DOE/CE/23810-7

THEORETICAL EVALUATION OF R22 AND R502 ALTERNATIVES

Final Report

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January 1993

Prepared for
The Air-Conditioning and Refrigeration Technology Institute
Under
ARTI MCLR Project Number 650-50900

This research project is supported, in whole or in part, by U.S. Department of Energy grant number DE-FG02-91CE23810: Materials Compatibility and Lubricants Research (MCLR) on CFC-Refrigerant Substitutes. Federal funding supporting this project constitutes 93.67% of allowable costs. Funding from non-government sources supporting this project consists of direct cost sharing of 6.33% of allowable costs; and in-kind contributions from the air-conditioning and refrigeration industry.

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ABSTRACT

The report presents the performance evaluation of nine R22 alternatives and three R502 alternatives. The study was conducted using a semi-theoretical model, CYCLE-11, with a pure cross-flow representation of heat transfer in the evaporator and condenser. The Carnahan-Starling-DeSantis equation of state was used for calculating thermodynamic properties. Transport properties were not involved in the simulations.

Simulations were conducted for "drop-in" performance, for performance in a modified system to assess the fluids' potentials, and for performance in a modified system equipped with a liquid-line/suction-line heat exchanger. The results - presented on a relative basis to R22 and R502 performance - include the volumetric capacity, coefficient of performance, pressure increase across the compressor, and compressor discharge pressure and temperature.

SCOPE

The evaluation of nine R22 alternative fluids was performed at two cooling and two heating operating conditions that represent 35.0°C (95°F), 27.8°C (82°F), 8.3°C (47°F), and -8.3 (17°F) temperature rating points for a residential heat pump [1]. The R22 alternatives considered in this study are: R32/125 (60/40), R32/125/134a/290 (20/55/20/5), R32/125/134a (10/70/20) and (30/10/60), R290, R32/227ea (35/65), R32/134a (25/75) and (30/70), and R134a.

Three R502 alternatives were evaluated at one operating condition typical for commercial refrigeration installations. The heat-transfer-fluid temperature entering the evaporator and condenser was -23.3°C (-10°F) and 35°C (95°F), respectively. The evaluated R502 alternatives are: R32/125/143a (10/45/45), R125/143a (45/55), and R125/143a/134a (44/52/4).

This study is restricted to the thermodynamic merits only and does not concern itself with other attributes like transport properties, toxicity, and flammability. The pure cross-flow representation of the evaporator and condenser used in these simulations makes the results of this study most applicable to systems equipped with cross-flow heat exchangers.

This is the only and final report covering the study performed.

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SIGNIFICANT RESULTS

INTRODUCTION

Due to an implication in the destruction of the stratospheric ozone layer, chlorine-containing refrigerants are scheduled for phase-out by the Montreal Protocol and subsequent regional regulations. Chlorofluorocarbons (CFCs), which are characterized by a high ozone depletion potential (ODP), are slated for complete production phase-out in the United States by the end of 1995. For hydrochlorofluorocarbons (HCFCs), the 1992 revision of the Montreal Protocol stipulates the first production ceiling for the year 1996 and the final elimination in 2030.

In response to the phase-out regulations, the Air-Conditioning and Refrigeration Institute (ARI) established the Alternative Refrigerants Evaluation Program (AREP) to pool together efforts of ARI member companies in search of alternative fluids for R22 (HCFC-22) and azeotropic mixture R502 (48.8% HCFC-22 and 52.2% CFC-115). The objective of this report is to provide a computer evaluation of alternative refrigerants and refrigerant mixtures. This evaluation is restricted to thermodynamic merits only and does not concern itself with other attributes like transport properties, toxicity, or flammability.

Impact of Fluid Properties on System Performance - General Remarks

To better understand the results presented in this report, let's briefly recite why different fluids exhibit different performance in a given application. The reason lies in their different thermodynamic and transport properties. Although all properties are important, some are more important than others.

In the thermodynamic properties category, the most important parameters are the normal boiling point temperature - or related to it critical temperature (T_{cr}) - and heat capacity. Figure 1 depicts the impact of the critical temperature by

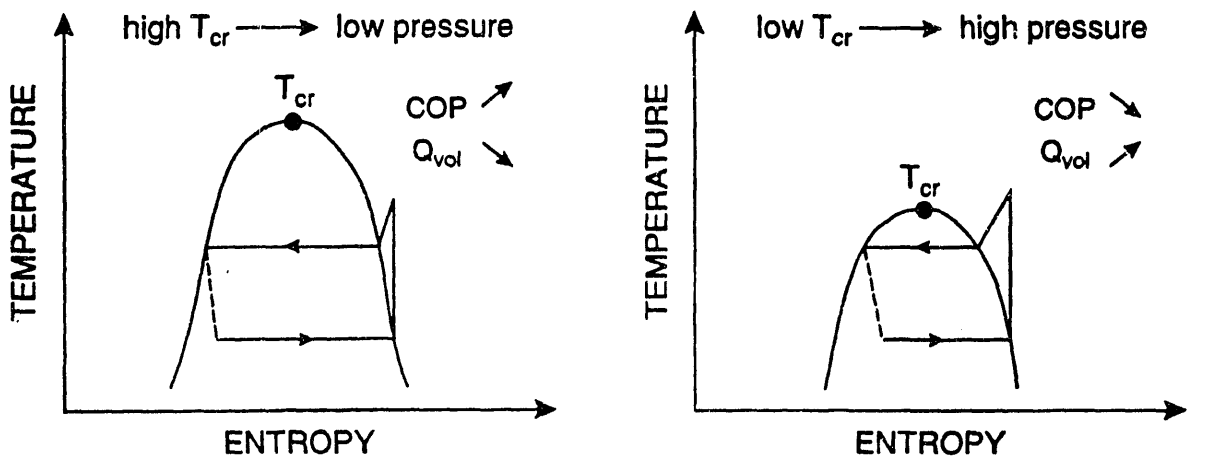

Figure 1. Impact of critical temperature on cycle performance.

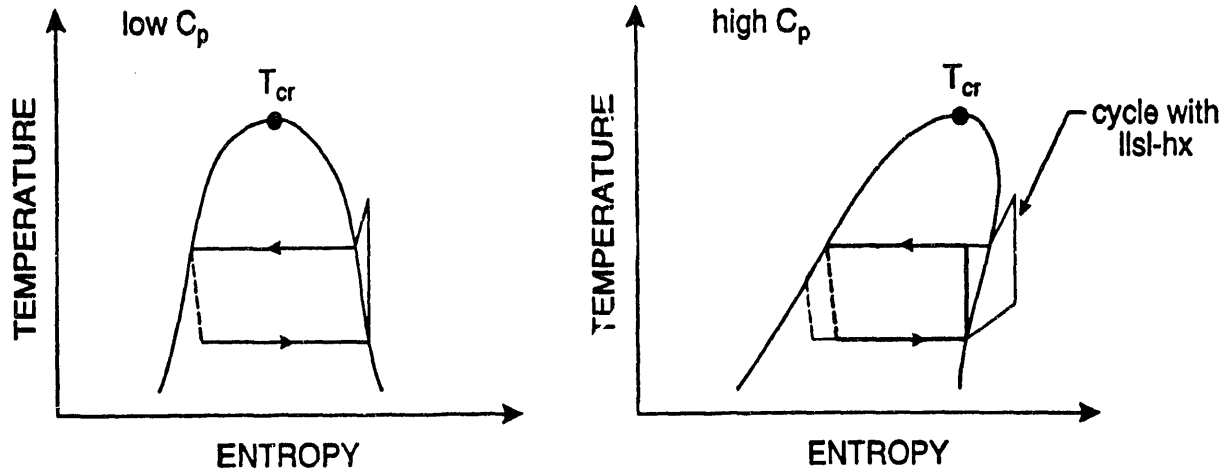

Figure 2. Impact of heat capacity on refrigeration cycle

showing a reversed Rankine cycle realized by two different fluids. The more volatile fluid (lower T_{cr}) starts evaporation at a higher vapor quality and has a larger superheated vapor horn - both attributes contribute to a lower coefficient of performance (COP). However, this refrigerant will have a higher volumetric capacity (Q_{vol}) since a lower critical temperature results in a higher pressure at the compressor inlet. Thus, the trade-off between the COP and volumetric capacity is unavoidable; a fluid that exceeds another fluid in volumetric capacity will most likely have a lower COP. It may be noted that differences in volumetric capacity are much sharper than differences in the COP.

