International Refrigeration and Air Conditioning Conference

School of Mechanical Engineering

1992

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THE POTENTIAL OF HFC-134a AND HFC-152a TO REPLACE CFC-12 IN MEDIUM TEMPERATURE HEAT PUMP APPLICATIONS

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ABSTRACT

As part of a joint Canadian R&D program investigating environmentally acceptable alternatives to CFC refrigerants a series of tests has been completed to investigate refrigerants HFC-134a and HFC-152a as potential replacements for CFC-12 in medium temperature heat pump applications. The thermodynamic performances of the working fluids were measured on a drop-in basis for a system operating at evaporating and condensing temperatures consistent with such applications. The tests were conducted in a well instrumented water/water heat pump test facility, using an open-drive reciprocating compressor and counter flow heat exchangers.

Test measurements covered an evaporating temperature range from 11.0°C to 34.0°C (51.8°F to 93.2°F), which is broadly representative of the source temperatures for industrial heat pump applications. A constant refrigerant condensing temperature of 70.0°C (158.0°F) was maintained throughout the tests to represent typical heat exchange conditions required for process water heating. Tests were controlled to maintain a constant degree of condenser subcooling and superheat at compressor inlet.

Test data were used to compare important performance characteristics of the three refrigerants including heating capacity, heating coefficient of performance, refrigerant mass flow, compressor shaft power, and volumetric refrigerating capacity. The paper includes a brief description of the test facility and measurement techniques used.

INTRODUCTION

Many industrial processes produce waste heat at temperatures up to 35°C (95°F). The heat available at such temperatures is not useful for direct heat exchange purposes but can be effective in heat pump applications. Many examples of such applications exist presently in industries such as lumber drying and food processing (Reay et al. 1987). The elimination of CFC-12 as a medium temperature heat pump working fluid for condensing temperatures up to approximately 80°C (176°F) has created a requirement to identify an acceptable replacement with a zero ozone depletion potential. The obvious replacement would be HFC-134a, which is the leading candidate to replace CFC-12 for lower temperature applications. Another possible replacement refrigerant is HFC-152a if some level of flammability is acceptable.

Test measurements conducted at this laboratory and reported previously (Snelson et al. 1991) found that for a constant condensing temperature lower overall cycle efficiencies occurred at lower evaporating temperatures for refrigerant HFC-134a when considered as a drop-in substitute for CFC-12. These tests also indicated that overall cycle efficiencies of HFC-134a were higher than CFC-12 at higher evaporating temperatures (approaching 10°C (50°F)). These results, indicating improved performance with increasing evaporating temperatures, encouraged the authors to investigate the potential of HFC-134a under typical operating conditions applicable to CFC-12 in medium temperature heat pumps.

This paper describes a series of tests undertaken to provide a detailed comparison of the system performances of HFC-134a and HFC-152a with CFC-12 for evaporating temperatures above 10°C (50°F), with a constant condensing temperature of 70.0°C (158.0°F). These conditions are considered to be representative of typical source and sink temperatures encountered in industrial heat pump applications.

EXPERIMENTAL EQUIPMENT

Test Facility

Test measurements were conducted in the water/water heat pump test facility shown in Figure 1. A schematic flow diagram of the facility is shown in Figure 2. The system contains three closed loops.

The refrigerant circuit uses an open-drive, two-cylinder reciprocating compressor driven by a variable-speed, 3,730 W (5 hp) electric motor. An accumulator located in the compressor suction line is used during start-up conditions only, then valved off during testing. Refrigerant flow is controlled manually using an electrically operated expansion valve. The heat exchangers are counterflow tube-in-tube configurations, and each is divided into four equal horizontal straight sections of 1,016 mm (40 in) long copper tubing, joined with short U-tube interconnecting pieces. In the evaporator, the refrigerant flows inside double-fluted tubes of 28.6 mm (1.125 in) OD that are located inside smooth tubes of 31.7 mm (1.25 in) ID. The refrigerant flows through the surrounding annuli in the condenser sections. Each heat exchanger is surrounded by 152 mm (6 in) of fibreglass insulation.

