HEAT PUMPS IN COLD CLIMATES

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KEY WORDS

Air conditioner, heat pump, residential, modeling, steady-state, cooling, heating, energy efficiency ratio, R-410A, R-134a, R-22, propane

ABSTRACT

This computer program is a public-domain system design tool for application to air-to-air heat pumps. The main aspects of the program are reviewed with emphasis on the newest features of the current fifth-generation version (Mark V) and an upcoming more fully HFC-capable release (Mark VI). Current model predictions are compared to test data for a leading HFC alternative to HCFC-22 in heat pumps. Examples are shown of some user interfaces that have been recently developed for the program.

To demonstrate the design capabilities of the model for R-22 alternatives, a refrigerant-side optimization was conducted to find the best balance of heat transfer versus pressure drop for HCFC R-22, HFCs R-134a and R-410A, and the natural refrigerant propane. COP was maximized while refrigerant charge and tube size were minimized. Independent design parameters were fraction of total area in the outdoor coil, tube diameter and number of circuits for each heat exchanger, and condenser subcooling. Heat exchanger design tradeoffs are discussed for a heat pump relative to air conditioners and heating-only units. A design optimized for heating-only operation is presented.

INTRODUCTION

History of DOE/ORNL Model

The DOE/ORNL Model is a publicly available system design tool developed for research into heat pump performance improvement. The program has seen a number of version releases, as shown in Figure 1, from its origin at MIT where basic models for compressors and heat exchangers were developed by Hiller and Glicksman (1976). Ellison and Creswick combined these FORTRAN programs and produced the first ORNL Version (1978). The program was publicly released as the Mark I version (Fischer and Rice 1983). The first PC version was Mark III in 1985 (Fischer et al, 1988), followed by an initial variable-speed version with charge inventory capability (Rice 1988). The variable-speed model was completed in Mark IV (Rice 1991) with the addition of electronically commutated motors (ECMs) and design parametric capability. Pure, azeotropic, and near-azeotropic HFC refrigerants were added in Mark V (Rice and Jackson 1994). Work is underway on Mark VI which will include R-407C and various heat exchanger circuiting configurations. The informal names for the newer models are shown in parentheses in Figure 1.

The distribution version of the model has been PC-DOS-based since Mark III. Inter-City Products, a leading U.S. heat pump manufacturer, developed an interface for Digital Equipment VAX computers in the mid-80s and made further changes to use the program as their primary system design tool (DeVos 1993). Trane developed an expert system interface to use the program for interactive heat exchanger design (Bergt 1989). More recently, Copeland and ORNL are developing more user-friendly Windows and Web interfaces that are briefly discussed later.

The program has also served as a referenceable foundation for modeling alternative-fuel and -source (engine-driven, ground- and water-source) heat pump systems. Hughes (1985) worked with Catan and Baxter (1985) to cost-optimize water- and ground-source heat pumps using the Mark II version. Monahan et al (1987) used the Mark III program to simulate a kinematic Stirling-engine-driven unit and Battelle (Fischer et al, 1987) modeled an IC-engine-driven design that became the York Triathlon heat pump (GRI 1994, Nowakowski 1995).

Lawrence Berkeley Laboratory adopted the Mark IV version in 1992 as their engineering model for use in evaluating minimum efficiency standards for heat pumps and air conditioners (Rosenquist 1997). The model has also been adapted for use by the Australian manufacturers association AREMA in their unitary certification program (Morrison, 1993)

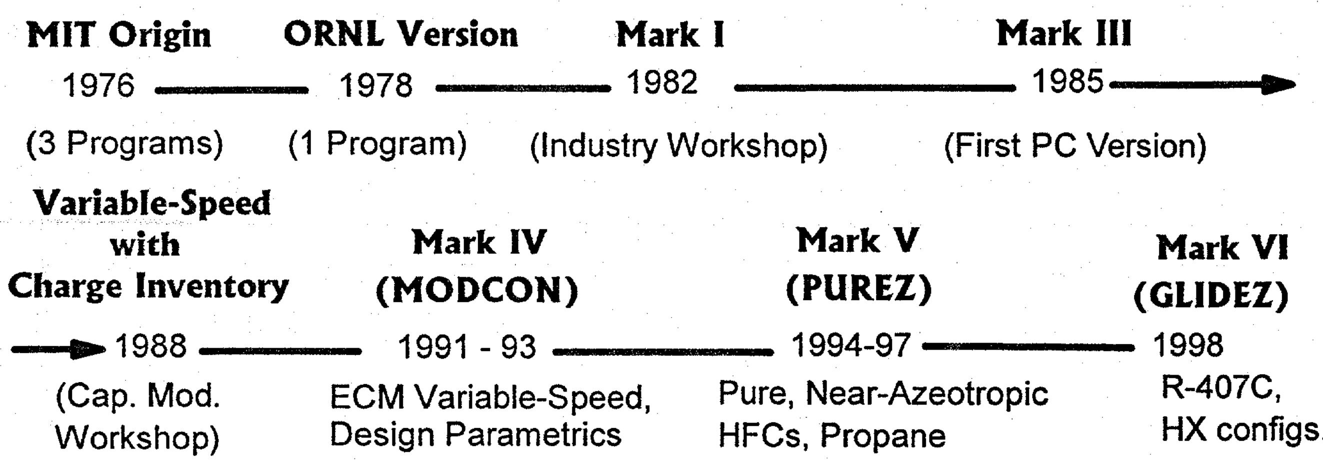

Figure 1. History of DOE/ORNL Heat Pump Design Model

Key Features of Current and Upcoming Versions

Component Modeling Capabilities

The DOE/ORNL Heat Pump Design Model predicts the steady-state cooling and heating performance of electric-driven vapor compression heat pumps. Transient (cyclic or frosting/defrosting) effects are not considered. The program has physically based heat transfer models for each single- and two-phase refrigerant region of fin-and-tube, air-to-refrigerant heat exchangers. Simple parallel refrigerant circuiting is assumed. Air-side dehumidification and evaporator sensible-heat-ratios (SHRs) are calculated.

Single- and variable-speed compressors and fans can be simulated. Compressors are modeled using curve-fit representations to manufacturers' performance map data; fan power can be specified or calculated from fan efficiency and air-side pressure drop models. A full range of flow control devices are available including capillary tubes, short-tube orifices, TXVs, TEVs, or specified condenser subcooling. The short-tube orifice models have been recently upgraded using the work of Kim and O'Neal (1994a, 1994b) to include refrigerant-specific models for R-134a and R-22. Refrigerant charge requirements can be calculated at design conditions or specified for determining off-design performance.

Refrigerant Capabilities

In the presently available Mark V program, there are six chlorine-containing refrigerants—CFCs R-12, R-114, and R-502, and HCFCs R-22, R-123, and R-124. There are eight HFCs suitable as R-22 or R-502 alternatives—the single component refrigerants R-134a, R-32, R-125, R-143a, and R-152a—and the azeotropic or near-azeotropic refrigerants R-410A, R-507, and R-404A. One hydrocarbon, R-290 (propane), completes the list.

The thermodynamic properties of these refrigerants are modeled with either the Martin-Hou (1955) or Downing (1974) equation-of state representations as provided by U.S. manufacturers Allied-Signal (1993, 1995) or DuPont, respectively. The transport properties are from ASHRAE (1976) or from the U.S. manufacturers of the newer HCFCs and the HFCs (DuPont 1993 and Zheng 1994).

For the upcoming Mark VI version, we have incorporated improved EOS representations for R-410A (DuPont 1996) based on an extended Martin-Hou approach from Bivens and Yokozeki (1996). Similar representations are included for R-410B (Yokozeki 1995) and R-407C (Chitti 1995) with transport properties provided by DuPont (1994, 1995). Additional modifications to the heat exchanger routines are underway to account for the effect of the refrigerant glide of R-407C in the two-phase region. Cross-parallel and cross-counterflow circuiting configurations are being added to supplement the existing cross-flow correlations.

We also have plans to add generalized capillary tube and short-tube orifice equations more suitable for use with R-410A, R-407C, and all the other refrigerants that are available in the program. Presently the program defaults to R-22 correlations when refrigerant-specific correlations are not available.

New Interfaces

Copeland recently added a Windows 95™ interface on the Mark V program with their COPESIM-HP™ version (1997) to aid international customers in system applications of their compressors. Figure 2 shows the main screen from the Windows interface. The input choices can be viewed and modified by selecting the components or the connecting lines on the diagram. Copeland's version provides database access to their full line of reciprocating and scroll compressors from 5.27 to 52.7 kW (1.5 to 15 tons). The compressor database screen is shown in Figure 3. Users can search for the compresssor size, type, phase, line frequency, for either R-22 or R-410A refrigerants and available models will be displayed for selection.

At ORNL, we are currently working on an Internet-Web-capable interface for the program which will allow users to run single cases or design parametrics remotely from any machine capable of running standard Web browsers, e.g., a PC running Windows 3.1, 95, or NT; a Macintosh; or a Unix machine. We have recently made the program operational on the Web for single cases; however, of the interface elements, only the output summary screen is complete at present. An example of this screen is given in Figure 4 for an R-410A heat pump at the high temperature heating rating condition. In a future version, we hope to include the capability of online x-y and contour plots of users' design parametrics analysis.

