INTERNATIONAL CFC AND HALON ALTERNATIVES CONFERENCE

December 3-5, 1991
Baltimore Convention Center
Baltimore, Maryland

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CONFERENCE PROCEEDINGS

Performance of Alternative Refrigerants From a System's Perspective

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Abstract

In this paper the results of modeling a representative 3-ton air-to-air heat pump running with the baseline refrigerant (HCFC-22) and zero ODP alternative refrigerants will be presented. The model is validated using test results of the 3-ton heat pump. Baseline simulation runs with HCFC-22 utilize compressor maps generated from compressor calorimeter tests. Alternative refrigerant simulation runs are made with two pure fluids (HFC-134a and HFC-152a) and an azeotropic mixture (60/40 wt%) of HFC-32 and HFC-125 (Allied-Signal Inc. U.S. Patent 4,978,467). The expansion device, the heat exchangers, and line sizes are optimized for each refrigerant. Compressor losses and efficiencies from the baseline runs are used for the alternative refrigerant runs to simulate an optimized compressor for these refrigerants. Pre-optimized and post-optimized system performance is presented to demonstrate the impact of this process on the ranking of replacement refrigerants.

Introduction

A move away from familiar refrigerants is currently in progress. This move is spurred on by concerns for the earth's protective ozone layer and accelerated global warming. Although the main focus in the near-term is the elimination of CFC's, HCFC's, viewed as interim replacement candidates in many applications, will also be eventually banned from production. According to present domestic legislation, only refrigerants without any ozone depletion potential would be acceptable long-term replacements.

However, a switch to refrigerants that do not deplete the ozone layer only addresses the first of two environmental concerns. The impact on global warming also requires attention, (although by switching to zero ODP HFC's, the direct greenhouse warming potential of the refrigerant is significantly reduced from that of the CFC's). Global warming will be impacted by both the direct greenhouse potential of the gas in the atmosphere and by the fossil fuel demands of the system the refrigerant is charged into. This makes energy efficiency a critical parameter that must be considered when selecting alternative refrigerants. However, too often refrigerants are ranked solely on the basis of simple thermodynamic cycle efficiencies. Factors such as heat transfer and pressure drop characteristics are ignored with this type of comparison. Even when a comparison is based on calorimeter testing, the results can lead to incorrect conclusions if the system is not optimized for each refrigerant's thermodynamic and transport properties.

A testing program that involves optimization of all components for a given refrigerant would be prohibitively expensive and time-consuming. It is for this reason that a computer simulation of a complete system, coupled with limited testing, would provide a more cost-effective and timely means of ranking alternative refrigerants for a given refrigeration system.

Heat Pump Modeling

The Mark III Heat Pump System Design Model Program developed by The Oak Ridge National

Laboratory, [1] [2], was modified to include the thermodynamic and transport properties of the following refrigerants: R134a, R32, R123, R124, R125, R143a, R152a, and the 60/40 (wt%) blend of R32/R125. Where available, transport property correlations based on measured data were used. When not available, predictive methods were used to estimate the properties [3] [4] [5].

The geometry and the compressor performance map of a 3-ton heat pump utilizing a scroll compressor was input into the model. The model was run (using R22) in the cooling mode at outdoor temperatures of 95°F and 82°F and an indoor temperature of 80°F db, 67°F wb, (ARI "Test A" and "Test B" conditions respectively). The model was also run in the heating mode at an outdoor temperature of 47°F db, 43°F wb and an indoor temperature of 70°F (ARI high temperature heating test condition). The results were compared to the heat pump manufacturer's test data of this heat pump at the same conditions. As can be observed from the comparison displayed on Table I, the model results compare quite well with the actual test data.

