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COMPRESSOR CALORIMETER PERFORMANCE OF REFRIGERANT BLENDS—COMPARATIVE METHODS AND RESULTS FOR A REFRIGERATOR/ FREEZER APPLICATION

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ABSTRACT

A protocol was developed to define calorimeter operating pressures for nonazeotropic refrigerant mixtures (NARMs) that corresponded with the saturated evaporator and condenser temperatures commonly used for pure refrigerants. Compressor calorimeter results were obtained using this equivalent-mean-temperature (EMT) approach and a generally applied Association of Home Appliance Manufacturers (AHAM) procedure at conditions characteristic of a domestic refrigerator-freezer application. Tests with R-12 and two NARMs indicate that compressor volumetric and isentropic efficiencies are nearly the same for refrigerants with similar capacities and pressure ratios.

The liquid-line temperature conditions specified in the AHAM calorimeter rating procedure for refrigerator-freezer compressors were found to preferentially derate NARM performance relative to R-12. Conversion of calorimeter data taken with a fixed liquid-line temperature to a uniform minimal level of condenser subcooling is recommended as a fairer procedure when NARMs are involved.

Compressor energy efficiency ratio (EER) and capacity data measured as a result of the EMT approach were compared to system performance calculated using an equivalent-heat-exchanger-loading (EHXL) protocol based on a Lorenz-Meutzner (L-M) refrigerator-freezer modeling program. The EHXL protocol was used to transform the calorimeter results into a more relevant representation of potential L-M cycle performance.

The EMT method used to set up the calorimeter tests and the AHAM liquid-line conditions combined to significantly understate the cycle potential of NARMs relative to that predicted at the more appropriate EHXL conditions. Compressor conditions representative of larger heat exchanger sizes were also found to give a smaller L-M cycle advantage relative to R-12.

BACKGROUND

Much of the effort being expended to reverse stratospheric ozone depletion and diminish atmospheric global warming is associated with finding substitute refrigeration fluids that are more environmentally acceptable with equal or have better thermodynamic performance than chlorofluorocarbons (CFCs) or hydrochlorofluorocarbons (HCFCs). Nonazeotropic (zeotropic) blends of refrigerants are seriously being considered as replacements for environmentally damaging pure refrigerants (Didion and Bivens 1990). Out of necessity, the experimental work on alternatives and advanced refrigeration systems requires the use of compressors that were originally designed and optimized for regulated or soon-to-be-regulated refrigerants. Assessments of how well these compressors perform with refrigerant mixtures are necessary to evaluate relative refrigerant performance in system simulations and in hardware tests.

Similarly, a problem exists in comparing the performance of nonazeotropic refrigerant mixtures (NARMs) to pure fluids in a refrigeration cycle. A NARM evaporates and condenses over a temperature range, whereas a pure compound undergoes an isothermal phase change at fixed pressures. This "glide" in temperature between the bubble point and dew point for a NARM makes it problematical to compare mixture cycle performance in terms of "saturated refrigerant" heat exchanger temperatures. Set-up conditions for compressor calorimeter tests call for establishing condensing and evaporating pressures, which, for pure refrigerants, correlate directly with unique heat exchanger operating temperatures.

Existing Procedures and Comparison Methods

Calorimeter tests with specific blends are being conducted to quantify compressor and, in some instances, fluid performance. This leads to questions of how to test mixtures with a compressor calorimeter in a manner that is comparable to that for pure refrigerants and whether the resultant calorimeter performance is indicative of relative system performance. The relevant current procedures (ASHRAE 1978; ARI 1990) were written to provide a uniform basis for rating and comparing the capacity and

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efficiency of compressors using pure refrigerants. These procedures assume isothermal evaporation and condensation. It is not readily apparent how these standards can be applied to provide a fair ranking of compressor performance with refrigerant blends.

One approach has recently been proposed by Sundaresan (1992) for near-azeotropes and is based on testing at pressures that maintain constant midpoint refrigerant temperatures in the heat exchangers. The midpoint temperatures are defined as arithmetic averages, with the evaporator midpoint based on the evaporator inlet two-phase temperature determined for a specified condenser subcooling and on the evaporator exit dew-point temperature. The condenser midpoint is the average from bubble point to dew-point temperatures. This approach has been adopted by ARI (1992) for its R-22 alternatives program where compressor calorimeter testing is under way with 10 alternative refrigerants that include a range of pure refrigerants, azeotropes, near-azeotropes, and zeotropes.

Approaches to defining "equivalent" temperatures for a gliding temperature heat exchange process based on *entropic* averages have been proposed by Alefeld (1987), Herold (1991), and Bansal et al. (1992). The present study uses an equivalent mean temperature based on *enthalpic* averages (which is intermediate in complexity between the more approximate arithmetic average used by ARI and the various entropic averages).

McLinden and Radermacher (1987) have shown that the relative performance of NARMs in comparison to pure refrigerants is totally dependent on the temperatures (which correspond directly to pressures set on a compressor calorimeter) chosen to equate the NARM and pure refrigerant evaporator and condenser operating temperatures. For this NARM calorimetry work, a methodology was developed to calculate the NARM pressure in the evaporator or condenser that corresponds to an averaged temperature across the NARM evaporating or condensing region. This technique is referred to as the equivalent-mean-temperature (EMT) approach and will be described in more detail in a later section.

The EMT method used here expands upon one of the simplified approaches (equal midpoint temperatures) noted by McLinden and Radermacher in their review of possible comparison methods. For their analysis, McLinden and Radermacher used a more rigorous method of maintaining equivalent total heat exchanger loading. The CYCLE7 program for heat pumps and CYCLE11 model for single-evaporator refrigerators or heat pumps (Domanski 1990) from the National Institute of Standards and Technology (NIST) were later developed as approximations to fixed total heat exchanger loading where the effective mean temperature differences (EMTDs) on each heat exchanger were held fixed. Rice and Sand (1990) modified CYCLE7 for the L-M refrigerator-freezer cycle (CYCLE-2) and developed a procedure to exactly maintain equivalent total heat exchanger loading (EHXL) by appropriate normalization of the condenser EMTD. Equivalent total heat exchanger loading is defined here for the refrigerator-freezer application as the total UA (conductance-area product) for all evaporator and condenser heat exchangers per unit of refrigeration capacity ($UA_{TOT}/\dot{Q}_E$).

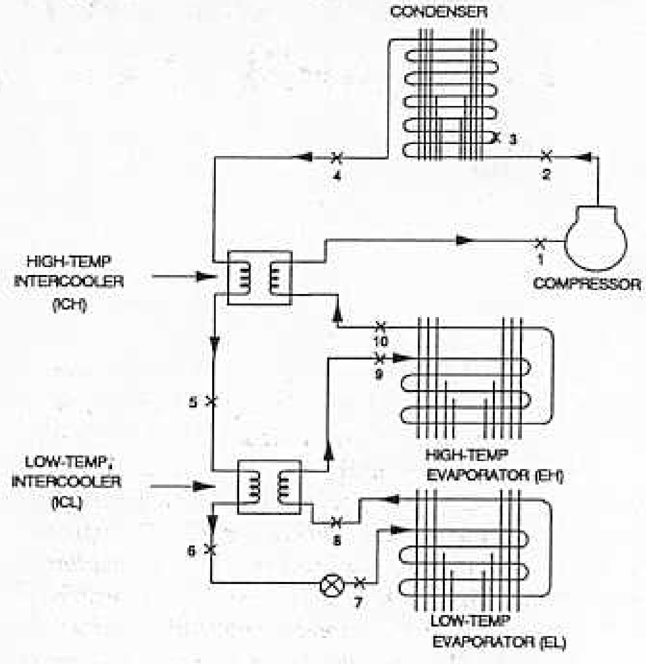

Figure 1a *Schematic diagram of Lorenz-Meutzner refrigerator-freezer circuit.*

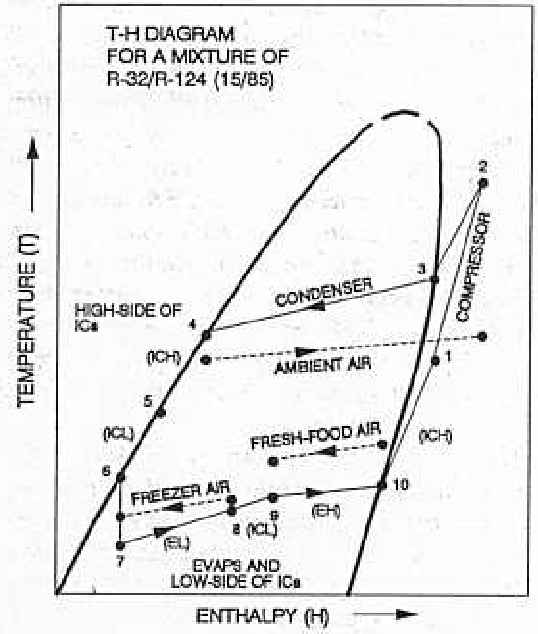

Figure 1b *Refrigerant temperature-enthalpy state-points for the Lorenz-Meutzner, refrigerator-freezer cycle with superimposed condenser, freezer, and fresh-food compartment air temperatures.*

Figure 1 *Lorenz-Meutzner refrigeration cycle.*

Lorenz-Meutzner Refrigeration Cycle

The authors' present experimental and system modeling work with pure and mixed refrigerants was the direct result of research aimed at applying NARMs in a Lorenz-Meutzner refrigeration cycle (Sand et al. 1992, 1993) as an energy-efficient alternative for R-12 in domestic refrigerator-freezers (r/f). A schematic of the L-M r/f circuit (Lorenz and Meutzner 1975) and a corresponding temperature versus enthalpy diagram for a NARM refrigerant operating in this cycle are shown in Figure 1. Temperature-enthalpy diagrams are especially useful in describing cycles using NARMs because temperature profiles for both fluids in the heat exchange steps can be readily illustrated, and the thermodynamic advantages of a refrigerant with a non-isothermal phase change are more apparent (Granyrd et al. 1991). Figure 1b shows how the temperature glide of an evaporating NARM can achieve a closer approach to the two air temperatures needed in the freezer and fresh food compartments of a refrigerator/freezer. This temperature glide matching on both sides of the heat exchanger decreases the irreversibilities in a counterflow heat exchange process, which translates to reduced compressor work for a fixed refrigeration capacity and heat exchanger size.

