Vapour compression refrigerator performance with R-12/R-134a refrigerant mixtures

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A series of laboratory tests was carried out, using a small reciprocating-compressor refrigeration system, to measure the system performance characteristics over a range of R-12/R-134a refrigerant mixture compositions as well as using the pure components. The test facility was carefully instrumented and data for various operating conditions were recorded with a computer data acquisition system. The tests covered a range of evaporating temperatures from 0 to 10° C, for three different condenser water exit temperatures. The test data revealed that it is feasible or even beneficial to perform a "refrigerant metamorphosis" from R-12 to R-134a. A refrigerant metamorphosis is a gradual conversion of the composition of the refrigerant charge through progressive topping-up, as leakages occur, with a refrigerant different from that originally used. It was found experimentally and verified by simulation that increases in the cooling capacity and coefficient of performance (COP) were experienced for the near-azeotropic mixtures of these two refrigerants. Pure R-134a gave a superior COP to pure R-12 for high evaporating temperatures, but mixtures of the two gave even higher COPs. No negative characteristics (e.g. high saturation pressure or high compressor discharge temperature) were observed. Predictions from theoretical simulations deviated in certain respects from the actual measurements and the reasons for these deviations are discussed.

Nomenclature

p_{sat}	saturation pressure (kPa)
pi	compression ratio
P	compressor shaft power (kW)
q_e	specific refrigeration effect (kJ/kg)
$q_{ev,\mathrm{real}}$	"realistic" refrigeration effect per unit volume (kJ/m^3)
T_{cond}	condensing temperature (°C)
T_{evap}	evaporating temperature (°C)
$T_{water,cond}$	condenser water outlet temperature (°C)
$T_{2,\mathrm{real}}$	"realistic" temperature after compression (°C)
$w_{in,\mathrm{real}}$	"realistic" specific work input (kJ/m^3)

Introduction

The rapid phasing out of chlorofluorocarbons (CFCs) during the 1990s in terms of the Montreal Protocol posed severe challenges to the refrigeration industry.[1] One of the biggest concerns has been the continued operation of existing machines once the production and importation of CFCs ceased. A number of alternative courses of action have been followed which included the use of stockpiled CFC refrigerants, the recycling of those refrigerants, retrofitting systems at substantial cost,[2] and performing a "refrigerant metamorphosis". For many systems nearing the end of their service life a retrofit would not have been economically justifiable. A refrigerant metamorphosis, however, appeared to be a viable option to keep systems in operation with a minimum of downtime or additional cost,

The idea of a refrigerant metamorphosis was developed by the Institut für Luft- und Kältetechnik in Dresden, Germany[3] and it amounts to a gradual substitution of the original refrigerant by a suitable replacement. Thus the normal leakage losses of the 'old' refrigerant are made up by simply topping up from time to time with a suitable 'new'product (for example, R-12 losses being replaced using R-134a). The mixed-refrigerant system could continue to operate for some time with the original mineral oil, which is made possible by the mineral oil solubility of the 'old' refrigerant. A minimum of 20 to 30% (by mass) of the 'old' refrigerant would have to be retained to ensure proper oil circulation, after which the oil would have to be replaced with polyol ester oil in the case of R-134a. (The mineral oil may, however, be replaced at any time before that critical lower limit is reached.) Depending on a system's leakage losses, its operating life could be extended by a number of years before the oil would have to be replaced. A system with an annual loss of 20% of its base refrigerant, for example, could continue to operate for more than six years before the mineral oil would need to be replaced.

The question arises what the effects on the performance of a refrigerator would be at various stages of a refrigerant metamorphosis. This paper investigates the effects that the use of various R-12/R-134a mixture compositions, as well as pure R-134a, would have on the performance of a reciprocating-compressor refrigeration system initially operating with R-12. These findings are compared with theoretical cycle simulations, which were carried out using a computer program written for the purpose of evaluating the performance of single-stage vapour compression refrigeration systems. The program incorporates a ther-

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mal model of the refrigeration cycle, in which the thermophysical properties of the refrigerant at pertinent points are determined using the REFPROP[4] software.