Heat input to the evaporator is provided by a water-glycol circuit which contains a storage tank, circulating pump, Coriolis effect mass flowmeter, electric heater, and interchanger. The temperature of the water-glycol system is controlled by the heater, and flow to the evaporator can be modulated using throttle valves and a bypass line.

Heat produced in the condenser is transferred into a water circulation loop in which water is pumped from a storage tank through the inside tubes of the condensing sections. The water loop also contains a Coriolis effect mass flowmeter and interchanger, and the water temperature is controlled by partial heat rejection to a chilled-water system in a separate heat exchanger. The interchanger is used to transfer most of the heat produced in the condenser to reheat the water-glycol circuit. Water flow rates into the condenser can also be regulated by valves and a bypass line.

Instrumentation

Platinum RTD precision digital thermometers with an accuracy of ±0.01°C (±0.018°F) are used to measure the temperature of water and water-glycol entering and leaving the various heat exchanger sections, where the differences between inlet and outlet temperatures are likely to be small. In-stream refrigerant temperatures are measured with copper-constantan thermocouples inserted into thermowells positioned in the bulk of the refrigerant flows. The uncertainty of the thermocouple temperature measurements is ±0.5°C (±0.9°F). Elsewhere, system temperatures are measured with thermocouples soldered directly to the outside of the copper tubing and covered with at least 25.4 mm (1.0 in) of foam insulation.

Refrigerant pressures are measured using pressure transducers connected to static pressure taps located at strategic points in the system. The pressure transducers were calibrated to ±0.1% of full scale.

Refrigerant mass flow is measured directly with a Coriolis effect mass flowmeter mounted in the liquid line leaving the condenser. All the mass flowmeters were calibrated by the manufacturer and provided an accuracy of ±0.2 % of measurement.

Power input to the compressor is measured with a torque sensor mounted between the electric motor and compressor drive shaft. The sensor is a strain gauge type with an accuracy of ±0.1% of full scale, ie. $\pm 5.6\times10^{-2}$ Nm (±0.5 lb in) in this case.

Figure 1. Water/water heat pump test facility

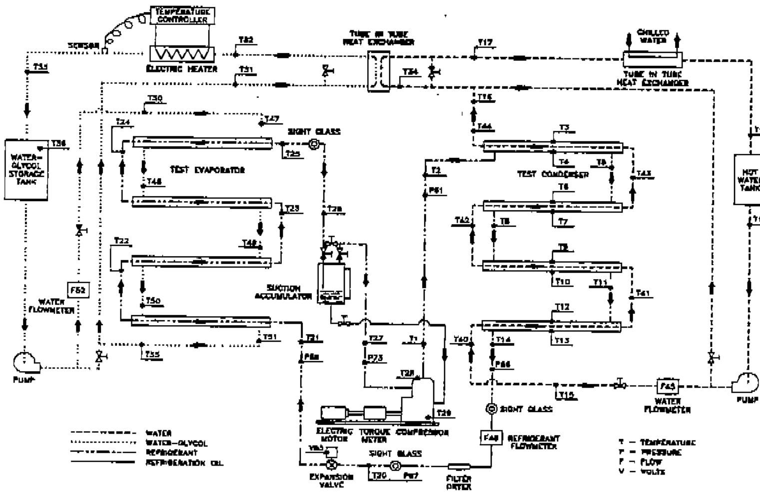

Figure 2. Schematic flow diagram of the water/water heat pump test facility

Data Acquisition

Channel output signals from instrumentation points are fed to a data acquisition/control unit and processed by a desktop computer. The only exceptions are the precision digital thermometers, which communicate directly with the computer via an interface bus. A computer program which controls the collection of data also performs all necessary calculations to carry out a complete heat balance check on the system. The refrigerant thermodynamic properties required for calculation purposes were determined from a commercially available software program (Morrison and Gallagher 1990).