An L-M r/f was constructed from a commercially produced, 18-ft^3 (0.51-m^3), top-mount unit that incorporated separate freezer and fresh food evaporators (Topping 1982). This prototype unit was operated with its original R-12 compressor. To better assess the resulting R-12 versus NARM system performance, it was necessary to perform comparable calorimetry tests with this compressor using R-12 and the NARM refrigerants tested in the L-M refrigerator/freezer.

PROBLEM DEFINITION AND APPROACH

Problems with Using Compressor Tests as a Basis for Refrigerant Screening

Compressor calorimeter tests are a convenient way to determine (using refrigerant-to-source and -sink energy balances) whether the predicted thermodynamic properties of enthalpy for a pure or mixed refrigerant are reasonably accurate for estimated application conditions. This type of experimental confirmation is useful as a first-level check on whether a candidate refrigerant can perform in a given compressor roughly as expected for a given set of suction and discharge pressures. Such testing does not, however, provide a sufficient basis for making judgments of the relative potential of one blend relative to another in a given application unless the compressors are shown to perform equivalently and the conditions for comparable testing of each blend are carefully determined in advance.

Once a protocol is selected for compressor calorimeter testing using NARMS, the question becomes how to best use the resulting test data:

1. to characterize relative compressor efficiency with blends and
2. to relate calorimeter performance to potential system performance.

Answers to both aspects of this question are needed to determine what basis exists, if any, for using compressor calorimeter tests with NARMs as a means for refrigerant screening. Evaluation of the relative compressor efficiency should indicate whether the compressor performed significantly worse for certain blends irrespective of the theoretical COP advantage of one mixture over another at "equivalent" conditions. Translation of calorimeter EER and capacity into potential system performance involves determining the "equivalent" conditions for various pure and mixed refrigerants in a specific application with the same or different cycle configurations.

Misleading conclusions can result if the calorimeter capacity and EER for different refrigerants are used directly to infer that the refrigerant with the higher tested EER and/or capacity is better. Such an interpretation implies that the compressor performed equivalently for all the considered refrigerants and that the test conditions were representative of those that would be found in comparably designed equipment for the intended application. Therefore, before refrigerants can be ranked for a given application on the basis of calorimeter tests, it is necessary to determine whether the implied equivalences have been met. If either assumption is found to be faulty, methods to correct the rankings for these effects should be considered or appropriate caveats stated.

Relative differences in basic compressor performance can be determined through the use of volumetric and isentropic efficiencies derived from measured capacity, EER, and refrigerant property data. Test conditions representative of comparably designed equipment can be predicted by cycle models that impose constraints of equivalent total heat exchanger loading and equal source and sink inlet and exit temperatures, as advocated by McLinden and Radermacher (1987).

Limiting Case Equivalence of EHXL and EMT Methods The equivalent heat exchanger loading (EHXL) method reduces to the equivalent mean temperature (EMT) approach only in the limiting case of pure refrigerants and then only under conditions of a fixed cycle configuration and equal condenser versus evaporator capacity ratios ($\dot{Q}_C/\dot{Q}_E$ ratios). For pure refrigerant alternatives to R-12, such as R-134a and R-152a, the EHXL approach gives equal evaporator temperatures and condenser temperatures to within 0.7F° (0.4C°) of those for R-12. For other pure refrigerants, the comparable condenser temperature may be somewhat different (e.g., for R-22 the comparable condenser temperature is more than 2F° [1.1C°] lower). A calorimeter test procedure based on equivalent mean temperatures does not represent equivalent heat exchangers except in these limited cases.

Potential Problems of Any Equivalent Mean Temperature Approach with NARMS

EMT approaches only give a rough approximation of the conditions seen by a compressor for NARMs in a Lorenz-Meutzner r/f cycle. Factors that will cause the EMT approach to err in predicting the comparable temperature and pressure levels in advanced designs employing blends include:

1. Widely differing $\dot{Q}_C/\dot{Q}_E$ ratios of various blends.
2. Gliding refrigerant temperatures that better, or more poorly, match the external source and sink glides relative to a pure refrigerant and result in mean refrigerant temperatures closer to, or further from, the source and sink temperatures.
3. Nonlinear temperature versus enthalpy profiles in the two-phase regions that can increase or decrease the mean refrigerant temperature obtained with fixed heat exchanger loading.
4. The effect of different cycles such as the dual-evaporator L-M cycle for mixtures relative to the single-evaporator refrigerator cycle commonly used with R-12.
5. The effect of liquid line subcooling in the low- and high-temperature intercoolers, which raises the low-side operating pressure without changing the mean temperature of a refrigerant blend.
6. The influence of heat exchanger configurations that deviate from pure counterflow and of reduced mixture heat transfer coefficients in widening the differences between the mean temperatures of the refrigerant and the source or sink fluid.

Advantages of Present Approach

The adopted approach for interpreting calorimeter data obtained on an EMT basis provides a means to predict more representative (and more refrigerant-to-refrigerant comparable) system performance potential using available calorimeter data. The method used has the following advantages:

1. basic compressor efficiencies are determined and presented in a more generalized manner,
2. means are provided to predict the relative cycle potential of tested blends for (a) a specific compressor and equivalent heat exchangers and (b) with cycle-specific effects such as dual evaporators and benefits of additional subcooling,
3. the procedure can be applied after-the-fact to compressors previously tested using various protocols, and
4. a basis is provided for determining the system performance loss due to NARM heat exchanger degradations in advanced breadboards using compressor calorimeter data.

Approach Overview

The calorimeter testing procedure is first described in more detail followed by presentation of the calorimeter test results and derived efficiencies. The issue of how well the compressor performed for the refrigerants tested is addressed.

Next the procedure that was used to convert the calorimeter data to a more accurate representation of relative L-M system performance is reviewed. This includes curve-fit representations of the calorimeter-derived efficiency data for each tested refrigerant and the system analysis conditions and assumptions made for the single- and dual-evaporator cycles.

The L-M system results obtained from application of the conversion procedure to the tested blends are compared to single-evaporator r/f performance with R-12. These system-based EER and capacity gains for NARMs relative to R-12 are contrasted with those trends shown from the calorimeter data alone. Operating conditions for the EMT and EHXL methods are compared to help explain the observed performance differences. Also examined is the degree of preferential NARM performance derating with the AHAM 90°F (32.2°C) liquid line temperature relative to more representative system subcooling levels. The findings from the application of the EHXL method to the EMT-based compressor data are summarized and the apparent implications are discussed.

CALORIMETER TESTING PROCEDURE

For r/f applications, compressor capacities and energy efficiency ratios (EERs) are usually measured at nine operating conditions corresponding to 110°, 120°, and 130°F (43.3°, 48.9°, and 54.4°C) condenser temperatures and −20°, −10°, and 0°F (−28.9°, −23.3°C, and −17.8°C) evaporator temperatures. A 90°F ambient temperature for the compressor, superheating of the suction gas to 90°F, and subcooling of the liquid line to 90°F (32.2°C) are additional testing conditions specified in the draft procedure developed by the AHAM Consortium for its round-robin compressor tests of CFC alternatives (Swatkowski 1989). Forced ventilation may or may not be used to cool the hermetic compressor shell. Refrigeration capacity in this procedure is determined by measuring the electrical energy needed to change subcooled liquid refrigerant at 90°F to superheated refrigerant at 90°F (32.2°C). As noted by Sandvordenker (1992), the test procedure requires accurate PVT data (and accurate liquid heat capacity data if the subcooled data are corrected to 90°F [32.2°C]).

Compressor calorimeter tests were conducted with R-12 and the NARMs R-32/R-124 and R-22/R-141b in the 670-Btu/h (196.3-W) R-12 compressor that was original equipment on a conventional r/f that was modified to the L-M design. The calorimeter test apparatus and environmental control chamber have been described elsewhere by Sand et al. (1992). Liquid refrigerant was subcooled to 90°F prior to the expansion valve and suction vapor was superheated to 90°F (32.2°C). Compositions of the two NARMs were chosen to provide roughly the same volumetric capacities, and therefore operating pressures, as R-12 when calculated using the EMT protocol.

EMT Definition Used for L-M Refrigerator/Freezer Application

The equivalent-mean-temperature definition used in this analysis is an enthalpically averaged value. Mathematically, the pressure needed to establish an EMT was calculated by dividing the two-phase evaporation or condensation portion of the refrigeration cycle (from bubble point to dew point) into 10 equal increments of enthalpy change. The Carnahan-Starling-DeSantis (CSD) refrigerant property routines of REFPROP V2.0 (Gallagher et al. 1991) were used to calculate mixed refrigerant temperatures at each of the enthalpy increments. These temperatures were then averaged arithmetically to determine the EMT. An iterative procedure was used to find the pressure at which this EMT matched the "saturation" temperature for a pure refrigerant. The pressures that resulted in EMTs corresponding to the nine-point R-12 saturation temperatures were then used as the suction and discharge pressures at which the NARM was tested on the calorimeter. These condensing and evaporating *pressures* are given in Table 1.