HEAT PUMP MODEL VALIDATION

USING MAP-BASED COMPRESSOR MODEL

	OUTDOOR COND.		INDOOR COND.		MODEL RESULTS		TEST RESULTS		PERCENT DEVIATION	
MODE	DEG.F DB	DEG.F WB	DEG.F DB	DEG.F WB	CAPACITY	EER	CAPACITY	EER	CAPACITY	EER
COOLING	95	75	80	67	33950	8.74	33136	8.87	2.5	-1.5
COOLING	82	65	80	67	35950	10.59	35219	10.55	2.1	0.4
HEATING	47	43	70	60	34232	10.94	34046	10.69	0.5	2.3

Table I

In order to run the model with alternative refrigerants, the compressor sub-model had to be switched from a map-based model to a model based on volumetric and isentropic efficiencies that are supplied for given conditions (compressor performance maps with alternative refrigerants are not available). These efficiencies were obtained from the model results previously described. It was assumed that the compressor efficiencies obtained with R22, could be obtained with the alternative refrigerants.

Four series of runs of the model in the cooling mode at ARI "Test B" [6] conditions was completed. The efficiency of the heat pump at these conditions is the dominant factor in determining a heat pump's or an air conditioner's SEER.

In the first series, the geometry of the heat exchangers was not changed. The displacement of the compressor with the alternative refrigerant was increased or decreased to achieve the same capacity. Evaporator superheat and condenser subcooling was held constant which implied changes to the expansion device to achieve these conditions. The results of this series are shown in column "A" of Table II. The absolute and relative (to R22) values of COP and compressor displacement are given.

In the next series of runs, the circuiting of the heat exchangers were optimized for maximum COP.

OPTIMIZATION PROCESS - COOLING ONLY

COP (@ EQUAL CAPACITY)
AND REQUIRED COMPRESSOR DISPL. (CU.IN.)

REFRIGERANT		A ORIGINAL DESIGN VALUE	A ORIGINAL DESIGN RATIO	B OPT. # OF CIRC'TS VALUE	B OPT. # OF CIRC'TS RATIO	C B + L/S HT. EXCHR. VALUE	C B + L/S HT. EXCHR. RATIO	D B + C + OPT. HX's VALUE	D B + C + OPT. HX's RATIO
BASELINE - R22	COP	3.10	1.000	3.19	1.028	3.19	1.028	3.25	1.047
	DISPL.	2.85	1.000	2.85	1.000	2.85	1.000	2.76	0.968
R32/R125	COP	2.99	0.964	3.21	1.036	3.22	1.039	3.27	1.053
	DISPL.	1.93	0.677	1.84	0.646	1.84	0.646	1.82	0.639
R134a	COP	3.03	0.977	3.13	1.010	3.19	1.027	3.22	1.038
	DISPL.	4.54	1.593	4.42	1.551	4.30	1.509	4.25	1.491
R152a	COP	3.14	1.011	3.20	1.032	3.22	1.038	3.24	1.045
	DISPL.	4.83	1.695	4.75	1.667	4.71	1.653	4.65	1.632

NOTES:

- MODEL RESULTS FOR COOLING MODE AT 82 F OUTDOOR TEMPERATURE AND 80 F DB, 67 F WB INDOOR TEMPERATURE (ARI "TEST B" CONDITIONS).
- COLUMN "B" REPRESENTS THE RESULTS OF OPTIMIZING THE NUMBER OF CIRCUITS IN BOTH THE EVAPORATOR AND CONDENSER TO MAXIMIZE COP.
- COLUMN "C" USES THE DESIGN FROM COL. "B" AND ADDS A SUCTION/LIQUID LINE HEAT EXCHANGER. THE EFFECTIVENESS OF THE HX IS 40%.
- COLUMN "D" ADDS OPTIMIZATION (AT CONSTANT COST) OF THE EVAP. AND COND., CIRCUITRY IS RE-OPTIMIZED, AND L/S HX IS INCLUDED IF BENEFICIAL.
  NOTE THAT FOR BOTH R134a AND R152a, THERE ARE NO HX CHANGES (AT CONSTANT COST) THAT WOULD IMPROVE PERFORMANCE.
  THE SUCTION LINE DIAMETER WAS INCREASED FROM 3/4" TO 7/8" FOR R-134a AND R-152a (THIS CHANGE WAS MADE IN COLUMN "D").

