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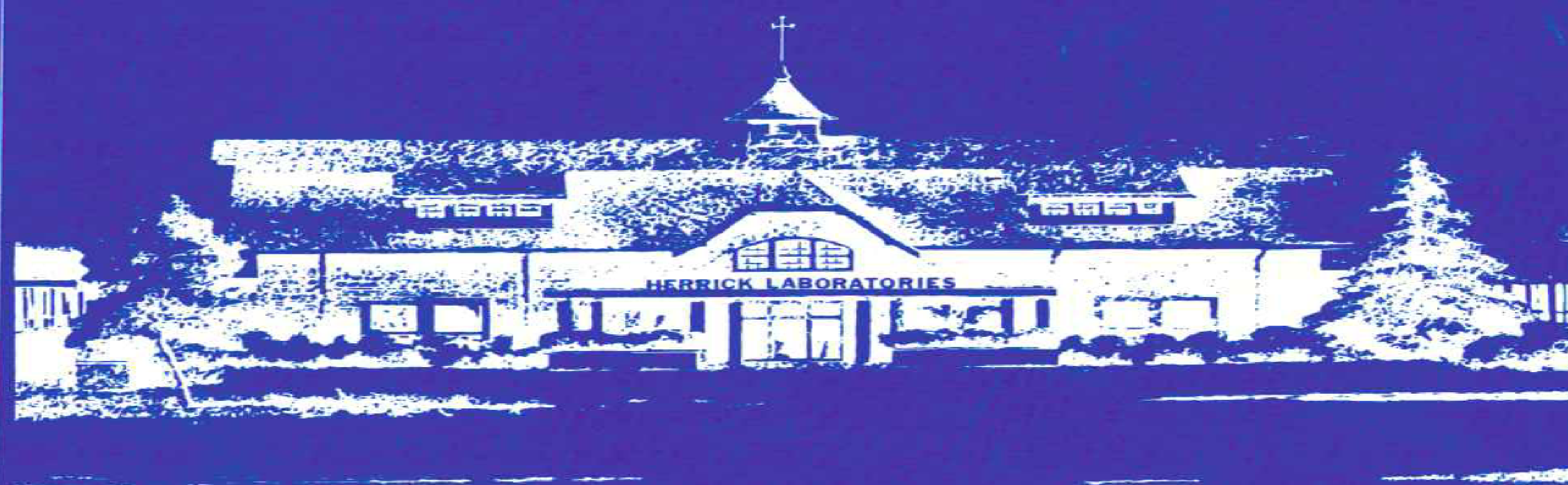

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Performance of Alternative Refrigerants In Heat Pumps and Air Conditioners

Mark W. Spatz
Jing Zheng
Allied-Signal Inc.
20 Peabody Street
Buffalo, NY 14210
Phone: (716) 827-6238 Fax: (716) 827-6221

Abstract

In this paper the results of both modeling and testing a representative 3-ton air-to-air heat pump running with the baseline refrigerant (HCFC-22) and zero ODP alternative refrigerants will be presented. 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. Pre-optimized and post-optimized system performance is presented to demonstrate the impact of this process on the ranking of replacement refrigerants. System tests of the heat pump and compressor calorimeter tests were run with the 32/125 azeotrope and a 32/134a (25/75 wt%) blend.

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 which are 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]. R134a, R152a, and the R32/R125 azeotropic mixture were chosen as candidate replacement fluids for the modeling study. Zeotropic blends of R32, R134a, and R125 were not modeled due to the need for extensive modifications of the model to evaluate such blends. However, a blend of R32 & R134a was included in the system test program discussed later in this paper.

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	34048	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 were 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. 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 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.

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

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). 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

