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EXPERIMENTAL ANALYSIS OF A WINDOW AIR CONDITIONER WITH R-22 AND ZEOTROPIC MIXTURE OF R-32/125/134a

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Energy Renewable and Research Section
Energy Division
Oak Ridge National Laboratory

August 1995

Prepared for the
U.S. Department of Energy
and
E. L. DuPont De Nemours, Co. Inc.

Prepared by
OAK RIDGE NATIONAL LABORATORY
Oak Ridge, Tennessee 37830-6070
managed by
MARTIN MARIETTA ENERGY SYSTEMS, INC.
for the
U.S. DEPARTMENT OF ENERGY
under contract DE-AC05-84OR21400

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[1] On loan from the Department of Applied Thermodynamics and Refrigeration, Royal Institute of Technology, Sweden.

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ABSTRACT

This study is the result of the cooperative research and development agreement (CRADA) between Oak Ridge National Laboratory and E. I. Du Pont De Nemours and Company, Inc., (CRADA No. 92-0161) for testing the use of heat exchangers as the evaporator and condenser in an air-conditioning rig. Heat exchangers at typical realistic operating conditions were tested with R-22 and with its potential replacement, a ternary mixture of R-32(30%)/R-125(10%)/R-134a(60%). A test rig was built that provided for operation of the low-temperature exchanger (evaporator) with flooded coils.

The test results indicated that the performance of the evaporator heat exchanger using ternary mixture, in terms of cooling capacity, would be around 7.4% less than the performance using R-22. The cooling capacity for both refrigerants improved with flooded evaporator operation by 8.6% for R-22 and by 15% for ternary mixture. Compared with R-22 operation, operation with ternary mixture results in slightly higher compressor discharge pressure, lower compressor discharge temperature, slightly lower compressor power consumption, and a higher compressor high-low pressure ratio. Temperature glide for ternary mixture, for both evaporator and condenser, was clearly evident, but not as pronounced as expected because of the pressure drop (and thus the temperature drop) along the coils. Further improvement of the performance of ternary mixture is possible if the evaporator is arranged in a counter-cross-flow configuration to take advantage of the temperature glide. Current evaporator designs are mostly concurrent-cross-flow, which is more appropriate for single-component refrigerants or azeotropic refrigerant mixtures.

ACKNOWLEDGMENTS

The authors would like to thank Dr. D. Bivens of Du Pont for providing us with the ternary refrigerant mixture and for reviewing of the test rig design and test results. The authors would like to thank Mr. J. Nelson of Refrigeration Research, Inc., for providing us the accumulator-heat exchanger used in our test setup. This project was jointly sponsored by E. I. Du Pont De Nemours & Co. and the U.S. Department of Energy. Oak Ridge National Laboratory is managed by the Lockheed Martin Energy System Inc. under contract DE-AC05-84OR21400 with the U.S. Department of Energy.

1. INTRODUCTION

The Montreal Protocol called for the gradual phaseout of all chlorofluorocarbon fluids (CFCs) to protect the stratospheric ozone layer. The London Amendments to the Protocol and the U.S. Clean Air Act called for the eventual total phaseout of CFCs and hydrochlorofluorocarbon fluids (HCFCs). R-22 is one of the most widely used HCFCs for applications such as residential room air conditioners, heat pumps, and supermarket refrigeration systems. While much work has been done to identify replacements for R-11 and R-12, relatively little experimental work with off-the-shelf heat exchangers has been done to identify a replacement for R-22 in residential air conditioning and heat pump applications. Radermacher and Jung (1993) theoretically analyzed the performance of several R-22 replacements, but ternary mixture was not one of them. Fischer and Sand (1993) screened potential R-22 replacements. Their modeling effort indicated that ternary mixtures could have an increase of up to 4% in the coefficient of performance (COP) and slightly reduced capacity. They also mentioned that with a change in the composition of three refrigerants, the capacity could be increased by up to 20%. Domanski and Didion (1993) evaluated R-22 alternatives thermodynamically. Among their findings, using R-22 performance as the baseline data, are that a ternary mixture [R-32(30%)/R-125(10%)/R-134a(60%)] has almost identical capacities and COP, but higher discharge pressures and lower discharge temperatures. This mixture has been regarded as one of the most likely R-22 replacements; however, little experimental work with practical heat exchanger configuration used on space conditioning equipment and operating conditions is available in the public domain.

A cooperative research and development agreement (CRADA) was established between E. I. Du Pont De Nemours and Company, Inc., and Oak Ridge National Laboratory (ORNL) in July 1993 for heat exchanger testing. The heat exchangers serve as the condenser and evaporator for air conditioning applications, using R-22 and the ternary mixture. The purpose of this study is to experimentally analyze the performance of the mixture, tested under a near-drop-in situation with off-the-shelf heat exchangers, and compare it with the results of tests using R-22. In this study, only refrigerant-side tests were performed.

The results indicated that the performance of the ternary mixture is less than that of R-22 by a 7.4% drop in heat exchanging capacity at 95°F ambient. However, when the evaporator coil is flooded, the improvement of the ternary mixture performance is more than that of R-22 performance. The performance of ternary mixture is improved to within less than 2% to that of R-22.

2. TEST SETUP

The test setup includes a heat exchanger as the evaporator, another heat exchanger as the condenser, and a compressor. Capillary tubes are used as an expansion device connected to the inlet of the evaporator. Figures 1 and 2 shows the heat exchangers tested. The evaporator heat exchanger is composed of four circuits. Thermocouples are installed on each circuit. The top circuit has more thermocouples than the other three because, judging from the air movement, the top circuit will probably be the most efficient section of the evaporator. However, the detailed temperature measurement on one circuit still only provides a general picture of the heat exchanger performance. The condenser heat exchanger is composed of four rows but only two circuits. Only one circuit has thermocouples on it because the performance should be fairly even with only two circuits.

Figures 3 and 4 show the schematic of the test rig and the arrangement of the rig in the chambers. Figure 3 shows an accumulator/heat exchanger (AHX) incorporated in the test loop with a refrigerant charge of 0.41 lb. For flooded evaporator tests, additional refrigerant (0.44 lb) is charged into the test loop until low-pressure refrigerant accumulates in the AHX (Mei and Chen, 1994). Warm, high-pressure refrigerant from the condenser flows through the heat exchanger coil in the AHX and boils the low-pressure refrigerant in the AHX. The heat exchange in the AHX cools the high-pressure refrigerant and results in a high level of refrigerant subcooling before the refrigerant reaches the expansion devices (4 capillary tubes in this study). Because the refrigerant enters the evaporator in a highly subcooled condition, the refrigerant is not all evaporated in the evaporator coils. Two-phase refrigerant from the evaporator exit flows to the AHX. The liquid is trapped in the AHX to be boiled off by the warm liquid from the condenser flowing through the heat exchanger coil in the AHX. The AHX thus provides flooded evaporator coil operation without liquid slugging back to the compressor.

The refrigerant volumetric flow rate is measured by a turbine flow meter. The pressures are measured by five pressure transducers. All the temperatures are measured with type T thermocouples. Their locations are shown in Fig. 3.

