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Executive Summary

As part of the Department of Energy/Oak Ridge National Laboratory Building Equipment Research Program, the National Institute of Standards and Technology constructed an experimental, easily reconfigurable, water-to-water, breadboard heat pump apparatus in order to compare pure R22 to nonazeotropic refrigerant mixtures. Performance of the heat pump charged with a range of compositions of the binary mixtures R22/R114 (chlorodifluoromehtane/dichlorotetrafluoroethane) and R13/R12 (chlorotrifluoromethane/dichlorodifluoromethane) were compared to R22. The advantage claimed for mixtures in this application is improved thermodynamic efficiency as a result of gliding refrigerant temperatures in the evaporator and condenser in low lift, high glide applications typical of air conditioning.

All tests, with the exception of one series designed to show the effect of varying condenser glide, were conducted with evaporator entering and leaving water temperatures of 80°F (26.7°C) and 55°F (12.8°C) and condenser entering and leaving water temperatures of 82°F (27.8°C) and 117°F (47.2°C) simulating an air conditioning application. Each test series was conducted with constant evaporator capacity as an additional requirement for comparability. Tests were conducted for three different evaporator configurations which produced substantially different overall thermal conductance values between the evaporating refrigerant and the heat source water. Tests were also conducted with and without intracycle heat exchange between the condensed refrigerant liquid and the evaporating two-phase refrigerant throughout the length of the evaporator, a modification which theoretically should show substantial benefits for mixtures but not for pure refrigerants. All tests were conducted with a flooded evaporator and with saturated vapor entering the open drive compressor.

For the three evaporator heat exchanger configurations for which a full range of compositions were tested, the optimum composition of each mixture performed better than pure R22. As the thermal conductivity at the evaporator was improved, the superiority of the mixtures over R22 increased. The pure refrigerant reached a pinch point where the water temperature was nearly the same as the R22 temperature and no heat transfer could take place in the remainder of the evaporator. The mixtures with a glide matching that of the water could use the whole evaporator more effectively.

The best efficiency measured with the mixture R22/R114 was approximately 32% better than the best efficiency measured with R22. With the R22/R114 mixture, the intracycle heat exchanger had no effect on mixture performance. The best efficiency measured with the mixture R13/R12 was approximately 16% better than R22 using the intracycle heat exchanger and approximately 8% better without.

The ineffectiveness of the intracycle heat exchange with R22/R114 was primarily from the nonlinear relationship between enthalpy and temperature in its two phase region. At low qualities the constant enthalpy lines were nearly isotherms as they would be for a pure refrigerant. Most of its glide occurred at qualities over 20%. This nonlinearity of enthalpy versus temperature also impacts strongly on simple computer models of system performance which normally assume linear temperature profiles.

An additional observation was with the less effective heat exchanger configuration. Glide matching was less in importance and the optimum composition was shifted in the direction of the mixture component which was best because of compressor efficiency, pressure drop, or thermodynamic properties.

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, mixtures can take advantage of heat exchanger efficiency that, in a gliding temperature application, a pure refrigerant is incapable of utilizing. Secondly, 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.

Key words: air conditioning, heat pump, intracycle heat exchange, nonazeotropic refrigerants, refrigerant mixtures, refrigerants, refrigeration

Acknowledgements

There are many individuals who contributed to the success of this project. The authors acknowledge, in particular, Dr. Reinhard Radermacher of the University of Maryland for the fundamental thermodynamic concepts upon which the project was based, Maciej Chwalowski and David Aaron for experimental operation and data acquisition, and Daniel Gaggioli and Donald Ward for construction of the rig.

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Unit Conversion Table

Given British	Multiply by	To Obtain SI
inch	25.4	mm
foot	0.3048	m
cuft	28.317×10^{-3}	ms
psi	6.8948	kpa
°F	$t_c = (t_f - 32)/1.8$	°C
°F	$T_r = t_f + 459.67$	R
°C	$T_k = t_c + 273.15$	K
Btu	1054.4	J
Btu/hr	0.2929	m^3/hr
cuft/min	1.699	m^3/hr
hp	0.7457	kW
in-lb	0.11298	Nm
ton(refr)	3516.8	W

1. Introduction

In recent years the research interest in the use of nonazeotropic binary mixtures as a working fluid for refrigeration systems has increased greatly. Previous work [1] has shown that a simple substitution of a mixture into a refrigeration system designed for single component refrigerants yields only minimal performance increases. Yet, elementary theoretical considerations suggest that considerable improvement should be realized. For this reason, an experimental heat pump rig capable of easy adjustment and redesign was built for comparison of mixed and single component refrigerants. This rig is referred to as a breadboard heat pump because of this emphasis on easy system reconfiguration to allow optimization with different refrigerants.

After checkout tests of the breadboard heat pump with pure R22 had been performed, a series of tests at different mixture compositions for four evaporator configurations were performed with mixtures of R22/R114 (chlorodifluoromethane/dichlorotetrafluoroethane), which is known to improve cycle efficiency [2], and R13/R12 (chlorotrifluoromethane/dichlorodifluoromethane) which may have application in composition shifting cycles for heat pumps.

All mixture tests were performed with counterflow heat exchangers using water as both the evaporator heat source and condenser heat sink. All tests, with the exception of one series designed to show the effect of varying condenser glide, were performed at the same nominal water temperatures: evaporator water inlet and outlet of 80°F (27°C) and 55°F (13°F) and condenser water inlet and outlet of 82°F (28°C) and 117°F (47°C), respectively. This water-to-water application with moderate temperature glides and low temperature lift was chosen as representing building air conditioning and as also likely to benefit from the gliding temperature effect of mixtures.

Besides observation of basic cycle efficiency and capacity, data on the effect of heat exchanger characteristics which best exploit mixtures, the linearity of enthalpy versus temperature, and intracycle heat exchange between condensed and evaporating refrigerant are presented. Tabulated experimental data are presented in the appendix. The material presented in detail in this report has previously been published in summary form [3, 4].

2. Theory

Binary refrigerant mixtures consist of two refrigerants. Depending on the constituent molecules, either azeotropic or nonazeotropic behavior will be obtained. An azeotropic mixture behaves like a pure refrigerant for a particular composition. At a fixed pressure, it boils or condenses at a constant temperature (Figure 1) and the liquid and vapor phases have the same composition. Under similar conditions a nonazeotropic mixture undergoes a temperature glide while boiling or condensing and has differing vapor and liquid phase compositions.

Figure 2 shows this behavior of a nonazeotropic mixture. For the condensation of a mixture at composition x_m the process is as follows: Superheated vapor cools (5) until it reaches the dew point (4). Rather than maintaining the same

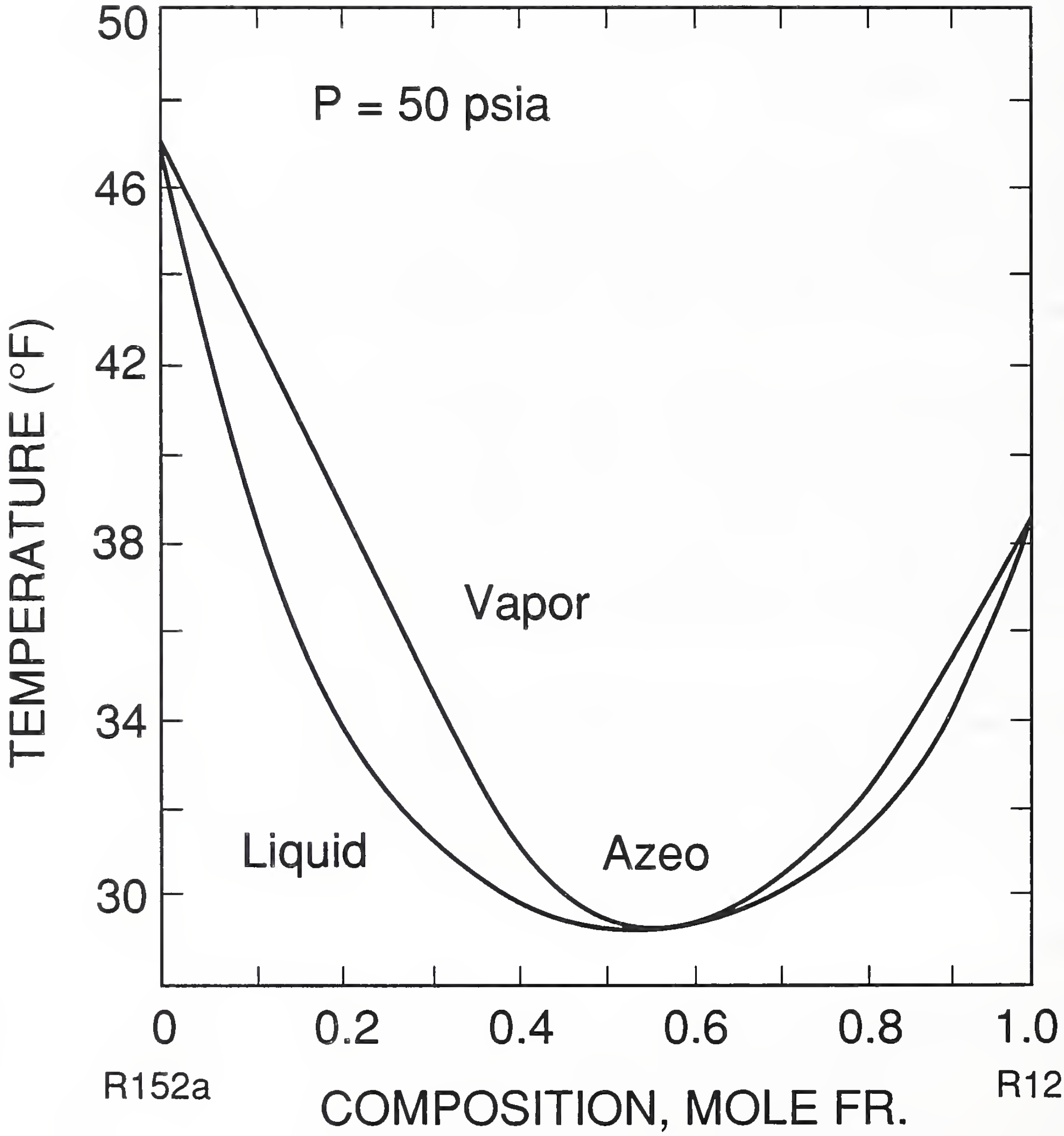


Figure 1. Phase Diagram of Azeotrope R-500

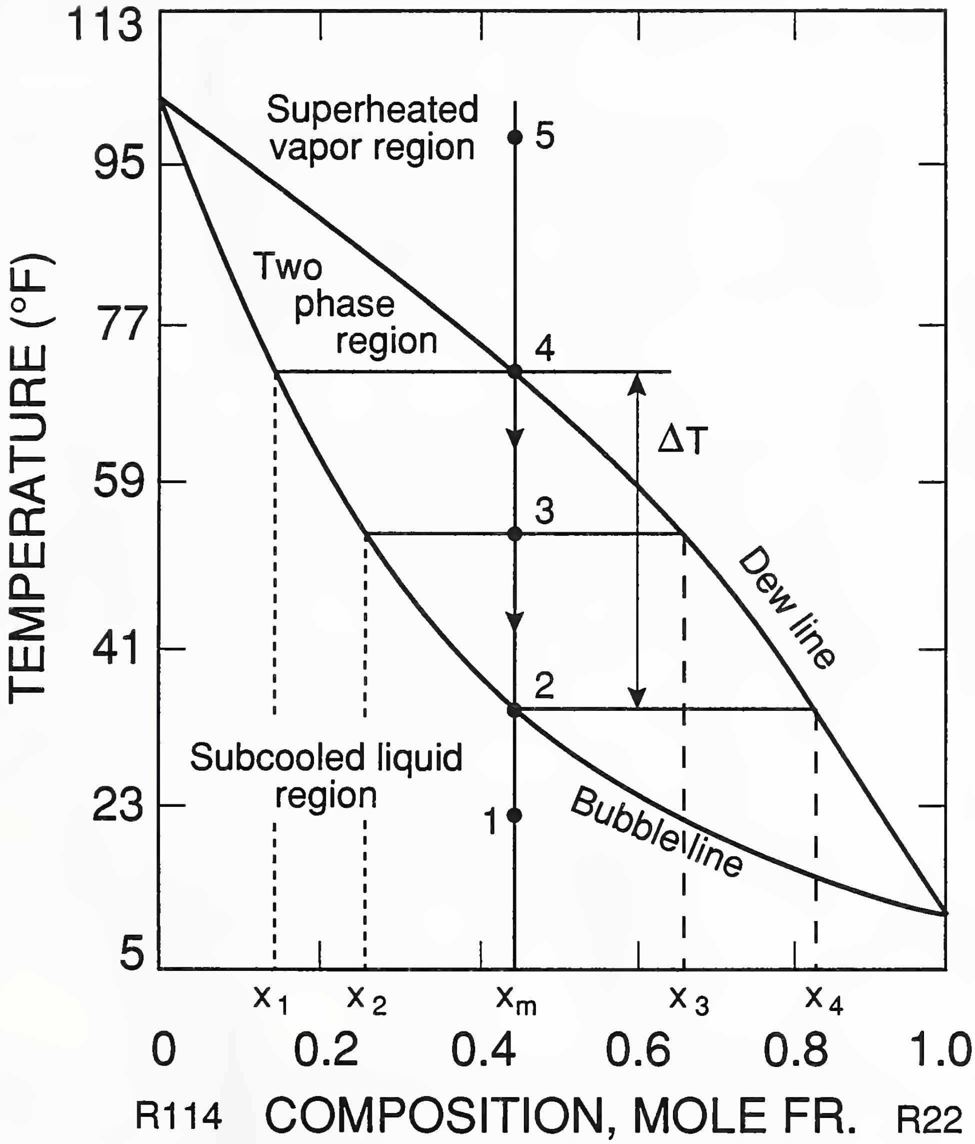


Figure 2. Phase Diagram of Binary Nonazeotrope

composition in the vapor and liquid phase throughout the condensation process, different compositions exist in vapor and liquid phase. The first drop of liquid has a composition enriched in the less volatile component, at composition x_1. The remaining vapor composition now shifts towards the more volatile component, R22. While the temperature drops, the vapor composition follows the dew line, whereas the liquid composition follows the bubble line. i.e., compositions x_2 and x_3 respectively at point three. Finally, the last bit of vapor has a composition enriched in the more volatile component, x_4, while at that point the liquid has a composition of x_m (2). Once the refrigerant mixture leaves the two phase region, the liquid has the composition x_m.

At fixed pressure, the two phase region for nonazeotropic mixtures exists over a temperature range, ΔT in Figure 2, instead of a single temperature as is the case for pure refrigerants or for an azeotrope. This feature allows the mixture, flowing counter current to the heat exchange fluid, to follow the same glide (temperature change) as is present in the heat sink and heat source of heat pumps or air conditioners. This reduces the average temperature difference between the refrigerant and the heat exchange fluid which is a source of thermal irreversibility.

A refrigeration cycle with changing temperatures of the refrigerant in the heat exchangers was first described by Lorenz and, therefore, the heat pump cycles employing nonazeotropic mixtures are said to be Lorenz cycles. This is in contrast to the Carnot cycle with constant refrigerant temperatures in the heat exchangers as found for single component refrigerants. The Carnot principle, stating that the Carnot cycle efficiency can not be exceeded, is limited to applications of constant temperature heat sinks and heat sources. An improvement in system efficiency for a heat pump or air conditioner can be obtained by departing from the ideal Carnot cycle and employing the Lorenz cycle with changing temperatures of the refrigerant. The reason for this improvement can be found in the nature of the heat source and heat sink. Since the heat exchange fluids are not changing phase while traveling through the heat exchanger, they have to change temperature in order to exchange heat. This means that a temperature glide will always exist at the heat source/heat sink side. In an ideal Carnot cycle the temperature of the refrigerant remains the same throughout the heat exchangers. As can be seen on the T,s-diagram (Figure 3) this mismatch in temperatures requires additional unnecessary work for the Carnot cycle as compared to the Lorenz Cycle (shaded areas in Figure 3).

It has to be noted that the improvement in system performance is influenced a great deal by the temperature differences obtained in the heat exchangers as stated by McLinden and Radermacher [5]. If the average temperature difference between the refrigerant and the heat source/sink fluid in each heat exchanger for the Lorenz cycle is larger than the Carnot temperature difference would have been for the same application (which is possible primarily because typically there is a significant decrease in the mixture evaporative heat transfer coefficient relative to either of its components) then the Lorenz cycle will not be advantageous. Special care has to be taken, therefore, in designing the heat exchangers for mixtures.

