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ABSTRACT

System tests were performed comparing the zeotropic refrigerant blend R-407A, (HFC-32/HFC-125/HFC-134a (20/40/40 wt.%)), and R-502 to investigate the effect of heat exchanger configuration on the overall system performance. The comparison was carried out in a vapour-compression cycle test facility in which the inlet conditions of each heat exchanger, on the heat transfer fluid side, was held constant. Tests were performed on a constant capacity basis by varying the compressor speed. The heat transfer fluids for the condenser and evaporator, which were both divided into four sections, were water and water-methanol respectively. The evaporator and condenser, both of a tube-in-tube type construction, were each operated in parallel and counter-flow.

Test data were used to compare important compressor performance characteristics including shaft power and suction pressure. System performance characteristics were also measured including refrigerant mass flow, and cooling coefficient of performance (COP). Conditions on the heat transfer fluid side of the heat exchangers were kept constant for each configuration, so that any changes to the system performance would be due to changes on the refrigerant side. Factors affecting the overall performance were refrigerant side heat transfer coefficients and the capacity distribution in each heat exchanger.

INTRODUCTION

Due to the ongoing phase out of CFCs, many studies have recently been performed investigating the benefits and disadvantages of using zeotropic refrigerant mixtures. Much has been written about the theoretical thermodynamic advantage that may be obtained in a counter-flow configuration to take advantage of glide matching of heat source and sink temperatures (Pannock et. al. 1992). Studies have shown that for the amount of temperature glide available with R-407A (less than 7°C), modest performance benefits can be gained through glide matching. The magnitude of the effect on system performance when using pure counter-flow or parallel flow heat exchangers has received less attention. In general, parallel flow situations are avoided in the design of heat exchangers, but in some cases parallel flow is unavoidable. Often a heat exchanger will operate as some combined function of counter and parallel flow. Several studies (Linton et. al. 1994, Haines et. al. 1994, Berge et. al. 1993), have shown varying performance of R-407A as compared to R-502. This may be due in part to the different heat exchanger configurations. This paper will focus on the two extreme conditions of pure counter-flow and parallel flow to investigate effects on the relative performance of R-407A compared to R-502 in an experimental vapour compression test loop. The loop used a water-methanol circuit as the heat source and a water circuit as the heat sink.

The refrigerant blend R-407A is a zeotropic mixture of HFC-32/HFC-125/HFC-134a (20/40/40 wt.%) that is being used as a zero ozone depletion substitute for R-502. At typical condensing temperatures the theoretical temperature glide of R-407A is about 4°C (7.2°F) and for typical freezer temperatures it has a theoretical temperature glide of about 6°C (10.8°F). However, in an actual evaporator the available temperature glide is generally less than the theoretical, due to the liquid refrigerant that flashes to vapour when passing through an expansion device. Pressure drop in a heat exchanger can also affect the amount of glide available by changing the dew or bubble point temperatures. This tends to decrease the available glide in an evaporator and increase it in the condenser.

TEST DESCRIPTION

Test Facility

The study was performed in the Alternative Refrigerants Test Facility located at the Thermal Technology Centre, NRC and a complete description of the test facility can be found in Linton et. al. (1994). The facility consisted of three closed loops. The refrigerant loop used an open drive two cylinder reciprocating compressor driven by a variable speed 3730 W (5 HP) electric motor. Shaft power input to the compressor was measured with a torque and speed sensor mounted between the drive shaft and the electric motor which was used to drive the compressor. The torque sensor was a strain gauge type with an accuracy of $\pm 5.6\times10^{-2}$ Nm ($\pm$0.5 lb-in).

The heat exchangers were of a tube-in-tube configuration with a double fluted enhanced tube of 28.6 mm (1.13 in) OD. inside of a straight tube of 31.7 mm (1.25 in) ID. Each heat exchanger was divided into four equal horizontal straight sections 1016 mm (40 in) in length, which were joined together with short U-tube interconnecting pieces. Each quarter section was instrumented such that an accurate overall heat balance could be conducted over the entire heat exchanger. Both the evaporator and condenser were equipped with valves so that the flow direction of the heat transfer fluid in either heat exchanger could be reversed. The evaporator heat source was provided by a water-methanol circuit and heat was rejected from the condenser into a water circulation circuit. In the condenser, the refrigerant flowed in the annulus and in the evaporator it flowed through the inner tube.