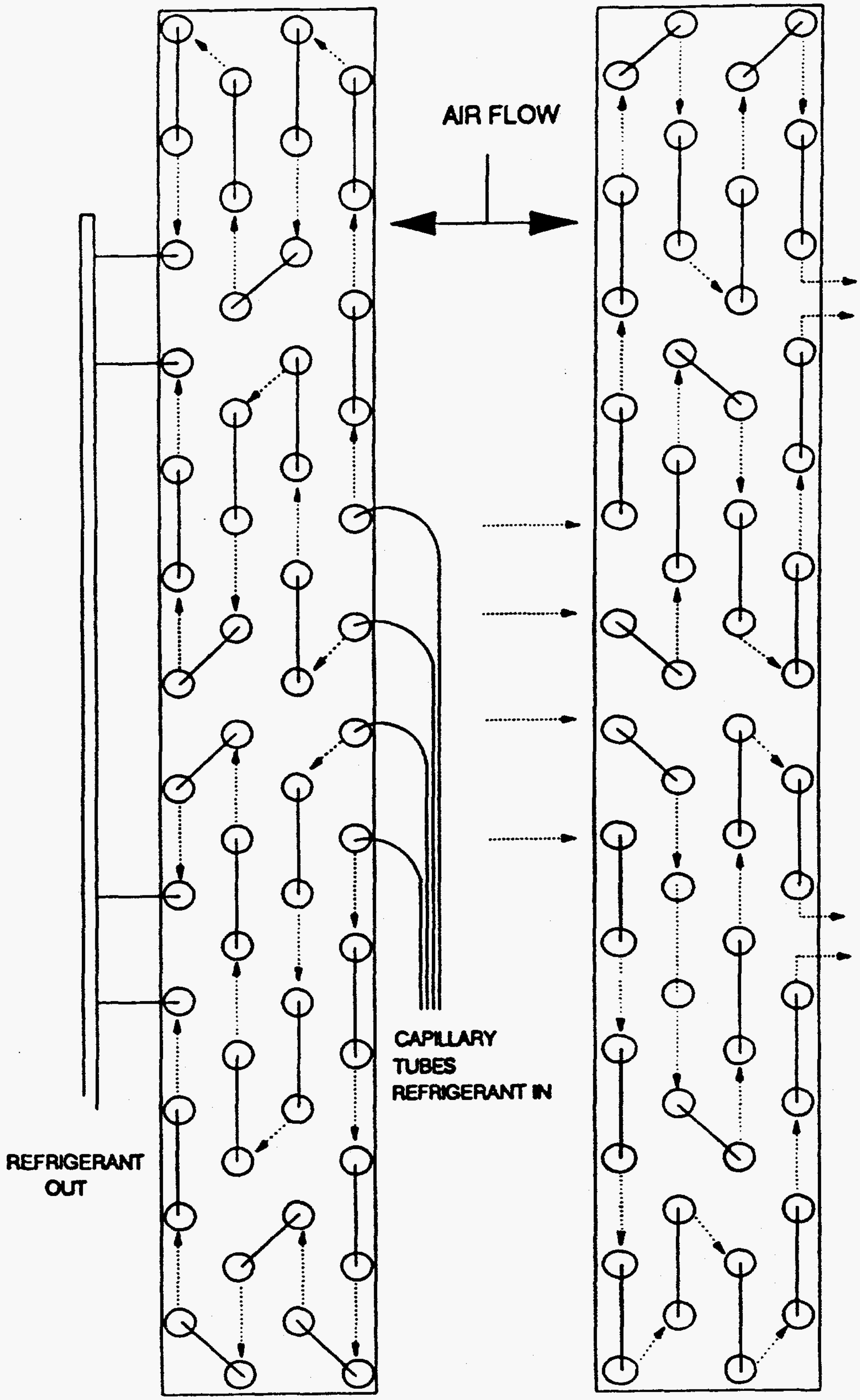

Fig. 1 Evaporator heat exchanger

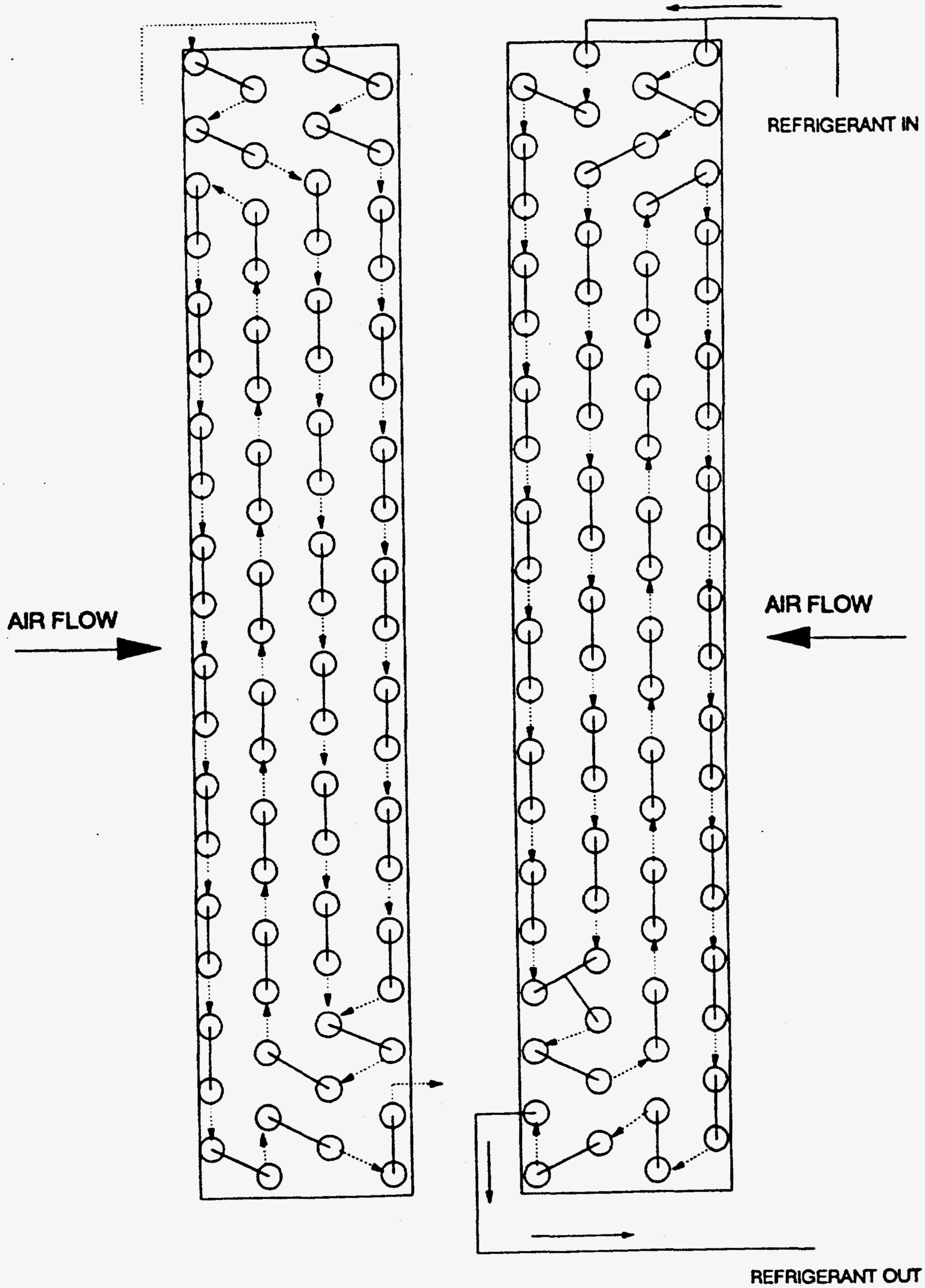

Fig. 2 Condenser heat exchanger

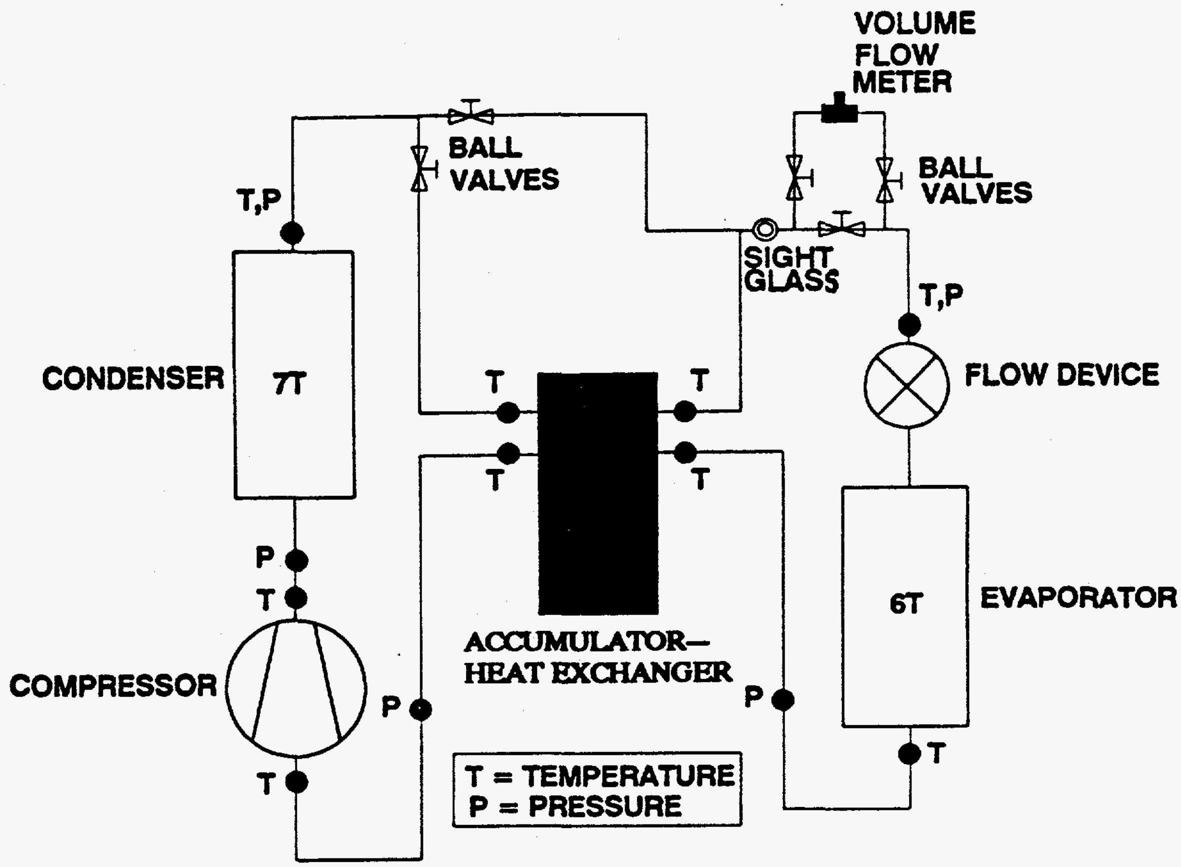

Fig. 3 Schematic of test rig setup

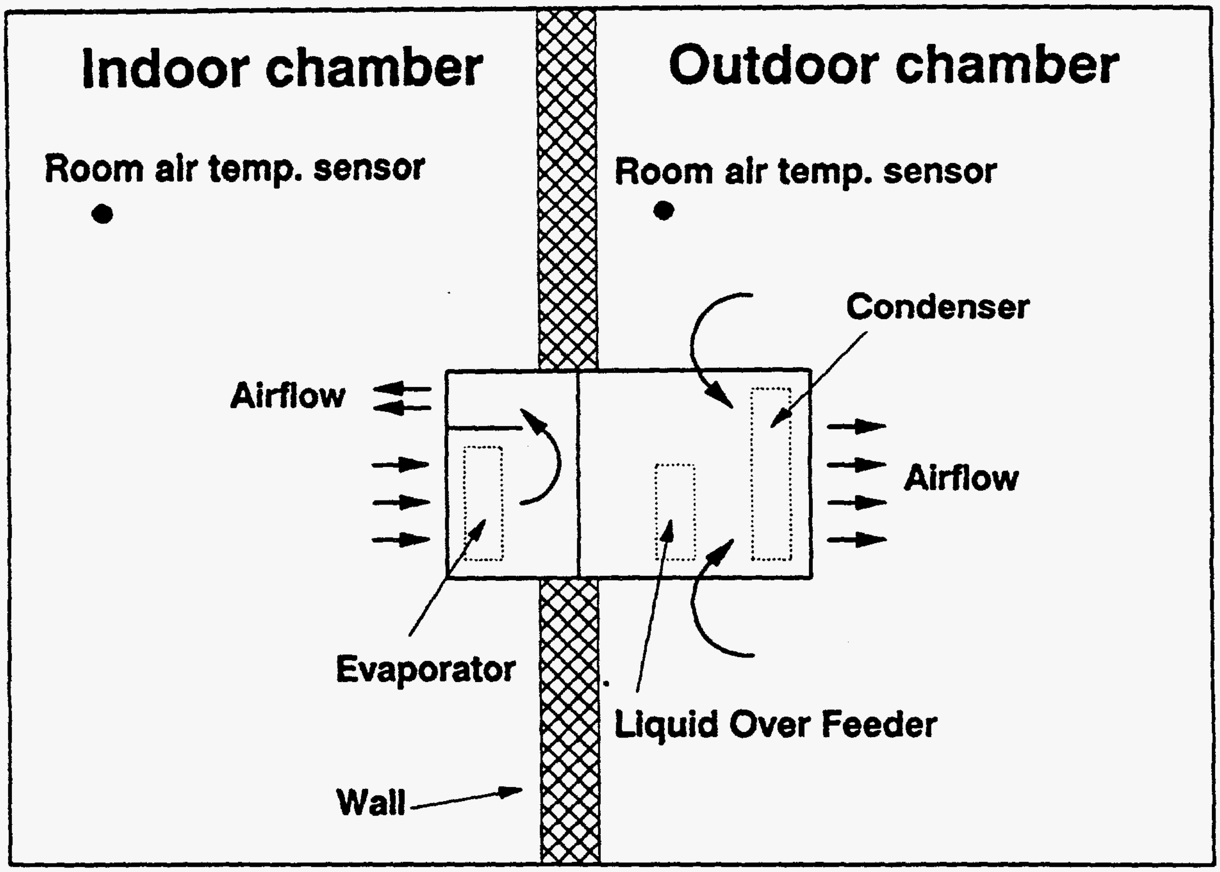

Fig. 4 Test setup in environmental chamber

3. TEST PROCEDURES

The tests were for two different types of refrigerants, R-22 and the ternary mixture. The evaporator heat exchanger was operated at nonflooded and flooded conditions. The indoor chamber conditions were maintained at 80°F and 52% RH; the outdoor chamber relative humidity was set at 27%; and temperature, which simulates the ambient temperature, was varied from 80°F to 120°F. For nonflooded heat exchanger operation, the maximum outdoor chamber temperature was set at 110°F because of the excessive compressor discharge pressure. All tests were performed at steady state operating conditions. Table 1 is the test matrix for R-22 and ternary mixture.