Later refinements to the CSD equation of state coefficients for R-32, R-124, and R-141b and better estimates for the binary interaction coefficients for these blends (Morrison and McLinden 1991) necessitated correction of the original EER and refrigeration capacity results calculated from the calorimeter measurements taken at these pressures. Table 2 lists the condensing and evaporating pressures necessary to give EMTs corresponding to saturated R-12 heat exchanger conditions that were calculated using the more recent CSD refrigerant property data. Larger corrections had to be made to the R-32/R-124 results.

The R-32/R-124 and R-22/R-141b NARMs used for compressor calorimeter testing were prepared by weighing appropriate amounts of the pure components into a gas cylinder cooled in dry ice. A gas manifold was added to the calorimeter that permitted convenient addition and removal of these NARMs from the liquid phase. Results presented in the following section are for NARM concentrations as prepared in the sampling cylinders and charged from the liquid phase.

RESULTS OF EMT-BASED COMPRESSOR CALORIMETER TESTS

Calorimeter EER and Capacity Results

The 670-Btu/h (196.3-W) compressor calorimeter results for EERs and refrigeration capacities of R-12 and the R-32/R-124 and R-22/R-141b NARMs obtained using the EMT protocol and the 90°/90°F (32.2°C) liquid line/suction line conditions of the AHAM procedure are given in Table 3. The EER and capacity values are adjusted to the corrected pressures of Table 2 to reflect the improved CSD refrigerant property data as discussed earlier.

The tabulated results show a small improvement in EER and capacity over R-12 for the R-32/R-124 NARM at the adjusted conditions and a small decrease in performance for the R-22/R-141b mixture. Figure 2 is a plot of the results from Table 3 for the 120°F (48.9°C) condensing temperature, which further illustrates these observations.

Derived Compressor Isentropic and Volumetric Efficiencies

EER and capacity values obtained from compressor calorimeter tests are a combination of refrigerant property effects and two basic compressor efficiencies. The refrigerant property effects determine the ideal EER and capacity under the tested operating conditions. The compressor (overall) isentropic and volumetric efficiencies (defined from compressor shell inlet to outlet) are the power and pumping efficiencies, respectively, that convert the ideal EER and capacity to actual calorimeter performance.

To examine how efficiently a given compressor will perform with a variety of refrigerants, these two efficiencies need to be determined. They are obtained from calorimeter and test condition data using refrigerant property information (along with data on the compressor nominal speed and displacement). The compressor power and pumping efficiencies are derived directly from the calorimeter power and mass flow rate values and depend only on the compressor conditions (i.e., the compressor suction and discharge pressures, the suction gas temperature, and the air temperature and flow rate over the compressor shell).

Isentropic Efficiency Overall compressor isentropic efficiency is defined as

$$\eta_{ISEN} = \dot{m}_R \cdot \Delta h_{ISEN} / \dot{W}$$

where $\dot{m}_R$ is refrigerant mass flow rate given by $\dot{m}_R = \dot{Q}_E / \Delta h_E$, and $\dot{Q}_E$ and Δh_E are evaporator capacity and enthalpy change, respectively; Δh_{ISEN} is the enthalpy change for an isentropic compression from shell inlet conditions to shell outlet pressure, and $\dot{W}$ is compressor motor input power.

Volumetric Efficiency Compressor volumetric efficiency is defined as

$$\eta_{VOL} = \dot{m}_R / (S D \rho_C)$$

where S is the nominal compressor speed in revolutions per minute (rpm), D is the total compressor displacement per revolution, and ρ_C is the refrigerant density at the compressor shell inlet.

The refrigerant mass flow rates $\dot{m}_R$ were available from measurements with a gyroscopic mass flowmeter and were used directly instead of values derived from capacity data. Refrigerant mass flow rates derived from refrigerant property routines are likely to be less accurate than direct

TABLE 1
Evaporator and Condenser Pressures Used to Perform Equivalent-Mean-Temperature Calorimeter Tests[1]

		Equivalent Mean HX Temperatures					
		−20°F (−28.9°C)	−10°F (−23.3°C)	0°F (−17.8°C)	110°F (43.3°C)	120°F (48.9°C)	130°F (54.4°C)
Refrigerant	Mass% Composition	Equivalent Pressures					
		psig (kPa)					
R-12[2]	100%	0.57 (106)	4.50 (133)	9.17 (165)	136.2 (1046)	157.3 (1192)	180.5 (1353)
R-32/R-124	15%/85%	−0.82 (96.2)	2.87 (122)	7.30 (114)	135.2 (1039)	157.4 (1193)	181.9 (1362)
R-22/R-141b	80%/20%	−0.70 (97.0)	3.31 (125)	7.75 (156)	143.5 (1096)	166.9 (1258)	192.9 (1439)

1 Data from Sand et al. (1992).

2 Saturation temperatures and pressures for R-12.

TABLE 2
Evaporator and Condenser Pressures Corrected to Give Equivalent-Mean Temperatures in the Heat Exchangers Newer CSD Results[1]

		Equivalent Mean HX Temperatures					
		−20°F (−28.9°C)	−10°F (−23.3°C)	0°F (−17.8°C)	110°F (43.3°C)	120°F (48.9°C)	130°F (54.4°C)
Refrigerant	Mass% Composition	Equivalent Pressures					
		psig (kPa)					
R-12[2]	100%	0.57 (106)	4.50 (133)	9.17 (165)	136.2 (1046)	157.3 (1192)	180.5 (1353)
R-32/R-124	15%/85%	0.85 (108)	5.02 (137)	10.03 (171)	154.7 (1174)	179.6 (1346)	207.0 (1536)
R-22/R-141b	80%/20%	−0.52 (98.3)	3.36 (125)	8.04 (158)	145.2 (1108)	168.9 (1272)	195.0 (1453)

1 Pressures calculated using the Carnahan Starling DeSantis refrigerant property routines in REFPROP© V2.0 from the National Institute of Standards and Technology. Interaction coefficients were calculated using an algorithm described by Morrison and McLinden (1991) based on dipole moments and molecular volumes.

2 Saturation temperatures and pressures for R-12.

measurements because of uncertainties in mixture property predictions.

Isentropic and volumetric efficiencies were derived from the nine-point calorimeter data for the tested 670-Btu/h (196.3-W) reciprocating r/f compressor. The actual test pressures used for the calorimeter work from Table 1 were used in the efficiency calculations rather than the corrected NARM pressures of Table 2.

Efficiency Results The resultant values of volumetric and isentropic efficiency for R-12 and the R-32/R-124 and R-22/R-141b NARMs are shown in Figures 3 and 4, respectively, plotted versus compressor pressure ratio. In Figure 3, the volumetric efficiency data are seen to group closely together and could be approximated by a common decreasing linear function of pressure ratio. The data for the R-22/R-141b mixture (with the highest tested pressures of the three refrigerants) exhibit a slightly higher efficiency at a given pressure ratio than the other two cases, especially at the higher pressure ratios. *At a typical application pressure ratio of 9, the volumetric efficiencies are about 55% based on an assumed nominal speed of 3450 rpm.* However, at a given EMT condition, it can be determined from Figure 3 and Tables 1 and 2 that the predicted pressure ratios for the NARMs are 12% to 16% higher than for R-12. Under the EMT assumption, the volumetric

TABLE 3 (IP Units)
Comparison of Calorimeter Results Based on EMT Method and AHAM 90°/90°F Conditions

Mean HX Conditions		Calorimeter Performance				
Evaporator Mean Temp.	Condenser Mean Temp.	R-12	R-22/R-141b	R-32/R-124	R-22/R-141b	R-32/R-124
(°F)	(°F)		EER		ΔEER vs R-12	ΔEER vs R-12
-20	110	4.02	3.97	4.11	-1.3%	2.0%
"	120	3.76	3.66	3.80	-2.6%	1.3%
"	130	3.54	3.35	3.51	-5.2%	-0.7%
-10	110	4.74	4.62	4.89	-2.4%	3.2%
"	120	4.41	4.29	4.60	-2.8%	4.2%
"	130	4.15	3.97	4.32	-4.4%	4.2%
0	110	5.47	5.22	5.41	-4.6%	-1.1%
"	120	5.08	4.87	5.16	-4.1%	1.6%
"	130	4.77	4.54	4.92	-4.7%	3.3%
(°F)	(°F)		Capacity (Btu/h)		ΔCap vs R-12	ΔCap vs R-12
-20	110	573.6	535.2	557.8	-6.7%	-2.7%
"	120	520.8	479.6	496.9	-7.9%	-4.6%
"	130	470.0	420.5	429.2	-10.5%	-8.7%
-10	110	795.4	759.7	812.0	-4.5%	2.1%
"	120	742.8	704.2	751.2	-5.2%	1.1%
"	130	692.0	645.2	683.4	-6.8%	-1.2%
0	110	1060.6	1032.0	1119.3	-2.7%	5.5%
"	120	1008.3	976.8	1058.7	-3.1%	5.0%
"	130	957.4	917.8	990.8	-4.1%	3.5%

R-22/R-141b mixture is 80% R-22 by mass
R-32/R-124 mixture is 15% R-32 by mass

Calorimeter Results Are Based On Refrigerant-Side Measurements
For Consistency With Compressor Curve-Fit Representations

Results For Mixtures Were Corrected To New EMT Pressures
Calculated With NIST CSD coefficients from REFPROP V2.0

