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OAK RIDGE NATIONAL LABORATORY

MARTIN MARIETTA

# MODELED PERFORMANCE OF NON-CHLORINATED SUBSTITUTES FOR CFC-11 AND CFC-12 IN CENTRIFUGAL CHILLERS

Research Project RP2891-14
Final Report

J. R. Sand
S. K. Fischer

MANAGED BY
MARTIN MARIETTA ENERGY SYSTEMS, INC.
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## RECURRING ACRONYMS

| | |
|---|---|
| AETD | average effective temperature difference |
| COP | coefficient of performance |
| $\Delta T$ | changes in temperature |
| EPA | United States Environmental Protection Agency |
| EPRI | Electric Power Research Institute |
| HCFC | hydrochlorofluorocarbon |
| HFC | hydrofluorocarbon |
| IC | interaction coefficient |
| kW/t | kilowatts per ton of refrigeration |
| LMTD | log mean temperature difference |
| NARM | nonazeotropic refrigerant mixture |
| NEARM | nearly azeotropic refrigerant mixture |
| ODP | ozone depletion potential |
| ORNL | Oak Ridge National Laboratory |

## ACKNOWLEDGMENTS

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## ABSTRACT

A number of partially fluorinated alkanes and ethers were identified from earlier technical publications and from a joint EPRI/EPA project as potential alternatives for CFC and HCFC refrigerants. These larger molecules, by virtue of their more elaborate structure, have larger vapor phase heat capacities ($C_p$s), larger molecular weights, and lower critical temperatures than currently used refrigerants, which result in decreased volumetric capacities and coefficients of performance (COPs) in simple cycle applications.

These compounds were fitted to the Lee-Kessler-Plöcker and Carnahan-Starling-DeSantis equations of state, and refrigerant property routines based on these equations were used to simulate their performance in a centrifugal chiller as pure refrigerants, nearly azeotropic refrigerant mixtures (NEARMs), and nonazeotropic refrigerant mixtures (NARMs). Results indicate that several chlorine-free compounds give modeled chiller performance comparable to R-11 and R-123 and better than R-12 and R-134a. Modifications to the current chiller cycle—such as liquid subcooling and suction gas superheat—may offer unique advantages for more complicated, larger refrigerant molecules.

An algorithm based on molecular bond energies and vapor phase $C_p$s was used to estimate the flammability of these alternatives and of blends made from them. Blends of these refrigerants may be required to mitigate the flammability of some alternatives that show the best performance.

The low temperature lift in a chiller cycle makes the heat exchanger efficiency improvements obtained with NARMs a more significant fraction of the overall cycle efficiency. Using NARMs (or zeotropes) as refrigerants in chillers may represent a way to replace environmentally damaging refrigerants and to improve cycle efficiency. Considerable hardware and circuiting changes will be required to obtain these efficiency benefits, however.

Depending on desirable and permissible deviations from the currently used chiller operating conditions and on the acceptability of flammable refrigerant combinations, ideal COP improvements of 5–10% over R-11 are possible with NARMs.

## SUMMARY

Centrifugal water chillers represent one of the most efficient applications of electrical energy for the purpose of air conditioning currently available. Roughly 70,000 centrifugal chillers are in use in the United States, and nearly 110,000 are installed throughout the world.[1,2] Eighty percent of these water chillers use R-11 as the refrigerant primarily because of its high cycle efficiency but also because these compressors operate at slower impeller speeds, and R-11 is easier to handle and store.

The phaseout of R-11 as a result of the Montreal Protocol, recent reports indicating a worsening of the stratospheric ozone depletion, and chronic toxicological effects of R-123 have heightened efforts to find a chlorine-free alternative that can be used instead of R-11 or R-123 in large, direct-driven centrifugal chillers. Several potentially useful fluorinated ethers and propanes were fit to the Lee-Kessler-Plöcker (LKP) and Carnahan-Starling-DeSantis (CSD) equations of state (EOSs). Refrigerant property routines based on these equations were then used to evaluate cycle performance estimates at various state points in a chiller circuit from known operating temperatures and pressures.
Modeled cycle performance of pure compounds, nearly azeotropic refrigerant mixtures (NEARMs), and nonazeotropic refrigerant mixtures (NARMs) of these chlorine-free alternatives were compared to that of currently used refrigerants using appropriate computerized cycle simulations. The effects of several variations to standard chiller operating conditions were also modeled to illustrate differences between pure refrigerants and mixtures and differences between these larger molecules and previously used refrigerants that have relatively simple molecular structures.

Results indicate that several chlorine-free compounds have modeled performance comparable to that of R-11 and R-123 and better than that of R-12 and R-134a. Compounds showing the best performance are estimated to be flammable, so blends with nonflammable alternatives may be desirable. Performance improvements ranging from 5% to 10% over CFC-11 are possible if cycle conditions are changed to suit the particular characteristics of these new compounds and mixtures of these new compounds.

Modifications of the basic chiller cycle, such as deliberately adding subcooling or liquid-to-suction line heat exchange, will benefit the performance of refrigerants with more complex molecular structures and larger molecular heat capacities than currently used refrigerants. These modifications will involve significant changes to the design and substantial increases in the complexity of chillers because of the large volumes of refrigerant that must be circulated to achieve acceptable cooling capacities.

# 1. BACKGROUND AND INTRODUCTION

Current scientific evidence indicates that stratospheric chlorine concentrations below two parts-per-billion will be necessary to reverse "ozone hole" formations over the Earth's polar regions each spring and to prevent measurable decreases of stratospheric ozone at lower latitudes. This makes it unlikely that hydrogen-containing chlorofluorocarbon (HCFC) alternatives with non-zero ozone depletion potentials (ODPs), no matter how small, will be accepted over the long term as refrigerants or blowing agents for foamed insulations. Pressure to eventually eliminate all high volume uses of chlorine-containing refrigerants provides a strong incentive to find hydrofluorocarbon (HFC) or alternative chlorine-free compounds with pressure-volume-temperature (PVT) characteristics similar to those of R-11 and R-123 for new and existing large centrifugal chiller applications.

Centrifugal water chillers represent one of the most efficient applications of electrical energy for the purpose of air conditioning currently available. Roughly 70,000 centrifugal chillers are in use in the United States, and nearly 110,000 are installed throughout the world.[1,2] Eighty percent of these water chillers use R-11 as the refrigerant primarily because of its high cycle efficiency but also because these compressors operate at slower impeller speeds, and R-11 is easier to handle and store.

The phaseout of R-11 as a result of the Montreal Protocol, recent reports indicating a worsening of the stratospheric ozone depletion, and chronic toxicological effects of R-123 have heightened efforts to find a chlorine-free alternative that can be used instead of R-11 or R-123 in large, direct-driven centrifugal chillers.

In addition to the numerous environmental and chemical requirements of an acceptable alternative refrigerant for current centrifugal designs, the proposed alternative should have a high critical temperature, a low-to-moderate gas heat capacity (40–100 $\mathrm{Jmol^{-1}K^{-1}}$), and a high critical pressure.[3] Given the limited number of chemical compounds that satisfy most of these requirements, some compromises in refrigerant properties or changes in the current chiller design may be required.

Several potentially useful fluorinated ethers and propanes were synthesized and partially characterized in a joint Electric Power Research Institute (EPRI)/Environmental Protection Agency (EPA) project aimed at finding new classes of organic compounds that could be used for chlorofluorocarbon (CFC) applications with fewer detrimental effects on the earth's stratospheric ozone layer. Some essential physical property measurements were part of this project, and these were used to fit the compounds to the Lee-Kessler-Plöcker (LKP) and Carnahan-Starling-DeSantis (CSD) equations of state (EOSs). Refrigerant property routines based on these equations were then used to evaluate cycle performance estimates at various state points in a chiller circuit from known operating temperatures and pressures. Other partially fluorinated ethanes and ethers suggested in earlier American Society for Heating, Refrigerating, and Air-Conditioning Engineers (ASHRAE) publications were also included in the study.[4,5]

## 2. COMPOUNDS MODELED

The compounds that were screened for chiller performance are listed in Table 1. Structural formulas are given in addition to normal boiling points and critical temperatures to provide a correlation with assigned refrigerant numbers. A standardized method for designating propanes has been established by ASHRAE,[6] but no standard method has been adopted for ethers. The numerical designators given in Table 1 provide a convenient and efficient shorthand method for referring to these compounds in subsequent tables and results.

The system used in this table assigns an "E" prefix for ethers. The numbering system is not definitive for methyl-ethyl-ethers because the position of the ether linkage in these molecules is not uniquely defined by the number or the lowercase suffix letters. The "C" prefix is added for cyclic molecules, and the "EE" prefix refers to the two ether moieties in $CF_3$-O-$CF_2$-O-$CF_3$ (EE-218).

Whenever possible, boiling points and critical temperatures were taken from published or pre-publication data coming out of the laboratories at the University of Tennessee and Clemson University, which synthesized these compounds. As part of the program, a minimum number of physical property measurements were made on the synthesized products.[7,8] For some compounds, critical properties were found in the Thermodynamics Research Center Thermodynamics Tables.[9] If no experimentally measured physical property data were available, required data were estimated using techniques outlined in reference 10.

Critical temperatures, pressures, and volumes were estimated using the group contribution method of Joback that is described on pages 12–23 of reference 10. Average absolute percentage errors of 0.8%, 5.2%, and 2.3% can be anticipated for critical temperatures, critical pressures, and critical volumes, respectively, if this method is used for compounds whose normal boiling points are known.

Acentric factors that were needed for the LKP routines were estimated from Eq. (1), using Table 1 measured or estimated boiling points, critical temperatures, and critical pressures.

Table 1. Chlorine-free CFC alternatives evaluated as CFC-11 and CFC-12 replacements in centrifugal chillers

| Designation | Molecular formula | Normal boiling point (°F) | Critical temperature (°F) | Source of property information |
|---|---|---|---|---|
| E-254cb | $CHF_2$-O-$CF_2CH_3$ | 97.6 | 373.0 | ASHRAE papers[a] |
| E-245cb | $CF_3$-O-$CF_2CH_3$ | 93.3 | 365.3 | ASHRAE papers[a] |
| R-152 | $CH_2FCH_2F$ | 87.2 | 397.1 | ASHRAE papers[a] |
| E-143 | $CH_2F$-O-$CHF_2$ | 86.1 | 368.3 | ASHRAE papers[a] |
| R-245ca | $CHF_2CF_2CH_2F$ | 78.8 | 345.8 | EPA/EPRI project |
| R-245fa | $CF_3CH_2CF_2H$ | 59.5 | 315.4 | EPA/EPRI project |
| R-236ea | $CF_3CHFCHF_2$ | 43.7 | 285.9 | EPA/EPRI project |
| R-143 | $CH_2FCHF_2$ | 41.0 | 316.0 | ASHRAE papers[a] |
| R-236ca | $CHF_2CF_2CHF_2$ | 41.0 | 282.1 | EPA/EPRI project |
| E-134 | $CHF_2$-O-$CHF_2$ | 40.4 | 308.2 | ASHRAE papers[a] |
| R-236cb | $CH_2FCF_2CF_3$ | 34.2 | 278.0 | EPA/EPRI project |
| R-236fa | $CF_3CH_2CF_3$ | 30.0 | 267.2 | EPA/EPRI project |
| E-227ca | $CF_3$-O-$CF_2CHF_2$ | 24.5 | 238.4 | EPA/EPRI project |
| EE-218 | $CF_3$-O-$CF_2$-O-$CF_3$ | 14.4 | 210.6 | EPA/EPRI project |
| R-227ea | $CF_3CHFCF_3$ | 4.6 | 218.2 | EPA/EPRI project |
| R-245cb | $CH_3CF_2CF_3$ | −0.9 | 224.5 | EPA/EPRI project |
| E-143a | $CH_3$-O-$CF_3$ | −10.8 | 220.7 | EPA/EPRI project |
| CE-216 | $CF_2$-$CF_2$-$CF_2$-O | −20.4 | 191.6 | EPA/EPRI project |
| E-125 | $CF_3$-O-$CHF_2$ | −43.5 | 178.2 | EPA/EPRI project |

[a] Summarized in E. Vineyard, J. Sand, and T. Statt. "Selection of Ozone-Safe Nonazeotropic Refrigerant Mixtures for Capacity Modulation in Residential Heat Pumps." *ASHRAE Trans.* vol. 95, 1989, pp. 34–46; and W. Kopko. "Beyond CFCs: Extending the Research for New Refrigerants." *Proceedings of ASHRAE's 1989 CFC Technology Conference.* Gaithersburg, MD; National Institute of Standards and Technology, September 1989, pp. 39–46.

$$\omega = \frac{3}{7}\left[\frac{T_b / T_c}{1 - T_b / T_c}\right] \times \log P_c - 1 \quad , \tag{1}$$

where

$\omega$ = acentric factor,
$T_b$ = normal boiling point (K),
$T_c$ = critical temperature (K),
$P_c$ = critical pressure (atmospheres).

Originally, the acentric factor was used to describe a molecule's acentricity or nonsphericity, but it is currently used more to indicate the complexity of molecular geometry and the polarity of molecules. PVT and thermodynamic results calculated from the LKP routines are quite sensitive to the value assigned to this factor, as will be shown later in this report.

Ideal gas heat capacities that were needed for both the LKP and CSD equation of state routines were obtained using the group contribution method worked out by Joback and described on pages 154–157 of reference 10. The four term polynomial obtained from this method was converted to a corresponding three term form, which was required for the CSD routines, using a standard statistical program.

# 3. EQUATION OF STATE CONSIDERATIONS

Fortran and basic computer codes based on several well established EOSs have been developed so that refrigerant PVT behavior and thermodynamic properties can be calculated from known values. These codes can conveniently be used to model the performance of pure and mixed refrigerants in air conditioning and refrigeration cycles on a computer. Some routines, like those based on the Martin-Hou EOS, give very accurate results but require many coefficients and constants specific to the refrigerant being evaluated. Values for these coefficients and constants must be derived from extensive physical property measurements. In addition, these more elaborate EOS routines have not been fitted with the required mixing rules and algorithms to handle mixtures of refrigerants.

Several simpler (two or three term) EOSs—like the Redlich-Kwong-Soave, Peng-Robinson, CSD, and LKP—have the necessary mixing rules and computational capabilities to simulate the characteristics of refrigerant mixtures. While these may not be as accurate as the Martin-Hou or Benedict-Webb-Rubin representations, they do not require as much physical property data to characterize a compound or as much computer-run time for implementation. Most are accurate enough to prescreen candidates for relative comparisons.

Of all the computer routines that encompass most of the newer refrigerant compounds and handle mixed refrigerants, the CSD code, which was started at and is currently supported by the National Institute of Standards and Technology (NIST), and the LKP routines, which were introduced by Dr. Kruse's group at the University of Hannover and were modified by Steve Fischer at Oak Ridge National Laboratory, are the most readily available.[11]

Initial efforts at modeling these new fluorinated alkanes and fluorinated ethers employed the LKP EOS routines for the following reasons:

- The LKP is a reduced properties EOS that requires less measured physical property data for each new compound.

- The LKP code contains an algorithm for estimating unknown interaction coefficients for refrigerant blends.

- The LKP routines gave better estimates for the cycle performance of polar and nonpolar refrigerants than the CSD code.

The LKP EOS requires data for the critical temperature, critical pressure, normal boiling point, and molecular weight and a correlation for ideal gas heat capacity in terms of temperature. It also requires an acentric factor for each compound, which can be obtained from vapor pressure data or calculated from critical point values.

An essential parameter for accurately calculating the thermodynamic properties of mixtures is an interaction coefficient (IC), which characterizes and quantifies the extent of ideality or non-ideality of the solutions formed by combinations of two or more compounds. The most reliable ICs are based on experimentally measured PVT data for known mixtures. Plöcker's refinement of the work of Lee and Kessler includes a correlation he developed between the interaction coefficient and the critical temperature and molar volumes.[12]

Since work on blends was one of the major components of this evaluative work, all of the evaluated compounds were coded into the LKP routines.