Table 1. Heat exchanger test matrix for R-22 and ternary mixture

Chamber temperature, °F		Chamber relative humidity, %		Nonflooded coil tests	Flooded coil tests	Remarks
Indoor	Outdoor	Indoor	Outdoor	*	*	4.1 lb of refrigerant charge for non-flooded coil tests. An additional .44 lb was charged for flooded coil operation
80	80	52	27	*	*	
80	85			*	*	
80	90			*	*	
80	95			*	*	
80	100			*	*	
80	105			*	*	
80	110			*	*	
80	115				*	
80	120				*	

The test procedures were basically the same for both refrigerants, except the procedure for the system cleaning during the change of one refrigerant to the other.

1. The indoor chamber was set at 80°F and 52% RH.

2. The units were tested at outdoor chamber temperatures from 80°F to 120°F, with an increment of 5°F, for both refrigerants with flooded coil operation. It was found that when the evaporator coils were flooded, the compressor discharge pressure became lower. For conventional system operation (nonflooded coils), the highest outdoor chamber temperature was set at 110°F to avoid excessively high discharge pressure.

3. For flooded coil tests, the heat exchange coil inside the accumulator was activated.

3. For flooded coil tests, the heat exchange coil inside the accumulator was activated. Additional refrigerant was charged into the unit until the output data indicated the vapor superheat was gone.

4. When the refrigerant was changed from R-22 to ternary mixture, the following procedures were followed:
   A. The system was evacuated.
   B. The compressor was cut out and the oil drained. New ester-based oil was added and the compressor was turned on for a minute or so. The oil was again drained. New ester-based oil was again charged, and the compressor was reconnected to the system.

4. TEST RESULTS

The test results can be divided into several groups—heat exchanger coil temperature distribution, heat exchanger performance, and component performance, particularly compressor performance.

4.1 HEAT EXCHANGER COIL TEMPERATURE DISTRIBUTION

4.1.1 Evaporator Coil Temperature Distribution

4.1.1.1. Nonflooded Coil Temperature Distribution. Figures 5 and 6 show the evaporator top circuit temperature distribution as a function of the ambient temperature for R-22 and ternary mixture, respectively, under nonflooded coil conditions. In R-22 operation, the temperature decreases along the circuit because of the pressure drop. At lower ambient conditions, 85°F and 90°F, the temperature at less than half of the coil length starts to increase sharply. This indicates the refrigerant is dried out. At higher ambient temperatures, the drying out is delayed to 8/14 of the circuit length. At lower ambient temperatures, the condenser heat exchanger performance improves, resulting in a lower compressor discharge pressure, which in turn, results in a lower mass flow rate. That is why refrigerant dries out more quickly at lower ambient temperatures. Generally, higher ambient temperature results in higher refrigerant temperature. In this test, a 5°F ambient temperature rise will increase the coil temperature by about 3°F for most of the circuit length. At the coil exit, the temperature differences are narrowed to only 5°F for all ambient conditions. This is because the circuit is dried out at the exit and the refrigerant temperature is close to air temperature. For ternary mixture operation, as shown in Fig. 6, the temperature along the circuit increases instead of decreasing, even with pressure drop along the coil, for the section before refrigerant dry-out. The maximum temperature glide is about 6.5°F. The refrigerant inlet temperature is lower than that for R-22 operation. At around 6/14 of the circuit length, the ternary mixture temperature becomes higher than R-22 coil temperature. The coil exit temperature of ternary mixture is around 5 to 7°F higher than that of R-22.

4.1.1.2 Flooded Coil Temperature Distribution. Figures 7 and 8 show the R-22 and ternary mixture evaporator top circuit temperature distribution with flooded coil operation. The dry-out region is almost all gone, except near the exit where the refrigerant temperature increases sharply. This is a good indication that the refrigerant flows in the four evaporator circuits are not evenly distributed, because the refrigerant temperature measured after the point where all four circuits merge is lower than the exit temperatures shown in the figures. That means that some evaporator circuits are flooded and some are not. The exit temperatures at different ambient conditions are not approaching each other as in the nonflooded test. The figures show that ternary mixture refrigerant temperatures are actually lower than those of R-22 for most of the top circuit length.

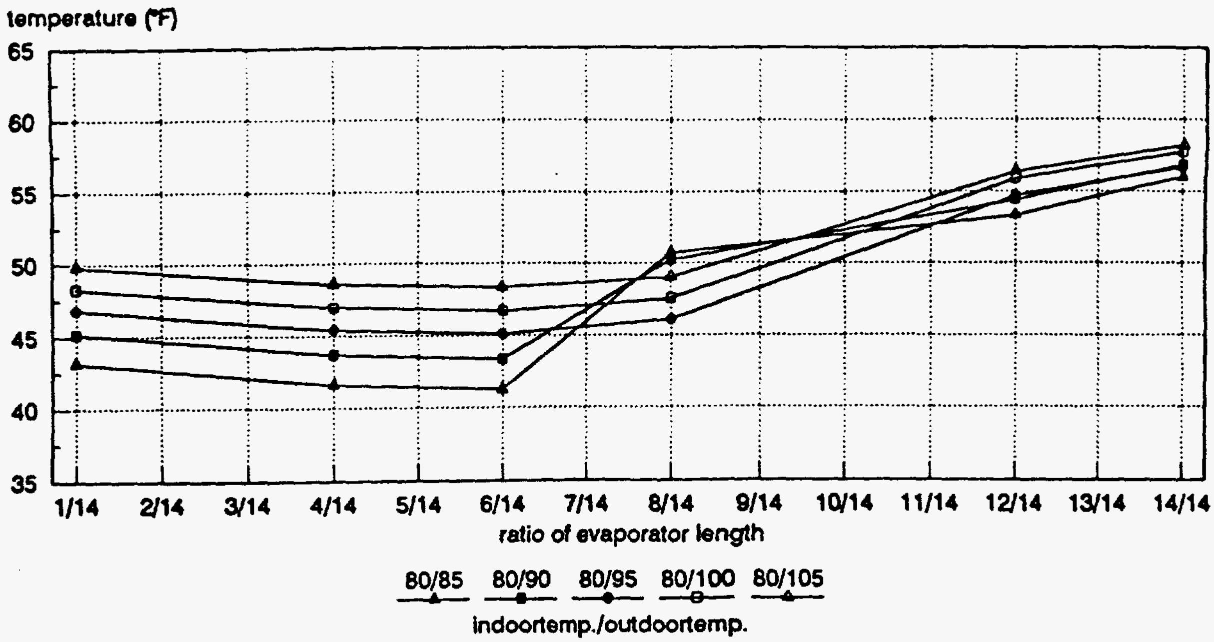

Fig. 5 Evaporator top circuit temperature distribution—R-22, nonflooded coils

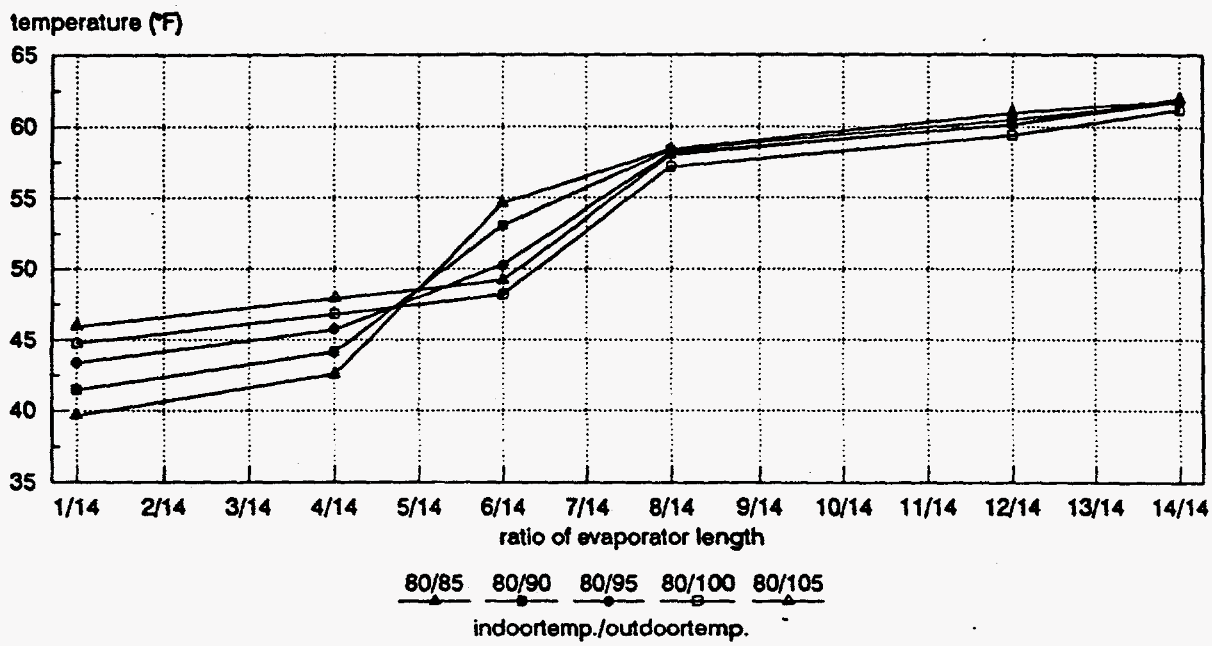

Fig. 6 Evaporator top circuit temperature distribution—ternary mixture, nonflooded coils

