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Experimental Testing and Computer Simulation of a Reciprocating Refrigeration Compressor

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ABSTRACT

This paper describes experimental testing of a two-cylinder open-drive reciprocating refrigeration compressor with spring-loaded ring-type valves, and a comparison of experimental results to predicted performance using a computer simulation. The cylinder pressure, and motions of the suction and discharge valves will be reported. The refrigerant used was R-12, although the results could be indirectly applied to other refrigerants, in particular for validation of computer models.

Experimental measurements were made simultaneously in real time, at each degree of crankshaft rotation. Measurements were taken at steady state condition. Saturated suction temperatures ranged from 1 °F to 34 °F (-17 to 1 °C), while the saturated discharge temperature was held roughly constant at 91 °F (33 °C). The mass flow rate of refrigerant ranged from 245 to 521 lb/hr (111 to 236 kg/hr). The experimental and predicted cylinder pressures agreed quite well, although there were differences between the two during the suction and discharge processes, which can be attributed to effects which were not accounted for in the computer model. Static behavior of the valves, such as when the valve opens, was predicted quite well. However, prediction of the dynamic motion of the valves by the computer model was not entirely accurate – indicating that refinements are needed in the modeling of the valve dynamics.

INTRODUCTION

The main objectives of the experimental testing were to make real-time, simultaneous measurements of the suction and discharge valve motion – and also cylinder, suction plenum, and discharge plenum pressures.

The compressor that was tested had a nominal capacity of 5.2 tons, although one of the pistons was removed (to support future grid-based modeling). The valves in this compressor are kept closed by springs, and open when the pressure force across the valve is greater than the spring force holding the valve on its seat. The refrigerant used was R-12, although the results could be indirectly applied to other refrigerants, in particular for validation of computer models. A test stand was built to provide the compressor with controllable suction and discharge conditions. The type of test stand used is called a desuperheating test stand and is of a type chosen by ASHRAE as standard equipment for rating positive displacement refrigeration compressors [1] (it is called gaseous refrigerant flow meter by ASHRAE). The refrigerant flow rate through the compressor was measured using an orifice plate (with a diameter ratio, B, of 0.298), and was checked by a calorimetric analysis on the condenser and mixing chamber of the test stand. It should be noted that since the test stand had an oil separator, and many of the articles in the literature state that oil circulation in a system with an oil separator is less than 1%, refrigerant properties were calculated using equations for pure R-12. The oil level seen in the sight glass on the compressor did not noticeably drop during the experimentation. The test stand used a low-speed data acquisition system for taking measurements from pressure transducers, type T thermocouples, and a condenser water flowmeter. These data were used to calculate refrigerant state points, and refrigerant flow rates – both from the orifice plate and from a calorimetric method. A PC-based high-speed data acquisition system was used to measure the motions of the ring valves and the three pressures in the compressor: suction plenum, discharge plenum, and cylinder.
Published accounts on experimental measurements of valve motion in compressors are common, but the vast majority are concerned with reed, or flapper, valves. Computer modeling is a common method used to predict the dynamic behavior of ring valves, because of the sometimes prohibitive expense in time or money of experimental work and because of the physical difficulty of getting instrumentation installed in the compressor [2]. A computer simulation for predicting valve motions and cylinder pressure in a reciprocating compressor was developed at the University of Minnesota [3]. The model predictions will be compared to the experimental results. It is hoped the data obtained will be used by computer modelers as a way to check the validity of compressor models.

APPARATUS AND EXPERIMENTAL PROCEDURES

• Compressor

Apparatus

One of the main reasons for choosing the type of compressor used in this research was that it lent itself easily to numerical modeling (axisymmetric inlet and outlet to the cylinder) and experimental testing (ring valves which are relatively easy to instrument). The compressor - Carrier model 5F20 - is a two-cylinder reciprocating, open-drive type, with a nominal capacity of 5.2 tons (18 kW) per ARI standard 514-60. The compressor normally comes with a capacity control feature but it was not used in this testing. For this research, one of the pistons was removed from the compressor, although the connecting rod bearing was left on the crankshaft to minimize any unbalancing effect. The cylinder bore is 2.50 inches (6.35 cm) and the piston stroke is 2.00 inches (5.08 cm). The compressor was driven by a 7.5 Hp (5.6 kW) 3-phase electric motor (208 V), with a nominal speed of 1750 rpm. A modified cylinder head was manufactured so there would be enough room for the various transducers and their wiring and to allow for electrical feed-throughs.