Table 3 (SI Units).						
Comparison of calorimeter results based on EMT method and AHAM 32.2°/32.2°C conditions						
Mean HX Conditions		Calorimeter Performance				
Evaporator Mean Temp.	Condenser Mean Temp.	R-12	R-22/R-141b	R-32/R-124	R-22/R-141b	R-32/R-124
(°C)	(°C)	COP			ΔEER vs R-12	ΔEER vs R-12
-28.9	43.3	1.18	1.16	1.20	-1.3%	2.0%
"	48.9	1.10	1.07	1.11	-2.6%	1.3%
"	54.4	1.04	0.98	1.03	-5.2%	-0.7%
-23.3	43.3	1.39	1.35	1.43	-2.4%	3.2%
"	48.9	1.29	1.26	1.35	-2.8%	4.2%
"	54.4	1.22	1.16	1.27	-4.4%	4.2%
-17.8	43.3	1.60	1.53	1.59	-4.6%	-1.1%
"	48.9	1.49	1.43	1.51	-4.1%	1.6%
"	54.4	1.40	1.33	1.44	-4.7%	3.3%
(°C)	(°C)	Capacity			ΔCap vs R-12	ΔCap vs R-12
			(W)			
-28.9	43.3	168.1	156.8	163.4	-6.7%	-2.7%
"	48.9	152.6	140.5	145.6	-7.9%	-4.6%
"	54.4	137.7	123.2	125.7	-10.5%	-8.7%
-23.3	43.3	233.0	222.6	237.9	-4.5%	2.1%
"	48.9	217.6	206.3	220.1	-5.2%	1.1%
"	54.4	202.8	189.1	200.2	-6.8%	-1.2%
-17.8	43.3	310.8	302.4	327.9	-2.7%	5.5%
"	48.9	295.4	286.2	310.2	-3.1%	5.0%
"	54.4	280.5	268.9	290.3	-4.1%	3.5%

R-22/R-141b mixture is 80% R-22 by mass
R-32/R-124 mixture is 15% R-32 by mass

Calorimeter Results Are Based On Refrigerant-Side Measurements
For Consistency With Compressor Curve-Fit Representations

Results For Mixtures Were Corrected To New EMT Pressures
Calculated With NIST CSD coefficients from REFPROP V2.0

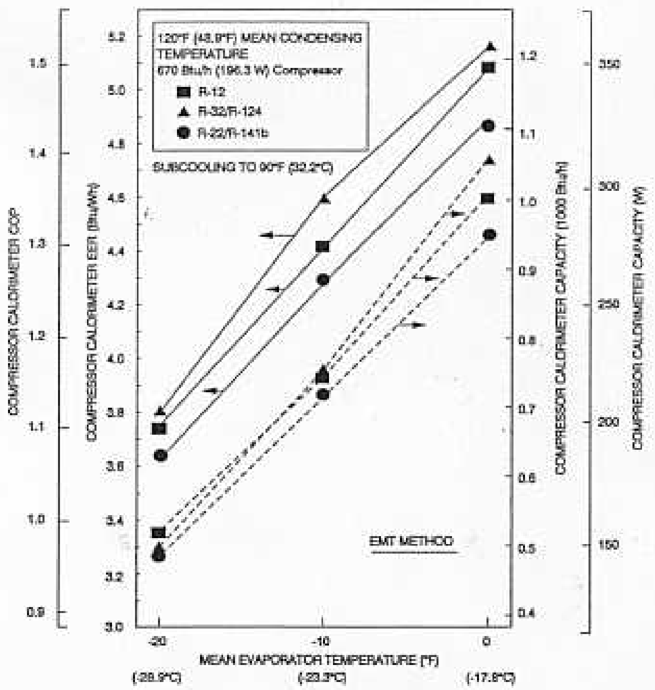

Figure 2 *Corrected experimental EERs and refrigeration capacities for a 670 Btu/h (196.3 W) R-12 compressor tested with R-12 and with R-22/R-141b and R-32/R-124 NARMs at 120°F (48.9°C) condensing EMT and at AHAM 90°/90°F (32.2°C) liquid line/suction line rating conditions.*

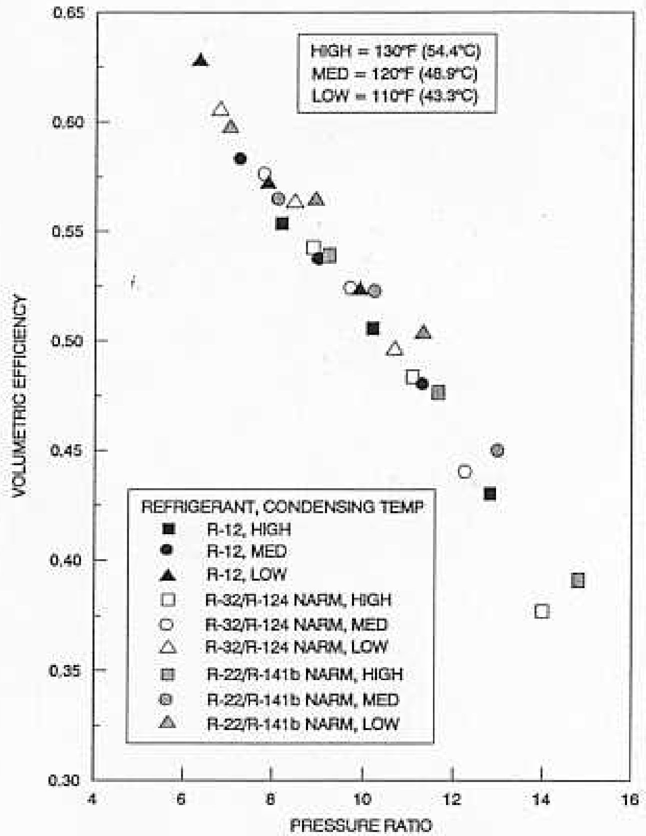

Figure 3 *Volumetric efficiencies versus pressure ratios for a 670 Btu/h (196.3 W) R-12 compressor tested with R-12 and with R-22/R-141b and R-32/R-124 NARMs at the AHAM 90°/90°F (32.2°C) rating conditions.*

efficiencies for the NARMs will be lower than for R-12 by as much as 14%.

The grouping of isentropic efficiencies in Figure 4 show a similar small deviation (slightly wider than for volumetric efficiency) for the range of refrigerants tested. The trend of decreasing isentropic efficiency with increasing pressure ratio is less pronounced and a specific functional relationship is less apparent. As in Figure 3, the R-22/R-141b mixture again shows a slightly higher efficiency at a given pressure ratio than for the R-12 and R-32/R-124 cases, but only for pressure ratios above 9. *At the typical application pressure ratio of 9, the isentropic efficiencies are about 45%.*

In Figure 5, the isentropic efficiency values have been plotted as a function of evaporator exit saturated vapor (dew-point) temperature. In this representation, the data for each tested refrigerant separate into distinct yet similar families for low, medium, and high values of mean condensing temperature. (Note that the higher refrigerant exit temperatures of up to 35°F [1.7°C] are plausible for NARMs in a dual-evaporator r/f with counterflow heat exchanger configuration and an entering air temperature of 38°F [3.3°C] or greater.) Just as for the R-12 case, where evaporator temperatures of −20°, −10°, and 0°F (−28.9°, −23.3°, and −17.8°C) apply, the NARMs show similar trends of increasing efficiency for their corresponding low, medium, and high equivalent exit temperatures. Unlike the volumetric efficiencies, the isentropic efficiencies of the tested refrigerants are about the same at the corresponding EMT conditions.

Representation of Compressor Efficiencies Versus Operating Conditions

An existing compressor map-fitting program (Rice 1991) to obtain curve fits to volumetric and isentropic efficiency from mass flow and power data was modified to work with the CSD equation-of-state routines. The modified map-fitting program was used to obtain, for the tested refrigerants, six-coefficient biquadratic representations of the isentropic efficiency and four-coefficient curve-fits to volumetric efficiency. The equations are of the form

$$\eta_{ISEN} = C_1 T_C^2 + C_2 T_C + C_3 T_E^2 + C_4 T_E + C_5 T_E T_C + C_6$$

and

$$\eta_{VOL} = C_1 (P_R - 1) + C_2 (P_R - 1) P_C + C_3 (P_R - 1) P_C^2 + C_4,$$

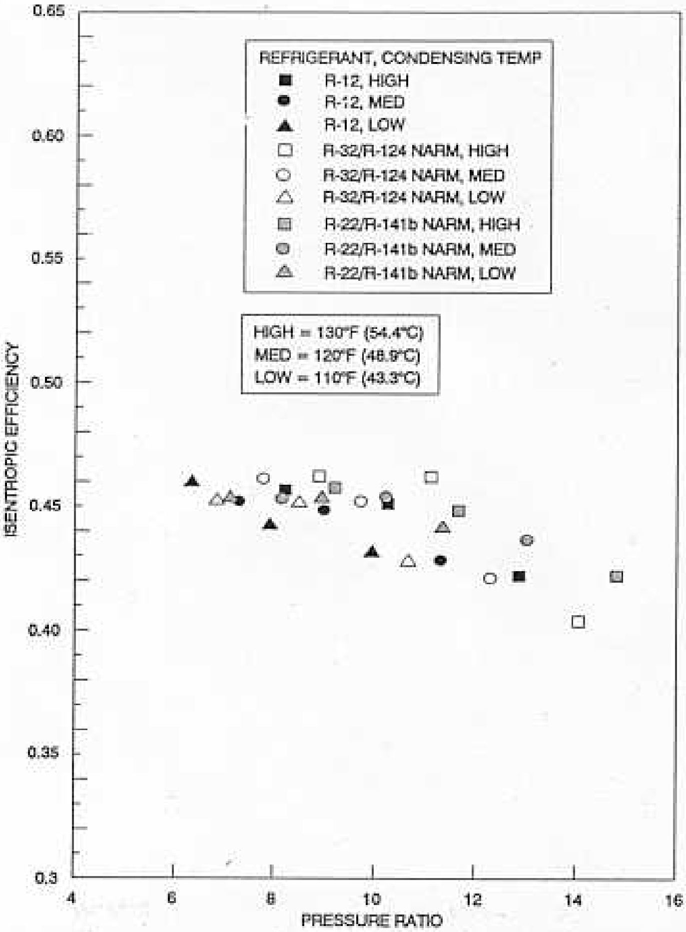

Figure 4 *Isentropic efficiencies versus pressure ratios for a 670 Btu/h (196.3 W) R-12 compressor tested with R-12 and with R-22/R-141b and R-32/R-124 NARMs at the AHAM 90°/90°F (32.2°C) rating conditions.*