Ideal cycle comparisons made by using thermodynamic values taken from ASHRAE tables and those calculated from CSD and LKP refrigerant property routines are shown in Table 2. The CSD EOS is based on a "hard sphere" theory of molecular structure, which is a poor approximation for molecules like propane and ammonia. The reduced property/corresponding state approach used in the LKP EOS thus seems to work better. One of the initial concerns in this work was that the PVT and thermodynamic behavior of ethers was to be estimated from a small amount of

Table 2. ideal cycle comparisons using ASHRAE, CSD, and LKP refrigerant property routines

(40°F evaporator, 100°F condenser, 0 F° superheat[b]/subcooling)

| | Evaporator pressure (psia) | Condenser pressure (psia) | Refrigerant circulated (cfm/ton) | Isentropic COP | R-11 (%)[a] |
|---|---|---|---|---|---|
| R-11 | | | | | |
| ASHRAE Tables | 7.05 | 23.64 | 15.85 | 7.570 | 100.0 |
| CSD Routines | 7.03 | 23.52 | 15.93 | 7.546 | 99.7 |
| LKP Routines | 7.06 | 23.68 | 15.85 | 7.544 | 99.7 |
| R-123 | | | | | |
| ASHRAE Tables | 5.78 | 20.77 | 18.85 | 7.453 | 98.2 |
| CSD Routines | 6.35 | 22.41 | 17.46 | 7.113 | 94.0 |
| LKP Routines | 5.12 | 19.17 | 20.63 | 7.410 | 97.9 |
| R-114 | | | | | |
| ASHRAE Tables | 15.13 | 46.15 | 8.99 | 6.864 | 90.7[b] |
| CSD Routines | 15.15 | 45.84 | 9.08 | 6.843 | 90.4[b] |
| LKP Routines | 15.16 | 45.75 | 9.12 | 6.851 | 90.5[b] |
| R-12 | | | | | |
| ASHRAE Tables | 51.70 | 131.72 | 3.06 | 7.061 | 93.3 |
| CSD Routines | 51.60 | 131.35 | 3.10 | 7.051 | 93.1 |
| LKP Routines | 51.39 | 130.63 | 3.10 | 7.069 | 93.4 |
| R-500 | | | | | |
| ASHRAE Tables | 60.72 | 155.80 | 2.62 | 6.702 | 88.5 |
| CSD Routines | 62.31 | 153.76 | 2.71 | 6.929 | 91.5 |
| LKP Routines | 60.72 | 155.03 | 2.66 | 6.903 | 91.2 |
| R-134a | | | | | |
| ASHRAE Tables | 49.76 | 138.90 | 2.99 | 6.937 | 91.6 |
| CSD Routines | 50.25 | 140.57 | 2.93 | 6.973 | 92.1 |
| LKP Routines | 51.03 | 140.77 | 2.96 | 6.925 | 91.5 |
| R-22 | | | | | |
| ASHRAE Tables | 83.25 | 210.67 | 1.91 | 6.984 | 92.3 |
| CSD Routines | 83.01 | 210.26 | 1.90 | 6.991 | 92.4 |
| LKP Routines | 83.28 | 210.13 | 1.91 | 6.995 | 92.4 |
| R-290 (Propane) | | | | | |
| ASHRAE Tables | 78.78 | 189.04 | 2.26 | 6.851 | 90.5 |
| CSD Routines | 78.90 | 190.01 | 2.26 | 6.759 | 89.3 |
| LKP Routines | 78.96 | 189.45 | 2.25 | 6.831 | 90.2 |
| R-717 (Ammonia) | | | | | |
| ASHRAE Tables | 73.11 | 211.40 | 1.70 | 7.261 | 95.9 |
| CSD Routines | 63.96 | 142.53 | 2.81 | 2.460 | — |
| LKP Routines | 74.90 | 215.85 | 1.67 | 7.205 | 95.2 |

[a] Relative to ASHRAE tables value.
[b] Superheat added to avoid wet compression.

physical property data. Table 2 indicates that LKP routines seem to give better results with a more diverse class of compounds.

As the modeling work progressed, it was necessary to determine the speed of sound in the compressor suction gas in order to evaluate the rotational mach number of the chiller impeller. Attempts to approximate the speed of sound based on numerical differentiation at two slightly different state points bracketing suction conditions were not successful. Since the CSD routines contained a subroutine that returned the velocity of sound from known conditions at the suction, the LKP routines were used to calculate vapor pressures and saturated liquid and vapor densities, which were then used to generate CSD EOS coefficients. The appropriate CSD routine was used to obtain values for the velocity of sound in the chiller modeling work, but all of the modeled predictions were ultimately obtained from the LKP routines and code.

An indication of how well the LKP property routines represent actual experimental PVT data was obtained by comparing the saturated vapor pressure measurements made on several compounds at the University of Tennessee and Clemson University to those predicted by the LKP routines. Figure 1 shows the percent deviation of vapor pressures calculated using the LKP routines from values reported for some of the partially fluorinated propanes being evaluated in this study. Figure 2 shows similar data for five of the fluorinated ethers being investigated. The worst deviations were near 4% for E-125 in Fig. 2, but errors of less than 2% were generally more common.

More recent results presented at the 1991 International CFC and Halon Alternatives Conference in Baltimore indicate a different boiling point and critical temperature for E-125,[13] which may improve the results shown in Fig. 2. In the 40°F to 100°F region, important for chiller modeling, the deviation of calculated from experimental values averaged less than 1%. This accuracy was considered acceptable for a screening exercise. When PVT data were available for these alternative fluids, they were used to adjust the acentric factor ($\omega$) to improve the fit between calculated and experimental results. In most cases, these adjusted values replaced ones that were estimated from critical temperatures, normal boiling points, and critical pressures.

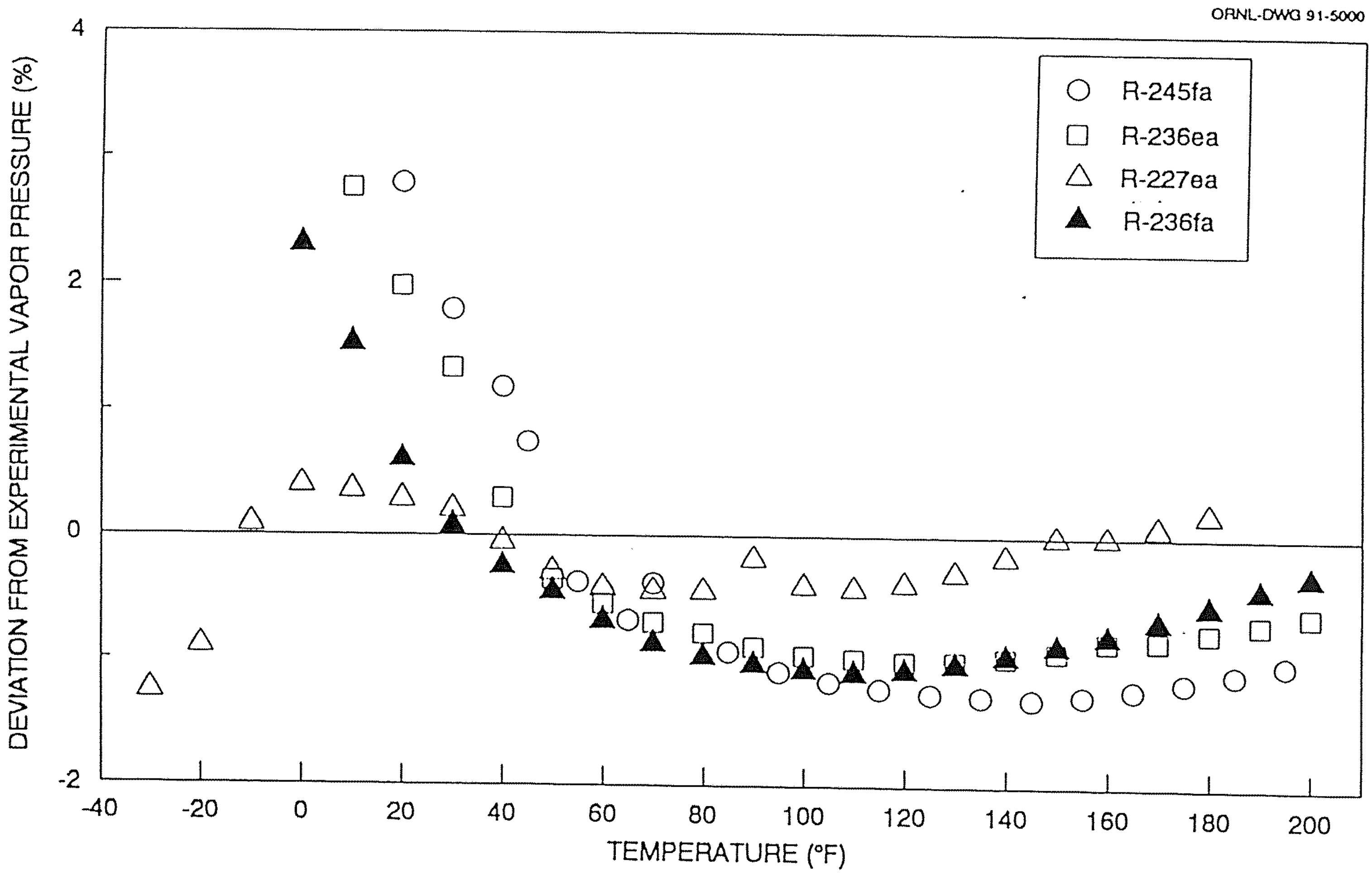


Fig. 1. Deviations of vapor pressures calculated by LKP routines from experimental values—propanes from EPRI/EPA project.

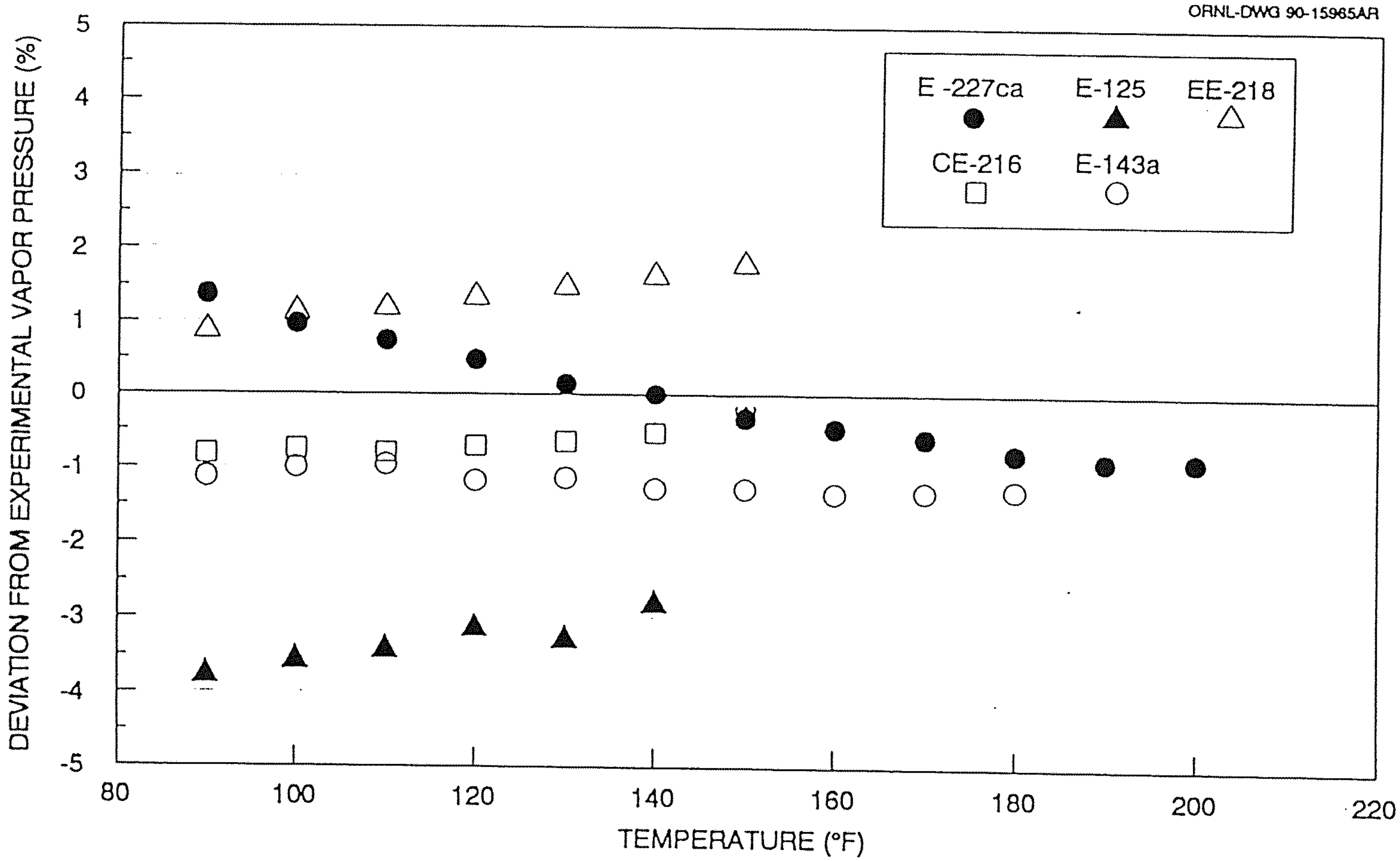


Fig. 2. Deviations of vapor pressures calculated by LKP routines from experimental values—ethers from EPRI/EPA project.

An example showing the sensitivity of the calculated vapor pressure results for E-125 to values used for the acentric factor is shown in Fig. 3. A value for $\omega$ of 0.3350 was calculated using Eq. 1. New vapor pressures were calculated over the range for which experimental data were available using 0.311, 0.300, 0.290, 0.270, and 0.260 as values for $\omega$. Figure 3 shows that the average deviation between calculated and experimental results can be reduced from $-3.7\%$ to $+0.3\%$, depending on the value chosen for $\omega$. In this instance, in which the experimental data are over such a narrow temperature range, and the "slope" of the deviations line becomes sharp at $\omega$ values that give the best least squares fit, the value of 0.311 was chosen as a compromise between an acentric factor that best fits the data and one that is not likely to give large deviations at the temperature extremes. A similar acentric factor fitting analysis was applied to all of the compounds in Figs. 1 and 2 and to any of the chlorine-free alternatives for which experimental PVT data became available.

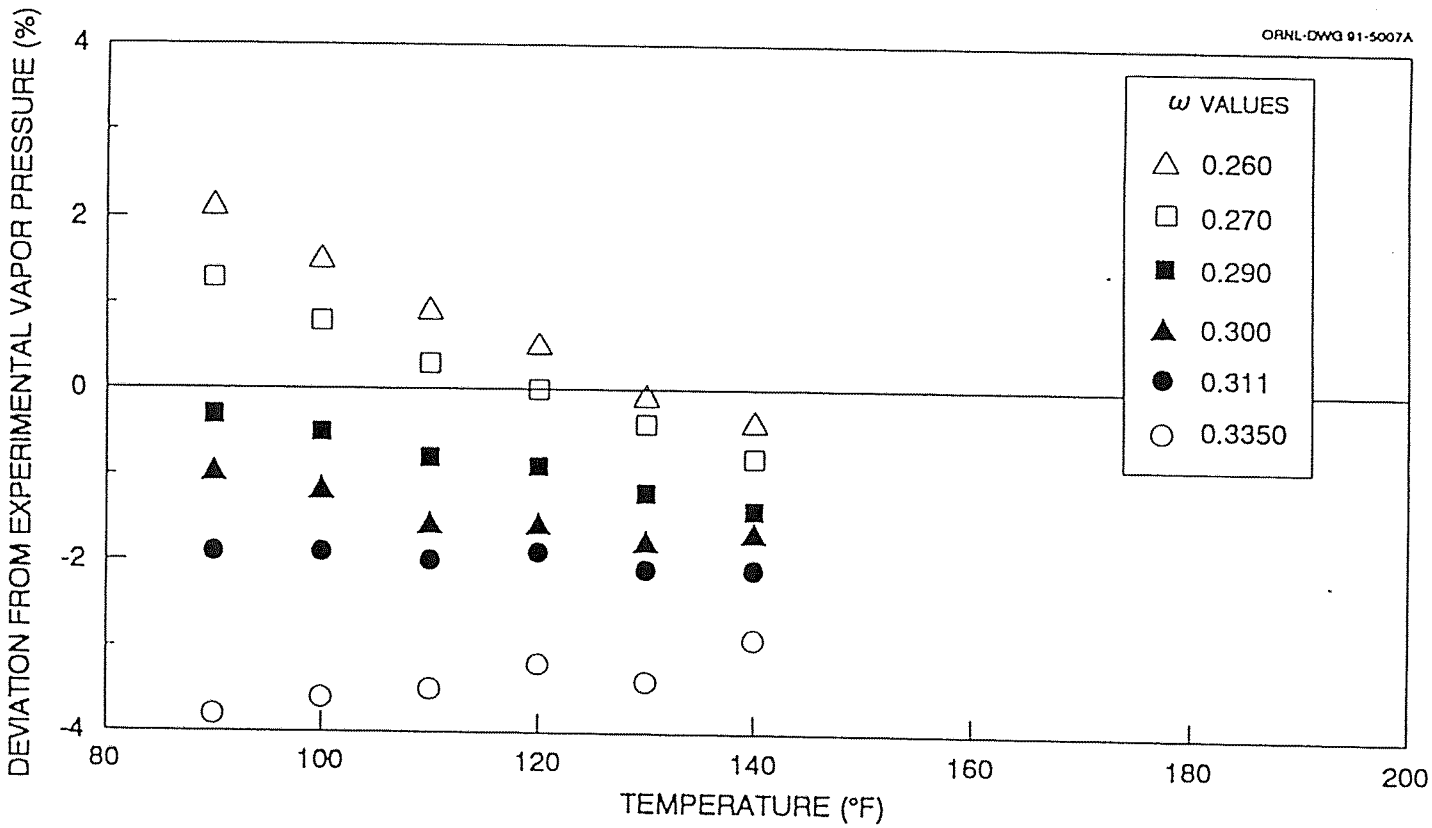


Fig. 3. E-125 vapor pressure deviations by LKP routines as a function of acentric factor $\omega$.