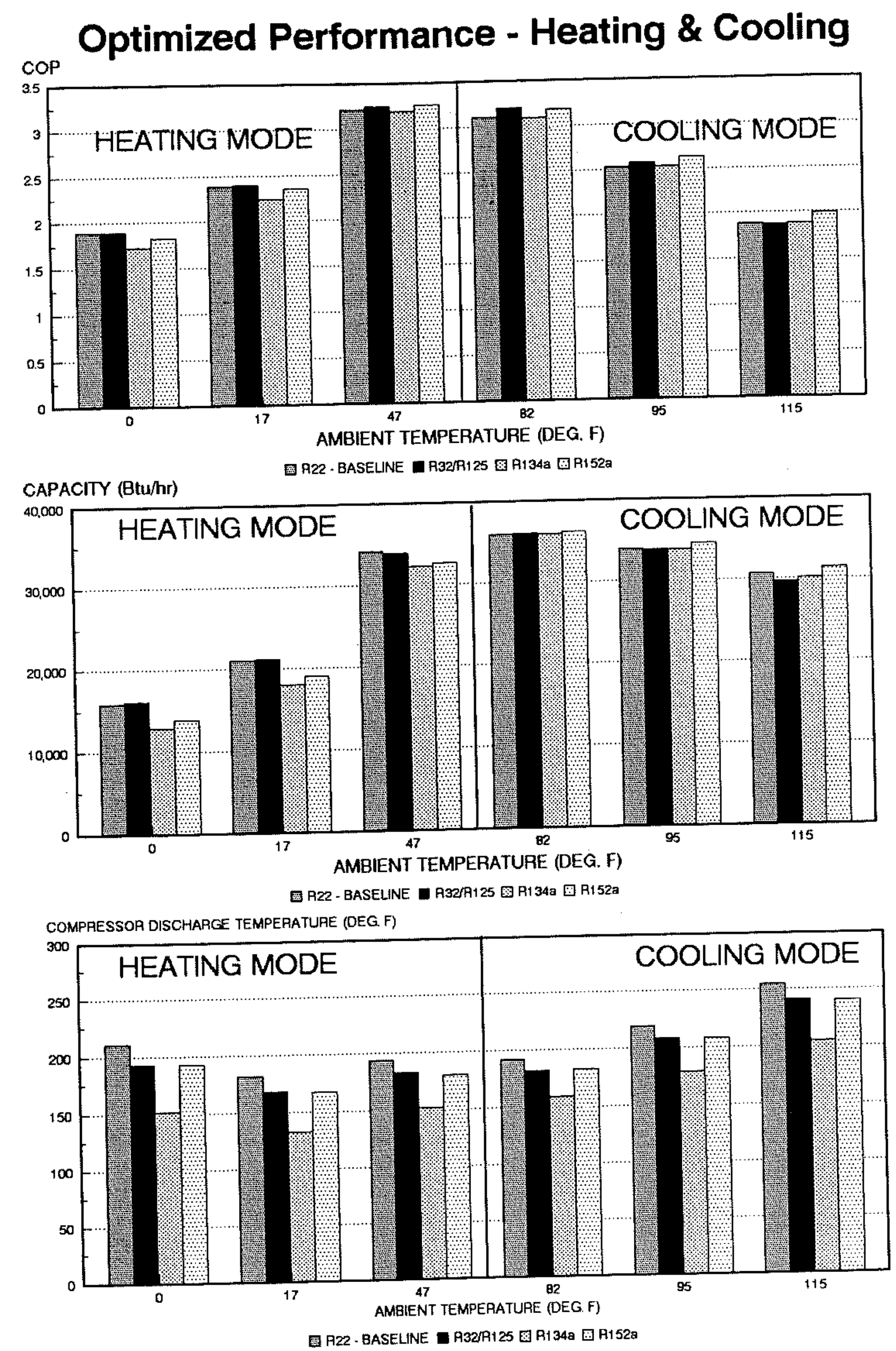

Figure 1

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 1.

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.

System & Compressor Test Results

Although detailed system modeling does take into account many more variables than simple thermodynamic cycle calculations, it cannot replace actual tests of components and systems when trying to evaluate alternative refrigerants. In order to evaluate alternative refrigerants in representative hardware, the 3-ton heat pump, that was used in the modeling effort, was installed in an air calorimeter test facility. This facility provides controlled conditions for both the indoor and outdoor units. The heat pump was instrumented with refrigerant and air side temperature and pressure sensors, an indoor air flow meter, a Coriolis refrigerant mass flow meter, and watt transducers.

One of the issues that needs to be addressed when testing alternative refrigerants is the variation in capacity of these fluids. If a fluid with either higher or lower capacity than R22 is dropped into a heat pump or air conditioner designed for R22, the compressor, heat exchangers, and other components would either be oversized or undersized and would result in additional impacts on efficiency (in addition to the refrigerant's impact). To compensate for this variation in capacity, a special compressor that is designed to run with an inverter was installed in the test heat pump. An inverter was also installed in the test facility which enabled the compressor to run at varying speeds. The speed variation and thereby displacement variation was used to match the capacity of the fluid under test with R22.