TEST MEASUREMENTS

The refrigerant condensing temperature was maintained constant at 70.0°C (158.0°F) throughout the tests by controlling the flow and temperature of water into the condensing section. The degree of subcooling at the condenser outlet was maintained at approximately 12.0°C (21.6°F) throughout the tests by adjusting the total refrigerant charge in the system. The refrigerant evaporating temperatures were varied over a range of 11.0°C to 34.0°C (51.8°F to 93.2°F) by adjusting the temperature and flow rate of water-glycol entering the evaporator. The refrigerant superheat at the compressor inlet was maintained at approximately 7.4°C (13.3°F) throughout the tests by manually adjusting the expansion valve. The compressor speed was held constant at 992 rpm.

Each set of test readings were taken under steady-state conditions which were typically reached in about one hour. The data acquisition system scanned all channel inputs frequently while the system was coming to steady state. When a satisfactory condition was reached, the corresponding scan data were retrieved, and the data acquisition program was used to process the raw data and make all the necessary system performance calculations.

On completion of the baseline tests with CFC-12, the compressor and refrigerant circuit were drained, flushed with CFC-11, and evacuated several times using first nitrogen and then HFC-134a to break the vacuum, according to a recommended procedure provided by the refrigerant manufacturer. The system was then recharged with HFC-134a, and the alkylbenzene-based lubricating oil was replaced by a polyalkyleneglycol (PAG) synthetic oil with a viscosity of 32 mm^2/s (150 SUS) at 40°C (104°F) supplied for this purpose by the refrigerant manufacturer. The same series of tests were then repeated for HFC-134a. After the HFC-134a testing was completed, the compressor and system were again drained and the system evacuated several times using first nitrogen and then HFC-152a to break the vacuum. The system was then recharged with HFC-152a and the same type of PAG synthetic oil. The same series of tests were then repeated for HFC-152a.

TEST RESULTS AND DISCUSSION

For heat pump applications the energy output from the system is measured by the condenser capacity. The experimental values of condenser capacities were obtained from mass flow measurements in the water circuit and from accurate differential temperatures of the water flow between the inlet and outlet of the condenser. For each refrigerant the corresponding condenser capacity was calculated for each evaporating temperature used in the series of tests. The results are plotted in Figure 3. This graph shows that all three refrigerants have approximately the same condenser capacity at the lower evaporating temperatures, but as the temperature increases the capacities of HFC-134a and HFC-152a increase, relative to CFC-12. HFC-134a produces the highest increase in condenser capacity, showing an increase of approximately 10% relative to CFC-12 at the higher temperatures.

Compressor shaft power inputs were determined from torque meter readings obtained under steady state conditions during the tests. The corresponding results obtained for the three refrigerants are shown in Figure 4. The shaft power requirement for HFC-152a and CFC-12 are approximately the same over the range of evaporating temperatures tested. However, HFC-134a has a relatively higher shaft power requirement, with the difference increasing up to about 9% compared to CFC-12 as the evaporating temperature increases.

Measurements of the refrigerant mass flow rate plotted in Figure 5 show widely different mass flow rates for the three refrigerants. HFC-134a and HFC-152a have much lower mass flow rates than CFC-12, and in the case of HFC-152a, the flow rate is approximately half that of