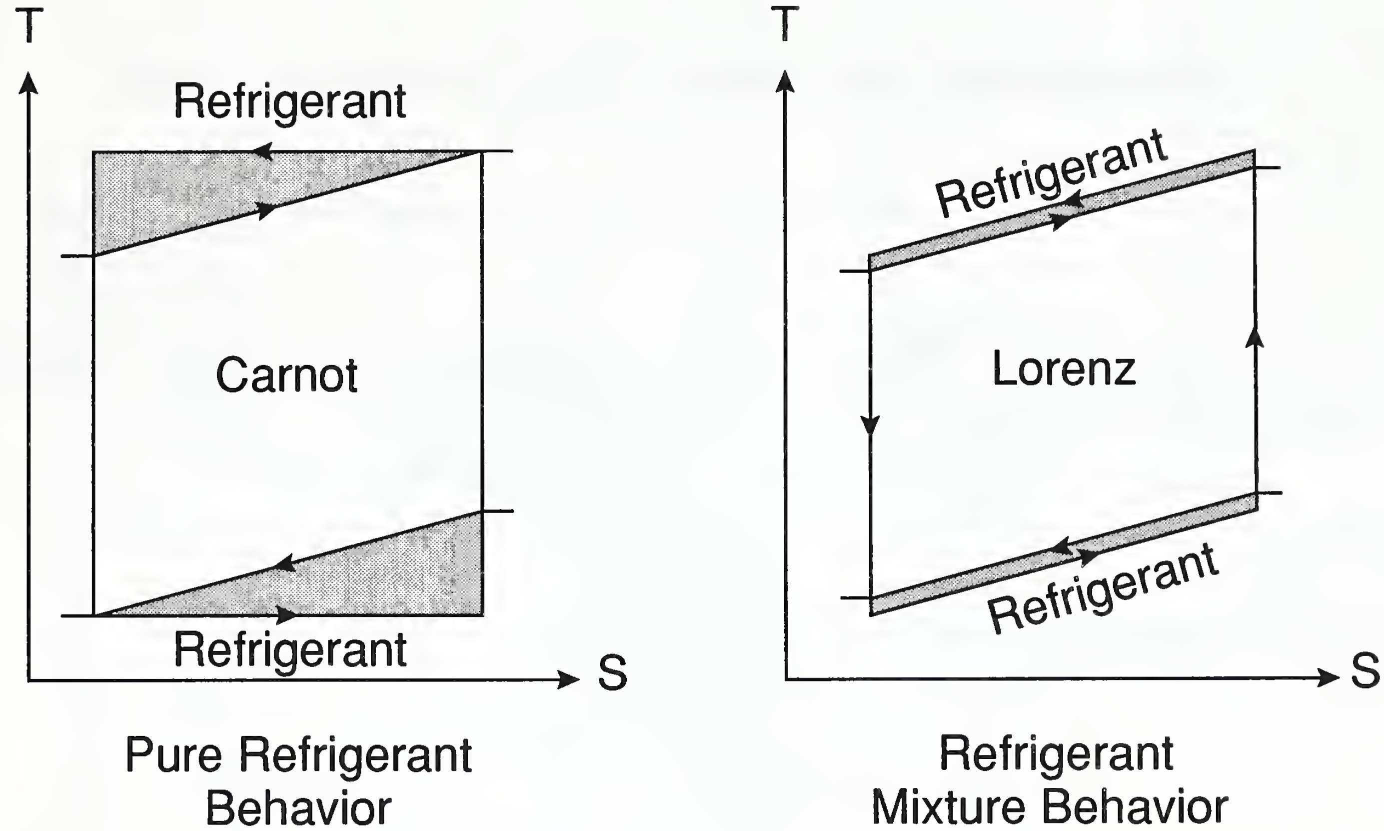


Figure 3. Efficiency Advantage of Lorenz Cycle

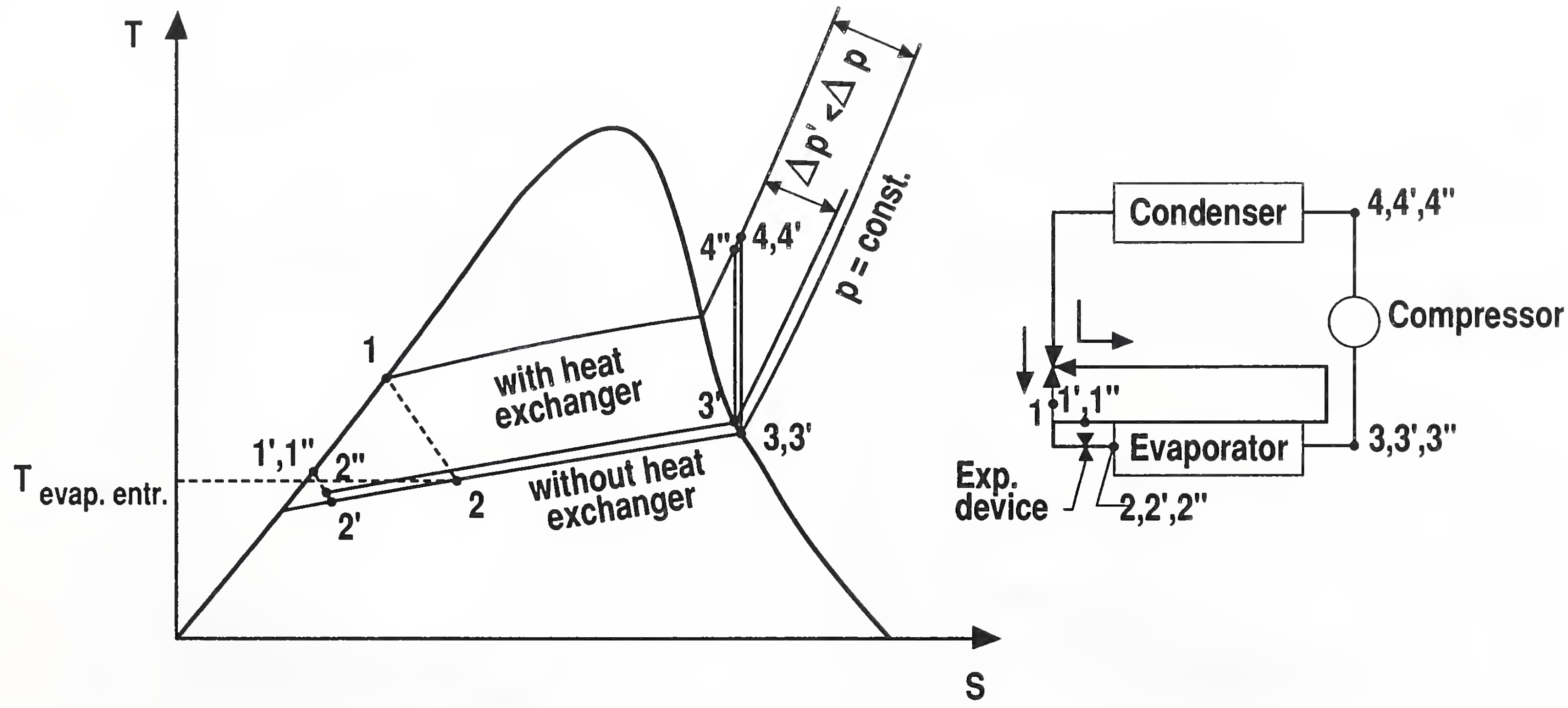


Figure 4. Effect of Subcooling on Nonazeotropic Refrigerant Mixture

An additional improvement in cycle efficiency can be obtained for a heat pump employing a nonazeotropic refrigerant mixture by introducing an internal heat exchange between the evaporating liquid and the condensed fluid, as described by Vakil [6]. Nonazeotropic mixture undergo a temperature glide while evaporating or condensing at constant pressure. If the liquid can be substantially subcooled before the expansion process, the evaporation could start at a lower temperature at the same pressure, as can be seen from Figure 4 (point 2'). For a fixed system capacity, the average refrigerant to heat exchange fluid temperature difference will be fixed. In order to do this, the evaporator pressure will be higher for the process with substantial subcooling (2", 3"). The higher evaporator pressure results in less pressure lift in the compression cycle, therefore reducing the amount of work to be supplied.

3.1 Test Apparatus

The breadboard heat pump is shown schematically in Figure 5, as built in Figure 6 and photographically in Figure 7 and Figure 8. The water heat source and sink were selected because of their simplicity of capacity measurements and of achieving counterflow and intracycle heat exchange. Both water loops, evaporator and condenser, are closed loops. The water is circulated by centrifugal pumps and the flow rates are controlled with manual valves to achieve the desired outlet temperatures of the heat exchangers.

A constant evaporator heat flux was desired to compare the different refrigerants [5]. Because of the different properties of the refrigerants to be examined, this required a variable speed compressor. Since compressor tests with a mixture of R22/R114 were done at the University of Hannover (FRG) for an open two piston reciprocating compressor [7], this same compressor model was selected for the breadboard heat pump. This was achieved by driving an open compressor with a variable speed motor. A strain gage torquemeter and tachometer were incorporated in the shaft drive to allow power measurement. An oil separater was installed at the compressor outlet to minimize the effect of this variable (oil concentration) on system performance.

The condenser heat exchangers are coiled aluminum extrusions (Figure 9). These heat exchangers result in good heat transfer between the refrigerant and the water, but do not allow measurement of temperatures other than at their entrance and exit.

The evaporator heat exchangers (Figure 10) were chosen to allow for temperature measurements all along the flow. Two 20 ft. (6 m) long extruded copper tubes were, therefore, selected for the evaporator heat exchanger. For convenience in installation, these tubes were bent into a hairpin shape reducing their overall length to ten ft. (3 m). As obtained from the manufacturer, the heat exchanger tubes consisted of only two passages, a center path with 'micro-fins' on the walls and a set of rectangular channels around this inner tube. The fins and channels are spiraled, changing top to bottom every 6 in. (150 mm).

A third passage was added on the outside of the evaporator to obtain a heat exchanger for a third fluid, thus allowing for heat exchange between the evaporator refrigerant and the condensed liquid refrigerant throughout the length

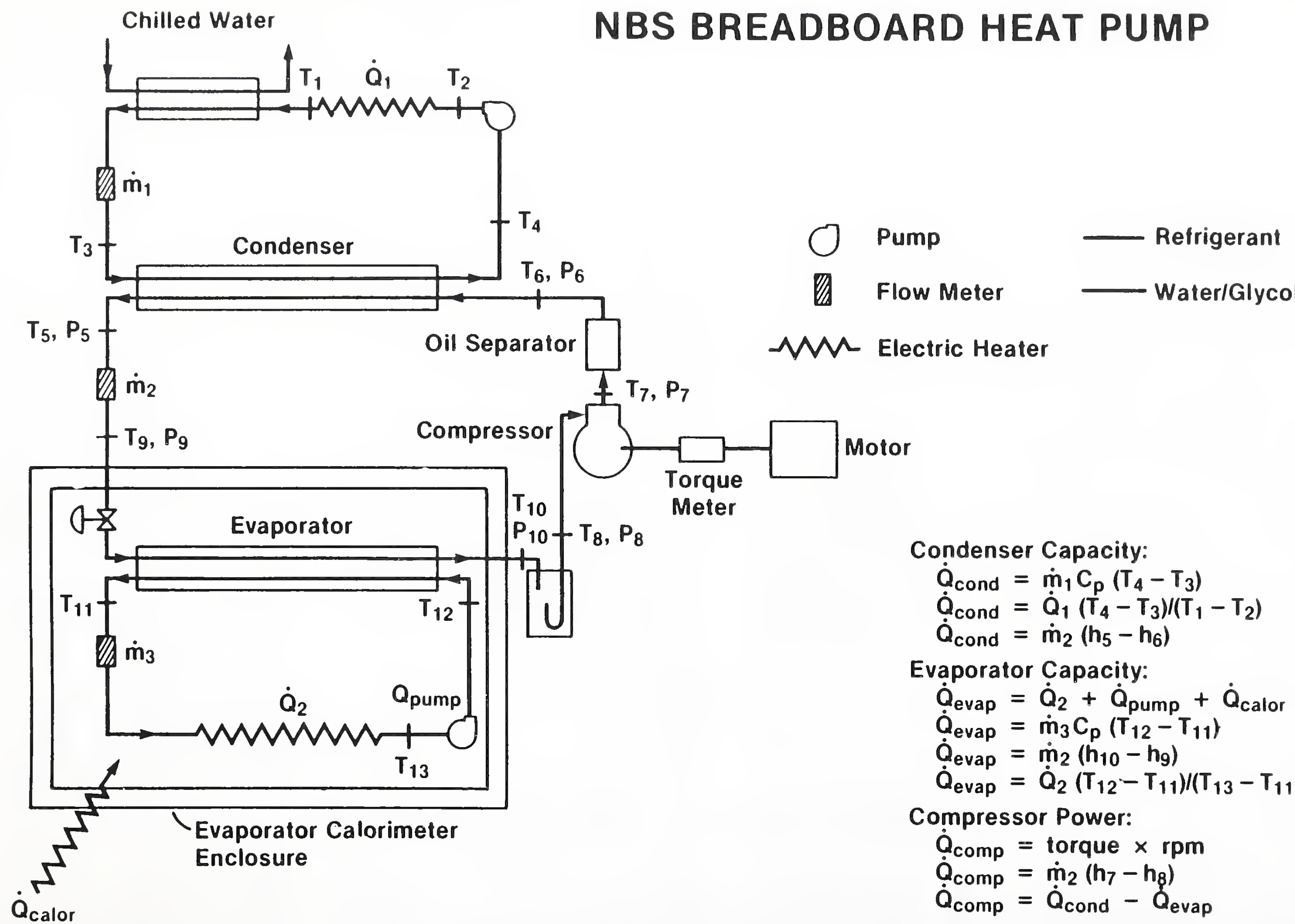


Figure 5. Schematic of Breadboard Heat Pump.

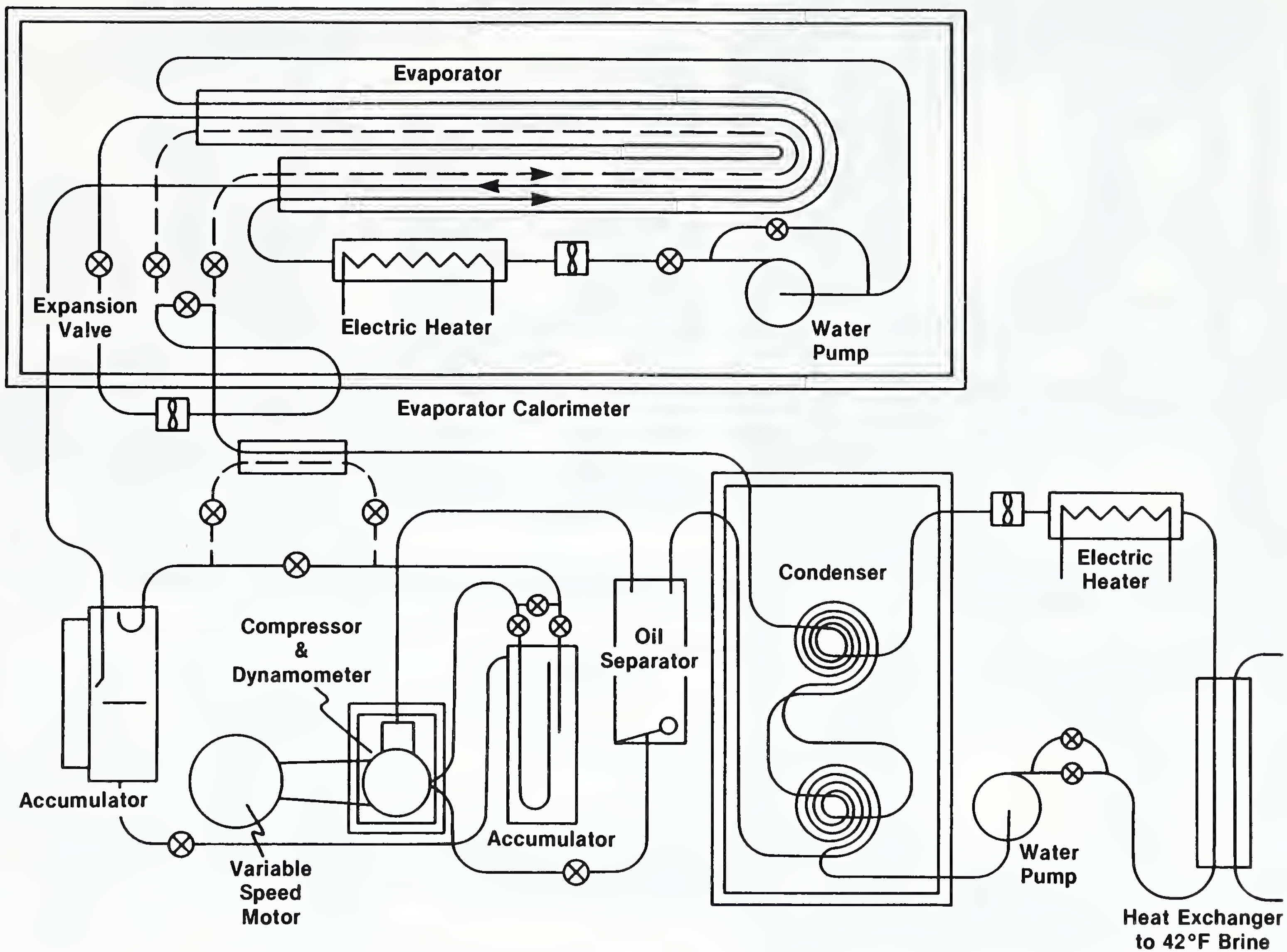


Figure 6. As-built Drawing of Breadboard Heat Pump

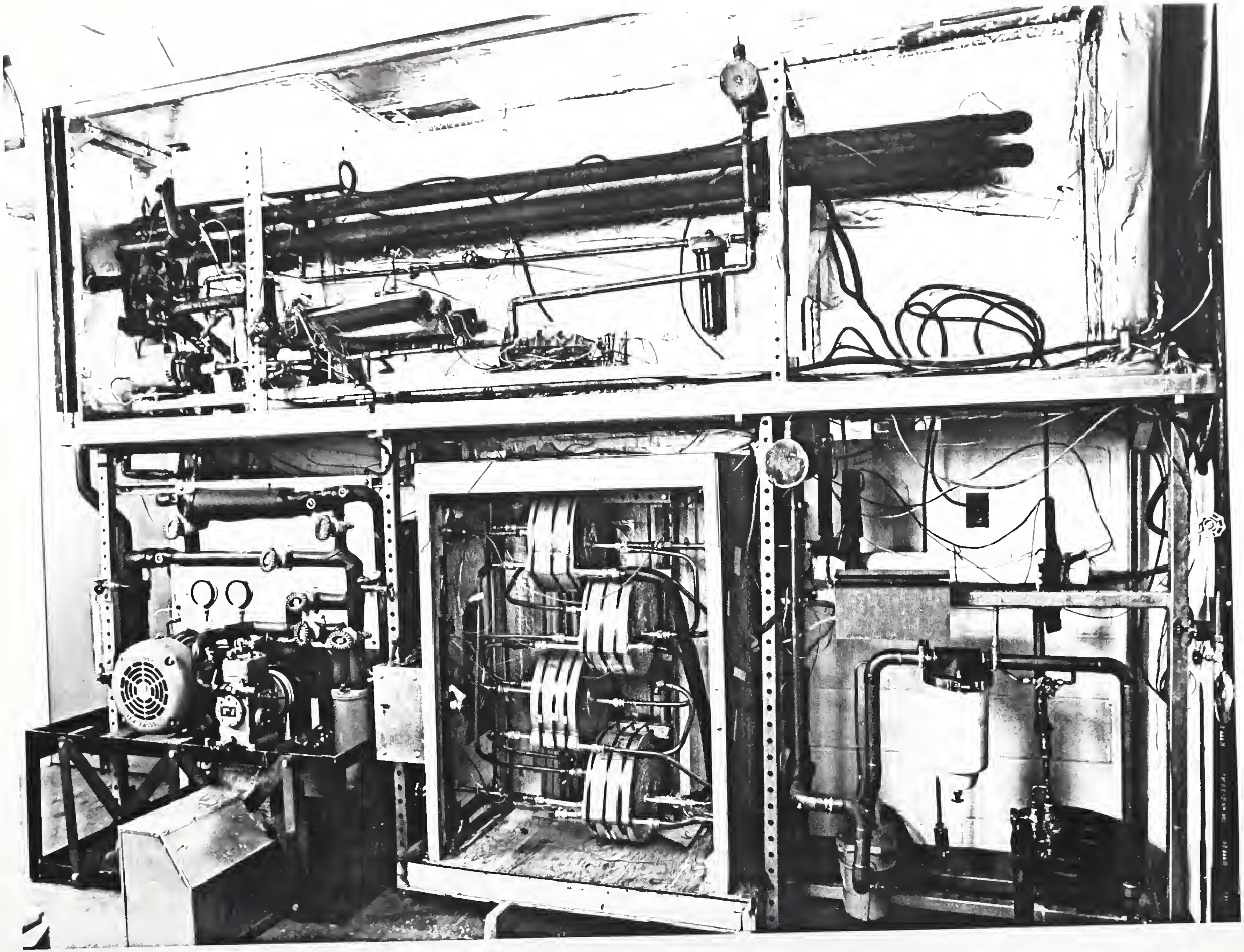


Figure 7. Photograph of Breadboard Heat Pump with Insulating Covers over Evaporator and Condenser Removed.

