Some Effects of Injection Oil on a Screw Compressor for Application in Refrigerant R-134a Air-Conditioning Systems

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Zusammenfassung

Ergebnisse der experimentellen Untersuchungen von Öleinspritzung mit einem Schraubenkompressor, der für Gebrauch in den R-134a Klimaanlagen bestimmt ist, werden überprüft. Einfluss der Ölströmungsgeschwindigkeit, Öltemperatur, die Stelle der Einspritzöffnungen und der Gebrauch von zwei Art Düsen wird umfaßt. Experimentelle Untersuchungen der Eigenschaften des Öls, das Kältemittel während die Verdichtung auflöst und freigibt, werden auch besprochen. Ausführliche Messungen des Verdichtungsprozesses und einer thermodynamischen Simulation werden verwendet, um das Verhältnis zwischen die Einzelheiten der Öleinspritzung und der resultierenden Kompressorleistung zu erklären.

Abstract

Results of experimental studies of oil injection with a screw compressor designed for use in R-134a air-conditioning systems are reviewed. Effects of oil flow rate, oil temperature, the location of oil injection ports and the use of special nozzles are covered. Experimental studies of the properties of the oil, which will dissolve and release refrigerant vapor during the compression cycle, will also be discussed. Detailed measurements of the compression process and thermodynamic simulations are used to explain the relationship between the details of the oil injection and the resulting compressor performance.

1 Introduction

The work reported here is part of the development of high performance R-134a screw compressors for use in air-conditioning systems. Specifically, the studies carried out are intended to find practical methods of using oil injection to achieve the best performance in current production compressors. In addition, tests and analyses were carried out to identify the underlying reasons why the different oil injection methods affected the performance of the compressor and to use this knowledge to improve existing thermodynamic design models.

A literature survey shows that studies of oil injection effects have been carried out in the Fachgebiet Fluidenergiemaschinen (FEM) at the University of Dortmund, references /1/ and /2/. Work is also reported at this conference and in FEM Research Reports publications, references /3/, /4/ and /5/. Plans and some of the test results for this project were reviewed with FEM as part of a cooperative effort for this study of the R-134a screw compressor.

There were three major experimental elements to the investigation. A series of performance tests were carried out during which selected oil injection parameters were varied and overall compressor performance measured. At the same time, high-speed pressure transducers were installed in order to acquire indicator diagrams. Finally, a special test device was built to allow us to observe the oil flow characteristics of the plain bore and spray type nozzles used in the compressor tests. In addition to the flow visualization, this test rig allowed us to measure the amount of refrigerant vapor escaping from the oil as a result of the pressure drop at the point of injection.

Section 2 describes the overall compressor performance tests. Here, the various test configurations are defined and the effects of injection parameters on overall compressor performance are presented. Results from the indicator diagram measurements are reviewed in Section 3. Injection visualization and refrigerant release from the injected oil are discussed in Section 4. Comparisons of the test results with calculations using a thermodynamic simulation of the compressor are given in Section 5 and a summary and conclusions are offered in Section 6.

2 Injection Effects on Compressor Performance

A transparent housing view of the compressor is shown in Figure 1. Oil enters the compressor at the filter and is then sent to lubricate the bearings and to the injection into the compression chamber in the rotors for cooling and sealing. In this study, only variations in the rotor injection oil are considered. In the normal production compressor, oil enters into the rotor compression chamber through a simple bored hole near the cusp -- the intersection of the male and female rotor bores. For this investigation, two port locations are tested. The port location is defined by the volume ratio of the compression space at the point at which the space is first in communication with the oil injection port. The port locations are designated as location A, nearest the inlet end of the rotors, and location B, set nearer the discharge end. The ports are at volume ratios of 1.2 and 1.5, respectively and are located 38 mm apart -- about 16% of the overall rotor length. Port location B is as far into the compression as possible. Oil supplied from a discharge side oil separator and sump flows due to the pressure difference between the sump and the point of injection. If the injection point is moved too near the discharge port, pressure at the port can be higher than the pressure in the sump and no oil can flow to the rotors.