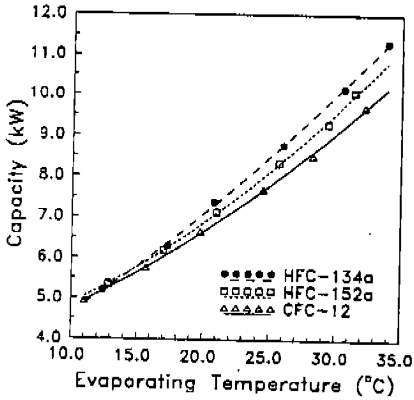

Figure 3. Condenser capacity

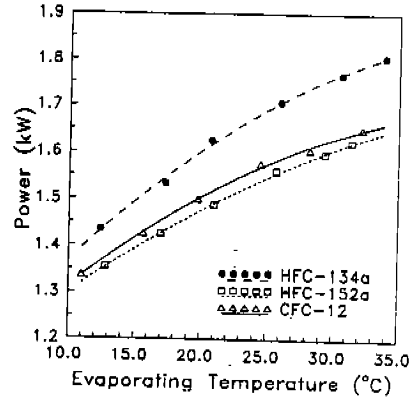

Figure 4. Compressor shaft power

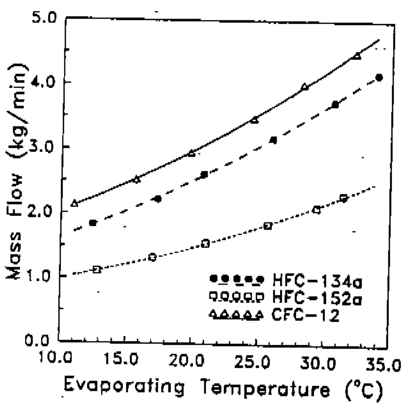

Figure 5. Refrigerant mass flow

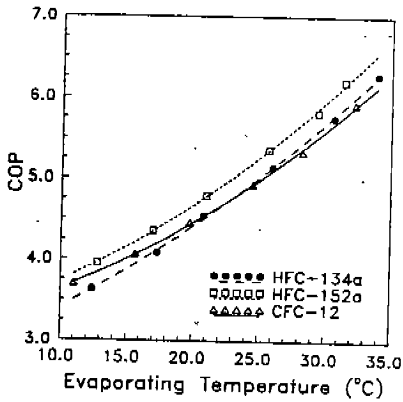

Figure 6. Coefficient of performance - heating

CFC-12. These differences in flow rate measurements combine with the different fluid enthalpy properties to produce the capacity curves referred to in Figure 3.

The coefficient of performance (COP) for heating was derived by dividing the condenser capacity by the shaft power input to the compressor. These results are shown in Figure 6. The graph shows that a crossover point occurs for the CFC-12 and HFC-134a curves at approximately 22.5°C (72.5°F) evaporating temperature. Below this point HFC-134a has a lower COP than CFC-12, and above this point HFC-134a has a slightly higher COP than CFC-12. Previous system performance tests were conducted on these three refrigerants by the authors (Linton et al. 1992) to determine the effect of condenser liquid subcooling on system performance at lower evaporating and condensing temperatures. Results obtained from those tests indicate that the crossover point for the CFC-12 and HFC-134a curves shown here in Figure 6 could be expected to shift to a lower evaporating temperature with increased liquid subcooling. Figure 6 also shows that HFC-152a has better performance than CFC-12 and HFC-134a for the entire range of evaporating temperatures tested. This reflects the corresponding higher heating capacity and lower compressor work requirements for HFC-152a.

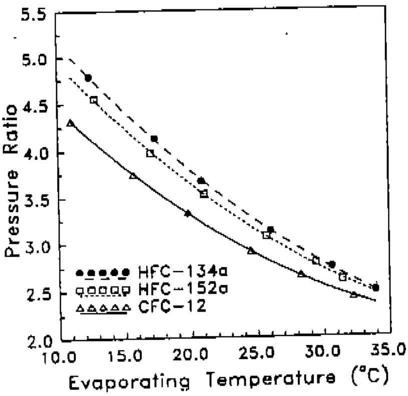

Figure 7. Compressor pressure ratio

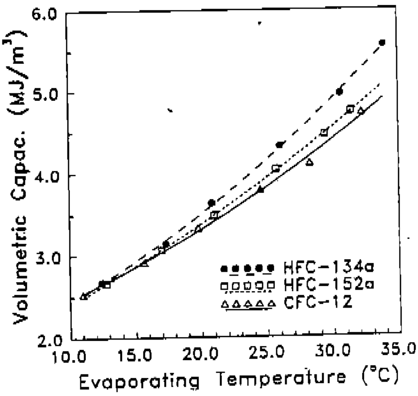

Figure 8. Volumetric refrigerating capacity

Another factor which has been shown to affect the performance of HFC-134a is the type and viscosity of compressor lubricant used. The test data reported here using the PAG synthetic oil can be compared to recent test results obtained with HFC-134a and lower viscosity ester type lubricants (Riffe 1991). These results showed improved compressor efficiency while still providing adequate lubrication of the compressor. The stability of HFC-134a and ester type lubricants has also been investigated in sealed tube tests and working systems, and has been reported to be equal to or better than CFC-12 and mineral oils (Hansen 1991).