In addition to the system tests, a second compressor (identical to the compressor installed in the heat pump) was installed in a secondary refrigerant compressor calorimeter. The impacts of the inverter and speed variation was quantified in these tests.

Results of System Tests

Conditions: 82 F Outdoor; 80 F DB/67 F WB Indoor (ARI Test B)

	Refrigerant		
Parameter	R-22	R32/125 (60/40 wt%)	R32/134a (25/75 wt%)
Frequency/Volts	60 Hz./230 Volts	40 Hz./153 Volts	65 Hz./ 249 Volts
Capacity (Btu/hr)	39720	40365	40187
Compressor Power (Watts)	3835	3805	3790
Total Power (Watts)	4435	4405	4390
E.E.R. (Btu/hr.-W)	8.96	9.16	9.15
Sat. Temp. @ Suction (Deg F)	41	44	39
Sat. Temp. @ Disch. (Deg F)	118	115	118
Refrigerant Mass Flow (lb/hr)	550	495	489

Note: Saturation temp. for the blend refers to the midpoint of the glide at the respective pressure.

Table III

A series of baseline tests with R22 was conducted with both the heat pump and compressor calorimeter. Following these tests, the mineral oil lubricant was drained from the compressors. In the heat pump, polybutylene glycol (PBG) lubricant was charged into the compressor and run with R22. PBG was used because of its mutual miscibility with mineral oil and the modified PAG lubricant that was used with the HFC refrigerants. This served to remove nearly all of the mineral oil left in the system. The compressor was then charged with a mixture of PBG and modified PAG, and finally with 100% modified PAG lubricant along with the HFC refrigerant.

System tests were run at ARI Test A & B conditions for the 32/134a blend and at Test B conditions (82°F outdoor temperature) for the 32/125 azeotropic blend. Testing 32/125 was limited to 82°F due to the high current draw at higher temperature when running at lower speed (lower frequency & voltage). Tables III and IV show the results of these tests. The capacity match frequency for 32/125 was approximately 40 hertz and 65 hertz for 32/134a. The efficiencies were slightly greater (approx. 2%) for both 32/125 and 32/134a. It is worthwhile noting that the saturation temperatures at the suction to the compressor were higher for 32/125 than for 22 and also lower at the compressor discharge. This is indicative of higher heat transfer coefficients and the effect of the higher vapor pressure on the pressure drop impact on saturation temperature. The saturation temperature at the compressor suction for 32/134a is actually the midpoint of 80% of the glide (assuming the blend enters the evaporator at 20% of the temperature difference between the bubble and dew points) at the suction pressure. The lower saturation temperature would indicate either poorer heat transfer or pressure loss characteristics for the blend. Test results at 95°F for the 32/134a blend show the same trends as the 82°F point.

Results of System Tests

Conditions: 95 F Outdoor; 80 F DB/67 F WB Indoor (ARI Test A)

Parameter	Refrigerant R-22	Refrigerant R32/134a (25/75 wt%)
Frequency/Volts	60 Hz./230 Volts	65 Hz./ 249 Volts
Capacity (Btu/hr)	36728	36524
Compressor Power (Watts)	4125	3958
Total Power (Watts)	4725	4558
E.E.R. (Btu/hr.-W)	7.77	8.01
Sat. Temp. @ Suction (Deg F)	43	41
Sat. Temp. @ Disch. (Deg F)	130	130
Refrigerant Mass Flow (lb/hr)	541	473

Note: Saturation temp. for the blend refers to the midpoint of the glide at the respective pressure.

Table IV

Compressor calorimeter tests have been completed for R22 and for the 32/125 azeotrope. Tests of the 32/134a blend will be run shortly. Figure 2 shows the performance of R32/125 (at 40 hertz) as compared to R22 (at 60 hertz). If comparing the two refrigerants at the same evaporating and condensing temperatures, the capacity of 32/125 is less (5 to 15%) over most of the temperatures tested. The efficiency is about the same at 100°F condensing and somewhat lower at 120°F. Tests were also performed at 40 and 60 hertz for both R22 and 32/125 and the results are shown on figure 3. There is a 3 to 5% efficiency penalty for operating at the lower frequency over nearly all the test points. If this factor were