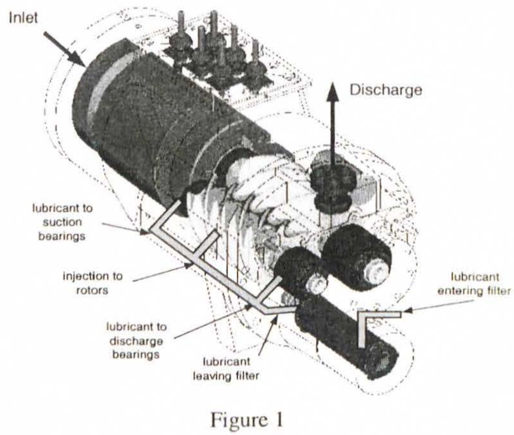
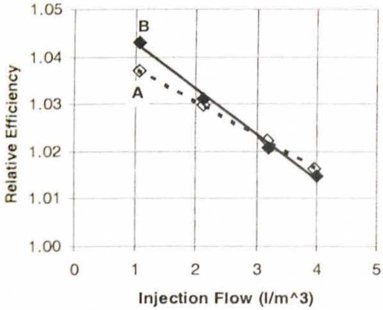


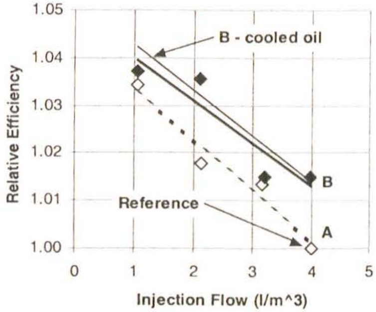
Figure 1
Screw Compressor with Oil Injection

Tests were run with varying amounts of oil injected into either port A or port B. In addition, tests were run with both cooled and uncooled oil. Results are shown in Figure 2.

In all cases, the compressor was operating at a pressure ratio of 4.6 with a male rotor rotational speed of 3550 min^{-1}. Efficiency shown in the figure is referred to the efficiency of the run with highest flow rate of uncooled oil through port A. Oil flow is shown as the injection oil flow in liters/min relative to the compressor inlet refrigerant volume flow rate in m^3/min. For the cooled oil runs, oil temperature is maintained at 57°C.



2a -- Cooled Oil



2b -- Uncooled Oil

Figure 2

Effect of Oil Flow Rate, Port Location and Oil Cooling

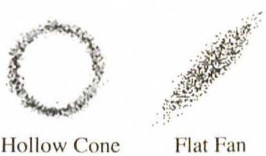
In the figures, data with the open symbols and the dotted line (a linear fit to the data) are for port location A; solid symbols and heavy, solid lines are for port location B. The linear fit of the data for cooled oil and port location B from Figure 2a is shown also in Figure 2b as the thin, solid line.

Examination of the data in these figures reveals the following:

1. Lower oil flow rate improves performance in all cases.
2. There is not much difference in the rate of change of performance with changing oil flow rate, but the effect of flow is greatest for the case of uncooled oil through port A.
3. Use of port location B is generally better than port location A except at the higher flow rates of cooled oil.
4. Cooled oil provides higher performance than uncooled oil, especially at high flow rates with port location A.

Combinations of injection location, flow rate and temperature can create a wide range of performance results. In these tests, the best performance (cooled oil, location B, low flow) resulted in an increase of 4.3% in efficiency compared to the baseline (uncooled oil, location A, high flow rate).

Some of the data from these tests is discussed in more detail in conjunction with the analysis of the indicator diagram results presented in Section 3.