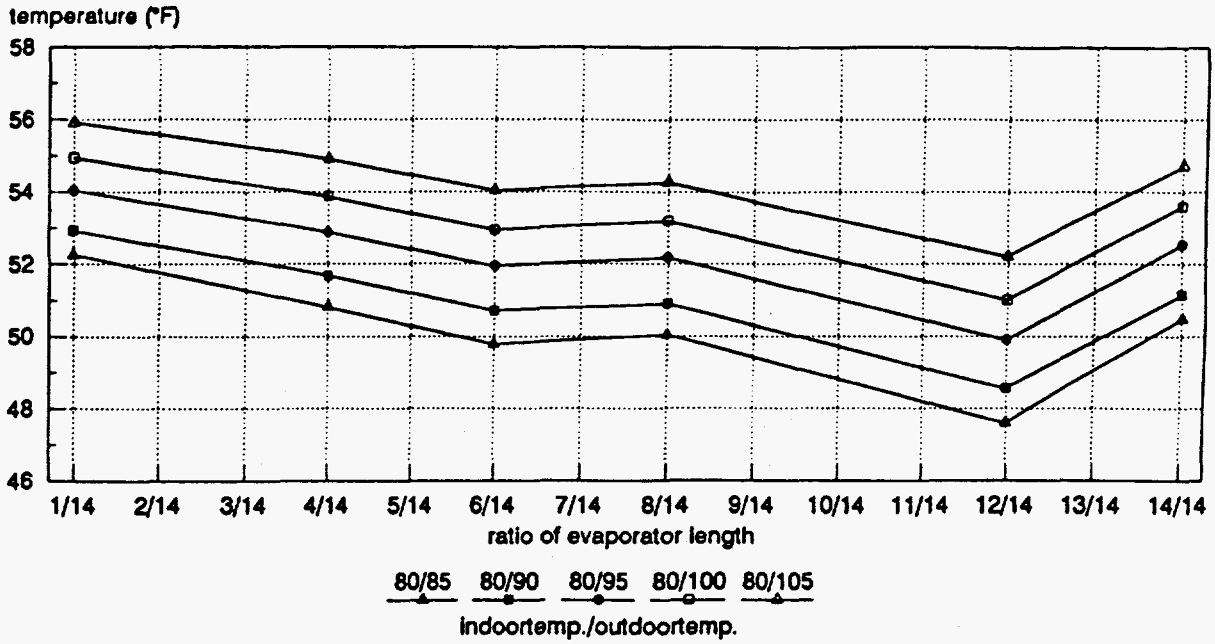

Fig. 7 Evaporator top circuit temperature distribution—R-22, flooded coils

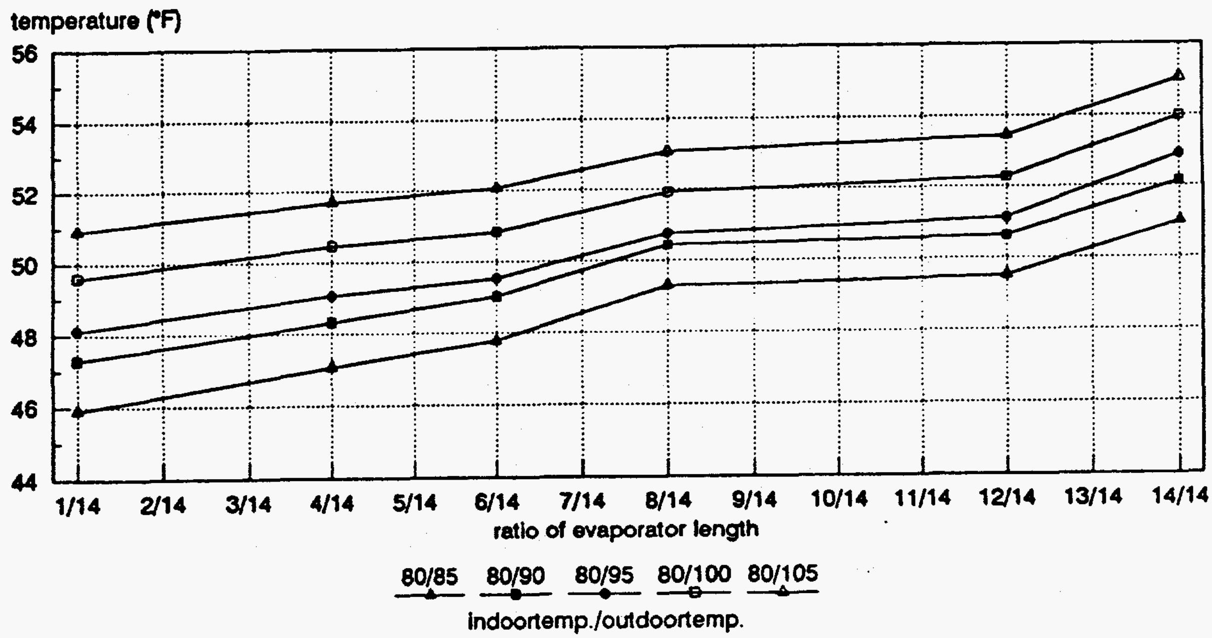

Fig. 8 Evaporator top circuit temperature distribution—ternary mixture, flooded coils

This indicates that a flooded evaporator probably enhances circuit performance more for the ternary mixture than for R-22. The temperature glide of around 4°F for the ternary mixture is clearly shown in Fig. 8.

The concurrent-cross-flow evaporator heat exchanger design is suitable for a single refrigerant application where refrigerant temperature becomes lower because of the pressure drop along the coil. The same arrangement works against non-azeotropic refrigerant mixtures such as the ternary mixture tested. While the air is being cooled down, the coil temperature keeps increasing because of the temperature glide, even with the pressure drop along the coil. If the evaporator heat exchanger is counter-cross-flow, the performance of this mixture should improve and the range of temperature glide should be extended.

4.1.2 Condenser Coil Temperature Distribution

4.1.2.1 Nonflooded Coil Temperature Distribution. Figures 9 and 10 chart the temperatures along the condenser coil as a function of ambient temperature for R-22 and the ternary mixture, respectively. The condenser coil arrangement is in counter-cross-flow form between air and refrigerant. For R-22, the coil temperatures clearly indicate the three condenser regions: superheated region, up to 8/58 of the coil; two-phase region, from 8/58 to 40/58 of the coil; and subcooled region for the rest of the coil. For the ternary mixture, the profile of the coil temperatures is similar; but because of the temperature glide, we are not sure where the subcooling region starts. The mixture has a lower coil temperature than does R-22 at all ambient temperatures.

4.1.2.2 Flooded Coil Temperature Distribution. Figures 11 and 12 show the temperatures along the top condenser circuit under flooded conditions. The inlet temperature is greatly reduced. This is because the vapor refrigerant at the compressor inlet is constantly at or near saturated condition with little or no superheat when the evaporator coil is flooded. In turn, the compressor discharge temperature becomes lower. However, the exit temperature seems higher. The reason could be the increase in mass flow rate for flooded coil operation. With lower compressor discharge temperature but higher mass flow rate, it is not surprising that the refrigerant at the condenser exit has higher temperature than in the nonflooded coil operation.

The temperature glide for the ternary mixture is small. If the temperature profiles shown in the figures are compared, the R-22 two-phase region of the circuit from 20/58 to 40/58 has a temperature drop of only about 1 or 2°. But the drop is around 4°F for the mixture. The concurrent-cross-flow coil arrangement will help the system performance with the mixture. However, because most heat exchangers for condenser application have only two circuits, coil arrangement will probably have limited influence on the condenser performance.

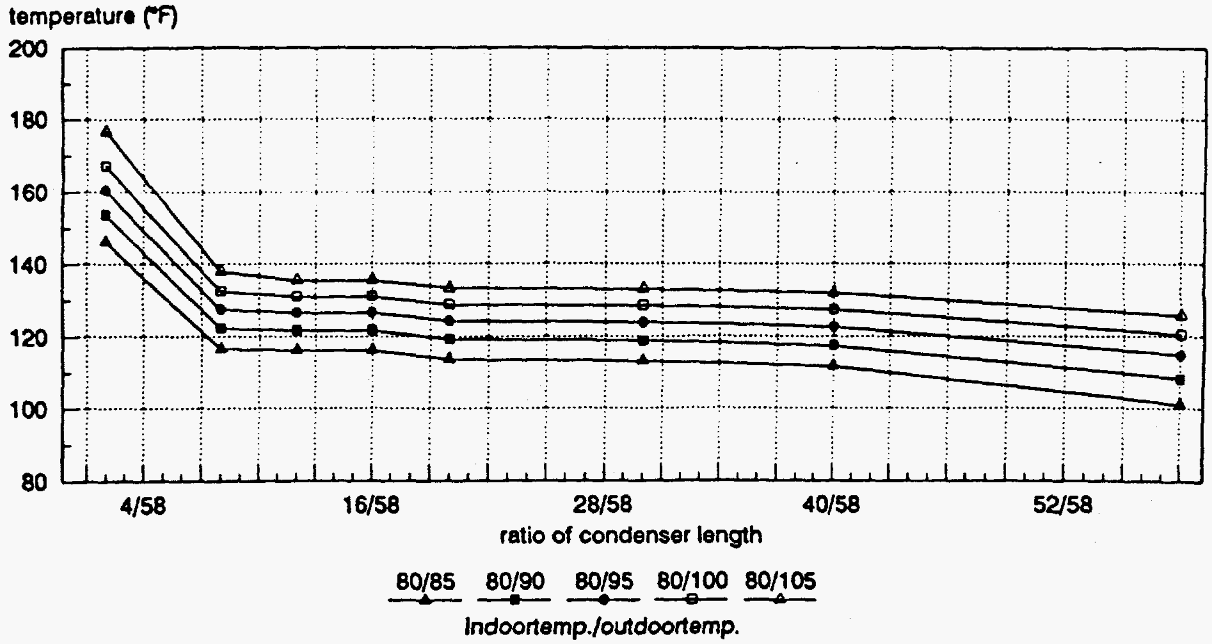

Fig. 9 Condenser top circuit temperature distribution—R-22, nonflooded coils

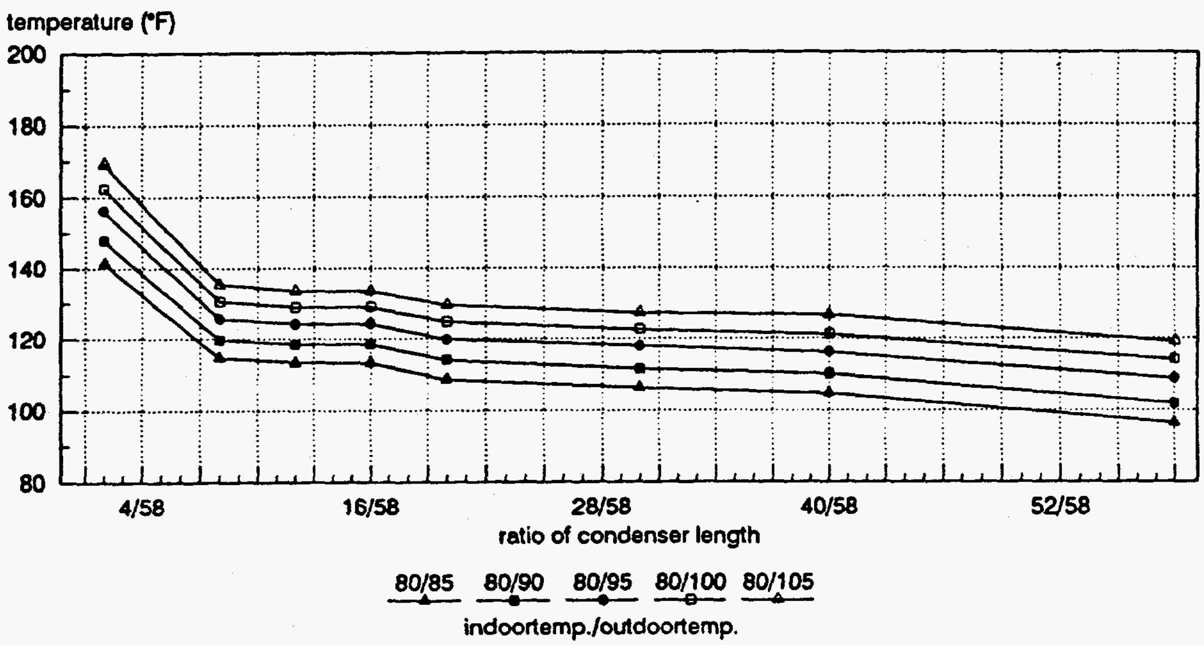

Fig. 10 Condenser top circuit temperature distribution—ternary mixture, nonflooded coils

