

Experimental Evaluation of Two Refrigerant Mixtures in a Breadboard Air Conditioner

by

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Abstract

An experimental, water-to-water, breadboard heat pump (that is one designed to be easily reconfigured) was constructed for comparison of pure R22 to the refrigerant mixtures R22/R114 and R13/R12. Three evaporator configurations were extensively studied. In all cases the best mixture outperformed R22. The best efficiency with R22/R114 was 32% higher and with R13/R12 was 16% higher than the best efficiency measured with R22. Other observations were, first, that mixtures can take advantage of heat exchanger efficiency that, in a gliding temperature application, a pure refrigerant is incapable of utilizing. Secondly, that heat exchange between the condensed and evaporating refrigerant is beneficial to some mixed refrigerants. Finally, mixtures exhibit nonlinearity of enthalpy versus temperature in the two phase region which has significant impact on both heat exchanger and cycle design.

Introduction

The National Bureau of Standards has evaluated two nonazeotropic refrigerant mixtures, R22/R114 and R13/R12, in a water-to-water breadboard heat pumping apparatus at temperatures typical of air conditioning applications. As is shown in figure 1, the best performing test series with the mixture R22/R114 resulted in a coefficient of performance of 6.9 (EER = 23.6), 32% higher than the best efficiency obtained with pure R22. In figure 2, the best efficiency with the mixture R13/R12 was 16% better than R22.

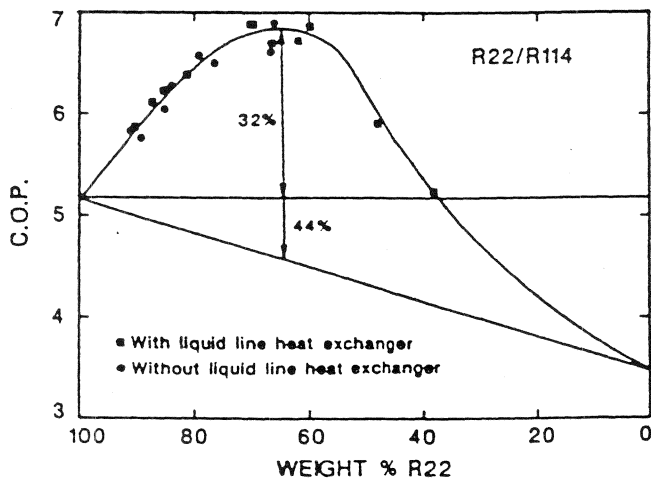


Figure 1: Performance of the mixture R22/R114 for evaporator configuration No. 4.

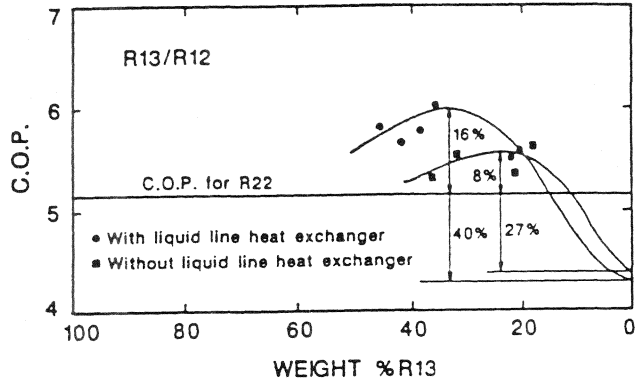


Figure 2: Performance of the mixture R13/R12 for evaporator configuration No. 4.

Test Conditions

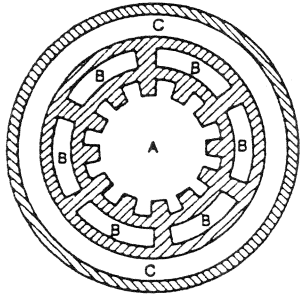
The two primary criteria used to define comparable conditions for evaluating the tested refrigerants were fixed water temperatures and constant capacity. The fixed water temperatures chosen as typical of air conditioning were 80°F (26.7°C) inlet and 55°F (12.8°C) outlet for the evaporator, and 82°F (27.8°C) inlet and 117°F (47.2°C) outlet for the condenser. Because water temperatures were fixed, refrigerant temperatures and pressures were dependent on refrigerant properties. Since the evaporator area was to be kept constant for each test series, the constant heat flux criteria required constant capacity. For this reason, as refrigerant properties changed it was necessary to change compressor speed.