Design Capabilities

The model has some special features that make it uniquely capable of performing certain types of design analyses. One- or two-variable parametrics can be performed for a wide range of design, control, and operating values. These include flow rate and ambient parameters and heat exchanger design variables. For many of the heat exchanger (HX) variables such as fin pitch, no. of tube rows, face area, or total area fraction, the total finned HX area on one or both coils can be held constant by having the model simultaneously adjust, in a prescribed manner, some or all of the remaining parameters related to finned area. In some cases, the adjustments can be made so that fan power requirements are held constant. Such options were used in this analysis to hold total finned area on both coils constant while adjusting the fraction of finned area in the outdoor coil and maintaining constant fan power requirements.

The user's choice of over 100 dependent output values can be monitored as the independent design or operating parameters are varied. The desired output values are provided in a table that can be easily imported into a spreadsheet or 2- or 3-D graphics packages for viewing as surface contours or 3-D images. Examples of contour surfaces are shown later in this paper.

The heat pump output capacity (cooling or heating) can be held constant while design parameters are varied. This is a useful option at design point conditions where it is often desirable to maintain a design capacity while heat exchanger parameters are studied. The specified capacity is maintained by the model by adjusting the compressor displacement as needed at each parametric combination. In addition to this autosizing capability for the compressor

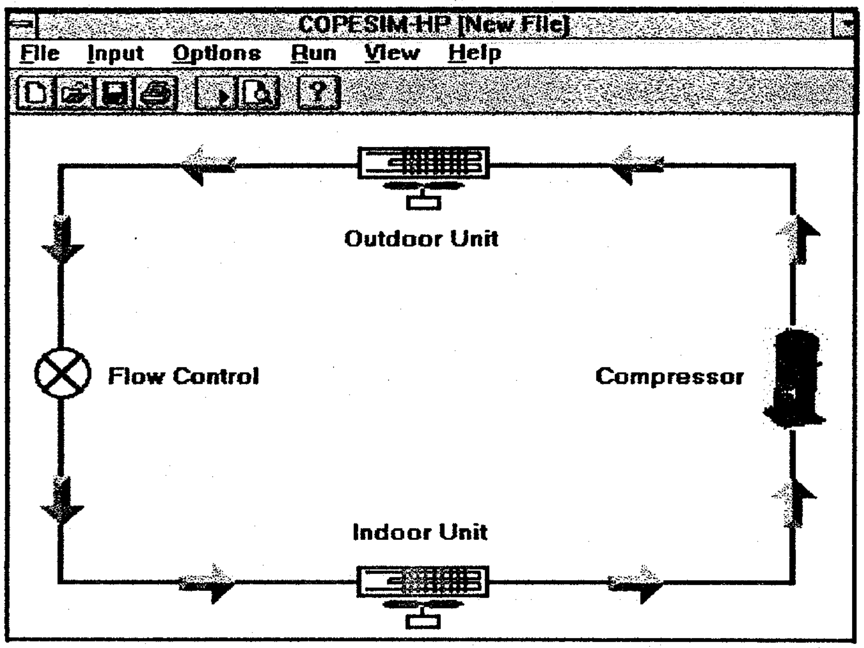

Figure 2. Example of Main Screen from COPESIM-HP™ Windows Interface to the DOE/ORNL Heat Pump Design Model

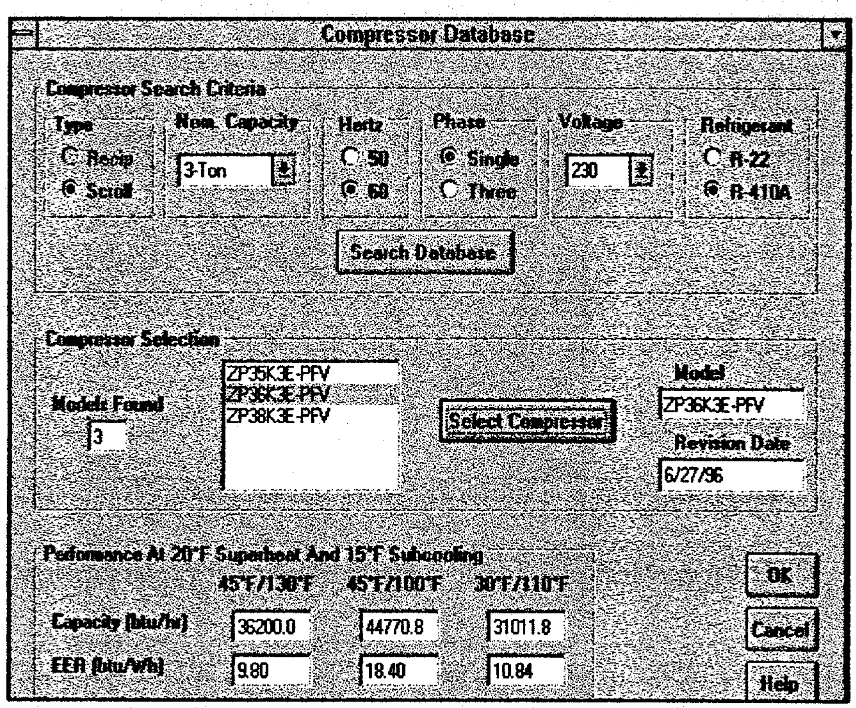

Figure 3. Example of Compressor Database Screen from COPESIM-HP™ Windows Interface to the DOE/ORNL Heat Pump Design Model

--High Temp. Heating Condition, HFC Heat Pump--

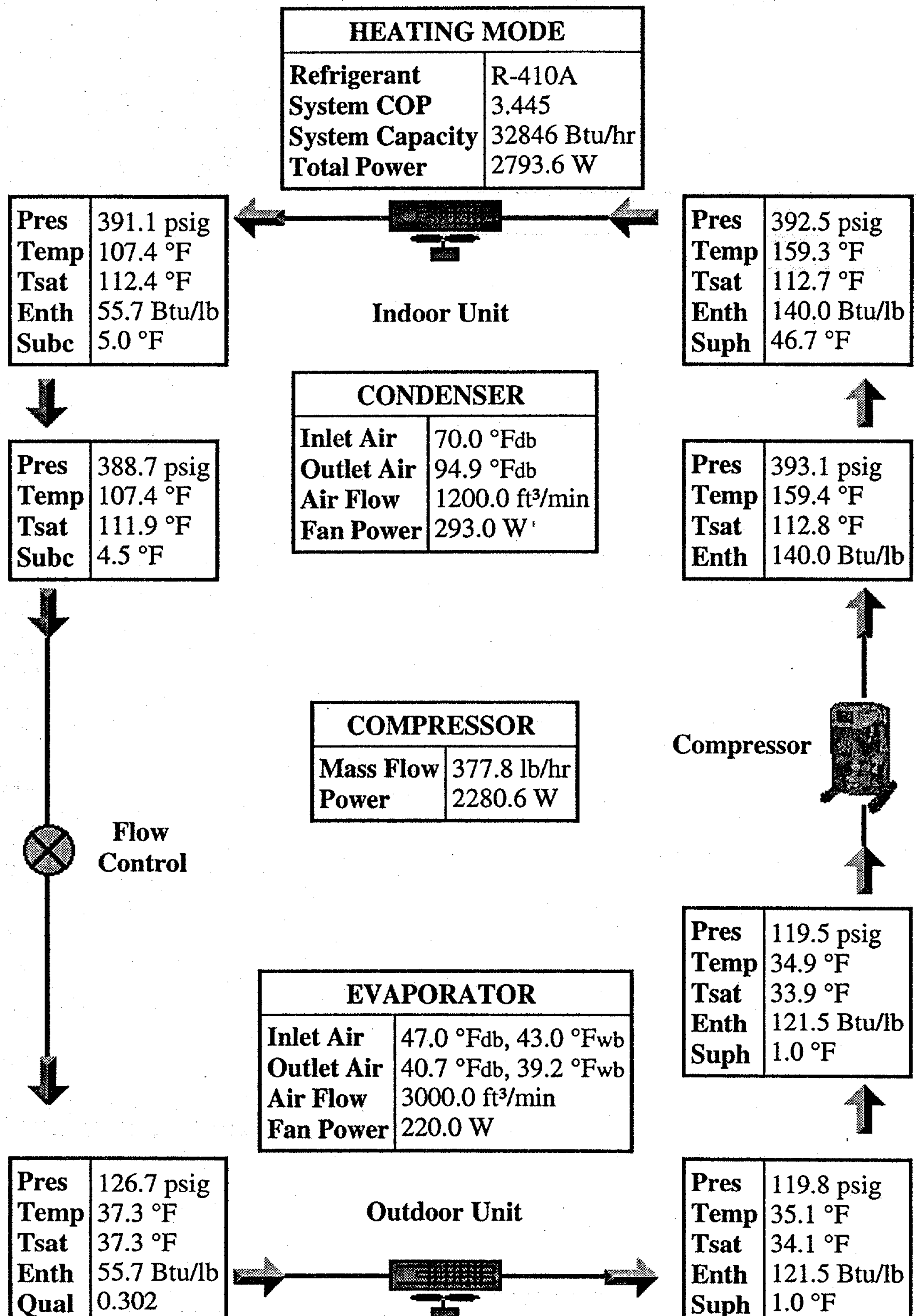

Figure 4. Sample Output Screen from Web Interface under Development for the DOE/ORNL Heat Pump Design Model

displacement, all the motors and the flow controls can be autosized at a design condition. Once the optimum design configuration is found, the compressor displacement, motor and flow control sizes, and the required refrigerant charge can be fixed for off-design analyses.