For high-speed measurements of suction and discharge valve motion and compressor cylinder, discharge plenum, and suction plenum (i.e. crankcase) pressures, the compressor was fitted with 2 eddy-current displacement transducers, and 3 diaphragm-type pressure transducers. A digital magnetic transducer (also referred to as a di-mag transducer) was used to provide a reference signal indicating the piston position.

The displacement transducers had to be installed in counterbored holes approximately 0.5 inches (1 cm) in diameter and 0.3 inches (0.8 cm) deep since any metal surrounding the transducers affects their output. Calibration curves were supplied, but the installation was non-standard enough to warrant calibrating them after they were installed in their respective compressor sub-assemblies. The effect of temperature on the output was checked by reading the signal when the compressor was at room temperature and not running (both valves would be closed), then running the compressor for 1 or 2 hours, turning the compressor off and reading the outputs again. It was found that - for both transducers - the output changed less than 0.2 volts, which corresponds to approximately 0.001 inch (0.025 mm) of valve travel: the nominal sensitivity of the transducers is 0.2 V per thousandth of an inch. The cylinder pressure transducer was installed in the center of a modified bolt which was one of three bolts holding the discharge valve stop in place. The discharge-plenum pressure transducer was installed in the custom cylinder head, and the suction-plenum pressure transducer was installed in a hole tapped axially through the compressor's oil fill plug. The di-mag transducer was used to obtain timing information for the other measurements. A notch was machined in the compressor-motor coupling, and the di-mag transducer was positioned to detect the passing of the notch as the coupling rotated. When the leading edge of the notch was directly below the sensing tip of the transducer, the compressor piston was 95.2 degrees (± 0.6 degrees) before top-dead-center. Therefore, the output of the transducer could be used to reference the high-speed measurements to top-dead-center.

Operation

When acquiring high-speed data, the compressor was allowed to operate about 1 hour after startup before the first set of data was taken, but it was only necessary to wait 15-30 minutes after changing to a new set of conditions to take the next set of data.

For measuring signals from the instrumented compressor, a high-speed, 4-channel data acquisition card installed directly into an IBM model AT computer was used. This card can gather data at a rate of 45 kHz per channel if 4 channels are monitored, using Direct Memory Access. The range of the card is ± 10 Volts, and the resolution is 12 bits - which translates into a voltage resolution of 4.88 mV. Data were taken at approximately every degree of crankshaft rotation. More precisely, the measurements were spaced (1/360)T seconds apart, where T is the period of the compressor rotation. The compressor rotational speed was measured by timing the signals from the di-mag transducer. The uncertainty in the angular position of the crankshaft for the high-speed
measurements was calculated as 1.7 degrees. The data were checked after each run to make sure the acquisition process was started when both of the valves were closed. If this wasn't the case, the test was run again. This was necessary because the valve motions, especially the suction valve, were not identical from revolution to revolution as will be seen later.

**RESULTS**

High-speed measurements of the valve motions and pressures were taken at 4 different compressor inlet pressures. The discharge pressure was maintained at a nearly constant level - corresponding to a saturated discharge temperature (SDT) of approximately 90 °F (32 °C). The saturated suction temperature (SST) corresponding to the 4 different suction pressures were 1.5, 10.8, 21.7, and 33.3 °F (-16.9, -11.8, -5.7, 0.7 °C). Results from the lowest SST will be reported here. For a given discharge pressure, the upper limit on SST was constrained by a needle valve in the test stand through which most of the refrigerant flows, while the lower limit is constrained by the low pressure drop across the orifice flowmeter. The refrigerant flowrate for the 4 tests ranged from 245 lb/hr (111 kg/hr) at the lowest SST to 521 lb/hr (236 kg/hr) at the highest SST.