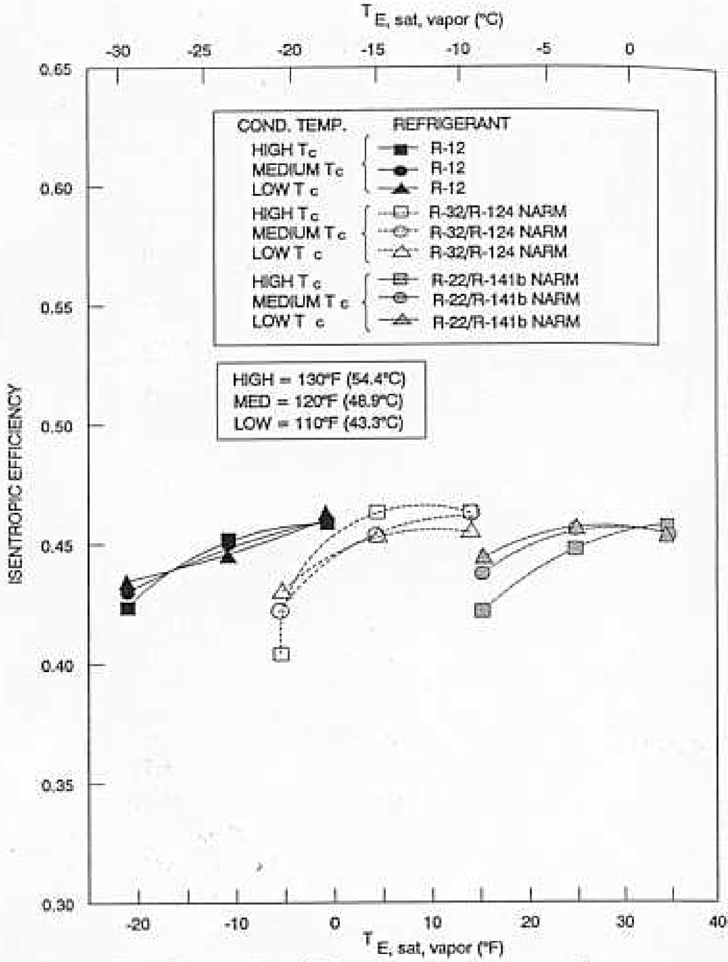

Figure 5 *Isentropic efficiencies versus saturated evaporator dew-point temperatures for a 670 Btu/h (196.3 W) R-12 compressor tested with R-12 and with R-22/R-141b and R-32/R-124 NARMS at the AHAM 90°/90°F (32.2°C) rating conditions.*

where $C_{(1 \ldots 6)}$ are the curve-fit coefficients; T_C and T_E are the condenser inlet and evaporator exit saturation (dew-point) temperatures, respectively; P_R is the pressure ratio; and P_C is the absolute condensing pressure. For limited sets of data, Rice found that curve fits to volumetric and isentropic efficiency were sometimes better behaved outside of the tested data range than similar representations of refrigerant mass flow rate and compressor power.

Ten-coefficient compressor performance equations, which have recently been recommended by ARI Standard 540-91 (ARI 1991), were not used for this analysis for various reasons. A significant deterrent was that the compressor data were available at only nine test points. Also, cubic equations can be more unreliable than the quadratic forms used here if extrapolations are required.

Because the curve-fit equation used for isentropic efficiency is a function of dew-point temperatures, it is clear from Figure 5 that the equation will apply only for the specific refrigerant and the mixture concentration for which the coefficients were generated. The functional form of the curve fit chosen for volumetric efficiency is of a more generalized nature because it is primarily a linear function of pressure ratio with a secondary quadratic dependence on discharge pressure. However, separate curve fits for each refrigerant were obtained for volumetric efficiency as well for maximum representation accuracy. The accuracy of the curve fits to the derived efficiency values was within ±2%.

Summary of Basic Compressor Performance

Figures 3 through 5 have shown that, for the trio of refrigerants tested in a common compressor, the compressor efficiencies have essentially the same absolute dependence on pressure ratio. This says the compressor will perform equally well with any of the considered refrigerants if the operating pressure ratios are the same. Conversely, the refrigerants with the lowest operating pressure ratios will have the highest compressor efficiencies.

Therefore, the factors that remain to determine which refrigerants will perform best in the test compressor are the operating pressure ratios, the ideal work requirements, and the ideal volumetric capacities under comparable system conditions. For the tested compressor, application of the appropriate operating conditions in the considered cycle will be the determining factor as to whether either blend can realize a performance advantage from higher theoretical EER or capacity and/or from the multiple benefits of reduced pressure ratios.

ANALYSIS—CONVERSION TO SYSTEM PERFORMANCE

A procedure was developed to convert compressor performance results with specific mixtures into potential system EER and capacity for the Lorenz-Meutzner r/f application. The concept of equivalent heat exchanger loading at fixed external source and sink conditions was applied to pure and mixed refrigerant r/f applications. Compressor conditions representative of the selected cycle are used to determine comparable rankings of NARM potential with a given compressor design.

Modifications to Lorenz-Meutzner System Modeling Program

CYCLE-Z is an EMTD-based single- and dual-evaporator computer model that is capable of maintaining fixed total heat exchanger loading.[1] The CYCLE-Z model was modified to meet the needs of the EHXL comparison protocol. Since this model was introduced in 1990, it has been modified to handle additional r/f modeling needs with options to:

1. conduct a numerical nonlinear enthalpy versus temperature analysis across the two-phase refrigerant regions of the heat exchangers (per routines adapted from Domanski [1990]),
2. maintain compressor suction gas temperature at a specified level through the use of a high-temperature intercooler,
3. specify levels of condenser subcooling, and
4. maintain fixed total (high- and low-temperature) intercooler subcooling.

The definitions for the overall heat exchanger EMTDs for condensers and dual evaporators required to hold a constant total system heat exchanger loading per unit of refrigerating capacity ($UA_{TOT}/\dot{Q}_E$) are as previously described by Rice and Sand (1990). Equivalent heat exchanger performance is assumed for all the refrigerants, which implies fan-forced airflow and countercrossflow configurations with sufficient passes to approach pure counterflow. Work by He et al. (1992) on the recircuiting of existing r/f coils suggests that such countercrossflow configurations are plausible. The overall heat exchanger conductance value, U, for the pure and mixed refrigerants is also assumed the same. This should be a reasonable approximation for applications such as domestic refrigerator/freezers that are air-side heat transfer limited.

Additional changes to the modeling program were needed specifically for the present evaluation. The CYCLE-Z model was modified so that compressor maps of volumetric and isentropic efficiency and a fixed compressor displacement could be used to compute system capacity and EER values specific to the tested compressor and refrigerants. This option allows direct use of output data files from the mixture map-fitting model discussed earlier.

Another program modification permitted determination of the overall heat exchanger EMTDs necessary to obtain, for a standard single-evaporator R-12 r/f configuration, specified evaporator and condenser saturation temperatures. CYCLE-Z was modified so that the saturation temperatures for either or both heat exchangers could be specified in lieu of heat exchanger EMTDs.

Conversion Procedure

The conversion from EMT to EHXL conditions for a Lorenz-Meutzner r/f configuration was accomplished in two steps. First, baseline heat exchanger loadings were determined for a single-evaporator R-12 r/f operating with the curve-fitted R-12 compressor map. These baseline loadings were determined by running the CYCLE-Z model for fixed external source and sink conditions at selected sets of evaporating and condensing temperatures. Evaporator temperatures of −20°, −15°, and −10°F (−28.9°, −26.1°, and −23.3°C) were selected to cover the range of low- to high-efficiency freezer coil designs, respectively. Condensing temperatures of 130°, 120°, and 110°F (54.4°, 48.9°, and 43.3°C) were chosen to cover a similar efficiency range of possible high-side designs. External fluid conditions were based on the single-design-point, 90°F (32.2°C) closed-door test condition.

Second, these baseline loadings obtained for the single-evaporator R-12 case were applied to L-M cycles using the tested blends of R-22/R-141b and R-32/R-124 and the appropriate compressor maps.[2] The range of equivalent L-M refrigerator/freezer heat exchanger loadings were applied at comparable ambient, freezer, fresh food, and compressor return gas temperatures; air-side temperature glides; and fresh food-to-freezer load ratios.