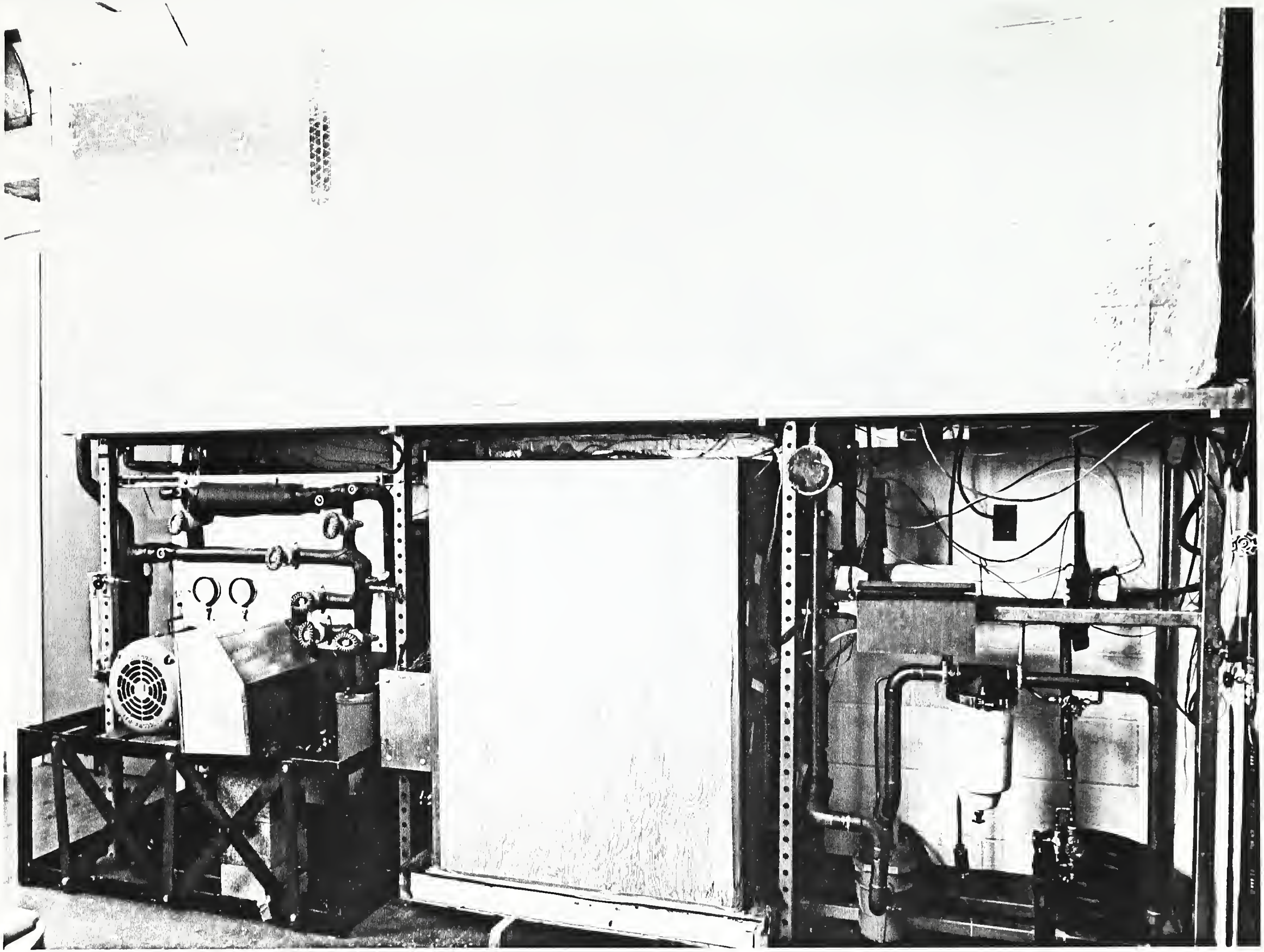


Figure 8. Photograph of Breadboard Heat Pump with Insulating Covers over Evaporator and Condenser Installed.

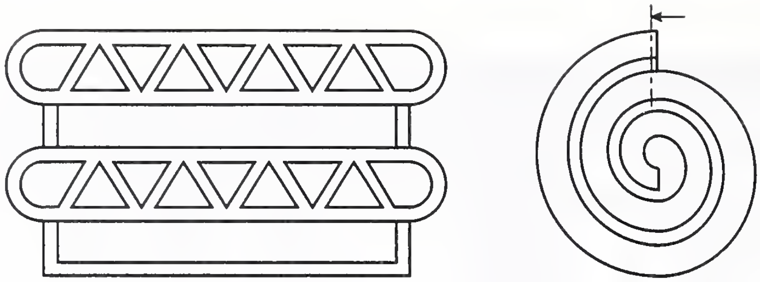
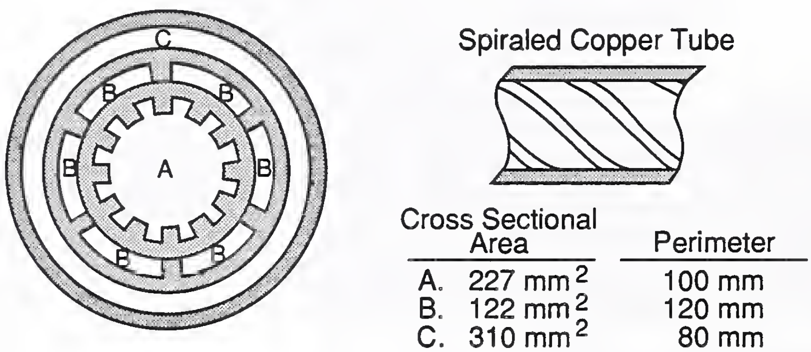


Figure 9. Condenser Heat Exchangers (not to scale)



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

Figure 10. Evaporator Extruded Heat Exchanger (not to scale)

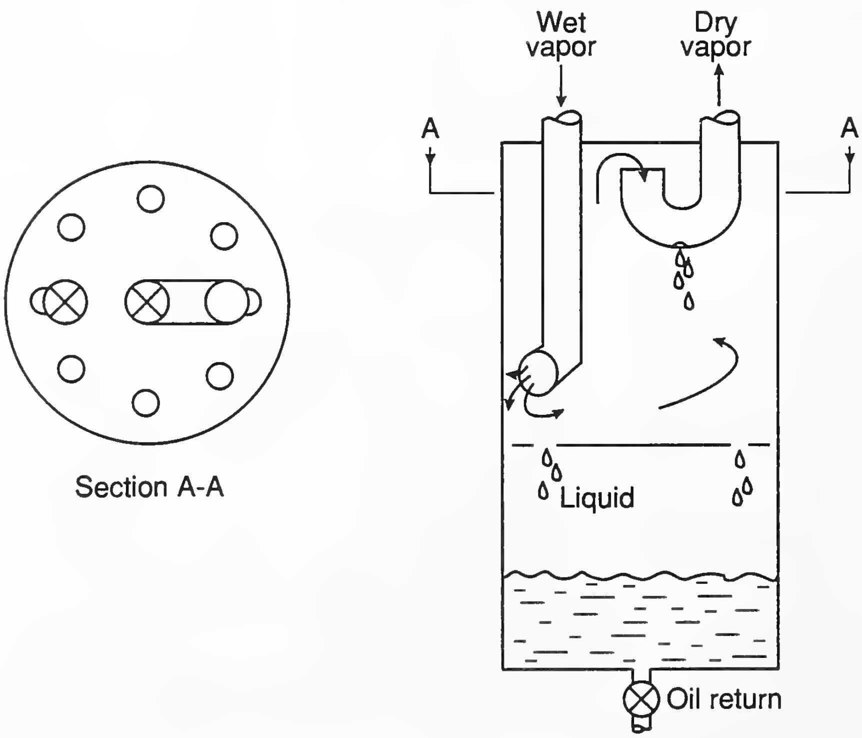


Figure 11. Accumulator Construction (not to scale)

of the evaporator. The two heat exchanger tubes are connected in series. resulting in a total evaporator heat exchanger length of 40 ft. (12 m).

Thermocouples were soldered to the outside of the evaporator heat exchangers every two ft. (60 cm) allowing measurement of the temperature profile in the outermost channel. For the rectangular channels, measurement was limited to the crossover between ten ft. (3 m) section of the 40 ft. (12 m) heat exchanger. This resulted in five thermocouples for these passages. For the center passage, thermocouples are located every two ft. (60 cm) in a sealed 1/8 inch (3.2 mm) thin-wall stainless steel tube. These thermowells are mounted in the center of the innermost channel to measure the in-stream temperature of the flowing fluid. Further measurement devices were several other thermocouples, thermopiles and pressure transducers located at critical points in the cycle.

Another important feature of the breadboard heat pump is a liquid/vapor separating accumulator in front of the compressor serving as a refrigerant storage vessel as well (Figure 11). The ability to store refrigerant in the accumulator makes the breadboard heat pump charge-insensitive by allowing flooded evaporator operation, making charging the heat pump easier. For the first series of system checkout tests, a large, commercially available accumulator was employed. This accumulator, however, contributed to changes in compressor performance at different speeds because of the large size of its oil return holes. The amount of liquid returned to the compressor was found to vary with velocity of the suction gas flowing through the accumulator and with the liquid level. Due to different amounts of liquid refrigerant droplets carried into the cylinder of the compressor at different speeds, the changes in system performance over the compressor speed range were amplified. A liquid/vapor separating accumulator with an oil return line which can be valved off during tests was, therefore, constructed and installed. The original accumulator was left installed for compressor protection during startup and for charge storage, but was valved off during test data periods.

Since this configuration supplies only saturated vapor to the suction line of the compressor, one would not be able to run with misty vapor leaving the evaporator, which was desired so as not to use a large amount of evaporator area drying out the droplets entrained in the center of the flow. A fixed heat (60 W) was, therefore, provided to the accumulator by wrapping it with an electric heating tape and then insulating. This small amount of heat was not included in the capacity calculations. This accumulator decreased the variation in system efficiency substantially. See section four, Compressor Speed Tests, for measured efficiency versus speed.

3.2 Capacity Measurements

The breadboard heat pump provides several capacity measurement methods for both evaporator and condenser as summarized in Figure 5. In the condenser, heat is gained by the water from the refrigerant. This heat is rejected to chilled water at 45°F (7°C) in a separate heat exchanger. A trim heater past this heat exchanger adjusts the inlet water temperature to the condenser heat exchanger to the desired 82°F (28°C). The electrical power to this variable heater is measured. The temperature differences across the condenser heat exchanger and across the electric heater are also measured. The ratio of these two temperature differences multiplied by the electric heat gives the condenser capacity.

A second capacity measurement method for the condenser is based on water side calculations. The average density and specific heat for the water are calculated based on the water temperatures. In conjunction with the flow rate, one can obtain the heat gained by the water. To be able to measure losses to the environment, the condenser heat exchanger is contained in an insulated box. The heat loss through the wall has been determined in relation to the temperature difference. The temperature difference between the inside of the box and the environment multiplied by the heat loss constant gives the heat lost to the environment. The total capacity is the sum of the heat gained by the water and the heat lost to the environment. A third capacity measurement method is based on refrigerant side measurements. The flow rate of the refrigerant is measured. The enthalpies leaving and entering the condenser are obtained from properties programs according to the measured temperatures and pressures and capacity calculated as the product of mass flow and enthalpy change.

In the evaporator, heat is transferred from the water to the refrigerant. The heat exchanger with all water pipes and the pump is contained in an insulated box, thus allowing a capacity calculation based on the calorimetric method. All heat sources, i.e., electric heat, pump power, and wall losses/gains are added together to give the first of four capacity measurement methods. This is felt to be the most reliable method because of the accuracy with which electric power can be measured in comparison to flow rates or small temperature differences and is, therefore, used for the performance analysis. Evaporator capacity was also measured by the three methods (water side, comparison heater, and refrigerant side) as described for the condenser.

Since an open compressor was employed, the compressor power was measured as shaft power, eliminating electrical efficiency of the motor. A shaft dynamometer (strain gage torquemeter with magnetic pickup tachometer) had been installed between the compressor shaft and a pulley which was belt driven by a variable speed motor.

Two additional checks on compressor power were a refrigerant side comparison and the difference between the condenser and evaporator capacity. For these two methods, the compressor jacket loss was subtracted before comparison to the shaft power measured by the dynamometer.

An insulated box was installed around the compressor, enabling measurement of the compressor jacket heat loss. This heat loss was found to be quite small (typically 100 to 200 Btu/hr (30 to 60 W)) reflecting the manufacturer's intention of providing primary compressor cooling by the incoming refrigerant vapor. For system efficiency calculation the shaft power was used.

All the different methods employed throughout the test program agreed closely (within 6%), confirming the validity of the test results, with the exception of occasional periods when flowmeter mechanical failure was experienced.

3.3 Test Procedure

Two nonazeotropic binary refrigerant mixtures were tested in the breadboard heat pump, R22/R114 and R13/R12. Tests were conducted at the same capacity for each mixture, to achieve equal average heat flux per unit area of the heat exchangers for all tests as one of the major criteria for comparability of refrigerants. One exception was pure R114, which would have required an excessively high compressor speed to reach the desired capacity. The capacity was chosen to be 14,000 Btu/hr (4.1 kW), which was obtained with pure R22 at 500 RPM compressor speed. Heat sink and heat source entering and leaving water temperatures were kept constant for all tests as the second major criterion for comparability [5]. The temperatures were 82°F (28°C) and 117°F (47°C) entering and leaving the condenser and 80°F (27°C) and 55°F (13°C) entering and leaving the evaporator. Since all test series were conducted at the same capacity and entering and leaving evaporator water temperatures were fixed, it follows that the evaporator water flow rate was also fixed. The compressor speed had to be adjusted for every test to meet the required capacity due to differences in suction vapor specific volume and latent enthalpy between the different refrigerants. The expansion device was set simultaneously to give marginal subcooling at the condenser outlet and to provide evaporation throughout the entire evaporator. The subcooling was determined by the disappearance of vapor bubbles in a sight glass in the liquid line. The flooded evaporator was determined by the presence of liquid droplets leaving the evaporator in a sight glass in this pipe and by the presence of liquid in the accumulator.

The mixture compositions examined cover the whole range from 0 to 100% for the R22/R114 test series, whereas for the second refrigerant mixture (R13/R12) compositions were limited to less than 45% of R13 by weight because of excessive pressures in the condenser that would have existed with more R13. The composition of the mixture was determined for each test by taking a vapor sample from the compressor discharge line and analyzing it with a gas chromatograph. When filling the sample bottles the entering refrigerant was throttled to keep the sample bottle pressure below saturation at room temperature. This prevented condensation of refrigerant on the sample bottle interior walls, which would have resulted in a composition shift in the vapor sample. At least two analyses for each sample were performed to validate the sample and reduce the chance of operator error. If condensation had occurred in the sample bottle, the two analyses would have differed because of evaporation as the sample was drawn into the chromatograph. The evaporator capacity and the amount of work supplied to the compressor were calculated according to the methods described under Test Apparatus. The coefficient of performance (COP) was determined for each test based on the capacity calculated by the calorimeter method, and the measured shaft power. The power required by the heat source/sink water loop pumps was not included in the COP calculation, because it was felt to be external to the basic equipment cycle and, instead. part of a systems application.

4. Compressor Speed Tests

A set of preliminary system performance tests at different compressor speeds was conducted with pure R22. The examined speed range covers the speeds used for the mixture tests, explicitly 500 to 1200 RPM except near 750 RPM when the compressor showed excessive vibration in its mounting. Since we felt these vibrations could damage the apparatus, no mixture tests were run at this speed.

The suction and discharge pressures were kept constant for all speeds for these compressor tests. The pressures, 104 psia (720 kPa) suction and 258 psia (1780 kPa) discharge line, were those which existed in the apparatus for the given water conditions (80/55°F) (27/13°C) evaporator and 82/117°F (28/47°C) condenser) at 500 RPM compressor speed. Two maintain these pressures for all speeds the water flow and temperature were changed. The expansion device was adjusted for each speed to give marginal subcooling and some flooding of liquid refrigerant into the accumulator. The results for these tests are shown on Figure 12.

As one can observe in this figure, the measured performance (excluding the point at 750 RPM where excessive vibration was experienced) lies within a band of ± 5%. The system performance was felt to be adequately consistent for the speed range of 500 to 1200 RPM to allow the start of mixture testing. No corrections were made to the performance obtained for different mixtures at different speeds.

5.1 Heat Exchanger Variations

A major task of this project was to analyze the influence of different heat exchangers on the performance of a heat pump employing nonazeotropic refrigerant mixtures. Four different configurations of the three passages in the evaporator were tested for this part of the project. Detailed instrumentation and observation have been limited to the evaporator because this heat exchanger has a greater effect on system performance than the condenser. According to the nomenclature of Figure 10 the four evaporator configurations were:

	Configuration			
	1	2	3	4
Evaporating Refrigerant	C (outside)	B	A	A
Liquid Refrigerant	B (rectangles)	C	B	C
Water	A (inside)	A	C	B

Expectations were that configurations 2 and 4 would perform better than 1 and 3 since the evaporating refrigerant and water (which constituted the majority of the transferred heat) were in the channels which had enhanced surfaces. Furthermore, these channels were adjacent. which limited the losses due to heat passing through the intermediate passage.

The performance of configuration 1 and 3 are strongly influenced by the surface characteristic of the outermost channel. The smooth copper provides a poor heat exchange area resulting in large temperature differences between the water and the refrigerant for these two configurations. The relative heat exchanger performance is shown in the temperature plots for the four configurations for pure R22 in Figure 13. It can be observed that R22 uses the full length of the heat exchanger for configuration 1 and 3, but only half the heat exchanger for

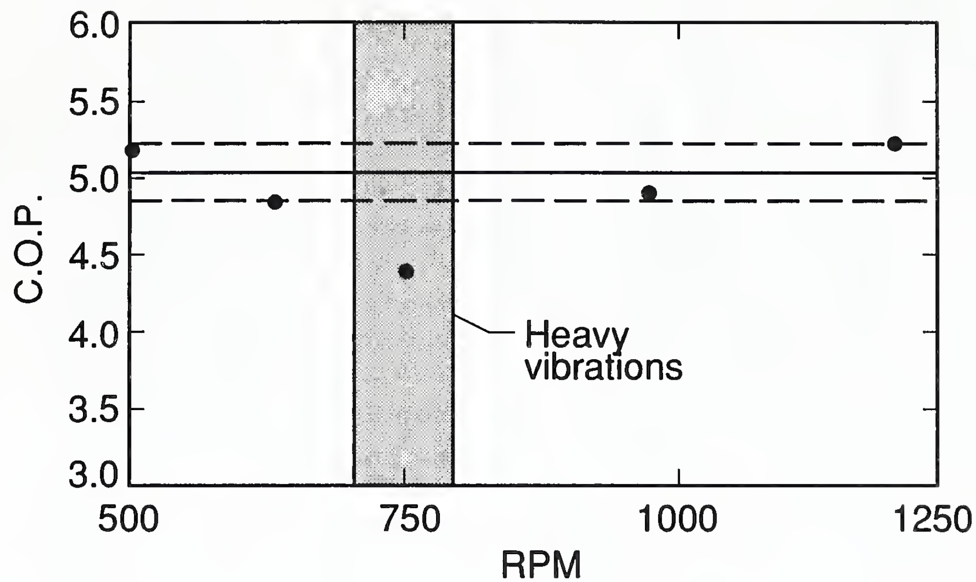
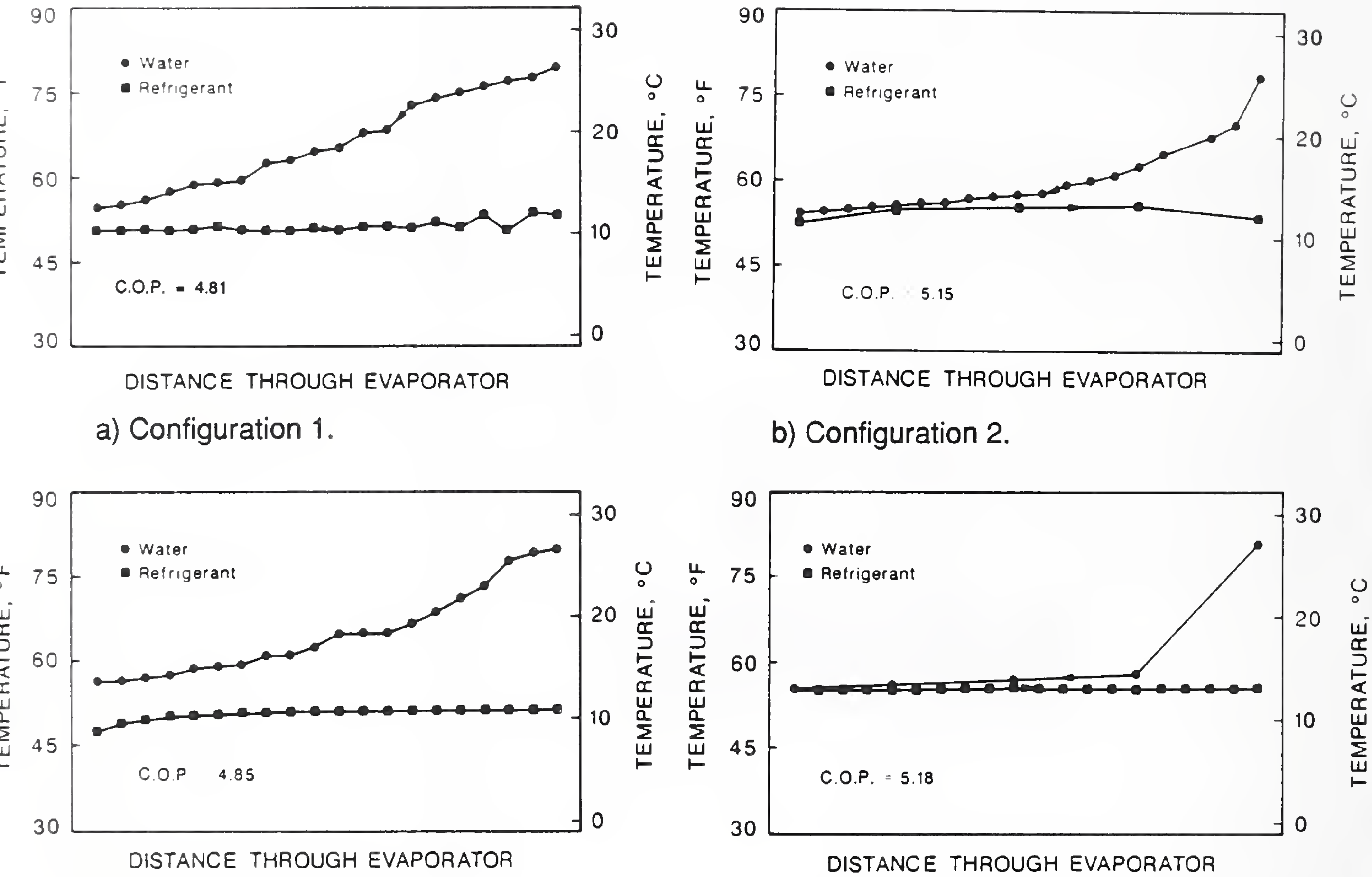


Figure 12. Performance of the Heat Pump versus Compressor Speed



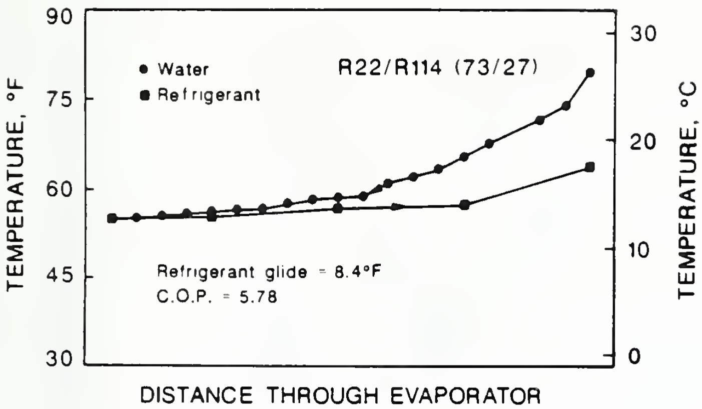
a) Configuration 1.

b) Configuration 2.