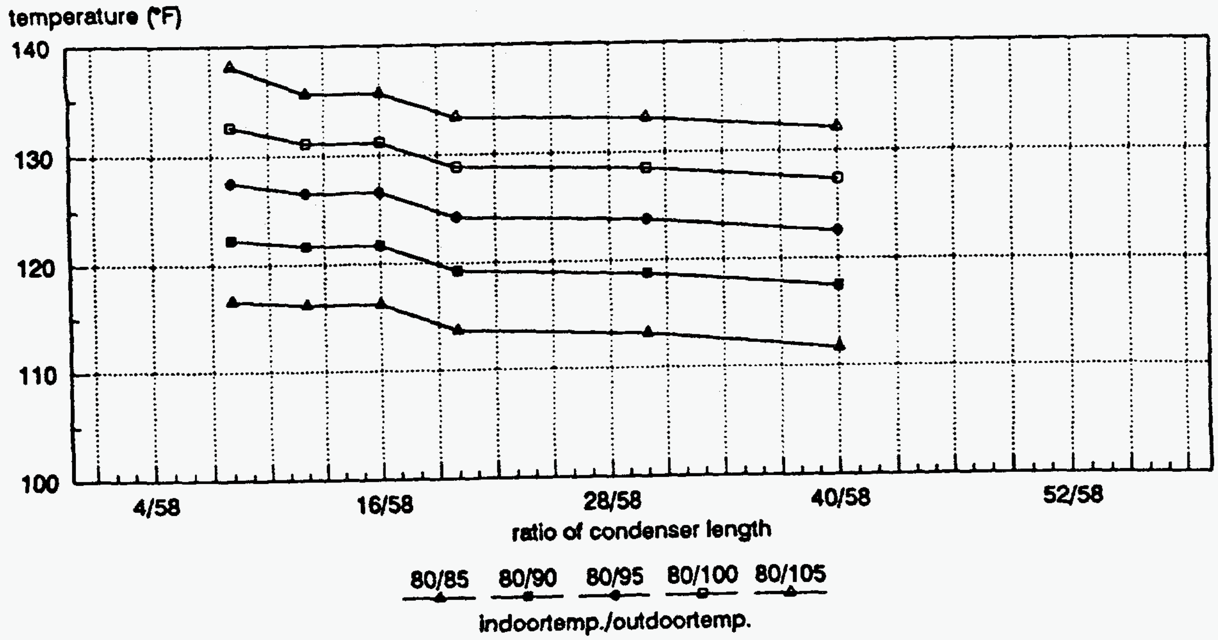

Fig. 11 Condenser top circuit temperature distribution—R-22, flooded coils

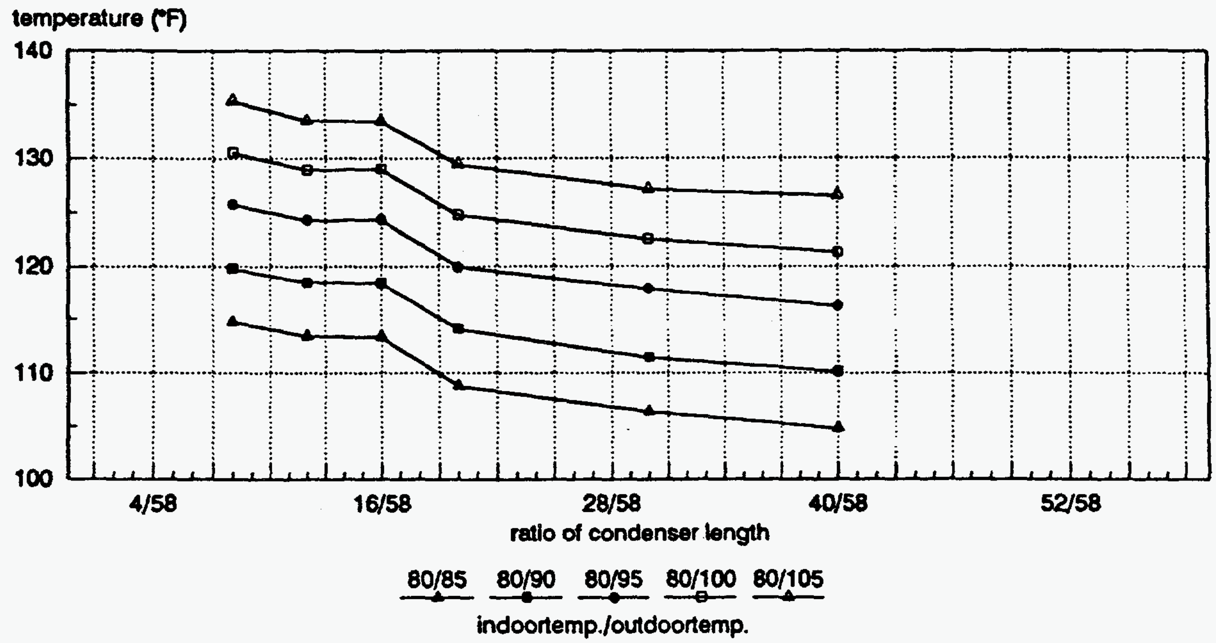

Fig. 12 Condenser top circuit temperature distribution—ternary mixture, flooded coils

4.2 EVAPORATOR HEAT EXCHANGER CAPACITY

The amount of heat exchange is calculated from the measured refrigerant temperature and pressure at the inlet and exit of the exchanger for the nonflooded coil test. The enthalpy difference times the mass flow rate is the cooling capacity. For flooded-coil tests, refrigerant at the evaporator exit is two-phase, and the enthalpy at that point cannot be determined. Additional temperature measurements are needed to calculate the refrigerant enthalpy at the compressor inlet, and then subtract the heat absorbed by the warm refrigerant liquid from the condenser flow through the heat exchanger coil in the accumulator. Figure 13 shows the evaporator heat exchanger cooling capacity as a function of ambient temperature. Capacities drop as the ambient temperature increases. The ternary mixture in nonflooded coil test has the lowest cooling capacity. At 95°F ambient and nonflooded coil conditions, the mixture's cooling capacity is about 1900 Btu/h lower than that of R-22, about a 7.5% reduction. However, at the same ambient temperature but with a flooded coil, the cooling capacities of the ternary mixture and R-22 differ by only 500 Btu/h (less than a 2% capacity penalty). The mixture at flooded coil condition outperforms R-22 at nonflooded operation by over 1500 Btu/h. In the nonflooded coil test at 95°F ambient the mixture has a cooling capacity of around 23,100 Btu/h. However, once the coil is flooded, the capacity at the same ambient jumps to 26,600 Btu/h, an increase in cooling capacity of 3500 Btu/h, a 15% improvement. For R-22 at the same ambient, the cooling capacity increase of flooded coil over nonflooded coil operation is about 2100 Btu/h, or an 8.4% improvement. It is clear that flooded coil operation has more effect on the mixture than on R-22.

4.3 COMPRESSOR PERFORMANCE

The compressor performance is calculated from refrigerant-side measurements of the compressor suction and discharge temperatures and pressures, and of the refrigerant mass flow rate. Compressor power consumption is measured directly from a watt mater.

4.3.1 Compressor Discharge Pressure

Figure 14 shows the compressor discharge pressure as a function of the outdoor temperature for R-22 and the mixture. The discharge pressures for R-22 are lower than those of the mixture. The differences become greater when the ambient air temperature is higher. At 80°F, the refrigerants have the same discharge pressures, except for the mixture at flooded coil condition, which is about 15 psi higher. At 95°F ambient, the pressure differences become more apparent. The mixture's pressure is around 6 psi higher than R-22 in nonflooded coil operation. For flooded coil condition, R-22 discharge pressure actually decreases by 6 psi over nonflooded coil operation. However, the mixture discharge pressure for flooded coil condition is even higher than for nonflooded coil condition by about 3 psi. At 120°F ambient and flooded coil operation, the discharge pressure of the mixture is almost 40 psi higher than that of R-22. The reason that the mixture has higher pressure at flooded coil operation could be explained this way: the heat exchange coil in the AHX is boiling off