## 4. CHILLER PARAMETERS MODELED

A simple, single-stage chiller model based on saturated refrigerant temperatures in the evaporator and condenser was developed using LKP refrigerant property routines to calculate (1) condenser and evaporator pressures, (2) net refrigerating effect (Btu/lb), (3) isentropic compressor work (equivalent head), (4) ideal isentropic coefficient of performance (COP), (5) mass and volume flow rates per ton of refrigeration, and (6) sonic velocity at suction conditions. All of these values could be readily calculated from the refrigerant property subroutines that were available and that had been used in previous publications to compare the value of alternative refrigerants.[14,15]

When it was required, the model added just enough superheat in the evaporator to prevent formation of a two-phase refrigerant during isentropic compression. The cooling effect of this superheat was added to the net refrigerating effect of the refrigerant and was included in the calculated COP. This was done to keep results from this study comparable with those from previous studies.[16,17] Larger molecules with larger vapor phase heat capacities ($C_p$s), like the propanes and three-carbon ethers in this study, exhibit "wet isentropic compression" more often than simpler one- and two-carbon refrigerants.[18]

Other parameters, specific to the performance of centrifugal chillers, that were modeled in this study were (1) impeller tip speeds, (2) rotational mach numbers, (3) relative efficiencies due to rotational mach numbers, (4) stage efficiencies, (5) impeller revolutions per minute, and (6) impeller pumping capacities. Typical values for compressor head coefficient, tip flow coefficient, mechanical efficiency, impeller diameter, and pumping capacity factor were assumed for these calculations.

The impeller tip speed ($\mu_{TS}$) was calculated using Eq. (2), which requires a value for the energy that must be added to the refrigerant by the compressor (equivalent head) and an assumed value for the compressor head coefficient ($\mu_{HC}$).

where

$E_{stage}$ = stage efficiency,
$E_{mechanical}$ = 0.80 (assumed),
$E_R$ = relative efficiency due to rotational Mach number.

Impeller revolutions per minute and refrigerant pumping capacities can be calculated with an assumed impeller diameter and a pumping capacity constant in Eqs. (5) and (6).

$$N = \frac{\mu_{TS} \times 12 \times 60}{\pi \times d} , \tag{5}$$

where

$N$ = rotational speed (rpm),
$\mu_{TS}$ = impeller tip speed (ft/s),
$\pi$ = pi (3.1416),
$d$ = impeller diameter (in.).

$$Q = 0.18 \times N \times D^3 \ , \tag{6}$$

where

$Q$ = impeller pumping rate ($ft^3/min$),
$N$ = rotational speed (rpm),
$D$ = impeller diameter (ft),
0.18 = pumping capacity constant.

Chiller efficiencies in terms of kilowatts per ton of refrigeration (kW/t) were computed by factoring in all of these centrifugal specific parameters. This measure of efficiency, which is more dependent on the characteristics of turbomachinery than the isentropic COP, could then be compared and contrasted with this COP, which was totally dependent on the thermodynamic properties of the fluid and which assumed ideal, 100% efficient operation.

The effects of transport properties like suction gas viscosity and liquid or vapor thermal conductivities were not modeled because none of these thermophysical properties were available, and their effect on chiller efficiency is much more complicated to evaluate than that of thermodynamic properties.

Cycle conditions simulating a 100°F condenser and 40°F saturated evaporator, which are fairly standard for a centrifugal water chiller, were chosen.[19] No refrigerant subcooling or superheat was added unless superheating was necessary to avoid wet isentropic compression.

A flow chart diagram for this chiller model is shown in Figure 5. The only features that differentiate this code from a simple model, where condenser and evaporator temperatures and pressures are equated with saturated refrigerant conditions at the appropriate heat exchanger temperatures, are a check to see if superheat is needed to avoid wet compression and the chiller specific calculations using equations previously described.

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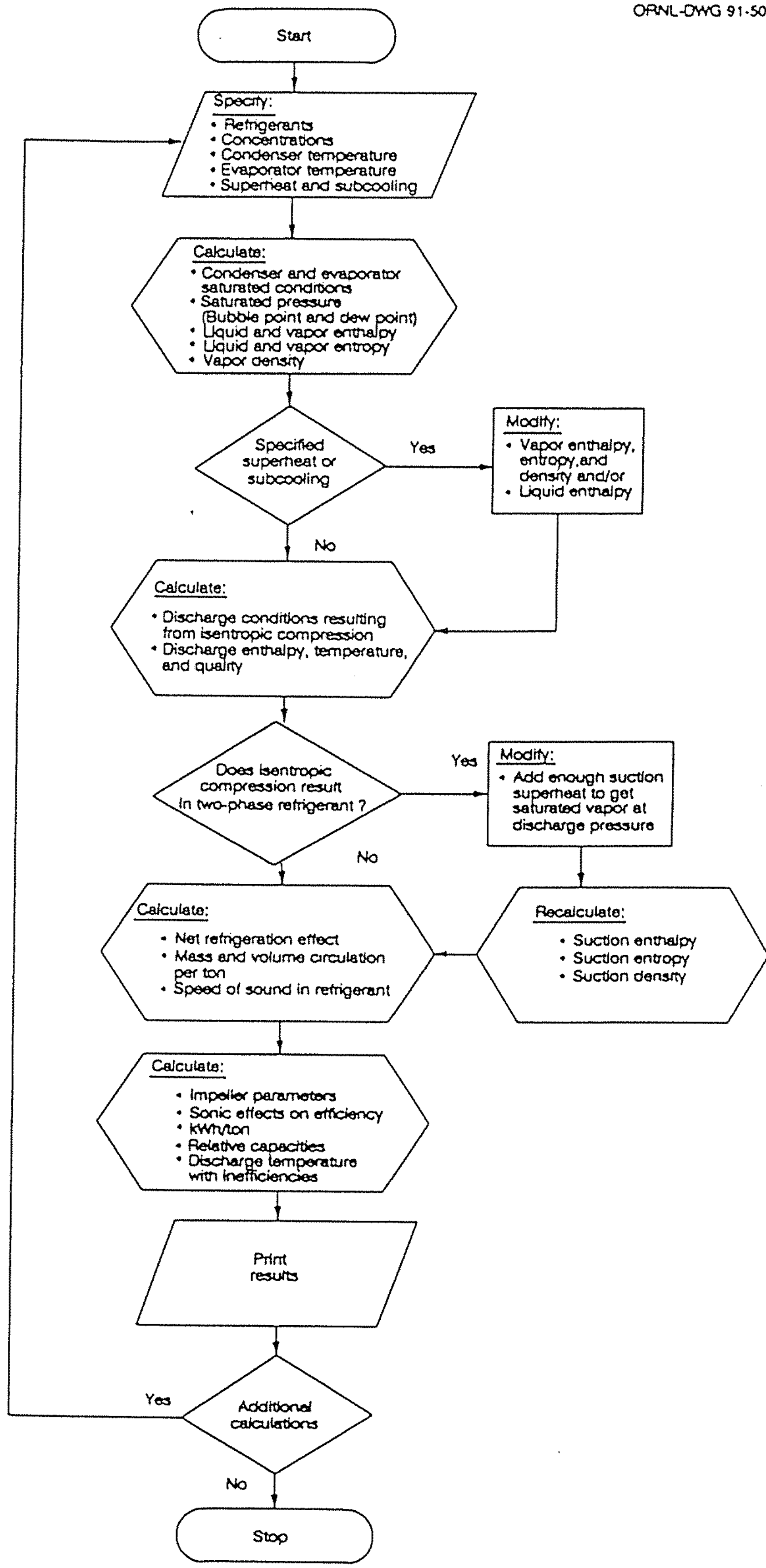


Fig. 5. Pure and NEARM cycle model flow chart for chiller performance.

## 5. PURE REFRIGERANT AND NEARM MODELING RESULTS

Table 3 summarizes the ideal isentropic COP and kW/t performance results for fluorinated alkanes and fluorinated ethers that gave modeled results equal to or better than R-134a performance. More extensive tables for all of the compounds evaluated in this study, which list all of the modeled parameters, are presented in Appendix B. Results for R-11, R-123, R-114, and R-134a are also shown in Table 3 for comparison. Also tabulated are the amount of superheat required over saturated evaporator conditions to avoid wet isentropic compression and rough estimates of atmospheric flammability based on summed bond energies and molar heat capacities.

Only R-152, E-143, and R-143 yield COPs and kW/t values that are comparable to those of R-11. Very little physical property information was found for R-152 and E-143, so results for these two compounds are based primarily on estimated properties. Rat inhalation studies indicate that R-152 is acutely toxic at concentrations greater than 75 ppm.[20] Some experimental performance data is available for R-143.[21] All of these compounds appear to be flammable.

Under these modeling conditions, all of the other compounds in Table 3 show comparable or better performance than R-134a, which is already being used to replace R-12 in gear-driven centrifugal applications. The presence or absence of subcooling and superheat, as well as variations in the transport properties of the fluids, may change their relative rankings. As reported by McLinden, incorporation of liquid line subcooling or liquid-to-suction line heat exchange into this simple model enhances the performance of refrigerants that have more complex molecular structures than the relatively simpler molecules previously used.[18]

Table 3 also suggests the potential benefits of blending two or more refrigerants to make NEARMs in which the superior cycle performance of one component is complimented by the nonflammability of the other(s). Unfortunately, no clearly nonflammable alternatives are available with boiling points near that for E-143, but a nonflammable

Table 3. Modeled performance results of chlorine-free alternatives in chillers—compounds with results comparable to or better than R-134a

(40°F saturated evaporator, 100°F saturated condenser, 0 F° superheat[a]/subcooling)

| Refrigerant | Formula | Normal boiling point (°F) | Modeled COP | Kilowatt per ton | Superheat added (F°) | Flammability indexing |
|---|---|---|---|---|---|---|
| E-254cb | $CHF_2$-O-$CF_2CH_3$ | 97.6 | 7.38 | 0.641 | 7.8 | Flammable[b] |
| E-245cb | $CF_3$-O-$CF_2CH_3$ | 93.3 | 7.33 | 0.643 | 11.7 | Uncertain-to-flammable[b] |
| R-152 | $CH_2FCH_2F$ | 87.2 | 7.60 | 0.615 | 0.0 | Flammable |
| E-143 | $CHF_2$-O-$CH_2F$ | 86.1 | 7.50 | 0.628 | 0.0 | Flammable[b] |
| *R-123* | *$CHCl_2CF_3$* | *81.7* | *7.42* | *0.629* | *2.0* | *Nonflammable* |
| R-245ca | $CHF_2CF_2CH_2F$ | 78.1 | 7.33 | 0.639 | 7.0 | Uncertain |
| *R-11* | *$CCl_3F$* | *74.9* | *7.54* | *0.614* | *0.0* | *Nonflammable* |
| R-245fa | $CF_3CH_2CHF_2$ | 59.5 | 7.26 | 0.642 | 7.6 | Uncertain |
| R-236ea | $CF_3CHFCHF_2$ | 43.7 | 7.14 | 0.650 | 11.7 | Nonflammable |
| R-236ca | $CHF_2CF_2CHF_2$ | 41.0 | 7.11 | 0.651 | 12.1 | Nonflammable |
| R-143 | $CHF_2CH_2F$ | 41.0 | 7.49 | 0.611 | 0.0 | Flammable |
| R-134 | $CHF_2$O-$CHF_2$ | 40.4 | 7.32 | 0.633 | 0.0 | Uncertain[b] |
| *R-114* | *$CClF_2CClF_2$* | *38.5* | *7.12* | *0.645* | *14.2* | *Nonflammable* |
| R-236cb | $CF_3CF_2CH_2F$ | 34.2 | 7.08 | 0.651 | 12.9 | Nonflammable |
| R-236fa | $CF_3CH_2CF_3$ | 30.0 | 7.04 | 0.654 | 12.9 | Nonflammable |
| R-134 | $CHF_2CHF_2$ | −3.5 | 7.17 | 0.635 | 2.6 | Nonflammable |
| R-152a | $CHF_2CH_3$ | −13.0 | 7.17 | 0.633 | 0.0 | Flammable |
| *R-134a* | *$CF_3CH_2F$* | *−15.7* | *6.97* | *0.653* | *0.0* | *Nonflammable* |

[a] Enough superheat added to avoid wet compression; cooling effect of superheat added to COP.

[b] Validity of index for ethers uncertain.

ether (E-134) and several nonflammable propanes have boiling points (and vapor pressures) similar to those of R-143.

Modeled chiller results for some flammable/nonflammable NEARM combinations of these alternatives are shown in Table 4. Several nonflammable NEARMs and NEARMs with uncertain flammability containing E-143, R-143, and E-134 were evaluated. No NEARM with a performance better than that of R-11 was found.

**Table 4. Modeled performance results of chlorine-free NEARM alternatives in chillers—blends with results comparable to or better than R-134a**

(40°F saturated evaporator, 100°F saturated condenser, 0 F° superheat[a]/subcooling)

| NEARM | Composition (mass fractions) | Modeled COP | Kilowatt per ton | Superheat added (F°) | Flammability index |
|---|---|---|---|---|---|
| *R-11* | *1.0* | *7.54* | *0.614* | *0.0* | *Nonflammable* |
| E-143/R-245ea | 0.21/0.79 | 7.32 | 0.642 | 3.5 | Uncertain[b] |
| E-143/R-245ea | 0.45/0.55 | 7.38 | 0.638 | 0.0 | Uncertain[b] |
| R-143/R-236ea | 0.11/0.89 | 7.19 | 0.644 | 8.9 | Nonflammable |
| R-143/R-236ea | 0.48/0.52 | 7.34 | 0.627 | 0.0 | Uncertain |
| R-143/E-134 | 0.20/0.80 | 7.36 | 0.628 | 0.0 | Nonflammable[b] |
| *R-114* | *1.0* | *7.12* | *0.645* | *14.2* | *Nonflammable* |
| R-143/R-236ca | 0.11/0.89 | 7.17 | 0.645 | 9.3 | Nonflammable |
| R-143/R-236ca | 0.48/0.52 | 7.33 | 0.627 | 0.0 | Uncertain |
| R-152a/R-134 | 0.19/0.81 | 7.17 | 0.636 | 0.0 | Uncertain |
| *R-134a* | *1.0* | *6.97* | *0.653* | *0.0* | *Nonflammable* |

[a] Enough superheat added to avoid wet compression; cooling effect of superheat added to COP.
[b] Validity of index for ethers uncertain.

# 6. ESTIMATING FLAMMABILITY

Flammability of alternatives continues to be a concern for refrigerant and equipment suppliers. The issue has technical implications, but much of the uncertainty centers around convenience and liability problems. Engineering controls and industrial safeguards exist for the handling of flammable materials, but they represent compromises not previously required in this industry.

Predictions of flammability for these partially fluorinated compounds were made with an estimating scheme provided by one of the fluorocarbon manufacturers. Although the company was willing to share information presented in this report, they requested that the algorithm not be published because it is still under development, and its validity for predicting flammability for ethers is questionable.

One of the advantages of this method is that a flammability index value is produced on a continuous scale (Fig. 6). This index indicates how flammable or nonflammable a compound is and estimates how effectively one compound will prevent flammability if it is used in a mixture with another.

An analytical approach like this is needed to help identify potentially useful compounds and blends whose precise flammability characteristics can be verified in laboratory tests.