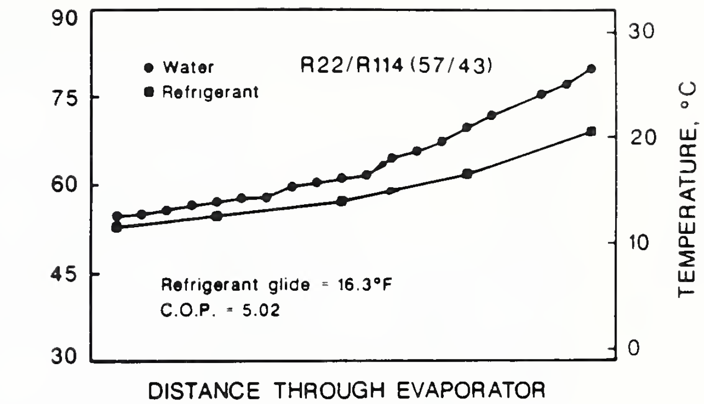
c) Configuration 3.

d) Configuration 4.

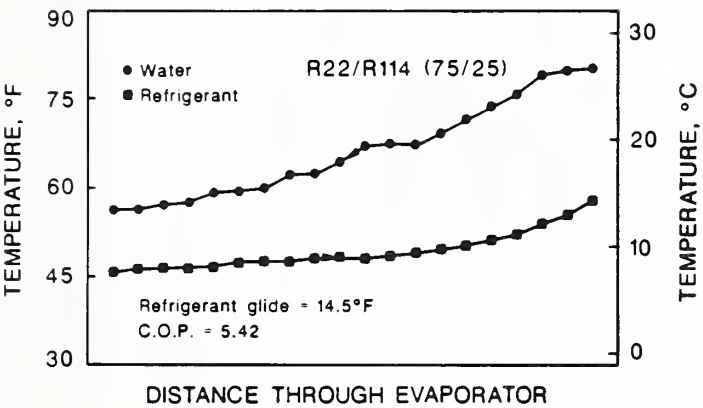
Figure13. Water and Refrigerant Evaporator Temperature Profiles Using Pure R22.



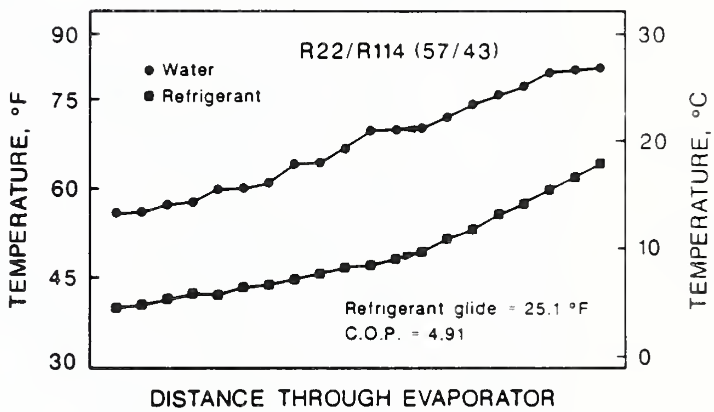
a) Configuration 2, best COP.



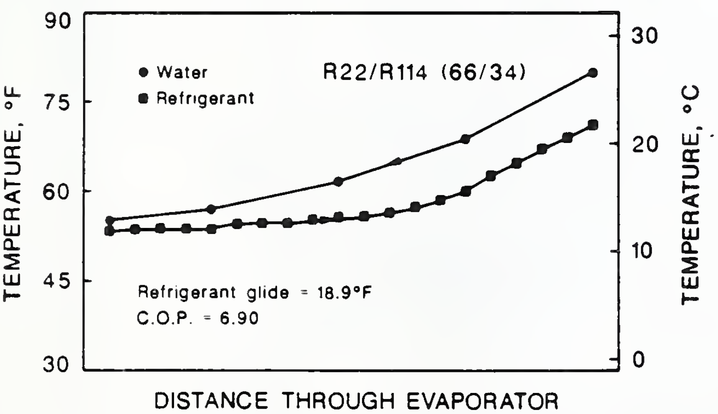
b) Configuration 2, best glide.



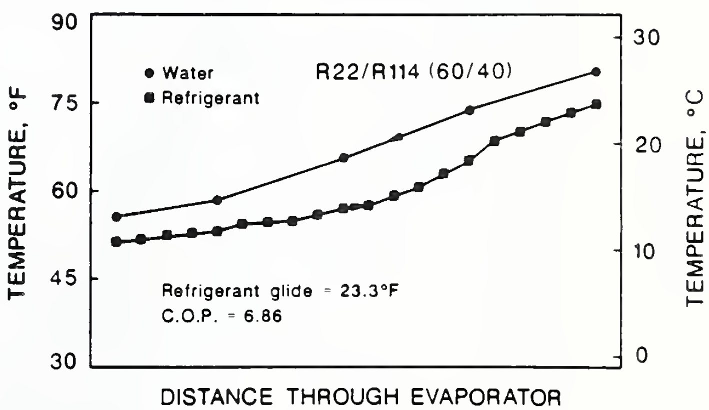
c) Configuration 3, best COP.



d) Configuration 3, best glide.



e) Configuration 4, best COP.



f) Configration 4, best glide.

Figure 14. 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).

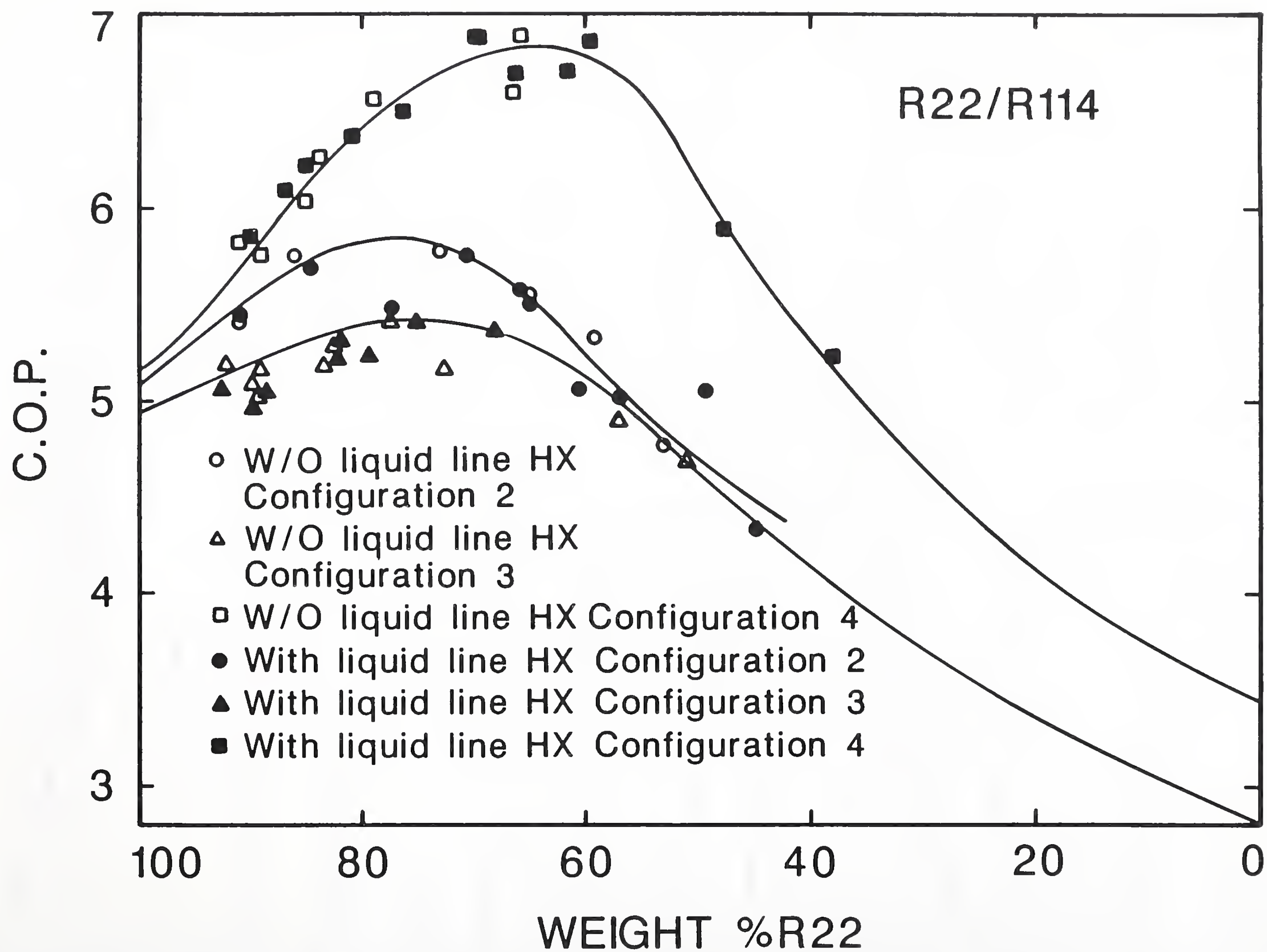


Figure 15. Performance of the Mixture R22/R114 for Evaporator Configurations 2,3 and 4.

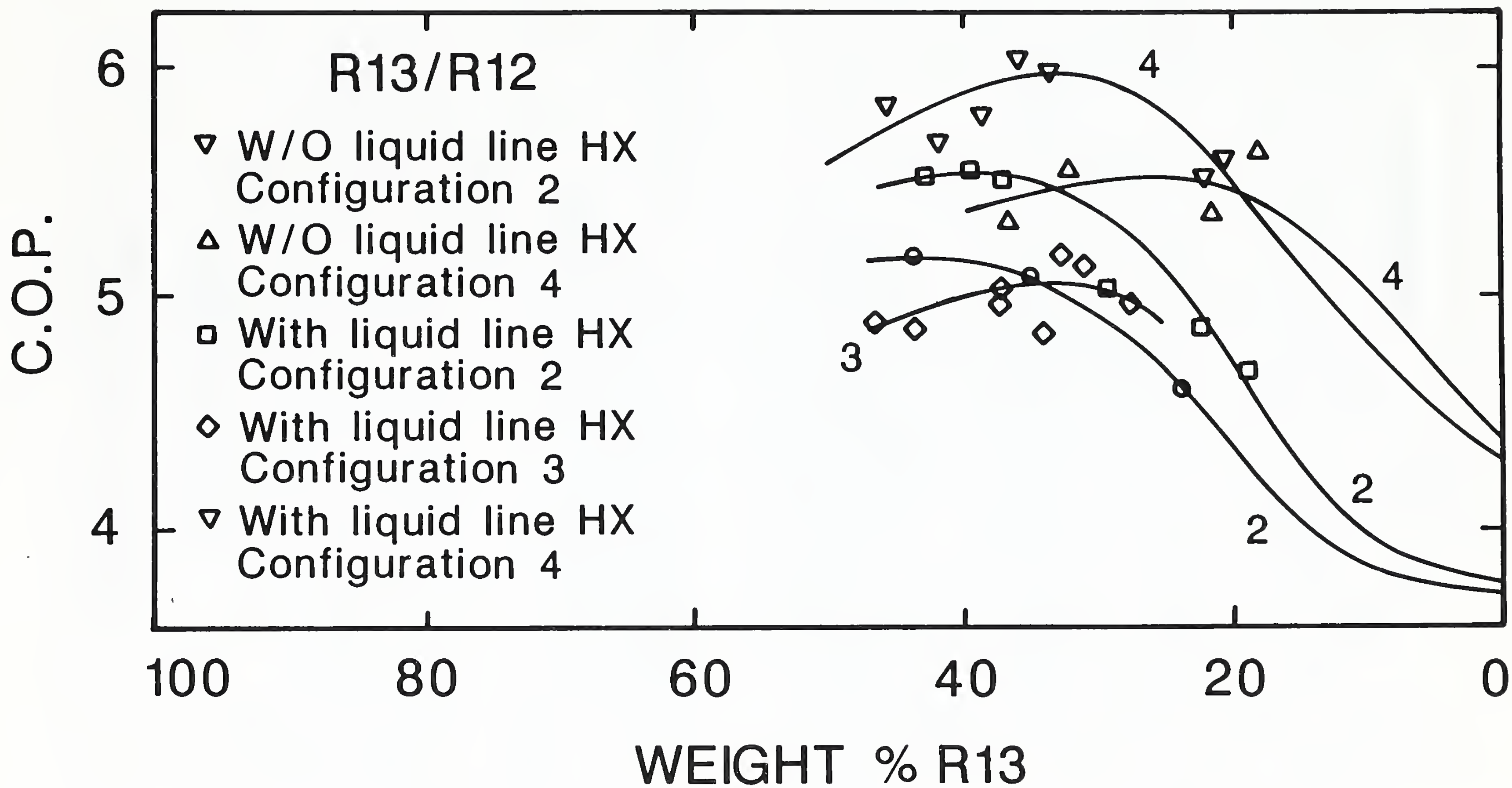


Figure 16. Performance of the Mixture R13/R12 for Evaporator Configurations 2, 3, and 4.

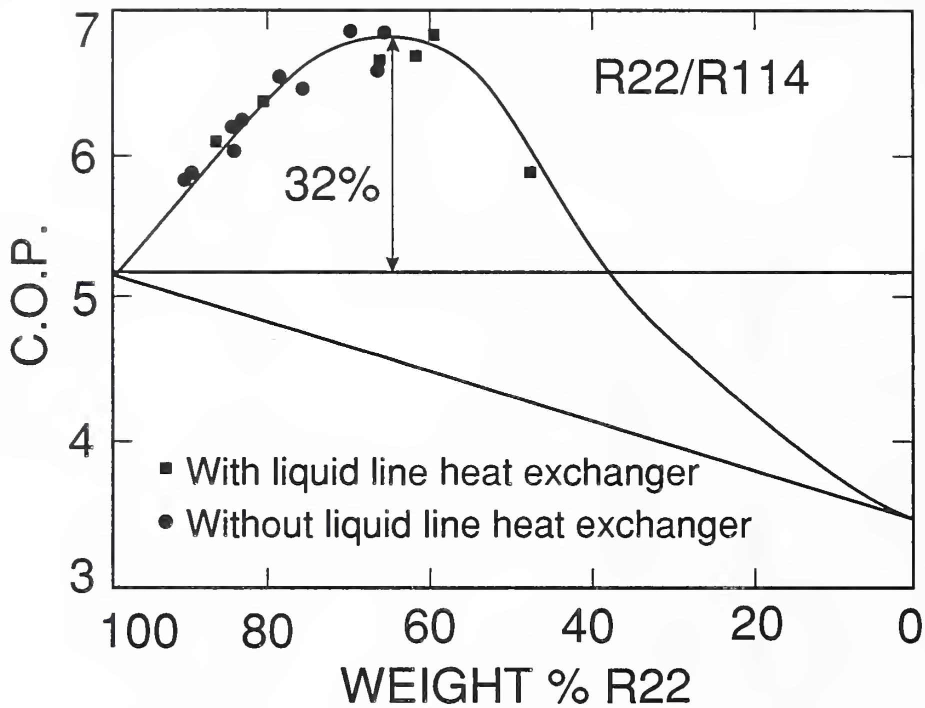


Figure 17. Performance of the Mixture R22 / R114 for Evaporator Configuration No. 4.

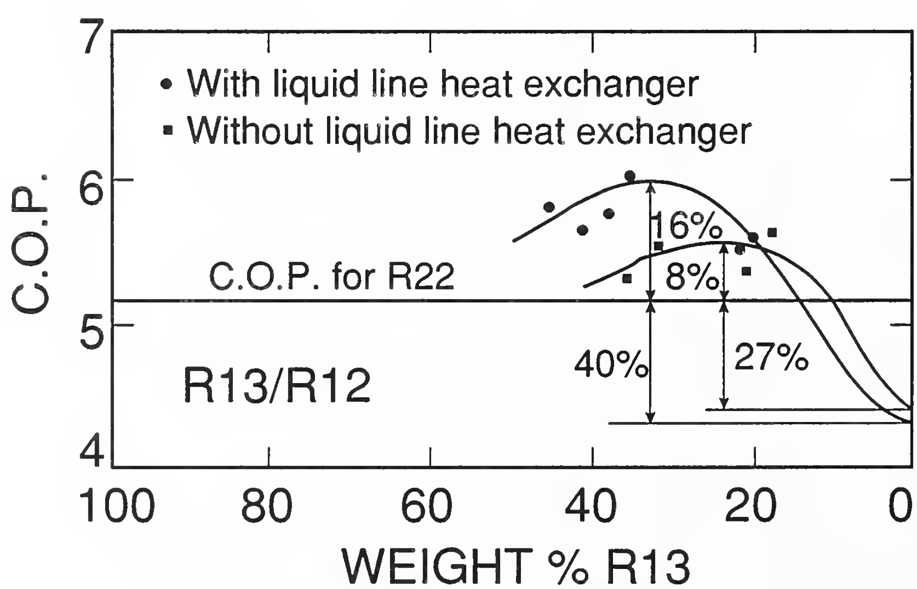


Figure 18. Performance of the Mixture R13 / R12 for Evaporator Configuration No. 4.

between the two end points of the efficiency curve in Figure 17, indicating a potential for increase in COP of 44% for R22/R114.