Refrigerant flow was regulated by a microprocessor control unit that opened and closed an electronic expansion valve. Refrigerant mass flow was measured directly with a Coriolis effect mass flowmeter mounted in the liquid line leaving the condenser. All the mass flowmeters were calibrated to provide an accuracy of $\pm$0.5% of measurement. Platinum RTD precision digital thermometers with an accuracy of $\pm$0.02°C ($\pm$0.036°F) were used to measure the temperature of water and water-methanol entering and leaving the various heat exchanger sections. In-stream refrigerant temperatures were measured with copper-constantan thermocouples inserted into thermowells positioned in the bulk of the refrigerant flow. The uncertainty of the thermocouple temperature measurements was $\pm$0.6°C ($\pm$1.1°F). Refrigerant pressures were measured using pressure transducers connected to static pressure taps located at strategic points in the system. The pressure transducers were calibrated to $\pm$0.25% of full scale.

The heat exchanger capacities were calculated from the heat transfer fluid flowrate, specific heat capacity and temperature difference. These values were checked against the refrigerant side using the refrigerant flowrate and change of enthalpy. The thermodynamic properties for R-407A and R-502 were calculated using REFPROP version 4.01. For the data presented, 81% of the data points had an evaporator heat balance within 2.5%. The evaporator heat balance was corrected for oil circulation which was found to be about 1.5%. Suspected sources of error in the evaporator heat balance which were not accounted for were, refrigerant held in the circulating oil, preferential oil solubility, refrigerant properties and the specific heat capacity of the water-methanol at low temperatures.

Test Method

In compressor calorimeter type tests, the refrigerant conditions at the compressor inlet and outlet are defined. This has lead to much debate as to how the evaporating and condensing temperatures for zeotropes should be defined. An alternate method is to set the heat source and sink inlet temperatures and flowrates and let the refrigerant temperatures float. The advantage of this method is that a definition of the evaporating and condensing temperatures is unnecessary. Tests using this method can also be made on a constant capacity basis if the compressor volumetric flowrate can be varied. This will lead to equal inlet and outlet conditions for the heat transfer fluids and therefore, the heat transfer coefficients on the heat transfer fluid sides will be constant. Any changes in performance of the heat exchangers can then be attributed to heat transfer effects on the refrigerant side. This is the approach used in this study and is similar to a method suggested by McLinden and Radermacher (1987).

Baseline results were obtained using R-502 and an alkylbenzene oil with a viscosity of 32 mm^2/s at 37.8°C (100°F). For the first test series, the evaporator was operated in counter and parallel flow while a counter-flow configuration was maintained in the condenser. In the evaporator, water-methanol inlet temperatures and flowrates were selected to achieve evaporating temperatures of -15°C (5°F) and -25°C (-13°F) for counter-flow and parallel flow configurations using R-502. A separate base case was established for parallel and counter-flow so that the compressor speed would be the same for both flow configurations using R-502. This was done to minimize changes in the volumetric efficiency of the compressor. For a given evaporating temperature, the cooling capacities for counter and parallel flow were equal due to the selection of the water-methanol inlet temperatures. The combination of the selected flowrates and inlet temperatures resulted in a water-methanol temperature difference of about 5°C (9°F) which was about 1°C (1.8°F) higher than the available temperature glide of R-407A in the evaporator. A superheat of 5°C (9°F) was maintained at the outlet of the evaporator. On the condenser side, which was operated in counter-flow, inlet water temperatures and flowrates were selected that resulted in a condensing temperature of 43.3°C (110°F) for R-502. A subcooling of 6°C (10.8°F) was maintained at the condenser outlet. When the R-502 baseline tests were completed, the system was recharged with R-407A and a polyol ester oil with a viscosity of 32 mm^2/s at 37.8°C (100°F). The inlet conditions for each heat exchanger determined from the R-502 base tests were then used for the corresponding R-407A tests. The compressor speed was varied as necessary to match the capacity of each corresponding R-502 test. The volumetric capacity of the compressor changes slightly with the speed, and no correction was applied for this effect. Since the capacity was constant for a given test condition, the flowrate of a particular refrigerant was similar in parallel and counter-flow.