Assumed Conditions Baseline cycle conditions and assumptions for single-evaporator R-12 configuration were:

- 38°F (3.3°C) fresh food and 5°F (−15°C) freezer settings, giving for the freezer coil a mixed entering air temperature of 8.9°F (−12.8°C) (for an equal fresh food-to-freezer heat load) and a 9F° (5C°) air glide,
- 90°F (32.2°C) condenser air inlet and 104.4°F (40.2°C) air exit temperatures,
- 0F° (0C°) condenser subcooling and evaporator superheat,
- 90°F (32.2°C) suction gas temperature obtained through liquid-line subcooling in the high-temperature

TABLE 4 (SI Units)
System Performance Comparisons Between R-12 and Two Blends Under Equivalent Heat Exchanger Loading

R-12 Baseline Conditions		System Performance				
Evaporator Temp.	Condenser Temp.	R-12	R-22/R-141b	R-32/R-124	R-22/R-141b	R-32/R-124
(°C)	(°C)	COP			ΔEER vs R-12	ΔEER vs R-12
−28.9	43.3	1.08	1.27	1.33	17.0%	22.4%
"	48.9	0.97	1.15	1.19	18.5%	23.0%
"	54.4	0.87	1.03	1.05	18.2%	20.2%
−23.3	43.3	1.28	1.43	1.50	11.7%	17.7%
"	48.9	1.14	1.30	1.37	14.2%	20.9%
"	54.4	1.02	1.17	1.23	14.9%	21.0%
(°C)	(°C)	Capacity (W)			ΔCap vs R-12	ΔCap vs R-12
−28.9	43.3	154.2	208.0	208.4	34.9%	35.2%
"	48.9	133.9	185.9	179.3	38.9%	33.9%
"	54.4	115.2	163.5	147.4	41.9%	28.0%
−23.3	43.3	213.9	280.2	292.7	31.0%	36.9%
"	48.9	191.0	255.9	258.8	34.0%	35.5%
"	54.4	169.6	231.2	221.6	36.3%	30.6%
R-22/R-141b mixture is 80% R-22 by mass R-32/R-124 mixture is 15% R-32 by mass						

intercooler,

- compressor shell heat loss of 70% of compressor input power (based on air-over compressor calorimeter tests conducted by Sand et al. [1992]), and
- a one-to-one freezer-to-fresh-food heat load ratio.

Conditions and assumptions for the L-M configurations were:

- equivalent heat exchanger loadings at each baseline R-12 condition,
- other values the same as for the R-12 single-evaporator case except for
- 38°F (3.3°C) fresh food and 5°F (−15°C) freezer coil entering air temperatures and 9F° (5C°) air glides for both, and
- 90F° (50C°) total subcooling from the two intercoolers.[3]

Evaporator temperatures of −5°F (−20.5°C) and 0°F (−17.8°C) were not included because they resulted in unfeasible heat exchanger sizes for the fixed 38°F/5°F (3.3°C/−15°C) temperature settings. Because of this constraint, it was not possible to cover the full range of evaporator conditions as in the nine-point compressor map test matrix.

TABLE 4 (IP Units)						
System Performance Comparisons between R-12 and Two Blends under Equivalent Heat Exchanger Loading						
R-12 Baseline Conditions		System Performance				
Evaporator Temp.	Condenser Temp.	R-12	R-22/R-141b	R-32/R-124	R-22/R-141b	R-32/R-124
(°F)	(°F)		EER		ΔEER vs R-12	ΔEER vs R-12
−20	110	3.70	4.33	4.53	17.0%	22.4%
"	120	3.30	3.91	4.06	18.5%	23.0%
"	130	2.97	3.51	3.57	18.2%	20.2%
−10	110	4.36	4.87	5.13	11.7%	17.7%
"	120	3.88	4.43	4.69	14.2%	20.9%
"	130	3.48	4.00	4.21	14.9%	21.0%
(°F)	(°F)		Capacity (Btu/h)		ΔCap vs R-12	ΔCap vs R-12
−20	110	526.3	710.0	711.4	34.9%	35.2%
"	120	457.0	634.5	611.8	38.9%	33.9%
"	130	393.2	557.9	503.2	41.9%	28.0%
−10	110	729.9	956.2	999.0	31.0%	36.9%
"	120	651.9	873.4	883.3	34.0%	35.5%
"	130	579.0	788.9	756.5	36.3%	30.6%
R-22/R-141b mixture is 80% R-22 by weight R-32/R-124 mixture is 15% R-32 by weight						

COMPARISONS OF SYSTEM PERFORMANCE PREDICTIONS VERSUS CALORIMETER RESULTS

Relative System Performance

Once the conversion procedure was applied to the R-12 and NARM cases, a comparable set of cycle-representative refrigerant-side conditions and system EERs and capacities was obtained. These system-based predictions for the three refrigerants being considered are tabulated and compared in Table 4 as was done previously for the calorimeter results in Table 3 (with the exception of the 0°F [−17.8°C] evaporator condition).

Comparison of the gains shown in Table 4 with those from Table 3 shows that the EHXL-based results predicted for NARMs are much larger than for the EMT-based calorimeter results. The predicted gains for the NARMs relative to R-12 range from 12% to 23% in EER and 28% to 42% in capacity for Table 4 as compared to −5% to +4% in EER and −11% to +6% in capacity for Table 3. The relative EER rankings between blends are similar to those from the calorimeter tests while the capacity rankings are generally reversed. Of the EER gains shown in Table 4, only 2.2 and 4.5 percentage points are due to the slightly higher isentropic efficiencies of the R-22/R-141b and R-32/R-124 mixtures, respectively. Of the capacity gains, only 2.6 and 1.7 percentage points, respectively, are due to higher volumetric efficiencies.

The relative system EERs and capacities for the 120°F (48.9°C) equivalent condensing condition are shown in

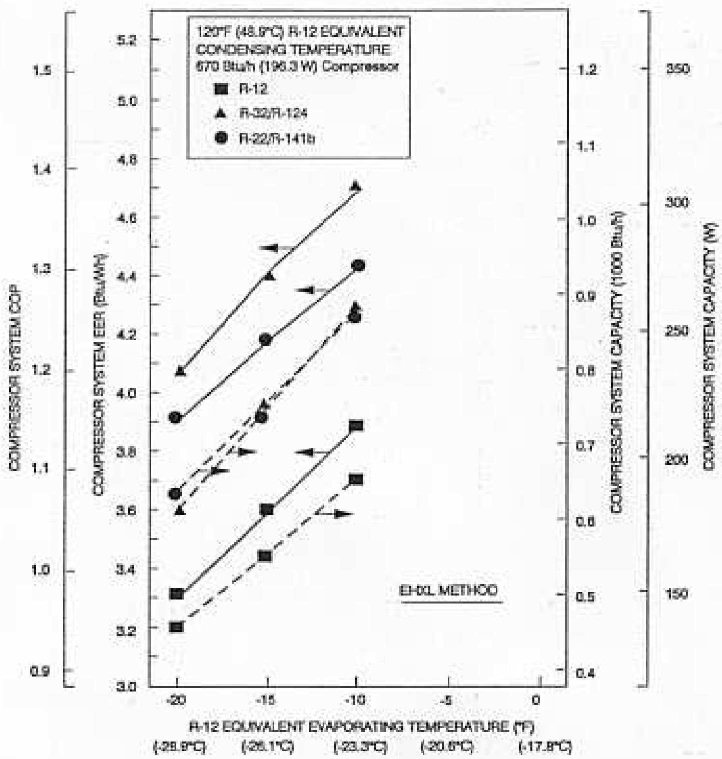

Figure 6 *Compressor system EERs and refrigeration capacities for R-12 and for R-22/R-141b and R-32/R-124 NARMs calculated using the equivalent-heat-exchanger-loading (EHXL) protocol for a Lorenz-Meutzner r/f at a 120°F (48.9°C) equivalent condensing condition.*

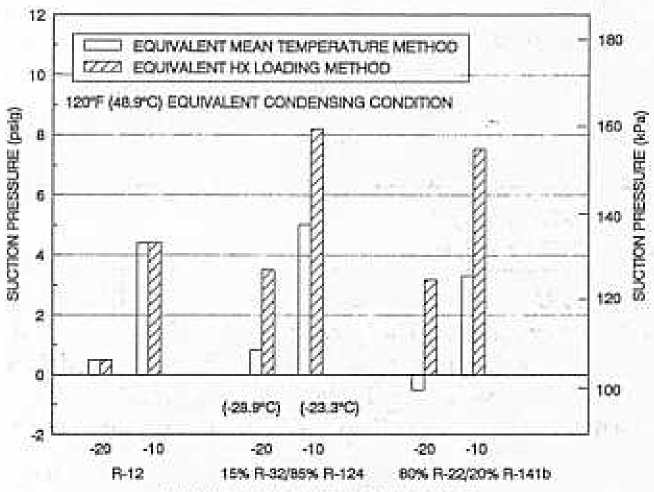

Figure 7 *Suction pressures for the equivalent-mean-temperature (EMT) and equivalent-heat-exchanger-loading (EHXL) methods of comparing pure and mixed refrigerants.*

Figure 6, which is of similar form to Figure 2 but with strikingly better results for the NARM cases. (Note the narrower range of equivalent evaporating temperatures in Figure 6 than in Figure 2.)

Operating Condition Comparisons

The resulting EHXL operating conditions for the pure and mixture cases are compared in Figures 7 through 9 to those for the EMT approach (as corrected in Table 2). The EHXL method predicts improved (higher) suction pressures for the mixtures, as shown in Figure 7. This is because increases in the mean evaporating temperature are predicted due to better overall glide matching with the refrigerated air conditions and the benefit of additional subcooling from use of the low temperature subcooler in the L-M cycle.

The EHXL-predicted discharge pressures shown in Figure 8 were, somewhat surprisingly, less favorable (also higher) for the mixtures than those calculated with the EMT method, implying higher mean temperatures than for the R-12 case. Further evaluation of the refrigerant versus air temperature profiles in the condenser revealed that the mixtures had significant overglides and, as a result, experienced more severe pinch points than with R-12. This result demonstrates that overgliding in heat exchangers can be as or more detrimental than undergliding (e.g., pure refrigerant cases). These possible detrimental effects with NARMs have been discussed by Didion and Bivens (1990).

