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ABSTRACT

A series of tests has been completed comparing the system performance of four long term replacements HP62 (44% R-125/ 54% R-143a/ 4% R-134a), AZ-50 (50% R-125/ 50% R-143a), Klea 60 (20% R-32/ 40% R-125/ 40% R-134a), and Klea 61 (10% R-32/ 70% R-125/ 20% R-134a) as potential alternatives for the CFC-containing low temperature refrigerant R-502. The tests were carried out under controlled conditions in a well-instrumented alternative refrigerants test facility, using an open-drive reciprocating compressor and counter-flow heat exchangers. Measurements were conducted at two different condensing temperatures: 54.4°C (130.0°F) and 43.3°C (110.0°F). The refrigerant evaporating temperature was varied over a range of -30.0°C to -15.0°C (-22.0°F to 5.0°F).

Test data were used to compare important system performance characteristics with R-502 including relative evaporator capacity, relative cooling coefficient of performance (COP), and compressor discharge temperature. The effect of additional refrigerant liquid subcooling on the evaporator capacity and COP was also evaluated.

INTRODUCTION

R-502 is used extensively as a working fluid in low temperature commercial refrigeration equipment. It is an azeotropic blend of HCFC-22 (48.8%) and CFC-115 (51.2%). This refrigerant was originally developed to overcome the high discharge temperatures and oil return problems of HCFC-22 at low operating temperatures. R-502 also has superior hermetic motor cooling characteristics compared to HCFC-22. However, CFC-115 is an ozone depleting refrigerant that is covered under the Montreal Protocol and will be phased out by the end of 1995. The 1992 consumption of R-502 in Canada was about 600,000 kg.

All of the proposed near term R-502 replacements now being offered contain HCFC-22, which is also scheduled for phase-out starting in 2004, with virtually a complete phase-out by 2020. The designers and suppliers of new low temperature refrigeration equipment require a long term alternative solution to the replacement of R-502 to avoid another costly redesign of equipment in the future. Near azeotropic refrigerant blends that contain HFCs (ozone depletion potential of zero) have been developed as long term replacements for R-502. Table 1 lists the commercial name and the ASHRAE Standard 34 (1992) refrigerant number for the blends that were evaluated by this Laboratory, its refrigerant components, and the percent composition (by weight) of those components. The Table also provides the global warming potentials (GWP) of these blends (compared to CFC-11). For reference the GWP of R-502 is 3.74.

Table 1. Long Term R-502 Alternatives Evaluated

Refrigerant	Components	% Composition (by wt.)	GWP
HP62 (R-404A)	R-125/143a/134a	44/52/4	0.94
Klea 61 (R-407B)	R-32/125/134a	10/70/20	0.70
AZ-50 (R-507)	R-125/143a	50/50	0.96
Klea 60 (R-407A)	R-32/125/134a	20/40/40	0.49

This paper provides a summary of a series of tests that were undertaken to provide a comparison of the system thermodynamic performance of R-502, and the above refrigerant blends. The flexibility of this equipment enabled the investigation to cover a broad range of evaporating temperatures representative of conditions anticipated in low temperature commercial refrigeration system applications. A more detailed description of the test results for some of the above blends can be found in Linton et al. (1994), and Snelson et al. (1994).

EXPERIMENTAL EQUIPMENT

Test Facility

Test measurements were conducted in an alternative refrigerants test facility. The system contains three closed loops, (a more detailed description of this test facility is included in the above references).

The refrigerant circuit uses an open-drive, two-cylinder reciprocating compressor driven by a variable-speed, 3,730-W electric motor. Refrigerant flow is controlled using an electronically operated expansion valve. The heat exchangers are of counterflow tube-in-tube configuration, and each is divided into four equal horizontal straight sections of 1,016 mm long copper tubing, joined with short U-tube interconnecting pieces. In the evaporator, the refrigerant flows inside double-fluted tubes of 28.6 mm OD located inside smooth tubes of 31.7 mm ID. The refrigerant flows through the surrounding annuli in the condenser sections. The suction line to the compressor contains an electrically heated refrigerant superheater to maintain a constant return gas temperature to the compressor if required.

Heat input to the evaporator is provided by a water-methanol circuit that contains a storage tank, circulating pump, Coriolis effect mass flowmeter, electric heater, and interchanger. The temperature of the water-methanol system is controlled by the heater, and flow to the evaporator can be modulated by adjusting the pump speed.

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-methanol circuit. Water flow rates into the condenser can also be regulated by adjusting the pump speed.