The particular investigation described in this paper provides an example of the use of computer simulations, verified by experimental measurements, to predict system performance with refrigerant mixtures.[5] There is a great deal of interest in the potential of refrigerant mixtures for various refrigeration and heat pump applications. Because of the large number of possible mixtures, the use of computer simulations is essential for evaluation purposes.

Experimental programme

Test facility

The test facility is shown diagrammatically in Figure 1.

The refrigerant circuit uses an open-drive two-cylinder reciprocating compressor driven by a 4 kW electric motor. The refrigerant flow can be controlled by any of three expansion devices (the two not in use are valved off), namely an R-12 thermostatic expansion valve (TEV), an R-134a TEV and a hand throttling valve. The heat exchangers have counterflow shell and tube configurations with finned tubes to increase the heat transfer between the water and the refrigerant. Water is pumped through the heat exchangers from a storage tank and the water temperature may be controlled by means of a cooling tower.

Instrumentation

Refrigerant pressures are measured using pressure transducers connected to static pressure tappings located at the inlets and outlets of the heat exchangers. The low- and high-pressure transducers are calibrated to ±0.5 and ±1.25 kPa, respectively. Platinum resistance temperature detectors (RTDs) inserted into thermowells are used to measure the temperature of the refrigerant at strategic points of the refrigerant and water circuits with an accuracy of ±0.5°C.

The refrigerant flow rate is measured with a positive-displacement flow meter mounted in the liquid line at the exit of the condenser. This type of flow meter is insensitive to density and viscosity changes and it has an accuracy of ±0.5% of the flow rate. The water flow rates of the condenser and evaporator are metered with turbine flow meters with an accuracy of ±2% of full scale.

The power input of the compressor is measured with an electronic power transducer that continuously multiplies the corresponding instantaneous current and voltage values of the electric motor.

Data acquisition

All instruments have 4–20 mA output signals that are converted into corresponding voltages. A computer program triggers channels of an analogue-to-digital card at set intervals to read these voltages. The A-D card can sample a maximum of 16 analogue channels, but an additional expander card, linked to the A-D card, was used to increase the number of sampled channels. The computer program processes the digital values into units of temperature, pressure, flow, or power.

Measurements

For the comparative study as many variables as possible were kept constant, so that the effect of altering one parameter, the refrigerant composition, could be investigated. Thus the water inlet temperature to the heat exchangers was maintained at 20°C by controlling the fan of the cooling tower, to cool the water of the storage tank. In this way it was possible to control the inlet water temperature to within ±0.1°C. Tests were performed for evaporating temperatures ranging from 0 to 10°C, increasing in 2.5°C steps. This was repeated for three condenser water exit temperatures ($T_{water,cond} = 35, 40, 45°\text{C}$). The condenser exit water temperature was used as the reference condition, since it was physically not possible to measure the condensing temperature in the two-phase region. This was because the refrigerant temperature could only be measured at the inlet and outlet of the condenser, where the refrigerant is in superheated and subcooled states, respectively. The condensing temperature could in principle have been determined from the saturation pressure by using a program capable of determining the thermophysical properties of refrigerant mixtures (such as REFPROP) but this would have introduced additional uncertainties. The refrigerant superheat and the degree of subcooling ranged between 1 to 4 K and 4 to 8 K, respectively.

All readings were taken under steady state conditions that typically were reached in about 45 minutes. The data acquisition system sampled all input channels every 15 seconds and once steady conditions had been reached data were recorded over a period of 10 to 15 minutes. The REFPROP computer program was used to determine the thermophysical properties from the mean values of the sampled data, after which the performance of the refrigerant could be evaluated.

Theoretical simulations

Computer simulations of the performance of the refrigeration system depicted in Figure 1 were carried out for various evaporating and condensing temperatures, over the full range of R-12/R-134a mixtures from pure R-12 to pure R-134a. The simulations took into account the actual performance characteristics of the reciprocating compressor. Figure 2 shows how a changing mixture composition was predicted to affect the performance of the cycle. The subscript "real" denotes "realistic" values taking the particular compressor characteristics into account. Thus $q_{ev,\text{real}}$ is the specific refrigeration effect in kJ/m^3 taking into account the volumetric displacement rate of the compressor whereas q_e is the specific refrigeration effect in kJ/kg, which does not reflect that the refrigerant mass flow rate does not remain constant with change in mixture composition because of changes in density.