The method, as it is applied in this report, should be further refined for ethers. The E-134 candidate that is being seriously considered as a replacement for R-114 is known to be nonflammable, but Fig. 6 places it in the "uncertain" area between flammable and nonflammable compounds. These results for E-134 may indicate that the method being used is overestimating the flammability of E-254cb, E-245cb, and E-143, of all which can function as substitutes for R-11.

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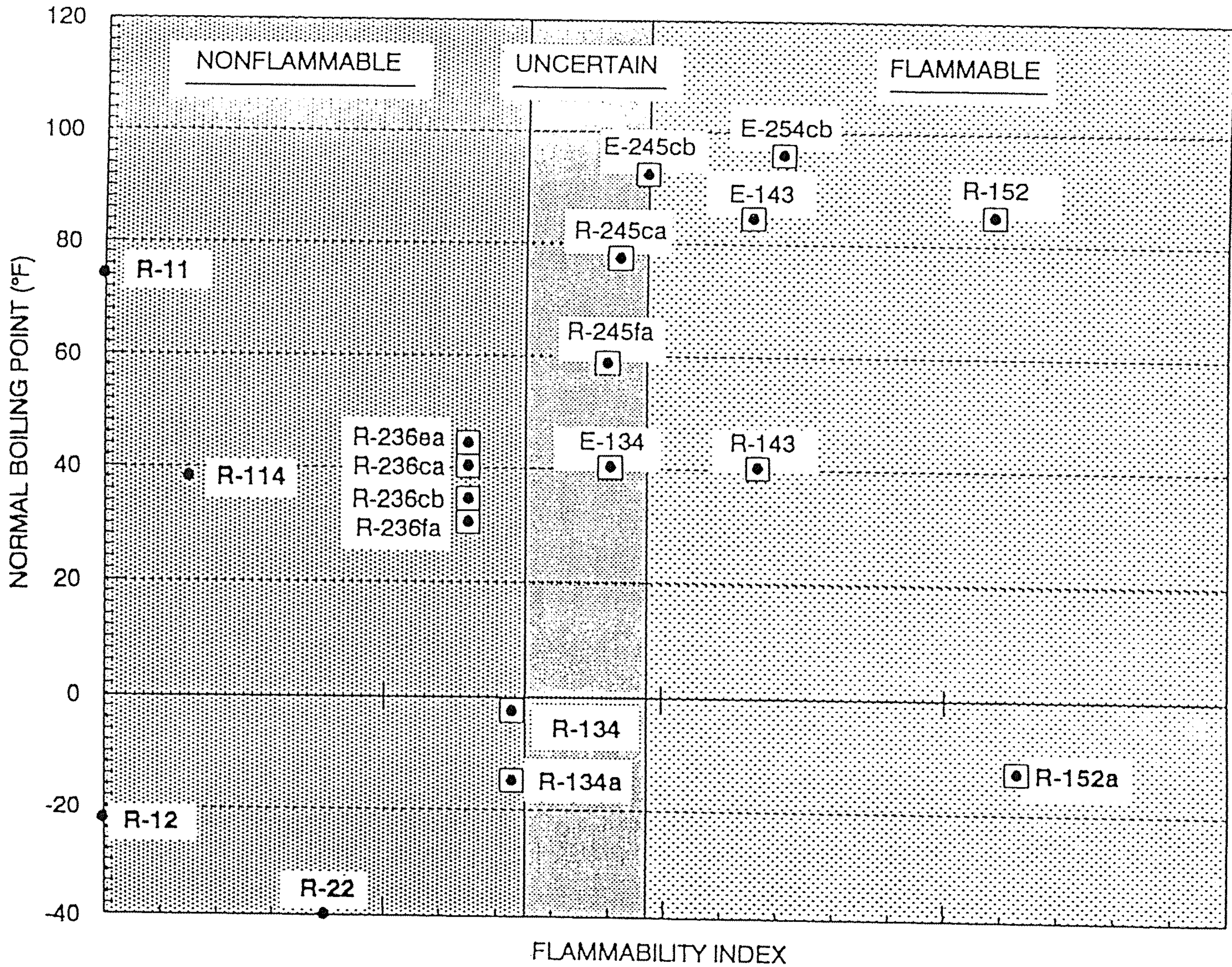


| | |
|---|---|
| R-11 | $CCl_3F$ |
| R-12 | $CCl_2F_2$ |
| R-22 | $CHClF_2$ |
| R-114 | $CClF_2CClF_2$ |
| R-134a | $CF_3\,CH_2F$ |
| R-134 | $CHF_2\,CHF_2$ |
| R-152a | $CHF_2\,CH_3$ |
| R-152 | $CH_2FCH_2F$ |
| R-143 | $CHF_2CH_2F$ |
| R-245ca | $CHF_2\,CF_2\,CH_2F$ |
| R-245fa | $CF_3\,CH_2CHF_2$ |
| R-236ca | $CHF_2\,CF_2\,CHF_2$ |
| R-236cb | $CF_3\,CF_2\,CH_2F$ |
| R-236ea | $CF_3\,CHFCHF_2$ |
| R-236fa | $CF_3\,CH_2\,CF_3$ |
| E-134 | $CHF_2$-O-$CHF_2$ |
| E-143 | $CHF_2$-O-$CH_2F$ |
| E-254cb | $CHF_2$-O-$CF_2\,CH_3$ |
| E-245cb | $CF_3$-O-$CF_2\,CH_3$ |

Fig. 6. Flammability indexing of EPRI/EPA chlorine-free refrigerants.

# 7. CONSIDERATIONS FOR NARMS IN CHILLERS

While NARMs have been suggested as potential energy saving alternatives to CFCs in most air conditioning and refrigeration applications,[22] the performance benefits that can be anticipated with NARM refrigerants depend on the temperature changes experienced by the sensible fluid and on the overall temperature lift of the application.[23]

The heat exchanger efficiency improvements that result from matching the evaporating or condensing temperature profile of an NARM to the sensible temperature change of the source or sink fluid rely on the use of counterflow heat exchangers. Counterflow heat exchange with liquids as the sensible fluid can easily be obtained in tube-in-tube heat exchangers, but a commercially available, efficient, flooded or refrigerant-to-air heat exchanger with a counterflow design has not yet been developed.

The fractional energy savings that result from improving the efficiency of heat exchange are larger if the temperature difference between the heat source and heat sink in the vapor compression cycle (temperature lift) is smaller. This statement applies more to cooling mode air conditioning applications than to refrigeration or heat pumping. Chillers offer a unique opportunity for NARMs because the sensible fluid for both heat exchangers is usually water, and the temperature change of the water through the heat exchangers is carefully controlled. In addition, the temperature difference between the evaporator and condenser in a chiller is usually relatively small.

Unfortunately, the shell-and-tube heat exchangers on most big chillers currently produced incorporate very little counterflow design. NARMs in flooded evaporators would very likely experience fractional distillation.

The relative performance advantage of NARMs over pure refrigerants can also be improved by increasing the temperature change of the sensible fluid through the heat exchangers. The isothermal evaporation and condensation of pure refrigerants are better suited than those of NARMs to minimal temperature changes in the sensible fluid. This suggests that if chillers are redesigned for use with NARMs, there may be energy saving advantages that can be obtained by forcing larger temperature changes in the chilled or cooling tower water circuits. "Superchilled water" can be produced in a

NARM chiller, with small losses in the efficiency of the compressor cycle. These losses may be compensated for by energy and material savings resulting from lower flow rates, smaller fans or pumps, and smaller building duct sizes.

Minimizing the temperature difference between refrigerant and sink or source streams in a heat exchanger means that the *size* of the heat exchanger has to be increased to maintain the same capacity.[23] In situations where the equipment is operating at partial loads, however, NARMs can make better use of oversized heat exchangers than pure refrigerants.[24]

# 8. CYCLE PROGRAM FOR NARMS

Using the Lorenz equation [Eq. (7)], estimates of NARM refrigerant glides in a chiller evaporator can be made from the normal boiling points of mixed components.[25]

$$\Delta T_{Max} = 0.04\,(\Delta t_{bp})^{1.616}\ , \tag{7}$$

where

$\Delta T_{Max}$ = maximum temperature glide for a 50/50 mole% mixture,
$\Delta t_{bp}$ = normal boiling point difference.

Predicted glides for 50/50 mole% mixtures of refrigerants modeled in this study are shown in Fig. 7.

Assuming that maximum glides of 15 F° to 30 F° will be most suitable for chiller applications, refrigerant pairs forming NARMs with this glide range can be selected. These binary pairs are indicated by the cross-hatched regions of Fig. 7. Potentially nonflammable NARM pairs made up of a flammable/nonflammable or nonflammable/nonflammable combination are also indicated. These predictions were based on the flammability index values shown in Fig. 6.

A 15 F° to 30 F° refrigerant glide criterion for refrigerant mixtures is larger than the 10 F° sensible fluid glide usually used for chiller modeling. The larger NARM glide was chosen because blends other than 50/50 mole% will show smaller glides more in line with water-side temperature changes and because cycle modeling comparisons with operating conditions more favorable for NARM refrigerants were anticipated.

A new computer model based on heat exchanger log mean temperature differences (LMTDs) or average effective temperature differences (AETDs) had to be developed for comparing chiller performance with NARMs to results previously generated for single component and NEARM refrigerants. The earlier model assumed saturated evaporator and condenser temperatures that are not relevant to a refrigerant whose temperature changes as it evaporates or condenses in the heat exchanger.

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| | E-245cb | R152 | E-143 | R-245ca | R-245fa | R-236ea | R-236ca | R-143 | E-134 | R-236cb | R-236fa | R-134 | R-152a | R-134a |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| E-254cb | 0.3 | 1.2 | 1.4 | 3.4 | 9.9 | 17.5 | 18.9 | 18.9 | 19.2 | 22.7 | 25.2 | 48.3 | 55.8 | 58.1 |
| E-245cb | | 0.5 | 0.7 | 2.3 | 8.2 | 16.3 | 16.7 | 16.7 | 17.0 | 20.4 | 22.7 | 45.1 | 52.5 | 54.7 |
| R152 | | | 0 | 1.0 | 6.0 | 12.4 | 13.7 | 13.7 | 13.9 | 17.1 | 19.3 | 40.6 | 47.7 | 49.8 |
| E-143 | | | | 0.8 | 5.6 | 11.9 | 13.2 | 13.2 | 13.4 | 16.5 | 18.7 | 39.8 | 46.9 | 49.0 |
| R-245ca | | | | | 3.1 | 8.5 | 9.6 | 9.6 | 9.8 | 12.6 | 14.5 | 34.2 | 40.9 | 42.8 |
| R-245fa | | | | | | 2.4 | 3.1 | 3.1 | 3.3 | 5.2 | 6.6 | 22.5 | 28.3 | 30.0 |
| R-236ea | | | | | | | 0.1 | 0.1 | 0.2 | 1.1 | 1.9 | 14.1 | 19.0 | 20.5 |
| R-236ca | | | | | | | | 0 | 0 | 0.6 | 1.3 | 12.8 | 17.6 | 19.0 |
| R-143 | | | | | | | | | 0 | 0.6 | 1.3 | 12.8 | 17.6 | 19.0 |
| E-134 | | | | | | | | | | 0.5 | 1.2 | 12.6 | 17.3 | 18.7 |
| R-236cb | | | | | | | | | | | 0.3 | 9.8 | 14.1 | 15.4 |
| R-236fa | | | | | | | | | | | | 8.1 | 12.2 | 13.4 |
| R-134 | | | | | | | | | | | | | 1.1 | 1.6 |
| R-152a | | | | | | | | | | | | | | 0.1 |

POTENTIAL NARMS

UNLIKELY NARMS

**POTENTIALLY NONFLAMMABLE BINARY NARM PAIRS**

| | |
|---|---|
| E-254cb + R-236ea | E-245cb + R-236ea |
| E-254cb + R-236ca | E-245cb + R-236ca |
| E-254cb + E-134 | E-245cb + E-134 |
| E-254cb + R-236cb | E-245cb + R-236cb |
| E-254cb + R-236fa | E-254cb + R-236fa |
| R-152 + R-236cb | R-245fa + R-134 |
| R-152 + R-236fa | R-245fa + R-152a |
| R-236ea + R-152a | R-245fa + R-134a |
| R-236ca + R-152a | E-143 + R-236cb |
| R-143 + R134a | E-143 + R-236fa |
| E-134 + R-152a | R-236ca + R-134a |
| E-134 + R-134a | R-236ca + R-134a |
| | R-236cb + R-134a |

**FLAMMABLE NARM PAIRS**

| | |
|---|---|
| E-254cb + R-143 | R-143 + R-152a |
| E-245cb + R-143 | |

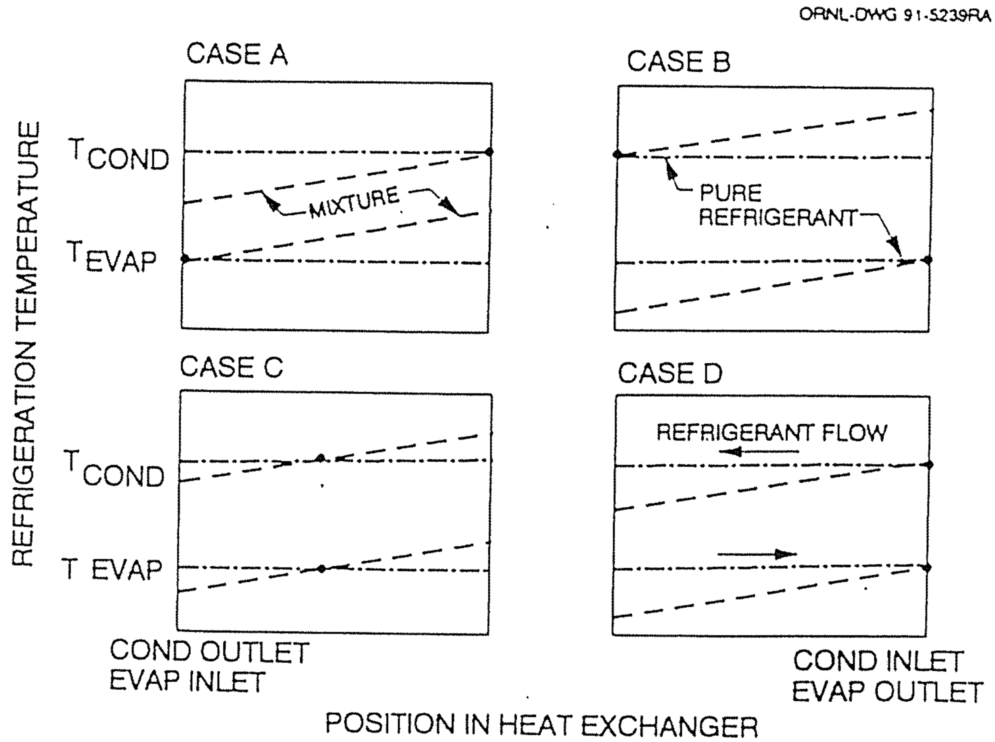
Fig. 7. Predicted evaporator glides (°F) for 50/50 binary mixtures of EPRI/EPA refrigerants.

McLinden and Radermacher have shown that fair comparisons between mixture and pure component performances can be made if temperature changes of the refrigerant and the heat transfer fluid in the application being modeled are considered.[26]

The rationale for this can be explained by the drawings in Fig. 8. Mixture and pure refrigerant evaporator and condenser temperatures are shown for four different cases, A, B, C, and D. In all of these, the pure refrigerant temperatures and temperature lift for the simulated cycle remain the same. The dew point of the mixture is equated to the pure refrigerant saturated condenser temperature, and the mixture's bubble point is equated to the saturated evaporator temperature in case A. Case B assumes that the bubble point and dew point of the mixture simulate the saturated condenser and evaporator temperatures, respectively, of the pure refrigerant. Case C uses the glide midpoint for the mixtures to approximate the saturated evaporator and condenser conditions, and Case D assumes that the mixture's dew point is used for both heat exchanger temperatures. The effects of these assumptions on system COP are shown as a function of NARM concentration in the lower drawing in Fig. 8. These variations are all predictable in terms of the actual temperature lifts that result with the assumptions used in each case.

As a result of this analysis, McLinden and Radermacher[26] recommend that comparisons between pure and mixed refrigerant cycle performances be made by specifying the entering and leaving temperatures of the secondary heat transfer fluid and the total heat exchanger area per unit of capacity. The CYCLE 7 program from NIST attempted to accomplish this.