As seen in Figure 9, the NARM pressure ratios predicted by the EHXL approach are 11% to 16% lower than the EMT-based ratios. Whereas the EMT pressure ratios for the blends were about 13% higher than R-12, the predicted system values in the Lorenz-Meutzner cycle are seen to be slightly lower. This is because of the more favorable suction pressures, which more than offset the slightly higher discharge pressures.

Effect of Liquid Line Temperature Assumption on Calorimeter Versus System Performance

Comparison of the EERs and capacities of R-12 for the system-based model in Table 4 with those in Table 3 shows that the calorimeter-based values have been derated by 16% at the −10°/130°F (−23.3°/54.4°C) standard rating condition. This occurred even though the operating pressures for R-12 remained the same for the EMT and the EHXL cases, as shown in Figures 7 and 8. The derating was caused by the assumption of no condenser subcooling in the system model used for the results in Table 4, instead of the 90°F (32.2°C) entering liquid line temperature as specified by AHAM for the calorimeter test results given in Table 3. This result indicates that, for pure refrigerants, the AHAM calorimeter-based EERs and capacities significantly overstate system performance. (For comparing performance between compressors using R-12 or other pure refrigerants, this is not a problem.)

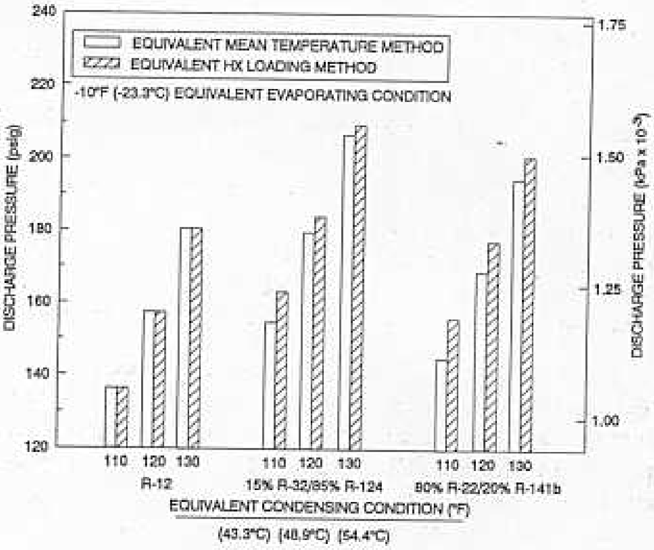

Figure 8 *Discharge pressures for the equivalent-mean-temperature (EMT) and equivalent-heat-exchanger-loading (EHXL) methods of comparing pure and mixed refrigerants.*

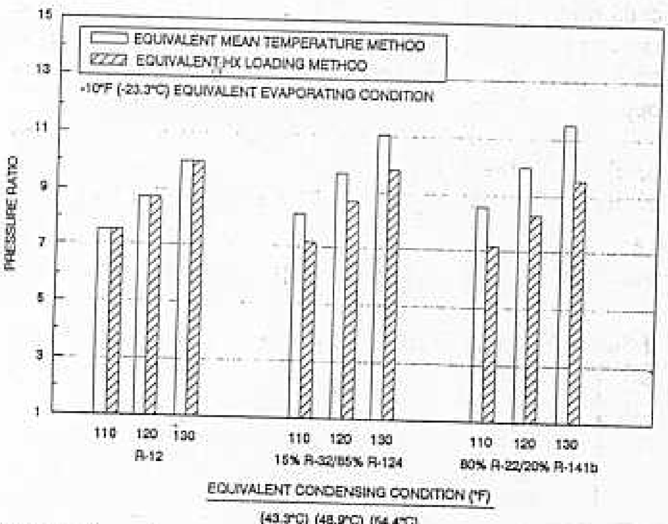

Figure 9 *Pressure ratios for the equivalent-mean-temperature (EMT) and equivalent-heat-exchanger-loading (EHXL) methods of comparing pure and mixed refrigerants.*

The AHAM procedure assumes (apparently for testing convenience) that the liquid line is cooled to 90°F (32.2°C) for all condensing temperatures, by some combination of condenser and ambient subcooling. Data taken at our laboratory on production r/f designs indicate that little condenser and ambient subcooling occurs in practice. Thus the system modeling assumption of *no* condenser subcooling appears to be closer to existing practice than is the AHAM test procedure.

Both the CYCLE-Z system analysis and the AHAM procedure do, however, count the liquid line subcooling required to superheat the suction gas to 90°F (32.2°C) as additional refrigerating capacity. Because in the AHAM test procedure, the liquid line is already held at 90°F, the energy required to superheat the suction gas can only be counted as refrigerating capacity if it is assumed to be obtained by further subcooling of the liquid line below 90°F (32.2°C).

When mixtures are considered, the AHAM liquid line condition used for the calorimeter testing has the effect of preferentially benefiting the pure refrigerant cases. Because mixtures have gliding temperatures in the condenser, the blends can exit the condenser at a considerably lower temperature than the pure refrigerant. The blends therefore do not receive as much advantage from further subcooling to 90°F (32.2°C) as in the case of R-12. Table 5 shows the amount of preferential subcooling given to R-12 by the AHAM procedure as compared to that for the two considered blends for the corrected EMT pressures of Table 2. The difference is about 17F° (9.4C°) for the R-22/R-141b case and around 19F° (10.6C°) for R-32/R-124.

The effect of replacing the fixed liquid line temperature assumption with a fixed low amount of condenser subcooling is a significant net gain in capacity and EER for the blends relative to R-12 performance, 7% to 8% for the limiting case of no subcooling in the present analysis. When referenced to the system gains for the blends that are given in Table 4, this translates into 38% to 50% of the predicted EER gain and 15% to 25% of the predicted capacity gain.

Possible EMT Testing Improvement A simple improvement to the calorimeter-predicted results would be to test according to the AHAM procedure and correct all the results back to 0F° (0C°) subcooling as described in the ASHRAE 23-78 compressor test procedure (ASHRAE 1978). This would provide more realistic EER estimates of the relative NARM r/f cycle potential. Even with this correction, the R-12 case still benefits from more "capacity-enhancing" liquid line subcooling than do the mixtures because of the greater amount of suction line superheating required for a pure refrigerant between the lower evaporator exit temperature and a fixed suction gas temperature. (The additional low intercooler subcooling assumed the L-M cycle does not contribute directly to greater evaporator capacity because the subcooling is done at the expense of two-phase evaporator capacity with a zero net gain.)

NARM EMT Testing Cautions A further correction should be applied to the NARM calorimeter tests for cases where the condenser exit temperature of the blend (bubble point) is lower than 90°F (32.2°C). In these instances, the NARM capacity should be decreased to the two-phase refrigerant enthalpy at 90°F (32.2°C). Otherwise, these mixtures can show performance advantages that are not thermodynamically consistent with the implicit source and sink temperatures. From Table 5, this condition occurred only for R-32/R-124 at the lowest equivalent mean condenser temperature. (Checks should also be made when calculating the mean evaporator temperature for the EMT approach so that the NARM dewpoint at the evaporator exit does not exceed the implied temperature of the fresh-food compartment. If so, the exit evaporator temperature for the EMT procedure should be limited to a reasonable fresh-food condition.)

EHXL Capacity Implications

The large increases in EHXL capacity for the NARMs relative to R-12 in Table 4 as compared to Table 3 suggest that *the EMT method can be especially inadequate for use in predicting the displacement or mixture concentration required for an L-M mixture cycle to achieve a desired capacity.*

Fixed Mixture Concentrations The EER results for refrigerants, either pure or mixed, that have a larger capacity than R-12 at a given compressor displacement can be interpreted in various ways. First, the predicted EERs can be viewed as achievable efficiencies in systems of the same total effective heat exchanger size and capacity as the R-12 baseline with displacements for the NARM cases scaled linearly downward (which assumes no loss in compressor efficiency with downsizing). Alternatively, the EER levels can be obtained at the predicted capacities for the tested compressor when applied to systems with proportionally larger effective heat exchanger sizes.

Adjusted Mixture Concentrations If the mixture concentration is to be changed to maintain a desired capacity with the tested compressor and equivalent heat exchanger size, the EER levels would be expected to change in response to changing mixture properties (glide, density, etc.) and pressures with concentration and any effect of the new concentration on the basic compressor efficiencies. For the tested blends, the EERs predicted in Table 4 would drop by an estimated 2% to 3% for R-32/R-124 and by 3% to 4% for R-22/R-141b, assuming the compressor efficiencies did not change. If a more generalized compressor efficiency model were available, a plot of mixture capacity and EER versus concentration could be generated with the CYCLE-Z model for the tested compressor at each level of assumed heat exchanger loading and the EERs could be determined at the concentrations that matched the R-12 single-evaporator capacity.

Performance Trends with Increasing Effective Heat Exchanger Size

EHXL Results Upon further examination, Table 4 contains some other surprising trends as well. One might expect that the EER gains for the NARMs relative to the R-12 would be greatest at the largest effective heat exchanger sizes (represented by the −10°/110°F [−23.3°/−43.3°C] condition). This is because NARMs can, under proper glide matching, make more cycle advantage of larger heat exchangers than pure (undergliding) refrigerants. Just the opposite EER trends occur in Table 4, however.