The improvement in cycle efficiency for the mixture R13/R12 compared to the efficiency obtained with pure R22 and pure R12 is shown in Figure 18 to be 16% and 40% respectively with intracycle heat exchange and 8% and 27% respectively without intracycle heat exchange. For the test set performed with R13/R12, it was not possible to perform tests with high concentrations of R13 since the pressure would have been far beyond the highest acceptable pressures.

Another observation that can be made is that at any temperature glide less than the maximum possible with a mixture there will be two compositions that will give the required glide. The choice of which of these two compositions to use would be based on other refrigerant properties For instance with the R22/R114 mixture, the composition high in R22 would be chosen because of the better compressor efficiency and capacity with higher R22 concentrations. With the R13/R12 mixture the composition low in R13 would be chosen because of the high discharge pressures with high concentrations of R13.

The capacity per compressor revolution for both mixtures in evaporator configuration 4 is shown in Figure 19. Compared to pure R22, the 65% R22/35% R114 mixture which showed a 32% efficiency improvement (Figure 17) resulted in a capacity loss of 8%. The 34% R13/66% R12 mixture which showed a 16% efficiency improvement over pure R22 (Figure 18) resulted in a capacity increase of 21%.

5.2 Linearity of Enthalpy versus Temperature

An interesting feature can be observed in the temperature plots, the differing location of the pinch points for the different refrigerant mixtures. The pinch point, the closest temperature approach between the refrigerant and the heat exchange fluid, is at the two ends of the heat exchanger for the R22/R114 mixture, whereas for the R13/R12 mixture the pinch point is in the center of the heat exchanger. This is shown in Figures 20 where a line has been drawn connecting the entering and leaving points to emphasize this nonlinearity.

The nonlinear shape of the temperature profile during evaporation results from the differing properties of the pure refrigerants in combination with mixture effects. The dominant effect in determining the shape of the profiles appears to be the relative amounts of the more and less volatile components. With the R22/R114 mixture with 60% R22 the bulk of heat necessary to vaporize the mixture is required at temperatures close to the bubble point temperature, i.e., at temperatures closer to the boiling point of the more volatile R22. As a result the temperature versus enthalpy profile is concave. With the R13/R12 mixture, the concentration of the more volatile R13 is only 20% and, thus, the bulk of the heat of vaporization is concentrated at the higher temperatures resulting in a convex profile.

A secondary effect is the different heats of vaporization for the two components. This is most pronounced with R22/R114 where the heat of vaporization of R22 is substantially higher than that of R114. Because the first bubbles of

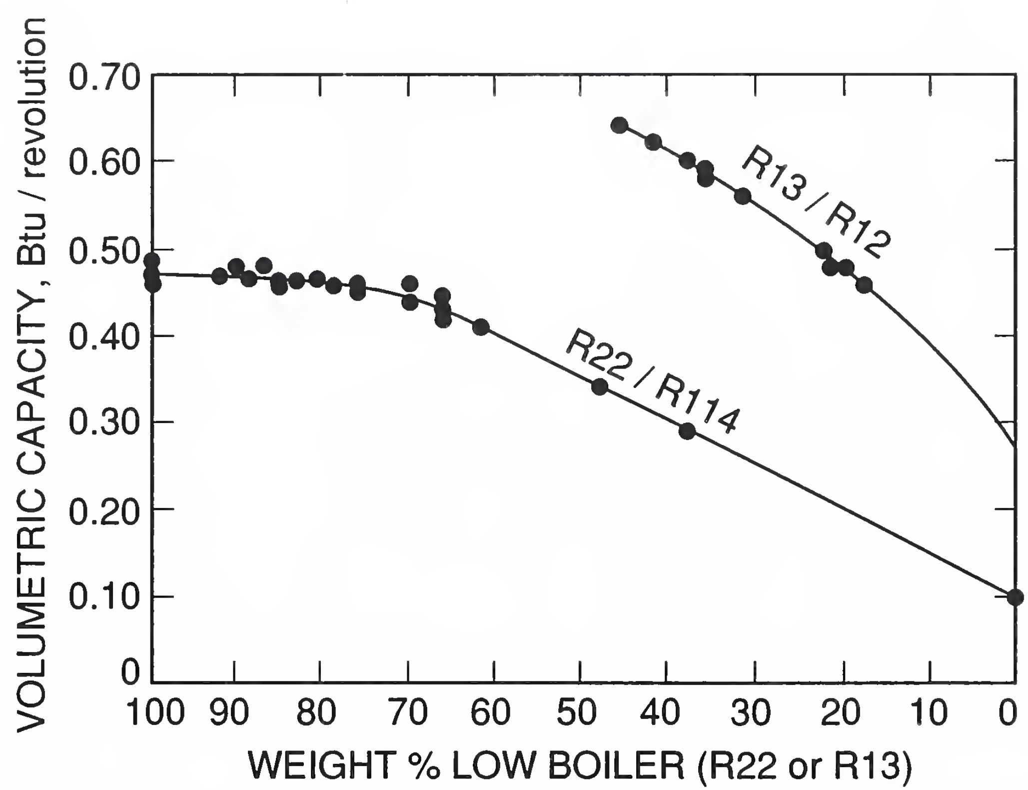


Figure 19. Capacity per Compressor Revolution for the Mixtures R22 / R114 and R13 / R12 in Evaporator Configuration No. 4.

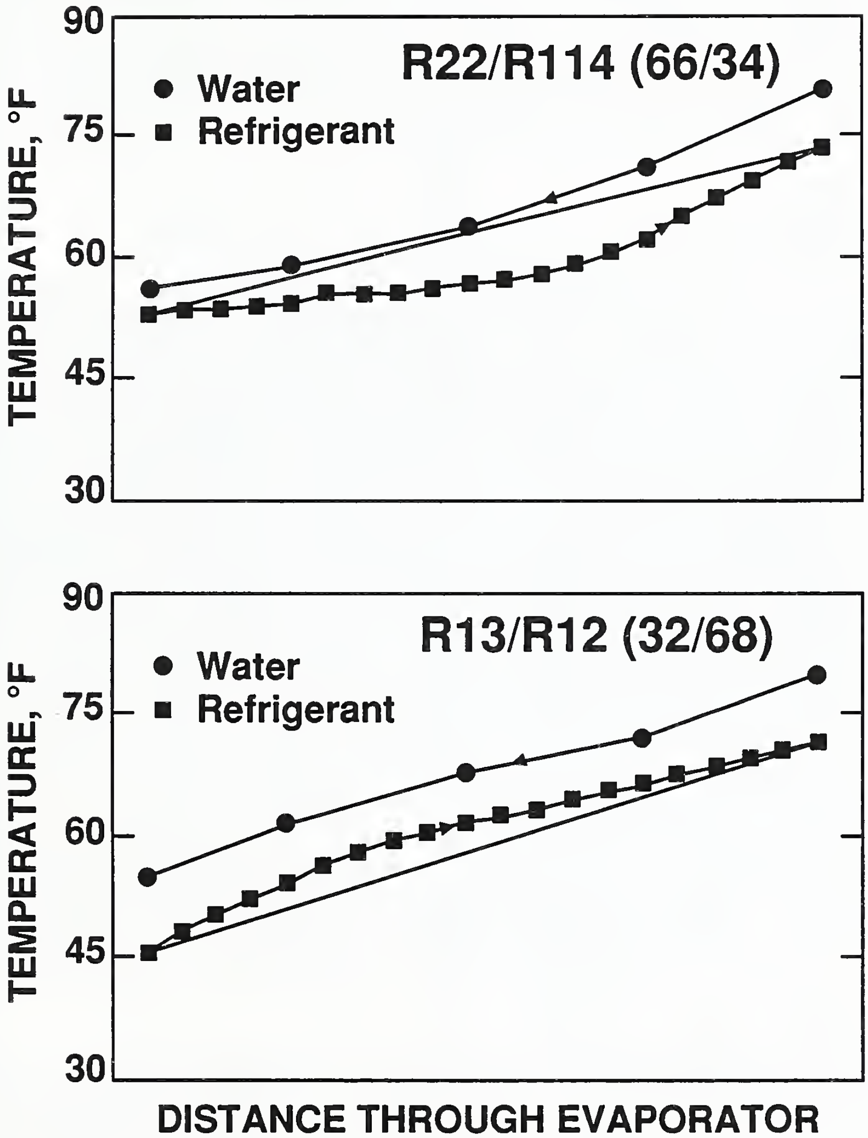


Figure 20. Comparison of Typical Evaporator Temperature Profiles for the Mixtures R22 / R114 and R13 / R12

vapor generated are enriched in R22 the higher heat of vaporization for this refrigerant further contributes to the concave shape. This behavior is important for the prediction of system performance by the use of simple computer programs which assume a linear temperature glide throughout the heat exchanger since the actual glide may deviate quite far from linearity. These errors would be greatest with efficient heat exchangers where the temperature differences would be small - the application that has been seen to be most favorable to mixtures.

The convex shape of the temperature profile of R13/R12 tends to be amplified by a pressure drop as present in the evaporator, whereas the concave shape of R22/R114 tends to be leveled. Another aspect of this nonlinearity is discussed in the next section, Liquid Line Heat Exchanger.

5.3 Liquid Line Heat Exchanger

The final part of this project was concerned with the validity of the proposed improvement obtained by using an intermediate heat exchange between the evaporating refrigerant and the condensed liquid [6]. The liquid line heat exchanger improves efficiency with mixtures by shifting a portion of the evaporator capacity to a lower vapor quality and, thus, lower temperature. Tests have been conducted with both refrigerant mixtures R22/R114 and R13/R12 to evaluate this theoretical proposal. As one can see from Figures 17 and 18 noticeable improvement in efficiency is only obtained for the mixture of R13/R12. The reasons for this different behavior can be seen in the pressure-enthalpy diagrams for the two refrigerant mixtures. First the two-phase isotherms are nearly flat at low quality for R22/R114 (Figure 21) while those for R13/R12 (Figure 22) show a substantial slope throughout the two phase region. Because of this flatness the heat exchanger has little value for R22/R114 for the same reasons that cause it to have little value for pure refrigerants with their truly horizontal isotherms. Referring to Figures 21 and 22, the shift of the evaporator entering condition from point 2 to 2' can be seen to result in a higher evaporator pressure (which improves efficiency) at the same average evaporator temperature for R13/R12 while the shift from point 2 to 2' has no effect on evaporator pressure for R22/R114. Secondly, it can be seen that the shape of the two-phase dome results in flashing more of the R13/R12 mixture between the approximate operating points (30°C condensing, 10°C evaporating) at which the tests were run, indicating more potential for reducing entering quality.

The flatness of the R22/R114 isotherms at low qualities are another aspect of the nonlinearity of enthalpy versus temperature. In this case, this departure from ideality renders ineffective the performance of this heat exchanger for one mixture (R22/R114) while increasing its value for the other (R13/R12).

Another way in which heat exchange between the evaporator and condenser improve cycle efficiency is described in [9]. Referring to figure 23, if condensation occurs by heat exchange with the evaporating refrigerant the discharge pressure will be lowered reducing the power input to the cycle. Countering this benefit, the refrigerant mass flow involved in this heat exchange will be pumped around the cycle without contribution to capacity. Maximum efficiency would occur where the tradeoff between these two effects was optimized. For this cycle

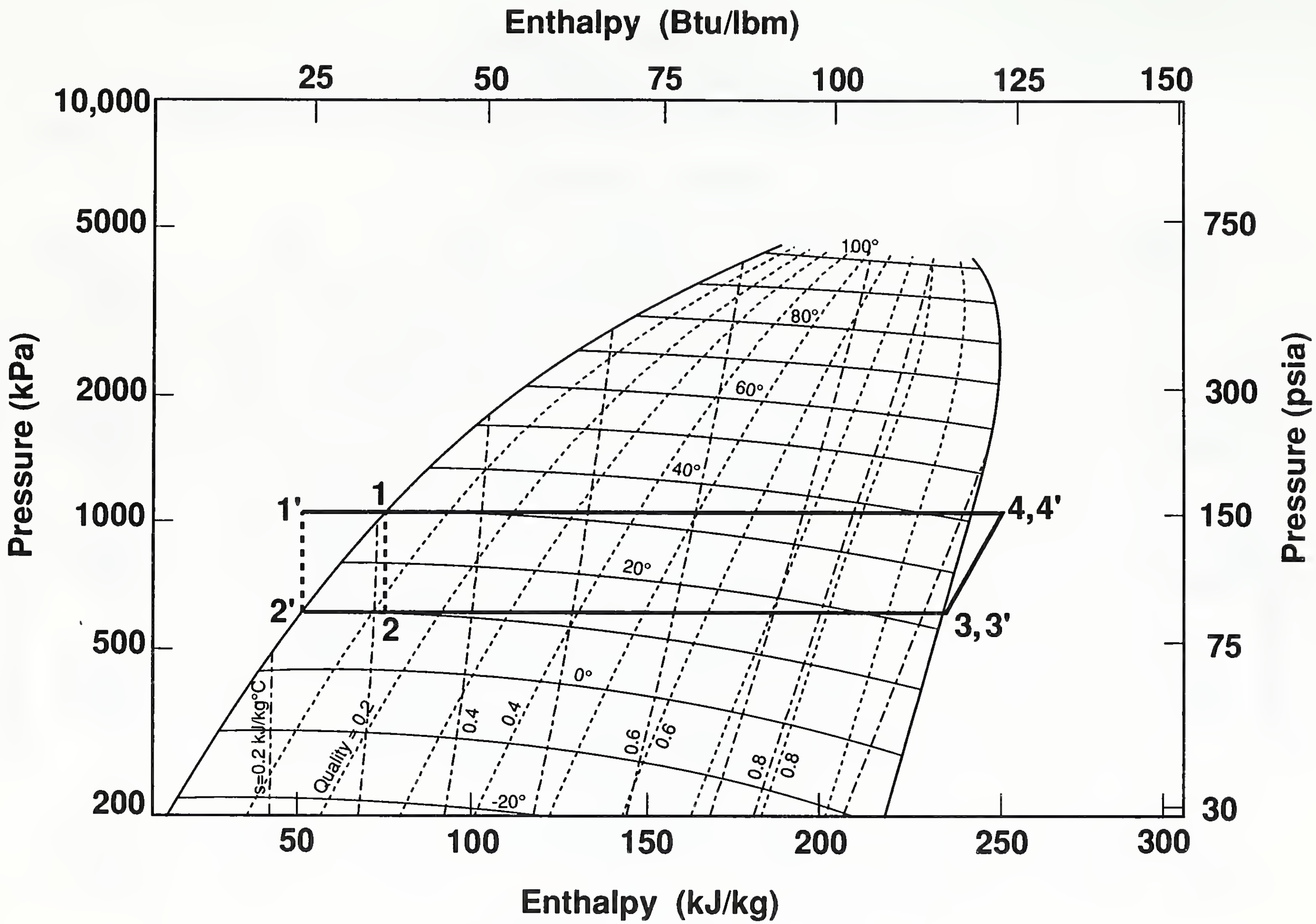


Figure 21. Effect of Heat Exchange Between Subcooled Liquid and Evaporating Refrigerant of 60% / 40% Mixture of R22 / R114

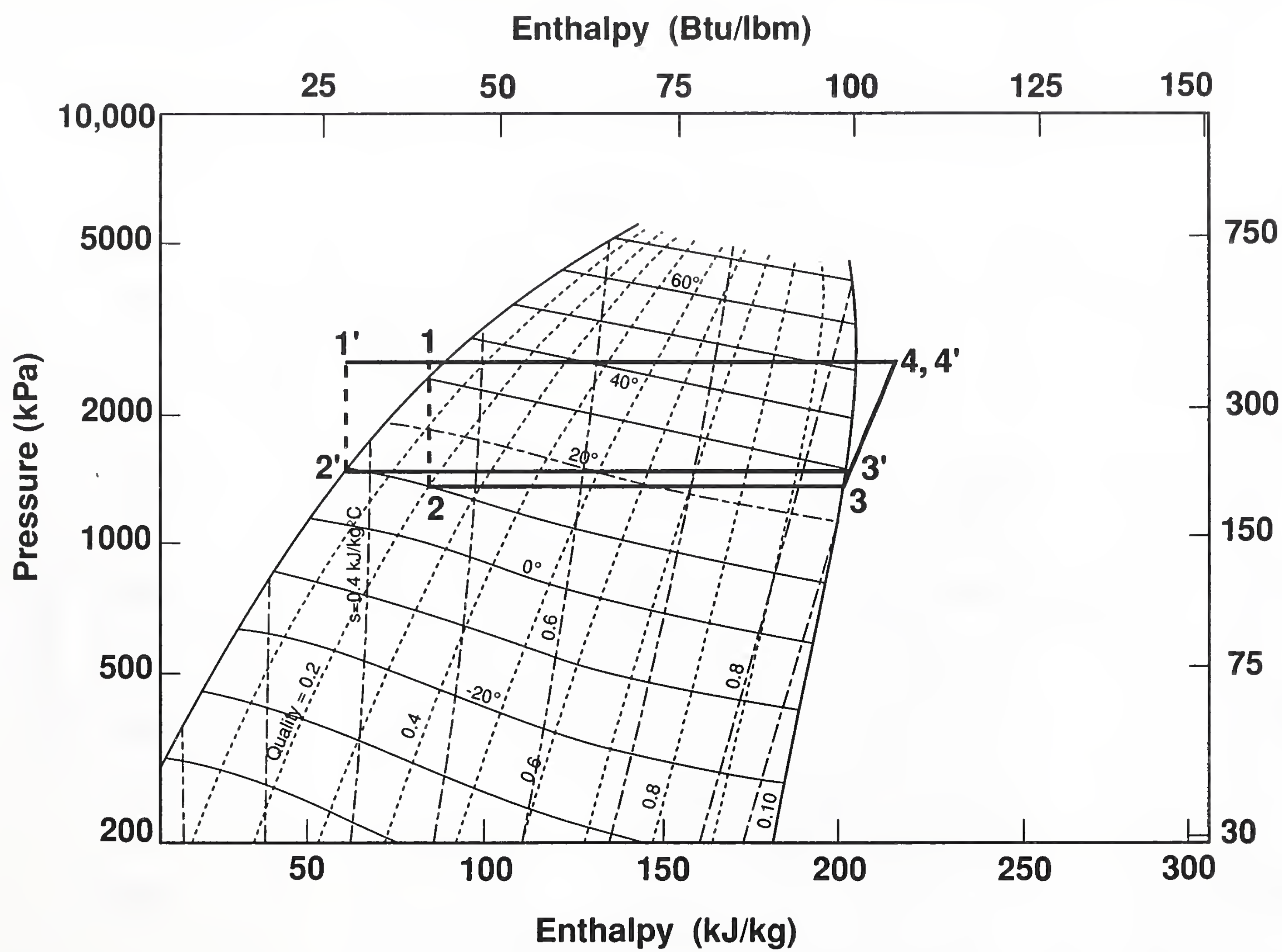


Figure 22. Effect of Heat Exchange Between Subcooling and Evaporating Refrigerant of 40% / 60% Mixture of R13 / R12