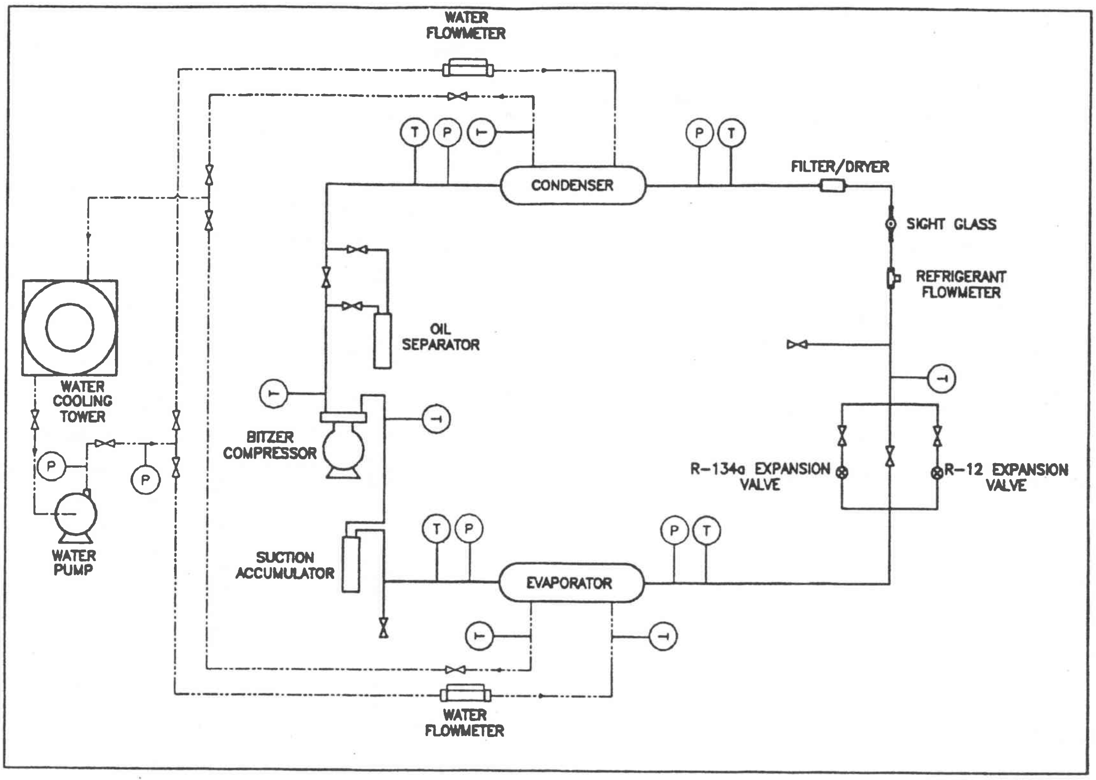

Figure 1 Flow diagram for experimental refrigeration system

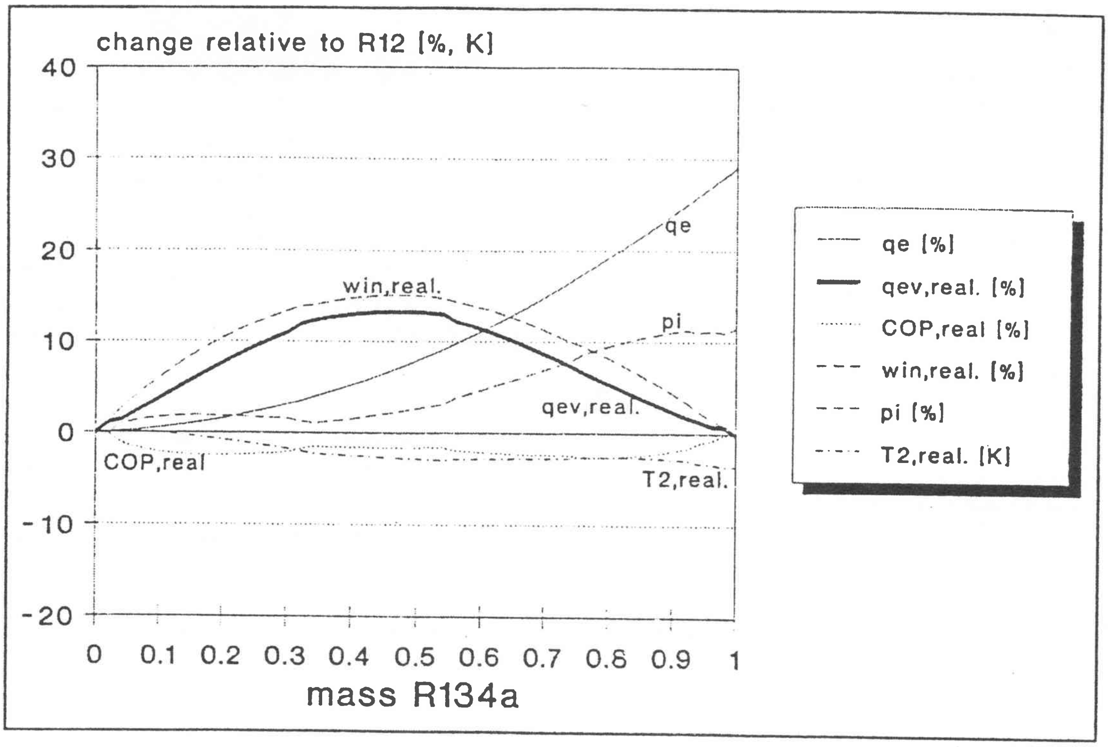

Figure 2 R-12/R-134a metamorphosis simulated at T_{evap} = 0°C and T_{cond} = 40°C

It is evident from Figure 2 that the "realistic" values of both refrigeration effect and work input per m^3 of refrigerant mixture increase and then decrease as the composition changes from R-12 to R-134a, resulting in very little change in the COP. The refrigeration effect q_e per kg, on the other hand, increases progressively with increase in the R-134a fraction. The compression ratio increases after there is more than 50% of R-134a to a maximum about 11% higher than with pure R-12.

Test results and discussion

The major factors of interest in a system undergoing a refrigerant metamorphosis are: the system pressures, the compressor discharge temperature, the cooling capacity, the power requirement, and the coefficient of performance. Both experimental data and theoretical predictions from computer simulations are discussed below for each of these factors.

System pressures

Experimental results:

Over the range of evaporating temperatures tested (0°C $\leq T_{evap} \leq$ 10°C) the saturation pressure of R-134a was somewhat lower than that of R-12. This behaviour is consistent with theoretical predictions and with previous measurements,[6] which found that the saturation pressure of R-134a only becomes higher for temperatures above 15°C. The saturation pressures of the mixtures were considerably higher than those of the pure components. A 50/50 mixture of R-12/R-134a experienced the greatest pressure rise, the saturation pressure being approximately 25% higher than that of pure R-12, for equivalent evaporating temperatures (Figure 3). At this composition the mixture behaves as a near-azeotrope, i.e. it almost forms a minimum boiling temperature azeotrope (maximum saturation pressure). An azeotrope is a mixture that behaves as a single fluid, at a particular combination of the constituents;[7] during change of phase the composition of the liquid phase is thus the same as that of the vapour phase.