From a consideration of the refrigerant temperature profiles for the NARMs relative to the air streams (an example of which is given in Figure 1b), it was found that

TABLE 5
Subcooling to AHAM Liquid Line Condition for R-12 Versus Two Blends

	Degrees of Subcooling to 90°F (32.2°C) Ambient		
		Refrigerant	
Condenser Mean Temp. °F (°C)	R-12	R-22/R-141b	R-32/R-124
	F°(C°)	F°(C°)	F°(C°)
110 (43.3)	20 (11.1)	3.3 (1.8)	-0.1 (-.06)
120 (48.9)	30 (16.7)	13.5 (7.5)	10.7 (5.94)
130 (54.4)	40 (22.2)	23.7 (13.2)	21.4 (11.9)

R-22/R-141b mixture is 80% R-22 by mass
R-32/R-124 mixture is 15% R-32 by mass

Constant Equivalent Mean Temperature Conditions narrower pinch points are being realized on one or the other evaporator for each of the considered NARMs than for the pure refrigerant case (at the freezer for the R-32/R-124 mixture and at the fresh-food evaporator for the R-22/R-141b case). It appears that while the NARMs show a significant performance boost over the single-evaporator R-12 case due to the better overall glide matching over the full −4°F to 38°F (−20°C to 3.3°C) evaporator air-side glide range, *local pinch points develop as more heat exchanger area is added which tend to reduce rather than enhance the NARM L-M cycle gain.* This occurs because the sum of the two air-side glides in the individual heat exchangers is smaller than the required overall air-side evaporator glide (18F° versus 42F° [10C° versus 23.3C°]).[4] In the condenser, both mixtures overglide relative to the air-side temperature profile and have closer pinch points than R-12; therefore, additional condenser area is of diminishing benefit as well.

The trends in NARM capacity gain relative to R-12 with respect to increased *evaporator* area (increased evaporator temperature) are roughly similar to those for EER, with predominately lower gains from larger heat exchangers. With respect to increased *condenser* area (decreased condenser temperature), NARM capacity gains relative to R-12 track those in EER for R-22/R-141b but reverse trend for R-32/R-124. Some of this difference in trend is due to the effects of a more rapidly rising discharge pressure and pressure ratio for R-32/R-124 than for R-22/R-141b which results in a more rapid falloff in volumetric efficiency.

It is apparent from the preceding observations that heat exchanger size tradeoffs for NARMs in the L-M refrigerator/freezer application can be counterintuitive and involve various competing effects. Related observations have been made by Smith et al. (1990). For the limited set of cases considered here, the NARMs generally show larger potential benefits with smaller heat exchanger areas.

EMT Results These trends of decreasing NARM gain with larger heat exchangers are not observed for the EMT results as given in Table 3. There, for the −20° to −10°F (−28.9° to −23.3°C) evaporator range, the mixture EER changes relative to R-12 for conditions representing larger heat exchangers are either neutral or positive (e.g., R-32/R-124).

DISCUSSION

Relevance of Results to EMT Calorimeter Testing of NARMs

Best Approach to System Conditions Debate regarding the most appropriate way to calculate EMTs should focus on which methods will predict pressures closest to the expected *system* operating conditions with NARMs for a given application rather than on which method is the most rigorous calculation of the mean refrigerant temperature in a heat exchanger.

Testing Convenience Ease of calorimeter setup should also be an important consideration. In this regard, the approach taken in this work for the evaporator calculation, where the mean temperature is calculated based on the evaporator bubble-point temperature instead of the temperature at the inlet quality (which would be based on an unrealistic liquid line temperature), seems an appropriate simplification. It allows the use of the same evaporator pressure with different condenser pressures and results in a higher suction pressure for NARMs than those that would be specified with the more exact mean temperature.

Bracketing the Operating Envelope The 10-point averaging of the two-phase temperature is recommended because it will help ensure that calculated condenser pressures are not unnecessarily under- or overestimated for refrigerants with strong two-phase nonlinearities. Based on the results shown in Figure 8, the condenser EMTs for use with NARMs in the Lorenz-Meutzner cycle could be increased by about 5°F (2.8°C) to ensure that the tested pressure range is beyond the expected envelope of operation. (This would also provide some adjustment margin for possible effects of revised mixture concentrations on pressure levels for system modeling conversions.) The overriding goal of these adjustments would be to test the compressor over a pressure range close to but slightly wider than the expected conditions so that system models will be interpolating within rather than extrapolating beyond the compressor maps.

Correcting to Fixed Subcooling Calorimeter capacities and EERs should, at a minimum, be corrected to fixed condenser subcooling levels if they are to be compared between different mixtures and if system EHXL conversion procedures are not available.[5] However, where sufficient refrigerant property information is available, an examination of the compressor volumetric and isentropic efficiencies may prove more illuminating, especially when compressor performance is the primary issue. These efficiency data will also provide information of more general utility to the compressor designer. Such data for a range of NARMs, compressor sizes, and types are needed for the development and validation of more-refrigerant-generalized compressor efficiency representations.

Different Approaches to Converting from Calorimeter to System Conditions

Fixed External Conditions, Variable Heat Exchanger Sizes The r/f application is particularly well-suited to allowing compressor performance to be transformed into comparative system performance with NARMs because r/f's have a single, predominant design point with fixed source and sink temperatures. Because of this, the calorimeter tests are conducted over a rather narrow range of conditions relative to those for an air conditioner or heat pump. With the assumption of *fixed* source and sink temperatures, the effective size of the heat exchangers was the design *variable* chosen to simulate a range of condenser and evaporator conditions.

Variable External Conditions, Fixed Heat Exchanger Sizes To consider a range of source and sink temperatures, the heat exchanger loadings should be held constant only at one design point. At off-design points, ambient temperatures would be varied while heat exchanger hardware would remain fixed at the original design point. In this manner, a range of compressor conditions under *variable* source and sink temperatures could be simulated for each candidate refrigerant, at a *fixed* level of total heat exchanger size. Off-design heat exchanger loadings would be free to vary as driven by the capacity characteristics of the individual refrigerant alternatives. This approach might be more appropriate for an air-to-air heat pump.

Use in Determining NARM Heat Exchanger Losses

Hardware Performance Limits Calorimeter tests conducted with the suction and discharge pressures predicted from the EHXL approach and the evaporator capacity adjusted to 0°F (0°C) condenser subcooling should obtain the EER and capacity gains predicted in Table 4. Similar predicted performance gains for a specific breadboard hardware configuration of an advanced r/f design can serve to provide an upper limit on expected NARM performance gains relative to a pure refrigerant with a given compressor.

Heat Exchanger Performance Losses This information can be used in combination with breadboard test data to quantify how much the heat exchanger performance is deviating from the desired approach to counterflow performance and to identify which heat exchanger is operating further from modeled expectations. The presented calorimeter conversion procedure provides a way that NARM compressor performance can be decoupled from NARM heat exchanger performance so that the relative compressor and heat exchanger losses for a particular mixture in advanced breadboard designs can be more easily determined.

Revisable Performance Targets By using a calorimeter map representation and system model, the proposed conversion approach can be applied after the compressors have been tested by any type of EMT protocol. Changes in model assumptions, external conditions, design parameters, and cycle configurations can be made and the effects on limiting case performance with NARMs determined without having to retest the compressor. Having a compressor map suitable for a range of mixture concentrations would further broaden the capability of this approach.

SUMMARY AND CONCLUSIONS

Two protocols for comparing pure and mixed refrigerant performance based on calorimeter tests have been compared. An EMT approach is the most appropriate for use in establishing NARM pressures for calorimeter testing. An EHXL method applied in conjunction with the calorimeter data yields operating conditions that are more cycle-representative and predicts higher performance for blends in a Lorenz-Meutzner configuration relative to a single-evaporator, R-12 baseline. Higher suction and discharge pressures and 11% to 16% lower pressure ratios were predicted for the blends with the EHXL protocol as compared to the EMT approach. Mixture EER gains predicted from the EHXL protocol were 14 to 22 percentage points higher than the EMT-based results with capacity increases of 30 to 50 percentage points higher relative to R-12.

Two means of comparing compressor performance with mixtures are recommended: one based on isentropic and volumetric efficiencies and the other based on EHXL system capacity and EER. Compressor calorimeter data for NARMs obtained by use of the EMT protocol and AHAM 90°/90°F (32.2°C) liquid line/suction line conditions provide sufficient information to derive the basic efficiencies and interpolate these to system performance. Comparison of basic compressor efficiencies showed that the R-12 compressor performed as well or better with either of the two mixtures.

The EMT-based calorimeter results for EER and capacity can give misleading indications of mixture performance rankings and trends relative to R-12. Conversion of the calorimeter data from the tested AHAM 90°F (32.2°C) liquid-line temperature to 0F° (0C°) of condenser subcooling is recommended as a minimum step toward correcting the mixture EER predictions (+7 to 8 percentage points) versus R-12.

Full conversion of the calorimeter test data to system performance with equivalent heat exchangers using a refrigerant screening model such as CYCLE-Z is recommended to obtain the most reliable refrigerant rankings, capacity predictions, and trends with regard to effective heat exchanger size.

With the EHXL method, compressor conditions representative of increased heat exchanger size were found generally to *reduce* the L-M cycle performance advantage of NARMs relative to R-12 whereas for the EMT approach the effect was positive to neutral. This result emphasizes the importance of considering the system implications of simplifying calorimeter assumptions when evaluating mixtures in given applications and cycle configurations.

The suggested procedures can be used with suitable EMTD- or UA-based thermodynamic screening models to transform calorimeter results into more representative indicators of mixed-refrigerant cycle performance potential. By providing an upper limit on expected system performance with a given compressor, EHXL-based results can be used with experimental system data to quantify heat exchanger losses in advanced breadboard designs.

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