The test variables other than refrigerant compositions were the evaporator overall heat transfer coefficient times area (UA) and the use of intracycle heat exchange. The evaporator heat exchanger had three passages as shown in figure 3. By routing water or evaporating refrigerant through passages of different cross section or surface area, different pressure drops and water-to-refrigerant heat exchange coefficients were obtained. Three configurations were examined for a full range of refrigerant mixture compositions. The third evaporator passage was available for heat exchange between the condensed liquid refrigerant and the evaporating refrigerant in counter flow throughout the length of the evaporator. The tested evaporator configurations are summarized in table 1.

Configuration

	1	2	3	4
Evaporating Refrigerant	C	B	A	A
Liquid Refrigerant	B	C	B	C
Water	A	A	C	B

Table 1: Tested evaporator configurations (letters refer to Figure 3).



	Cross Sectional Area	Perimeter
A	227 mm ²	100 mm
B	122 mm ²	120 mm
C	310 mm ²	80 mm

Figure 3: Evaporator cross section (not to scale).

Apparatus

A schematic diagram of the breadboard heat pump is shown in figure 4. The open compressor is belt driven by a variable speed motor. A shaft dynamometer (strain gage torquemeter plus magnetic pickup tachometer) mounted between the compressor pulley and flywheel was used as the primary cycle power input measurement. From the compressor the refrigerant passes through an oil separator and then enters the condenser which rejects its heat to a counterflow water loop. Condenser capacity was measured by a water side measurement and by comparison of the water temperature rise to that over the metered electric trimming heater. The manual expansion valve was adjusted to maintain slight subcooling as observed at the condenser exit sightglass.

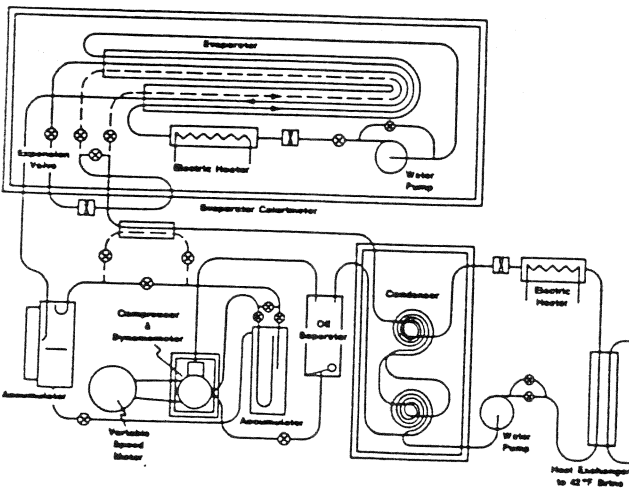


Figure 4: Schematic of breadboard heat pump.

The primary measure of evaporator capacity which was used to calculate cycle efficiency was provided by measuring the total heat input (electric heaters, water circulating pump, calorimeter through-the-wall heat gain) to the calorimeter box which surrounded the evaporator and its water loop. Secondary measures of evaporator capacity were a water side calculation and comparison of the water temperature drop through the evaporator to the water temperature rise through the metered heaters used to match the evaporator capacity.

A special accumulator which lacked an oil return hole was used at the evaporator outlet to allow flooded coil operation with only saturated vapor returning to the compressor so that the evaporator area in two-phase refrigerant-side heat transfer would stay constant and the system be charge insensitive. Mixed refrigerant composition was determined by analysis of compressor discharge gas samples using a gas chromatograph.

Figure 5 shows variation of system efficiency with compressor speed for pure R22 while holding suction and discharge pressures constant. Efficiency was felt to be sufficiently constant to allow mixture testing without correction for this variable. Because of vibrations in the compressor mounting and drive systems (resulting in a marked decrease in measured system efficiency) tests were not run near the speed of 750 rpm.