Hollow Cone Flat Fan

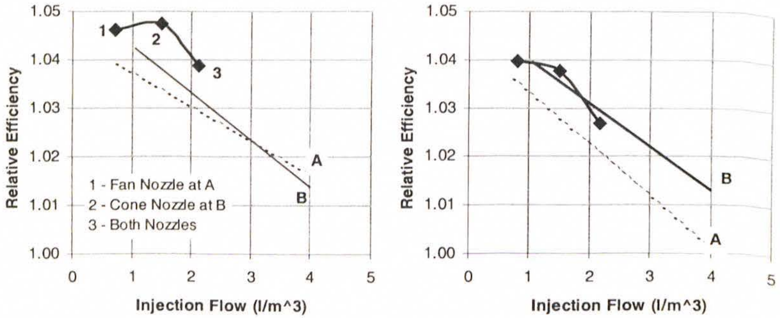
Figure 3

Nozzle Spray Patterns

In addition to the plain bore injection ports, two nozzle types were studied. Tests were run with a hollow cone type nozzle in position B and a fan nozzle in position A. Spray patterns for the two nozzle types are illustrated in Figure 3.

Data was taken for uncooled oil with each nozzle running separately and with injection through both nozzles at the same time. In addition to the pattern of the spray flow, both nozzles are intended to provide atomization of the oil, increasing the number of droplets in the flow. The goal is to increase the total surface area of oil droplets in the flow and thereby raise the effectiveness of the oil-to-refrigerant heat transfer process. The flat fan nozzle used has a maximum flow rating of about 5 l/min and the hollow cone nozzle is rated at about 10 l/min. An external flow control valve that was used to vary the flow rate for the tests with the plain bores was removed when testing the nozzles to eliminate the pressure loss and allow the maximum possible flow through the nozzles.

Results are shown in Figure 4. Data with the solid symbols and heavy solid line is for the tests with the nozzles. The lowest flow point is from use of the fan nozzle in location A. The hollow cone nozzle in location B provides the data at the intermediate flow rate and running both nozzles at the same time gives the third point at the highest flow.



4a -- Cooled Oil

4b -- Uncooled Oil

Figure 4

Effect of Fan and Hollow Cone Nozzles on Compressor Efficiency

The light weight, dashed line in each figure shows the performance of the plain bore configuration in location A (from Figures 2a and 2b); the light weight, solid line is for the plain bore at location B.

From the data in Figure 4, it appears that use of the nozzles has some benefit. The fan nozzle in position A provides higher performance than the plain bore at the same flow for both cooled and uncooled oil. The same is true for the hollow cone nozzle at position B. Within the uncertainty of the data, the fan and cone nozzles provide about the same benefit relative to their baselines with the plain bores - the fan nozzle data compared to the plain bore in location A and the cone nozzle compared to the plain bore in location B. On average, the use of nozzles improves efficiency relative to use of the plain bores by 0.5%.

Some insights into why the performance variations occur can be found in results of the indicator diagram measurements and comparisons with results computed using the thermodynamic model. Results of these analyses are discussed in Sections 3 and 5.

3 Indicator Diagram Analysis

The effects of the various oil injection parameters on overall performance can be seen in the differences in the pressure-time traces recorded for all of the tests. The oil has several direct effects on performance:

1. Reduction in gas space volume in the compression chamber and a corresponding increase in the time rate of change in the gas space volume.
2. Heat exchange with the refrigerant vapor.
3. Sealing of leakage gaps.
4. Power required for friction and acceleration of the oil.
5. Introduction of vapor from the high pressure oil separator through release of dissolved refrigerant and carrying and release of dissolved refrigerant through leakage gaps.

All of these factors will affect the pressure in the compression chamber. To illustrate the effect of varying oil injection parameters, data from the runs with the plain bore injection ports for the points of highest and lowest efficiency shown in Figure 2 are compared. The run with the lowest efficiency is that with the highest flow of uncooled oil through port A (test run number 14) while the highest efficiency is achieved with injection of cooled oil at the lowest flow rate through port B (test run number 18). The pressure-time data for these points are compared in Figure 5.

