CYCLE PERFORMANCE TESTING OF NONAZEOTROPIC MIXTURES OF HFC-143a/HCFC-124 AND HFC-32/HCFC-124 WITH ENHANCED SURFACE HEAT EXCHANGERS

E.A. Vineyard, P.E. J.C. Conklin, Ph.D., P.E. A.J. Brown

ABSTRACT

In an effort to improve the efficiency of residential heat pumps using alternative refrigerants, two nonazeotropic refrigerant mixtures (NARMs) were tested over a range of heat exchanger capacities to determine their cooling mode performance at U.S. Department of Energy (DOE) heat pump rating conditions of 82°F (27.8°C). The two mixtures, HFC-32/HCFC-124 (30%/70%) and HFC-143a/HCFC-124 (75%/25%), were selected on the basis of a previous study that screened refrigerant pairs using such factors as boiling point, stability, ozone-depletion potential (ODP), and coefficient of performance (COP) to determine suitable candidates for residential heat pump performance. Three refrigerant-side heat transfer enhancements were tested to determine their effects on overall system performance. Comparisons were made on the basis of COP as a function of capacity. The results for one of the heat exchanger combinations, a segmented evaporator and finned condenser, were quite promising. Improvements in COP, relative to that for R-22 (an HCFC), ranged from 9% to 17% for the HFC-32/HCFC-124 (30%/70%) mixture and from 5% to 9% for the HFC-143a/HCFC-124 (75%/25%) NARM. Another combination, a smooth tube evaporator with a perforated foil insert and finned condenser, had similar gains at low capacities but experienced decreased performance at the higher capacities. The final combination, a smooth tube evaporator with a perforated foil insert and smooth tube condenser with a bent-tab insert, resulted in poor performance.

INTRODUCTION

In the early 1970s, scientists first presented the theory that chlorofluorocarbons (CFCs) slowly migrate into the stratosphere where sunlight can split off free chlorine molecules that react with ozone and reduce its concentration (Molina and Rowland 1974). This theory, along with other findings, led the global community to ratify the Montreal Protocol, a landmark agreement to protect the stratospheric ozone layer from emissions of chlorinated and brominated compounds (UNEP 1987). The agreement calls for a reassessment every four years. However, with scientific facts indicating that ozone depletion is increasing, the first reassessment was moved up two years and released in 1989. Based on the results of the 1989 study, contracting parties to the Montreal Protocol accelerated the phaseout of CFCs. The modified agreement called for a total phaseout by the year 2000 instead of the original 50% reduction based on 1986 levels (UNEP 1991).

The emerging issue of greenhouse gases and their effects on the atmosphere has recently received greater attention following the release of scientific data by the United Nations Environment Programme and World Meteorological Organization that show carbon dioxide to be the main contributor to increased global warming (UNEP 1991). In response to the report, some refrigerant suppliers plan to limit the production of those hydrochlorofluorocarbons (HCFCs) with an environmental impact comparable to or greater than HCFC-22 (*ACHR News* 1991). In addition, a climate change treaty addressing phaseout periods for HCFCs was signed by several heads of state at the United Nations Conference on the Environment and Development in June 1992.

The twin issues of ozone depletion and global warming have sparked an intensive search for replacement refrigerants for use by the heat pump, refrigeration, and air-conditioning industries. These substitutes may be in the form of pure fluids or mixtures. Stringent requirements for replacement refrigerants have eliminated a large number of potential fluids from consideration. Some of these requirements are that the refrigerant must be nontoxic, stable, compatible with lubricating oils, similar in thermodynamic performance, and available at reasonable cost. Although not a requirement, most manufacturers prefer nonflammable alternatives.

Replacement refrigerants are currently being investigated at several national laboratories, in private industry, and through government-industry consortia. NARMs, one class of refrigerants, exhibit a unique characteristic in that the saturated temperature at constant pressure is a function of the vapor quality, that is, the phase-change process is not isothermal. The change in temperature as the NARM changes phase is called the *temperature glide*. This noniso-

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thermal phase-change behavior can improve heat exchanger effectiveness as well as thermodynamic cycle efficiencies, as shown in a previous study (McLinden and Radermacher 1987). However, investigators have measured heat transfer coefficients for binary NARMs in smooth tubes that were less than those for either of the pure constituent refrigerants (Jung et al. 1989). Additional heat transfer area for NARM heat exchangers would therefore be required to achieve the same heat exchanger capacity.