For this study, the CYCLE 11 program developed by Domanski and McLinden[27] at NIST was selected as a starting point. This program incorporated algorithms to account for the nonlinearity of refrigerant-side temperature changes in the heat exchangers and an average effective temperature difference (AETD) between the refrigerant and secondary heat transfer fluid rather than a log mean temperature difference (LMTD) to specify heat exchanger performance. Routines that Rice developed for the CYCLE Z refrigerator-freezer model[28] were added to the NIST model to control condenser and evaporator sizes per unit of refrigeration output. These routines have the effect of equalizing the amount of heat transferred in the evaporator and condenser, thus keeping results from runs with different fluids and different operating conditions more comparable.



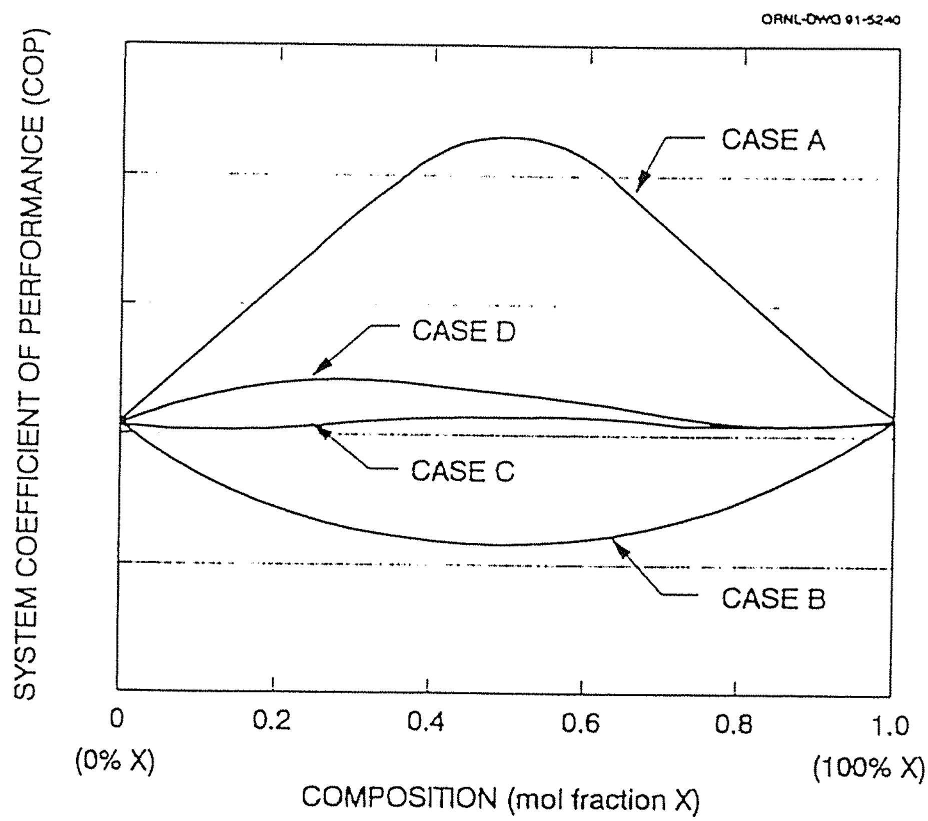




Fig. 8. Pure refrigerant vs. NARM cycle performance comparisons.

Some of the CYCLE 11 input parameters, such as the AETD in the heat exchangers, were adjusted so that this NARM model gave the same chiller performance results for R-11 as those obtained with the simpler, saturated heat exchanger temperature model used earlier to evaluate pure compounds and NEARMs.

A branching routine was added to this model that automatically adds evaporator superheat if isentropic compression of vapor from the evaporator results in wet compression at the specified operating conditions. This is a concession to practical operation of centrifugal chillers that was employed in the previous simulations. However, it penalizes the performance of most of the NARM pairs that come out of this study because superheat makes the evaporator less efficient and often causes evaporator pinching when an AETD or LMTD is used to specify the heat exchangers.

All of the combinations listed as potential candidates in Fig. 7 were evaluated with the NARM chiller model described above. Weight percentage concentrations were varied from 10% to 100%. Several potentially nonflammable NARM combinations that gave modeled performance better than R-11 performance (COP and kW/t) were indicated. It is important to note that the model assumes heat exchangers not typical of those on currently manufactured large commercial chillers.

Adding evaporator superheat to avoid two-phase isentropic compression caused pinching in the evaporator for many of the runs in which condenser and chilled water glides were held at 10 F°. A fairer way to make comparisons between these fluids thus was needed. Neglecting the wet compression or factoring in a compressor inefficiency that results in corresponding superheating of the discharge gas were both considered, but a method of adding required superheat analogous to that used in refrigerator-freezers and commercial refrigeration was eventually used. Superheat, if needed to avoid wet compression, was added in the form of suction-to-liquid line heat exchange. In this manner, evaporator pinching was avoided, and conditions beneficial to NARMs and these larger refrigerant molecules were attained. Results from the modeling calculations employing this suction-to-liquid line heat exchange are given in Table 5 for the following conditions:

- condenser water entering = 85°F,
  leaving = 95°F;
- condenser average effective temperature difference (AETD) = 9.1°F;

**Table 5. NARM pairs evaluated for chiller application**

(10°F glides, chilled water, cooling water)

| NARM pair | Concentration maximum COP (mass fraction) | COP maximum |
|---|---|---|
| R-236ea/E-254cb | 0.40/0.60 | 7.68 |
| R-236ca/E-254cb | 0.40/0.60 | 7.68 |
| R-143/E-254cb | 0.70/0.30 | 7.81[a] |
| E-134/E-254cb | 0.70/0.30 | 7.72 |
| R-236cb/E-254cb | 0.20/0.80 | 7.70 |
| R-236fa/E-254cb | 0.40/0.60 | 7.73 |
| R-236ea/E-245cb | 0.30/0.70 | 7.74 |
| R-236ca/E-245cb | 0.20/0.80 | 7.72 |
| R-143/E-245cb | 0.60/0.40 | 7.82[a] |
| E-134/E-245cb | 0.20/0.80 | 7.76[a] |
| R-236cb/E-245cb | 0.10/0.90 | 7.72 |
| R-236fa/E-245cb | 0.10/0.90 | 7.71 |
| R-236cb/R-152 | 0.30/0.70 | 7.89[a] |
| R-236fa/R-152 | 0.20/0.80 | 7.86 |
| R-236cb/E-143 | 0.20/0.80 | 7.81 |
| R-236fa/E-143 | 0.20/0.80 | 7.83[a] |
| R-134/R-245fa | 0.10/0.90 | 7.70 |
| R-152a/R-245fa | 0.80/0.20 | 7.70 |
| R-134a/R-245fa | 0.10/0.90 | 7.69 |
| R-152a/R-236ea | 0.40/0.60 | 7.61 |
| R-134a/R-236ea | 0.20/0.80 | 7.49 |
| R-152a/R-236ca | 0.60/0.40 | 7.66 |
| R-152a/R-143 | 0.20/0.80 | 7.83[a] |
| R-152a/E-134 | 0.50/0.50 | 7.73[a] |
| R-134a/E-134 | 0.10/0.90 | 7.68 |
| R-134a/R-236cb | 0.20/0.80 | 7.53 |

[a] Selected for additional evaluations.

- evaporator water entering = 55°F,
  leaving = 45°F;
- evaporator average effective temperature difference (AETD) = 9.1°F;
- isenthalpic expansion; and
- 100% isentropic/polytropic efficiency.

Concentrations given in Table 5 are those mass fractions that gave the maximum COP. Only marginal gains in COP over R-11 (approximately 4–5%) were obtained with NARMs at this condition.

Several pairs that gave higher COP results were modeled using the same conditions but at concentrations that indicate a nonflammable combination (Table 6). In most cases a compromise of performance is required to obtain a nonflammable mixture. This is especially true in the example with R-152a as one of the NARM components.

Cycle simulation runs with 20 F° water-side glides were performed by changing the condenser water entering and leaving temperatures from 85°F and 95°F to 85°F and 105°F, respectively, and the chilled water temperatures from 55°F and 45°F to 55°F and 35°F, respectively. These runs caused the COP of R-11 to drop from 7.582 to 5.812 and resulted in no NARM COPs greater than 6.50. Most of the pinched evaporator problems resulting from added evaporator superheat were removed with this 20 F° glide, however. The larger temperature lift degrades cycle efficiency, but the *relative performance* of NARMs to pure refrigerants improves for the reasons previously indicated. Comparisons between R-11 and R-236ea/E-245cb NARM performances at different entering and leaving water conditions are tabulated in Table 7.

These results clearly illustrate that superheat for NARMs must be added outside of the evaporator and that suction-to-liquid line heat exchange may be necessary to obtain NARM performance comparable to R-11 performance. Data in Table 7 also show the rapid deterioration of pure refrigerant performance relative to NARMs if larger secondary fluid temperature changes are specified for the refrigeration cycle. Increasing the temperature lift has a predictable effect on both NARM and pure refrigerant performances.

## Table 6. Selected NARM pairs flammability mitigation by dilution

(10°F glides simulating 40°F evaporating/100°F condensing)

| NARM pair | Concentration for maximum COP (% mass) | Maximum COP | Concentration for uncertain flammability status and COP at that concentration[1] | |
|---|---|---|---|---|
| | | | Concentration (% Mass) | COP |
| E-134/E-245cb | 20/80 | 7.76 | 20/80 | 7.76 |
| R-236cb/R-152 | 30/70 | 7.89 | 82/18 | 7.74 |
| R-236fa/E-143 | 20/80 | 7.83 | 46/54 | 7.68 |
| R-152a/E-134 | 50/50 | 7.73 | 4/96 | 7.56 |
| R-152a/R-143 | 20/80 | 7.83 | No nonflammable combination possible | |
| R-143/E-254cb | 70/30 | 7.81 | No nonflammable combination possible | |
| R-143/E-245cb | 60/40 | 7.82 | No nonflammable combination possible | |
| CFC Comparisons | | | | |
| R-11 | — | 7.54 | — | 7.54 |
| R-123 | — | 7.42 | — | 7.42 |

[a] Validity of flammability index for ethers uncertain.

Table 7. NARM (R-236ea/E-245cb) performance in chiller applications—superheat/glide variations

| | NARM | R-11 | NARM | NARM | NARM | R-11 | NARM | R-11 | NARM | R-11 |
|---|---|---|---|---|---|---|---|---|---|---|
| Concentration for maximum COP (% mass) | 20/80 | 100 | 10/90 | 30/70 | 40/60 | 100 | 80/20 | 100 | 80/20 | 100 |
| Condenser | | | | | | | | | | |
| Entering (°F) | 85 | 85 | 85 | 85 | 85 | 85 | 80 | 80 | 80 | 80 |
| Leaving (°F) | 95 | 95 | 95 | 95 | 105 | 105 | 100 | 100 | 100 | 100 |
| Evaporator | | | | | | | | | | |
| Entering (°F) | 55 | 55 | 55 | 55 | 55 | 55 | 60 | 60 | 60 | 60 |
| Leaving (°F) | 45 | 45 | 45 | 45 | 35 | 35 | 40 | 40 | 35 | 35 |
| COP | 7.49 | 7.54 | 7.24 | 7.74 | 5.98 | 5.81 | 7.49 | 6.92 | 6.94 | 6.45 |
| Kilowatts/ton | 0.626 | 0.614 | 0.651 | 0.604 | 0.727 | 0.825 | 0.625 | 0.677 | 0.685 | 0.733 |
| Other | No superheat | | Evaporator superheat | Suction/liquid heat exchange | Evaporator superheat | | Evaporator superheat | | Evaporator superheat | |
| | Wet compression<br>Uncertain flammability | | Evaporator pinch | | Large lift | | Smaller lift | | Super chilled water | |

The effects of no superheat or various methods of adding suction superheat on the ideal cycle COPs of several of these chlorine-free alternatives and of an NEARM and an NARM formed from these alternatives are shown in Fig. 9. Values for R-11 are given for reference, and four cases are illustrated for each refrigerant. The first condition, indicated by the darkly shaded bar, is a modeling run in which no superheat was added to cycle runs. No effect is seen on the R-11 results because this is the only refrigerant for which isentropic compression of the saturated vapor from evaporator pressures to condenser pressures does not result in a two-phase fluid.
When enough evaporator superheat is added to avoid wet compression, and the corresponding enthalpy change is added to the refrigerating effect of the refrigerant (and to the COP), results indicated by the second bar on Fig. 9 are obtained. Adding this superheat in the evaporator for the NARM run in which AETDs or LMTDs are specified for the heat exchanges results in a pinched evaporator.

If suction superheat and corresponding liquid line subcooling are added by a suction-to-liquid line heat exchanger, COP results indicated by the third bar are obtained for each of these alternatives. No change is shown for R-11 because it did not require superheat to avoid wet isentropic compression.

The effects of 10 F° of superheat over and above the amount required for dry compression (and the equivalent amount of liquid line subcooling) on *all* of the refrigerants in Fig. 9, including R-11, is shown by the last bar. The COP increase indicated by this cross-hatched bar shows the relative incremental increases in refrigerant performance associated with the superheat and subcooling corresponding to a 10 F° increase in suction gas temperature. Molecules with larger vapor and liquid phase heat capacities and the NARMs benefit the most from this intercycle heat exchange.

A summary of the effects of parameter variations carried out in this NARM chiller work is given below:

- NARM composition affects refrigerant-side glides, which relate directly to cycle performance.

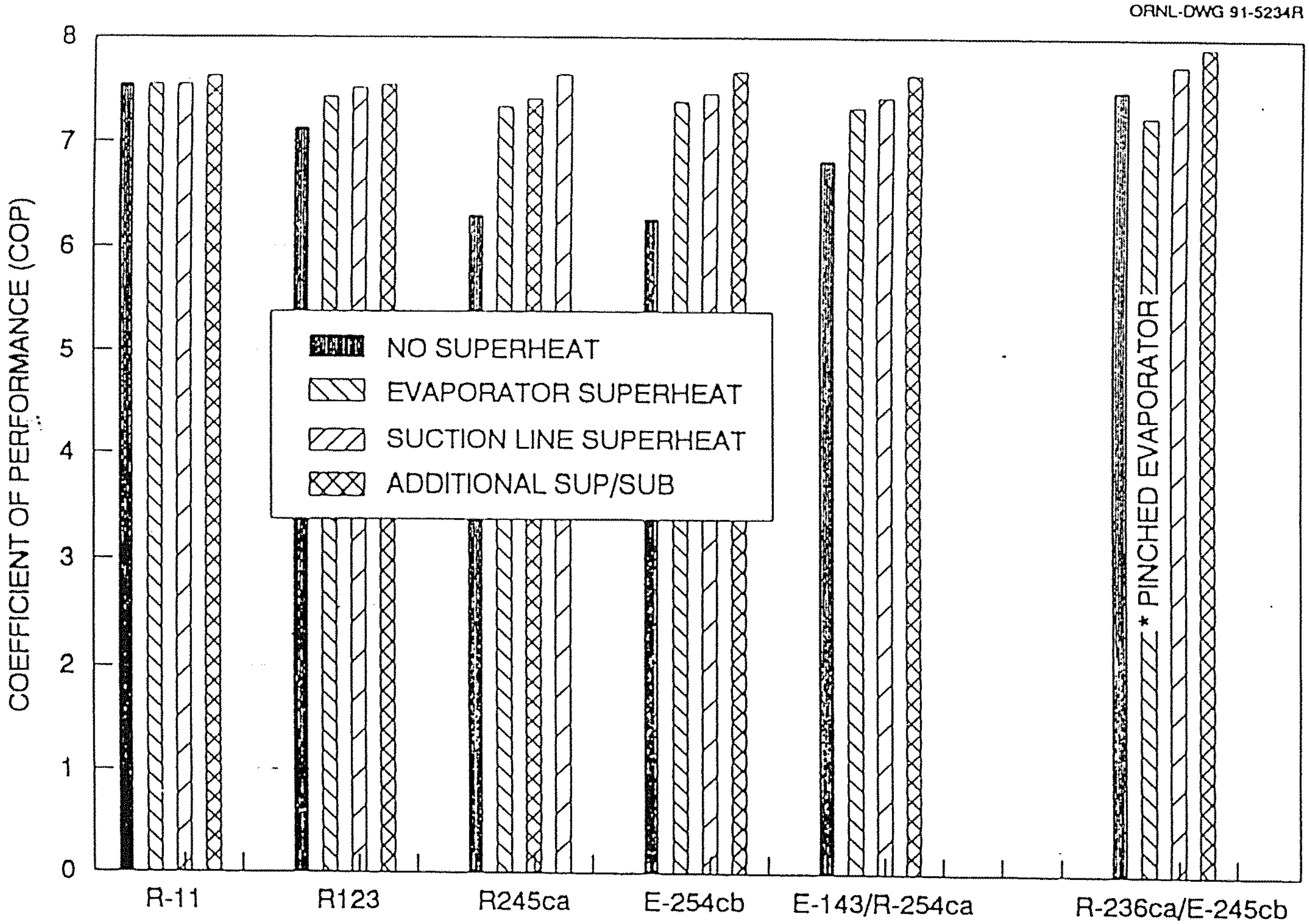



Fig. 9. Effects of superheat/subcooling on alternative refrigerants, NEARM, and NARM cycle performance.