For the work presented here, this option was used to keep the same capacity for all refrigerants at design conditions. The required compressor displacements and motor sizes so determined for each refrigerant were then fixed for off-design analyses. Refrigerant charge was not fixed in this analysis but the required charge at the design point was noted so that at off-design condtions the optimum amount of condenser subcooling would not exceed the design charge.

APPROACH

Refrigerant-Side Optimization Approach

Previous Work

Spatz (1991) added R-134a, R-152a, and the azeotrope R-32/R-125 (60/40 wt%) to the Mark III ORNL Heat Pump Model to evaluate the relative system performance of these R-22 alternatives. He looked first at optimum numbers of circuits for cooling only and then for heat pump duty; he found that the R-32/R-125 mix required fewer circuits than R-22. The systems were then optimized for tube diameter and number of tubes using a constraint of fixed coil cost. The cost algorithms were not given nor were the optimum configurations for the azeotrope; however, the COP was shown to improve most for the azeotrope. The net result was that the azeotrope showed system-optimized performance that was significantly improved from the theoretical cycle COP as compared to R-22. This R-32/R-125 refrigerant mixture, in slightly modified composition of 50/50 wt% due to flammability considerations, became the leading R-22 alternative, R-410A.

Spatz and Zheng (1993) tested a 3-ton unitary heat pump in cooling mode at the low- and high-temperature rating points and confirmed the performance gains predicted from optimizing the number of circuits for R-32/R-125 (60/40 wt%) relative to R-22 (Zheng and Spatz 1993). Additional system tests on heat exchanger recircuiting in other equipment were reported by Spatz and Zheng (1994) with similar results. Details of the circuiting changes were not given.

Wattelet et al (1994) and Dobson et al (1993,1994) looked at optimal tube diameters in evaporators and condensers for R-22, R-134a, and R-410A in smooth tubes. They assumed a nominal flow of 10 g/s (80 lbm/hr) per circuit (or in some cases a constant capacity per circuit) for both condensers and evaporators. A fixed air-side resistance was used along with refrigerant-side correlations developed based on their test data. They found optimum tube sizes (where primary tube material was minimized) at about 3.6, 4.3, and 5.3 mm (0.14, 0.17, and 0.21") for R-410A, R-22, and R-134a, respectively, for both evaporators and condensers. The reason for choosing a priori the flow rate or capacity values on a per circuit basis rather than for the HX as a whole was not stated.

Ragazzi and Pedersen (1996) looked at tube diameters and numbers of rows that minimized HX irreversibility in wet and dry evaporators for R-407C. Two circuits with smooth tubes were assumed in the analysis. For R-407C, the optimum internal diameter was found to be about 16mm (0.63") for a dry coil. They found that with increased tube diameters up to the optimum, the air-side heat transfer conductance increased more than the refrigerant-side decreased. The air-side correlations were based on Gray and Webb (1986), the same as those in the ORNL model.

Douglas et al (1996) looked at cost-optimum smooth tube diameters for R-22 and a number of HFC and hydrocarbon pure and mixture alternatives. They used water-to-refrigerant heat exchangers configured to have equivalent air-to-refrigerant resistance ratios of unitary air conditioners. Cooling COP and capacity were fixed and system cost was minimized. HX cost was proportional to tube material. The analysis was done assuming one circuit in each heat exchanger. For R-22, optimum inner diameters were found to be 13.5mm (0.531") in the evaporator and 8.8mm (0.35") in the condenser (Douglas 1997). For the R-22 alternatives of most interest here—R-134a, R-290 (propane), R-407C, and R-410A—the optimum diameter ratios relative to R-22 for evaporators/condensers were 1.14/1.02, 0.98/0.99, 0.965/0.935, and 0.885/0.88, respectively.

From the previous work, it can be seen that a variety of optimum values have be obtained depending on the assumptions made, correlations used, and the optimization criteria selected. The number of parallel circuits, base HX air-to-refrigerant-side resistance ratios, refrigerant-side heat transfer and pressure drop correlations, and primary versus finned material minimization are specific areas of possible significant difference between researchers.

Our analysis suggests that number of parallel circuits cannot be traded equivalently with tube diameter—with a fixed total HX length—while maintaining constant heat transfer and pressure drop. Because of this, it appears important that both number of circuits and tube diameter be considered as independent variables to obtain more global optimums. Except for the earlier noted work of Spatz and Zheng with the Mark III ORNL code, the author could not find other analyses where tube diameter and number of circuits were varied simultaneously and modeled as complete air-to-refrigerant heat exchanger assemblies in a total system.

Present Work

In this work, we looked at HCFC R-22 and three non-chlorine containing alternatives—two nonflammable HFCs R-134a and R-410A and one flammable hydrocarbon R-290 (propane) in a system with fixed total HX finned (secondary) surface area and refrigerant-side tube length. Fixed total finned surface area was defined as the sum of the products of face area, tube and fin spacings, and number of tube rows for each coil.

The approach taken to optimize the refrigerant-side design was the following. We considered as heat exchanger design parameters 1) the tube inner diameter and number of circuits in each coil, 2) the fraction of the total HX finned area in the outdor coil, and 3) the condenser subcooling at each rating point. Compressor inlet superheat was set at 5.6°K (10°R) in cooling and 0.6°K (1°R) in heating.

These design parameters were judged sufficient to determine the optimal tradeoffs between refrigerant-side heat transfer and pressure drop for each refrigerant. The refrigerant-side heat transfer and pressure drop correlations are unchanged from Fischer and Rice (1983).

The design objective was to maximize EER in cooling mode and COP in heating while minimizing tube sizes and refrigerant charge. Minimum tube sizes and charge give the lowest copper and refrigerant cost, lower cycling losses, and, for R-410A, thinner tubes required to meet the higher pressure requirements. The constraints imposed on the analysis were a fixed design cooling capacity of 9.38 kW (2.67 tons), fixed total HX finned area, a maximum design sensible-heat-ratio (SHR) of 0.75, and a constant fan power of 513 W. The Hughmark void fraction method (Rice 1987) was used in calculating the refrigerant charge in the two-phase regions of the HXs.

For this analysis, design conditions are considered to be at the DOE "A" test point as given by the ARI (1994) high-temperature cooling condition shown in Table 1. Off-design conditions considered are the low-temperature cooling and high- and low-temperature heating test points given in Table 1.

TABLE 1
Basic Steady-State Rating Conditions for Air-to-Air Heat Pumps

Rating Conditions	Heating Mode		Cooling Mode	
	Low-Temp DB/WB* ℃ (℉)	High-Temp DB/WB* ℃ (℉)	Low-Temp DB/WB* ℃ (℉)	High-Temp DB/WB* ℃ (℉)
Outdoor Coil	-8.3/-9.4 (17/15)	8.3/6.1 (47/43)	27.8 (82)	35 (95)
Indoor Coil	21.1 (70)	21.1 (70)	26.7/19.4 (80/67)	26.7/19.4 (80/67)

*WB given for evaporating cases only.

Validation of Baseline Configuration

The program was calibrated with and validated against data sets taken on a nominal 9.5kW (2.7-ton) R-22 heat pump system tested in the cooling mode with both R-22 and R-410A as reported by Allied-Signal (Spatz and Zheng 1993). System configuration and compressor map data for the R-22 and R-410A compressors used in these tests were provided by Zheng (1993). The model was calibrated against R-22 system data at the low temperature cooling (SEER rating) condition. Here the compressor map, heat exchanger, and line pressure drop models were calibrated (by adjusting relevant input multipliers) to match closely the refrigerant-side conditions and the measured compressor mass flow rate and power. This calibrated model was used to predict performance at the high-temperature cooling rating condition for R-22 tests and at the low- and high-temperature cooling conditions for R-410A. All three validation tests were within –2% in EER and capacity of the data of Spatz and Zheng (1993). The heat pump configuration in these tests and the required heat exchanger and line calibrations were used as the baseline for this analysis (Rice and Jackson 1994).

No heating mode tests were conducted on this unit. However, the model has been tested previously in heating mode by Fischer and Rice (1983), Fagan et al (1987), by Miller (1988), Spatz (1991), and by Rosenquist (1997). Because the heat exchanger, compressor, and line loss models for the baseline unit could not be independently calibrated in heating mode, a larger error is to be expected here for the absolute predictions of heating COP and capacity. A summary of other validation work and known model limitations is given by Rice (1991).

Independent validations at Copeland (1997) on three R-22 10.5 kW (3-ton) systems found EER predictions to within +1.5 to +3.8% and capacities from +0.7% to +2% at the high-temperature [35°C (95°F)] cooling rating condition. At the high-temperature heating condition [8.3°C(47°F)], one validation was reported for R-22 and showed COP predictions within +1.5% and capacity to within -0.3%. Results were reported for one R-410A system for the low-temperature [27.8°C(82°F)] rating point in cooling; the EER was within +1.7% and capacity within +4.2%.