Table II

An example of this process is shown on Figures 1 and 2. The optimum number of evaporator circuits for R22 is six (the original number of circuits), while for the R32/R125 blend, the optimum is four. In the condenser (when evaluating the condenser, the optimum number of evaporator circuits is used) the optimum number of circuits is two for R22 and one for R32/R125. Since the capacity of the heat pump with R32/R125 using four evaporator and one condenser circuit is higher than the optimized R22 heat pump, the displacement of the compressor was reduced to equalize capacity. It is at this point the efficiency and displacement comparison is made and the results are shown in column "B" of Table II for all refrigerants analyzed. It should be noted that the actual heat pump has three condenser circuits rather than two (as suggested by the optimization) most likely due to the need for compromise between cooling and heating performance.

Liquid-suction heat exchange impacts are listed in column "C" of Table II (using the optimum number of circuits from column "B"). A heat exchanger could be easily incorporated into a split-system air conditioner by pressing the liquid line against the suction line and insulating the two together. It was estimated that for typical 25 feet suction/liquid lines the effectiveness of this exchanger would be approximately 40%. However, the incorporation of a liquid-suction heat exchanger into a heat pump would

Optimizing Number of Evaporator Circuits

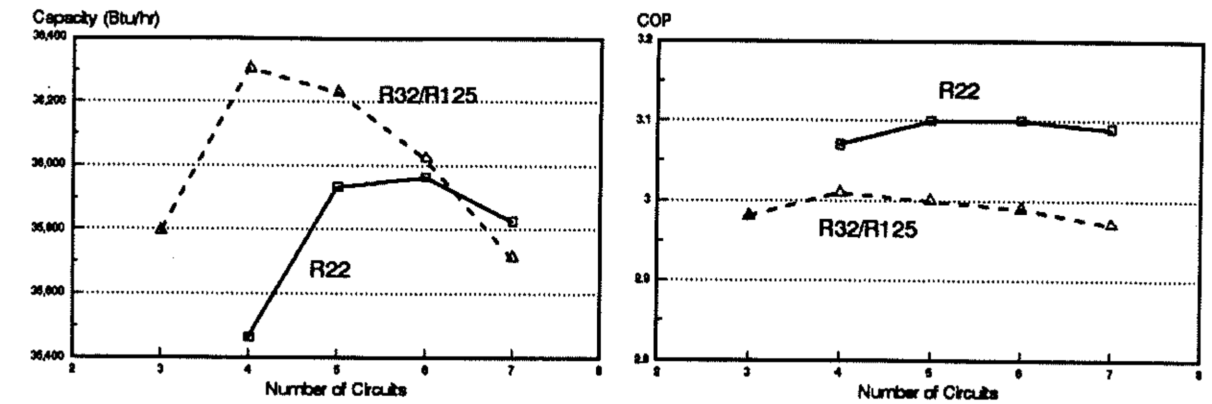

Figure 1

Optimizing Number of Condenser Circuits

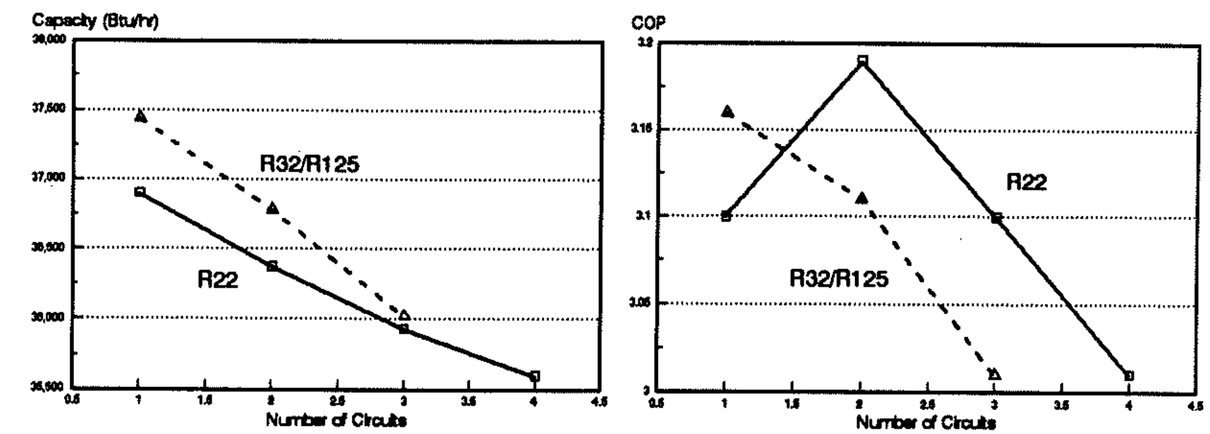

Figure 2

be more complicated and costly. The only refrigerant that significantly benefits from this heat exchanger is R134a (an approx. 2% increase in efficiency).

The fourth column ("D") of Table II lists the results of including the optimization of both the evaporator and condenser coils in the changes made to the original design of the heat pump. The cost of these coils were estimated using an estimation routine that factors in material cost, labor cost, and overhead to arrive at coil cost. Changes such as tube diameter and number of tubes, were made to the coil that did not change the total cost of the coil. During this process the number of circuits in the heat exchangers were evaluated again to arrive at an optimum COP based on coil geometry and number of parallel circuits. It should be noted that for both R134a and R152a there were no coil changes that would improve performance while maintaining a constant coil cost and the refrigerant lines transporting vapor were increased from 3/4" diameter to 7/8" diameter.