A second series of tests similar to the first were conducted in which the condenser was operated in parallel and counter-flow and the evaporator was kept in a counter-flow configuration. Water and water-methanol temperatures were selected such that for R-502, evaporating temperatures of -15°C (5°F) and -25°C (-13°F) and a condensing temperature of 43.3°C (110°F) were achieved.

TEST RESULTS AND DISCUSSION

Figure 1 shows the relative cooling coefficient of performance (COP) of R-407A versus R-502 when switching from parallel to counter-flow in the evaporator. The first two bars show the results for the tests which had an evaporating temperature of -15°C (5°F) for R-502. The graph shows that for parallel flow, the COP of R-407A was 0.96 that of R-502 and in counter-flow it was 1.05. The third and fourth bars show similar results for R-502 evaporating at -25°C (-13°F), except that in counter-flow the relative COP of R-407A was 0.99 as compared to R-502.

The mass flowrate of R-407A is generally about 30% less than R-502 for similar capacity requirements. For this reason, the two different evaporating temperatures for R-502, which resulted in two different capacity levels, were employed to investigate the effect of mass flux on the relative performance of the heat exchanger configurations. The measured mass flow rate of R-407A was 31% lower than that of R-502 for both evaporating temperature levels. At the higher evaporating temperature, (higher capacity), the relative COP of R-407A, as compared to R-502, increased by 9% when switching from parallel to counter-flow. At the lower evaporating temperature (lower capacity), switching from parallel to counter-flow only increased the relative COP by 3%. This smaller increase in the relative COP may have been partially due to the reduction in the refrigerant mass flux (and velocity) which adversely affected the heat transfer performance of R-407A more than that of R-502.

Since the evaporator capacity was constant for each test condition, the compressor power is inversely proportional to the COP values shown. In switching from parallel to counter-flow in the evaporator, the COP

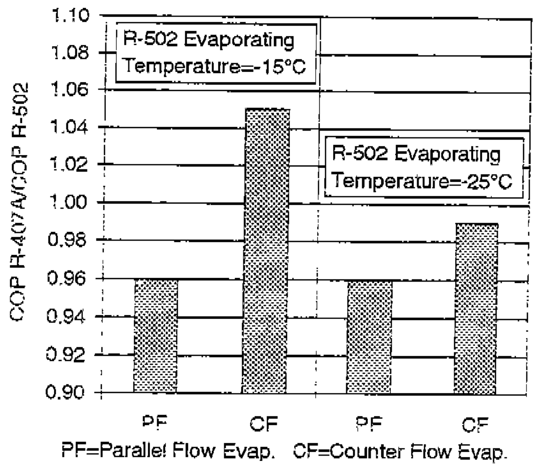

Figure 1 Relative COP of R-407A vs. R-502

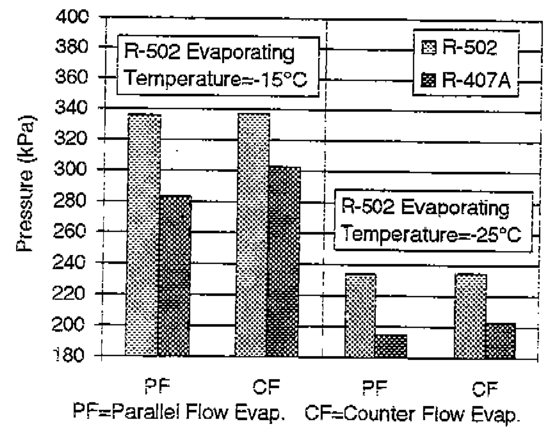

Figure 2. Compressor Suction Pressure

increased due to a decrease in compressor power. The decrease in compressor power could be mostly attributed to an increase in the suction pressure. Figure 2 shows that in switching from parallel to counter-flow, the compressor suction pressure of R-407A increased by 20 kPa (2.9 psi) at the higher evaporating temperature level, and by 8 kPa (1.2 psi) at the lower evaporating temperature. This increase was most likely due to a combination of better temperature profile matching and increased heat transfer performance which lead to a decrease in the overall temperature difference between the fixed water-methanol temperature and the floating refrigerant temperature. The suction pressures for R-502 remained relatively constant for each of the two evaporating temperatures due to the way the tests were defined, and not due to the heat transfer performance of R-502.