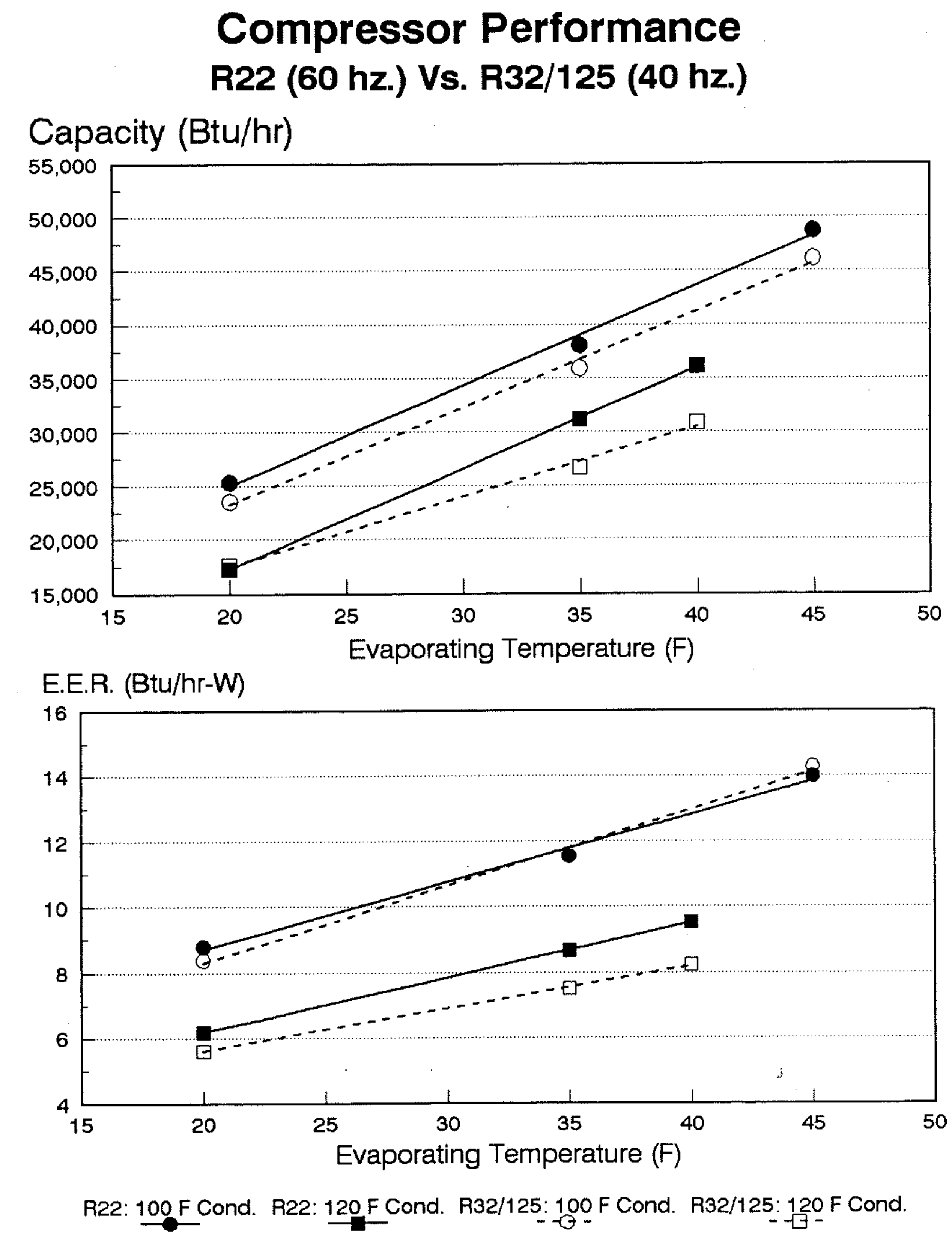

Figure 2

applied to the system test results, it would increase the efficiency of the heat pump running with 32/125 by this factor, resulting in an efficiency gain of at least 5% better than R22. Tests will be run with the 32/134a blend to determine the impact of running the compressor at 65 hertz. It is expected that any efficiency impact would be less.