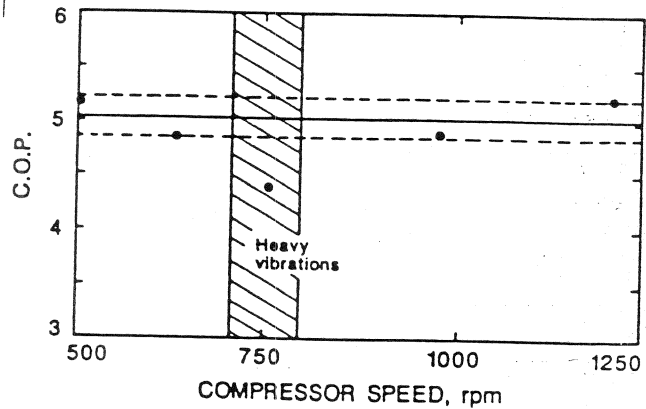
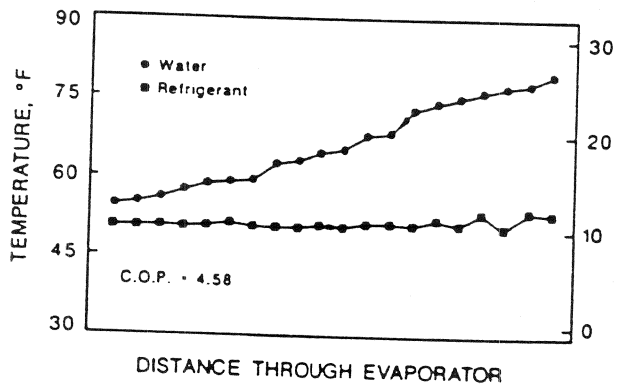


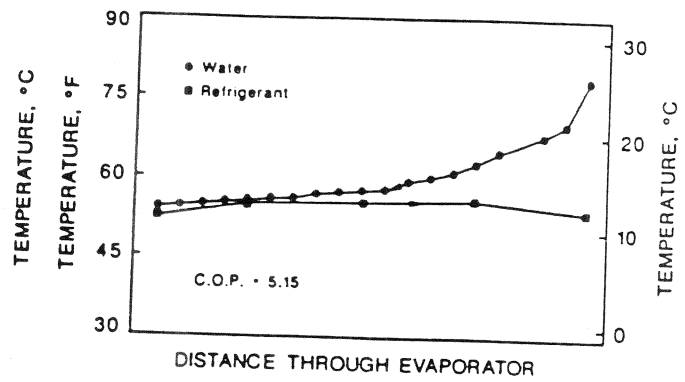
Figure 5: System efficiency as a function of compressor speed for pure R22.

Discussion and Results

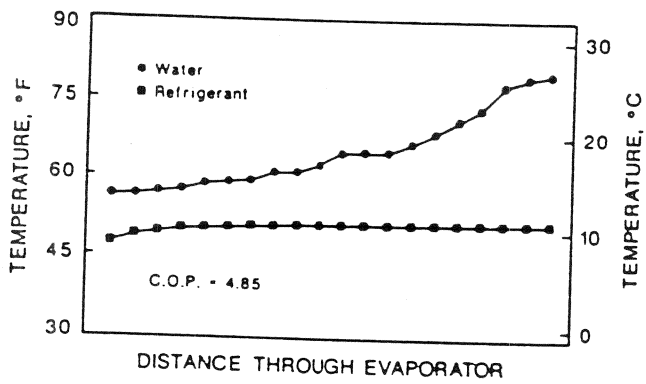
Water and refrigerant temperature profiles through the evaporators for pure R22 are shown in figure 6. The four parts of this figure are for four different evaporator configurations (i.e. flow routed through different passages) which resulted in different overall heat transfer coefficient times area (UA) values. In the best configuration (figure 6d-configuration 4) the water temperature has reached that of the refrigerant in a quarter of the length of the evaporator. The remainder of the evaporator is