TEST MEASUREMENTS

To achieve a fair comparison of a zeotrope to a single refrigerant or an azeotrope, the refrigerant cycle operating conditions need to be defined. The evaporating temperature is defined as the mean of the suction pressure dew point and the evaporator inlet temperature, and the condensing temperature as the mean of the discharge pressure dew point and the discharge pressure bubble point. The superheat and subcooling are measured from the suction pressure dew point and the discharge pressure bubble point respectively.

The refrigerant condensing temperature was evaluated at two different temperatures; 54.4°C (130.0°F) to simulate summer time extreme air cooled condensing conditions and 43.3°C (110.0°F) to represent more moderate air cooled condensing conditions. The condensing temperature was maintained at the required level by controlling the flow and temperature of water into the condensing section. Many HFCs have proved to be more sensitive to the amounts of system liquid subcooling than the CFCs and HCFCs they are replacing, (Linton et al. 1992; Domanski and Didion 1993). To investigate the influence that increased subcooling had on the blends the amount of liquid subcooling at the condenser outlet was varied between 6.0°C and 18.0°C (10.8°F and 32.4°F) by adjusting the total refrigerant charge in the system. The system performance was measured at 6.0°C (10.8°F) subcooling increments for each evaporating temperature. The refrigerant evaporating temperature was varied over a range of -30.0°C to -15.0°C (-22.0°F to 5.0°F) by adjusting the temperature and flow rate of water-methanol entering the evaporator. The refrigerant superheat at the evaporator outlet was maintained at approximately 5.0°C (9.0°F) throughout the tests by controlling the refrigerant flow with the electronic expansion valve. The refrigerant temperature at the compressor inlet was maintained at 18.3°C (65.0°F) for all the tests by adjusting the amount of electric heat added to the suction line superheater. The compressor speed was held constant at 1,940 rpm.

All test readings were taken under steady-state conditions that typically were 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 the necessary system performance calculations.

The baseline tests were first completed with R-502 and a commercially available alkylbenzene type lubricating oil with a viscosity of 32 mm^2/s (150 SUS) at 40°C (104°F). The system was then thoroughly cleaned out before testing any of the blends. All the blends were tested with a commercially available polyol ester oil with a viscosity of 32 mm^2/s (150 SUS) at 40°C (104°F).

TEST RESULTS AND DISCUSSION

The relative evaporator capacity of HP62 with respect to R-502 is shown in Figure 1 for 6.0°C and 18.0°C (10.8°F and 32.4°F) subcooling and for the 43.3°C (110.0°F) and 54.4°C (130.0°F) condensing temperatures. Figures 2, 3, and 4 show the relative evaporator capacities for Klea 61, AZ-50, and Klea 60 respectively at the same operating conditions. From Figure 1 it can be seen that the relative capacity of HP62 ranges from about 0.94 to 1.05 depending on the operating conditions and the amount of liquid subcooling present. The relative capacity of HP62 increases slightly as the evaporating temperature rises, and there is a definite increase in the relative capacity as the amount of subcooling is increased. Finally, the higher condensing temperature has a detrimental effect on the relative capacity of HP62. Figure 2 shows that the relative capacity of Klea 61 ranges from about 0.88 to 1.05 depending on the operating conditions and the amount of liquid subcooling present. The evaporating temperature has a stronger effect on the relative capacity of Klea 61 than it does on HP62. Finally, additional subcooling improves the capacity of Klea 61, but a higher condensing temperature has a negative effect on the relative capacity. From Figure 3 it can be seen that the relative capacity of AZ-50 ranges from about 0.95 to 1.05 depending on the operating conditions and the amount of liquid subcooling present. The relative capacity of AZ-50 increases slightly as the evaporating temperature rises, and there is a definite increase in the relative capacity as the amount of subcooling is increased. The relative capacity of AZ-50 is very similar to that of HP62. Figure 4 shows that the relative capacity of Klea 60 ranges from about 0.93 to 1.07 depending on the operating conditions and the amount of liquid subcooling present. Similar to Klea 61, the evaporating temperature has a stronger effect on the relative capacity of Klea 60 than it does for HP62 and AZ-50. Klea 60 is the only blend where additional subcooling does not show any additional improvement in the capacity relative to R-502.