Heat capacity affects performance profoundly through its impact on the outline of the two-phase dome. A refrigerant with a high heat capacity has its two-phase

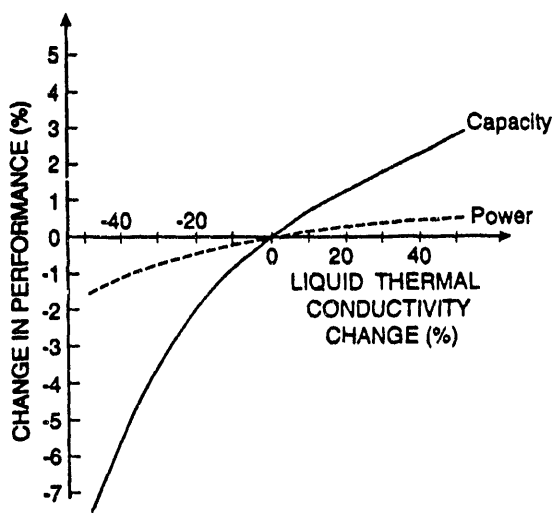

Figure 3. Impact of liquid thermal conductivity on performance of air-to-air heat pump

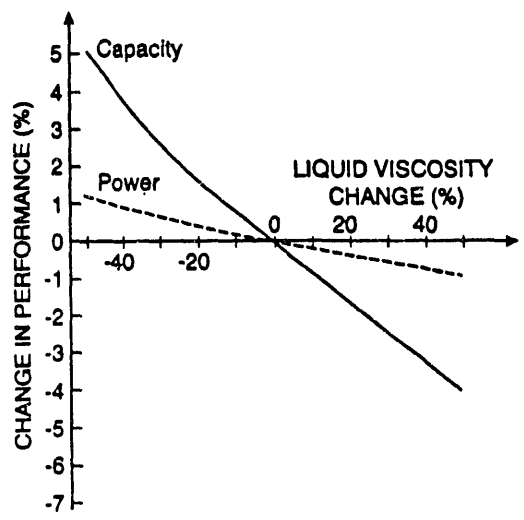

Figure 4. Impact of liquid viscosity on performance of air-to-air heat pump

dome skewed to the right, which may result in significant flashing losses and undesirable "wet compression", as showed in Figure 2. A cycle with a liquid-line/suction-line heat exchanger (llsl-hx) - shown by a thin line on the diagram for high-heat-capacity fluid - is relieved from these undesirable conditions and may have a better COP than a basic reverse Rankine cycle [2].

The impact of transport properties will vary with the heat exchanger effectiveness, heat transfer surfaces, and with a heat transfer resistance on the side of heat transfer fluids (HTFs). For an air-to-air heat pump charged with R22 and employing a smooth-tube evaporator and condenser, liquid thermal conductivity and viscosity were found to be most influential, as showed in Figures 3 and 4 taken from reference [3]. These figures are provided only to allow estimation of the influence of liquid thermal conductivity and viscosity; transport properties are not involved in the following performance evaluations.

Performance Comparison Methods: "Drop-In" vs. Equal Heat Exchanger Loading

We have to distinguish between two basic methods for performance comparison of different fluids:

1. "drop-in" evaluation, and
2. evaluation at a constant heat exchanger loading.

In the "drop-in" method, a new refrigerant is evaluated in a machine designed for the original refrigerant (the expansion device may be modified to assure the same subcooling at the condenser outlet). This is a typical first-cut test performed in a laboratory. "Drop-in" testing, although telling us the performance we may expect in a given machine, does not give us a fair comparison between the fluids. This lack of objectivity results from the changed system capacity when the same system is charged with refrigerants having different volumetric capacities. Different system capacities cause different heat fluxes through the evaporator and condenser resulting in different temperatures in the condenser and the evaporator for transfer of heat to/from the heat transfer fluids, if the heat exchangers remain the same, as is assumed.

The second, more objective comparison methodology (McLinden and Radermacher [4]) requires a constant ratio of capacity to the total heat transfer area (evaporator plus condenser) for all the fluids analyzed. This requirement - difficult to implement in laboratory testing but easy to impose in computer simulations - assures the same heat exchanger area loading despite differences in volumetric capacity between refrigerants studied.

The second approach is more objective; however, "drop-in" applications are also of interest. Since both approaches have their merits, this study includes both "drop-in" simulations and simulations in modified equipment at the same heat flux through the evaporator and condenser.

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Modeling Tools

Three semi-theoretical cycle models, derived from CYCLE-11 [5] and described in Appendix B, were used to perform simulations. The models do not include transport properties; they carry the implicit assumption that transport properties (and the overall heat transfer coefficients) are the same for the fluids studied. With the understanding of the assumptions involved, our confidence in the CYCLE-11 programs was established through various joint laboratory/simulation projects, e.g. the study of R22, R32/134a, and R32/152a [6].

Thermodynamic properties are represented in the cycle models through the Carnahan-Starling-DeSantis (CSD) equation of state, as implemented in the REFPROP package [7]. Not all pure fluids considered in this study and their interaction coefficients are included in REFPROP. The needed data for fluids not covered by REFPROP were obtained from G. Morrison [8].

It has to be noted that the CSD equation of state does not accurately represent highly polar fluids like R32, which is included in this report. The CSD equation of state was the only option available to us. A new REFPROP version based on the Carnahan-Starling-DeSantis-Morrison (CSDM) equation of state [9] - a CSD successor that can handle polar refrigerants - did not cover all the needed fluids at the time of this study.

EVALUATION OF R22 ALTERNATIVES

R22 alternatives were evaluated at conditions approximating two cooling and heating rating points for residential heat pumps [1]. These conditions, represented by temperatures of the heat transfer fluids (air), are shown in Table 1. R22 alternative fluids considered in this study are shown in Table 2. The thermodynamic data included in Table 2 were obtained from REFPROP. Refrigerant 22 - specified as fluid number 0 - was used for reference purposes in presenting the simulation results.

"Drop-In" Evaluation
(Unchanged System, Constant Heat Exchangers)

"Drop-in" simulations were performed using CYCLE-11.UA, which requires selecting (pre-coding) a system to be simulated, similar to "drop-in" tests in a laboratory where a certain heat pump has to be selected for testing. In the case of CYCLE-11.UA, a compressor, condenser and evaporator must be designated.

A physical description of the compressor is needed, which, at the minimum, requires the compressor swept volume and RPM as input. From two options available for the compressor simulation, a more detailed representation of a hermetic compressor was selected for "drop-in" runs, which required specification of the electric motor efficiency, polytropic efficiency, and pressure drop and heat transfer parameters for the processes taking place between four key locations within a hermetic compressor [Appendix B]. The values for these parameters were selected to obtain pressure drop and temperature change typical

Table 1. Inlet and Outlet Temperatures of Heat Transfer Fluid (Air) used for Evaluating R22 and R502 Alternatives

	R22				R502
	Cooling		Heating		Refrigerating
	°C (°F)	°C (°F)	°C (°F)	°C (°F)	°C (°F)
Condenser Inlet	35.0 (95)	27.8 (82)	21.1 (70)	21.1 (70)	35 (95)
Condenser Outlet	43.2 (110)	37.4 (99.3)	32.5 (90.5)	28.1 (82.6)	40.6 (105)
Evaporator Inlet	26.7 (80)	26.7 (80)	8.3 (47)	-8.3 (17)	-23.3 (-10)
Evaporator Outlet	14.4 (58)	13.8 (56.8)	2.7 (36.9)	-11.3 (11.7)	-27.2 (-17)

for R22 at a 35°C (95°F) temperature test. The value of 0.9 was selected for both the electric motor efficiency and polytropic efficiency. The chosen values for the two efficiencies and pressure and heat loss parameters resulted in an overall "black-box" polytropic efficiency of approximately 0.7.

The evaporator and condenser were simulated as pure cross-flow heat exchangers, and were represented by their respective overall conductances, UA_e and UA_c. The values for UA_e and UA_c were selected such that 7.8°C (46°F) and 46.1°C (115°F) refrigerant saturation temperatures at the evaporator and condenser outlet, respectively, were obtained at R22 simulations at the 35°C (95°F) test condition. Additionally, the following parameters were imposed as constants in all simulation runs:

- zero degree of vapor superheat at the evaporator outlet,
- zero degree of liquid subcooling at the condenser outlet,
- 34.5 kPa (5 psi) pressure drop in the evaporator,
- 34.5 kPa (5 psi) pressure drop in the condenser.

Once a heat pump was defined, simulations were performed at four operating conditions listed in Table 1. Note that the assumption of the constant temperature profile of air in "drop-in" simulations implies different mass flow of air, depending on the capacity of the system charged with a particular refrigerant. The impact of a resulting changes in the air-side heat transfer coefficient and fan powers are not considered by the simulation model.

Five indicators of performance were selected for presenting the simulation results: capacity, COP, pressure increase across the compressor, compressor discharge pressure, and compressor discharge temperature. All results are presented on a relative scale with respect to performance of R22.