Although the overall evaporator capacity was constant for each test condition, the distribution of the capacity through the evaporator was different for each refrigerant. Figures 3 and 4 show how the capacity of each refrigerant was distributed through the four equal area sections of the evaporator, for the higher evaporating temperature level (higher capacity). The vapour qualities of each refrigerant entering the evaporator were very similar, (0.37 for R-502 and 0.35 for R-407A), and did not change appreciably with flow configuration. In parallel

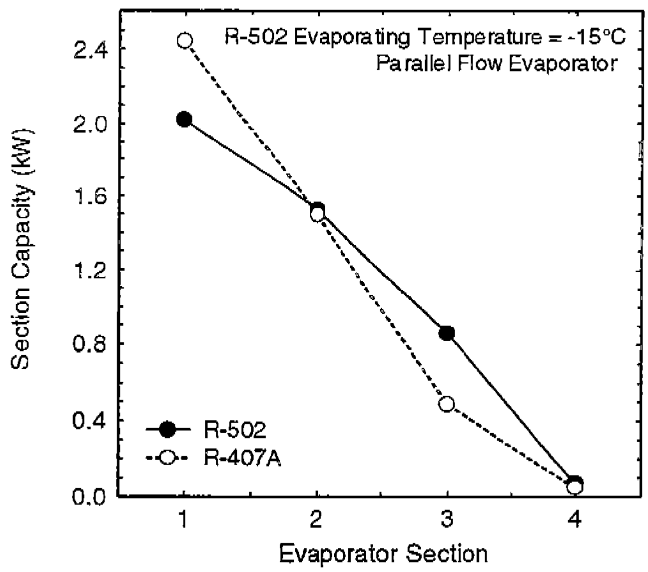

Fig. 3. Parallel Flow Evaporator Capacity Distribution

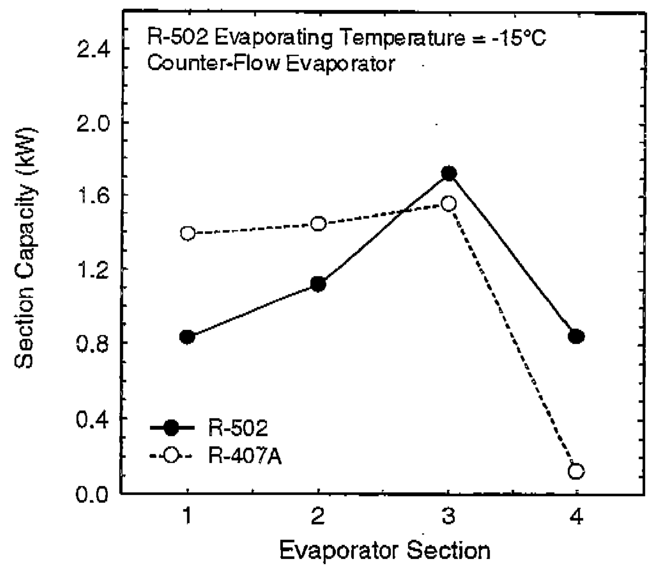

Fig. 4. Counter-Flow Evaporator Capacity Distribution

flow, the greatest amount of heat transfer occurred in the first section and then decreased linearly as evaporation proceeded. This was to be expected as the largest temperature difference between the two working fluids in parallel flow occurs at the inlet of the heat exchanger and then decreases. In the last section, where only superheating of vapour occurred, there was very little heat exchange. This was due to a small temperature difference and low vapour heat transfer coefficients. Figure 4 shows the distribution of the evaporator capacity in a counter-flow configuration. The capacity was more evenly distributed than in the parallel flow case, with the third section having the largest capacity. In the fourth section, the amount of heat transferred to the R-502 was much greater than that for R-407A. This was due to the lower vapour quality entering the section (0.89 for R-502 versus 0.99 for R-407A) which lead to better heat transfer. In both flow configurations, R-407A transferred more of its capacity in the first sections as compared with R-502. This may be partially due to the fact that, for R-407A at the beginning of the evaporation process, the refrigerant that is being boiled off is rich in the component with the lowest boiling point (HFC-32) which also has the highest latent heat of vapourization