The baseline design for this work is a 10.5 SEER R-22 unit. The R-22 saturation temperatures entering and leaving the compressor were 7.4°C (45.3 °F) and 51.1°C (124°F). The two-phase refrigerant-to-total HX resistance ratios were 0.19 for the evaporator and 0.42 for the condenser at design cooling conditions.

The compressor performance for each refrigerant was represented by scroll compressor maps based on refrigerant-specific test data provided by ARI (1993, 1995). The displacements were scaled by the model (about tested sizes that were closest to the desired application capacity) assuming that the compressor efficiencies remained unchanged.

RESULTS

Example Analysis for R-410A

Results of the refrigerant-side optimization analysis for R-410A, as the leading R-22 alternative, will be described to demonstrate the design and off-design tradeoffs involved in choosing an optimum heat pump configuration. Starting at the high-temperature cooling design condition, we first looked to see what was the optimum HX area fraction and condenser subcooling, with capacity, total HX finned area on both coils, and fan power requirements held constant. Figure 5 shows contours of EER at the design point versus independent variables of outdoor area fraction and condenser subcooling with total cooling capacity held constant. An apparent optimum is found at about 0.58 outdoor area fraction and 8.3°K (15°R) subcooling. However, the SHR value of 0.78 at this optimum is higher than desired for acceptable dehumidification. The "X" shown in Figure 5 is the baseline design configuration which had an acceptable SHR of 0.75. The unconstrained optimum EER was 10.54 while the SHR-constrained value is 10.42.

Cooling Mode Optimums

With the condenser subcooling and heat exchanger area fractions determined, the effect of tube ID and number of circuits at the design condition were studied. Figure 6 shows the effect of these parameters on EER for the outdoor coil serving as a condenser in the cooling mode. Here a maximum EER of 10.5 is obtained with 2 circuits and a tube ID of 8.9mm (0.35") although a ridge of nearly constant EER exists where smaller tube sizes perform nearly as well

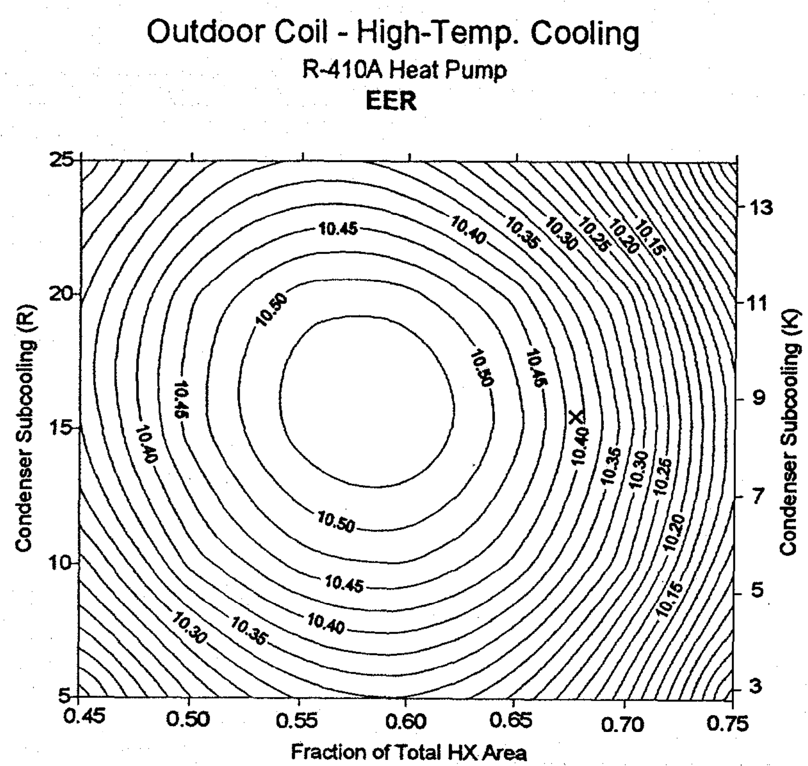

Figure 5. EER Levels for R-410A as a Function of Outdoor Coil Fraction of Total HX Area and Condenser Subcooling -- High-Temperature Cooling Condition.

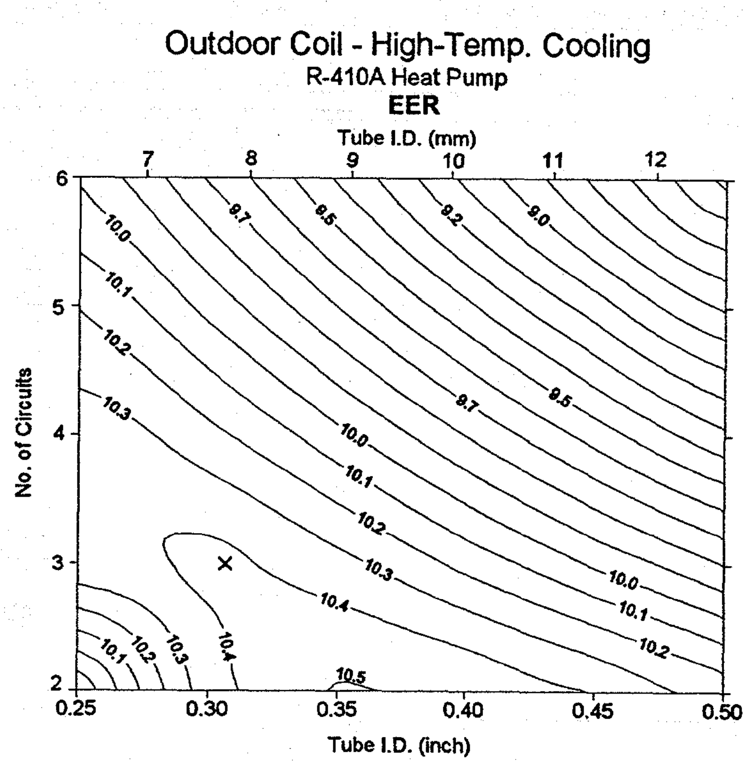

Figure 6. EER Levels for R-410A as a Function of Tube Diameter and No. of Circuits in the Outdoor Coil -- High-Temperature Cooling Condition.

with more circuits. The decreasing EER above and to the right of this ridge is the result of decreasing refrigerant-side heat transfer with more circuits and larger tubes where the heat transfer per unit length falls off inversely with either parameter to about the 0.8 power. Below and to the left of the ridge, the lower EERs at the smallest tube IDs and fewest circuits show the stronger negative effects of excessive pressure drop—which increases inversely with tube ID to about the 5th power and with number of circuits to the 3rd power.

Figure 7 shows the required system refrigerant charge in lbm (1Kg = 2.2 lbm) for the same variation in tube ID and number of circuits. From Figures 6 and 7 together, one can see that a tube ID of 7.9mm (0.31", a nominal tube size of 5/16" OD expanded) and 3 circuits, marked by an "X" on the plots, gives an EER drop of less than 0.1 unit (less than 1%) but a one pound (0.454 kg) drop in required refrigerant charge.

The combination of 4 circuits and tube ID of 6.4mm (0.25", a nominal tube size of 1/4" OD expanded) results in an additional 0.1 EER drop in performance with another one pound (0.454 kg) reduction in total charge. We chose the combination given by the "X" based on best EER with reduced charge and one U.S. tube size reduction relative to an R-22 design. This combination also has a small heating performance advantage over the 2 and 4 circuit alternatives. If refrigerant-charge reduction (or equal tube sizes in both coils) was a strong objective, the 6.4mm ID, 4 circuit alternative could be used.

For the smaller indoor coil acting as an evaporator in cooling mode, similar trends are seen—except they are shifted to higher numbers of circuits and slightly smaller diameters as shown in Figure 8. Here an optimum value of 6.4mm (0.25") and 5 circuits as shown by the "X" was chosen as a good tradeoff between EER and charge size.

Heating Mode Tradeoffs

In the heating mode, the maximum pressure drops occur at the milder ambients where the capacities are higher. Since pressure drop effects are dominant over heat transfer, the parametrics of number of circuits vs tube ID were focused at the high-temperature heating condition.

These tradeoffs are shown in Figure 9 where the "X" at 5 circuits and 6.4mm (0.25") tube ID is the cooling mode optimum. Here one can see that 2 circuits would improve heating COP at the same tube ID. This is because in heating, the indoor coil acting as a a condenser can operate at much higher velocities and improved heat transfer before pressure drop effects start to hurt performance. However, because fewer circuits would seriously compromise cooling performance with higher pressure drops, the cooling requirements govern here.

For the outdoor coil acting as an evaporator in heating, the optimum number of circuits and tube ID is generally compatible with those in cooling. The only caveat is that in heating, the selected optimums are closer to the COP dropoff point due to pressure drop effects. One more circuit in the outdoor coil would give a little more room for error in pressure drop estimations due to refrigerant oil effects or other uncertainties but cooling EER would suffer more than heating COP would gain.

Optimal Subcooling

For the off-design conditions, the best subcooling conditions were determined without consideration of what type of flow control would be required to reach them. The one constraint imposed was that the required charge at any off-design condition could not exceed that at the design cooling point.