Table 2. R22 Alternative Fluids Considered in the Study

	Refrigerant	Weight composition %	$T_{dew\ point}$ (1)		T_{glide} (2)		T_{cr} (3)		Molecular weight kg/kmol
			°C	°F	°C	°F	°C	°F	
0	R22	100	-40.9	-41.6	0	0	96.2	205.1	86.47
1	R32/125	60/40	-53.1	-63.5	0	0	84.0	183.1	67.27
2	R32/125/134a/290	20/55/20/5	-45.2	-49.4	8.5	15.4	80.3	176.5	86.80
3	R32/125/134a	10/70/20	-42.4	-45.1	5.4	9.7	83.5	182.2	102.94
4	R290	100	-40.0	-40.0	0	0	96.7	206.0	44.10
5	R32/125/134a	30/10/60	-36.0	-32.8	7.6	13.7	97.4	207.3	80.13
6	R32/227ea	35/65	-35.3	-31.5	20.6	37.2	75.3	167.5	94.78
7	R32/134a	30/70	-34.4	-30.0	7.3	13.1	100.0	212.1	79.19
8	R32/134a	25/75	-33.1	-27.6	7.0	12.6	101.4	214.6	82.26
9	R134a	100	-26.2	-15.1	0	0	101.2	214.1	102.03

(1) at atmospheric pressure

(2) $T_{glide} = T_{dew\ point} - T_{bubble\ point}$

(3) REFPROP estimates

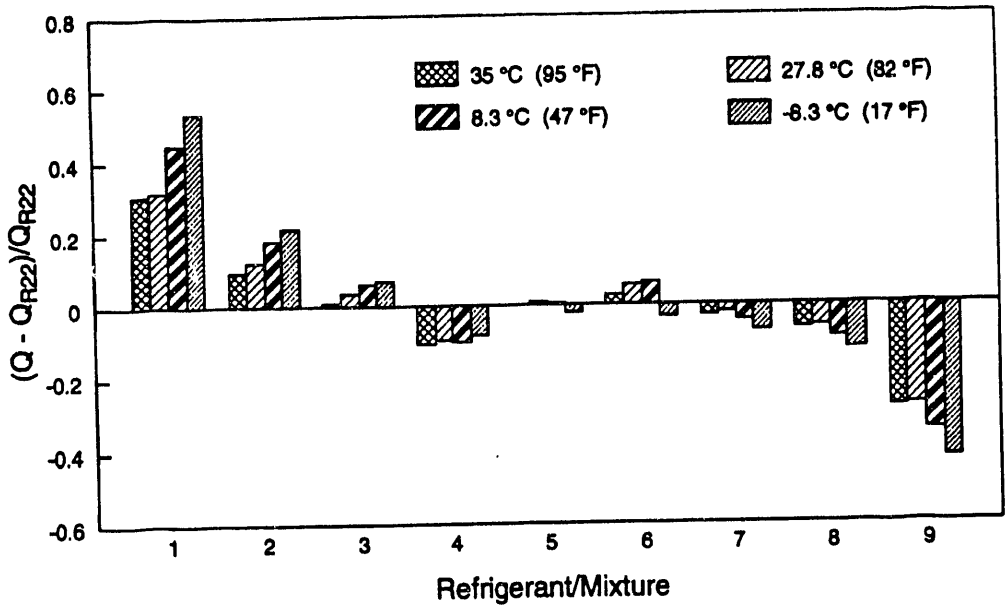

Figure 5. Volumetric capacity for "drop-in" simulations

Figure 5 shows a difference in capacity between candidate fluids and R22 divided by the capacity of R22. The four bars shown for each fluid represent results at four operating conditions denoted on the figure by the temperature of air entering what would be the outdoor coil - a customary designation of operating conditions for residential heat pumps.

The results are consistent with basic expectations relating volumetric capacity to refrigerant pressure. The highest pressure fluid, R32/R125 mixture (#1), has the highest capacity, while the lowest pressure fluid, R134a (#9), has the lowest capacity. The range of capacity difference for the screened fluids with respect to R22 is almost ± 50 percent.

Figure 6 shows a difference in COPs on a relative basis. If we imposed this figure over Figure 5, we could notice that refrigerants having a lower capacity display a higher COP than R22. We may notice that the difference in COP is not as large as the difference in capacity; in the extreme cases the COP difference does not exceed ± 20 percent.

Figure 7 presents the difference in pressure increase across the compressor between the candidate fluids and R22 divided by the value for R22. The results presented in this figure are indicative of the forces carried by the compressor bearings. A general trend in Figure 7 is consistent with the trend displayed in Figure 5: higher capacity fluids exhibit a higher pressure increase across the compressor because they operate at higher reduced temperatures, at which dP/dT gradient at saturation is greater. We can observe the same pattern in Figure 8, which shows the difference in discharge pressures. Significant differences for high pressure fluids with respect to R22 [for example, over 1200 kPa (174 psi) for fluid #1] indicate possible safety problems if those fluids are tested in unmodified R22 equipment.

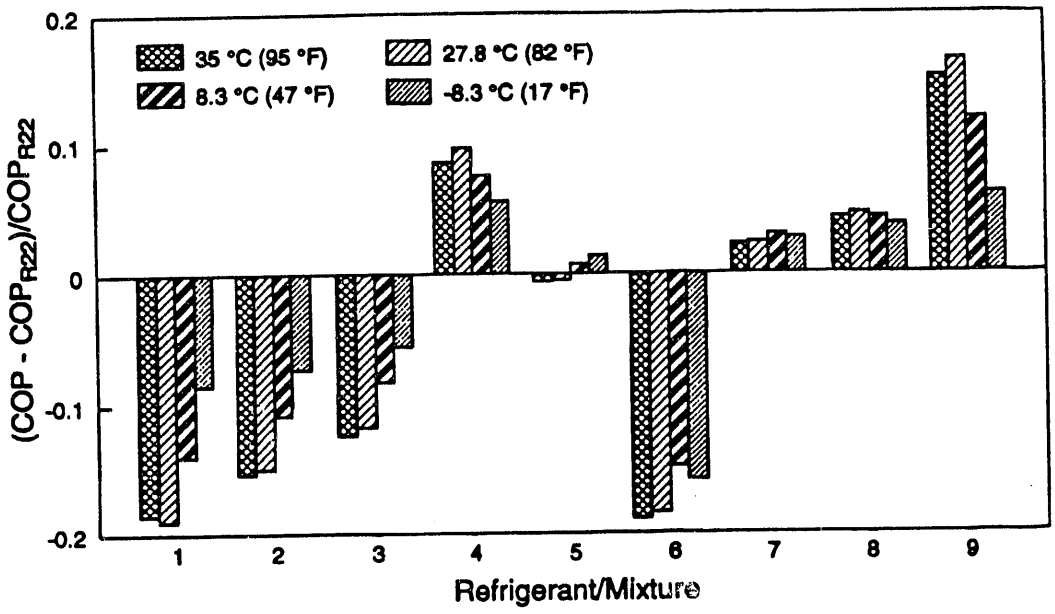

Figure 6. Coefficient of Performance for "drop-in" simulations

Figure 9 displays a difference between compressor discharge temperature for candidate fluids and R22. This information is useful in the consideration of lubricant and refrigerant stability. Except for fluid #1, all remaining refrigerants had a lower discharge temperature than R22.

In addition to the above simulation results, it is of interest to examine the average temperature in the evaporator since this temperature is a measure of the dehumidification capability of the fluid considered. As shown in Figure 10,

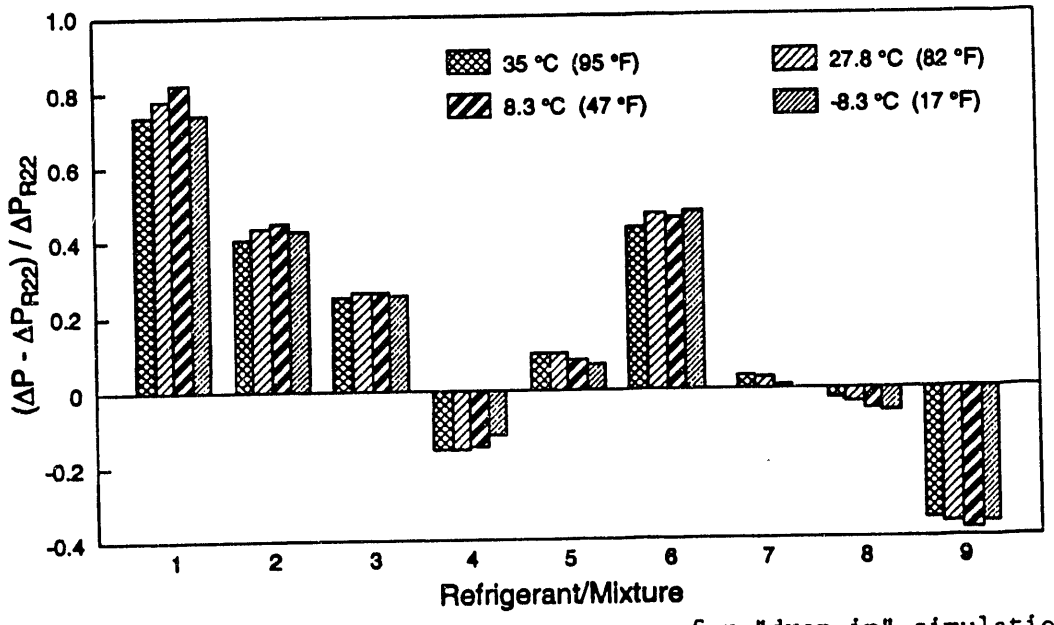

Figure 7. Pressure increase across compressor for "drop-in" simulations

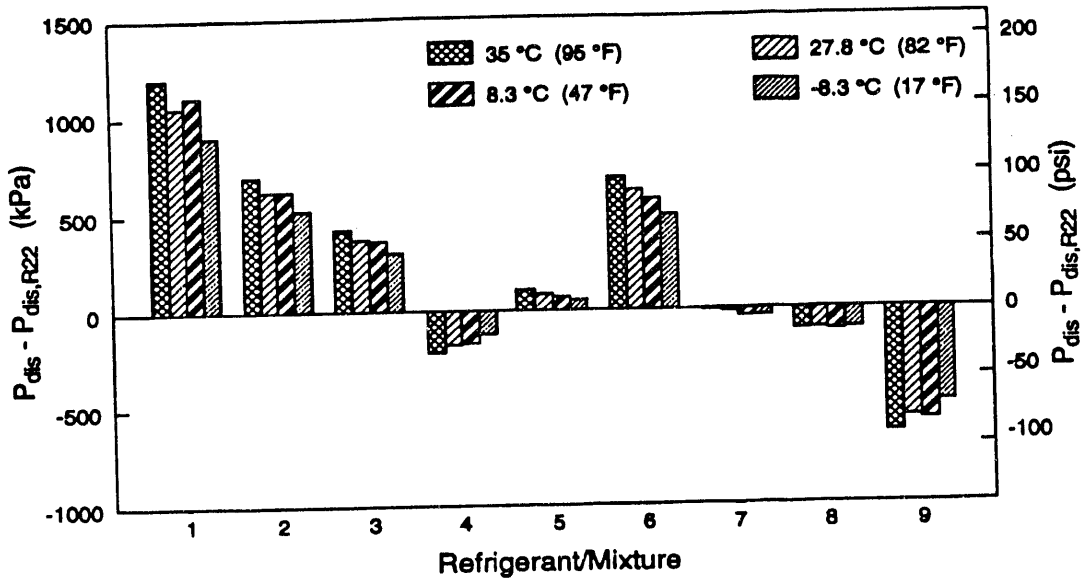

Figure 8. Compressor discharge pressure for "drop-in" simulations

temperatures different from R22 were observed - from a high pressure R32/R125 mixture having the lowest average temperature to a pure R134a having the highest average temperature, exceeding 10°C (50°F).