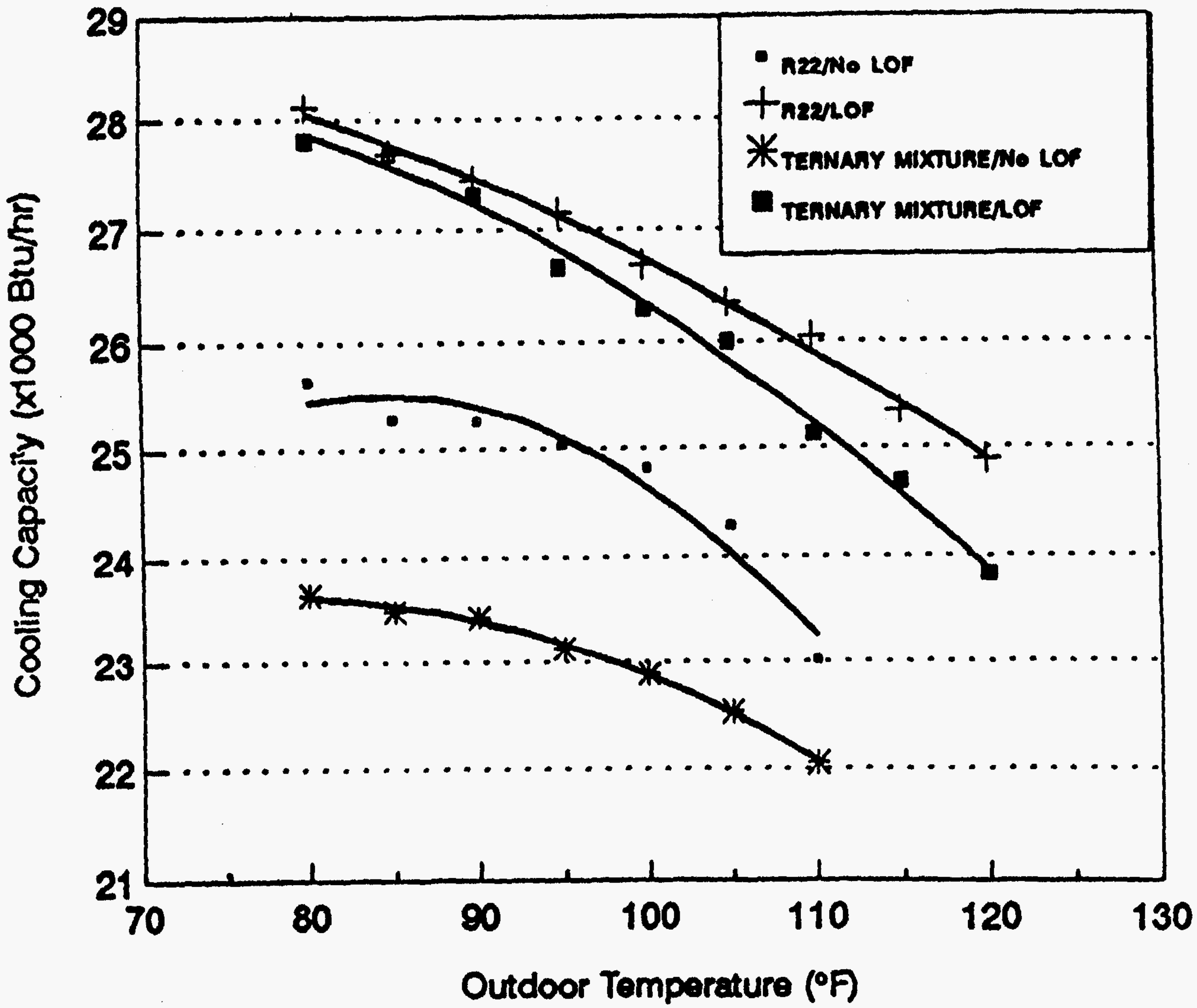

Fig. 13 Evaporator cooling capacity as a function of outdoor temperature

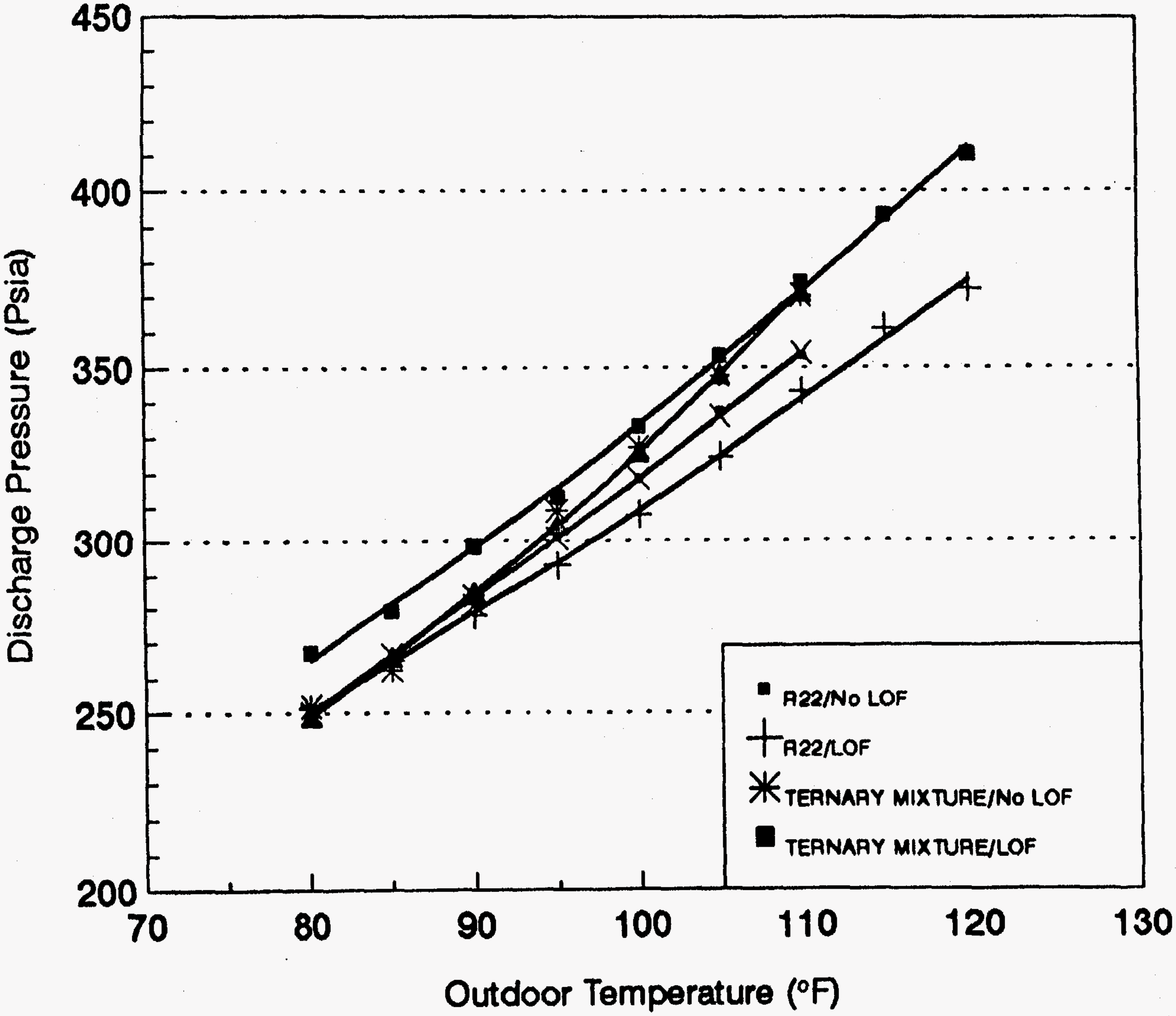

Fig. 14 Compressor discharge pressure as a function of outdoor temperature

refrigerant R-32 and R-125 first because these compounds of the ternary mixture have lower boiling points than R-134a. The net result would be that at flooded coil operation, the compressor is actually circulating a mixed refrigerant that is richer in R-32 and R-125, and thus has a higher discharge pressure and higher system cooling capacity.

4.3.2 Compressor Discharge Temperature

Figure 15 shows the compressor discharge temperature as a function of ambient temperature. The mixture has a lower discharge temperature than R-22. With a nonflooded coil, the temperature of R-22 is about 19°F higher than that of the mixture at 95°F ambient. With flooded coils, the difference at the same ambient is also about 19°F. The difference becomes greater at higher ambient temperature.

4.3.3 Compressor High-Low Pressure Ratio

The compressor discharge pressure and temperature usually do not give the compressor performance completely. But the compressor high-low pressure ratio for a certain refrigerant usually indicates the compressor operating efficiency: the lower the ratio, the higher the compressor operating efficiency. Figure 16 shows the compressor high-low pressure ratios of R-22 and the mixture for both nonflooded and flooded coil operations. The flooded coil operations have lower pressure ratios for both refrigerants: 13% lower for R-22 and 11% lower for the mixture. While the mixture has a higher ratio than R-22, that does not necessarily mean that the compressor is more efficient with R-22. The thermodynamic properties of the refrigerants have to be considered, together with the compressor power input. Then the compressor performance of different refrigerants can be determined and compared.

4.3.4 Mass Flow Rate

Figure 17 shows the mass flow rate as a function of ambient temperature. The mass flow rates for flooded coil operation are much higher than for nonflooded coil condition. Under flooded coil operation, the compressor constantly has saturated, or near saturated, suction inlet conditions, and thus has higher mass flow rate. The mass flow rate for R-22 is about 60 lb/h, or 16.4%, higher than that of the mixture at 95°F ambient with nonflooded coils, and the difference is consistent over the entire test range. For flooded coil operation, the mixture mass flow rate increases more than that of R-22. The difference reduces to 40 lb/h at 95°F ambient. The difference becomes smaller when the ambient temperature increases. This is an indication that flooded coil operation aids the performance of the mixture more than that of R-22 in terms of mass flow rate. Higher mass flow rate means a higher refrigerant-side heat transfer coefficient and potentially higher cooling capacity.