The equipment design that was optimized for cooling at an ambient temperature 82°F for each refrigerant was then run at ambient temperatures of 95°F (ARI "Test A") and 115°F (Maximum Operating Conditions). Figure 3 shows the performance comparison of the alternative refrigerants with R22. At 95°F, there is no significant change in relative performance from 82°F. When the ambient temperature reaches 115°F, there is a slight drop-off in performance of R32/R125 as compared to the other refrigerants. However, testing at this ambient temperature is for the purpose of determining the heat pump's or air conditioner's ability to run at these conditions and this temperature is rarely encountered in the field. It should be noted that the discharge temperatures for all the alternative refrigerants is lower than for R22. This should minimize the concern for satisfactory operation at this ambient temperature.

The computer model was then run using R22 in the heating mode at 47°F ambient and 70°F indoor temperature. Using the equipment design optimized for cooling, heating performance was significantly reduced from that of the original design. Optimizing for both heating and cooling "pushed" the R22 design back to the original configuration (which increased the level of confidence in the modelling results). The equipment designs for the alternative refrigerants were also optimized for both heating and cooling. Performance of these systems at ambient temperatures of 82°F, 95°F, and 115°F in the cooling mode and 47°F, 17°F, and 0°F in the heating mode is shown on Figure 4.

The cooling performance comparison among the refrigerants is about the same as when the systems are optimized for cooling only, however, the COP's are slightly reduced from the previous levels. In the heating mode, the performance of both R134a and R152a falls off at lower ambient temperatures, while R32/R125 keeps up with R22. The drop-off in capacity at lower temperatures would have an impact on the heat pump's energy consumption since the losses in capacity have to be made up with inefficient electric resistance heat in most cases. The discharge temperature of any of the alternatives do not present any problems since they all are lower than the baseline R22.