"Drop-in", constant-heat-exchanger-area simulations provide us with information concerning "drop-in" performance; however, these simulations do not facilitate an objective comparison. For example, a lower-pressure, lower-capacity fluid (like R134a) will work at a smaller temperature difference between the condenser

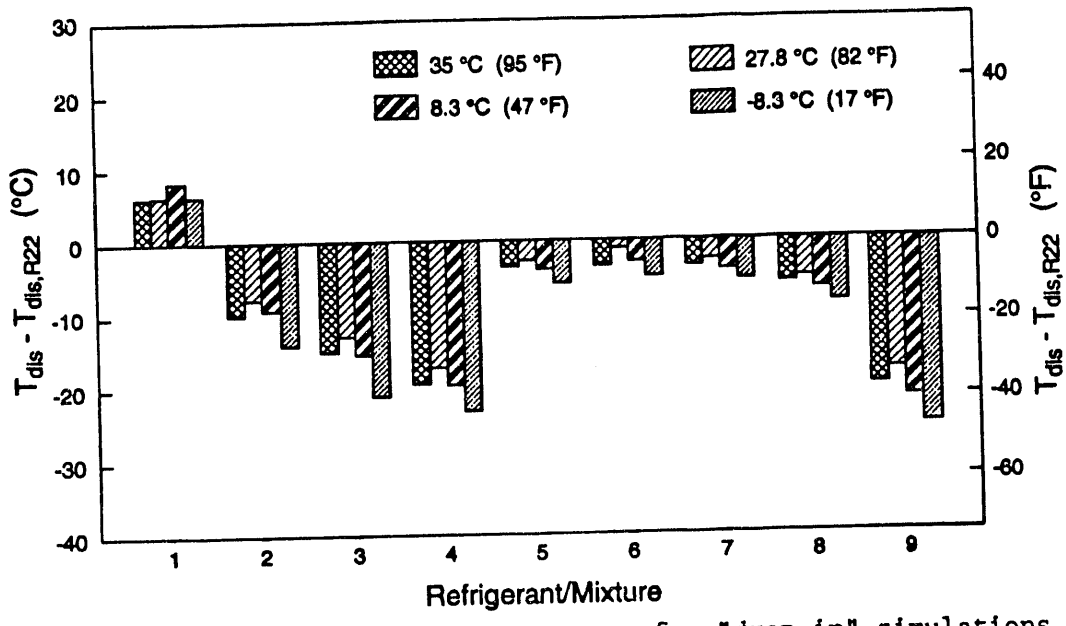

Figure 9. Compressor discharge temperature for "drop-in" simulations

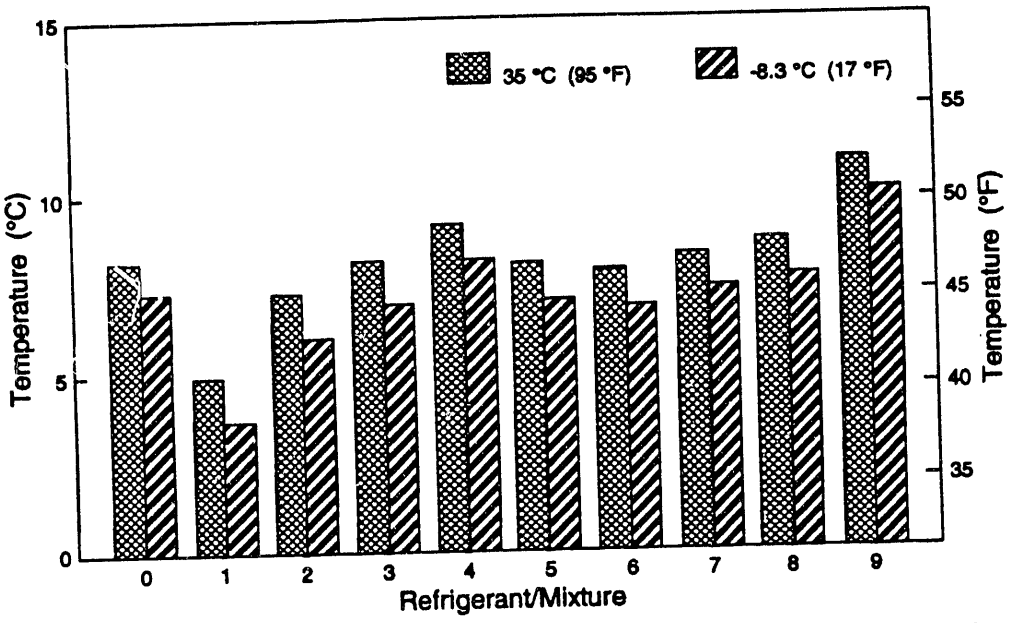

Figure 10. Average refrigerant temperature in the evaporator for "drop-in" simulations

and evaporator than R22 (because less heat has to be transferred). This results in a yet lower capacity and higher COP; i.e., a constant-heat-exchanger evaluation amplifies the fundamental trade-off between capacity and COP discussed at the beginning of this paper. The opposite COP vs. capacity trend can be noted for higher pressure fluids - with an important distinction that higher evaporator temperatures for lower pressure fluids may result in loss of latent capacity.

Evaluation at Constant Heat Exchangers Loading
(Modified Systems)

The objective of a second round of simulations was to provide a fair performance comparison between the alternative fluids. In contrast to the "drop-in" simulations where performance of all candidates was simulated using one heat pump, in these simulations a different heat pump was selected (coded) for each refrigerant to simultaneously satisfy the following two constraints at the cooling 35°C (95°F) condition:

$$(1) \quad \frac{Q_e}{UA_e + UA_c} = const = value\ for\ R22$$

$$(2) \quad T_{e,ave} = const = value\ for\ R22$$

The first constraint assures the same loading of the heat transfer area for each refrigerant. This constraint alone can be satisfied by an infinite number of possible distributions of the heat transfer area between the evaporator and condenser. The second constraint removes this degree of freedom by requiring a specific size of evaporator (represented by UA_e) so a predefined value of the average temperature in the evaporator, $T_{e,ave}$, is achieved.

Simulations were performed first for R22 at all four operating conditions using the CYCLE-11.UA model. The same thermodynamic cycle assumptions were used as for "drop-in" simulations with a difference in representation of the compressor. The pressure drop and heat loses where lumped together in a lower value of polytropic efficiency η_p=0.75, while the efficiency of electric motor was left unchanged at η_e=0.9.

Once UA_e+UA_c, Q_e, and $T_{e,ave}$ were obtained for R22 from simulation at the 35°C (95°F) condition, a modified system for each candidate fluid was devised that satisfied the outlined criteria. This could have been done using CYCLE-11.UA through iterative selection of UA_e and UA_c at a given, unchanged size of the compressor. In practice, UA_e and UA_c were selected using another model from the CYCLE-11 family, CYCLE-11.UADT (explained Appendix B), which explicitly provided the information required. Once UA values were established at the 35°C (95°F) condition for each candidate refrigerant (corresponds to selection of modified systems), simulations at the remaining three test conditions were performed using CYCLE-11.UA.