Figure 5 is the relative Coefficient of Performance (COP) of HP62 with respect to R-502 for 6.0°C and 18.0°C subcooling and for the 43.3°C and 54.4°C condensing temperatures. Figures 6, 7, and 8 show the relative COP for Klea 61, AZ-50, and Klea 60 respectively at the same operating conditions. Figure 5 shows that for all the conditions tested HP62 is less energy efficient than R-502. The graph also shows that HP62 has a relative performance that ranges from 0.89 to 0.99. There is a definite improvement in the relative COP of HP62 as the amount of subcooling is increased, and at the lower condensing temperature. Figure 6 is the relative performance of Klea 61 with respect to R-502, this graph also shows that for all the conditions tested Klea 61 is less energy efficient than R-502. The relative performance of Klea 61 ranges from 0.88 to 0.96. Like HP62 the lowest relative performance also occurs at the highest condensing temperature and the lowest subcooling. The relative performance of AZ-50 is shown in Figure 7 which shows that for all the conditions tested AZ-50 is also less energy efficient than R-502. From the graph it can be seen that AZ-50 has a relative performance that ranges from 0.87 to 0.97. There is a definite improvement in the relative COP of AZ-50 as the amount of subcooling is increased, and at the lower condensing temperature. Figure 8 is the relative performance of Klea 60 with respect to R-502. The graph shows that Klea 60 is the only blend tested that had a higher energy efficiency than R-502 for all the conditions tested. The relative performance of Klea 60 ranges from 1.01 to 1.06. Klea 60 was also the only blend tested that showed a reduction in the COP compared with R-502 for increasing levels of subcooling. Klea 60 had the highest glide temperature of the blends tested, with a typical glide temperature at the evaporating temperatures