The optimum subcooling levels found in cooling for R-410A were 8.3°K (15°R) at high-temperature cooling and 5°K (9°R) at low-temperature cooling. In heating, a low subcooling level of 3°K (5°R) was found to be best for the higher temperature point to minimize capacity and thereby heat exchanger loading and cycling losses. At the low temperature heating condition, a large value of subcooling of 21°K (38°R) was found to be best for system COP. Although the heat pump COP starts dropping more strongly above 11°K (20°R), the gains in capacity more than offset this on a system COP basis because the alternative is resistance heat with a COP of 1.

The one caution that we noted was that the pressure ratio at this higher subcooling is about 5.5 for R-410A. For lower pressure refrigerants, the pressure ratio would be even higher so that compressor operating limits and

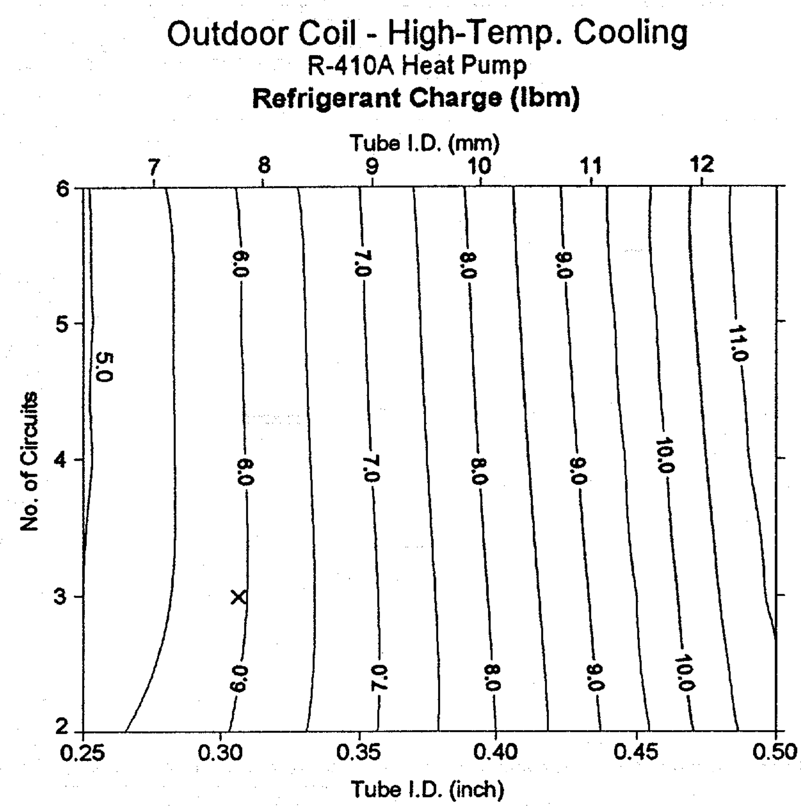

Figure 7. Refrigerant Charge Levels for R-410A as a Function of Tube Diameter and No. of Circuits in the Outdoor Coil -- High-Temperature Cooling Condition.

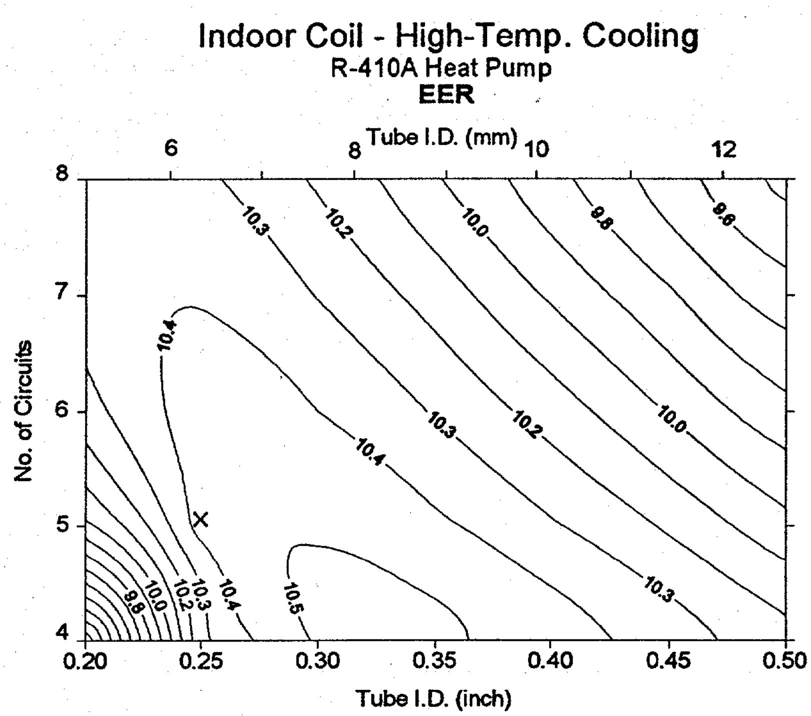

Figure 8. EER Levels for R-410A as a Function of Tube Diameter and No. of Circuits in the Indoor Coil -- High-Temperature Cooling Condition.

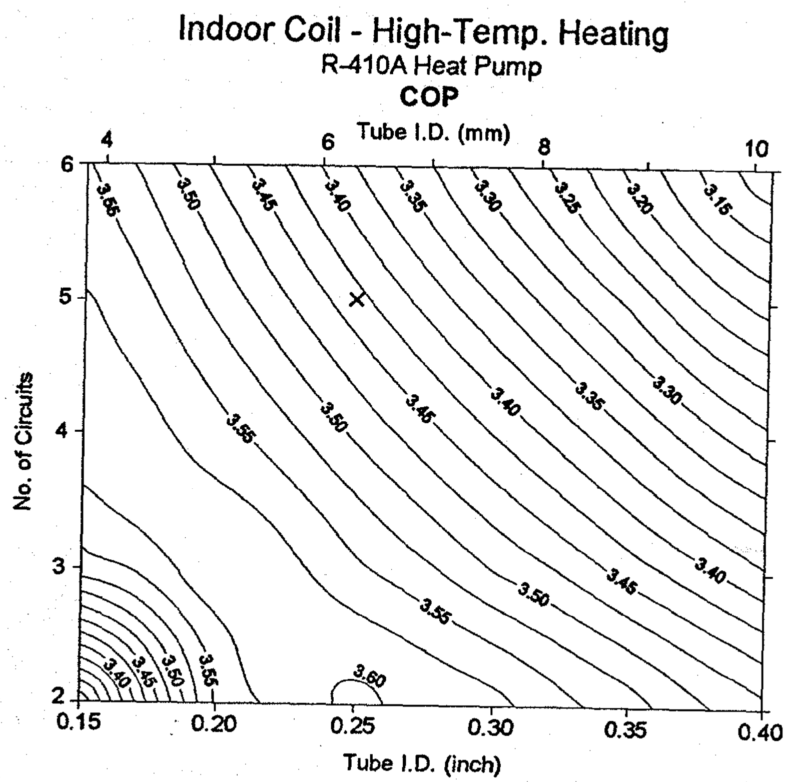

Figure 9. COP Levels for R-410A as a Function of Tube Diameter and No. of Circuits in the Indoor Coil, Heat Pump Design -- High-Temperature Heating Condition.

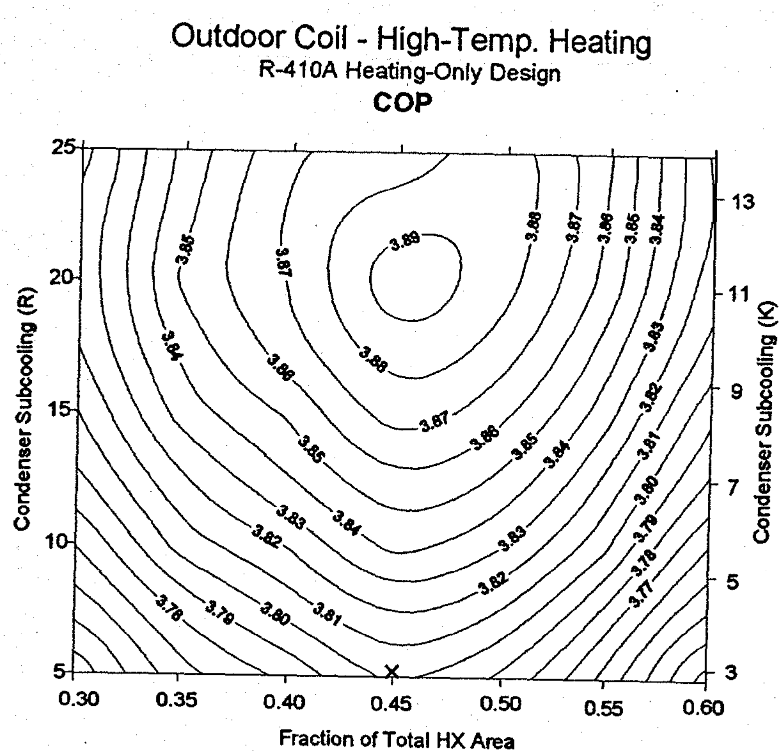

Figure 10. COP Levels for R-410A as a Function of Outdoor Coil Fraction of Total HX Area and Condenser Subcooling, Heating-Only Design -- High-Temperature Heating Condition.

reliability may become an issue. While existing flow controls may not be able to achieve these conditions, it was deemed useful to identify what the optimal operating levels were in both cooling and heating mode so that future control valves with these capabilities might be considered.