In an initial effort to determine the potential for improving heat pump system performance with NARMs, investigators performed tests in a breadboard heat pump apparatus with enhanced-surface counterflow heat exchangers to take advantage of the gliding temperature differences of the heat transfer fluid (Kauffeld 1990). The results of the study showed a 32% improvement in COP with a mixture of HCFC-22/CFC-114 (65%/35%), an alternative that is presently unacceptable. Temperature glides for the tests were 25°F (13.9°C) for the evaporator and 35°F (19.4°C) for the condenser. Building on that effort, a project was initiated to determine system performance for environmentally acceptable substitutes using different heat transfer surfaces at temperature glides that more closely match those found in present heat pump systems. Initial test results from the project, while showing improved system COPs with a mixture of HFC-143a/HCFC-124 (75%/25%) at low capacities, revealed that additional heat exchanger improvement was necessary if NARMs were to be a realistic alternative refrigerant for higher capacities (Vineyard and Conklin 1991).

Attempting to further improve system COP, two additional methods of heat transfer enhancement were explored, inserts and a segmented evaporator. The purpose of inserts was to improve the heat transfer with a minimal effect on pressure drop. The segmented evaporator was an attempt to enhance different portions of the heat exchanger to match the mode of heat transfer that was taking place in each section.

The non-CFC refrigerants R-143a and R-32 (both HFCs) and R-124 (an HCFC) were selected as mixture components based on previous research that screened refrigerants to determine suitable candidates for residential heat pump operation (Vineyard 1989). Following a series of tests to determine the optimal concentration based on COP, mixtures of HFC-143/HCFC-124 (75%/25%) and HFC-32/HCFC-124 (30%/70%) (percentages by mass) were tested at various capacities. R-22 (an HCFC) was also tested over the same capacity range as the NARMs so that the effects of surface performance on mixtures could be compared to that of a pure fluid.

APPARATUS

The test system, shown in Figure 1, is a highly instrumented test loop with the following components: a variable-speed compressor (500 to 3000 rpm) that allows for variable heat exchanger loadings, a variable-orifice refrigerant metering device (needle valve), and two sets of counterflow concentric-tube heat exchangers. The manually operated needle valve provides a degree of flexibility in controlling refrigerant conditions that is lacking in capillary tubes and thermostatically controlled expansion devices.

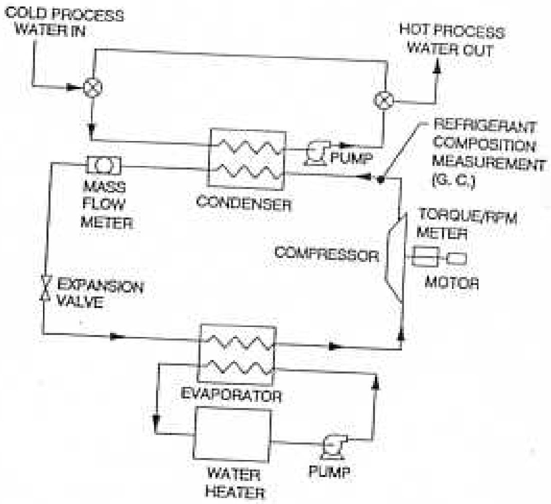

Figure 1 *Test loop.*

HEAT EXCHANGER CONFIGURATIONS

Two sets of heat exchangers were mounted side-by-side on the rig, each with a different enhancement on both the refrigerant and water sides (descriptions to follow). This enabled testing of different combinations of heat exchangers without extensive down time due to the discharging, recircuiting, and recharging. The surfaces were selected on the basis of manufacturer and heat transfer consultant recommendations for each heat exchanger. The heat exchangers were instrumented as shown in Figure 2, which allowed for heat transfer parameters to be determined for different tube sections.