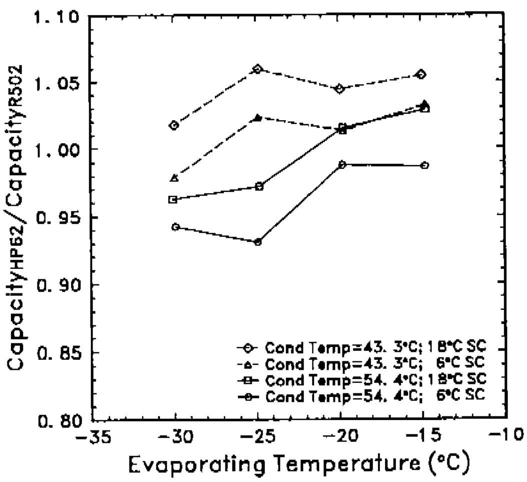

Figure 1. Relative evaporator capacity of HP62

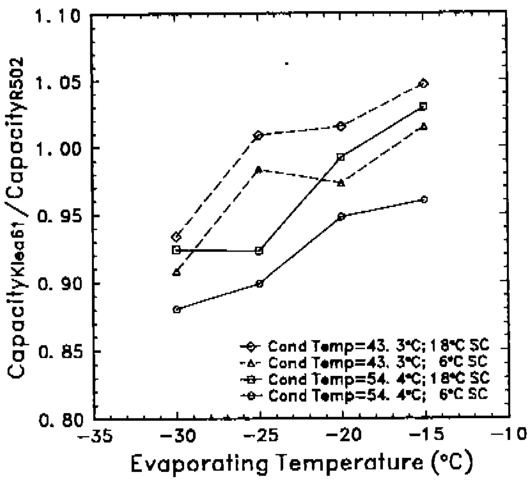

Figure 2. Relative evaporator capacity of Klea 61

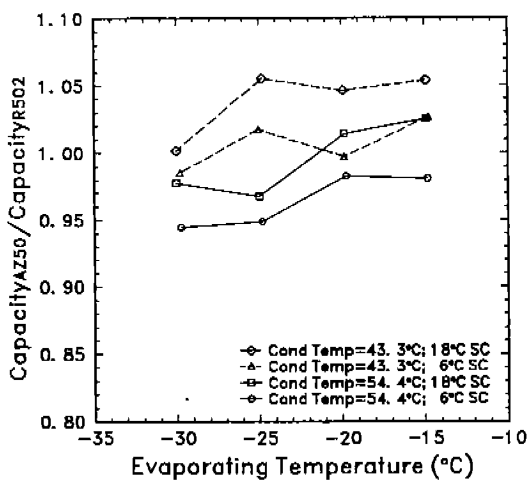

Figure 3. Relative evaporator capacity of AZ-50

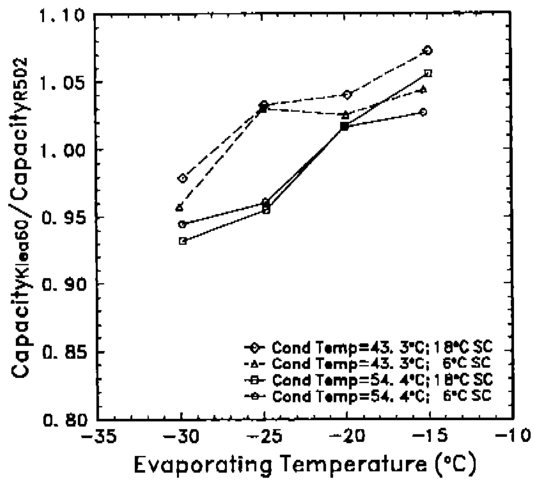

Figure 4. Relative evaporator capacity of Klea 60

tested of about 6.0°C (10.8°F). HP62 had a glide temperature of less than 1.0°C (1.8°F), and the glide for Klea 61 was about 4.0°C (7.2°F). AZ-50 had no glide as it is an azeotrope. The evaporator and condenser of the test facility were operated in a pure counter flow configuration. This factor can provide a performance advantage to a blend with a larger temperature glide due to the closer approaches of the source and sink temperatures.

Table 2 shows the relative change in system operating conditions for all the blends tested compared to R-502 for the 54.4°C (130.0°F) condensing temperature. The second column shows the relative change of the compressor discharge temperature for the range of evaporating temperatures tested. The discharge temperature was measured at the discharge valve of the compressor and with the refrigerant temperature at the compressor inlet held at a constant 18.3°C (65.0°F). The Table shows that HP62 and AZ-50 had about a 6.0°C (10.0°F) lower discharge temperature than R-502. Klea 61 had approximately a 1.0°C (1.8°F) lower discharge temperature than R-502. Klea 60 was the only blend that showed an increase in discharge temperature relative to R-502 with about an 8.0°C (14.4°F) increase. The third column of Table 2 shows the percent change of condensing pressure relative to R-502. HP62, Klea 61, and AZ-50 all showed an increase in condensing pressure ranging from 12% to 15%. Klea 60 had a condensing pressure that was about 5% higher than R-502. The fourth column of Table 2 is the percent change of the evaporating pressures for the range of evaporating temperatures tested. HP62 showed an increase in evaporating pressure ranging from 4.5% to 7.4%. Klea 61 had an increase in evaporating pressure that ranged from 0.2% to 5.9%. AZ-50 showed the largest change in evaporator pressure with a 8.2% to 10.0% increase.