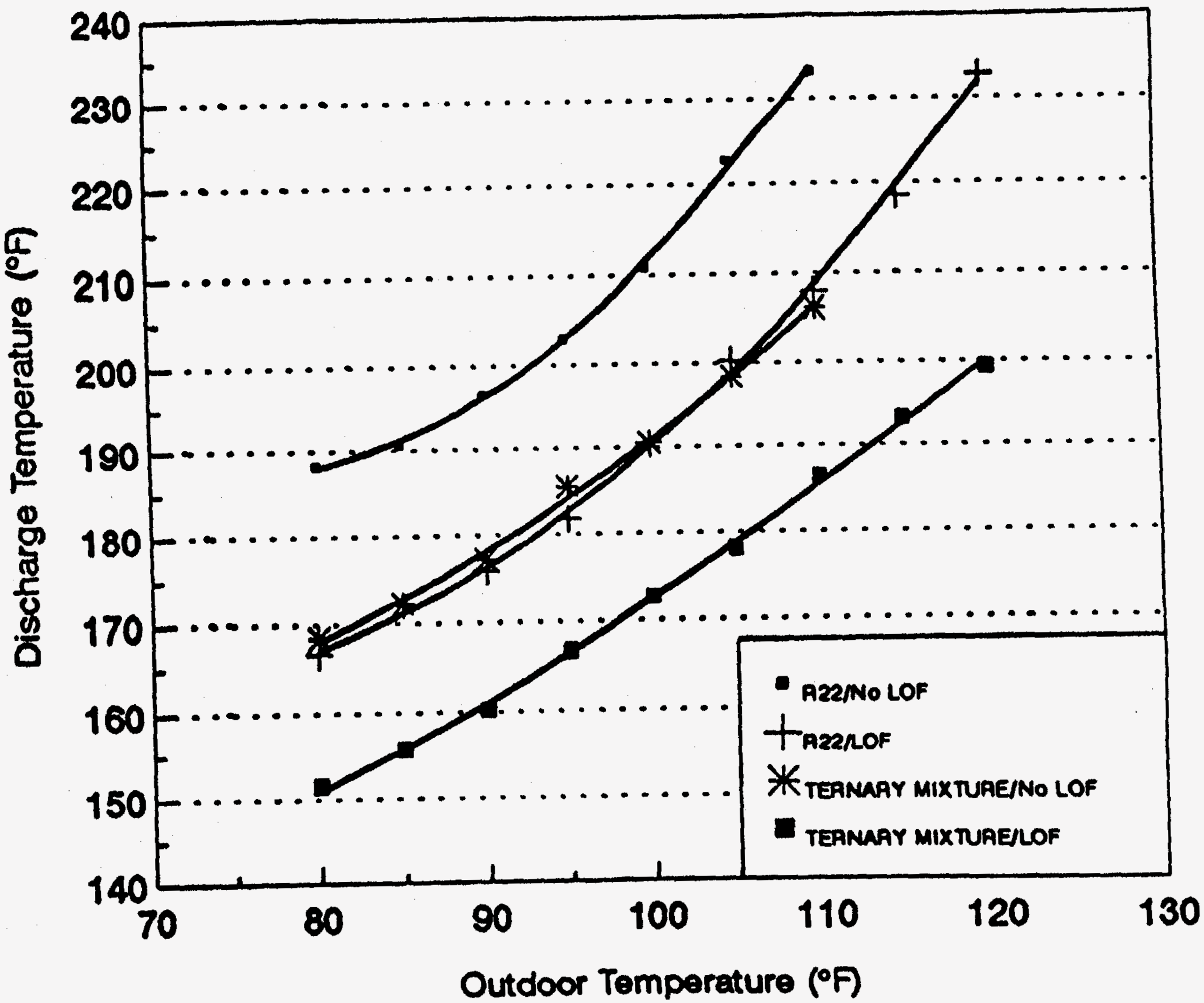

Fig. 15 Compressor discharge temperature as a function of outdoor temperature

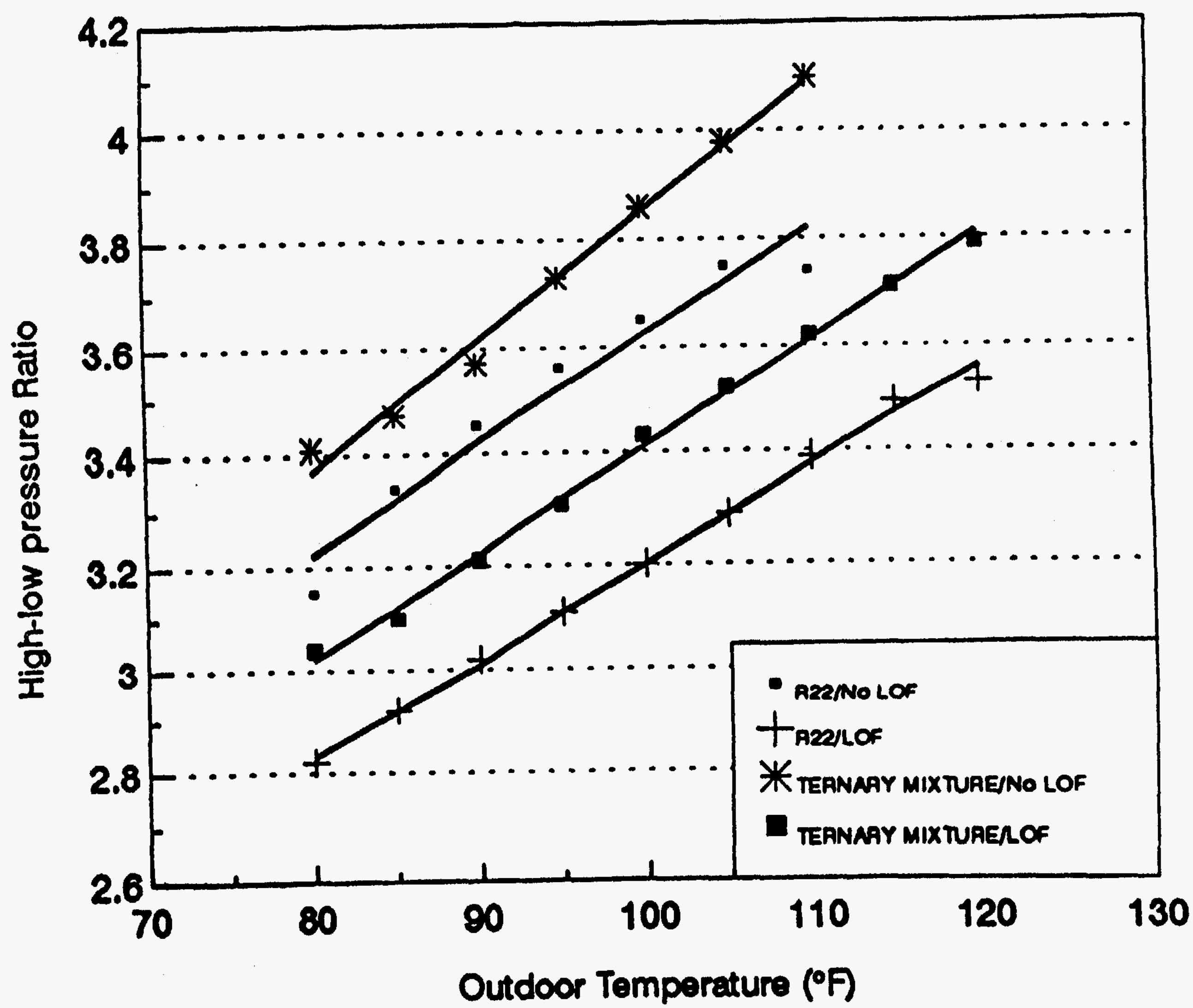

Fig. 16 Compressor high-low pressure ratio as a function of outdoor temperature

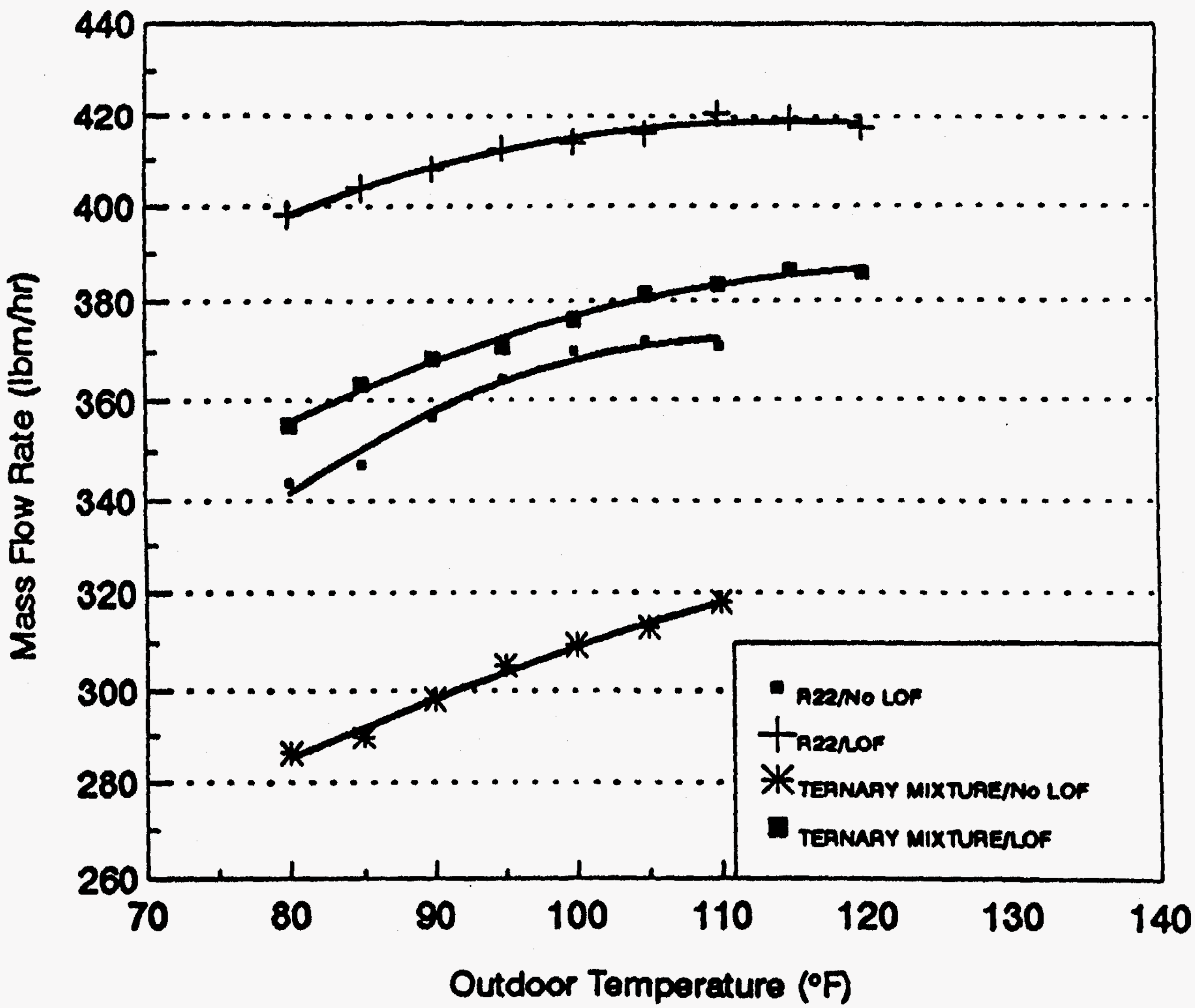

Fig. 17 Refrigerant mass flow rate as a function of outdoor temperature

4.3.5 Compressor Power Consumption

Figure 18 shows the system power consumption as a function of ambient temperature. Generally, the higher the ambient temperature, the higher the system power consumption because of higher mass flow rate and higher compressor discharge pressure. It is interesting to compare flooded coil and nonflooded coil operations. For R-22, the power consumption is almost identical for ambient temperature up to 100°F regardless of whether the coils are flooded, even though the flooded coil operation will produce over 12% more mass flow rate at 95°F ambient (see Fig. 17). For the mixture, flooded coil operation consumes slightly over 4% more power than nonflooded coil operation, while the mass flow rate for the flooded coil increases by more than 20% over the nonflooded coil. When the coil is flooded, the compressor discharge temperature becomes lower, which often implies less compressor power consumption. However, because of the increase of refrigerant mass flow rate, the compressor power consumption should increase. The two factors work against each other, resulting in higher mass flow rate for flooded coil operation without increasing compressor power in the same proportion.

4.3.6. System Coefficient of Performance (COP)

Figure 19 shows the system COPs as a function of outdoor temperature for LOF and baseline operations. At 95°F ambient, the COP for the ternary mixture is about 7.4% and 2.5% less than the COP for R-22 during baseline and LOF operations, respectively. LOF operation enhances the performance of the ternary mixture more than that of R-22: COP is improved by 6.8% for R-22 and by 9.7% for the ternary mixture over baseline operation.