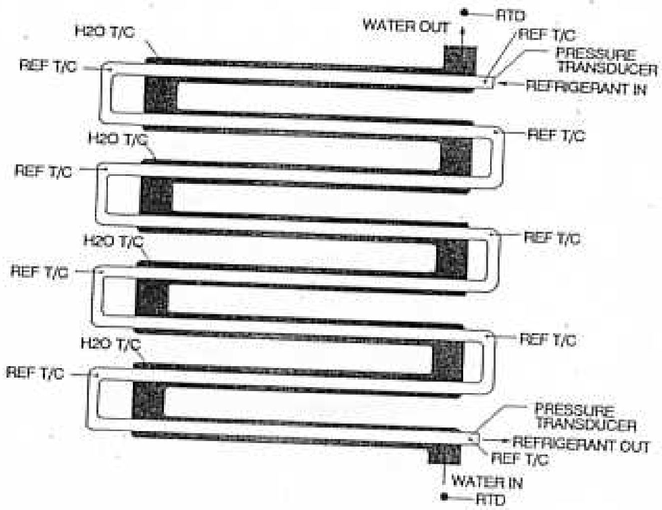

Figure 2 *Heat exchanger schematic.*

Smooth Surface

The smooth-tube heat exchangers consist of a center tube and an outer tube forming an annulus. The center tubes are copper with an outer diameter of 0.625 in. (15.88 mm) and an inner diameter of 0.545 in. (13.85 mm). There are eight horizontal passes, and the outer surfaces of the water annulus and refrigerant bends are insulated. Each horizontal section of the evaporator is 9.02 ft (2.75 m) long. The condenser section is 6.36 ft (1.94 m) long.

Inserts

There are only a limited number of publications that contain information on heat transfer enhancement with NARMs. Therefore, the most successful enhancement techniques for pure refrigerants were selected to determine which ones showed the most promise with NARMs. One augmentation, the perforated brass foil, has been used to enhance heat transfer from the outside surface of a tube in pool boiling and from the inside surface of a tube in flow boiling (Conklin and Vineyard 1992). The surface can be used as a retrofit heat transfer enhancement device for existing evaporators. The actual increase in heat transfer varies with application, hole geometry, and pure refrigerant (Palm 1991). For R-22, the heat transfer coefficient was doubled. Thus, foil inserts were a promising candidate for enhancing the evaporator heat transfer coefficients with NARMs.

The brass foils, 0.002 in. (0.05 mm) thick, were perforated by a simple method resulting in evenly spaced holes with a density of 323 holes per square inch (50 holes per square centimeter). The extruded material from the perforation operation has a height of approximately 0.009 in. (0.23 mm). The side of the foil with the extruded material was placed against the smooth tube wall in the evaporator as shown in Figure 3. The foils are held against the wall by the stiffness of the brass. Because the perforated foils were designed to improve nucleate boiling, they were only used in the first half of the evaporator where the quality would be lowest. Due to accessibility constraints, the foils were not inserted into the first pass.

Although there were no references to the use of bent tabs for condensation, they have shown 300% improvement in convective heat transfer for fluid streams with a low Reynolds number (Junkhan 1988). Bent tabs were tested in the condenser in an attempt to deflect the vapor in the center of the tube to the wall, which is desirable for condensing heat transfer. The bent tabs, shown in Figure 4, were fabricated from 0.0315 in. (0.8 mm) thick aluminum that was cut into strips 0.525 in. (13.33 mm) wide and 9.02 ft (2.75 m) long. The bent tabs were inserted in passes 2 through 7 of the smooth-tube condenser. Inserts in the first and last pass were unnecessary since those sections of the condenser are superheated and subcooled, respectively.

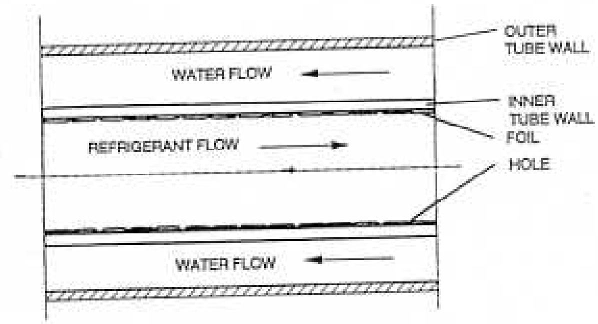

Figure 3 *Evaporator tube with perforated foil insert.*

Finned Surface

The finned-tube surface, a commercially available product shown in Figure 5, has 10 extruded spiral ridges on the inside tube surface with a fin height of 0.015 in. (0.38 mm). The tube material is copper with an outer diameter of 0.622 in. (15.8 mm) and an inner diameter of 0.539 in. (13.7 mm). The extruded ridges have a 47° helix angle with respect to the longitudinal axis of the tube centerline. The surface roughness is 32 microinches (0.8 micrometers). The outer surface of the refrigerant tube has closely spaced fins that increase the heat transfer to the water.