Heating-Only Analysis

A heating-unit-only optimization analysis was also performed for an R-410A system of equal total HX area. The capacity was fixed at the value for the optimum R-410A heat pump configuration at the high-temperature heating condition. We looked first the fraction of the total HX finned area in the outdoor coil that was optimal for heating-only operation free from the SHR constraint that was imposed in cooling mode with the heat pump design.

Figure 10 shows the results of this analysis where an optimal fraction of 45% outdoor coil area was found for a heating-only design as compared to 67% for the reversible heat pump case. The optimal subcooling from a seasonal perspective was chosen at the 2.8°K (5°R) value (rather than at the COP-only optimum of 11°K [20°R]) so as to utilize a larger displacement for the specified rated capacity at the high-temperature heating condition. The required charge is also reduced with low subcooling levels at mild ambients.

At the low-temperature heating condition, a 8.3°K (15°R) level of subcooling was found optimum and was consistent with the required charge for low subcooling at the mild ambient condition. In contrast to the heat pump design, no net benefit was found for the heating-only design to have significantly higher subcooling values at extreme ambients.

The choice of a low subcooling at the mild ambient condition gives a larger heating capacity at the more extreme conditions as subcooling is increased as compared to a design where subcooling was chosen at 11°K (20°R) at the rated capacity condition and dropped to the 8.3°K (15°R) level at the low-temperature point. (If fixed flow control devices were used, however, more subcooling would be needed at mild ambients to avoid floodback to the compressor at extreme ambients.)

The coil designs that were determined for a 45% outdoor coil area fraction in the heating-only design have the same fan power requirements as the reversible heat pump design but with a more heavily finned indoor coil. The optimal unit has 36% more indoor coil fin area, a 19% larger face area, and 13% higher fin pitch while the outdoor coil needs 17% less finned area, 12.6% less face area, and 26% lower fin pitch. The lower fin pitch on the outdoor coil should have benefits of lower frosting losses that are not considered here.

The required displacement for the heating-only case was found to be essentially the same as for the heat pump design of the same capacity. This was mainly because the same subcooling levels were assumed at rated capacity as in the heat pump design. The heating-only unit had capacity-neutral (offsetting) saturation temperatures that were 1.1 and 4.4°K (2 and 8°R) lower in the evaporator and condenser, respectively, than for the heat pump design.

The HX tubing configurations found optimum for the indoor and outdoor coil were 2 and 4 circuits and 6.4mm (1/4") and 7.9mm (5/16") tube ID, respectively. The required charge was 5% less than for the heat pump design. If charge inventory was a higher priority, the tube sizes could be reduced to 4.8 mm (3/16") on the indoor and 6.4mm (1/4") on the outdoor coil, with more circuits added for both coils, to lower the charge by another 21%.

Heat Exchanger UA Comparisons

Figures 11 and 12 show how much the heat exchanger UA balance is improved from a heat pump design to a heating-only design. In Figure 11, the relative HX UA levels in the R-410A heat pump design are shown for the high-temperature heating and cooling conditions. The reference UA level is that of the indoor coil in the heating mode. The relative air-side finned area of the indoor and outdoor coil is given by the dotted lines based on the optimal outdoor coil area fraction of 0.67 from Figure 5. The UA comparisons show that the evaporators and condensers (identified by the different patterns) have quite different UA levels from heating to cooling mode. Furthermore, the ratio of evaporator to condenser UA is 1.21 to 2.15 in cooling versus 2.86 to 1.0 in heating. The 2-to-1 area ratio from outdoor to indoor is the major factor here, but higher evaporating heat transfer coefficients relative to condensing, especially in the heating mode, contribute further to the mismatch.

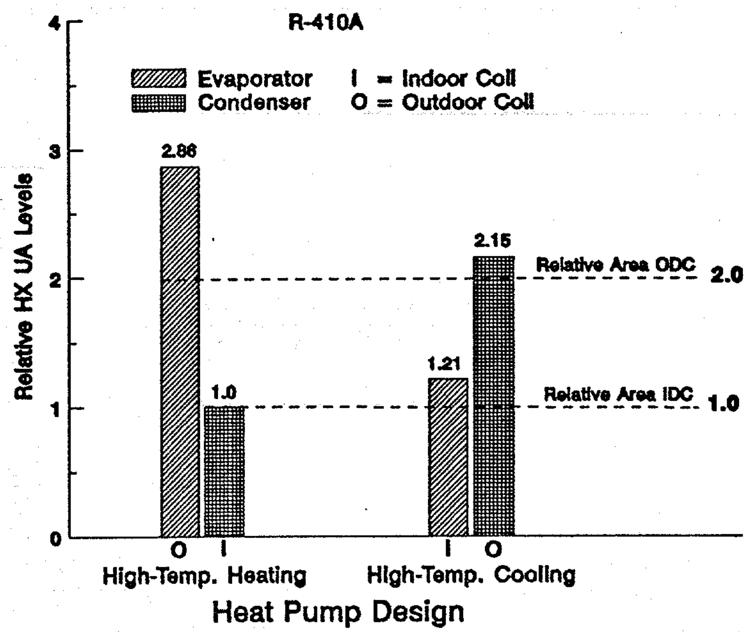

Figure 11. Relative Heat Exchanger UA Levels Between Evaporating and Condensing Functions for R-410A Heat Pump Design—at High-Temperature Heating and Cooling Conditions

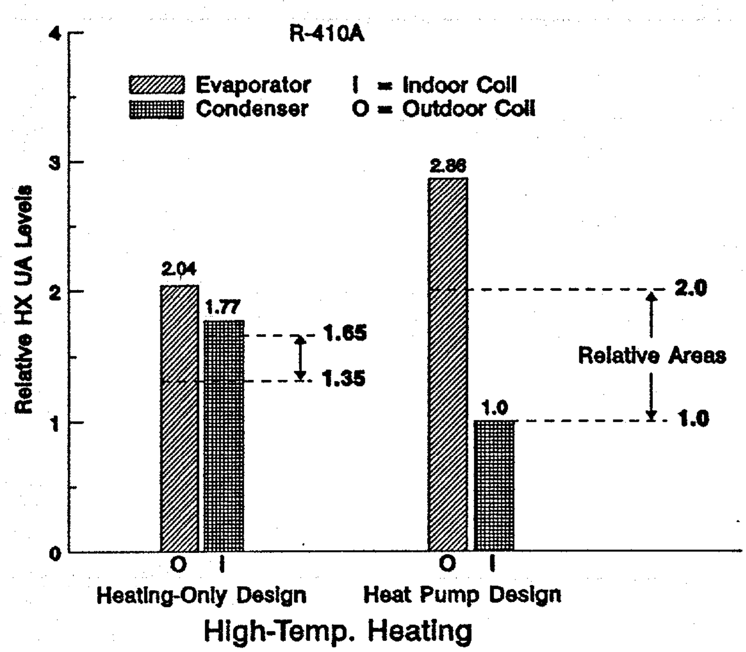

Figure 12. Relative Heat Exchanger UA Levels Between Evaporating and Condensing Functions for R-410A Heating-Only and Heat Pump Designs—at High-Temperature Heating Conditions

In Figure 12, the relative UAs in heating mode are compared between the heating-only design and the heat pump design. The relative areas corresponding to the optimal 0.45 outdoor coil area fraction from Figure 10 are also shown. The UA levels are seen to be much more balanced with only 15% higher evaporator than condenser UA as compared to 186% higher for the heat pump design. This more effective utilization of the total available HX area is the basis for the predicted improved performance. The result also shows that in heating mode, a slightly higher UA on the evaporator is optimum even though more heat is transferred in the condenser. This is in contrast to the cooling mode where more UA is needed in the condenser.

Seasonal Performance Benefits

For the heating-only design, rated HSPF gains of 11% are predicted (at the minimum design heating requirement [DHR] condition) in DOE Regions IV and V relative to the heat pump design. At maximum DHR, which reflects a higher more typical balance point just below 0°C (32°F) (instead of around the low-temperature heating ambient with a minumum DHR), gains of 4.2% in Region IV and 3.7% in Region V are predicted. These HSPF improvements for a heating-only design do not include further gains that would be realized from the elimination of reversing valve losses, defrost from hot gas bypass instead of cycle reversal, and less frosting degradation and fewer required defrost cycles from the wider fin spacing on the outdoor coil.

Design Tradeoffs

Tradeoffs of AC Design vs Heat Pump

In looking at the design tradeoffs between cooling and heating mode for a heat pump, one can see that the indoor configuration is driven by cooling mode in terms of relative coil size for sufficient moisture removal and by evaporator pressure drop limitations. The outdoor coil refrigerant-side design is influenced, to a somewhat lesser degree, by heating mode—again with regard to evaporator pressure drop limitations. A heat pump design requires that outdoor coil refrigerant velocities be lower than optimum in condensing (cooling) mode to keep evaporator pressure drops from being excessive in heating mode. Because of this, in an AC-only unit, the outdoor coil could have a smaller tube diameter or fewer number of circuits with a slight performance gain.