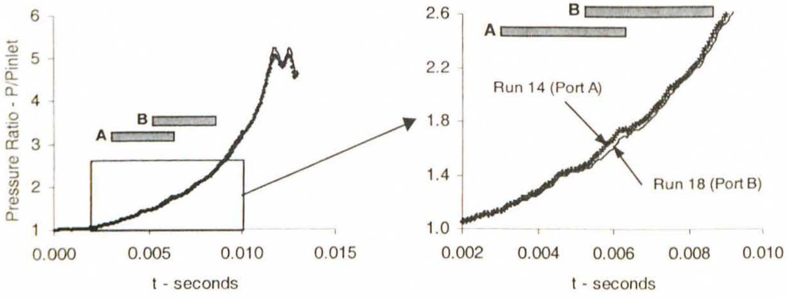


Figure 5
Pressure-Time Data for Plain Bore Injection Port Runs

The entire compression process is shown on the left side of the figure. The portion of the process during which oil is injected into the compression chamber (box in the chart on the left) is magnified and shown in the right hand side of the figure. The range of compression over which oil is injected is identified by the cross-hatched boxes, one for port location A and one for location B.

The relative performance of the two runs compared is shown in Table 1. The test using port A at the highest flow rate is the reference run for all tests and all performance parameters are shown with values relative to the data recorded for this reference condition. As can be seen in the table, the isentropic efficiency increased by 4.3% with the lower flow of cooled oil, injected through the higher pressure port B location. The improvement realized consisted of a 3.2% reduction in indicated work and a 1.1% increase in delivered flow (volumetric efficiency).

Table 1
Lowest (Run 14) and Highest (Run 18) Efficiency with Plain Bore Injection

Test:	14 - Reference	18
Injection Port	A	B
Oil Temperature	80°C	57°C
Oil Flow (l/m^3)	4.0	1.1
Isentropic Efficiency	1.000	1.043
Volumetric Efficiency	1.000	1.011
Power	1.000	0.9683
Indicated Work	1.000	0.9710

Using the measured pressures, the indicated work (area of the pressure-volume curve) of run 18 is 97.1% of that measured for run 14, only 0.3% different from the ratio of the measured input powers for the two runs.

Examining the pressure data in Figure 5, we can see that uncooled oil injected into port A results in higher pressures than seen in the case of the injection of cooled oil into port B, beginning almost immediately at the start of the injection process with port A.

Injection oil affects compressor performance as defined at the beginning of this section, and these factors are seen collectively in the pressure-time data. However, the individual effects cannot be separated using only the pressure data acquired during these tests. One way to begin to separate out the effects is through use of the thermodynamic model developed for design and analysis of refrigerant screw compressors /6/. The model contains analyses of the five effects listed. The effects of the change in gas space volume and the introduction of the vapor released from the refrigerant can be quantified fairly well in the model, the latter effect as a result of tests carried out during this study and reported here in Section 4. Following the review in Section 4, some analyses of the test results made using the thermodynamic simulation are presented in Section 5.

4 Oil Flow Through Injection Ports

A special tester was constructed to allow visualization of the oil flowing through the various injection port geometries used in the compressor tests. The test section is shown in Figure 6. A special cartridge insert is made with the bore or nozzle configuration to be tested. This cartridge is mounted in the test head, part A in the figure. There is a toothed-wheel on the test head that can rotate during testing to simulate the passing of the screw rotor lobe tips. Windows B, C and F allow for lighting, viewing, photographing and video taping of the flow leaving the injection port.

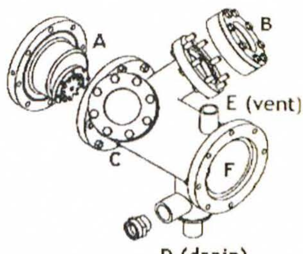
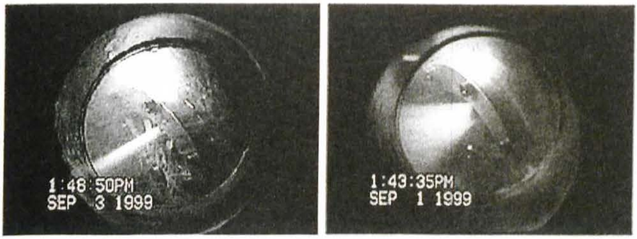