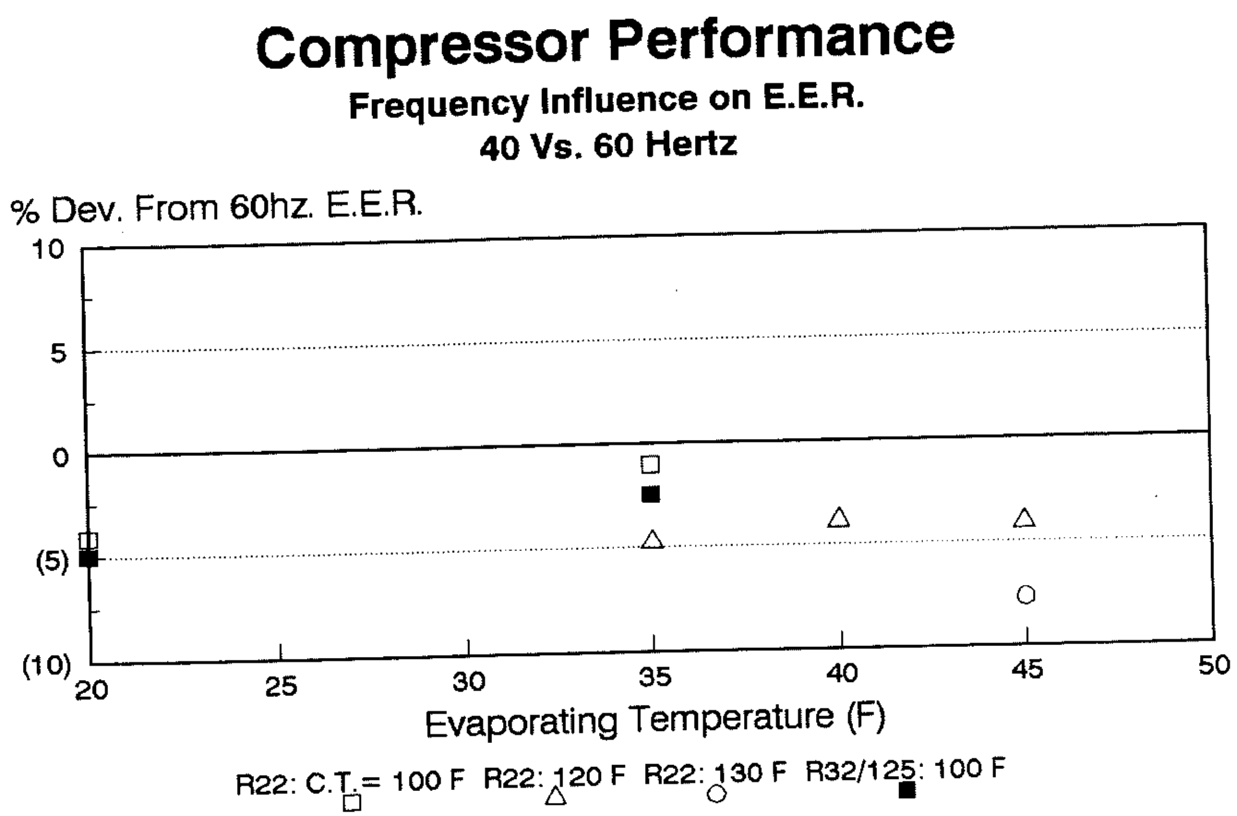

Figure 3

Conclusions and Planned Work

One can conclude that a number of alternative refrigerants could be used without any significant energy penalty in an air conditioner or heat pump. In fact, the efficiency of an optimized heat pump for R32/R125 could be significantly higher than the optimized R22 heat pump. Test results of a system optimized for R22 but using 32/125 as the refrigerant showed an efficiency improvement rather than the 3% deficiency the model predicted (due in large part to the higher than predicted heat transfer). Further efficiency gains would be expected through system optimization. The 32/134a blend also showed an efficiency improvement over R22. Optimization and taking advantage of the temperature glide could increase the efficiency further, but when trying to take advantage of the glide the impact on parasitic power consumption and other system characteristics needs to be considered[7]. 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 issue of blend segregation needs to be addressed by the air conditioning industry. For the R32/R125 azeotrope, the higher vapor pressure (approximately 50% higher than R22) could have a significant impact on the design and possibly the cost of the system. The flammability characteristics of R152a could either eliminate this refrigerant from consideration or at least impact the cost by requiring additional safety controls. 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 testing with a 32/125/134a blend and pure R134a, system tests using a compressor designed for 32/125, testing of optimized systems, and modifying a non-azeotropic blend version of the heat pump model to include additional refrigerant mixtures such as R32/R134a.

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Tree, Braun

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Ray W. Herrick Laboratories
School of Mechanical Engineering
Purdue University