Figure 7 provides a comparison of the compressor pressure ratio (compressor discharge pressure ÷ compressor suction pressure) for the three refrigerants over the corresponding range of evaporating temperatures. The graph shows that the pressure ratio for HFC-134a is approximately 15% higher than CFC-12 at the lower evaporating temperatures, with the two approaching each other at the higher evaporating temperatures. The compression ratios for all three refrigerants tend to converge as the evaporating temperature increases. The pressure ratio of CFC-12 is less than the other two refrigerants throughout the whole temperature range. Higher compression ratios influence the mechanical design requirements of the compressor, leading to the need for more robust construction of bearings, valves, gaskets, etc.

From the respective refrigerant properties data, the enthalpy and vapour density at compressor inlet can be determined from the measurements of suction pressure and temperature obtained for each test condition. Multiplying enthalpy by suction vapour density gives the corresponding volumetric refrigerating capacity, ie. the amount of heat removed from the heat source per unit volume of vapour drawn into the compressor. These results are plotted in Figure 8. The volumetric refrigerating capacity of the three refrigerants is identical at the lower evaporating temperatures, and diverges as the evaporating temperature increases. The volumetric refrigerating capacity is always greatest for HFC-134a at the higher evaporating temperatures tested. These curves indicate that for a given load a compressor operating with HFC-134a could be sized for a lower displacement than CFC-12. The potential size reduction could be up to 10% for the highest evaporating temperatures considered here.

CONCLUSIONS

A series of tests has been completed to investigate refrigerants HFC-134a and HFC-152a as potential replacements for CFC-12 in medium temperature heat pump applications. For all tests the refrigerant condensing temperature was maintained constant at 70.0°C (158.0°F). The refrigerant evaporating temperatures covered a range of 11.0°C to 34.0°C (51.8°F to 93.2°F). The refrigerant superheat at the compressor inlet was maintained at approximately 7.4°C (13.3°F) and the degree of subcooling at the condenser outlet was maintained at 12.0°C (21.6°F) throughout the tests.

The test measurements showed that as the evaporating temperature increases the condenser capacities of HFC-134a and HFC-152a increase, compared to CFC-12. HFC-134a showed the greatest increase in condenser capacity at the higher temperatures (up to 10% greater than CFC-12). However the significantly higher compressor shaft power requirements for HFC-134a leads to a COP which is close to that of CFC-12 over most of the temperature range of interest. For the COP a crossover point occurs at approximately 22.5°C (72.5°F) evaporating temperature, with HFC-134a having a slightly higher COP at temperatures above the crossover point. Previous work has shown that the crossover point could be expected to shift to a lower evaporating temperature with increased liquid subcooling. This implies that an improved performance relative to that obtained in these tests may be attained for HFC-134a by optimization of system operating conditions.

HFC-152a has better performance than CFC-12 and HFC-134a over the entire range of evaporating temperatures tested. This reflects the lower compressor work requirements for HFC-152a.

In summary these tests have indicated that HFC-134a will operate satisfactorily as a medium temperature heat pump working fluid, with performance equal to or better than CFC-12, depending on the system operating conditions. The performance of HFC-152a proved to be superior to CFC-12 and HFC-134a, but due to its flammability is unlikely to be considered as a working fluid, except as a component in refrigerant mixtures specially blended to be non-flammable.

ACKNOWLEDGMENTS

This test program was undertaken as part of an ongoing joint collaboration arrangement with Du Pont Canada Inc. and Energy Mines and Resources Canada.

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