The calorimetric method used for calculating the refrigerant flowrate differed from the flowrate measured using the orifice flowmeter by 1.5% at the lowest SST, and by 6% at the highest flowrate. This difference is within predicted uncertainties, with the exception of the test done at the highest flowrate (i.e. the highest SST).

**High-Speed Measurements : Compressor**

Cylinder pressures

The cylinder-pressure time history is shown in figure 1. Generally, the cylinder pressure was measured for 4 consecutive cycles of the compressor for any of the given conditions. The repeatability of the cylinder pressure from cycle to cycle was quite good - the maximum deviation of any of the measured cylinder pressures from the average (at any crankshaft angle) was ± 2 psi.

Valve motions - general

The suction and discharge valve displacements are shown in figure 2. The calculated maximum valve lifts for the suction and discharge valves are 0.050" and 0.070" (1.27 and 1.78 mm), respectively. Valve lift is defined as the distance the valve is displaced from its seat. The maximum valve lifts were determined by measurements of the compressor geometry. The uncertainty in the position of the valves was calculated as 0.003 inches (0.076 mm).

Suction valve motion

Referring to figure 2, the dynamic motion of the suction valve could be considered to have 2 degrees of freedom: a translation in the 'axial' direction, and a rotation about the radial direction. Although there are no data to test this hypothesis, it was considered to be the most likely explanation. A second transducer placed 180° opposite the first would prove or disprove this hypothesis. The cause of this motion might be non-uniform pressure forces on the crankcase side of the suction valve, or slight variations in the suction spring constants.

As can be seen in figure 2, variation in the closing angle of the suction valve was observed. In some of the runs, the valve actually closes after the piston has reached bottom-dead-center. This indicates that some outflow of refrigerant from the cylinder to the suction plenum may be occurring. Outflow of the suction gas back into the suction plenum is undesirable, since it results in a loss of pumping capacity which in turn reduces overall system efficiency. The valve openings, on the other hand, are very well behaved. For the two measurements shown in figure 2, the valves open at nearly identical crank angles. This was typical of the other runs also (note the distinction between 'cycle-to-cycle' valve openings at a given suction pressure, and valve openings at different suction pressures: the crank angle at which the suction valve opens is a function of suction pressure, whereas the crank angle at which the valve opens at a given suction pressure is constant from cycle to cycle).

Discharge valve motion

The motion of the discharge valve is qualitatively different than that of the suction valve. The discharge valve is open for a much shorter time, because the gas exhausted from the cylinder is much denser than when it was drawn into the cylinder. The oscillations seen in the suction valve motion are absent in the discharge valve motion - the motion is divided into an opening and closing
phase, both distinct from each other. In fact, the motion may be generally described as overdamped.

The discharge valve's motion is very repeatable from cycle to cycle at any given condition, as can be seen by comparing the two measurements of the same run. There is a 'dip' in the valve lift curves during the closing phase, which is intimately linked to the pressure difference across the valve and the instantaneous flow through the valve port. The one characteristic that sometimes varies from cycle to cycle is the maximum lift of the discharge valve, which in one case showed a 0.005 inch deviation. Like the suction valve, the cycle-to-cycle opening of the discharge valve occurs at the same crank angle. Regardless of suction pressure, the valve consistently closes about 3.5 degrees before top-dead-center.

Comparison Of Experimental Results To A Computer Simulation

The computer simulation used to predict the cylinder pressure and valve motions was developed previously by Liang and Kuehn [3]. It uses a lumped-parameter method to predict the cylinder pressure and valve motions for a positive displacement compressor with spring-loaded valves. Once the known geometric properties of the compressor and refrigerant states were input into the simulation, there were 4 unknown input parameters - of which clearance volume and valve damping coefficients were included. From parametric studies, it was found that the clearance volume affected the cylinder pressure only, and the suction valve damping coefficient affected the suction valve motion only (the discharge valve damping coefficient had an insignificant effect on the discharge valve motion).

Experimental vs. predicted cylinder pressures

The appropriate clearance volume was found by comparing the simulation results using various values for clearance volume with actual test data. Figure 1 shows the comparison of actual test data to the computer simulation. A clearance volume of 5% was used. Engineering data supplied for this compressor indicated that the clearance volume can vary from 3.3% to 5.2%. The increase in clearance volume caused by the hole around the suction displacement transducer was approximately 0.6%.