Figure 6
Injection Port and Nozzle Test Section

The test section is connected to the compressor on the test stand. Oil is provided from the compressor high pressure oil separator to the cartridge at location A. A valve and flow meter are provided in the line to control and measure the oil flow. Oil flows into the test section where flow patterns are visualized and recorded through the windows. The oil collects in the chamber and flows out through the drain, D, which is connected to the compressor inlet pipe. A valve in this line is used to insure that the chamber always has oil in the drain sump. Refrigerant dissolved in the oil in the compressor separator is released as the fluid expands to the lower test section pressure through the nozzle. The vapor can flow through the vent, E, which is also connected to the compressor inlet pipe. A valve in this line can be used to control the test section pressure and a flow meter is provided to measure the amount of refrigerant released from the injected liquid.

Flow through a plain bore injection port as seen in the tests is shown in Figure 7a. Figure 7b shows the flow through the fan nozzle. In these photos, the view is through window B (Figure 6) with the light source at window C. In this test, the wheel was stationary and the flow through the ports is directed between two of the teeth.



7a -- Plain Bore 7b -- Fan Nozzle

Figure 7
Visualization of Flow Through Injection Ports

The visualization tests were useful in determining the flow patterns from the ports, especially from the fan and cone nozzles. There was some question as to whether or not the nozzles would provide the proper shape and atomization due to the effect of the release of the refrigerant vapor. However, as can be seen in Figure 7b, the fan nozzle provided the desired flow pattern (Figure 3); the same was true for the cone nozzle (not shown).

In addition to the visualization, the measurement of the liquid flow in and the liquid and vapor flows out of the test section provided valuable data on the release of dissolved refrigerant from the injection oil.

A model of the release of oil was developed for use with the thermodynamic simulation. Oil-refrigerant property data provided by the lubricant supplier is used to define the amount of refrigerant dissolved in the oil based on the pressure and temperature of the liquid in the oil separator. This information allows computation of the enthalpy of the liquid. An iterative scheme is then employed to compute the amount of refrigerant in solution after an isenthalpic expansion to a lower pressure, as would occur at the injection port.

After the injection port tests, the measured and computed refrigerant flows could be compared. The results are shown in Figure 8 where the measured flow is compared to the calculated flow. The solid symbols show the data and open symbols the computed results.

The data collected for display in Figure 8 was taken with the compressor running at its fully loaded rating condition. In this case, the amount of refrigerant released from the oil at the low oil flow rates (data around 0.3 liters/m^3) is about 0.5% of the compressor's inlet mass flow rate. At the higher oil flows tested (data points around 1.1 liters/m^3) the rate of refrigerant released from the oil is about 1.5% of the total compressor flow rate.

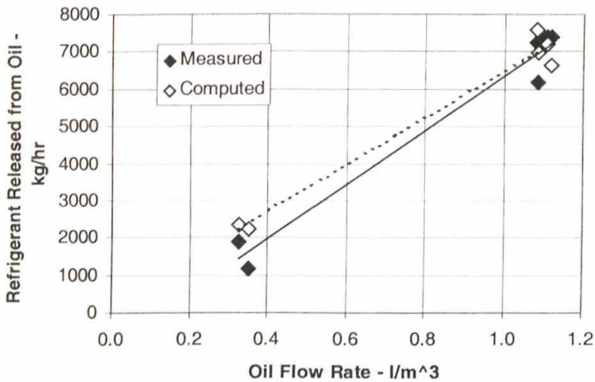


Figure 8
Refrigerant Released from Injected Oil in Nozzle Tester

A hypothesis of the study is that differences in the amount of refrigerant released from the injection oil and in the location of this release in the compression process will cause measurable changes in the overall compressor performance. Results from the nozzle tests in Figure 8 show that the computed refrigerant release rate characteristics are similar to the measurements. Thus, it should be possible to approximate the effect of this factor on compressor performance with a thermodynamic simulation of the compressor using the refrigerant release model in the analysis. Results of these studies are reviewed in the next section.