The simulation results for modified systems are presented in Figures 11 through 15. The results for volumetric capacity shown in Figure 11 are representative of the change in compressor displacement required to attain the capacity of R22. Compared to the results obtained for "drop-in" simulations, capacity differences are larger for modified systems (Figure 11) than for the "drop-in" evaluation (Figure 5). For example, high capacity refrigerant #1, which exceeded R22 in the "drop-in" evaluation on average by 40%, exceeded R22 in a modified system by over 50%. Also, a larger difference - in the opposite direction - is observed for low capacity R134a (#5). This is understood because the first constraint requires more heat transfer area for a higher capacity fluid, and a larger heat transfer area causes a lower temperature lift. This leads to an even higher capacity for

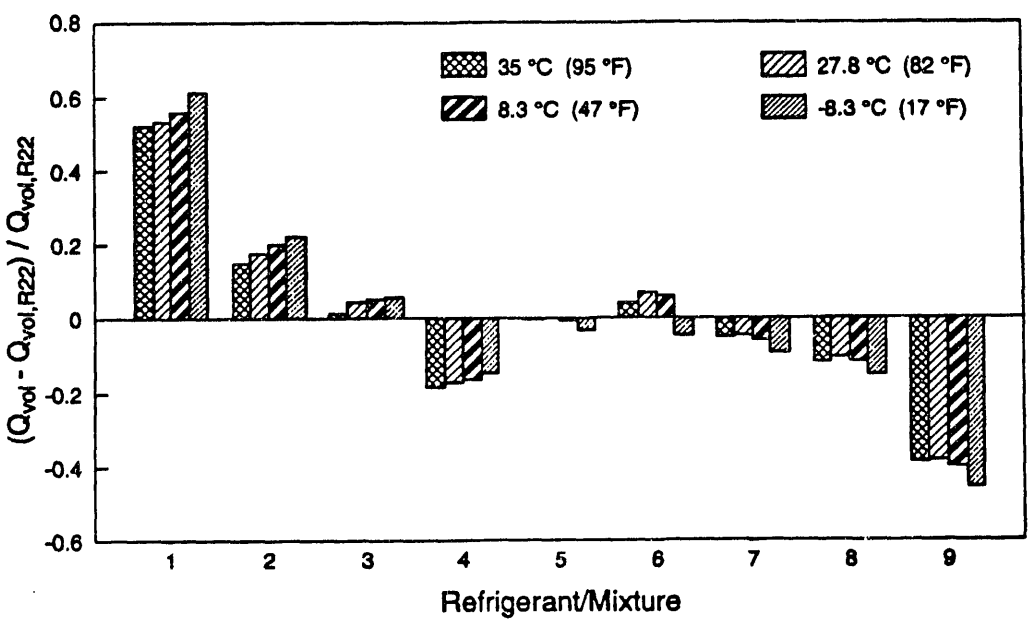

Figure 11. Volumetric capacity for modified systems

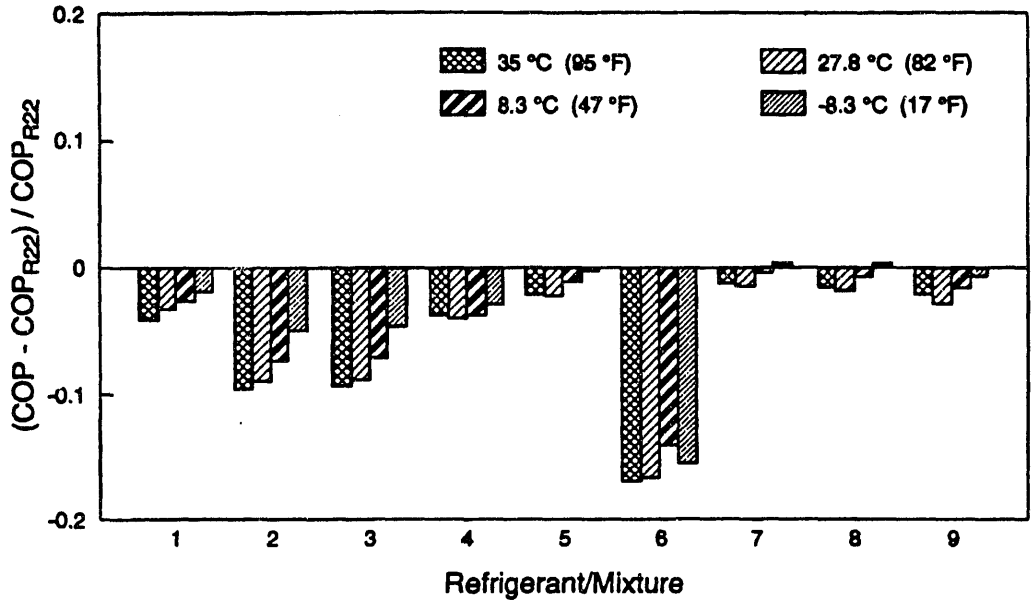

Figure 12. Coefficient of Performance for modified systems

a high capacity refrigerant, such as the R32/R125 mixture (#1). The opposite is true for lower capacity refrigerants.

The constant-heat-exchangers-loading criterium also affects the COP. Higher capacity refrigerants attained improved COPs, while lower capacity refrigerants attained lower COPs than found in the "drop-in" simulations. Consequently, the COPs obtained are less different between fluids, as is shown in Figure 12.

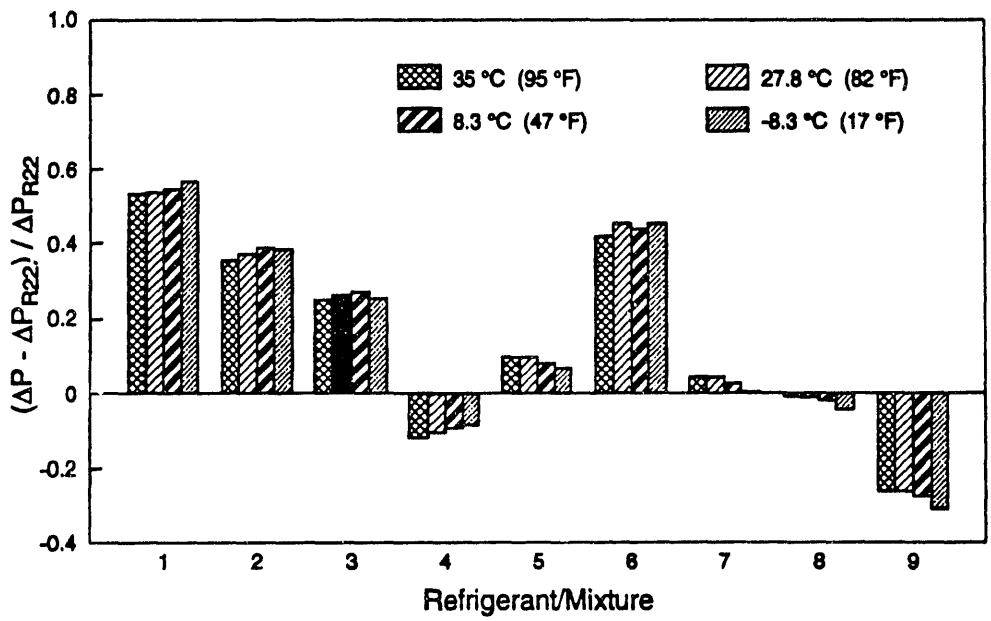

Figure 13. Pressure increase across compressor for modified systems

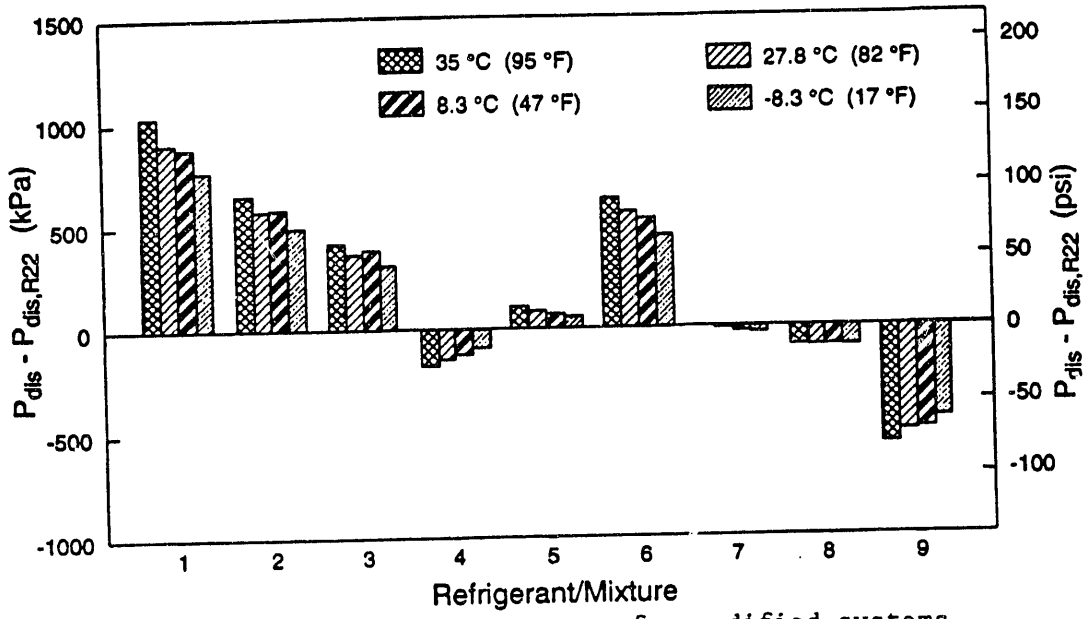

Figure 14. Compressor discharge pressure for modified systems

The disappointing aspect of Figure 12 is that none of the replacement candidates exceeded the COP of R22 at any of the four test conditions. This is even true for the low-pressure/low-capacity R134a for which we expected a better COP - in light of the fundamental COP/capacity trade-off discussed at the beginning of this report. There are two reasons for this somewhat surprising result. Firstly, the imposition of a constant 34.5 kPa (5 psi) pressure drop for the evaporator and the condenser for all refrigerants causes low-pressure R134a to experience a larger temperature change than R22, since low-pressure fluids have a larger