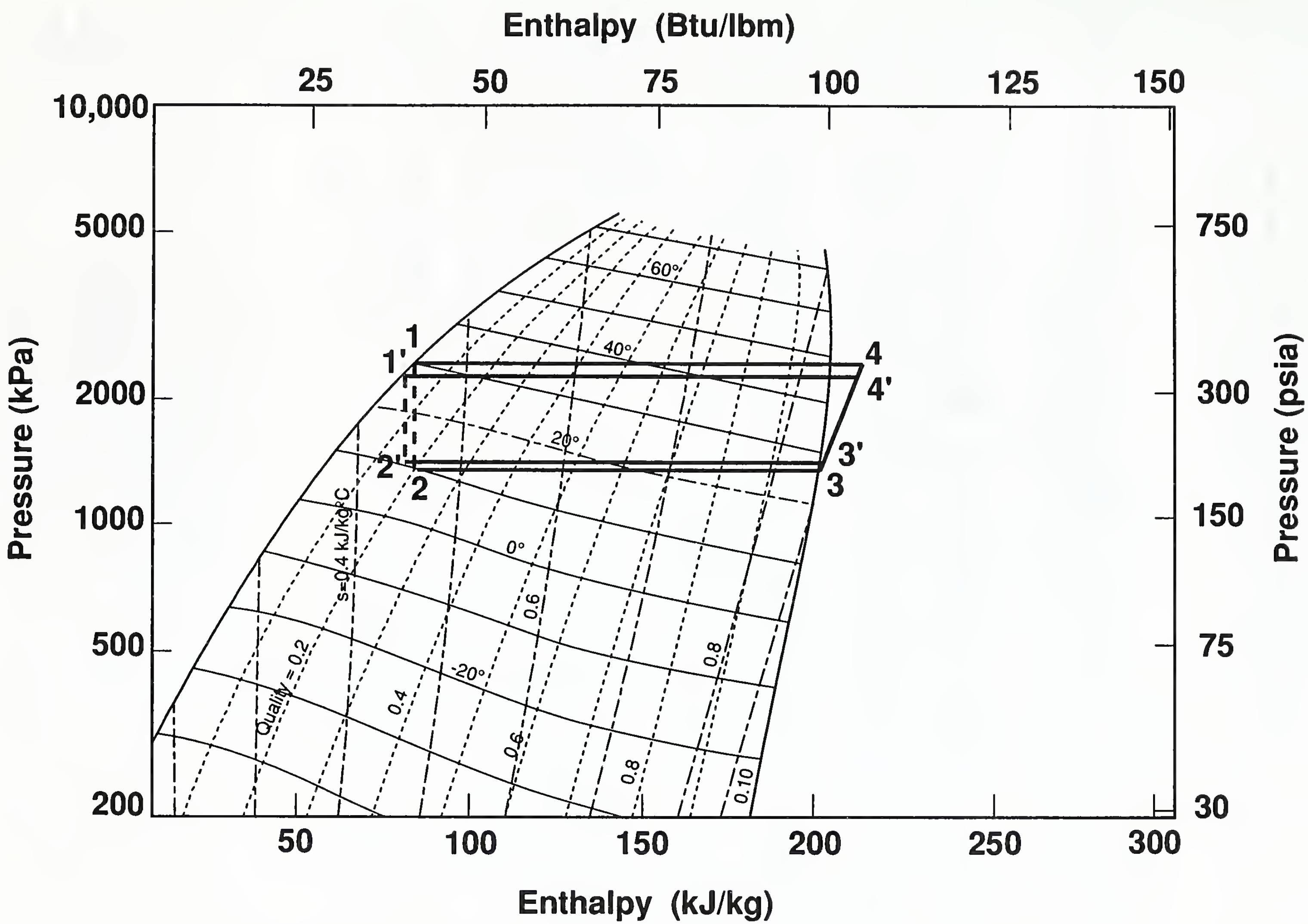


Figure 23. Effect of Heat Exchange between Condensing and Evaporating Refrigerant of 40% / 60% Mixture of R13 / R12

modification to be most effective, a pinchpoint at the condenser outlet and a convex temperature vs. enthalpy nonlinearity would be present. It should be noted that for these reported tests the subcooling criterion of a marginally clear sight glass could not be strictly adhered to because of system instabilities, i.e., cycling from a clear sight glass to a few bubbles. For this reason, it is felt that the observed efficiency improvement was most likely a combination at the benefits from increased subcooling [6] and condensation [9].

5.4 Variable Condenser Glide

The tests described in this section were performed approximately two years after the other tests described in this report, and because of piping changes, are not considered directly comparable to the other reported tests.

As was previously described, all other reported tests were performed with condenser entering and leaving temperatures of 82°F (28°C) and 117°F (47°C), respectively. Considering that an infinite heat sink is typically available on the condenser side, the only penalty to be incurred by reducing the glide in an actual system would be increased power to the heat sink fluid pumps. Reduced evaporator glide would not be reasonable because of dehumidification criteria.

To quantify the effect of reduced condenser glide, tests were performed at the above stated 35°F (19.4°C) condenser glide and at reduced glides of 30°F (16.7°C), 25°F (13.9°C), 20°F (11.1°C), 15°F (8.3°C) and 10°F (5.6°C). All tests were performed in evaporator configuration 4 with the refrigerant mixture 65% R22/35% R114 which had been shown to give the highest efficiency.

The condenser glide was reduced by reducing the heat sink (water loop) temperature leaving the condenser and holding the heat sink entering temperature constant. This approach was felt to be more representative of an actual installation than the more theoretically pure alternative of holding the average heat sink temperature constant. Hence, the cycle efficiency plotted in figure 24, increases as glide decreases because the mean heat sink fluid temperature decreases with decreasing glide, thus, reducing the temperate lift.

The percentage point improvement from the mixture is approximately constant between a 15°F (8.3°C) and 35°F (19.4°C) glide. Below 15°F (8.3°C) the efficiency improvement drops rapidly.

Discussion

It has been shown in this study that improvements in cycle efficiency can be obtained by the following the same temperature glides in the heat exchangers on the refrigerant side as are present at the heat sink and heat source side of a heat pump or an air conditioner. This statement was first made by Lorenz in 1894, but could not be realized in the vapor compression cycle with pure component refrigerants without a change in pressure which would have been necessary for those fluids to perform a temperature change in the two phase region and would have cancelled the efficiency improvement. Lorenz himself suggested the use of a compressor and an expander as heat exchangers, but

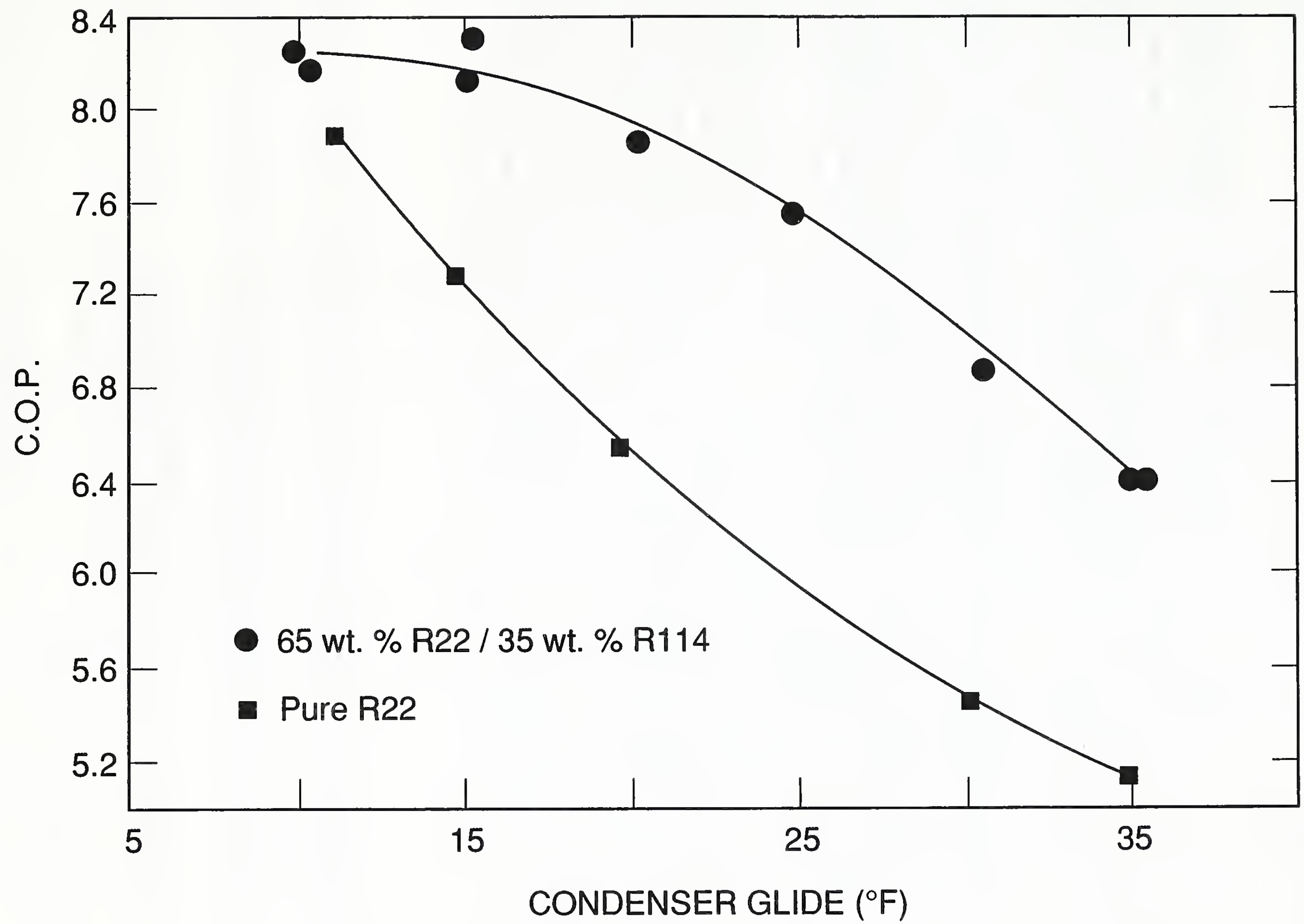


Figure 24. Effect of Reducing Condenser Heat Sink Fluid Outlet Temperature (Reduced Condenser Glide).

concluded that such a system would be too complex for practical use.

However, by using nonazeotropic refrigerant mixtures with their changing temperatures at constant pressures in the two phase region one can obtain a temperature glide on the refrigerant side in a conventional heat exchanger. A nonazeotropic mixture at the right composition can match the temperature glide of the heat exchange fluid in a counter flow heat exchanger.

This work shows that special care must be taken in the heat exchanger design to suit the requirements of the nonazeotropic mixtures. This is quite possibly one of the reasons why several earlier experimental studies failed in showing the projected improvements. Because pure refrigerants quickly encounter a pinch point as heat exchanger area is increased, mixtures can take much better advantage of increased areas. For instance, with the most effective heat exchanger configuration, the pure refrigerant could only effectively use a quarter of the heat exchanger area because of a pinch point and would have been expected to produce substantially the same efficiency with its heat exchangers reduced in size to a quarter of that employed. It must be emphasized, however, that all the comparisons between pure and mixed refrigerants presented in this report are for equal areas. A compromise has to be found for each application between increased cost for the heat exchangers and obtainable improvements in efficiency which results in cost savings during the operation of the system. Furthermore, the heat exchanger should not result in a high pressure drop since pressure drop affects the temperature glide of a mixture.

Finally, the validity of proposed improvements in cycle efficiency by the introduction of an internal heat exchange between the evaporating refrigerant and the condensed liquid has been proven. It has been shown that the amount of improvement is dependent on the shape of the vapor dome and temperature profiles and may be small for some mixtures. It was also shown that the introduction of such a heat exchanger does not significantly affect the cycle efficiency of single component systems. It can be stated that the potential for cycle improvement is present with mixtures. The employment of nonazeotropic refrigerant mixtures was shown to result in an efficiency improvement of as much as 32% over R22 when heat exchangers and temperature glides were selected properly.

Conclusion

For all heat exchanger configurations for which a full range of compositions were tested, the optimum composition of each mixture performed better than pure R22.

The best efficiency measured with the mixture R22/R114 (at a composition of 65% R22) was approximately 32% higher than the best efficiency measured with R22.

The best efficiency measured with the mixture R13/R12 (at a composition of 35% R13) was approximately 16% better than the best efficiency measured with R22.

The better the heat exchanger effectiveness the greater was the improvement which the mixtures show over pure R22.

Matching temperature glides is not the only consideration. In the evaporator configurations which had poorer heat exchanger effectiveness, the gliding temperature effect of mixtures was less important to cycle efficiency than other refrigerant properties. Hence, the optimum efficiency occurred at a composition deviating from that which produced a glide matching the heat source fluid in the direction of the component which had the best performance for other reasons (thermodynamic properties, compressor efficiency, pressure drop. etc.)

Pressure drop in the evaporator reduced temperature glide and hence reduced the cycle efficiency improvement by mixtures.

Nonlinearity of temperature with enthalpy was observed for the two mixtures tested. The R22/R114 mixture exhibited a slightly concave temperature profile through the evaporator and the R13/R12 mixture a slightly convex one. This observation is in conformance with predictions of the NIST equation of state [10] for these mixtures.

Intracycle heat exchange between the condensed liquid and evaporating refrigerant streams benefitted the R13/R12 mixture but not the R22/R114 mixture.

Intracycle heat exchange between the condensed liquid and evaporating refrigerant streams is expected to be beneficial for those mixtures which show a substantial loss of potential temperature glide as result of flashing through the expansion device.

Efficiency improvement by use of mixtures does not imply loss of capacity as a trade off.

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Appendix

Summarized Test Data

Configuration 1: Refrigerant Outside, Water In Center, Heat Exchange in Rectangular Passages, R22/R114

Test Number	1	2	3	4	5
Refrigerant Temp., °F:					
Entering Evap.	51.1	50.8	38.2	38.4	39.6
Middle of Evap.	50.6	50.4	53.7	54.5	52.8
Leaving Evap.	51.8	51.5	63.6	64.2	62.1
Entering Comp.	57.7	57.0	68.5	68.1	66.4
Entering Cond.	153.2	153.7	149.2	147.7	149.2
Middle of Cond.	112.4	112.6	109.1	108.7	108.9
Leaving Cond.	100.9	100.2	87.1	86.0	88.2
Before Exp. Valve	98.5	55.2	48.9	49.3	87.0
Water Temp., °F:					
Entering Evap.	79.7	79.9	80.0	80.5	79.5
Middle of Evap.	65.5	65.4	72.9	73.4	70.7
Leaving Evap.	55.6	55.3	54.8	55.3	54.6
Entering Cond.	81.8	82.4	81.9	81.9	82.2
Middle of Cond.	112.7	112.9	103.5	102.8	103.2
Leaving Cond.	117.2	117.3	117.5	117.4	117.0
Pressure, psia.:					
Leaving Evap.	99.3	99.0	59.1	58.9	58.5
Comp. Suction	97.6	97.2	55.5	55.4	55.0
Comp. Discharge	256.5	258.1	156.6	155.3	158.4
Leaving Cond.	251.2	251.6	151.1	149.5	152.5
Evap. Capacity kBtu/h:					
$\dot{m} * cp * \Delta T$	----	----	----	----	----
Electric Power	12.48	12.14	12.14	12.06	11.92
Calorimeter	12.96	12.71	12.55	12.46	12.37
Cond. Capacity, kBtuh:					
$\dot{m} * cp * \Delta T$	----	----	----	----	----
Electric Power	14.51	14.62	14.22	14.20	14.11
Comp. Speed, rpm	502	502	766	767	776
Comp. Power, kBtu/h	2.70	2.68	2.80	2.77	2.86
COP	4.81	4.73	4.49	4.50	4.33
Composition , % R22	100	100	58.6	62.9	52.1
Heat Exchanger	No	Yes	Yes	Yes	No

Configuration 2: Refrigerant in Rectangular Passages, Water in Center, Heat Exchange Outside, R22/R114

Test Number	6	7	8	9	10	11	12	13	14	15
Refrigerant Temp., ^{o}F:										
Entering Evap.	52.6	53.8	52.8	53.5	53.8	54.2	53.7	54.5	54.9	54.7
Middle of Evap.	56.9	56.8	58.3	57.2	56.8	56.9	56.6	57.2	57.1	56.6
Leaving Evap.	68.9	66.8	69.0	68.6	68.1	67.1	67.2	66.9	65.3	62.1
Entering Comp.	72.3	70.8	74.0	72.8	71.8	71.2	71.2	70.7	69.2	66.4
Entering Cond.	145.4	148.9	146.5	148.8	144.0	146.0	143.5	143.1	142.1	143.8
Middle of Cond.	107.3	108.1	102.6	108.6	109.7	109.7	110.9	111.5	111.4	112.9
Leaving Cond.	87.8	86.0	84.4	83.0	83.8	83.2	85.7	87.8	92.8	91.0
Before Exp. Valve	56.2	84.6	56.6	56.5	56.7	81.9	56.5	57.2	57.3	56.9
Water Temp., ^{o}F:										
Entering Evap.	79.9	80.2	80.2	80.1	79.8	79.8	79.6	79.6	79.6	79.8
Middle of Evap.	61.4	60.2	63.0	60.9	60.3	60.1	59.6	59.6	59.1	57.9
Leaving Evap.	54.6	55.0	55.2	55.0	55.0	55.1	54.7	55.3	55.2	54.6
Entering Cond.	82.1	81.9	82.0	82.0	82.0	82.2	82.0	82.0	82.1	82.1
Middle of Cond.	102.2	102.4	97.3	102.7	104.6	104.1	106.6	108.1	109.0	111.1
Leaving Cond.	117.0	116.9	117.5	117.1	116.9	117.2	117.2	117.0	117.0	117.0
Pressure, psia.:										
Leaving Evap.	65.8	63.0	60.9	69.3	74.2	71.8	77.2	80.6	85.0	88.4
Comp. Suction	61.6	59.1	56.5	65.3	70.3	68.3	73.3	77.0	80.9	84.6
Comp. Discharge	155.1	156.5	146.7	164.4	171.7	170.9	178.2	182.2	191.7	207.1
Leaving Cond.	148.8	149.8	140.2	158.3	166.0	165.6	172.3	176.5	186.2	201.3
Evap. Capacity, kBtu/h:										
$\dot{m} * cp * \Delta T$	11.14	11.52	11.55	11.62	11.65	11.33	11.63	11.26	11.01	11.54
Electric Power	12.15	12.36	12.24	12.21	11.98	11.77	11.96	11.67	11.85	12.22
Calorimeter	12.58	12.94	12.82	12.81	12.51	12.27	12.46	12.16	12.31	12.64
Cond. Capacity, kBtuh:										
$\dot{m} * cp * \Delta T$	----	----	----	----	----	----	----	----	14.04	14.67
Electric Power	13.85	14.63	14.16	14.53	14.03	13.64	13.83	13.48	13.63	14.18
Comp. Speed, rpm	669	723	721	650	592	601	568	551	534	529
Comp. Power, kBtu/h	2.51	2.71	2.54	2.53	2.27	2.30	2.24	2.18	2.14	2.31
COP	5.02	4.77	5.05	5.06	5.52	5.33	5.57	5.58	5.76	5.48
Composition , % R22	57	53	49	61	65	59	65	66	71	78
Heat Exchanger	Yes	No	Yes	Yes	Yes	No	Yes	Yes	Yes	Yes

Configuration 2: Refrigerant in Rectangular Passages, Water in Center, Heat Exchange Outside, R22/R114 cont'd.