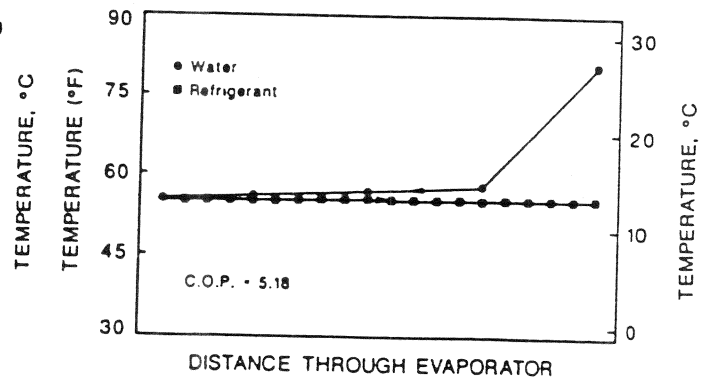
a) Configuration 1



b) Configuration 2

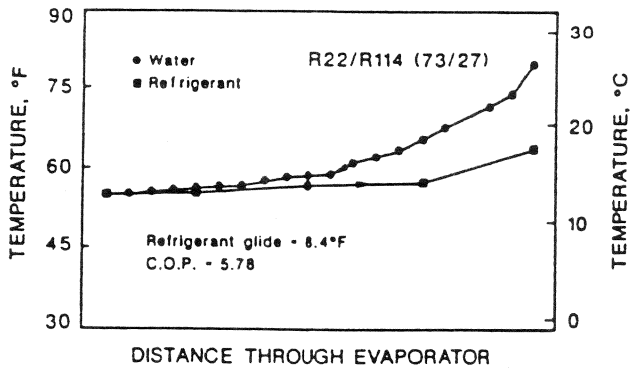


c) Configuration 3

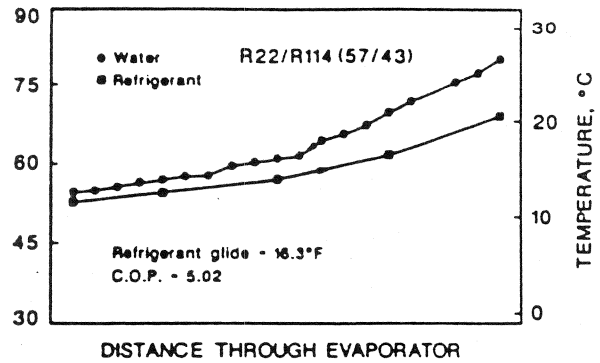


d) Configuration 4

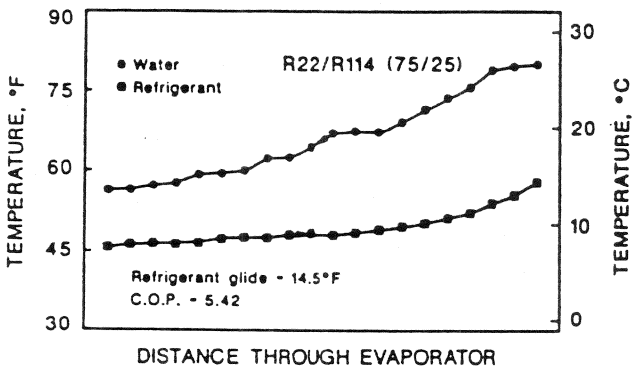
Figure 6: Water and refrigerant evaporator temperature profiles using pure R22.



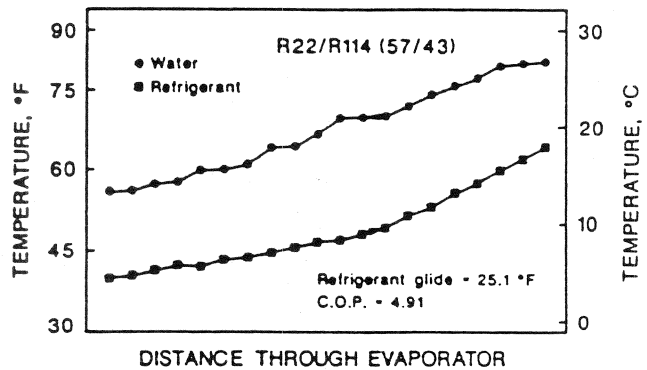
a) Configuration 2, best COP.



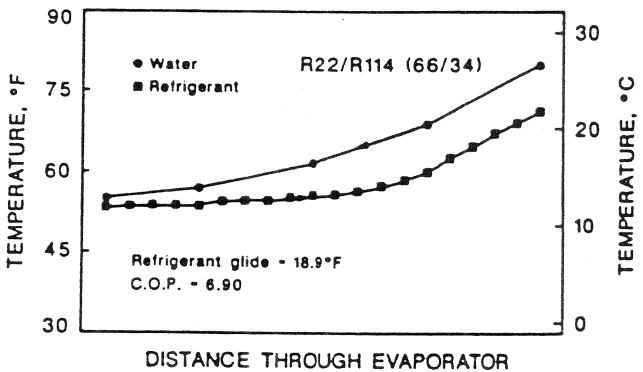
b) Configuration 2, best glide.