Segmented Evaporator

The segmented evaporator, shown in Figure 6, is composed of two sections with different surfaces. The surface in passes 1 through 4 is designed to promote

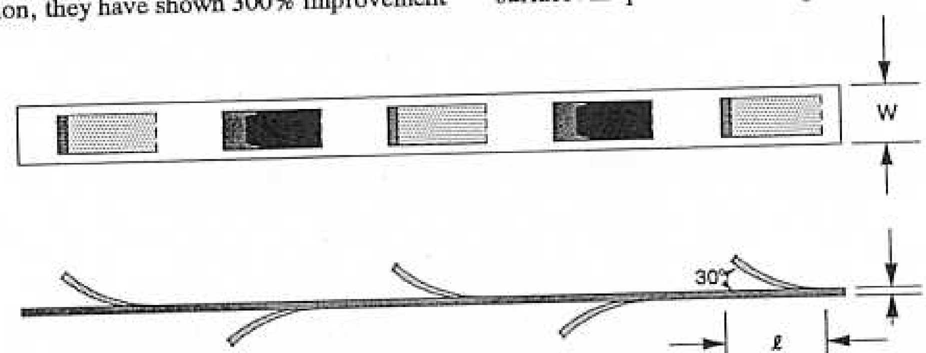

Figure 4 *Bent-tab insert.*

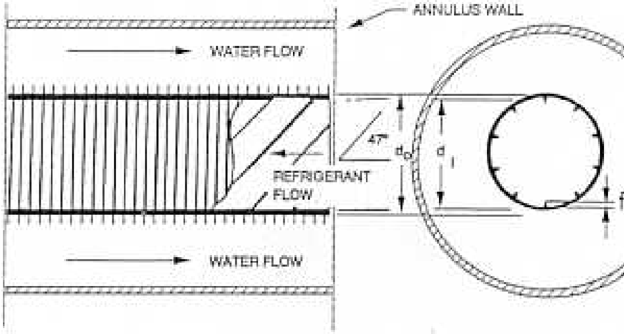

Figure 5 Finned surface.

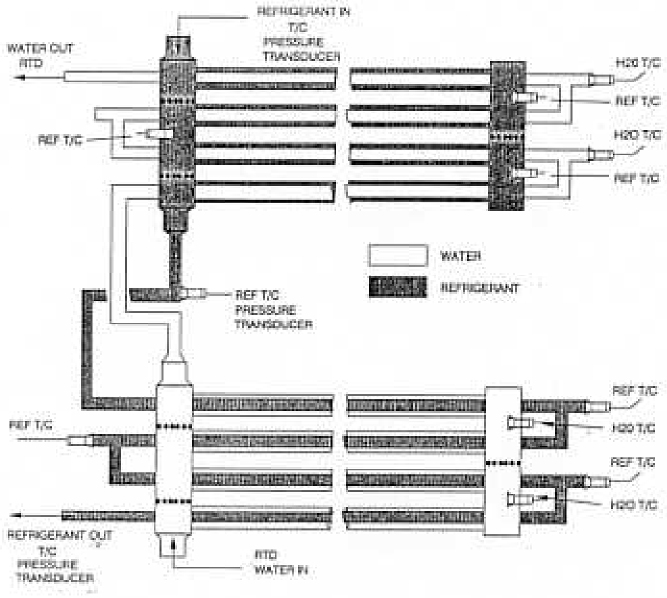

Figure 6 Segmented evaporator.

nucleate boiling in the low-quality region of the evaporator. Refrigerant flows in the annulus of the tube where it contacts a rolled-fin surface that is manufactured by raising integral low fins, cutting diagonally across these fins, and then rolling the fins to compress them to form mushroom-like pedestals. Re-entrant passageways are thus formed in a rectangular crosshatch pattern. The inner surface, where water flows, has extruded spiral ridges similar to the finned-surface tube described above. Figure 7 depicts a cross section of the rolled-fin surface.

The second section of the segmented evaporator, passes 5 through 8, has a finned-tube surface as described previously. In this section, where convection heat transfer is the dominant mode, refrigerant flows inside the tube and water flows in the annulus.

ALTERNATIVE REFRIGERANTS

On a global basis, air conditioners and heat pumps contain approximately 330,000 metric tons of R-22 (UNEP 1991). Essentially all are electrically driven vapor-pression units with hermetic compressors. The high de and thermal efficiency of R-22 have resulted in effective products compared to non-vapor-compres cycles or products using lower pressure refrigerants. 1992 revision of the Montreal Protocol calls for HCF essentially be phased out by 2020. In addition, the relat high direct GWP (1500) of R-22 may result in an accel ed phaseout if only direct global warming effects considered.