Thermodynamic Cycle Vs. System Modeling

It is interesting to compare the system modeling results to the results of thermodynamic cycle calculations. A simple thermodynamic cycle calculation was made using a 45°F evaporating temperature, a 115°F condensing temperature, 100% isentropic compressor efficiency, and no superheat or subcooling. Also, an calculation was performed that included 20°F subcooling and superheating and 70% isentropic compressor efficiency. The results of these calculations are presented in Figure 5 along with the system modeling results of the original design and the optimized designs.

Examining the results from the simple thermodynamic cycle (saturated cycle) results shows a drop in efficiency of 7.5% for R32/R125 and 1.5% for R134a, and an increase of 4.5% for R152a. The magnitude of these differences shrink significantly when subcooling and superheating are included in the analysis (also changes in the ranking occur). As shown in the system modeling results for the original configuration, the differences are reduced still further along with another ranking change when the entire system is modeled. When the system design is tailored for each refrigerant, there can be additional changes to the ranking of these refrigerants. These results emphasize the importance of evaluating alternative refrigerants in an optimized system.

Table III gives some insight into the reasons why the thermodynamic calculations fail to accurately compare alternative refrigerants. It shows some of the cycle temperatures and heat exchanger performance parameters optimized system (cooling and heating) running at 82°F. The basic assumption in the thermodynamic analysis is constant evaporating and condensing temperatures for all refrigerants. As can be seen from the compressor inlet and outlet saturation temperatures on Table III, they are not the same for all refrigerants. The reasons for the differences among these temperatures are changes in transport