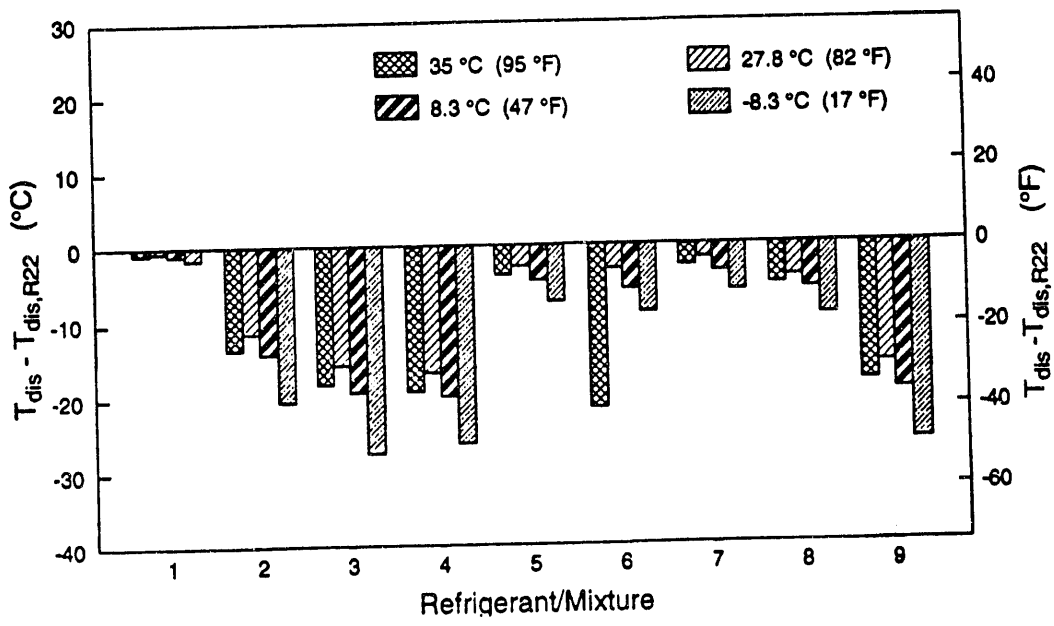

Figure 15. Compressor discharge temperature for modified system

dT/dP gradient at saturation. Secondly, much higher heat capacity of R134a makes it a worse performer in a basic reversed Rankine cycle. For simulations with zero pressure drop in the heat exchangers, the COP of R134a would match that of R22.

Refrigerant pressure increase across the compressor, compressor discharge pressure, and compressor discharge temperature - displayed in Figures 13, 14 and 15 - are self-explanatory. It is interesting to note from Figure 15 that the refrigerant discharge temperature for each of the candidate fluids was below the temperature for R22.

Table 3 contains a summary of the simulation results at a constant heat exchanger loading. The numbers in the table are the averages of the results for four operating conditions shown in Figures 11 through 15.

Impact of a Liquid-Line/Suction-Line Heat Exchange

Considering the lower discharge temperature of the alternative refrigerants, additional simulation runs were conducted using the modified systems (as previously defined) equipped with a liquid-line/suction-line heat exchanger (llsl-hx). This heat exchanger facilitates the heat transfer between a high-temperature liquid refrigerant leaving the condenser and a low-temperature suction vapor leaving the evaporator. The low discharge temperatures of the alternative refrigerants are indicative of their large heat capacities, which suggests performance benefit potential from the application of the llsl-hx [2].

Simulation results were performed for one cooling and one heating test condition, 35°C (95°F) and -8.3°C (17°F), respectively. It was assumed in these simulations that the llsl-hx would increase the suction vapor temperature to 23.9°C (75°F)

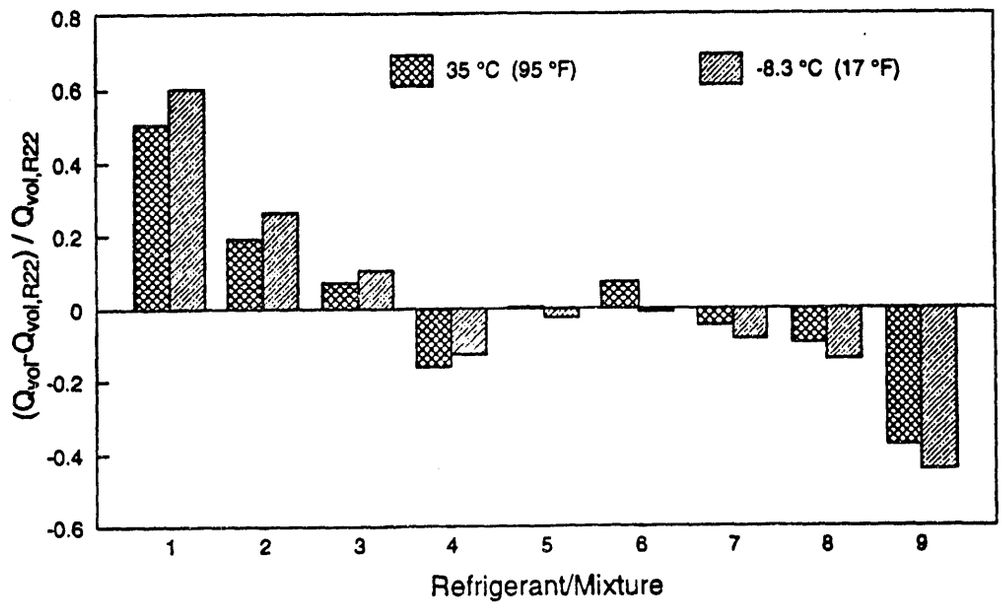

Figure 16. Volumetric capacity for modified systems with llsl-hx

Table 3. Summary[1] of Simulation Results for R22 Alternatives at Constant Heat Exchanger Loading

	Refrigerant	Weight composition %	$\frac{COP}{COR_{R22}}$	$\frac{Q_{vol}}{Q_{vol,R22}}$	$\frac{\Delta P}{\Delta P_{R22}}$	$P_{dis} - P_{dis,R22}$		$T_{dis} - T_{dis,R22}$	
						kPa	psi	°C	°F
1	R32/125	60/40	0.97	1.55	1.55	845	130	-1	-2
2	R32/125/134a/290	20/55/20/5	0.93	1.18	1.37	572	83	-15	-27
3	R32/125/134a	10/70/20	0.93	1.04	1.25	364	53	-20	-36
4	R290	100	0.94	0.83	0.90	-134	-19	-21	-37
5	R32/125/134a	30/10/60	0.98	0.98	1.09	77	11	-5	-8
6	R32/227ea	35/65	0.84	1.03	1.44	568	82	-6	-10
7	R32/134a	30/70	0.99	0.93	1.03	-15	-2	-4	-6
8	R32/134a	25/75	0.99	0.88	0.98	-92	-13	-6	-10
9	R134a	100	0.98	0.59	0.72	-498	-72	-20	-36

[1] the averages of the results for the four operating conditions presented in Figures 11 through 15