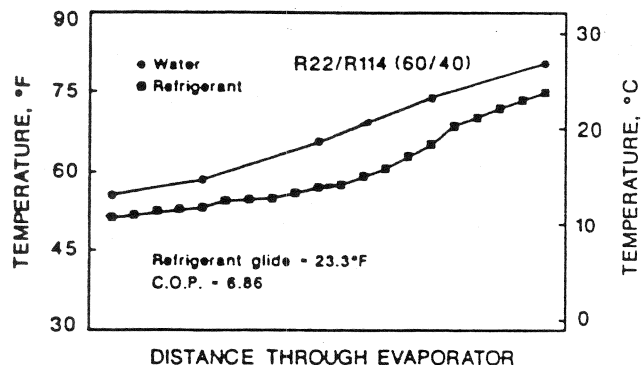
c) Configuration 3, best COP.



d) Configuration 3, best glide.



e) Configuration 4, best COP.



f) Configuration 4, best glide.

Figure 7: Evaporator temperature profiles for mixtures of R22/R114 which resulted in the highest COP and in a refrigerant glide most nearly matching that of the water (25°F, 3.9°C).

ineffective. This "pinch point" resulting from a mismatch of the refrigerant and water temperature glides limits the efficiency attainable with a pure refrigerant to that resulting from its thermodynamic properties and the LMTD (log mean temperature difference) when pinching first occurs.

As can be seen in figure 7 (for the mixture R22/R114) properly chosen mixtures do not have a pinch point. Unlike a pure refrigerant, their LMTD continuously decreases and cycle efficiency continuously increases with increasing heat exchanger size. It should be noted that mixtures always have a thermodynamic advantage over pure refrigerants because of Carnot versus Lorenz cycle considerations and this advantage is greatest when temperature lifts are low and pinch points approached. Whether this theoretical advantage results in better efficiency relative to a given pure refrigerant depends on the respective performance of the refrigerants in question.

Figures 7c and d are temperature profiles for a heat exchanger configuration (configuration 3 is shown, configuration 1 would be similar) in which pure R22 has not reached a pinch point. The best efficiency (Figure 7c) occurs with a mixture containing considerably more R22 (75% R22) than that resulting in the temperature glide most nearly matching the water temperature glide (Figure 7d, 57% R22). In figure 1 it could be seen that pure R22 resulted in a much higher cycle efficiency in the test apparatus than did pure R114. Hence, in this case where the mixture and the pure refrigerant can operate at comparable LMTD's, best performance occurs in a mixture rich in the better performing component. However, as can be seen in figure 8, the mixture still results in improved cycle efficiency compared to pure R22.

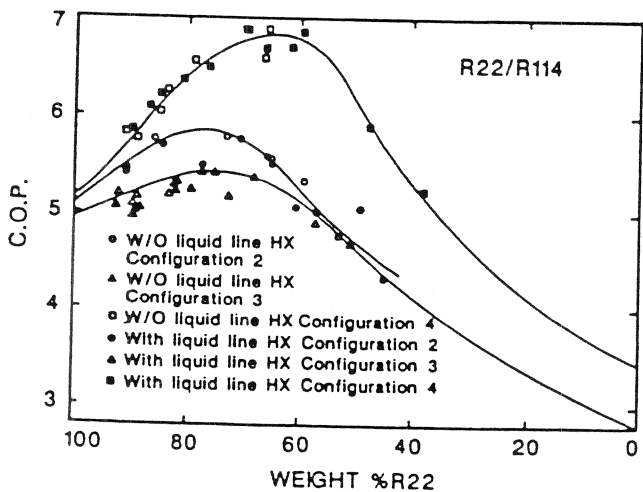


Figure 8: Performance of the mixture R22/R114 for evaporator configurations 2, 3 and 4.

Figure 7a and b are for a case in which there is a high evaporator UA and a high refrigerant pressure drop (configuration 2).