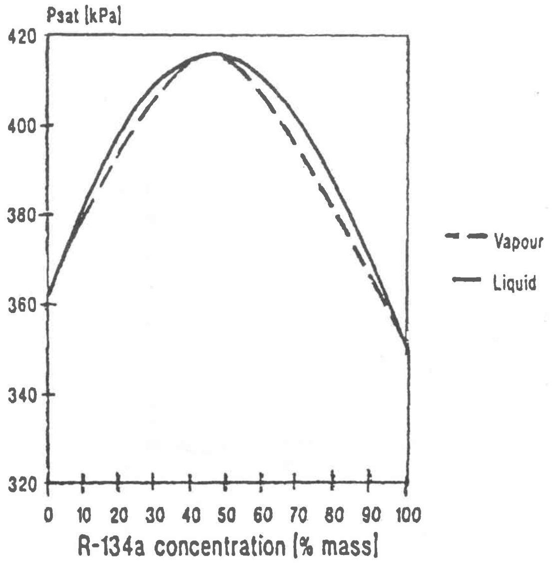

Figure 3 Saturation pressures for R-12/R-134a mixtures at a temperature of 5°C

The addition of R-134a to R-12 had a similar effect on the saturation pressures in the condenser. The pressure of a 50/50 mixture increased by 27.6% ($T_{water,cond}$ = 35°C) while that of pure R-134a was 7.3% higher. Under normal operating conditions, these pressure increases should not result in additional leakage losses or cause the piping to rupture, since these higher pressures are still well below the maximum permissible pressure. For systems operating with high condensing temperatures ($T_{cond} \geq$ 65°C) the higher pressures could become problematic.

The volumetric efficiency of a compressor is primarily a function of the compression ratio,[8] varying inversely with the compression ratio. The compression ratios of the mixtures were somewhat higher than for R-12 and they increased as the R-134a content increased. Pure R-134a was affected most and its measured compression ratio was 14.8% higher than that of R-12, as indicated in Table 1 (T_{evap} = 0°C, $T_{water,cond}$ = 35°C). These increases should have a negligible effect on the volumetric efficiency of the compressor.

Table 1 Comparison of simulated and measured compression ratio changes relative to R-12

Refrigerant (R-12/R-134a)	Measured		Simulated	
	T_{evap} = 10°C	T_{evap} = 0°C	T_{evap} = 10°C	T_{evap} = 0°C
75% / 25%	+3.7%	+3.3%	+1.7%	+1.7%
50% / 50%	+3.7%	+4.4%	+1.9%	+2.5%
25% / 75%	+8.2%	+8.0%	+5.8%	+7.9%
100% R-134a	+11.6%	+14.8%	+7.9%	+11.7%

Simulated results:

The compression ratios from the simulations were very similar to those based on the measured data (Table 1). For mixtures containing less than 50% R-134a, the compression ratio was less than 2.5% higher than that of R-12, while for higher R-134a concentrations the compression ratio increased gradually until a maximum was reached for pure R-134a (11.7% higher for equivalent operating conditions).

Compressor discharge temperature

Experimental results:

It is undesirable to operate a system at high compressor discharge temperatures, since the rate of acid formation is increased and carbonization of the oil in the head of the compressor is induced. This breakdown of the oil can cause excessive wear and a reduced compressor life. The volumetric efficiency of the compressor is also reduced at increased temperatures, since more superheating of the suction vapour occurs by heat conducted from the hot cylinder walls, causing the vapour to expand after entering the cylinder, resulting in a smaller mass of vapour filling the cylinder.

The discharge temperatures were all lower than those for pure R-12 for equivalent operating conditions, thus ensuring oil stability. As the R-134a content in the mixtures increased, so the discharge temperatures decreased, with pure R-134a experiencing the lowest temperatures as indicated in Table 2 (6.7°C lower than R-12 for $T_{evap} = 0°C$ and $T_{water,cond} = 35°C$). This slight reduction in temperature would be beneficial for the compressor efficiency and would counter the losses induced by the higher compression ratio.

Simulated results:

The predicted trends were very similar to the experimental data (Table 2), since the discharge temperature decreased for higher R-134a concentrations and pure R-134a had the lowest discharge temperature (3.5°C lower than R-12 for the conditions listed above).

Table 2 Comparison of simulated and measured compressor discharge temperature changes relative to R-12

Refrigerant (R-12/R-134a)	Measured		Simulated	
	$T_{evap} = 10°C$	$T_{evap} = 0°C$	$T_{evap} = 10°C$	$T_{evap} = 0°C$
75% / 25%	–0.3°C	–2.5°C	–0.6°C	–1.0°C
50% / 50%	–1.5°C	–4.5°C	–1.9°C	–2.9°C
25% / 75%	–2.8°C	–5.6°C	–1.7°C	–2.7°C
100% R-134a	–3.3°C	–6.7°C	–2.5°C	–3.5°C