Tradeoffs Between a Heat Pump and a Heating-Only Design

On the heating-side, a reversible heat pump design is more significantly compromised with respect to a heating-only version. The indoor coil is too small, with too many circuits, for best heating efficiency and the outdoor coil usually has too close a fin pitch to minimize frost degradations. Possible steady-state performance gains from a redistribution of HX area and more optimum refrigerant-side design were given earlier. While air-side redesign was not considered here, use of a larger indoor coil may also allow a lower airflow rate and higher exit temperatures, without as much COP penalty (especially with R-410A) as a heat pump design where the condensing saturation temperature is 4.4°K (8°R) higher with the smaller coil.

Tradeoff of Tube Sizes Vs Performance and Charge / Flow Control Issues

A heat pump design with larger tubes in the outdoor coil than the indoor has excess charge to accommodate in the heating mode because all the cooling-mode condenser charge must be stored in the smaller-size, smaller-tube indoor coil or in the accumulator. Smaller tubes in the large outdoor coil of a heat pump result in a more balanced charge between heating and cooling modes. How advantageous this is depends on the type of flow control device used. A system with a smaller refrigerant charge and no accumulator will usually reach equilibruim faster and have lower cycling losses.

For an adjustable-opening flow control design such as a TXV, which does not usually include an accumulator for storing excess charge, equal indoor and outdoor tube sizes can be beneficial in avoiding excessive subcooling levels at mild heating ambients. For fixed-opening flow control systems, however, larger outdoor tubes provide extra subcooling at milder ambients in heating mode so that, at low ambients, the condenser exit would remain subcooled. We found that the optimal condenser subcooling control for a heat pump in heating mode was to minimize subcooling at mild ambients to reduce overcapacity and unload the heat exchangers while, at colder ambients, increasing subcooling as much as available charge allowed to boost capacity and overall system COP including backup heat.

Summary of Optimum Configurations and Performance

Optimum Configurations and Charge Requirements

As in the heat pump example given for R-410A, the HX configurations and refrigerant subcooling control were also optimized for R-22, R-134a and R-290 (propane). The tube IDs and number of circuits were optimized for the best balance of heating and cooling performance. Condenser subcooling was optimized at each of the design and off-design conditions given in Table 1. The HX area ratios, face areas, fin pitches, tube spacings, number of rows, and fan power were held constant for all the heat pump cases at the values of the original baseline configuration. In Table 2, the optimal tube OD sizes, numbers of circuits, and required refrigerant charge levels are shown for each refrigerant.

TABLE 2
Heat Exchanger Configurations and Charge Requirements
for Refrigerant-Side-Optimized Heat Pump Designs
Predicted from the ORNL Mark V Design Model

Refrigerant	Indoor Coil		Outdoor Coil		Refrigerant Charge
	Tube OD[2] mm (inch)	Circuits (# in parallel)	Tube OD[2] mm (inch)	Circuits (# in parallel)	kg (lbm)
R-22 [orig.][1]	7.9 (5/16)	6	9.5 (3/8)	3	3.5 (7.6)
R-22	7.9 (5/16)	5	9.5 (3/8)	3	3.4 (7.4)
R-134a[3]	7.9 (5/16)	6	9.5 (3/8)	3	3.1 (6.9)
R-410A	6.4 (1/4)	5	7.9 (5/16) 6.4 (1/4)	3 4	2.7 (6.0) 2.2 (4.9)
R-290 (propane)	6.4 (1/4)	6	7.9 (5/16) 6.4 (1/4)	4 5	1.1 (2.4) 0.67 (1.5)[4]

All units have 9.4 kW (2-2/3 ton) design cooling capacity
[1]Original 10.5 SEER R-22 split system
[2]Tube ODs are nominal before expansion
[3]Suction lines increased from 19 to 22 mm (3/4 to 7/8")
[4]No accumulator, smaller connecting tubes, reduced subcooling

For R-22, the original (actual validation hardware) design is given first and is seen to compare closely with the optimized design. The predicted improved design has the same tube sizes with one less indoor circuit.

For R-134a, the tube sizes were found to be the same as for R-22 but more circuits were needed in the indoor coil. Also, the suction line had to be increased by one tube size to avoid excessive pressure drop. A 6% reduction in charge was predicted.

For R-410A and R-290, the optimal tube IDs are one tube size smaller than for R-22 with similar numbers of circuits. R-290 requires one more circuit than R-410A for both coils. The one size reduction in tube sizes for R-410A and propane results in 20 to 67% lower charge requirements than for R-22. For the outdoor coils, a second smaller-tube-size option is given for R-410A and R-290 which gives slightly lower performance but with significantly lower charge while requiring one additional circuit. For R-410A, the charge was reduced to 34% less than for R-22.

For the flammable refrigerant propane, additional measures were taken to see how low the charge could be taken. All line sizes and subcooling levels were reduced and the accumulator was eliminated. A lower level of 0.072 kg/kW (0.56 lbm/ton) was found. This charge is only 20% of the optimum R-22 charge. An analysis of the use of propane in split system air conditioners has been made recently by Keller, et al (1996). He estimated the required propane charge to be 50% of that for R-22 or R-410A at about 0.13 kg/kW (1 lbm/ton).

The amount of flammable refrigerant required per unit capacity is of interest in assessing the risk of propane designs in residential-size heat pump applications. Grob (1994) showed sample flammability calculations which indicated that less than 0.051 kg/kW (0.4 lbm/ton) would be needed before propane might be considered nonflammable in certain scenarios. It would appear that new HX designs that further minimize HX internal volume might approach these levels. Were this minimum charge level obtained for propane or if flammability requirements were relaxed in the future, it can be seen from Table 2 that HXs designed for R-410A could be used with propane with little or no modification.

Relative Low-Temperature Heating Capacity

As noted earlier, the compressor displacement of each refrigerant system was adjusted to have the same capacity at the design cooling condition. Of special interest in heating-dominated climates is how well the capacity level is maintained at the low-temperature heating condition. We found that all the alternatives were predicted to have lower capacities than R-22, with R-134a having 10% less, while R-290 and R-410A had 4.0 and 3.6% less respectively. All systems had the same optimal subcooling at this condition except for R-134a which was 2.2°K (4°R) lower to meet charge requirements.

The predictions for R-410A are reasonably consistent with the findings of Linton et al (1997). They found that, after accounting for a switch from reciprocating to scroll compressors by adjusting for the different volumetric efficiencies, the low temperature heating capacity of R-410A would have been about 3% lower than for R-22 (had both units had exactly the same design cooling capacity). This compares with 3.6% lower from our predictions.

In our analysis, scroll compressors were used for both R-22 and R-410A. Because the R-410A scroll was a nominal 5.3kW R-22 design vs 8.2kW for R-22, as in an actual application with equal cooling capacity, it has a bigger drop in volumetric efficiency at the low-temperature heating condition. This pumping efficiency dropoff for the smaller compressor causes a 2.7% lower capacity relative to R-22[1]. Adjusting Linton's 3% loss by this correction gives an experimentally based 5.7% lower capacity for R-410A vs R-22 as compared to the 3.6% loss predicted here. Thus, with the smaller scroll compressors, an R-410A system of 10.5 kW or less may lose between 4 and 6% in low-end heating capacity relative to R-22.

Seasonal Performance Comparisons

With the optimum configurations and refrigerant control determined for the four refrigerants, the model was exercised for all the rating conditions and the results used to determine SEER and HSPF levels for DOE Region IV. The results for seasonal performance relative to R-22 are shown in Figure 13. R-410A and R-290 show small performance gains relative to R-22 with R-410A looking best in cooling and propane doing better in heating. R-134a breaks even in cooling but loses ground in heating. The absolute performance of the R-22 baseline is also given.

The performance gains for R-410A and propane are from improved heat exchanger and compressor performance. R-410A and propane have theoretical cycle performances at the high-temperature cooling point of almost 7% and 1%, respectively, less than R-22 while R-134a has a 1% advantage over R-22. However, because of system effects, propane and R-410A gain back 5.2 and 9.1 percentage points, respectively, in performance at the design condition relative to R-22 due to better HX and compressor performance while R-134a loses 1.6 percentage points.

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[1]This lower volumetric efficiency in heating for the R-410A scroll should be added to the adjustments made by Linton (1997) because scrolls of this size or smaller lose more in performance from their fixed leakage losses than the baseline R-22 size.