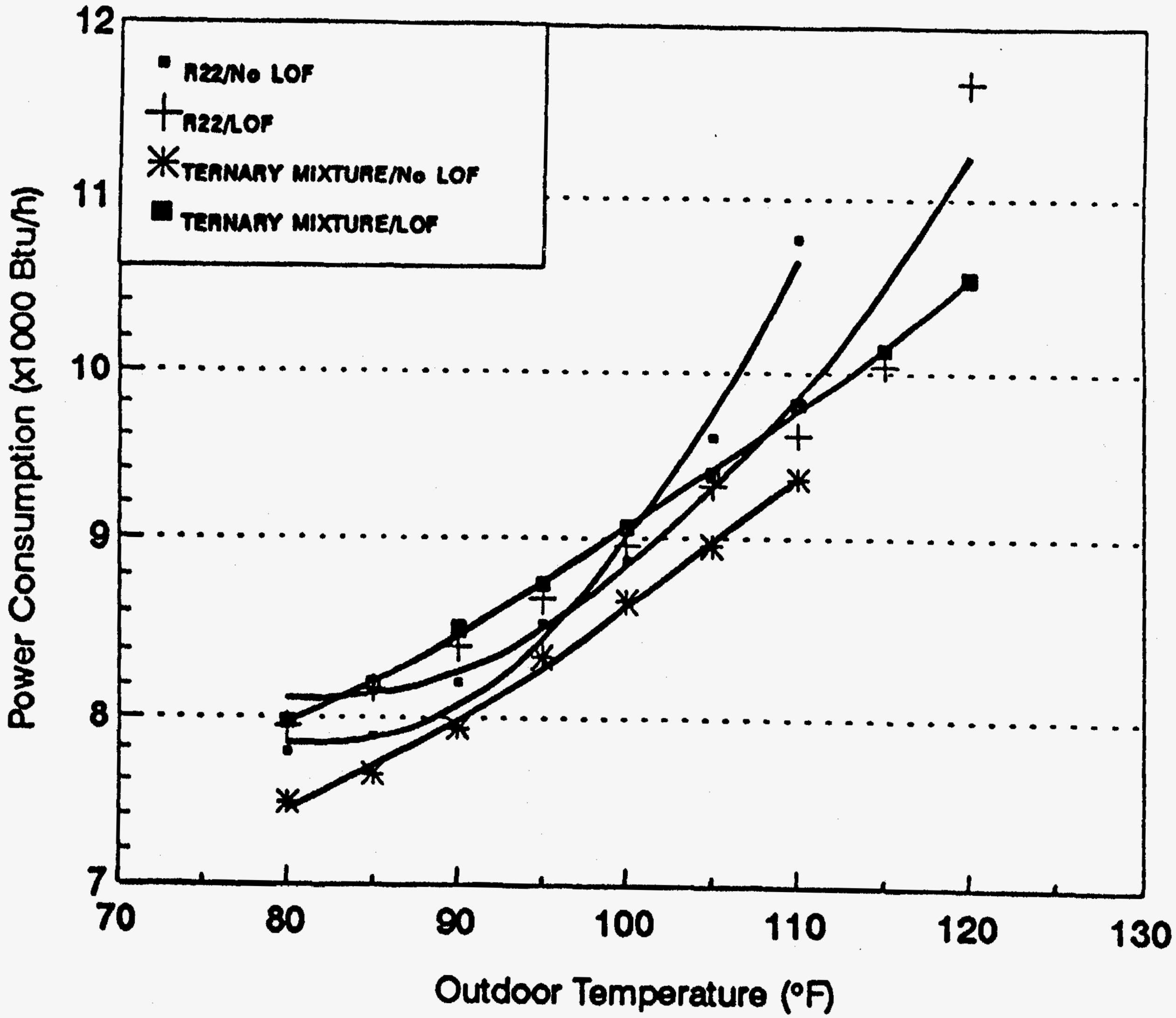

Fig. 18 Compressor power consumption as a function of outdoor temperature

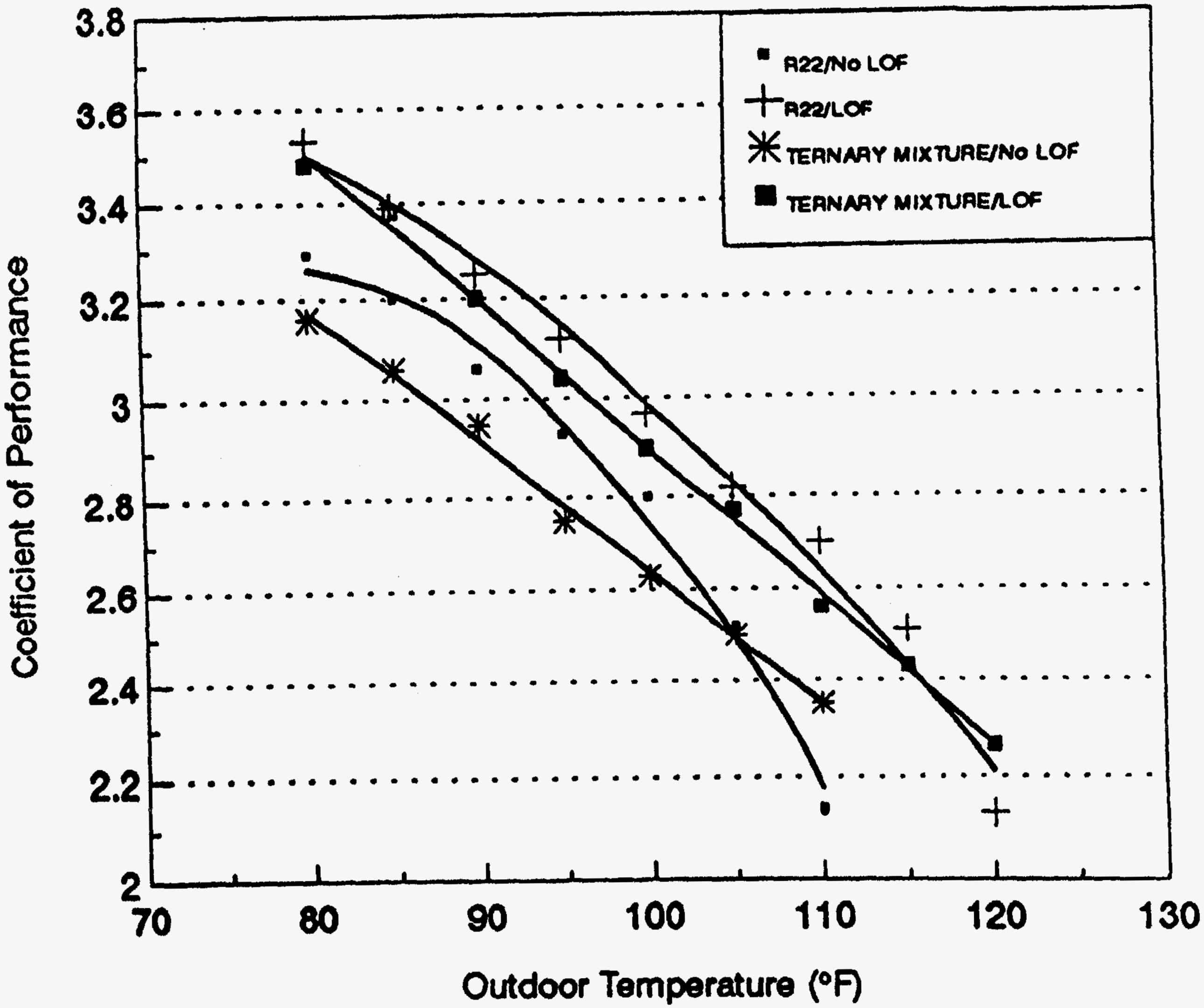

Fig. 19 System COP as a function of outdoor temperature

5. DISCUSSIONS AND CONCLUSIONS

Two heat exchangers, one used as an evaporator and the other used as a condenser, were extensively tested using R-22 and a ternary mixture of R-32/R-125/R-134a at nonflooded and flooded evaporator conditions. The test results can be summarized as follows.

1.For nonflooded evaporator operation, the mixture has a lower evaporator inlet temperature than that of R-22. But the temperature along the evaporator circuit continues to rise even as pressure drops along the coil. The temperature glide is only about 3°F before dry-out occurs. The average mixture temperature is higher than that of R-22. For the flooded coil test, the temperature glide for the mixture is around 4°F over a longer coil section.

2. Temperature glide for the condenser heat exchanger is not as clear as for the evaporator heat exchanger. Over the same coil section, R-22 shows a 2°F drop because of the pressure drop. The mixture, however, shows a 6°F drop.

3. The amount of heat exchanged by the evaporator with the mixture is not as high as with R-22, a drop of around 7.5% at 95°F ambient under nonflooded coil condition. In flooded coil tests, the cooling capacity increases for both R-22 and the mixture: R-22 has an increase of over 8%, while the mixture an increase of over 15%. The capacity difference between R-22 and the mixture under flooded evaporator operation is reduced (less than 2%).

4. Refrigerant mass flow rate increases with ambient temperature. Flooded coil operation increases the mass flow rate for both R-22 and the mixture. R-22 has a 17% higher mass flow rate than the mixturer at 95°F ambient and under nonflooded evaporator operation. For the flooded evaporator test, the mass flow rate of R-22 is still about 10% higher than that of the mixture. When the evaporator is flooded, the mass flow rate of the mixture increases from 305 lb/h to 370 lb/h, and that of R-22 increases from 364 lb/h to 412 lb/h.

5. R-22 requires more than 3% higher compressor power consumption than the mixture at 95°F ambient. Flooded evaporator operation has little effect on power consumption for R-22 for ambient temperature below 105°F. For the mixture, flooded coil operation increases the power consumption by about 4%.

6. The mixture has a higher compressor discharge pressure than R-22, and the difference is broader at higher ambient temperatures. At a low ambient, 90°F or lower, the discharge pressures for the mixture and R-22 are almost identical. Flooded coil operation lowers the discharge pressure for R-22 but increases it for the mixture.

7. The mixture has a lower compressor discharge temperature than R-22. Flooded coil operation lowers the discharge temperature for both refrigerants by about 20°F.

8. The mixture has a higher compressor high-low pressure ratio than does R-22. Flooded coil operation reduces the ratio by 11.8% for the mixture and by 13% for R-22.

The test results reveal that the overall performance of the mixture is close to that of R-22, but it falls behind in heat exchanger cooling capacity and results in higher compressor discharge pressure. One reason could be that the components used in the test are all designed for R-22 application. Modification of the components for the mixture operation would improve the performance; counter-cross-flow evaporator and condenser coils and a better expansion device are examples of needed changes. A counter-cross-flow evaporator heat exchanger would take advantage of the temperature glide and probably increase the temperature glide as well.

Flooded evaporator coils have a positive effect on both R-22 and the mixture operation. The test results show that flooded coil operation affects the mixture more than R-22. Operating with flooded evaporator coils, the mixture actually performs better than R-22 performs with nonflooded evaporator coils, in terms of evaporator cooling capacity, refrigerant mass flow rate, and compressor high-low pressure ratio.

6. RECOMMENDATIONS FOR FUTURE WORK

As the experiment was going on, Du Pont developed a new formula for the mixture we tested. The original composition of 30% R-32, 10% R-125, and 60% R-134a has been changed to 23% R-32, 25% R-125, and 52% R-134a to reduce flammability of the mixture. We recommend testing the updated ternary mixture refrigerant with concurrent-cross-flow and counter-cross-flow evaporator heat exchangers (Kuo, 1994). Because with flooded evaporator operation, even with a concurrent-cross-flow evaporator, the cooling capacity of the mixture is only 2% below that of R-22 at the same operating conditions. It would be interesting to test the mixture with a counter-cross-flow evaporator at flooded evaporator condition.

It could possibly outperform R-22 under those conditions. The recommended test will provide important information about the performance of this mixture using an off-the-shelf evaporator heat exchanger. Proper modification of the heat exchanger design to suit the mixture application could be derived from the test results.

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