Because of the tendency for saturation temperature to drop with pressure, much of the glide is lost. Both figure 7b and 7d are for a 57% R22 mixture, but the high pressure drop case has only a 16°F glide where the low pressure drop case has a 25°F glide. A second liability for mixtures in the high pressure drop configuration results from the constant heat flux test criterion. Refrigerant capacity is reduced with increased R114 concentrations which was compensated for by increased compressor speed resulting in higher gas velocities and, consequently, increased pressure drop. For this case in which pressure drop was high enough to have a significant effect on system performance, this resulted in best efficiency with a higher than expected concentration of R22.

Figure 7e and 7f are for configuration 4 which had low pressure drop and the best overall UA. In this case best efficiency occurs at an R22 concentration which results in a refrigerant temperature glide closer to the water temperature glide than in the other configurations and in the best cycle efficiency measured in this study. Referring to figure 1, a 32% improvement over pure R22 is shown. Simple cycle simulations predict equal efficiency for cycles using pure R22 and pure R114. Greatly reduced efficiency, believed to be caused by the compressor design being unsuited to the operating pressures, was observed with pure R114. If compressor designs were available to suit the various mixture requirements it is then estimated, as shown in figure 1, that a 44% improvement over pure R22 may be possible.

It should again be noted that this configuration which resulted in the greatest improvement over pure R22 employed four times the heat exchanger area that would be reasonable for a pure R22 system (see figure 6d). The pure R22 experiences a "pinch point" which limits its ability to effectively utilize large heat exchangers - a limitation that mixtures with appropriate temperature glides do not experience. If the heat exchanger were still larger, the system, when operating with the mixture, would be expected to have a still better efficiency which would not occur with pure R22.

Figure 9 summarizes the test results for evaporator configuration two, three, and four for the mixture R13/R12. Tests were not run at high compositions of R13 because of high discharge pressures.

An outstanding difference between the system performance for the mixture R13/R12, as shown in figure 9, and that with R22/R114, as shown in figure 8, is the effect of heat exchange between the evaporating refrigerant and subcooled liquid refrigerant leaving the condenser. This heat exchange dramatically improved efficiency for the mixture R13/R12 but has little effect for the mixture R22/R114.

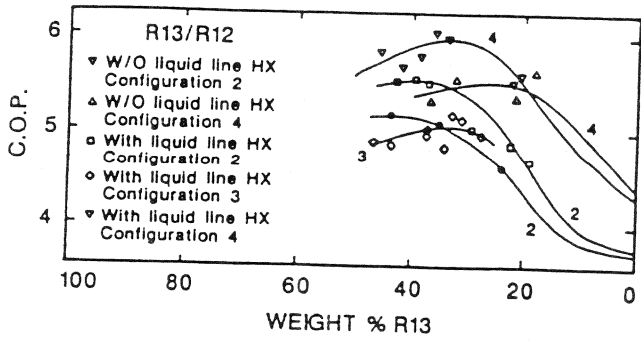


Figure 9: Performance of the mixture R13/R12 for evaporator configuration 2, 3 and 4.

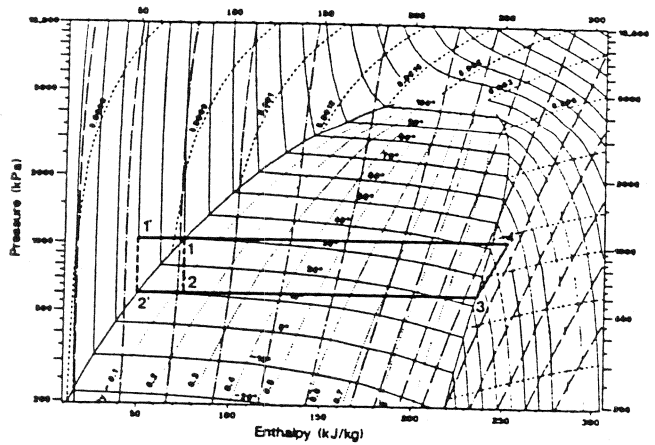


Figure 11: Effect of heat exchange between subcooled liquid and evaporating refrigerant a 60%/40% of R22/R11.