- Compositions can be used to control flammability, but performance usually has to be degraded to obtain a nonflammable combination.

- NARMs perform better than pure refrigerants, with larger differences between secondary fluid entering and leaving temperatures.

- Both NARMs and pure refrigerants are penalized by higher cycle temperature lifts. NARMs rapidly lose any fractional advantage over pure refrigerants if the lift is increased.

- Evaporator superheat at modeled conditions specifying fixed secondary fluid inlet and outlet conditions causes evaporator pinching.

- If superheat is added outside the evaporator in the form of suction-to-liquid line subcooling, it benefits NARMs and refrigerants with higher heat capacities more than it benefits R-11.

# 9. CONCLUSIONS AND RECOMMENDATIONS

Centrifugal chiller modeling work on several fluorinated two- or three-carbon alkanes and fluorinated ethers indicates that they are potential chlorine-free alternatives for R-11 and R-123. The use of fluorinated ethers as CFC alternatives should be more thoroughly investigated.

Several of the newer compounds that give the best modeled performance are flammable. Blends of flammable and nonflammable refrigerants with similar vapor pressures can be used to make a nonflammable NEARM with a cycle performance that is intermediate between components of the blend.

The flammability of fluorinated ethers should be more thoroughly characterized and evaluated so that a method for predicting the combustibility of pure compounds and blends of compounds can be developed using molecular structures and known physical properties.

Problems with the use of flammable refrigerants in various applications have to be objectively evaluated against potential gains in energy efficiency.

Modifications of the basic chiller cycle, such as deliberately adding subcooling or liquid-to-suction line heat exchange, will benefit the performance of refrigerants with more complex molecular structures and larger molecular heat capacities than currently used refrigerants. These modifications will involve significant changes to the design and substantial increases in the complexity of chillers because of the large volumes of refrigerant that must be circulated to achieve acceptable cooling capacities.

The low temperature lift typical of a chiller cycle is an ideal application for NARMs.

Substantial changes to the heat exchanger design and plumbing will be needed to obtain counterflow conditions. Different entering and leaving temperatures for the cooling tower and chilled water can be used to save energy and material in the external circuits without extensive degradation of the cycle efficiency. Liquid line subcooling has a much more beneficial impact on the cycle performance of NARMs than it does on pure refrigerant performance.

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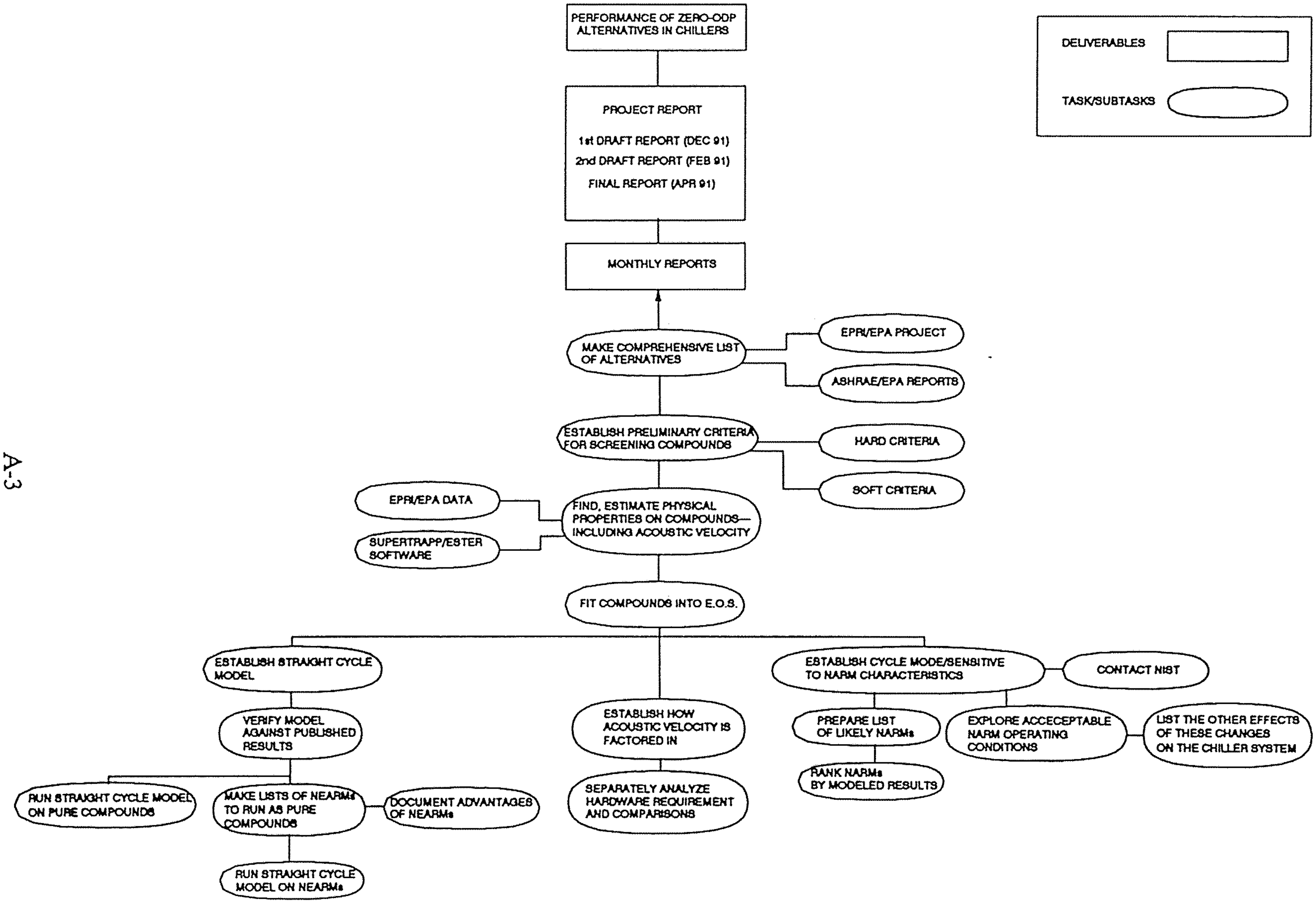
# APPENDIX A

# PROJECT OUTLINE

ORNL-DWG 90-15824R

# PERFORMANCE OF CHLORINE-FREE ALTERNATIVES IN CHILLERS: PROJECT PLAN



# APPENDIX B

# TABULATED PERFORMANCE OF CHLORINE-FREE REFRIGERANTS AND NEARMS IN CENTRIFUGAL CHILLERS

Table B-1
COMMERCIALLY AVAILABLE AND NEAR TERM ALTERNATIVES
(based on a cycle at 40°F evaporation/100°F condensation,
some superheat[a] /no liquid subcooling)

| Refrigerant | R-11 | R-134a | R-12 | R-22 | R-113 | R-114 | R-123 | R-142b | R-152a | R-141b | PROPANE R-290 | AMMONIA R-717 |
|---|---|---|---|---|---|---|---|---|---|---|---|---|
| Evaporator Pressure (psia) | 7.03 | 50.25 | 51.60 | 83.01 | 2.70 | 15.15 | 6.35 | 25.10 | 45.24 | 5.10 | 78.90 | 74.90 |
| Condenser Pressure (psia) | 23.52 | 140.57 | 131.35 | 210.26 | 10.51 | 45.84 | 22.41 | 72.06 | 125.34 | 18.01 | 190.01 | 215.85 |
| Pressure Ratio | 3.35 | 2.80 | 2.54 | 2.53 | 3.89 | 3.03 | 3.53 | 2.87 | 2.77 | 3.53 | 2.41 | 2.88 |
| Net Refrigerating Effect (Btu/lb) | 68.23 | 64.89 | 50.25 | 69.50 | 54.70 | 45.80 | 62.16 | 75.04 | 108.89 | 83.75 | 118.66 | 472.10 |
| Isentropic Compressor Work (Btu/lb) | 9.04 | 9.30 | 7.13 | 9.94 | 8.58 | 6.43 | 8.37 | 10.16 | 14.82 | 11.09 | 17.55 | 65.52 |
| Equivalent Isentropic Head (ft·lb/lb) | 7029 | 7234 | 5541 | 7728 | 6676 | 5002 | 6513 | 7904 | 11528 | 8623 | 13648 | 50955 |
| Isentropic COP | 7.545 | 6.972 | 7.049 | 6.990 | 6.368 | 7.117 | 7.419 | 7.379 | 7.348 | 7.549 | 6.757 | 7.20 |
| Mass Flow Rate (lb/min/ton) | 2.93 | 3.08 | 3.98 | 2.88 | 3.66 | 4.37 | 3.27 | 2.66 | 1.84 | 2.39 | 1.69 | 0.42 |
| Suction Vapor Density (ft³/lb) | 5.44 | 0.95 | 0.78 | 0.66 | 10.46 | 2.04 | 5.43 | 2.01 | 1.66 | 8.83 | 1.34 | 3.95 |
| Suction Flow Rate (cfm/ton) | 15.94 | 2.94 | 3.09 | 1.91 | 38.23 | 8.91 | 17.46 | 5.35 | 3.05 | 21.10 | 2.27 | 1.67 |
| Impeller Tip Speed (with 0.59 Head Coefficient) (ft/s) | 619.1 | 628.1 | 549.7 | 649.1 | 603.3 | 522.3 | 595.9 | 656.5 | 792.8 | 685.7 | 862.7 | 1667 |
| Refrigerant Sonic Velocity (ft/s) | 443.2 | 484.3 | 447.2 | 536.2 | 374.1 | 386.3 | 414.5 | 501.9 | 620.9 | 478.7 | 722.2 | 1320.0 |
| Rotational Mach Number | 1.397 | 1.297 | 1.229 | 1.211 | 1.613 | 1.352 | 1.438 | 1.308 | 1.272 | 1.432 | 1.197 | 1.263 |
| Relative Efficiency Due to Mach Number | 0.9486 | 0.9643 | 0.9739 | 0.9763 | 0.9072 | 0.9559 | 0.9415 | 0.9627 | 0.9672 | 0.9425 | 0.9784 | 0.9692 |
| Stage Efficiency (Assumed 80% Mechanical Efficiency) | 0.7589 | 0.7715 | 0.7791 | 0.7811 | 0.7258 | 0.7647 | 0.7532 | 0.7702 | 0.7738 | 0.7540 | 0.7827 | 0.7754 |
| Discharge Temperature (°F) | 112.4 | 105.7 | 109.8 | 128.31 | 100.0 | 100.0 | 100.0 | 104.3 | 118.5 | 106.1 | 108.7 | 214.9 |
| Kilowatts/ton | 0.6136 | 0.6532 | 0.6396 | 0.6434 | 0.7601 | 0.6455 | 0.6286 | 0.6181 | 0.6178 | 0.6172 | 0.6642 | 0.6291 |
| RPM for 20-in. Impeller | 7094 | 7197 | 6299 | 7439 | 6914 | 5985 | 6829 | 7522 | 9085 | 7858 | 9885 | 19100 |
| Tons for 20-in. Impeller | 371.0 | 2038.6 | 1696.6 | 3249.7 | 150.7 | 559.9 | 325.9 | 1171.2 | 2480.5 | 310.4 | 3635.6 | 9510.3 |
| Tonnage Compared with R-11 (%) | 100.0 | 549.5 | 457.3 | 875.9 | 40.6 | 150.9 | 87.8 | 315.7 | 668.6 | 83.7 | 980.0 | 2563.4 |
| Tonnage Compared with R-134a (%) | 18.2 | 100.0 | 83.2 | 159.4 | 7.4 | 27.5 | 16.0 | 57.5 | 121.7 | 15.2 | 178.3 | 475.0 |

[a] Enough suction superheat added to avoid "wet compression" where applicable; cooling effect of superheating applied to performance and COP.

Table B-2
COMMERCIALLY AVAILABLE AND NEAR TERM ALTERNATIVES
(based on a cycle at 40°F evaporation/100°F condensation,
no superheat [a]/no liquid subcooling)

| Refrigerant | R-11 | R-134a | R-12 | R-22 | R-113 | R-114 | R-123 | R-142b | R-152a | R-141b | PROPANE R-290 | AMMONIA R-717 |
|---|---|---|---|---|---|---|---|---|---|---|---|---|
| Evaporator Pressure (psia) | 7.03 | 50.25 | 51.60 | 83.01 | 2.70 | 15.15 | 6.35 | 25.10 | 45.24 | 5.10 | 78.90 | 74.90 |
| Condenser Pressure (psia) | 23.52 | 140.57 | 131.35 | 210.26 | 10.51 | 45.84 | 22.41 | 72.06 | 125.34 | 18.01 | 190.01 | 215.85 |
| Pressure Ratio | 3.35 | 2.80 | 2.54 | 2.53 | 3.89 | 3.03 | 3.53 | 2.87 | 2.77 | 3.53 | 2.41 | 2.88 |
| Net Refrigerating Effect (Btu/lb) | 68.23 | 64.89 | 50.25 | 69.50 | 54.70 | 43.48 | 61.84 | 75.04 | 108.89 | 83.75 | 118.66 | 472.10 |
| Isentropic Compressor Work (Btu/lb) | 9.04 | 9.30 | 7.13 | 9.94 | 8.58 | 8.83 | 8.69 | 10.16 | 14.82 | 11.09 | 17.55 | 65.52 |
| Equivalent Isentropic Head (ft·lb/lb) | 7029 | 7234 | 5541 | 7728 | 6676 | 6869 | 6759 | 7904 | 11528 | 8623 | 13648 | 50955 |
| Isentropic COP | 7.545 | 6.972 | 7.049 | 6.990 | 6.368 | 4.911 | 7.112 | 7.379 | 7.348 | 7.549 | 6.757 | 7.20 |
| Mass Flow Rate (lb/min/ton) | 2.93 | 3.08 | 3.98 | 2.88 | 3.66 | 4.61 | 3.23 | 2.66 | 1.84 | 2.39 | 1.69 | 0.42 |
| Suction Vapor Density (ft³/lb) | 5.44 | 0.95 | 0.78 | 0.68 | 10.46 | 1.97 | 5.40 | 2.01 | 1.66 | 8.83 | 1.34 | 3.95 |
| Suction Flow Rate (cfm/ton) | 15.94 | 2.94 | 3.09 | 1.91 | 38.23 | 9.10 | 17.47 | 5.35 | 3.05 | 21.10 | 2.27 | 1.67 |
| Impeller Tip Speed (with 0.59 Head Coefficient) (ft/s) | 619.1 | 628.1 | 549.7 | 649.1 | 603.3 | 612.0 | 607.1 | 656.5 | 792.8 | 685.7 | 862.7 | 1667 |
| Refrigerant Sonic Velocity (ft/s) | 443.2 | 484.3 | 447.2 | 536.2 | 374.1 | 379.7 | 413.6 | 501.9 | 620.9 | 478.7 | 722.2 | 1320.0 |
| Rotational Mach Number | 1.397 | 1.297 | 1.229 | 1.211 | 1.613 | 1.612 | 1.468 | 1.308 | 1.272 | 1.432 | 1.197 | 1.263 |
| Relative Efficiency Due to Mach Number | 0.9486 | 0.9643 | 0.9739 | 0.9763 | 0.9072 | 0.9074 | 0.9361 | 0.9627 | 0.9672 | 0.9425 | 0.9784 | 0.9692 |
| Stage Efficiency (Assumed 80% Mechanical Efficiency) | 0.7589 | 0.7715 | 0.7791 | 0.7811 | 0.7258 | 0.7559 | 0.7489 | 0.7702 | 0.7738 | 0.7540 | 0.7827 | 0.7754 |
| Discharge Temperature (°F) | 112.4 | 105.7 | 109.8 | 128.31 | 100.0 | 100.0 | 100.0 | 104.3 | 118.5 | 106.1 | 108.7 | 214.9 |
| Kilowatts/ton | 0.6136 | 0.6532 | 0.6396 | 0.6434 | 0.7601 | 0.9854 | 0.6596 | 0.6181 | 0.6178 | 0.6172 | 0.6642 | 0.6291 |
| RPM for 20-in. Impeller | 7094 | 7197 | 6299 | 7439 | 6914 | 7013 | 6957 | 7522 | 9085 | 7858 | 9885 | 19100 |
| Tons for 20-in. Impeller | 371.0 | 2038.6 | 1696.6 | 3249.7 | 150.7 | 642.2 | 331.8 | 1171.2 | 2480.5 | 310.4 | 3635.6 | 9510.3 |
| Tonnage Compared with R-11 (%) | 100.0 | 549.5 | 457.3 | 875.9 | 40.6 | 173.1 | 89.4 | 315.7 | 668.6 | 83.7 | 980.0 | 2563.4 |
| Tonnage Compared with R-134a (%) | 18.2 | 100.0 | 83.2 | 159.4 | 7.4 | 31.5 | 18.3 | 57.5 | 121.7 | 15.2 | 178.3 | 475.0 |

[a] "Wet" isentropic compression allowed.