5 Performance Analysis using a Thermodynamic Simulation Program

A comprehensive thermodynamic simulation for semi-hermetic screw compressors /6/ is used to compute the effect of changes in the oil injection details tested during this project. The effects of the various oil injection schemes on overall performance and indicator diagram details have been reviewed in Sections 2 and 3. One important detail of the oil injection, the amount of dissolved refrigerant released during the injection process, was measured as described in Section 4.

The purpose of the simulation analysis is to quantify the effect of the release of dissolved refrigerant on compressor performance. For this report, the simulation is carried out for test runs 14 and 18, tests using plain bore injection ports with the lowest and highest efficiencies, respectively. All modeling inputs are the same for the simulation of each run, except for the actual operating conditions and details of the oil injection. The nominal operating conditions specified for the tests are the same for both runs. In the actual tests, small differences in the inlet and discharge pressures resulted in operating pressure ratios for runs 14 and 18 of 4.7 and 4.6.

The primary differences in runs 14 and 18 are that run 14 was run with a high injection flow rate of uncooled oil into the port in location A while run 18 was run with a low flow of cooled oil into port B. Using computed pressures and temperatures in the oil separator and the calculated average pressure at the oil injection port, the oil-refrigerant model was used to determine the amount of refrigerant released from the oil during the injection process. The simulation program input allows for specification of the size and location of the injection port and details of the oil supply from the separator. A restriction in the oil supply line is used to simulate the control valve used in the tests. The value of the restriction factor was adjusted in the model until the computed injection flow rate was the same as that measured for each run.

Results of the simulation are shown in Figure 9. In this comparison, all efficiencies are compared to the actual test efficiency for run 14. The simulation resulted in a computed efficiency for run 14 equal to the measured value (relative efficiency = 1.00). The relative efficiencies from test and simulation for run 18 are 1.043 and 1.062, respectively.

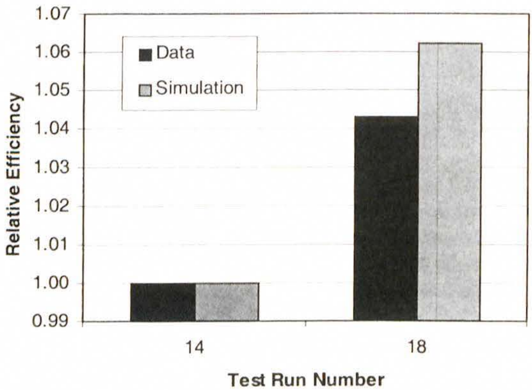
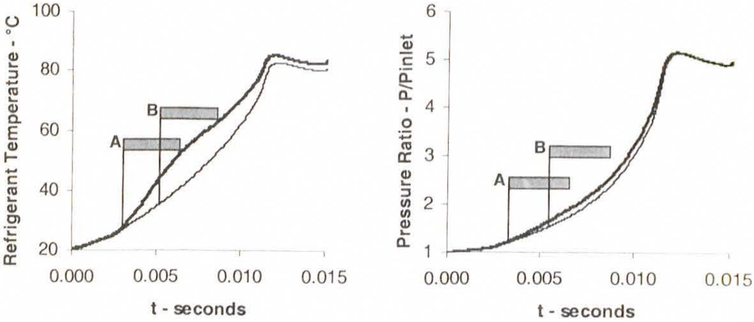


Figure 9
Measured and Computed Performance Comparison

According to the oil/refrigerant model, the amount of refrigerant released from the injected oil is 16.9% of the injected mass flow for run 14 and 20.9% for run 18. The actual amount of refrigerant released from the oil is 4.5% of the compressor inlet flow rate for run 14, where the total oil injection flow rate is highest, and 1.9% for run 18 with the lower injection rate.

In addition to the higher amount of refrigerant released, the higher flow rate of uncooled oil returns more heat to the compression process for run 14. Based on results from the simulation, the uncooled oil injected during run 14 adds 1.2 kW of heat to the compression while the lower flow rate of cooled oil for run 18 removes 2.2 kW from the compressed gas. The result is a net benefit for the run 18 injection configuration as the compression after the oil is injected takes place at a lower temperature.