R-32

R-32 is considered to be a possible replacement fo 22 since its normal boiling point is in the same ra However, its high pressure would probably restric consideration only for use as a component for a bl From an environmental standpoint, it would be an exce replacement since it has zero ODP and a relatively direct GWP (640) (IPCC 1990). R-32 is modera flammable, with flammability limits in air of 14% (lo and 31% (upper).

R-124

R-124 has been suggested for use in systems prese using R-114, such as centrifugal chillers. Although it chlorine-containing compound, its ODP (0.02) is less half that of R-22. Like R-32, it has a low direct GWP (that makes it an excellent constituent for mixtures w attempting to lower the overall GWP for a binary refri ant. Based on limited toxicity testing, a threshold value of 500 ppm is estimated.

R-143a

R-143a, like R-32, has been mentioned as a poss replacement for R-22. Its normal boiling point of −53.

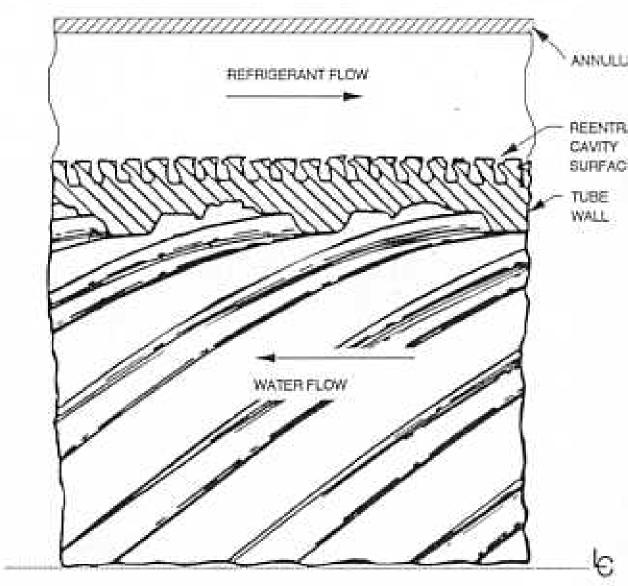

Figure 7 Rolled-fin surface.

(−47.6°C) is closer to that of R-22 than R-32. In addition, it has lower vapor pressure than R-32 and could possibly be used a pure replacement, although it would still have higher pressures than R-22. R-143a is also flammable, with a lower flammability limit of 9.2% and an upper flammability limit of 18.4% in air. Since it contains no chlorine, the ODP is zero. One of the potential drawbacks to R-143a is its high direct GWP, which is 2900.

TESTING PROCEDURE

The output of instrumentation, such as mass flowmeters, pressure transducers, and the thermocouples and resistance temperature detectors (RTDs), was monitored on a minicomputer. In addition, a dedicated gas chromatograph, located at the compressor exit, was used to measure the composition of the circulating charge in real time. Data were taken every 15 seconds and averaged over a 15-minute period during steady-state operation. All pressure transducers, flowmeters, thermocouples, and RTDs were calibrated prior to testing.

Water flow rates to the heat exchangers were pneumatically controlled to allow for rapid changes in test conditions. Inlet temperatures to the condenser are maintained with a mixing valve that introduces cool process water while bleeding off warmer water that has passed through the condenser. Inlet water temperatures to the evaporator are controlled by a bank of resistance heaters in the closed-loop system.

Test conditions were based on the U.S. DOE standard rating conditions for an air-source heat pump. Inlet water temperatures were set to values that yielded equivalent R-22 saturation pressures as those determined from previous testing with an air-source system at a national laboratory (Miller 1989). For all the tests, the refrigerants enter the evaporator at an approximate 25% quality and are completely evaporated, leaving with a slight amount of superheat.

The heat rates for the refrigerant side and the water side are determined from measured inlet and outlet temperatures and measured flow rates and then checked for a steady-state heat balance. Measured electrical power to the heaters and pump in the closed-loop evaporator was also used to check for a heat balance in steady-state operation. The necessary refrigerant thermophysical properties were determined from algorithms using the Carnahan-Starling-DeSantis equation of state (Morrison and McLinden 1986). During the testing, minimal subcooling at the condenser exit and minimal superheating at the evaporator exit were maintained to ensure two-phase conditions throughout most of the heat exchanger length.