Table B-3

SECOND GENERATION HFCs AND EPRI/EPA HFC PROPANES

(based on a cycle at 40°F evaporation/100°F condensation, some superheat [a]/no liquid subcooling)

| Refrigerant | R-11 | R-134a | R-134 | R-143 | R-152 | R-245ca | R-245cb | R-245fa | R-236ca | R-236cb | R-236ea | R-236fa | R-227ea |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| Evaporator Pressure (psia) | 7.03 | 50.25 | 38.61 | 14.43 | 5.02 | 6.10 | 36.85 | 9.52 | 14.38 | 16.65 | 13.51 | 18.34 | 33.29 |
| Condenser Pressure (psia) | 23.52 | 140.57 | 110.99 | 43.58 | 18.40 | 22.22 | 102.75 | 33.29 | 46.73 | 52.25 | 44.63 | 56.58 | 95.48 |
| Pressure Ratio | 3.35 | 2.80 | 2.88 | 3.02 | 3.67 | 3.64 | 2.79 | 3.50 | 3.25 | 3.14 | 3.30 | 3.08 | 2.87 |
| Net Refrigerating Effect (Btu/lb) | 68.23 | 64.89 | 69.41 | 99.81 | 157.46 | 72.21 | 47.93 | 68.56 | 54.94 | 52.77 | 56.07 | 51.29 | 38.84 |
| Isentropic Compressor Work (Btu/lb) | 9.04 | 9.30 | 9.77 | 13.32 | 20.71 | 9.85 | 7.09 | 9.44 | 7.72 | 7.44 | 7.85 | 7.28 | 5.80 |
| Equivalent Isentropic Head (ft·lb/lb) | 7029 | 7234 | 7598 | 10355 | 16108 | 7660 | 5515 | 7344 | 6005 | 5789 | 6108 | 5662 | 4510 |
| Isentropic COP | 7.545 | 6.972 | 7.101 | 7.492 | 7.598 | 7.327 | 6.756 | 7.256 | 7.111 | 7.085 | 7.136 | 7.041 | 6.694 |
| Mass Flow Rate (lb/min/ton) | 2.93 | 3.08 | 2.88 | 2.00 | 1.27 | 2.77 | 4.17 | 2.92 | 3.64 | 3.79 | 3.57 | 3.90 | 5.15 |
| Suction Vapor Density (ft³/lb) | 5.44 | 0.95 | 1.27 | 4.28 | 15.93 | 6.52 | 1.02 | 4.15 | 2.42 | 2.08 | 2.58 | 1.88 | 0.90 |
| Suction Flow Rate (cfm/ton) | 15.94 | 2.94 | 3.65 | 8.57 | 20.23 | 18.05 | 4.24 | 12.12 | 8.80 | 7.88 | 9.19 | 7.33 | 4.63 |
| Impeller Tip Speed (ft/s) (with 0.59 Head Coefficient) (ft/s) | 619.1 | 628.1 | 643.6 | 751.4 | 937.2 | 646.3 | 548.3 | 632.8 | 572.2 | 561.8 | 577.1 | 555.6 | 495.9 |
| Refrigerant Sonic Velocity (ft/s) | 443.2 | 484.3 | 489.9 | 565.1 | 648.9 | 443.0 | 420.4 | 440.8 | 410.2 | 408.8 | 410.9 | 407.3 | 374.3 |
| Rotational Mach Number | 1.397 | 1.297 | 1.267 | 1.330 | 1.444 | 1.459 | 1.304 | 1.436 | 1.395 | 1.374 | 1.405 | 1.364 | 1.325 |
| Relative Efficiency Due to Mach Number | 0.9486 | 0.9643 | 1.314 | 0.9594 | 0.9404 | 0.9378 | 0.9632 | 0.9419 | 0.9489 | 0.9523 | 0.9473 | 0.9540 | 0.9601 |
| Stage Efficiency (Assumed 80% Mechanical Efficiency) | 0.7589 | 0.7715 | 0.9618 | 0.7675 | 0.7523 | 0.7502 | 0.7706 | 0.7535 | 0.7591 | 0.7619 | 0.7578 | 0.7632 | 0.7681 |
| Discharge Temperature (°F) | 112.4 | 105.7 | 102.5 | 113.0 | 128.3 | 100.0 | 100.0 | 100.0 | 100.0 | 100.0 | 100.0 | 100.0 | 100.0 |
| Kilowatts/ton | 0.6136 | 0.6532 | 0.6429 | 0.6109 | 0.6146 | 0.6391 | 0.6748 | 0.6425 | 0.6507 | 0.6508 | 0.6496 | 0.6538 | 0.6832 |
| RPM for 20-in. Impeller | 7094 | 7197 | 7375 | 8611 | 10739 | 7405 | 6284 | 7252 | 6557 | 6438 | 6613 | 6367 | 5683 |
| Tons for 20-in. Impeller | 371.0 | 2038.6 | 1684.7 | 837.2 | 442.3 | 342.0 | 1235.5 | 498.7 | 621.0 | 680.4 | 599.8 | 723.9 | 1021.9 |
| Tonnage Compared with R-11 (%) | 100.0 | 549.5 | 454.1 | 225.7 | 119.2 | 92.0 | 333.0 | 134.4 | 167.4 | 183.4 | 161.6 | 195.1 | 275.4 |
| Tonnage Compared with R-134a (%) | 18.2 | 100.0 | 82.6 | 41.1 | 21.7 | 16.8 | 60.1 | 24.5 | 30.5 | 33.4 | 29.4 | 35.5 | 50.1 |

[a] Enough suction superheat added to avoid "wet compression" where applicable; cooling effect of superheating applied to performance and COP.

91-5017A

Table B-4

SECOND GENERATION HFCs AND EPRI/EPA HFC PROPANES
(based on a cycle at 40°F evaporation/100°F condensation,
no superheat[a] /no liquid subcooling)

| Refrigerant | R-11 | R-134a | R-134 | R-143 | R-152 | R-245ca | R-245cb | R-245fa | R-236ca | R-236cb | R-236ea | R-236fa | R-227ea |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| Evaporator Pressure (psia) | 7.03 | 50.25 | 38.61 | 14.43 | 5.02 | 6.10 | 36.85 | 9.52 | 14.38 | 16.65 | 13.51 | 18.34 | 33.29 |
| Condenser Pressure (psia) | 23.52 | 140.57 | 110.99 | 43.58 | 18.40 | 22.22 | 102.75 | 33.29 | 46.73 | 52.25 | 44.63 | 56.58 | 95.48 |
| Pressure Ratio | 3.35 | 2.80 | 2.88 | 3.02 | 3.67 | 3.64 | 2.79 | 3.50 | 3.25 | 3.14 | 3.30 | 3.08 | 2.87 |
| Net Refrigerating Effect (Btu/lb) | 68.23 | 64.89 | 69.41 | 99.81 | 157.46 | 70.82 | 45.97 | 67.05 | 52.65 | 50.33 | 53.86 | 48.85 | 36.18 |
| Isentropic Compressor Work (Btu/lb) | 9.04 | 9.30 | 9.77 | 13.32 | 20.71 | 11.24 | 9.05 | 10.96 | 10.02 | 9.89 | 10.07 | 9.72 | 8.47 |
| Equivalent Isentropic Head (ft·lb/lb) | 7029 | 7234 | 7598 | 10355 | 16108 | 8744 | 7038 | 8520 | 7788 | 7688 | 7832 | 7562 | 6585 |
| Isentropic COP | 7.545 | 6.972 | 7.101 | 7.492 | 7.598 | 6.295 | 5.078 | 6.117 | 5.254 | 5.088 | 5.345 | 5.021 | 4.270 |
| Mass Flow Rate (lb/min/ton) | 2.93 | 3.08 | 2.88 | 2.00 | 1.27 | 2.82 | 4.35 | 2.98 | 3.80 | 3.97 | 3.71 | 4.09 | 5.53 |
| Suction Vapor Density (ft³/lb) | 5.44 | 0.95 | 1.27 | 4.28 | 15.93 | 6.42 | 0.99 | 4.09 | 2.35 | 2.02 | 2.51 | 1.82 | 0.87 |
| Suction Flow Rate (cfm/ton) | 15.94 | 2.94 | 3.65 | 8.57 | 20.23 | 18.13 | 4.31 | 12.19 | 8.94 | 8.03 | 9.32 | 7.47 | 4.79 |
| Impeller Tip Speed (with 0.59 Head Coefficient) (ft/s) | 619.1 | 628.1 | 643.6 | 751.4 | 937.2 | 690.5 | 619.4 | 681.6 | 651.7 | 647.5 | 653.5 | 642.1 | 599.2 |
| Refrigerant Sonic Velocity (ft/s) | 443.2 | 484.3 | 489.9 | 565.1 | 648.9 | 439.8 | 414.5 | 437.1 | 404.4 | 402.6 | 405.3 | 400.9 | 366.31 |
| Rotational Mach Number | 1.397 | 1.297 | 1.267 | 1.330 | 1.444 | 1.570 | 1.494 | 1.559 | 1.611 | 1.608 | 1.612 | 1.602 | 1.636 |
| Relative Efficiency Due to Mach Number | 0.9486 | 0.9643 | 1.314 | 0.9594 | 0.9404 | 0.9162 | 0.9312 | 0.9184 | 0.9075 | 0.9082 | 0.9073 | 0.9096 | 0.9021 |
| Stage Efficiency (Assumed 80% Mechanical Efficiency) | 0.7589 | 0.7715 | 0.9618 | 0.7675 | 0.7523 | 0.7330 | 0.7449 | 0.7348 | 0.7260 | 0.7265 | 0.7258 | 0.7277 | 0.7217 |
| Discharge Temperature (°F) | 112.4 | 105.7 | 102.5 | 113.0 | 128.3 | 100.0 | 100.0 | 100.0 | 100.0 | 100.0 | 100.0 | 100.0 | 100.0 |
| Kilowatts/ton | 0.6136 | 0.6532 | 0.6429 | 0.6109 | 0.6146 | 0.7614 | 0.9286 | 0.7816 | 0.9209 | 0.9503 | 0.9056 | 0.9614 | 1.1400 |
| RPM for 20-in. Impeller | 7094 | 7197 | 7375 | 8611 | 10739 | 7912 | 7098 | 7810 | 7467 | 7419 | 7489 | 7358 | 6866 |
| Tons for 20-in. Impeller | 371.0 | 2038.6 | 1684.7 | 837.2 | 442.3 | 363.7 | 1373.0 | 534.0 | 696.4 | 770.2 | 669.4 | 820.9 | 1194.1 |
| Tonnage Compared with R-11 (%) | 100.0 | 549.5 | 454.1 | 225.7 | 119.2 | 98.0 | 370.0 | 143.9 | 187.7 | 207.8 | 180.4 | 221.2 | 321.85 |
| Tonnage Compared with R-134a (%) | 18.2 | 100.0 | 82.6 | 41.1 | 21.7 | 17.8 | 67.4 | 26.2 | 34.2 | 37.8 | 32.8 | 40.3 | 58.57 |

[a] "Wet" isentropic compression allowed.

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## Table B-5
## SECOND GENERATION HFCs AND EPRI/EPA HFC PROPANES
(based on a cycle at 40°F evaporation/100°F condensation, suction-to-liquid line heat exchanger to avoid "wet compression")

| Refrigerant | R-11 | R-134a | R-134 | R-143 | R-152 | R-245ca | R-245cb | R-245fa | R-236ca | R-236cb | R-236ea | R-236fa | R-227ea |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| Evaporator Pressure (psia) | 7.03 | 50.25 | 38.61 | 14.43 | 5.02 | 6.10 | 36.85 | 9.52 | 14.38 | 16.65 | 13.51 | 18.34 | 33.29 |
| Condenser Pressure (psia) | 23.52 | 140.57 | 110.99 | 43.58 | 18.40 | 22.22 | 102.75 | 33.29 | 46.73 | 52.25 | 44.63 | 56.58 | 95.48 |
| Pressure Ratio | 3.35 | 2.80 | 2.88 | 3.02 | 3.67 | 3.64 | 2.79 | 3.50 | 3.25 | 3.14 | 3.30 | 3.08 | 2.87 |
| Net Refrigerating Effect (Btu/lb) | 68.23 | 64.89 | 69.41 | 99.81 | 157.46 | 74.11 | 49.33 | 70.35 | 56.40 | 54.59 | 57.53 | 53.12 | 40.62 |
| Isentropic Compressor Work (Btu/lb) | 9.04 | 9.30 | 9.77 | 13.32 | 20.71 | 10.61 | 7.21 | 9.59 | 7.83 | 7.58 | 7.97 | 7.42 | 8.47 |
| Equivalent Isentropic Head (ft·lb/lb) | 7029 | 7234 | 7598 | 10355 | 16108 | 7783 | 5609 | 7460 | 6091 | 5893 | 6194 | 5767 | 4610 |
| Isentropic COP | 7.545 | 6.972 | 7.101 | 7.492 | 7.598 | 7.405 | 6.837 | 7.334 | 7.200 | 7.205 | 7.223 | 7.163 | 6.851 |
| Mass Flow Rate (lb/min/ton) | 2.93 | 3.08 | 2.88 | 2.00 | 1.27 | 3.70 | 4.05 | 2.84 | 3.55 | 3.66 | 3.48 | 3.76 | 4.92 |
| Suction Vapor Density (ft³/lb) | 5.44 | 0.95 | 1.27 | 4.28 | 15.93 | 6.64 | 1.03 | 4.23 | 2.46 | 2.12 | 2.62 | 1.92 | 0.92 |
| Suction Flow Rate (cfm/ton) | 15.94 | 2.94 | 3.65 | 8.57 | 20.23 | 17.92 | 4.19 | 12.02 | 8.71 | 7.78 | 9.10 | 7.22 | 4.53 |
| Impeller Tip Speed (with 0.59 Head Coefficient) (ft/s) | 619.1 | 628.1 | 643.6 | 751.4 | 937.2 | 651.4 | 553.0 | 637.8 | 576.3 | 566.8 | 581.2 | 560.7 | 501.4 |
| Refrigerant Sonic Velocity (ft/s) | 443.2 | 484.3 | 489.9 | 565.1 | 648.9 | 447.3 | 424.6 | 444.9 | 413.7 | 413.3 | 414.4 | 411.8 | 379.2 |
| Rotational Mach Number | 1.397 | 1.297 | 1.267 | 1.330 | 1.444 | 1.456 | 1.303 | 1.433 | 1.393 | 1.372 | 1.402 | 1.362 | 1.322 |
| Relative Efficiency Due to Mach Number | 0.9486 | 0.9643 | 1.314 | 0.9594 | 0.9404 | 0.9382 | 0.9635 | 0.9423 | 0.9473 | 0.9528 | 0.9476 | 0.9544 | 0.9606 |
| Stage Efficiency (Assumed 80% Mechanical Efficiency) | 0.7589 | 0.7715 | 0.9618 | 0.7675 | 0.7523 | 0.7506 | 0.7708 | 0.7538 | 0.7594 | 0.7622 | 0.7581 | 0.7635 | 0.7684 |
| Discharge Temperature (°F) | 112.4 | 105.7 | 102.5 | 113.0 | 128.3 | 109.4 | 106.4 | 108.8 | 107.5 | 109.2 | 107.5 | 109.1 | 108.8 |
| Kilowatts/ton | 0.6136 | 0.6532 | 0.6429 | 0.6109 | 0.6146 | 0.6323 | 0.6667 | 0.6357 | 0.6400 | 0.6400 | 0.6419 | 0.6426 | 0.6676 |
| RPM for 20-in. Impeller | 7094 | 7197 | 7375 | 8611 | 10739 | 7465 | 6337 | 7308 | 6604 | 6495 | 6659 | 6426 | 5745 |
| Tons for 20-in. Impeller | 371.0 | 2038.6 | 1684.7 | 837.2 | 442.3 | 347.2 | 1261.1 | 506.6 | 632.6 | 696.2 | 609.8 | 741.3 | 1056.2 |
| Tonnage Compared with R-11 (%) | 100.0 | 549.5 | 454.1 | 225.7 | 119.2 | 93.6 | 339.7 | 136.5 | 170.3 | 187.6 | 164.4 | 199.8 | 284.7 |
| Tonnage Compared with R-134a (%) | 18.2 | 100.0 | 82.6 | 41.1 | 21.7 | 17.0 | 61.9 | 24.85 | 31.0 | 34.1 | 29.9 | 36.4 | 51.8 |