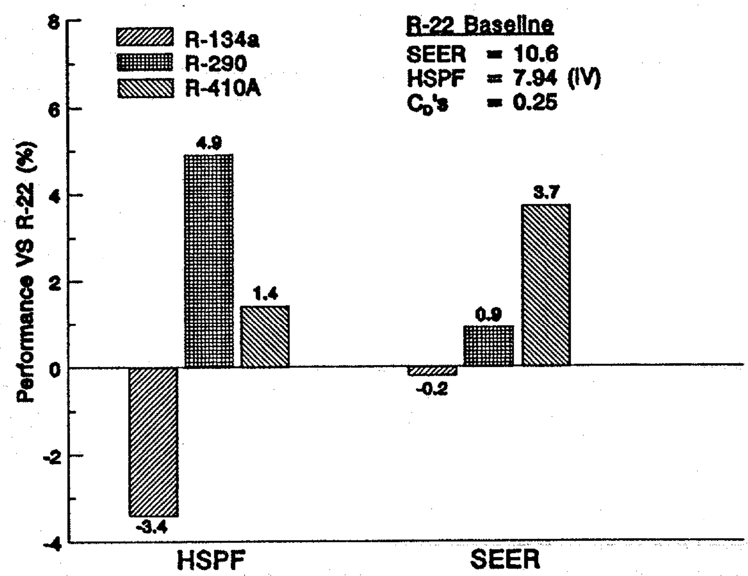

Figure 13. Seasonal Performance Comparisons of R-22 Alternatives with Refrigerant-Side-Optimized Designs

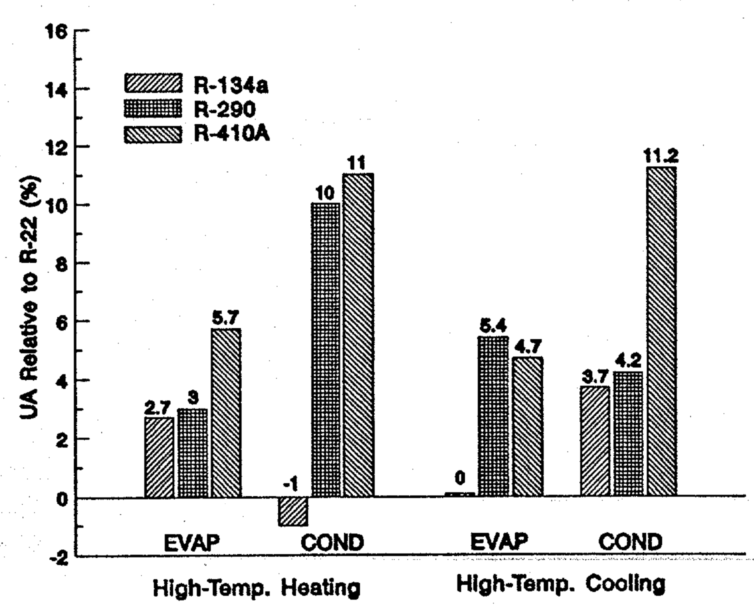

Figure 14. Heat Exchanger UA Comparisons of R-22 Alternatives with Optimized Designs at High-Temperature Heating and Cooling Rating Conditions

For R-410A, more than half of this gain is due to better HX performance with the rest from higher compressor efficiency. For propane, one-third of its system gains in cooling are in the HXs with the rest in the compressor.

Figure 14 shows the relative HX performance of the R-22 alternatives in terms of overall HX UA levels at the high-temperature heating and cooling conditions. For R-410A, the HXs had about 11% larger UAs in the condensing mode with about 5% gains when evaporating. For propane, the UA gains are 3 to 5% except for heating mode condensing with a 10% gain.

The other aspect of HX performance is the relative loss of refrigerant-to-air temperature difference from two-phase pressure drop. In terms of pressure drop effects, R-410A had 65 to 95% higher optimal pressure drops than R-22 with an effective saturation temperature drop of about 1.7°K (3°R) in the evaporators and condensers. R-22 had about the same effective temperature drop in the condenser and slightly less in the evaporator. R-134a had an average two-phase temperature drop of 2.8°K (5°R) in each HX, which is two-thirds larger than for the other refrigerants.

These results are with R-22 compressor designs unmodified for use with other refrigerants except for the different compressor sizes and the larger motor in the smaller R-22 compressor used for R-410A. Compressor designs tuned for a specific alternative would be expected to show further improvement.

Even with R-22 compressors, these results indicate that R-410A or propane designs can perform as well or better than R-22 with HX tubes one or two sizes smaller and with 20 to 80% reduced system charge.

DISCUSSION

One possible heating improvement to a heat pump design would be to provide some means of merging indoor circuits in heating mode. This would improve the low condensing coefficients that result from the large number of indoor circuits needed for the cooling mode. In heat pump designs, the indoor coil UA is about 35% of that for the outdoor coil while transferring 125% as much heat. Figures 8 and 9 show that a design with 6 circuits in cooling and 3 in heating would improve heating COP at the high-temperature rating point by over 5% with no loss of cooling performance. As the refrigerant flow reverses from cooling to heating, perhaps some check valve type of passive arrangement could be employed to accomplish this.

In a climate where heating is needed for most of the year and cooling requirements are minimal, a heating-only heat pump offers a number of advantages. The configuration changes and general performance benefits for coils sized and circuited for the heating function alone have been given for R-410A and should generally apply for R-22 and its alternatives.

Other circuiting performance advantages of a heating-only design would be available from the elimination of reversed flow operation that are not modeled here. Circuits could be merged in the subcooled region of the indoor coil to boost heat transfer without concern of excessive pressure drop in cooling. Also, the indoor coil could be arranged in cross-counterflow to better match the refrigerant two-phase (pressure-drop-induced) and subcooled temperature profiles. R-22 alternatives such as R-407C which has a moderate (5°K) two-phase temperature glide would benefit a few percent more in performance from a cross counterflow configuration in the indoor coil similar to the air conditioning benefits shown by Murphy et al (1995).

Another significant seasonal performance benefit of a heating-only heat pump is that the unit could be sized for the heating climate rather than the design cooling requirement. This would allow the balance point to be pushed below the peak of the [load x hours] distribution curve for the specific heating climate which would minimize backup heat needs. If this were done, the 11% higher HSPF rating values obtained at the minimum design heating requirement could be achieved at higher DHR conditions that are more typical.

A two-speed or dual compressor heat pump (Di Flora and Hatzikazakis 1994) could accomplish this shift of balance point with less cyclic operation and higher performance at the milder ambients. Such a heating-only design could provide some cooling at low capacity, but with reduced dehumidification capability. The operating cost savings of such designs would have to be weighed against the first cost of the larger capacity units to determine if such a product were viable in selected cold climate regions.

CONCLUSIONS

Conventional Heat Pumps

Heat pump designs were refrigerant-side optimized for R-22 and three non-chlorine containing alternatives, R-134a, R-410A, and propane. With the exception of R-134a, the R-22 alternatives were found to perform as well or better than R-22 on a seasonal basis with refrigerant tubes one or two U.S. sizes smaller. R-410A and propane had 20 and 67% lower charge requirements than R-22 with one tube size reduction and 34-80% lower with one further tube size reduction in the outdoor coil. For the high pressure R-410A, the smaller tube sizes mean that the tube walls do not have to be thicker than for R-22 to meet burst requirements. Charge balance implications of having an outdoor coil with the same tube size as the indoor coil must be considered in selecting a fixed-opening flow control device.

Two refrigerant-side means were identified to improve heating performance. A flow control strategy that would maintain a high level of subcooling at more extreme ambients was found to give higher capacity and better overall performance when resistance heat requirements were included. This does result in higher pressure ratios at these conditions, which may be an operational concern for the lower pressure refrigerants.

A way of merging indoor coil circuits to half as many in heating mode would improve performance by about 5%. Some type of passive check valve arrangement activated during flow reversal might be employed to accomplish this.

Heating-Only Heat Pumps

A heating-only optimal design was determined for the leading R-22 alternative, R-410A. This design uses the same total coil material and airflow as the heat pump designs and has the same heating capacity and fan power at the high-temperature rating condition. The indoor coil was increased from 33 to 45% of the total coil area and went from 5 to 2 circuits. The outdoor coil has a 26% lower fin density in addition to a reduced face area. With this design, outdoor to indoor UA ratios went from 2.9 to 1 previously to 1.15 to 1 and the condenser saturation temperaure dropped 4.4°K (8°R) at the mild ambient rating condition.

In terms of seasonal performance, the heating-only design was predicted to have 11% higher HSPF ratings for Regions IV and V. Further gains not included here would be realized from the elimination of reversing valve losses and lower frosting degradation from the more widely spaced fins. Sizing such heating-only units to meet more of the heating load would allow the higher HSPF ratings to translate into corresponding reductions in resistance backup requirements and total energy use.

RECOMMENDATIONS

The DOE/ORNL Heat Pump Design Model and the selected optimizing design methodology was found to predict optimum configurations close to those for existing R-22 heat pumps. The heat transfer and pressure drop correlations have been tested and in some cases developed for R-22 and other HCFC and CFC refrigerants in annular flow. However, the validity of the results for R-22 alternatives, especially at the smaller tube sizes have not been proven. The reported results should be confirmed with newer heat transfer and pressure drop correlations developed for use with HFC alternatives and extending beyond the annular flow regime. Using these newer correlations, a similar design analysis should be conducted for R-407C once this capability is available and the model has been validated. Further testing of the model with R-410A in the heating mode is also recommended.

The analysis done here was strictly for smooth tubes. As many heat exchanger coils now have rifled or grooved microfin tubes, a similar study of optimal tube diameters versus numbers of circuits is needed for the most industry-predominate surfaces. As the pressure drops with these internally augmented surfaces can be significantly higher than for smooth tubes, the tube sizes and numbers of circuits predicted here may be too small.

Microchannel HXs are the subject of active current investigation and have been recently demonstrated by Chapp (1997) in a split system air-conditioner with R-22. The concept appears to offer new opportunities to reduce refrigerant charge, HX size and material, and better optimize refrigerant-side performance (Heun and Dunn, 1996). An additional degree of freedom is available with the increased number of HX design parameters than is possible with only the limited tube sizes and circuits of conventional fin-and-tube design. A model of such a heat exchanger should be added to the DOE/ORNL program and system-optimized for R-22, R-410A, and propane.

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