Cooling capacity

Experimental results:

The cooling capacity, in kW of refrigeration, was measured on both the refrigerant side and the water side of the evaporator. The two cooling capacities should be equal in a perfectly insulated system, since the energy absorbed by the refrigerant in the evaporator should equal the energy lost by the water. In practice the two measuring techniques differed, especially for the mixtures, therefore only the cooling capacities obtained from the water side are considered in this paper. The major source of error for the cooling capacities on the refrigerant side seemed to stem from the computer program (REFPROP) used to evaluate the thermophysical properties of the mixtures. It was found that the version used of this program (version 3.03) modelled the pure components more accurately than the mixtures, which is the main reason for the results of the two techniques correlating better for the pure components.

For evaporating temperatures between 0 and 2°C the cooling capacity of R-12 was approximately equal to that of R-134a but as the evaporating temperature rose, the refrigerating effect of R-134a became higher than that of pure R-12. This improved cooling capacity of R-134a at relatively high evaporating temperatures was also observed in tests performed on large refrigeration systems by Corr *et al.*[9] as well as by a refrigerant supplier.[6]

Table 3 Change in cooling capacity relative to R-12 (constant condenser water exit temperature)

Refrigerant (R-12/R-134a)	$T_{water,cond} = 35°C$		$T_{water,cond} = 40°C$	
	$T_{evap} = 10°C$	$T_{evap} = 0°C$	$T_{evap} = 10°C$	$T_{evap} = 0°C$
75% / 25%	+15.6%	+21.2%	+17.9%	+21.5%
50% / 50%	+21.8%	+25.2%	+21.0%	+27.8%
25% / 75%	+22.6%	+20.5%	+25.4%	+21.5%
100% R-134a	+5.1%	–2.4%	+7.1%	–1.1%

As shown in Table 3, the R-12/R-134a mixtures all showed significant increases in the cooling capacity (per m^3) over the entire range of evaporating temperatures that were tested, for condenser water exit temperatures of both 35 and 40°C. These gains are, however, somewhat lower than those observed in tests conducted on a heat pump.[10]

Simulated results:

The simulations predicted an increase in the cooling capacities of the mixtures for evaporating temperatures above 0°C. At lower evaporating temperatures, mixtures with high R-134a concentrations, as well as pure R-134a, showed inferior cooling capacities. The simulated values showed lower gains than did the experimental data. For conditions equivalent to $T_{evap} = 0°C$ and $T_{water,cond} = 35°C$, the predicted increases were: +8.8% for a 75/25 mixture, +13.9% for a 50/50 mixture, +7.6% for a 25/75 mixture, and −0.3% for pure R-134a.

Power requirements

Experimental results:

The compressor power was measured directly by monitoring the electrical power consumption of the motor driving the compressor and it was also determined from the thermodynamic analysis of the refrigeration cycle. Since the latter method relied strongly on the accuracy of REFPROP, only the motor measurements were used in deriving the results considered here. The power recorded by the wattmeter was not a true indication of the shaft power, since it included the losses of the electric motor and the energy lost by driving the compressor with a V-belt. The readings were therefore corrected so that the actual shaft power was obtained.

The power requirements with the pure refrigerants differed significantly from those with the mixtures. The power curves, when using either of the two pure components only, were similar in shape and magnitude to each other; likewise, those of the various mixtures were similar to each other. The shapes of the power curves indicated that the pure components operated to the left of the power curve peak and the mixtures to the right, over the range of operating conditions tested. Peak power was therefore experienced at lower evaporating temperatures for mixtures than for the pure components. The changes in the shaft power are illustrated in Figure 4, for a constant condenser water exit temperature of 35° C and for various mixture compositions and evaporating temperatures.

It may be seen in Figure 4 that at low evaporating temperatures the power requirement first increased as the mixture changed to 50% of each component, then fell again. At higher evaporating temperatures on the other hand, the power first fell and then rose again as the proportion of R-12 decreased. The power requirement is a complex relationship that depends on both the evaporating and condensing temperatures. This effect on the power requirement of changing the mixture composition was, however, also observed at other, higher, condensing temperatures.