Due to the fact that each test condition had the same area and flow conditions on the water-methanol side, the heat transfer coefficients, (HTCs), on the heat transfer fluid side could be considered to be the same for both refrigerants. Therefore the overall HTC should only be a function of the refrigerant side HTC. The overall HTC, U_i was calculated for each heat exchanger section as,

$$U_i = Q_i \,/\, (A_i \times LMTD_i) \qquad (1)$$

where Q_i is the section capacity, A_i is the section area and $LMTD_i$ is the section logarithmic mean temperature difference. Note that this equation assumes that the temperature profiles of each fluid passing through a given section are linear. This is probably a safe assumption for the water-methanol and for the refrigerants during evaporation. However, in sections where there is transition from evaporation to superheating, the refrigerant temperature profile will not be linear and therefore the overall HTCs calculated for these sections will be subject to some error. In all cases the first two sections of the evaporator contained only two-phase flow.

Figures 5 and 6 show the overall HTCs for parallel flow and counter-flow respectively. In sections one and two in parallel flow, (where two-phase flow existed), the overall HTCs for R-407A were 9% and 13% lower respectively, than for R-502. In the third section, the overall HTC of R-407A decreased significantly with respect to R-502. This was a result of the higher capacity of R-407A in the first two sections, which lead to a higher quality entering the third section (0.93 for R-407A versus 0.88 for R-502). Therefore, dryout occurred sooner for R-407A than for R-502. In parallel flow, the evaporation process was completed for both refrigerants in the third