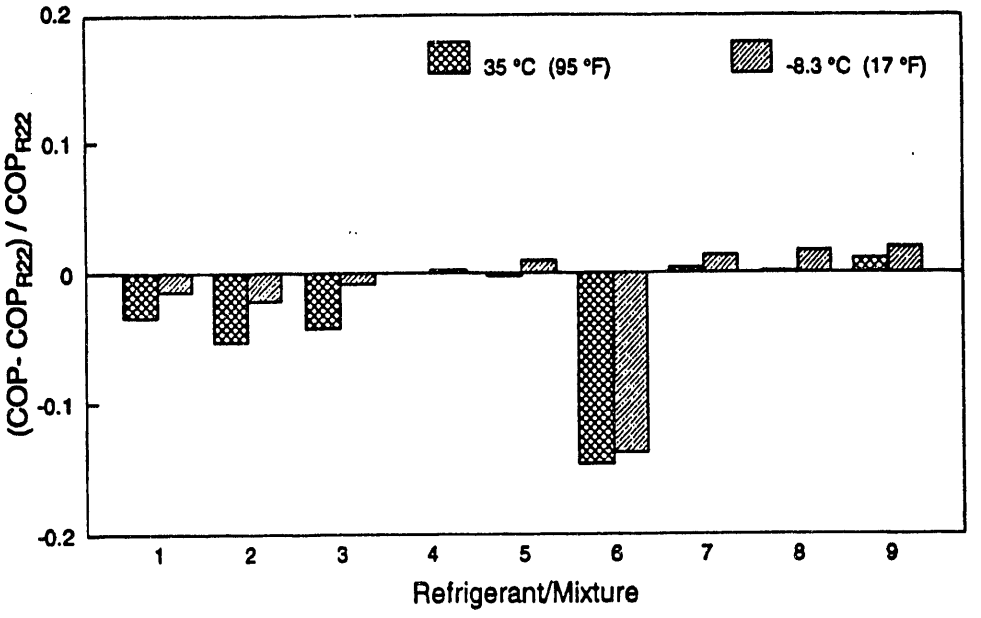

Figure 17. Coefficient of Performance for modified systems with llsl-hx

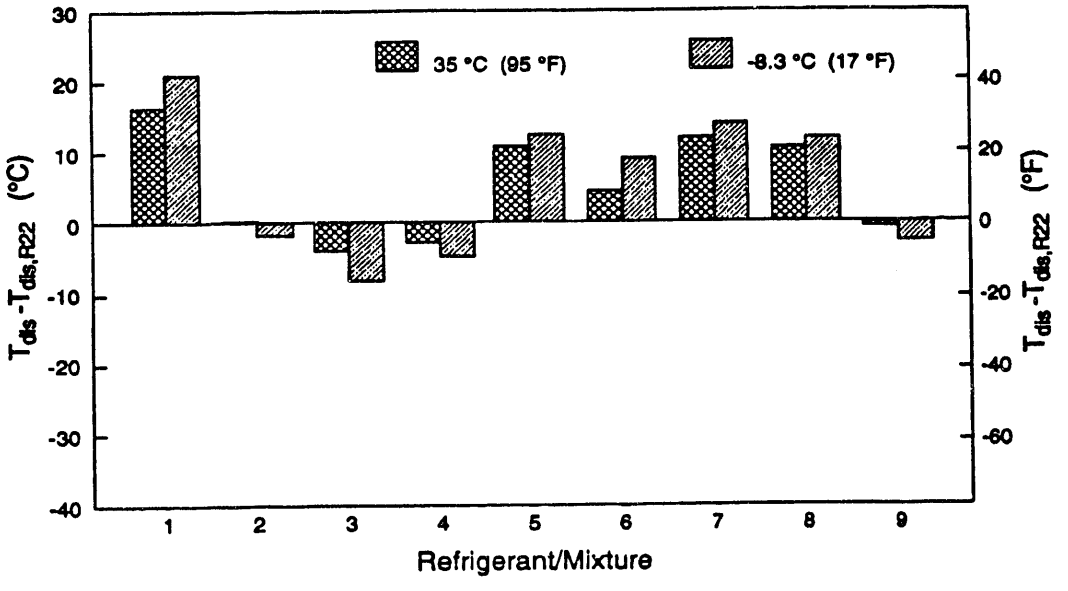

Figure 18. Compressor discharge temperature for modified systems with llsl-hx

at cooling and to 7.2°C (45°F) at heating. This required a llsl-hx of approximately 45% effectiveness.

Figures 16, 17, and 18 display simulation results for the volumetric capacity, COP, and compressor discharge temperature. Except to the R32/227ea mixture (#6), the COPs of the examined refrigerants are within 5 percent of R22, some exceeding

the COP of R22 (R22 did not benefit from the llsl-hx application). It is interesting to note that the discharge temperatures (Figure 18) are not excessive. This suggests that utilization of the llsl-hx may be warranted for some of the alternative refrigerants.

EVALUATION OF R502 ALTERNATIVES

A popular application for R502 [R22/115 (48.8/51.2)] is in commercial refrigeration. Three refrigerant mixtures, specified in Table 4, were evaluated as alternatives for R502 at conditions shown in Table 1.

Evaluation of R502 alternatives consisted of the same three types of simulations as performed for R22 alternatives:

(1) "drop-in" simulations in a R502 unchanged system,
(2) simulations in modified systems to assure the same loading of the heat exchanger area, and
(3) simulations in modified systems at the same heat exchanger loading and with a liquid-line/suction-line heat exchanger.

All simulations were performed with a pure cross-flow evaporator and condenser imposing 34.5 kPa (5 psi) pressure drop for each fluid. Zero degree of subcooling and superheat at the condenser outlet and evaporator inlet, respectively, were assumed. Simulations utilizing the llsl-hx were performed with 70°C (158°F) vapor entering the compressor.

Figure 19 presents simulation results for the three simulation series for a relative volumetric capacity, $(Q_{vol}-Q_{vol,R502})/Q_{vol,R502}$, a relative Coefficient of Performance, $(COP-COP_{R502})/COP_{502}$, and relative difference between the compressor

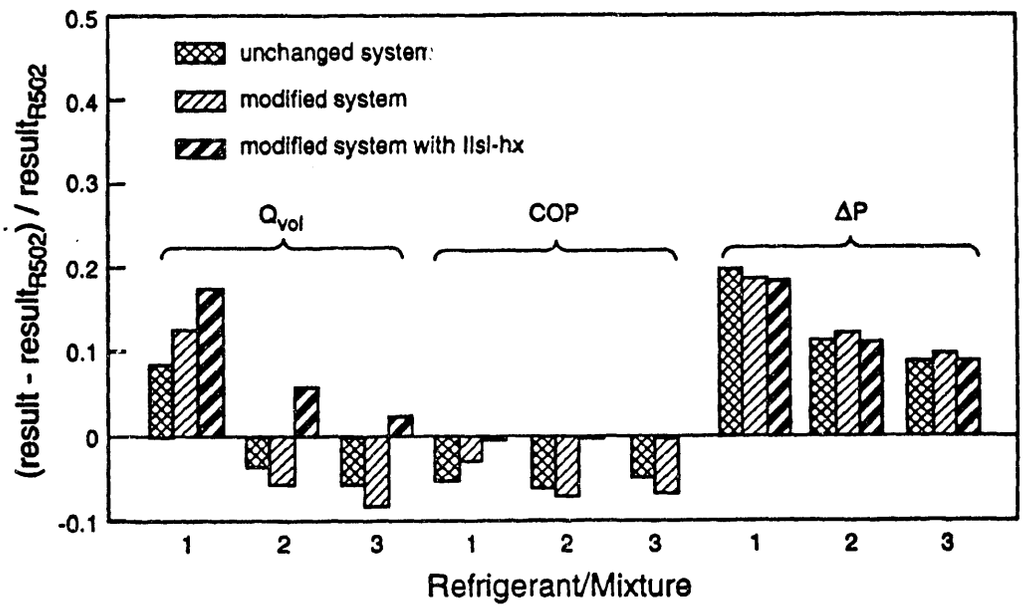

Figure 19. Relative performance of R502 alternative mixtures

discharge and suction pressures, $(\Delta P-\Delta P_{R502})/\Delta P_{R502}$, where ΔP is the difference between the compressor discharge and suction pressures for a given fluid. The figure shows that the volumetric capacity of mixture (1) exceeds that of R502, while capacity of mixture (2) and (3) is lower if llsl-hx is not employed. Each of the candidate mixtures have a lower COP by a few percents, and have a larger than R502 pressure increase across the compressor by approximately 18% for mixture (1), 12% for mixture (2), and 9% for mixture (3). Also, compressor discharge pressures are higher for the studied alternatives, but compressor discharge temperatures are lower than that for R502. A low discharge temperature of the alternative fluids is very important since R502 has been used in applications in which R22 would have too high temperature in the compressor. The summary of the simulation results at a constant heat exchanger loading is presented in Table 5.

CONCLUDING REMARKS

The presented simulation results describe performance of candidate fluids based on the thermodynamic properties. Two basic evaluation approaches were used. The first approach used was the "drop-in" evaluation. The simulation results obtained from this evaluation predict the performance of the candidate refrigerants in a system designed for the original refrigerant - with a possible modification of the expansion device (the same subcooling (zero) at the condenser outlet was assumed at simulations).

The second approach, the constant-heat-exchanger-loading evaluation, provides more objective information on the performance potential of the fluids screened. The simulation results obtained with this approach correspond to a test in a system modified specifically for each refrigerant to obtain the same heat flux through the evaporator and condenser at the design rating point. This simulation constraint assures that the condenser and evaporator pressures are not affected by the different volumetric capacities of the refrigerants studied. In general, higher pressure fluids demonstrated a higher volumetric capacity and a lower COP. A strong exception was the high-glide R32/227ea mixture, of which the performance was penalized by use of pure cross-flow heat exchangers.

Supplementary simulations were performed for the constant-heat-exchanger-loading constraint applying the liquid-line/suction-line heat exchanger. This simulations show that using the llsl-hx may be warranted for some of the candidate fluids.

Several assumption were used in these simulations, and it is important to have them in mind when analyzing the results. The simulations did not involve transport properties and carry the implicit assumption of the same overall heat transfer coefficient. Algorithms for transport properties for most of the fluids were not available in a form suitable for simulation at the time of the study. Only at the time of finishing up this report, we could compile the values for liquid thermal conductivity and viscosity (the most influential transport properties), and these are given in Appendix C. These values, in conjunction with Figures 3 and 4, may be used to estimate if a given candidate fluid offers more promise than is indicated by the simulation results.

Table 4. R502 Alternative Fluids Considered in the Study

	Refrigerant	Weight comp. %	$T_{dew\ point}$ (1)		T_{glide} (2)		T_{cr} (3)		Molecular weight kg/kmol
			°C	°F	°C	°F	°C	°F	
0	R502 [R22/115]	48.8/51.2	-45.4	-49.7	0	0	82.2	179.9	111.63
1	R32/125/143a	10/45/45	-48.8	-55.8	0.5	0.9	82.8	181.0	90.70
2	R125/143a	45/55	-48.1	-54.6	0	0	78.7	173.7	100.24
3	R125/143a/134a	44/52/4	-46.7	-52.0	0.9	1.6	79.9	175.8	97.61

(1) at atmospheric pressure

(2) $T_{glide} = T_{dew\ point} - T_{bubble\ point}$

(3) REFPROP estimates

Table 5. Summary of Simulation Results for R502 Alternatives at Constant Heat Exchanger Loading

	Refrigerant	Weight comp %	$\frac{COP}{COP_{R502}}$	$\frac{Q_{vol}}{Q_{vol,R502}}$	$\frac{\Delta P}{\Delta P_{R502}}$	$P_{dis} - P_{dis,R502}$		$T_{dis} - T_{dis,R502}$	
						kPa	psi	°C	°F
1	R32/125/143a	10/45/45	0.97	1.13	1.18	323	47	-4	-7
2	R125/143a	45/55	0.93	0.94	1.12	206	30	-12	-22
3	R125/143a/134a	44/52/4	0.93	0.92	1.09	164	24	-11	-20

Transport properties may also be used to explain the difference between the simulated COP of propane (lower than that for R22) and the experimental COP reported by Treadwell [10] (higher than the COP for R22). Propane's thermal conductivity of liquid and vapor is somewhat better than that of R22; however, it can be estimated that a significantly better, almost 50 lower, liquid viscosity of propane is responsible for approximately 4.5 percent of difference in the coefficient of performance.