It can be seen from the figure that the simulation models the pressure inside the cylinder quite well. The simulation predicts the cylinder pressure during the suction process to be approximately 5-10 psia higher than what was found from experiments. This can be explained by the model not accounting for some pressure drops found in the real compressor. The same behavior is seen in the cylinder pressure during the discharge process; and the same argument can be used to explain that behavior.

Experimental vs. predicted suction valve motions

Figure 3 contains simulated valve motions for 3 different suction valve damping coefficients as well as the experimental valve motions. The 3 different predictions shown demonstrate the qualitative effect of the damping coefficient on the valve's dynamic motion. It is observed that the simulated valve motions show the valve opening at the same angle and rate, the oscillatory motion beginning at the same angle, and the valve closing at the same angle. Note that the simulation indicates the valve will close significantly after the piston reaches bottom-dead-center which seems slightly counter-intuitive. The opening time and the opening rate of the valve are modeled quite well. However, the dynamic motion of the valve as measured by the author does not resemble what is predicted by the simulation. This indicates that refinements of the model used in the simulation are needed.

Experimental vs. predicted discharge valve motions

As was the case for the suction valve, the discharge valve motion had no noticeable effect on the cylinder pressure. The results presented in figure 3 show simulated valve motions for 3 different discharge valve damping coefficients as well as the experimental valve motion. Again, when considering the simulation results, the valve opens at same time and rate regardless of the damping coefficient. Unlike the suction valve, though, the damping coefficient had no effect on the dynamic motion of the discharge valve. Some speculation may be attempted as to why the simulation predicts the valve stays open longer than the experimental results indicate. The simulation uses the experimental values of the mass flowrate and plenum pressure as constraints - therefore, the valve has to be open long enough to allow for the prescribed mass to flow through it with a pressure drop across the valve determined by the geometry. This may explain why the area under the valve motion curves is larger for the simulation than for the experimental results. Recall that the simulation under-predicted the cylinder pressure; therefore the driving force for flow is less in the simulation - and the density of the refrigerant is less - and so the valve must be open for a longer period of time to allow the correct mass of refrigerant to flow through it. The valve is open when the piston reaches top-dead-center, which is not what one would expect or what is indicated from the experimental results. When comparing the experimental valve motion to the predicted motion, the opening time of the valve is not predicted as well as for the suction valve - the difference between experiment and
simulation is approximately 10 degrees. The opening rate of the valve is predicted rather well. The prediction of the overall shape of the valve motion curve can be thought to be better for the discharge valve than for the suction valve.

CONCLUSIONS

Overall, the test stand operated quite well. The orifice plate gave accurate mass flowrate measurements, which were more constant than the mass flowrates calculated from the energy balance.

Considering the high-speed measurements taken from the compressor, the cylinder pressure measurement was repeatable from cycle to cycle at all of the suction and discharge conditions. Valve motion data were very reliable, given the excellent correlation of valve openings from cycle to cycle. The suction valve motion was not repeatable from cycle to cycle, due to the valve moving in a non-planar fashion. The discharge valve motion was very repeatable from cycle to cycle.

The predicted cylinder pressures compared very well with the experimental results. Differences in cylinder pressures between experiment and simulation during the suction and discharge processes can be attributed to unmodeled pressure drops across the valves and/or valve ports. Qualitatively correcting for the unmodeled pressure drops reduces the error between experiment and simulation. Prediction of the dynamic motion of the valves by the computer model is not entirely accurate. Some 'static' behavior is predicted well – such as when the valves open, and the rate at which they open – but the valve dynamics require more refined modeling.

REFERENCES


Figure 1 - Experimental and simulated cylinder pressures. The experimental results are averaged from 4 consecutive cycles of the compressor.
Figure 2 - Experimental valve motions - suction and discharge valves. The measurements shown are for 2 consecutive cycles of the compressor.

Figure 3 - Comparison of experimental and simulated valve lifts. The experimental results, denoted with symbols, are from figure 2 (only one of the runs is shown). The simulated results are shown as solid lines, the difference between them being the damping coefficient (DC) used.