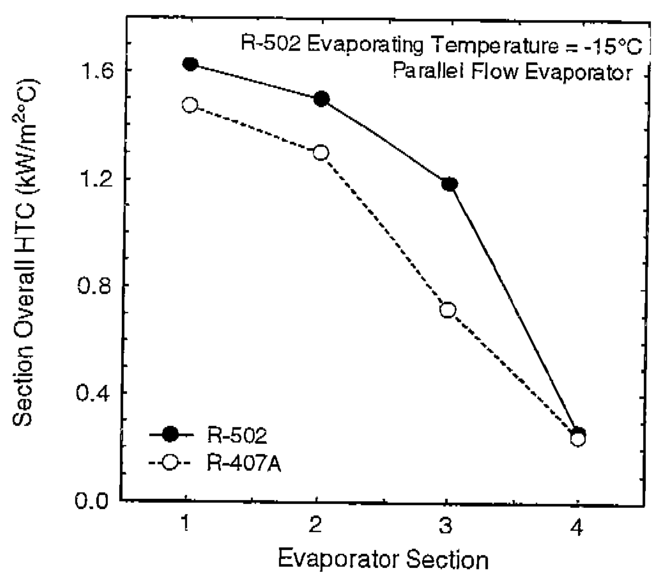

Fig. 5. Parallel Flow Evaporator Overall HTCs

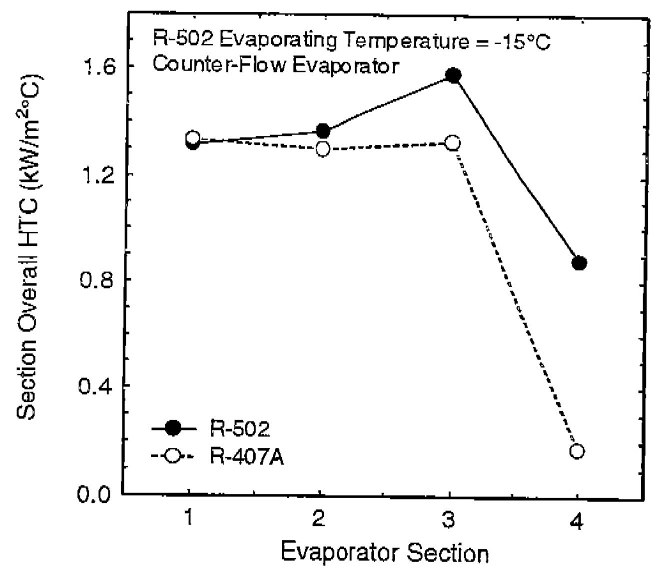

Fig. 6. Counter-Flow Evaporator Overall HTCs

section. In the fourth section, only superheating of vapour occurred and therefore the overall HTCs of both refrigerants were much lower than in sections containing two-phase flow.

In counter-flow, the overall HTCs of R-407A were +1%, -4% and -16% in the first three sections with respect to R-502. In the fourth section the overall HTC of R-407A decreased significantly compared to R-502. This was due to the higher vapour quality, (0.99 for R-407A versus 0.89 for R-502), entering the last section and mostly superheating taking place there. For R-502 in counter-flow, some evaporation still takes place in the fourth section, and therefore the HTC is a combination of evaporation of liquid refrigerant and superheating of vapour.

In the second series of tests, the condenser was operated in parallel and counter-flow and the evaporator was kept in a counter-flow configuration. Water and water-methanol temperatures were selected to produce R-502 evaporating temperatures of -15°C (5°F) and -25°C (-13°F) and a condensing temperature of 43.3°C (110°F) When switching from parallel to counter-flow in the condenser, the maximum relative COP increase of R-407A compared to R-502 was 3%.

CONCLUSIONS

A performance comparison was made between R-407A and R-502 in an experimental vapour compression test loop. Both the evaporator and condenser were operated in a parallel and counter-flow configuration. For a given test condition, the heat transfer fluid inlet conditions were kept constant and tests were performed on a constant capacity basis by varying the compressor speed.

In switching from parallel to counter-flow in the evaporator the relative COP of R-407A compared to R-502 increased from 0.96 to 1.05 at an R-502 evaporating temperature of -15°C (5°F) and a condensing temperature of 43.3°C (110°F). At an R-502 evaporating temperature of -25°C (-13°F), the relative COP of R-407A compared to R-502 increased from 0.96 to 0.99. The relative COP increase could be traced to an increase in the compressor suction pressure which lead to a decrease in the compressor power consumption. The evaporator was divided into four equal sections which allowed the capacity distribution to be plotted. This revealed that in both parallel and counter-flow, R-407A transferred more of its latent heat energy, as compared to R-502, in the first sections of the evaporator. When switching from parallel to counter-flow in the condenser, the maximum relative COP increase of R-407A compared to R-502 was 3%.

REFERENCES

J. Pannock, D. A. Didion and R. Radermacher. 1992. Performance Evaluation of Chlorine Free Zeotropic Refrigerant Mixtures in Heat Pumps, Computer Study and Tests, Proceedings of the 1992 International Refrigeration Conference at Purdue, West Lafayette, Indiana, July 14-17, Vol.1 pages 25-44

J. W. Linton, W. K. Snelson, A. R. Triebe, and P. F. Hearty. 1994. System Performance of Some Long Term R-502 Replacements, Proceedings of the 1994 International Refrigeration Conference at Purdue, West Lafayette, Indiana, July 19-22, pages 383-388

G. H. Haines, T. W. Dekleva, and R. E. Low. 1994. Performance Testing of R-502 Replacements Based on R-32/R-125/R-134a, Proceedings of the 1994 International Refrigeration Conference at Purdue, West Lafayette, Indiana, July 19-22, pages 395-400

J. Berge, S. L. Kwon, and L. Naley. 1993. Drop-In Test of Refrigerant Blends R-32/125/143a (10/45/45) and R-32/125/134a (20/40/40), Alternative Refrigerants Evaluation Program (AREP) report 87, Air-Conditioning and Refrigeration Institute (ARI), Arlington, Virginia

M. O. McLinden and R. Radermacher. 1987. Methods for Comparing the Performance of Pure and Mixed Refrigerants in the Vapor Compression Cycle, International Journal of Refrigeration, V.10, No.6, pages 318-325