Test Number	16	17	18	19	20	21	22
Refrigerant Temp., ^{o}F:							
Entering Evap.	54.7	52.8	54.5	52.2	55.3	52.8	54.8
Middle of Evap.	56.5	57.0	56.4	57.1	56.5	57.1	56.3
Leaving Evap.	63.1	58.8	57.5	53.3	51.8	56.2	54.8
Entering Comp.	68.7	64.0	62.6	58.7	57.6	61.8	60.4
Entering Cond.	145.3	145.6	145.5	150.0	150.6	148.4	147.7
Middle of Cond.	109.2	113.2	113.1	112.3	112.4	112.8	113.1
Leaving Cond.	82.2	88.7	85.4	100.8	98.6	92.7	89.4
Before Exp. Valve	81.2	56.4	84.1	56.1	96.9	56.5	87.6
Water Temp., ^{o}F:							
Entering Evap.	79.8	79.9	80.0	79.9	79.8	79.6	79.5
Middle of Evap.	58.4	57.9	57.4	57.4	56.7	57.6	56.9
Leaving Evap.	55.0	54.3	54.4	54.0	54.7	54.2	54.5
Entering Cond.	82.1	81.9	81.8	82.2	81.7	82.0	81.9
Middle of Cond.	104.0	111.9	111.6	112.4	112.4	112.4	112.6
Leaving Cond.	117.1	116.9	117.2	116.8	116.9	117.1	117.3
Pressure, psia.:							
Leaving Evap.	83.1	94.6	92.6	104.0	101.9	98.5	96.4
Comp. Suction	79.0	90.5	88.9	100.4	98.2	94.8	93.0
Comp. Discharge	196.9	225.6	225.6	257.4	257.8	237.6	239.1
Leaving Cond.	191.9	219.4	219.7	251.0	251.4	231.9	233.2
Evap. Capacity, kBtu/h:							
$\dot{m} * cp * \Delta T$	11.35	11.75	11.80	11.78	11.63	11.06	----
Electric Power	12.04	12.26	12.54	12.56	12.14	12.26	12.21
Calorimeter	12.61	12.81	12.96	13.01	12.59	12.69	12.62
Cond. Capacity, kBtuh:							
$\dot{m} * cp * \Delta T$	14.39	15.11	14.90	15.21	15.07	14.77	14.51
Electric Power	13.93	14.64	14.49	14.66	14.98	14.51	14.32
Comp. Speed, rpm	521	489	490	484	483	481	482
Comp. Power, kBtu/h	2.18	2.25	2.25	2.53	2.50	2.34	2.33
COP	5.78	5.69	5.75	5.15	5.03	5.43	5.41
Composition , % R22	73	85	86	100	100	91	91
Heat Exchanger	No	Yes	No	Yes	No	Yes	No

Configuration 2: Refrigerant in Rectangular Passages, Water in Center, Heat Exchange Outside, R13/R12

Test Number	23	24	25	26	27	28	29	30	31	32
Refrigerant Temp., ^{o}F:										
Entering Evap.	46.5	49.8	48.2	51.8	47.0	46.6	50.0	46.0	49.6	55.6
Middle of Evap.	64.7	60.9	61.9	59.8	62.6	64.6	61.6	64.2	62.3	56.7
Leaving Evap.	70.6	60.8	64.7	64.5	67.7	70.9	67.6	71.8	70.1	42.2
Entering Comp.	74.0	63.5	66.8	68.9	69.5	73.0	71.0	74.1	71.9	49.1
Entering Cond.	145.9	146.0	146.5	153.4	147.0	145.4	149.0	146.2	148.2	153.0
Middle of Cond.	94.7	110.8	104.2	106.7	106.5	105.0	107.2	107.4	102.2	112.2
Leaving Cond.	83.4	82.4	82.3	82.0	82.6	84.5	83.6	84.9	84.2	82.3
Before Exp. Valve	57.2	56.9	56.8	81.2	56.7	57.2	82.9	56.4	83.8	57.2
Water Temp., ^{o}F:										
Entering Evap.	79.5	79.3	79.2	79.7	79.7	79.6	79.3	79.8	79.6	79.7
Middle of Evap.	66.8	62.1	63.5	61.5	64.8	67.1	63.6	67.2	64.8	55.9
Leaving Evap.	55.1	55.0	54.8	55.1	54.8	55.1	54.8	54.3	54.6	55.2
Entering Cond.	82.3	81.8	82.1	82.0	81.8	82.3	83.6	84.9	82.3	82.0
Middle of Cond.	87.4	99.4	92.8	93.4	95.8	94.8	107.2	107.4	92.4	105.9
Leaving Cond.	117.2	117.2	116.9	117.1	117.2	116.7	149.0	142.2	116.8	117.2
Pressure, psia.:										
Leaving Evap.	130.4	88.1	101.0	93.3	115.3	135.0	123.2	148.0	145.1	53.2
Comp. Suction	127.2	83.9	97.2	89.6	111.5	131.4	120.3	144.6	142.2	47.3
Comp. Discharge	287.3	214.2	237.9	236.6	263.0	299.9	292.1	324.0	332.9	162.3
Leaving Cond.	277.8	207.7	230.5	230.5	255.1	290.3	282.9	313.5	322.8	157.6
Evap. Capacity kBtu/h:										
$\dot{m} * cp * \Delta T$	----	----	----	----	----	----	----	----	----	14.20
Electric Power	16.02	15.87	15.84	16.06	16.28	15.96	16.04	16.69	16.25	14.97
Calorimeter	16.50	16.34	16.30	16.49	16.76	16.46	16.54	17.18	16.80	15.64
Cond. Capacity, kBtuh:										
$\dot{m} * cp * \Delta T$	19.33	20.02	19.95	20.06	20.17	19.44	19.94	20.13	20.13	----
Electric Power	18.82	19.57	19.41	19.56	19.76	19.06	19.63	19.80	19.74	25.38
Comp. Speed, rpm	501	720	628	674	579	493	534	479	485	1178
Comp. Power, kBtu/h	2.97	3.50	3.36	3.59	3.33	2.99	3.25	3.12	3.25	4.18
COP	5.55	4.67	4.86	4.59	5.03	5.51	5.08	5.51	5.17	3.74
Composition , % R13	40	19	22	24	29	37	35	43	44	0
Heat Exchanger	Yes	Yes	Yes	No	Yes	Yes	No	Yes	No	Yes

Configuration 2: Refrigerant in Rectangular Passages, Water in Center, Heat Exchange Outside, R13/R12 cont'd.

Test Number	33
Refrigerant Temp., ^oF:	
Entering Evap.	56.6
Middle of Evap.	56.4
Leaving Evap.	41.8
Entering Comp.	52.7
Entering Cond.	156.8
Middle of Cond.	111.3
Leaving Cond.	82.3
Before Exp. Valve	81.3
Water Temp., ^oF:	
Entering Evap.	79.4
Middle of Evap.	55.3
Leaving Evap.	55.4
Entering Cond.	81.9
Middle of Cond.	100.8
Leaving Cond.	116.4
Pressure, psia.:	
Leaving Evap.	52.2
Comp. Suction	46.3
Comp. Discharge	160.6
Leaving Cond.	156.0
Evap. Capacity kBtu/h:	
$\dot{m} * cp * \Delta T$	14.15
Electric Power	14.67
Calorimeter	15.32
Cond. Capacity, kBtuh:	
$\dot{m} * cp * \Delta T$	----
Electric Power	24.81
Comp. Speed, rpm	1170
Comp. Power, kBtu/h	4.06
COP	3.77
Composition , % R13	0
Heat Exchanger	No

Configuration 2: Refrigerant in Rectangular Passages, Water in Center, Heat Exchange Outside, Pure Refrigerants in Parallel Flow in Evaporator

Test Number	34	35	36	37
Refrigerant Temp., ^{o}F:				
Entering Evap.	71.3	72.5	57.7	62.1
Middle of Evap.	67.4	67.2	61.7	60.4
Leaving Evap.	46.3	45.6	53.3	54.3
Entering Comp.	50.4	49.5	57.4	59.1
Entering Cond.	150.7	149.6	151.8	149.7
Middle of Cond.	111.7	112.3	112.7	112.8
Leaving Cond.	96.2	94.5	101.5	97.2
Before Exp. Valve	71.0	92.1	57.0	95.0
Water Temp., ^{o}F:				
Entering Evap.	80.1	79.5	79.8	80.1
Middle of Evap.	69.8	69.5	62.4	61.1
Leaving Evap.	55.1	54.6	54.9	55.4
Entering Cond.	82.0	81.9	81.8	82.1
Middle of Cond.	111.9	112.4	112.9	113.1
Leaving Cond.	116.6	117.1	117.4	117.0
Pressure, psia.:				
Leaving Evap.	71.0	71.1	106.3	103.7
Comp. Suction	51.3	50.8	99.4	101.8
Comp. Discharge	160.5	161.7	258.2	258.6
Leaving Cond.	155.4	156.5	251.8	252.4
Evap. Capacity kBtu/h:				
$\dot{m} * cp * \Delta T$	----	----	----	----
Electric Power	15.06	14.97	15.04	13.00
Calorimeter	15.70	15.60	15.71	13.62
Cond. Capacity, kBtuh:				
$\dot{m} * cp * \Delta T$	20.15	20.24	19.12	16.46
Electric Power	19.12	19.16	18.26	15.96
Comp. Speed, rpm	1163	1162	586	488
Comp. Power, kBtu/h	4.27	4.26	3.14	2.58
COP	3.67	3.66	5.00	5.28
Refrigerant	R12	R12	R22	R22
Heat Exchanger	Yes	No	Yes	No

Configuration 3: Refrigerant in Center, Water Outside, Heat Exchange in Rectangular Passages, R22/R14

Test Number	38	39	40	41	42	43	44	45	46	47
Refrigerant Temp., ^{o}F:										
Entering Evap.	47.6	47.3	49.3	46.7	48.2	45.9	46.9	44.4	45.4	40.8
Middle of Evap.	50.7	49.3	49.8	48.2	48.6	47.7	48.1	47.2	47.1	46.8
Leaving Evap.	51.8	53.4	53.1	57.2	56.8	60.4	59.5	64.0	60.3	65.9
Entering Comp.	57.0	58.7	58.2	74.7	61.7	64.6	64.0	68.9	64.7	69.6
Entering Cond.	151.1	148.8	148.4	154.8	147.4	144.0	144.8	145.8	146.4	147.2
Middle of Cond.	113.2	113.4	113.2	111.7	113.0	111.0	111.0	107.5	109.1	108.0
Leaving Cond.	97.8	92.4	87.9	92.8	90.9	89.6	87.0	86.0	87.5	86.0
Before Exp. Valve	52.9	52.5	86.3	52.3	89.3	51.9	86.0	51.1	86.3	49.0
Water Temp., ^{o}F:										
Entering Evap.	79.5	79.5	79.5	79.6	79.5	79.5	79.5	79.5	79.5	79.9
Middle of Evap.	64.4	64.3	64.6	65.6	65.5	66.7	66.5	67.6	66.4	69.4
Leaving Evap.	55.0	54.6	54.9	54.8	54.8	55.0	54.8	54.9	54.2	54.6
Entering Cond.	82.0	82.2	82.3	82.1	82.1	82.2	82.0	82.1	82.0	82.2
Middle of Cond.	113.4	113.0	112.9	110.1	111.6	108.7	108.7	104.4	106.5	102.8
Leaving Cond.	117.5	117.4	117.1	117.2	116.9	117.2	117.1	117.2	117.5	117.3
Pressure, psia.:										
Leaving Evap.	101.6	94.8	95.1	87.2	87.9	81.6	81.7	76.3	77.3	66.1
Comp. Suction	98.2	91.0	91.3	83.0	83.7	77.7	77.6	72.1	72.6	60.2
Comp. Discharge	261.0	241.9	242.6	214.4	220.0	198.7	200.5	182.4	190.0	163.0
Leaving Cond.	254.0	235.4	236.3	208.8	214.0	193.1	195.0	176.9	184.4	155.9
Evap. Capacity kBtu/h:										
$\dot{m} * cp * \Delta T$	13.49	13.81	13.58	13.73	13.72	13.62	13.64	13.54	13.78	13.79
Electric Power	12.96	13.24	13.16	13.31	13.18	13.30	13.31	13.24	13.53	13.56
Calorimeter	13.20	13.47	13.38	13.55	13.40	13.48	13.47	13.45	13.82	13.85
Cond. Capacity, kBtuh:										
$\dot{m} * cp * \Delta T$	16.31	16.58	15.92	16.19	16.32	15.95	16.23	16.12	16.64	16.60
Electric Power	16.12	16.07	15.60	16.09	16.09	15.47	15.88	16.04	16.40	16.12
Comp. Speed, rpm	505	535	515	560	561	587	586	618	634	741
Comp. Power, kBtu/h	2.72	2.66	2.58	2.56	2.58	2.49	2.49	2.50	2.67	2.82
COP	4.85	5.06	5.19	5.29	5.18	5.42	5.41	5.37	5.18	4.91
Composition, % R22	100	93	92	82	84	75	77	68	73	57
Heat Exchanger	Yes	Yes	No	Yes	No	Yes	No	Yes	No	Yes

Configuration 3: Refrigerant in Center, Water Outside,
Heat Exchange in Rectangular Passages, R22/R14 cont'd.

Test Number	48	49	50	51	52	53	54	55	56	57
Refrigerant Temp., ^{o}F:										
Entering Evap.	40.6	47.3	48.2	48.8	49.2	49.3	48.5	48.7	46.6	46.2
Middle of Evap.	48.7	47.9	48.6	48.3	48.6	49.0	48.1	48.1	47.4	49.5
Leaving Evap.	64.9	57.2	56.5	54.2	54.9	55.6	54.0	55.1	58.8	51.3
Entering Comp.	69.3	63.0	61.6	59.3	60.2	60.7	58.7	60.2	64.0	56.7
Entering Cond.	149.4	147.4	145.9	147.2	147.7	148.8	147.3	149.8	147.9	151.5
Middle of Cond.	107.7	112.9	113.1	112.8	112.4	112.9	113.3	113.2	112.5	112.5
Leaving Cond.	82.9	91.4	89.6	97.3	94.8	99.3	82.2	92.6	90.3	90.3
Before Exp. Valve	81.8	52.6	87.9	53.4	93.4	96.0	53.3	90.2	52.1	52.3
Water Temp., ^{o}F:										
Entering Evap.	79.5	79.4	79.2	79.4	79.2	79.1	79.2	79.3	79.1	78.9
Middle of Evap.	69.5	65.0	64.7	64.4	64.2	64.4	65.1	64.7	65.8	64.0
Leaving Evap.	54.7	54.6	54.6	55.0	54.9	55.0	55.2	54.6	54.7	54.8
Entering Cond.	82.0	82.1	82.0	82.1	82.1	82.2	82.2	82.1	82.1	82.2
Middle of Cond.	100.0	111.3	111.7	112.4	112.0	112.6	112.8	112.7	110.6	112.6
Leaving Cond.	117.4	117.1	116.9	117.2	116.5	117.1	117.3	117.4	116.9	116.7
Pressure, psia.:										
Leaving Evap.	60.8	86.6	87.7	91.9	92.1	92.7	91.5	91.4	83.9	99.8
Comp. Suction	54.9	82.4	83.6	88.0	88.4	88.1	87.4	87.3	79.6	96.0
Comp. Discharge	159.1	214.9	218.3	232.6	232.8	234.7	232.9	235.2	207.2	257.1
Leaving Cond.	153.1	209.4	212.7	226.8	226.7	228.5	226.7	229.1	201.3	250.6
Evap. Capacity kBtu/h:										
$\dot{m} * cp * \Delta T$	13.57	13.09	13.06	13.11	13.13	12.99	13.29	13.54	13.42	13.27
Electric Power	13.27	12.92	12.82	12.77	12.87	12.54	13.06	13.32	13.39	12.88
Calorimeter	13.57	13.19	13.11	13.08	13.17	12.84	13.35	13.62	13.69	13.16
Cond. Capacity, kBtuh:										
$\dot{m} * cp * \Delta T$	16.50	16.10	16.01	16.21	15.99	15.67	16.15	16.42	16.34	----
Electric Power	16.12	15.78	15.64	15.72	15.35	15.24	15.91	16.17	15.83	15.40
Comp. Speed, rpm	787	555	543	545	532	531	552	552	584	502
Comp. Power, kBtu/h	2.90	2.52	2.47	2.62	2.55	2.56	2.65	2.67	2.61	2.64
COP	4.69	5.23	5.30	4.99	5.16	5.01	5.03	5.10	5.24	4.99
Composition, % R22	51	82	83	90	89	89	89	90	79	100
Heat Exchanger	No	Yes	No	Yes	No	No	Yes	No	Yes	Yes

Configuration 3: Refrigerant in Center, Water Outside, Heat Exchange in Rectangular Passages, R22/R14 cont'd.

Test Number	58	59	60	61	62	63
Refrigerant Temp., ^{o}F:						
Entering Evap.	46.1	5.15	34.2	38.9	35.5	53.4
Middle of Evap.	49.2	50.9	49.8	48.3	49.6	53.0
Leaving Evap.	51.2	48.1	64.0	67.5	64.9	51.8
Entering Comp.	57.1	60.1	66.7	71.9	68.1	60.2
Entering Cond.	153.0	175.1	146.1	148.0	148.1	136.8
Middle of Cond.	113.2	109.3	102.6	104.8	85.1	113.5
Leaving Cond.	88.2	89.4	83.0	82.7	81.8	104.4
Before Exp. Valve	52.1	55.4	46.7	49.0	46.9	56.7
Water Temp., ^{o}F:						
Entering Evap.	78.9	80.1	79.9	80.5	80.1	79.6
Middle of Evap.	63.9	62.0	70.5	71.2	70.7	61.6
Leaving Evap.	54.6	54.5	54.8	55.7	54.6	55.6
Entering Cond.	82.0	82.2	82.2	82.4	82.1	82.4
Middle of Cond.	113.2	108.7	93.8	97.5	82.2	113.6
Leaving Cond.	117.6	117.0	117.3	117.4	117.1	117.4
Pressure, psia.:						
Leaving Evap.	99.3	18.6	44.9	59.7	49.4	19.7
Comp. Suction	95.8	12.0	37.1	53.4	42.8	15.9
Comp. Discharge	259.5	59.3	120.8	150.9	136.1	----
Leaving Cond.	252.9	47.3	113.8	145.3	129.5	----
Evap. Capacity kBtu/h:						
$\dot{m} * cp * \Delta T$	13.18	8.11	14.03	13.86	14.16	5.55
Electric Power	12.95	8.46	13.81	13.34	13.86	6.38
Calorimeter	13.21	8.79	14.19	13.71	14.23	6.63
Cond. Capacity, kBtuh:						
$\dot{m} * cp * \Delta T$	----	13.30	17.35	16.80	17.40	7.92
Electric Power	15.76	12.95	16.48	16.02	16.55	8.71
Comp. Speed, rpm	503	2805	1202	832	1056	1395
Comp. Power, kBtu/h	2.64	5.03	3.30	2.99	3.22	2.34
COP	5.00	1.75	4.31	4.59	4.41	2.84
Composition, % R22	100	0	24	49	38	0
Heat Exchanger	Yes	Yes	Yes	Yes	Yes	Yes

Configuration 3: Refrigerant in Center, Water Outside, Heat Exchange in Rectangular Passages, R13/R12