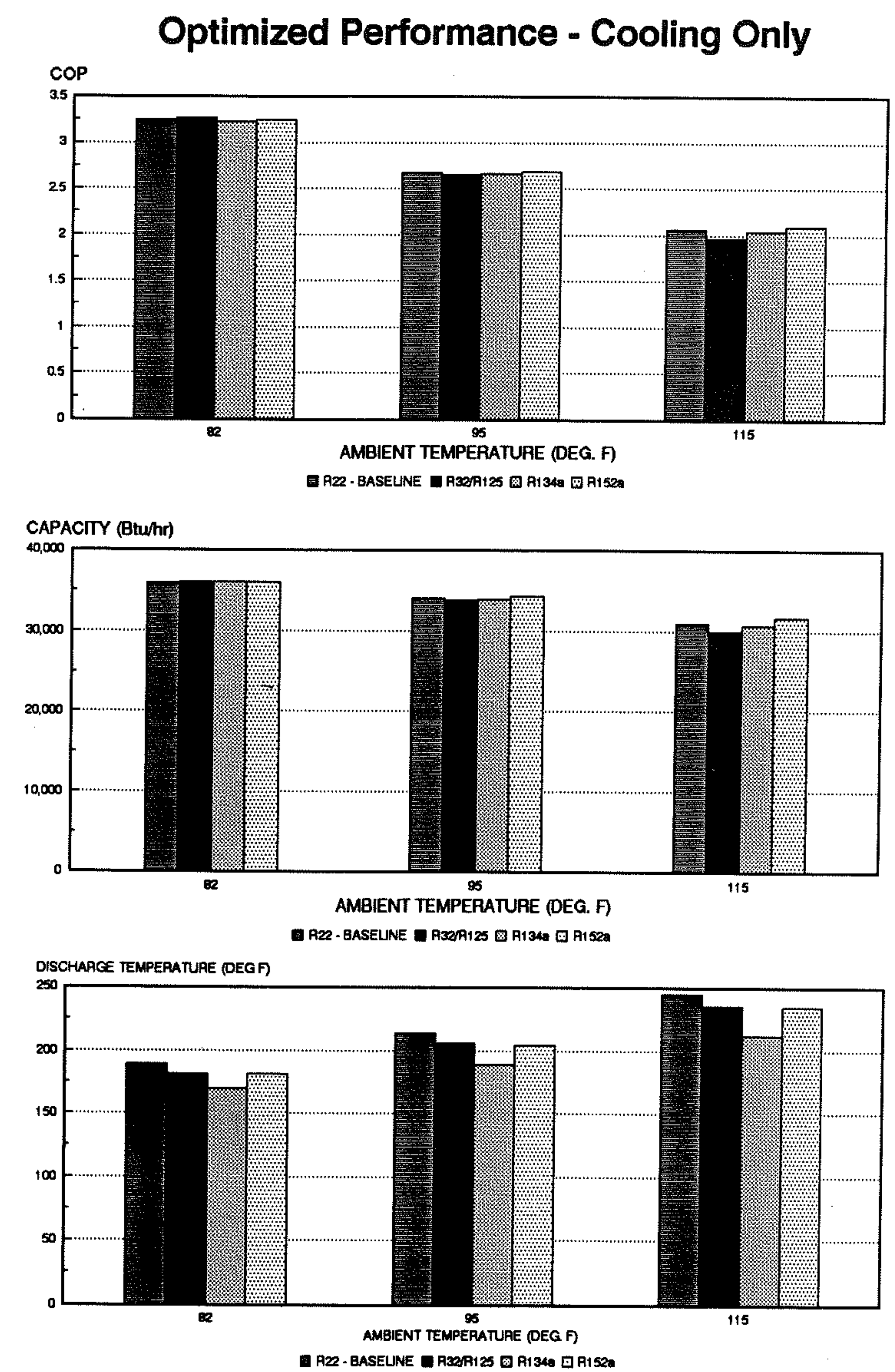

Figure 3

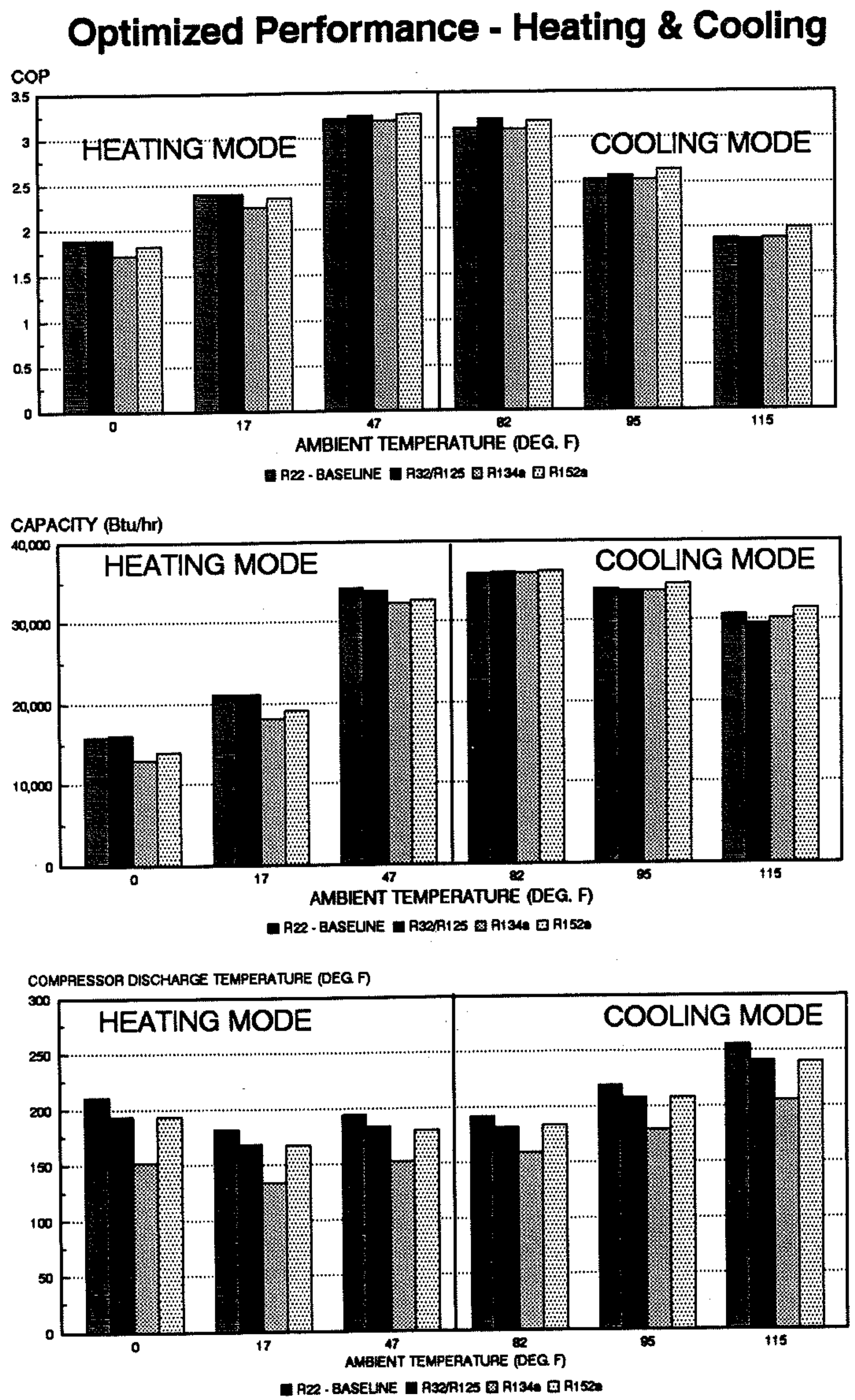

Figure 4

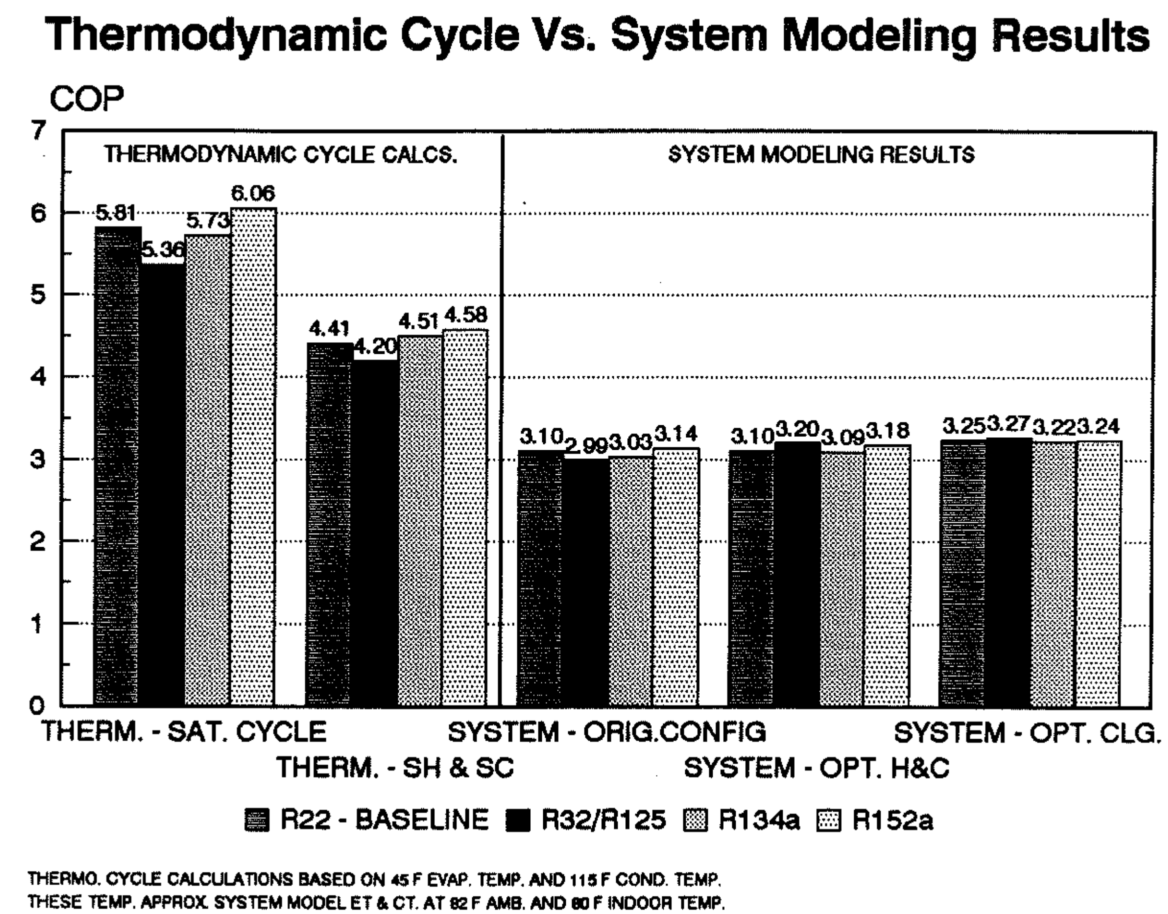

Figure 5

properties which impact both heat transfer and pressure loss characteristics and the impacts of refrigerant mass flow rate and vapor pressure. An example of how the refrigerant vapor pressure effects the optimum system is the comparison between the optimized R32/R125 evaporator and the R22 evaporator. The higher mass velocity through the R32/R125 heat exchanger results in both increased heat transfer and increased pressure drop (7.5 psi vs. 4.1 psi for R22), however the change in saturation temperature is about the same. The result is improved heat transfer and because there is no accompanying increase in saturation temperature drop (due to pressure drop), an increase in evaporation temperature for the R32/R125 system also results.

Conclusions