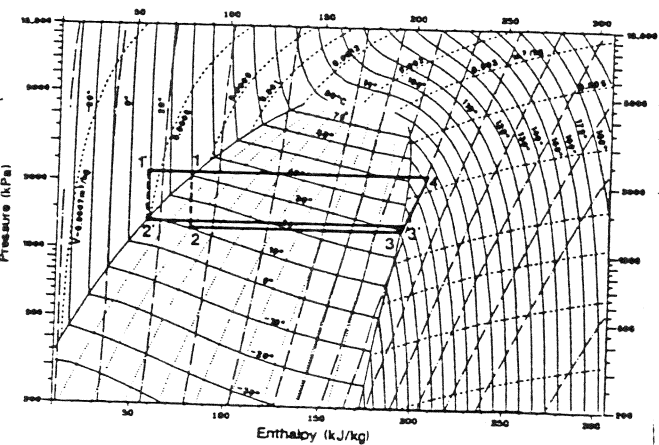


Figure 10: Effect of heat exchange between subcooled liquid and evaporating refrigerant a 40%/60% mixture of R13/R12.

The benefits to be expected from this heat exchange can be seen in figure 10, a pressure enthalpy plot of the cycle with and without this heat exchange for the mixture R13/R12. The intracycle heat exchange results in expansion from point 1' to point 2' instead of flashing from 1 to 2. Because of the slope of the isotherms in the two phase region evaporation from point 2' to point 3' results in the same average temperature as from point 2 to 3 but at a higher pressure resulting in less compressor work. Note that to achieve the requisite subcooling, heat exchange must take place in counterflow throughout the length of the evaporator because of the nearly vertical isotherms in the subcooled region.

Similar cycle plots for the mixture R22/R114 are shown in figure 11. It can be seen that the isotherms in the low quality portion of the two phase region are nearly flat resulting in no significant difference in evaporator pressure for the cycles with and without heat exchange as is also the case with pure refrigerants.

The reason that intracycle heat exchange can benefit one mixture and not the other is that enthalpy is not necessarily a linear function of temperature as is generally supposed. A further demonstration of the importance of this nonlinearity is shown in figure 12, comparing plots of refrigerant and water temperatures through the evaporator. In these figures a

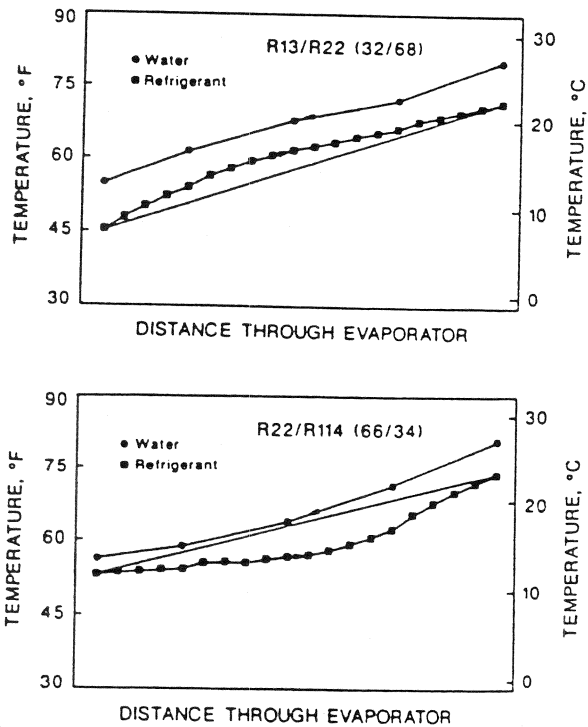


Figure 12: Comparison of typical evaporator temperature profiles for the mixtures R22/R114 and R13/R12.

straight line has been drawn between the entering and leaving refrigerant temperatures to emphasize that one refrigerant has a concave and the other a convex temperature profile. This manifestation of nonlinearity would interfere with cycle performance prediction using simple computer programs in which only inlet and outlet temperatures are specified and linearity is assumed within the heat exchangers. Additionally, the pinch points which result from this nonlinearity reduce cycle efficiency.

Conclusion

A mixture of R22/R114 was found to be 32% better, and a mixture of R13/R12 16% better, in efficiency than R22 in an air conditioning application. Some system design characteristics that make it likely that mixtures will substantially outperform pure refrigerants are counterflow heat exchangers, large heat exchangers, high temperature glide in the heat source/sink fluid, matching of refrigerant and heat source/sink temperature glides and a low temperature lift.

Nonlinearity of enthalpy as a function of temperature in the two phase region was observed to have an effect on system design with respect to intracycle heat exchange and cycle simulation using models.