The computed refrigerant temperatures and pressures during the compression process are shown in Figure 10. Calculated temperatures are shown in Figure 10a and pressures in 10b, presented as the ratio of the compression space pressure to the inlet pressure. The heavy line in the figure represents run 14 where uncooled oil is injected into port A. The computed temperatures and pressures for run 18 are shown with the lighter weight line. In this case, cooled oil is injected into port B, located further into the compression process than port A. The cross-hatched bars show the range in the compression process during which oil is injected into the ports A and B.



10a -- Temperature

10b -- Pressure Ratio

Figure 10
Computed Properties During the Compression Process

Data from Table 1 is reproduced in Table 2, where the results of the computed differences in performance between runs 14 and 18 are compared to the measured differences. As seen in Figure 9 and in the table details, the calculated efficiency improvement of run 18 relative to run 14 is greater than that actually realized in the tests.

Table 2
Lowest (Run 14) and Highest (Run 18) Efficiency with Plain Bore Injection

		Run 18	
Test:	**14 - Reference**	**Test**	**Computed**
Injection Port	A	B	B
Oil Temperature	80°C	57°C	57°C
Oil Flow (l/m^3)	4.0	1.1	1.1
Isentropic Efficiency	1.000	1.043	1.062
Volumetric Efficiency	1.000	1.011	1.010
Power	1.000	0.9683	0.9510
Indicated Work	1.000	0.9710	0.9489

One possible explanation for the difference in the measured and computed benefits is that the reduced injection flow rate -- 8 l/min for run 18 compared with 28 l/min for run 14 -- leads to less effective sealing in the internal clearances, with increased leakage offsetting the improvement realized from the reduction in refrigerant released into the compression with the injection oil. In the calculations, leakage flow areas and flow coefficients are the same for both runs.

Tests with the fan and cone nozzles showed that performance was improved slightly when compared to the plain bore tests (Figure 4) at the same injection rate, injection location and injection temperature. Computations with varying oil-to-refrigerant heat transfer and leakage flow coefficients show that a combination of increased heat transfer (which would arise from the increased surface area of droplets from the nozzle) and an increase in leakage were necessary to compute efficiency improvements seen in the tests. The model used is quite simple and work is still underway to understand the actual mechanisms by which the nozzles were able to result in the additional performance improvements seen in the tests.

6 Summary and Conclusions

Tests with a variety of oil injection options were carried out with an R-134a screw compressor. We looked at variations in flow rate, injection temperature, location of the injection port and port configuration -- plain bore or spray nozzle. The overall result was that compressor efficiency could vary by 4.3% depending on the choice of injection parameters when using a plain bore as the injection port. Use of spray-type nozzles provided an additional 0.5% efficiency improvement.

For the refrigeration compressor, use of a low flow rate of cooled oil injected relatively later into the compression process provided the best results. The improvement is due to a combination of a reduction in the amount of refrigerant carried from the higher pressure side in the oil to the compression space and a reduction in compression temperature. Calculations of these effects predict a larger improvement than was measured.

Indicator diagram measurements provided some insight into the effects of oil injection on the compression process. Comparisons of measured and computed indicator diagrams (not discussed in this report) show that calculations, while having the same general trends in indicator diagram changes, do not agree with the data in significant details. This means that further refinement of the heat transfer, refrigerant release and leakage models is necessary to arrive at a more fundamentally correct model. Improvements in the model can be made through better understanding of the injection flow details -- drop sizes, distribution of oil in the compression spaces, heat transfer coefficients, etc. A model of the effect of oil on leak path sealing that reacts to changes in the amount and distribution of oil in the compression spaces and to absorption and release of refrigerant during the compression process itself is also necessary.

A lower flow of cooled oil injected at higher pressures through a cone nozzle provides the best performance for the R-134a compressor. However, considerations of operating range, cost and noise will affect the final choice of injection details. Understanding of the effects of the choices and accurate modeling of an oil-injected refrigeration compressor allows the most appropriate configuration to be selected during the compressor design phase. The work reported here represents some progress in this direction. This project has also identified areas of lesser understanding where further progress can be made.

7 References

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