Most compressors are driven by electric motors. A reduction in compressor power would not pose a problem for an electric motor but a motor may become undersized if the power demand becomes too large. Generally motors tend to be slightly oversized, since they are sized for peak power requirements and not necessarily for design conditions. Thus a motor could usually be expected to cope with the additional power requirements induced by the mixtures, during normal operating conditions.

Simulated results:

The simulations predicted an increase in the shaft power for all mixture compositions, for an evaporating temperature above 0° C. For the lower evaporating temperatures the power requirement when using mixtures rich in R-134a, as well as pure R-134a, became less than that when using R-12. Maximum power gains did not exceed 16% and decreased as the evaporating temperature was reduced.

Coefficient of Performance

Experimental results:

Over the range of evaporating and condensing temperatures that were tested the mixtures were associated with substantially higher COPs than were the pure components, but the gain decreased as the evaporating temperature became less. The superior COPs of the mixtures over this range were due to the combination of their higher volumetric cooling capacities and their reduced power requirements (they were operating to the right of the peak of the power curve, whilst the pure components were near their peak). It is most likely that at the lower evaporating temperatures the pure components could have somewhat higher COPs, since the mixtures would then be at their peak power requirement, but the pure components would have moved away from it. Therefore it would be incorrect to predict that the COPs of R-12/R-134a mixtures are always superior to those of the pure components. The changes in the coefficient of performance are illustrated in Figure 5, for a constant condenser water exit temperature of 35° C and various mixture compositions and evaporating temperatures as before. It may be seen that the mixture composition does not greatly affect the values of the COP, but at evaporating temperatures above 0° C the COP rises and then falls back to nearly the original value as the mixture changes from all R-12 to all R-134a. The same also applied at higher condensing temperatures.

Simulated results

A small reduction in the COP was predicted by carrying out the simulations. For evaporating temperatures above 0° C, the COP of the mixtures was approximately 3% less than that of R-12. For lower evaporating temperatures, it was predicted that losses of up to 12% ($T_{evap} = -30°C$) could occur for certain mixture concentrations.

Conclusions

The effect of a refrigerant metamorphosis using R-12 and R-134a was experimentally investigated for various operating conditions. Carefully measured data were recorded for three mixture compositions as well as for the pure components, from which performance comparisons could be made. It was found that, for systems with reciprocating compressors and water-cooled heat exchangers, a metamorphosis could safely extend an existing refrigeration system's operating life without having to perform expensive modifications. Higher system pressures were measured for the mixtures, but for condensing temperatures below 65° C the elevated pressures were still well within the pressure ratings of the system components. Oil breakdown due to high compressor discharge temperatures would not occur, since the temperatures decreased with the addition of R-134a. Based on the test data it was found that a metamorphosis could be economically attractive because the mixtures experienced higher volumetric cooling capacities and higher COPs than the pure components. Theoretical simulations tended to be somewhat conservative, since they predicted lower cooling capacity gains for mixtures when compared with pure R-12 (the predicted percentage improvements were about half of those measured) and smaller increases in the COPs.

As a general conclusion, it was found that computer simulations could give reasonable-to-good predictions of system performance when operating with refrigerant mixtures. It could be expected that later versions of refrigerant property software such as REFPROP would give more accurate predictions of the properties of mixtures. Although not reported in this paper, a similar series of simulations was carried out for R-11 and R-123 mixtures[5] and this indicated that a metamorphosis in that case would lead to losses in cooling capacity (which could require the

