, a discharge plenum is used, as well as a discharge valve and a check valve. The two valves are intended to limit the direction of refrigerant flow during start-up and shut-down, but do have a dampening effect on the
pressure fluctuation during regular operation. The discharge plenum is directly intended to reduce pressure fluctuation by providing a large dead volume to absorb the pressure difference between the condensing line and the incoming compressed pocket of refrigerant. This dead space has a proportionally larger volume than the displacement of the compression chamber, thus any pressure change in the dead space due to the pressure difference between the dead space and the compression chamber is minimal.

The dual rotary compressor also employs discharge valves and a discharge plenum, although the discharge plenum in this case is the shell of the compressor, leading to a proportionally larger dead volume than in the scroll compressor.

4. Conclusions and Future Work

The gas fluctuations of two positive displacement compressors, a 5-ton scroll compressor and a 4-ton dual rotary compressor, were tested using a conventional compressor calorimeter. The compressors were not modified, so all manufacturer muffling techniques were left in place.

Based on the test conducted, it was observed that the current fluctuation methods are able to limit pressure fluctuation to 16 kPa (±8 kPa) using serial muffling methods. After these muffling methods, pressure fluctuations correlate with the evaporating temperature.

To further evaluate the potential for parallel muffling methods, the pressure fluctuation must be measured immediately at the discharge of the compression chamber, before manufacturer muffling methods are used.

Pending the results of modelling and tests with an internally placed pressure transducer, a parallel muffling design will be implemented to mitigate pressure fluctuation in the discharge line. Comparisons on compressor performance as well as pressure fluctuation between the parallel muffling technique and the manufacturer muffling methods will be made.

References

[1] Huang, P. X. (2012). Gas Pulsations: A Shock Tube Mechanism. International Compressor Engineering Conference (pp. 1304-1-10). West Lafayette: Purdue University.
[2] Huang, P. X., Yonkers, S., & Hokey, D. (2014). Gas Pulsation Control Using a Shunt Pulsation Trap. International Compressor Engineering Conference (pp. 1136-1-10). West Lafayette: Purdue University.
[3] Mösch, T. (2015, December 21). Performance tests of a scroll compressor with vapor injection using a compressor calorimeter. Rostock, Mecklenburg-Vorpommern, Germany.
[4] Nieter, J. J., & Gagne, D. P. (1992). Analytical Modeling of Discharge Flow Dynamics in Scroll Compressors. International Compressor Engineering Conference (pp. 85-96). West Lafayette: Purdue University.
[5] Sano, K., & Mitsui, K. (1984). Analysis of Hermetic Rolling Piston Type Compressor Noise, and Countermeasures. International Compressor Engineering Conference (pp. 242-250). West Lafayette: Purdue University.
[6] Schneider, E., Paoli, P., & Brun, E. (2005). Noise in Figures. European Agency for Safety and Health at Work.
[7] Soedel, W. (2007). Sound and Vibrations of Positive Displacement Compressors. Boca Raton: Taylor & Francis Group, LLC.
[8] Wu, H., Xing, Z., Peng, X., & Shu, P. (2004). Simulation of Discharge Pressure Pulsation Within Twin Screw Compressors. Part A: J, Power and Energy (pp. 257-264). Institution of Mechanical Engineers.