Test Number	64	65	66	67	68	69	70	71
Refrigerant Temp., °F:								
Entering Evap.	29.5	29.5	29.9	30.0	30.1	31.2	31.6	32.1
Middle of Evap.	50.8	50.9	51.0	51.4	51.3	52.2	52.2	51.4
Leaving Evap.	65.5	64.8	63.4	64.5	64.2	64.2	63.7	62.7
Entering Comp.	68.0	67.5	66.7	66.9	66.9	67.1	66.7	65.8
Entering Cond.	147.9	147.8	148.1	145.5	145.5	143.4	142.9	145.6
Middle of Cond.	101.5	99.7	99.0	99.2	99.3	99.4	94.7	105.2
Leaving Cond.	83.8	82.7	82.2	82.7	82.8	82.9	82.5	83.8
Before Exp. Valve	43.8	43.7	44.2	44.3	44.3	45.7	45.6	45.5
Water Temp., °F:								
Entering Evap.	79.9	79.6	80.0	79.9	79.7	79.7	80.0	79.6
Middle of Evap.	71.6	71.1	70.5	70.7	70.6	70.6	70.5	69.8
Leaving Evap.	54.7	54.5	54.6	54.6	54.7	55.3	55.3	55.0
Entering Cond.	82.2	82.1	82.2	82.2	82.1	82.2	82.1	82.2
Middle of Cond.	91.5	89.5	88.5	89.3	89.2	90.5	86.5	92.4
Leaving Cond.	117.1	117.2	117.3	117.0	117.0	117.2	117.1	117.2
Pressure, psia.:								
Leaving Evap.	138.5	132.4	114.8	120.0	119.8	113.6	109.4	103.3
Comp. Suction	132.5	126.9	109.9	115.0	114.6	108.9	104.9	98.3
Comp. Discharge	328.8	320.0	283.9	289.7	289.2	269.2	259.9	253.0
Leaving Cond.	318.2	309.6	274.4	280.1	279.5	260.2	251.2	154.7
Evap. Capacity kBtu/h:								
$\dot{m} * cp * \Delta T$	15.70	15.17	14.94	14.42	14.28	13.94	14.07	14.05
Electric Power	15.26	14.54	14.24	13.86	13.66	13.52	13.42	13.61
Calorimeter	15.49	14.79	14.47	14.14	13.92	13.78	13.67	13.87
Cond. Capacity, kBtuh:								
$\dot{m} * cp * \Delta T$	18.94	18.41	18.06	17.33	17.34	16.86	17.07	17.22
Electric Power	17.90	17.34	17.09	16.33	16.51	16.10	16.39	16.66
Comp. Speed, rpm	480	478	522	484	485	487	505	537
Comp. Power, kBtu/h	3.17	3.04	2.99	2.81	2.80	2.66	2.67	2.79
COP	4.89	4.86	4.83	5.03	4.97	5.18	5.12	4.96
Composition, % R13	46	44	34	37	37	33	31	28
Heat Exchanger	Yes	Yes	Yes	Yes	Yes	Yes	Yes	Yes

Configuration 4: Refrigerant in Center, Water in Rectangular Passages, Heat Exchange Outside, R22/R114

Test Number	72	73	74	75	76	77	78	79	80	81
Refrigerant Temp., ^{o}F:										
Entering Evap.	55.2	53.8	55.1	54.1	55.2	54.7	55.2	53.9	54.9	53.7
Middle of Evap.	54.8	54.5	54.5	54.8	54.7	55.3	54.6	54.6	54.4	54.2
Leaving Evap.	56.3	59.9	59.7	61.8	60.6	64.3	63.1	65.5	63.9	68.1
Entering Comp.	60.3	63.3	63.7	65.2	64.1	67.5	66.7	68.7	67.0	71.1
Entering Cond.	149.9	144.5	145.0	142.1	143.9	141.4	143.3	141.2	141.4	139.2
Middle of Cond.	112.5	113.2	112.7	113.1	112.9	112.7	112.9	113.2	113.3	112.2
Leaving Cond.	101.2	93.0	93.6	90.8	95.9	94.0	93.2	89.4	88.6	89.8
Before Exp. Valve	56.1	57.6	92.1	58.0	93.5	58.7	91.1	57.9	87.1	57.9
Water Temp., ^{o}F:										
Entering Evap.	80.4	80.1	79.8	80.2	79.7	79.9	80.4	79.7	79.7	79.7
Middle of Evap.	56.5	57.1	56.4	57.6	56.8	58.7	57.0	57.8	57.0	58.2
Leaving Evap.	55.1	54.6	54.8	55.0	54.9	55.5	55.0	54.7	54.7	54.6
Entering Cond.	82.1	81.9	81.9	82.0	82.1	81.9	82.0	81.9	81.9	82.1
Middle of Cond.	112.9	113.1	112.6	112.6	112.6	111.8	112.1	111.7	112.0	110.4
Leaving Cond.	117.1	117.4	116.9	116.9	117.1	117.1	117.4	117.0	117.0	117.4
Pressure, psia.:										
Leaving Evap.	107.7	103.6	103.6	102.4	102.6	100.9	100.1	98.1	98.0	94.8
Comp. Suction	104.8	101.1	100.6	99.3	99.6	97.8	96.7	94.8	94.7	91.7
Comp. Discharge	257.6	238.5	239.9	231.0	235.5	222.3	225.5	216.0	220.1	205.0
Leaving Cond.	250.5	232.0	233.2	224.2	228.6	215.5	218.6	209.3	213.7	198.4
Evap. Capacity kBtu/h:										
$\dot{m} * cp * \Delta T$	13.90	14.27	13.94	14.09	13.85	13.61	14.18	13.90	13.93	13.98
Electric Power	13.49	13.81	13.45	13.63	13.25	13.16	13.52	13.38	13.37	13.38
Calorimeter	14.17	14.51	14.14	14.31	13.98	13.88	14.26	14.05	14.05	14.01
Cond. Capacity, kBtuh:										
$\dot{m} * cp * \Delta T$	17.00	17.07	16.70	16.68	16.41	16.17	16.89	16.56	16.40	16.54
Electric Power	16.19	16.44	15.89	16.01	15.69	15.59	16.14	16.00	15.83	16.01
Comp. Speed, rpm	504	505	497	498	499	490	512	500	499	513
Comp. Power, kBtu/h	2.74	2.48	2.43	2.34	2.43	2.23	2.36	2.20	2.24	2.16
COP	5.18	5.86	5.83	6.10	5.76	6.22	6.04	6.37	6.27	6.50
Composition, % R22	100	90	91	87	89	85	85	81	84	76
Heat Exchanger	Yes	Yes	No	Yes	No	Yes	No	Yes	No	Yes

Configuration 4: Refrigerant in Center, Water in Rectangular Passages, Heat Exchange Outside, R22/R114 cont'd.

Test Number	82	83	84	85	86	87	88	89	90	91
Refrigerant Temp., ^{o}F:										
Entering Evap.	54.9	53.1	54.9	53.9	54.0	53.6	54.0	51.8	52.2	48.3
Middle of Evap.	54.6	54.6	54.7	55.3	55.3	56.5	55.4	55.1	57.2	62.3
Leaving Evap.	66.5	70.9	68.4	72.0	72.9	74.6	73.0	73.7	75.5	77.3
Entering Comp.	69.4	73.9	70.8	74.3	75.7	82.1	75.1	75.3	77.3	78.6
Entering Cond.	138.9	138.1	139.3	136.3	138.1	143.6	139.3	137.5	137.0	140.4
Middle of Cond.	111.8	109.3	111.2	109.0	107.1	111.0	110.0	107.0	104.3	101.9
Leaving Cond.	90.2	84.0	90.9	91.9	84.0	88.5	90.6	89.5	87.6	84.7
Before Exp. Valve	89.0	58.5	89.4	58.9	83.0	59.6	89.8	59.0	59.4	60.9
Water Temp., ^{o}F:										
Entering Evap.	79.6	80.1	80.2	79.8	79.6	80.3	79.6	80.0	80.1	80.2
Middle of Evap.	57.7	60.4	58.2	60.7	61.3	63.4	60.9	63.2	65.4	70.4
Leaving Evap.	54.9	55.0	55.0	55.3	54.9	55.8	54.8	55.0	55.3	55.4
Entering Cond.	81.9	82.0	81.9	82.0	81.8	82.0	81.9	82.0	82.3	82.4
Middle of Cond.	110.1	106.2	109.1	106.4	103.3	106.8	106.9	103.4	100.7	96.3
Leaving Cond.	116.6	117.3	117.4	116.7	117.3	117.1	117.2	116.8	116.6	116.6
Pressure, psia.:										
Leaving Evap.	95.8	90.9	93.6	90.5	87.8	87.3	86.7	82.7	81.6	70.0
Comp. Suction	92.1	87.2	90.6	87.1	83.5	83.2	82.8	78.6	77.2	62.9
Comp. Discharge	207.2	190.5	201.1	185.0	181.0	179.7	181.8	169.6	163.1	143.4
Leaving Cond.	200.7	184.3	194.9	179.1	175.0	173.1	175.2	163.1	156.7	136.6
Evap. Capacity kBtu/h:										
$\dot{m} * cp * \Delta T$	13.73	14.21	14.06	13.71	13.90	13.87	13.97	14.15	14.20	14.25
Electric Power	13.26	13.64	13.47	13.24	13.37	13.33	13.47	13.75	13.80	13.84
Calorimeter	13.94	14.34	14.12	13.09	14.03	14.02	14.14	14.41	14.42	14.46
Cond. Capacity, kBtuh:										
$\dot{m} * cp * \Delta T$	16.34	16.62	16.56	16.13	16.48	16.32	16.55	16.68	16.58	16.99
Electric Power	15.73	16.03	15.97	15.49	15.89	15.76	16.00	15.90	15.91	16.13
Comp. Speed, rpm	501	525	524	524	538	539	555	583	584	711
Comp. Power, kBtu/h	2.12	2.08	2.17	2.02	2.04	2.09	2.14	2.15	2.10	2.45
COP	6.57	6.88	6.50	6.88	6.90	6.70	6.61	6.72	6.86	5.90
Composition, % R22	79	70	76	70	66	66	66	62	60	48
Heat Exchanger	No	Yes	No	Yes	No	Yes	No	Yes	Yes	Yes

Configuration 4: Refrigerant in Center, Water in Rectangular Passages, Heat Exchange Outside, R22/R114 cont'd.

Test Number	92	93	94	95	96	97	98
Refrigerant Temp., ^oF:							
Entering Evap.	45.2	55.6	55.7	55.1	55.6	56.8	58.7
Middle of Evap.	64.3	54.9	55.0	54.9	55.4	56.6	58.4
Leaving Evap.	78.4	54.6	54.6	56.0	56.7	57.7	59.4
Entering Comp.	82.1	65.1	62.5	58.9	59.2	58.6	59.0
Entering Cond.	147.6	136.5	133.2	150.6	157.2	155.6	153.7
Middle of Cond.	102.4	113.7	113.6	112.3	112.3	112.3	111.5
Leaving Cond.	83.5	83.1	83.7	99.0	94.2	98.0	95.8
Before Exp. Valve	61.5	81.6	81.9	57.6	58.0	59.1	60.9
Water Temp., ^oF:							
Entering Evap.	80.3	80.4	80.1	79.3	85.3	94.3	105.5
Middle of Evap.	71.5	55.7	55.7	56.8	57.5	58.9	61.0
Leaving Evap.	54.5	55.0	55.1	55.1	55.6	56.8	58.7
Entering Cond.	82.4	82.4	82.1	81.9	94.2	77.2	66.9
Middle of Cond.	94.8	112.8	113.5	112.9	112.3	112.9	111.7
Leaving Cond.	117.6	117.1	116.8	117.0	157.2	118.5	119.0
Pressure, psia.:							
Leaving Evap.	58.0	20.6	20.7	108.2	108.8	111.2	114.6
Comp. Suction	51.0	17.1	17.2	104.1	104.1	103.7	104.8
Comp. Discharge	130.7	63.4	63.4	258.5	258.6	259.2	256.8
Leaving Cond.	124.0	58.2	58.0	251.0	250.9	250.8	246.6
Evap. Capacity kBtu/h:							
$\dot{m} * cp * \Delta T$	14.85	6.10	6.08	17.81	22.32	28.85	36.74
Electric Power	14.40	6.76	6.75	17.00	21.40	27.61	35.14
Calorimeter	15.01	7.03	6.98	17.55	22.00	28.14	35.66
Cond. Capacity, kBtuh:							
$\dot{m} * cp * \Delta T$	17.79	8.05	7.94	21.65	27.62	35.12	44.37
Electric Power	14.40	8.70	8.70	20.22	25.87	32.95	41.16
Comp. Speed, rpm	873	1199	1201	635	756	977	1203
Comp. Power, kBtu/h	2.87	2.02	2.02	3.62	5.02	5.75	6.82
COP	5.23	3.48	3.46	4.85	4.38	4.89	5.23
Composition, % R22	38	0	0	100	100	100	100
Heat Exchanger	Yes	No	No	Yes	Yes	Yes	Yes

Configuration 4: Refrigerant in Center, Water in Rectangular Passages, Heat Exchange Outside, R22/R114, Ethylene Glycol added to Water Loop

Test Number	99	100	101	102
Refrigerant Temp., ^{o}F:				
Entering Evap.	54.5	24.9	52.0	22.9
Middle of Evap.	54.3	24.7	54.3	23.8
Leaving Evap.	55.5	25.5	70.4	40.1
Entering Comp.	59.5	34.7	73.8	49.6
Entering Cond.	148.7	180.2	140.8	159.7
Middle of Cond.	113.0	110.1	109.6	110.8
Leaving Cond.	94.6	98.3	90.9	87.3
Before Exp. Valve	57.6	30.3	57.7	30.3
Water Temp., ^{o}F:				
Entering Evap.	79.9	49.8	80.1	49.9
Middle of Evap.	58.1	29.7	62.9	32.4
Leaving Evap.	55.0	25.4	54.8	25.0
Entering Cond.	82.3	82.2	82.3	81.9
Middle of Cond.	113.4	110.5	106.8	107.6
Leaving Cond.	117.4	117.3	116.8	117.5
Pressure, psia.:				
Leaving Evap.	106.7	63.2	87.7	52.3
Comp. Suction	103.7	58.7	82.7	47.6
Comp. Discharge	260.4	250.2	183.8	187.9
Leaving Cond.	253.3	243.6	177.0	181.1
Evap. Capacity kBtu/h:				
$\dot{m} * cp * \Delta T$	----	----	----	----
Electric Power	14.05	13.17	14.47	11.80
Calorimeter	14.15	13.23	14.61	11.69
Cond. Capacity, kBtuh:				
$\dot{m} * cp * \Delta T$	17.67	19.16	17.40	15.61
Electric Power	17.06	18.18	16.65	14.96
Comp. Speed, rpm	509	965	600	934
Comp. Power, kBtu/h	2.76	4.93	2.52	3.58
COP	5.13	2.68	5.81	3.26
Composition, % R22	100	100	66	68
Heat Exchanger	Yes	Yes	Yes	Yes

Configuration 4: Refrigerant in Center, Water in Rectangular Passages, Heat Exchange Outside, R13/R12

Test Number	103	104	105	106	107	108	109	110	111	112
Refrigerant Temp., ^{o}F:										
Entering Evap.	42.6	41.9	45.5	42.1	42.6	45.8	42.8	45.4	48.6	46.2
Middle of Evap.	66.3	65.4	63.8	64.8	64.4	62.3	63.8	61.7	60.0	60.7
Leaving Evap.	76.4	75.2	74.0	74.7	74.0	72.8	73.1	71.4	69.9	70.6
Entering Comp.	77.4	76.4	76.0	75.9	75.3	74.6	74.5	73.4	72.0	72.1
Entering Cond.	142.6	144.6	151.9	144.6	143.2	148.3	142.9	146.2	146.7	143.9
Middle of Cond.	102.7	102.5	104.8	105.6	102.2	103.7	106.2	100.6	99.8	102.0
Leaving Cond.	85.9	83.0	82.0	83.4	82.9	82.0	83.2	81.6	81.8	81.8
Before Exp. Valve	61.9	61.1	81.4	61.1	61.2	81.4	60.9	60.2	81.1	59.7
Water Temp., ^{o}F:										
Entering Evap.	80.3	79.9	79.8	80.0	80.2	80.0	79.5	80.4	80.0	80.0
Middle of Evap.	72.3	71.2	69.2	70.6	70.0	67.7	69.2	66.9	65.0	65.8
Leaving Evap.	55.2	54.8	55.0	54.5	55.1	54.6	54.8	54.9	54.5	54.5
Entering Cond.	82.1	82.1	82.1	82.1	82.2	82.0	81.9	81.8	82.0	82.0
Middle of Cond.	95.2	93.5	91.0	95.9	92.5	91.7	95.9	88.6	87.6	90.3
Leaving Cond.	117.2	116.5	117.3	117.0	116.7	117.2	116.6	117.0	117.2	116.8
Pressure, psia.:										
Leaving Evap.	166.6	155.3	142.2	148.3	139.5	131.6	132.8	113.8	109.1	105.7
Comp. Suction	157.6	148.0	133.4	140.0	131.1	123.2	124.7	105.0	102.0	98.8
Comp. Discharge	321.0	313.4	311.2	299.6	281.1	278.8	267.3	232.8	231.7	216.2
Leaving Cond.	309.5	302.2	301.8	289.3	271.1	270.0	257.6	225.6	225.4	209.7
Evap. Capacity kBtu/h:										
$\dot{m} * cp * \Delta T$	19.23	18.88	18.62	19.14	19.08	19.24	18.62	19.45	19.24	19.26
Electric Power	18.56	18.45	18.18	18.55	18.36	18.49	17.94	18.69	18.33	18.51
Calorimeter	18.99	18.93	18.70	19.10	18.99	19.07	18.54	19.25	18.85	19.06
Cond. Capacity, kBtuh:										
$\dot{m} * cp * \Delta T$	22.61	22.36	22.37	22.75	22.48	22.81	21.99	23.32	22.91	22.86
Electric Power	21.34	20.84	21.09	21.24	21.18	21.45	20.48	22.05	21.45	21.22
Comp. Speed, rpm	492	509	539	528	537	572	547	643	655	667
Comp. Power, kBtu/h	3.27	3.35	3.52	3.31	3.15	3.44	3.11	3.50	3.52	3.42
COP	5.82	5.66	5.32	5.77	6.02	5.54	5.78	5.50	5.35	5.58
Composition, % R13	46	42	36	38	36	32	34	22	22	20
Heat Exchanger	Yes	Yes	No	Yes	Yes	No	Yes	Yes	No	Yes

Configuration 4: Refrigerant in Center, Water in Rectangular Passages, Heat Exchange Outside, R13/R12 cont'd.

Test Number	113	114	115
Refrigerant Temp., ^{o}F:			
Entering Evap.	49.6	55.5	55.4
Middle of Evap.	59.2	54.9	54.2
Leaving Evap.	68.5	55.1	54.0
Entering Comp.	70.4	56.7	55.7
Entering Cond.	142.5	142.8	142.6
Middle of Cond.	100.0	112.4	112.5
Leaving Cond.	81.8	104.1	94.8
Before Exp. Valve	81.2	57.9	92.8
Water Temp., ^{o}F:			
Entering Evap.	79.9	79.9	79.9
Middle of Evap.	64.0	57.2	55.8
Leaving Evap.	54.6	55.2	54.7
Entering Cond.	81.9	82.0	81.9
Middle of Cond.	88.6	112.9	112.9
Leaving Cond.	116.8	117.1	116.9
Pressure, psia.:			
Leaving Evap.	103.4	67.3	67.1
Comp. Suction	96.1	60.5	59.5
Comp. Discharge	213.4	164.6	164.2
Leaving Cond.	206.7	157.5	157.5
Evap. Capacity kBtu/h:			
$\dot{m} * cp * \Delta T$	19.03	18.22	18.54
Electric Power	18.36	17.62	17.82
Calorimeter	18.92	18.17	18.40
Cond. Capacity, kBtuh:			
$\dot{m} * cp * \Delta T$	22.71	22.76	22.13
Electric Power	21.32	21.45	21.56
Comp. Speed, rpm	680	1138	1130
Comp. Power, kBtu/h	3.37	4.24	4.20
COP	5.62	4.29	4.38
Composition, % R13	18	0	0
Heat Exchanger	No	Yes	No

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11. ABSTRACT (A 200-WORD OR LESS FACTUAL SUMMARY OF MOST SIGNIFICANT INFORMATION. IF DOCUMENT INCLUDES A SIGNIFICANT BIBLIOGRAPHY OR LITERATURE SURVEY, MENTION IT HERE.)

An experimental, water-to-water, breadboard heat pump apparatus, that is one which could easily be reconfigured, was constructed for comparison of pure R22 to the mixture 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, mixtures can take advantage of heat exchanger efficiency that, in a gliding temperature application, a pure refrigerant is incapable of utilizing. Secondly, 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.

12. KEY WORDS (6 TO 12 ENTRIES; ALPHABETICAL ORDER; CAPITALIZE ONLY PROPER NAMES; AND SEPARATE KEY WORDS BY SEMICOLONS)

air conditioning; heat pump; introcycle heat exchange; nonazeotropic refrigerants; refrigerant mixtures; refrigerants; refrigeration.

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