Among other assumptions imposed during simulations, it is important to remember that the evaporator and condenser were represented as pure cross-flow heat exchangers. This implies that the simulations were conservative for zeotropic mixtures if application of counter-flow evaporator and condenser is anticipated; on the other hand, the simulation results are too optimistic for a system using flooded shell-and-tube heat exchangers.

COMPLIANCE WITH AGREEMENT

This report complies fully with the contract agreement.

PRINCIPAL INVESTIGATORS EFFORT

This study was carried out by the principal investigators with 80 percent of the effort contributed by P.A. Domanski and 20 percent by D.A. Didion.

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APPENDIX A

NOMENCLATURE

COP - coefficient of performance
C_p - heat capacity at constant pressure
P - pressure
ΔP = $P_{dis} - P_{suc}$
Q - capacity
Q_{vol} = Q/v_1 , volumetric capacity
T - temperature
v_1 - specific volume at compressor inlet (see Figure B1)

Subscripts:

ave - average
c - condenser
cr - critical (for pure fluids) or pseudocritical (for mixtures)
dis - at compressor discharge
e - evaporator
suc - at compressor suction

APPENDIX B

DESCRIPTION OF CYCLE-11 SIMULATION MODELS

The original CYCLE-11 model (later referred to as CYCLE-11.DT) is described in [5]. It identifies from 5 up to 11 key locations in a refrigeration system; the eleven-point cycle is applicable if a four-point representation of the compressor and a liquid-line/suction-line heat exchanger are used, as shown in Figures A1 and A2. In the four-point simulation of the compressor, heat transfer and pressure drop at the suction and discharge valves (locations 1-2 and 3-4) can be accounted for by assigning heat loss and pressure drop parameters, as opposed to a "black box" approach where these losses can be lumped together within an overall compressor efficiency.

CYCLE-11 models perform simulations at specified inlet and outlet temperatures of the heat transfer fluids (HTFs) at the evaporator and condenser. In CYCLE-11.DT, the evaporator and condenser are represented by temperature differences between refrigerant and HTF, ΔT_e and ΔT_c, which are input data. Thus, ΔT_e and ΔT_c have to be known, and this is not always the case if simulations are to be performed at different operating conditions (ΔT_e and ΔT_c change since capacity changes with operating conditions).

CYCLE-11.UA relieves us from the need of knowing ΔT_e and ΔT_c at all conditions. The evaporator and condenser are represented by the product of their overall heat transfer coefficient and area, UA_e and UA_c, which are the input instead of ΔT_e and ΔT_c.

Regardless of different input requirements, CYCLE-11.UA is amenable to the same convergence logic as CYCLE-11.DT. It is possible because, if we assume that the overall heat transfer coefficient does not change with operating conditions, then UA_e and UA_c stay constant, and since

$$Q = UA \cdot \Delta T \quad or \quad \Delta T = \frac{Q}{UA},$$

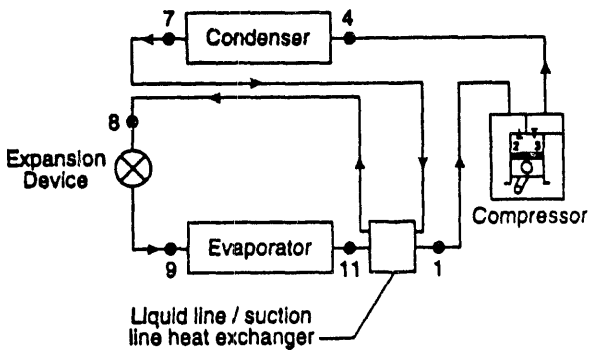

Figure B1. Schematic of a heat pump with a 11sl-hx

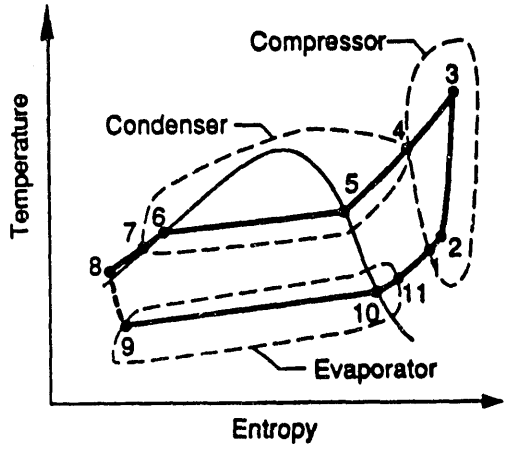

Figure B2. Temperature-entropy diagram for a heat pump with 11sl-hx working with zeotropic blend

and capacity (Q) can be calculated from the refrigerant mass flow rate and enthalpy change in the heat exchanger, ΔT can be calculated and the program will converge in the same manner as in CYCLE-11.DT.

Still another version, CYCLE-11.UADT, was used in this study to explicitly obtain the system performance at the 35°C (95°F) condition satisfying the following constraints:

$$(1) \quad \frac{Q_e}{UA_e + UA_c} = const = value\ for\ R22$$

$$(2) \quad T_{e,ave} = const = value\ for\ R22$$

The second constraint is synonymous with specifying the same ΔT_e since each fluid worked with the same temperature profile of the HTFs in the evaporator.

Considering that $Q = UA \cdot \Delta T$, we may rearrange the first constraint as follows:

$$\frac{UA_e + UA_c}{Q_e} = \frac{1}{\Delta T_e} + \frac{Q_c}{Q_e} \cdot \frac{1}{\Delta T_c}$$

where Q_c/Q_e equals the ratio of refrigerant enthalpy change in the condenser and evaporator. The Q_c/Q_e ratio varies between fluids because of their different outlines of the two-phase dome, but can be calculated during simulation. Knowing Q_c/Q_e, ΔT_e, and the left side of the equation, the first constraint uniquely defines ΔT_c, which with ΔT_e as input allows converging to the solution in a similar way as the DT version of CYCLE-11.

APPENDIX C

SELECTED PROPERTIES OF ALTERNATIVE FLUIDS

The molar heat capacity, thermal conductivity and absolute viscosity are given in Table C-1 and Table C-2 for R22, R502, and their alternative fluids. The values were generated using a not-realized-yet version of REFPROP and should be used with caution.

Table C-1. Selected Properties of R22 Alternative Fluids
(evaluated at saturation at 8°C (46.4°F) temperature)

	Refrigerant	Weight comp.	Vapor heat capacity	Liquid thermal conductivity		Vapor thermal conductivity		Liquid viscosity		Vapor viscosity	
	—	%	J/(mol·K)	mW/(m·K)	Btu/(h·ft·°F)	mW/(m·K)	Btu/(h·ft·°F)	micro-poise	lb/(ft·h)	micro-poise	lb/(ft·h)
0	R22	100	61.1	97.9	0.0566	9.87	0.00571	2046	0.495	121.6	0.0294
1	R32/125	60/40	63.5	112.4	0.0650	11.54	0.00672	1913	0.463	122.3	0.0296
2	R32/125/134a/290	20/55/20/5	82.0	83.3	0.0482	11.67	0.00710	1982	0.480	123.9	0.0230
3	R32/125/134a	10/70/20	90.5	76.9	0.0444	11.50	0.00665	2082	0.504	123.7	0.0229
4	R290	100	78.5	99.97	0.0577	16.05	0.00928	1165	0.282	79.9	0.0193
5	R32/125/134a	30/10/60	70.5	106.5	0.0616	11.20	0.00648	2358	0.570	116.7	0.0283
6	R32/227ea	35/65	81.3	not available							
7	R32/134a	30/70	69.6	109.5	0.0633	11.19	0.00647	2427.2	0.587	115.5	0.0279
8	R32/134a	25/75	71.9	106.5	0.0616	11.17	0.00645	2470.7	0.598	115.2	0.0288
9	R134a	100	87.2	88.5	0.0512	11.37	0.00657	2593.7	0.627	113.2	0.0274

Table C-2. Selected Properties of R502 Alternatives Fluids
(evaluated at saturation at -25°C (-13°F) temperature)

	Refrigerant	Weight comp.	Vapor heat capacity	Liquid thermal conductivity		Vapor thermal conductivity		Liquid viscosity		Vapor viscosity	
	—	%	J/(mol·K)	mW/(m·K)	Btu/(h·ft·°F)	mW/(m·K)	Btu/(h·ft·°F)	micro-poise	lb/(ft·h)	micro-poise	lb/(ft·h)
0	R502 [R22/115]	48.8/51.2	71.3	83.3	0.0481	7.9	0.00457	2911	0.704	106.4	0.0257
1	R32/125/143a	10/45/45	73.8	97.7	0.0565	9.8	0.00566	2584	0.625	103.0	0.0249
2	R125/143a	45/55	81.9	89.3	0.0516	9.9	0.00572	2481	0.600	102.7	0.0248
3	R125/143a/134a	44/52/4	79.3	89.9	0.0520	9.9	0.00572	2541	0.614	102.6	0.0248

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