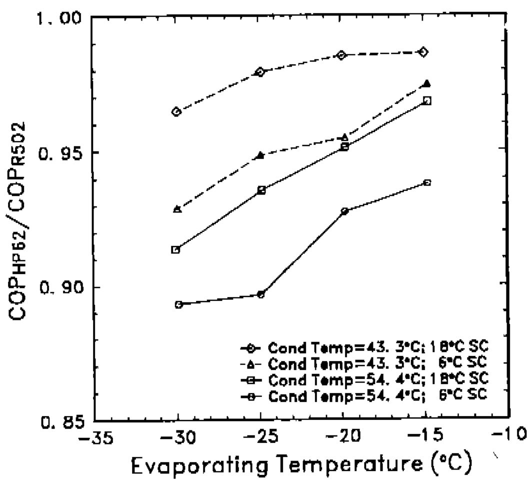

Figure 5. Relative COP of HP62

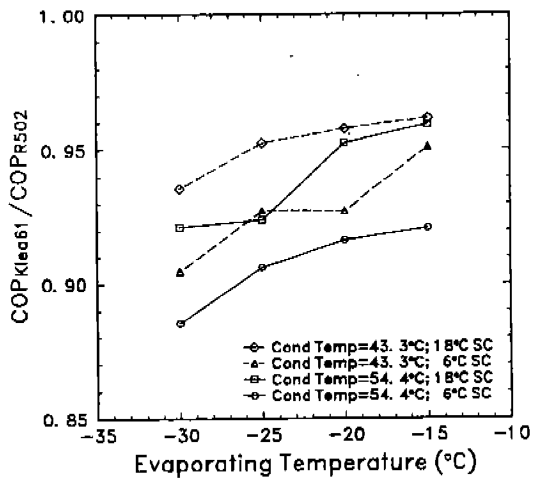

Figure 6. Relative COP of Klea 61

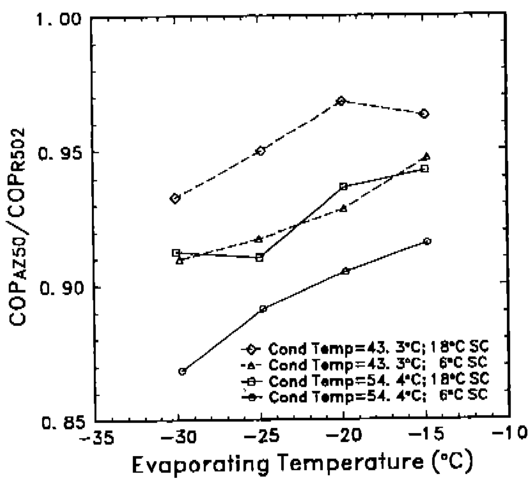

Figure 7. Relative COP of AZ-50

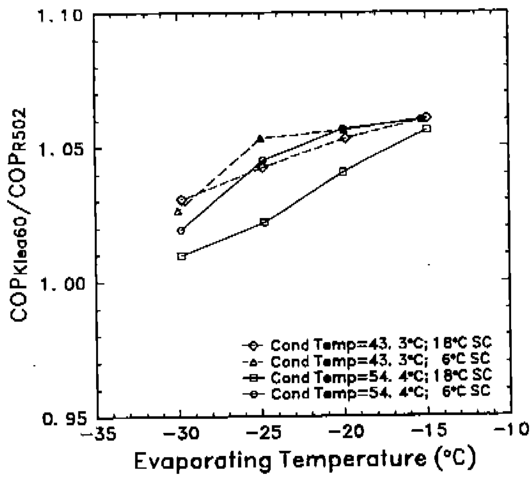

Figure 8. Relative COP of Klea 60

Finally, Klea 60 was the only blend that showed a decrease in evaporating pressure with a 11.6% to 14.0% drop in pressure relative to R-502. The last column of Table 2 is a comparison of the compressor pressure ratio for the range of evaporating temperatures tested. HP62 and AZ-50 showed a similar increase in compressor pressure ratio ranging from 3.2% to 5.3%. Klea 61 had an increase in compressor pressure ratio that ranged from 12.1% to 16.3%. Finally, Klea 60 had the largest increase in compressor pressure ratio ranging from 18.5% to 22.7% higher.

Table 2. Change in System Operating Conditions Compared to R-502 for the Constant Condensing Temperature of 54.4°C (130.0°F)

Refrigerant	Discharge Temperature	Condensing Pressure (%)	Evaporating Pressure (%)	Compressor Press. Ratio (%)
HP62 (R-404A)	-5.2°C to -6.0°C	+12.0	+7.4 to +4.5	+3.2 to +5.3
Klea 61 (R-407B)	-1.3°C to -2.1°C	+15.7	+5.9 to +0.2	+12.1 to +16.3
AZ-50 (R-507)	-4.8°C to -7.0°C	+14.3	+10.0 to +8.2	+3.4 to +4.9
Klea 60 (R-407A)	+7.4°C to +8.5°C	+4.7	-11.6 to -14.0	+18.5 to +22.7

CONCLUSIONS

HP62, and AZ-50 had similar evaporator capacities compared to R-502. The evaporating temperature has a stronger effect on the relative capacities of Klea 61 and Klea 60. Additional liquid subcooling increased the relative capacities of HP62, AZ-50 and Klea 61, but did not increase the relative capacity of Klea 60 when compared to R-502. Finally, a higher condensing temperature had a negative effect on relative capacities of all the blends tested.

HP62, Klea 61, and AZ-50 were all less energy efficient than R-502, but the relative performance of these blends improves as the evaporating temperature increases. There was a definite improvement in the relative performance of these blends as the amount of liquid subcooling was increased, and also when the condensing temperature was decreased. Klea 60 was the only blend tested that had a higher energy efficiency than R-502 for all the conditions tested. Klea 60 was also the only blend that showed a reduction in the COP compared with R-502 when there was additional liquid subcooling.

HP62 and AZ-50 had about a 6.0°C (10.0°F) lower discharge temperature than R-502. Klea 61 had approximately a 1.0°C (1.8°F) lower discharge temperature than R-502. Klea 60 was the only blend to show an increase in discharge temperature relative to R-502 with an approximate 8.0°C (14.4°F) increase.

HP62, Klea 61, and AZ-50 all showed an increase in condensing pressure in some cases as high as 15%. Klea 60 showed a much lower increase in condensing pressure that was about 5% higher than R-502. HP62, Klea 61, and AZ-50 also showed an increase in evaporating pressure in some cases as high as 10%. Klea 60 was the only blend that showed a decrease in evaporating pressure relative to R-502.

HP62 and AZ-50 showed a similar increase in compressor pressure ratio ranging from 3.2% to 5.3%. Klea 61 had an increase in compressor pressure ratio that ranged from 12.1% to 16.3%. Finally, Klea 60 had the largest increase in compressor pressure ratio ranging from 18.5% to 22.7% higher than R-502.

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