An alkylbenzene oil was used for lubrication of the compressor for both mixtures and R-22. The concentration of oil in the circulating refrigerant was on the order of 1% by weight, based on manufacturers' data. The effect of oil on the heat transfer coefficient or pressure drop was not investigated.

RESULTS

Performance results, consisting of a plot of COP versus capacity, are shown in Figures 8 and 9. Four tube surface combinations (condenser and evaporator), including the smooth tubes, were tested. The designation describing the tube surfaces in the figures gives the evaporator first and the condenser second. Where only one surface is specified, the same surface was used for both heat exchangers. Varying amounts of pressure drop are associated with each surface, depending on factors such as the number of fins, fin height, angle, and surface roughness.

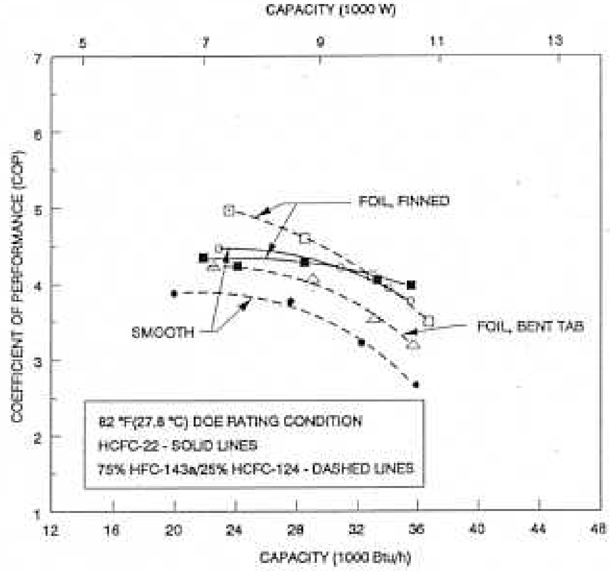

Figure 8 Refrigerant performance curves—inserts.

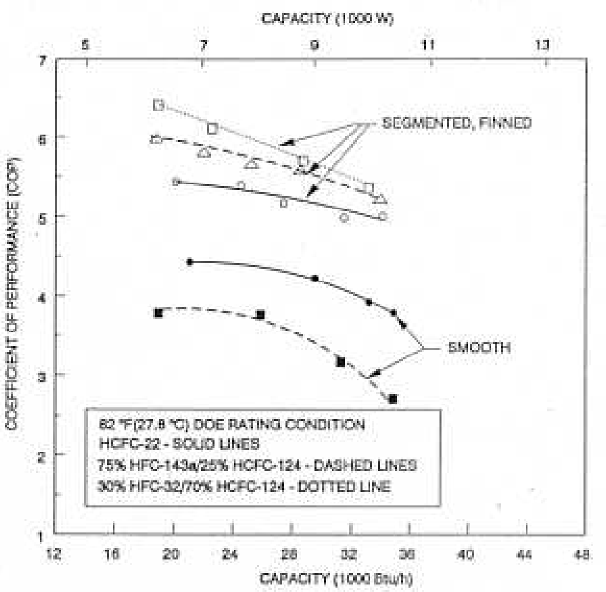

Figure 9 Refrigerant performance curves—segmented evaporator.

COPs were calculated by dividing the evaporator capacity by the input power to the compressor. Evaporator capacity was determined from a measurement of the electrical energy input to the heaters and pump, both of which are located in an insulated box along with the evaporator. Additional heat balances were calculated for the water and refrigerant sides but were determined to be less accurate. The compressor input power was calculated by multiplying the compressor speed by the torque.

Smooth Surface

The baseline smooth-tube performance for the NARM, as shown in Figure 8, was significantly less, 13% to 27%, than that of R-22 over the capacity range tested. This result was expected since NARMs have inherently lower heat transfer coefficients.

Inserts

The first combination that was investigated for improving the smooth-tube performance was the perforated foil evaporator and bent-tab condenser. For the insert tests, HFC-143a/HCFC-124 (75%/25%) was the only NARM tested, along with the R-22. Experimental results with this particular combination showed a large pressure drop in the condenser as a result of the bent-tab insert. The pressure drop was so large that there was no improvement in COP compared to smooth-tube performance.