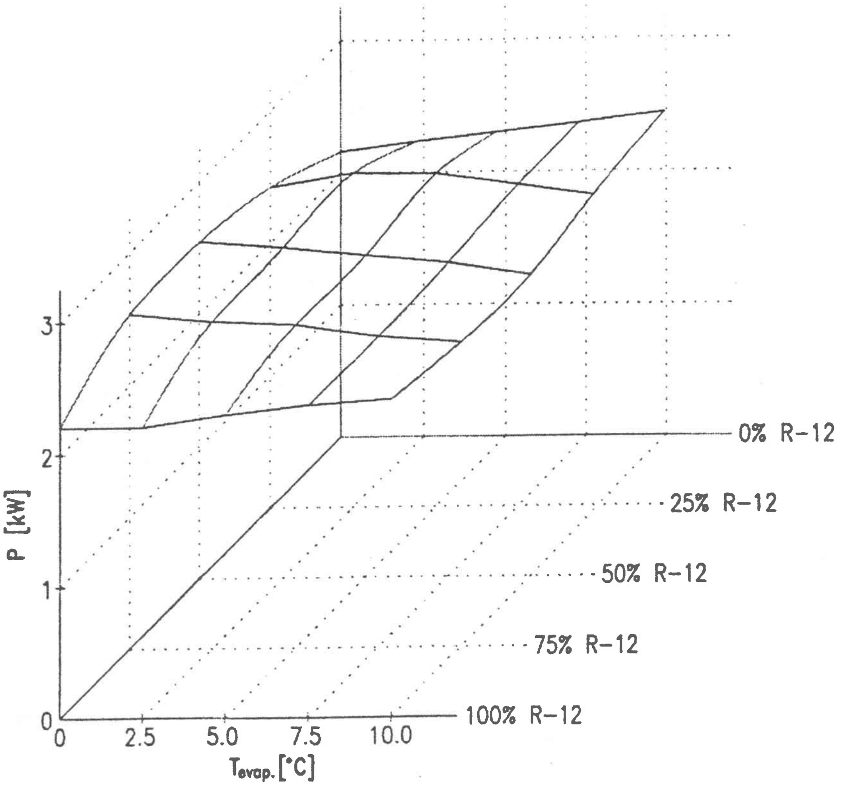

Figure 4 Measured compressor shaft power variation for condenser water exit temperature of 35°C

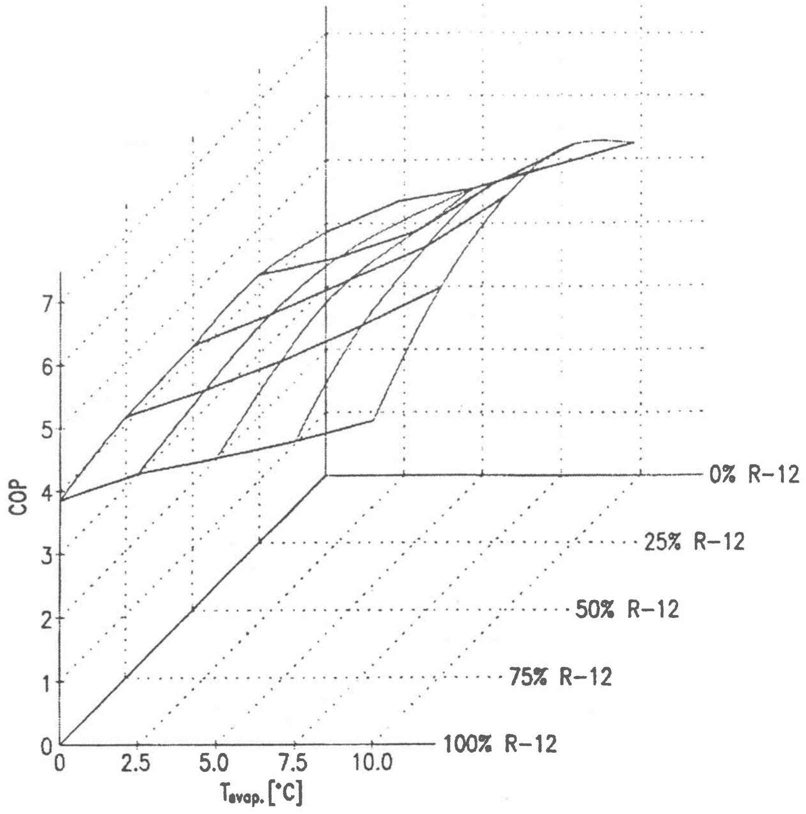

Figure 5 Variation in measured coefficient of performance for condenser water exit temperature of 35°C

adoption of measures such as increasing the compressor speed). This and other work has demonstrated both the usefulness of such simulations and the necessity of being able to verify the theoretical predictions by means of careful measurements.

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