One can conclude that all three alternative refrigerants evaluated could be used without any significant energy penalty in an air conditioner or heat pump. In fact, the efficiency of the optimized heat pump for R32/R125 is approximately 1.5% higher than the optimized R22 heat pump. Another point that must be made is the importance of evaluating alternative refrigerant in optimized systems rather than judging their potential with overly simplistic thermodynamic cycle calculations.

Other factors that could have a significant impact on the air conditioner or heat pump need to be addressed. The flammability characteristics of R152a would either eliminate this refrigerant from

REFRIGERANT THERMODYNAMIC STATE POINTS FOR CONSTANT AIR INLET TEMPERATURES

		REFRIGERANT			
COMPONENT	PARAMETER	R22	R32/R125	R134a	R152a
EVAPORATOR	INLET SAT. TEMP. (DEG F)	49.1	50.8	51.3	50.3
	EXIT SAT. TEMP. (DEG. F)	46.6	47.9	45.0	45.4
	HX EFFECTIVENESS	0.746	0.787	0.745	0.745
COMPRESSOR	INLET SAT. TEMP. (DEG F)	45.7	47.6	43.9	44.6
	EXIT SAT. TEMP. (DEG. F)	120.9	117.8	121.6	120.4
CONDENSER	INLET SAT. TEMP. (DEG F)	120.8	117.8	121.5	120.3
	EXIT SAT. TEMP. (DEG. F)	120.2	117.2	120.4	119.4
	HX EFFECTIVENESS	0.383	0.419	0.407	0.400

Table III

consideration or at least impact the cost by requiring additional safety controls. For the R32/R125 azeotrope, the higher condensing pressure (approximately 50% higher than R22) could impact the cost of the system. The increase (for R134a and R152a) or decrease (for R32/R125) in compressor displacement could also effect system cost. Once these factors are determined, a final comparison of alternative refrigerants in a constant cost system could be made.

Future work planned for this project include modifying a non-azeotropic blend version of the heat pump model to include additional refrigerant mixtures such as R32/R134a and testing the heat pump with alternative refrigerants.

Acknowledgements

The author wished to acknowledge his colleagues, especially H. M. Hughes for his helpful discussions, and Allied-Signal Inc. for permission to publish this work.

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