Insert-Finned A second series of tests was performed using a finned surface in the condenser and foil inserts in the evaporator. The results were more promising, as indicated in Figure 8. The foil-finned arrangement effected a 15% improvement in COP over that for R-22 at the lowest capacity. At the highest capacity, the NARM had a 6% lower COP than R-22 with the foil evaporator/finned condenser combination. The decrease in NARM performance as the heat exchangers became more fully loaded is the result of the increased pressure drop at higher flow rates. The pressure drop negates the NARM gliding temperature difference that enables a matching of the secondary heat transfer fluid temperature glide.

Segmented Evaporator-Finned Two NARMs, HFC-143a/HCFC-124 (75%/25%) and HFC-32/HCFC-124 (30%/70%), in addition to the baseline R-22, were tested in the segmented evaporator and finned condenser. The results for the NARMs, shown in Figure 9, were very encouraging. The improvements in COP over R-22 for the HFC-143a/HCFC-124 (75%/25%) mixture ranged from 5% to 9%, depending on the capacity. Results for the HFC-32/HCFC-124 (30%/70%) NARM showed even more improvement. At the higher capacity, a 9% improvement in COP was realized, while the lower capacity realized a 17% improvement.

The curves in Figure 9 indicate that the heat exchanger surface enhancements selected for the segmented evaporator improved the NARM performance more than that of R
This same trend, although not as pronounced, is seen in
results for the insert tests in Figure 8. Comparisons of
relative improvements over smooth-tube performance
the segmented evaporator/finned tube combination show
the COP for R-22 improves by 22% to 27% over the ra
of capacities. COPs for the HFC-143a/HCFC-
(75%/25%) mixture show a much larger improvement.
low capacities, a 62% improvement is realized. At
higher capacities, even larger COP increases of appro
mately 80% are obtained.

DISCUSSION

Previously, system performance with an HFC-14
HCFC-124 (75%/25%) mixture was shown to improve
low capacities using a fluted-surface evaporator and
finned-surface condenser (Vineyard and Conklin 199
Analyses of the heat transfer coefficients performed
conjunction with the system tests confirmed that for
evaporator and condenser surfaces tested, those with
highest heat transfer coefficients yielded the best syst
COPs (Conklin and Vineyard 1990, 1991). The h
transfer analyses also revealed that the fluted-surfa
evaporator heat transfer coefficients for NARMs were t
to three times less than those for R-22, while the finne
surface condenser coefficients were close to the same
both the mixture and R-22. On the basis of that inform
tion, follow-up work concentrated on improving he
transfer in the evaporator while focusing less on t
condenser.

For the insert tests, the optimum configuration was t
foil-finned tube combination, as indicated in Figure
Compared to results from an earlier study where a flute
finned-tube combination was used, the foil-finned arrang
ment did not perform as well (Vineyard and Conklin 1991
A comparison of heat transfer coefficients reveals that t
fluted surface has a much higher value than that of the fo
insert. Furthermore, the foil inserts have virtually no effe
on the heat transfer coefficient compared to smooth-tu
performance because the refrigerant quality of the NAR
entering the evaporator was too high for nucleate boilin
the regime in which the foil inserts are designed to enhanc
heat transfer, to occur (Conklin and Vineyard 1992
Therefore, the conclusion is that the performance improv
ment at low capacities for the foil-finned combination is th
result of improvements in the condenser.

The optimum configuration in all testing to date is th
segmented evaporator and finned condenser. This particula
combination combines the best condenser that was previous
ly investigated with the best evaporator from this effor
While heat transfer analysis has not been performed on th
segmented evaporator, it appears that there is much im
provement in the heat transfer coefficients based on prelimi
nary plots of the temperature profiles and the large increas
in COP.

CONCLUSIONS

The following conclusions apply only to the NARM test loop under the test conditions that were performed and with the specified mixtures. The results should not be construed as being applicable to other refrigeration systems, such as refrigerator-freezers or automotive air conditioning. Also, mixtures and concentrations with different gliding temperature differences and curvatures of the enthalpy-temperature line may yield different performance values.

- Significant system performance improvements, relative to R-22, can be realized with NARMs by using surface enhancements similar to those for the segmented evaporator and finned condenser.
- The R-32 NARM gave nearly twice as much performance improvement over the entire range of capacities that were tested, as that of the R-143a NARM.
- The inserts tested in this study were ineffective at improving the system COPs for NARMs.

ACKNOWLEDGMENT

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