Table B-6

ETHER REFRIGERANTS COMPARED TO R-11 AND R-134a
(based on a cycle at 40°F evaporation/100°F condensation,
some superheat[a]/no liquid subcooling)

| Refrigerant | R-11 | R-134a | E-134 | E-245cb | E-254cb | E-227ca | E-143 | E-143a | CE-216 | EE-218 | E-125 |
|---|---|---|---|---|---|---|---|---|---|---|---|
| Evaporator Pressure (psia) | 7.03 | 50.25 | 14.18 | 4.35 | 3.87 | 20.84 | 4.96 | 44.94 | 54.13 | 25.96 | 70.06 |
| Condenser Pressure (psia) | 23.52 | 140.57 | 47.88 | 16.73 | 15.29 | 63.19 | 19.27 | 121.67 | 144.11 | 78.60 | 178.64 |
| Pressure Ratio | 3.35 | 2.80 | 3.38 | 3.84 | 3.95 | 3.03 | 3.89 | 2.71 | 2.66 | 3.03 | 2.55 |
| Net Refrigerating Effect (Btu/lb) | 68.23 | 64.89 | 75.47 | 68.04 | 79.41 | 39.64 | 105.19 | 62.09 | 34.88 | 31.92 | 37.20 |
| Isentropic Compressor Work (Btu/lb) | 9.04 | 9.30 | 10.30 | 9.28 | 10.76 | 5.79 | 14.01 | 9.44 | 5.21 | 4.81 | 5.92 |
| Equivalent Isentropic Head (ft·lb/lb) | 7029 | 7235 | 8008 | 7217 | 8366 | 4505 | 10894 | 7343 | 4048 | 3740 | 4603 |
| Isentropic COP | 7.545 | 6.972 | 7.325 | 7.328 | 7.378 | 6.840 | 7.504 | 6.572 | 6.697 | 6.634 | 6.282 |
| Mass Flow Rate (lb/min/ton) | 2.93 | 3.08 | 2.65 | 2.94 | 2.52 | 5.04 | 1.90 | 3.22 | 5.73 | 6.26 | 5.38 |
| Suction Vapor Density (ft³/lb) | 5.44 | 0.95 | 3.09 | 8.28 | 10.52 | 1.36 | 10.65 | 1.17 | 0.53 | 0.92 | 0.50 |
| Suction Flow Rate (cfm/ton) | 15.94 | 2.94 | 8.20 | 24.33 | 26.51 | 6.84 | 20.25 | 3.75 | 3.03 | 5.75 | 2.67 |
| Impeller Tip Speed (with 0.59 Head Coefficient) (ft/s) | 619.1 | 628.1 | 660.8 | 627.3 | 675.4 | 495.6 | 770.8 | 632.7 | 469.8 | 451.6 | 501.0 |
| Refrigerant Sonic Velocity (ft/s) | 443.2 | 484.3 | 469.2 | 420.3 | 448.1 | 364.8 | 517.0 | 488.5 | 369.9 | 331.4 | 399.9 |
| Rotational Mach Number | 1.397 | 1.297 | 1.408 | 1.493 | 1.507 | 1.359 | 1.491 | 1.295 | 1.270 | 1.362 | 1.253 |
| Relative Efficiency Due to Mach Number | 0.9488 | 0.9643 | 0.9466 | 0.9315 | 0.9288 | 0.9548 | 0.9318 | 0.9648 | 0.9682 | 0.9544 | 0.9706 |
| Stage Efficiency (Assumed 80% Mechanical Efficiency) | 0.7589 | 0.7715 | 0.7573 | 0.7452 | 0.7430 | 0.7639 | 0.7455 | 0.7716 | 0.7746 | 0.7635 | 0.7765 |
| Discharge Temperature (°F) | 112.4 | 105.7 | 106.0 | 100.0 | 100.0 | 100.0 | 109.0 | 100.0 | 103.7 | 100.0 | 100.0 |
| Kilowatt/ton | 0.6136 | 0.6532 | 0.6333 | 0.6433 | 0.6408 | 0.6724 | 0.6279 | 0.6927 | 0.6772 | 0.6936 | 0.7202 |
| RPM for 20-in. Impeller | 7094 | 7197 | 7572 | 7188 | 7740 | 5679 | 8832 | 7251 | 5384 | 5175 | 5741 |
| Tons for 20-in. Impeller | 371.0 | 2038.6 | 769.8 | 246.3 | 243.3 | 691.7 | 363.5 | 1609.8 | 1481.1 | 750.5 | 1795.0 |
| Tonnage Compared with R-11 (%) | 100.0 | 549.5 | 207.5 | 66.4 | 65.6 | 186.4 | 98.0 | 433.9 | 399.2 | 202.3 | 483.8 |
| Tonnage Compared with R-134a (%) | 18.2 | 100.0 | 37.8 | 12.1 | 11.9 | 33.9 | 17.8 | 79.0 | 72.6 | 36.8 | 88.0 |

[a] Enough suction superheat added to avoid "wet compression" where applicable; cooling effect of superheating applied to performance and COP.

Table B-7

ETHER REFRIGERANTS COMPARED TO R-11 AND R-134a
(based on a cycle at 40°F evaporation/100°F condensation,
no superheat[a] /no liquid subcooling)

| Refrigerant | R-11 | R-134a | E-134 | E-245cb | E-254cb | E-227ca | E-143 | E-143a | CE-216 | EE-218 | E-125 |
|---|---|---|---|---|---|---|---|---|---|---|---|
| Evaporator Pressure (psia) | 7.03 | 50.25 | 14.18 | 4.35 | 3.87 | 20.84 | 4.96 | 44.94 | 54.13 | 25.96 | 70.06 |
| Condenser Pressure (psia) | 23.52 | 140.57 | 47.88 | 16.73 | 15.29 | 63.19 | 19.27 | 121.67 | 144.11 | 78.60 | 178.64 |
| Pressure Ratio | 3.35 | 2.80 | 3.38 | 3.84 | 3.95 | 3.03 | 3.89 | 2.71 | 2.66 | 3.03 | 2.55 |
| Net Refrigerating Effect (Btu/lb) | 68.23 | 64.89 | 75.47 | 65.70 | 77.76 | 38.04 | 105.19 | 50.05 | 34.88 | 27.73 | 38.17 |
| Isentropic Compressor Work (Btu/lb) | 9.04 | 9.30 | 10.30 | 11.62 | 12.41 | 9.39 | 14.01 | 21.48 | 5.21 | 9.00 | 6.96 |
| Equivalent Isentropic Head (ft·lb/lb) | 7029 | 7235 | 8008 | 9035 | 9653 | 7302 | 10894 | 16705 | 4048 | 6998 | 5410 |
| Isentropic COP | 7.545 | 6.972 | 7.325 | 5.652 | 6.261 | 3.837 | 7.504 | 2.329 | 6.697 | 3.000 | 5.196 |
| Mass Flow Rate (lb/min/ton) | 2.93 | 3.08 | 2.65 | 3.04 | 2.57 | 5.55 | 1.90 | 4.00 | 5.73 | 7.21 | 5.53 |
| Suction Vapor Density ($ft^3/lb$) | 5.44 | 0.95 | 3.09 | 8.08 | 10.36 | 1.29 | 10.65 | 1.08 | 0.53 | 0.87 | 0.49 |
| Suction Flow Rate (cfm/ton) | 15.94 | 2.94 | 8.20 | 24.59 | 26.64 | 7.18 | 20.25 | 4.33 | 3.03 | 6.25 | 2.70 |
| Impeller Tip Speed (with 0.59 Head Coefficient) (ft/s) | 619.1 | 628.1 | 660.8 | 701.9 | 725.5 | 631.0 | 770.8 | 954.4 | 469.8 | 617.7 | 543.1 |
| Refrigerant Sonic Velocity (ft/s) | 443.2 | 484.3 | 469.2 | 415.3 | 444.6 | 355.5 | 517.0 | 468.0 | 369.9 | 320.8 | 395.8 |
| Rotational Mach Number | 1.397 | 1.297 | 1.408 | 1.690 | 1.632 | 1.775 | 1.491 | 2.039 | 1.270 | 1.926 | 1.372 |
| Relative Efficiency Due to Mach Number | 0.9486 | 0.9643 | 0.9468 | 0.8897 | 0.9030 | 0.8687 | 0.9318 | 0.7904 | 0.9682 | 0.8266 | 0.9526 |
| Stage Efficiency (Assumed 80% Mechanical Efficiency) | 0.7589 | 0.7715 | 0.7573 | 0.7117 | 0.7224 | 0.6949 | 0.7455 | 0.6323 | 0.7746 | 0.6612 | 0.7621 |
| Discharge Temperature (°F) | 112.4 | 105.7 | 106.0 | 100.0 | 100.0 | 100.0 | 109.0 | 100.0 | 103.7 | 100.0 | 100.0 |
| Kilowatts/ton | 0.6136 | 0.6532 | 0.6333 | 0.8732 | 0.7766 | 1.3175 | 0.6279 | 2.3857 | 0.6772 | 1.7248 | 0.8871 |
| RPM for 20-in. Impeller | 7094 | 7197 | 7572 | 8043 | 8313 | 7230 | 8832 | 10936 | 5384 | 7078 | 6224 |
| Tons for 20-in. Impeller | 371.0 | 2038.6 | 769.8 | 272.6 | 260.0 | 838.7 | 363.5 | 2104.3 | 1481.1 | 944.4 | 1923.4 |
| Tonnage Compared with R-11 (%) | 100.0 | 549.5 | 207.5 | 73.5 | 70.1 | 226.0 | 98.0 | 567.2 | 399.2 | 254.5 | 518.4 |
| Tonnage Compared with R-134a (%) | 18.2 | 100.0 | 37.8 | 13.4 | 12.8 | 41.1 | 17.8 | 105.2 | 72.6 | 46.3 | 94.4 |

a "Wet" isentropic compression allowed

Table B-8
ETHER REFRIGERANTS COMPARED TO R-11 AND R-134a
(based on a cycle at 40°F evaporation/100°F condensation,
suction-to-liquid line heat exchanger to avoid "wet compression")

| Refrigerant | R-11 | R-134a | E-134 | E-245cb | E-254cb | E-227ca | E-143 | E-143a | CE-216 | EE-218 | E-125 |
|---|---|---|---|---|---|---|---|---|---|---|---|
| Evaporator Pressure (psia) | 7.03 | 50.25 | 14.18 | 4.35 | 3.87 | 20.84 | 4.96 | 44.94 | 54.13 | 25.96 | 70.06 |
| Condenser Pressure (psia) | 23.52 | 140.57 | 47.88 | 16.73 | 15.29 | 63.19 | 19.27 | 121.67 | 144.11 | 78.60 | 178.64 |
| Pressure Ratio | 3.35 | 2.80 | 3.38 | 3.84 | 3.95 | 3.03 | 3.89 | 2.71 | 2.66 | 3.03 | 2.55 |
| Net Refrigerating Effect (Btu/lb) | 68.23 | 64.89 | 75.47 | 69.59 | 80.73 | 41.11 | 105.19 | 65.56 | 34.88 | 33.39 | 42.54 |
| Isentropic Compressor Work (Btu/lb) | 9.04 | 9.30 | 10.30 | 9.40 | 10.87 | 5.89 | 14.01 | 9.60 | 5.21 | 4.89 | 6.35 |
| Equivalent Isentropic Head (ft·lb/lb) | 7029 | 7235 | 8008 | 7306 | 8450 | 4577 | 10894 | 7469 | 4048 | 3806 | 4935 |
| Isentropic COP | 7.545 | 6.972 | 7.325 | 7.409 | 7.430 | 6.985 | 7.504 | 6.826 | 6.697 | 6.823 | 6.704 |
| Mass Flow Rate (lb/min/ton) | 2.93 | 3.08 | 2.65 | 2.87 | 2.48 | 4.86 | 1.90 | 3.05 | 5.73 | 5.99 | 4.70 |
| Suction Vapor Density (ft³/lb) | 5.44 | 0.95 | 3.09 | 8.40 | 10.65 | 1.38 | 10.65 | 1.19 | 0.53 | 0.93 | 0.53 |
| Suction Flow Rate (cfm/ton) | 15.94 | 2.94 | 8.20 | 24.14 | 26.38 | 6.71 | 20.25 | 3.62 | 3.03 | 5.59 | 2.51 |
| Impeller Tip Speed (with 0.59 Head Coefficient) (ft/s) | 619.1 | 628.1 | 660.8 | 631.2 | 678.8 | 499.6 | 770.8 | 638.1 | 469.8 | 455.5 | 518.7 |
| Refrigerant Sonic Velocity (ft/s) | 443.2 | 484.3 | 469.2 | 423.5 | 450.9 | 368.3 | 517.0 | 493.8 | 369.9 | 335.2 | 418.2 |
| Rotational Mach Number | 1.397 | 1.297 | 1.408 | 1.490 | 1.565 | 1.556 | 1.491 | 1.292 | 1.270 | 1.359 | 1.240 |
| Relative Efficiency Due to Mach Number | 0.9486 | 0.9643 | 0.9466 | 0.9320 | 0.9291 | 0.9552 | 0.9318 | 0.9650 | 0.9682 | 0.9548 | 0.9724 |
| Stage Efficiency (Assumed 80% Mechanical Efficiency) | 0.7589 | 0.7715 | 0.7573 | 0.7456 | 0.7433 | 0.7642 | 0.7455 | 0.7720 | 0.7746 | 0.7638 | 0.7779 |
| Discharge Temperature (°F) | 112.4 | 105.7 | 106.0 | 107.7 | 106.3 | 107.5 | 109.0 | 107.3 | 103.7 | 107.4 | 122.6 |
| Kilowatts/ton | 0.6136 | 0.6532 | 0.6333 | 0.6364 | 0.6364 | 0.6585 | 0.6279 | 0.6670 | 0.6772 | 0.6743 | 0.6740 |
| RPM for 20-in. Impeller | 7094 | 7197 | 7572 | 7233 | 7778 | 5725 | 8832 | 7312 | 5384 | 5220 | 5944 |
| Tons for 20-in. Impeller | 371.0 | 2038.6 | 769.8 | 249.7 | 245.7 | 711.0 | 363.5 | 1683.7 | 1481.1 | 778.0 | 1974.5 |
| Tonnage Compared with R-11 (%) | 100.0 | 549.5 | 207.5 | 27.3 | 66.2 | 191.6 | 98.0 | 453.8 | 399.2 | 209.7 | 532.2 |
| Tonnage Compared with R-134a (%) | 18.2 | 100.0 | 37.8 | 12.2 | 12.0 | 34.9 | 17.8 | 82.6 | 72.6 | 38.2 | 96.9 |

Table B-9

NEARMS FROM SELECTED REFRIGERANTS
(based on a cycle at 40°F evaporation/100°F condensation, some superheat[a] /no liquid subcooling)

| Refrigerant | R-11 | R-134a | R-143/ R-236ea | R-143/ R-236ea | R-143/ R-236ca | R-143/ R-236ca | R-143/ E-134 | E-143/ R-245ca | E-143/ R-245ca | R-152a/ R-134 |
|---|---|---|---|---|---|---|---|---|---|---|
| Mass Fraction | 1.00 | 1.00 | 0.108/ 0.892 | 0.485/ 0.515 | 0.108/ 0.892 | 0.485/ 0.515 | 0.20/ 0.80 | 0.208/ 0.792 | 0.45/ 0.55 | 0.19/ 0.81 |
| Flammability Index Classification | Nonflam. | Nonflam. | Nonflam. | Uncertain | Nonflam. | Uncertain | Uncertain | Uncertain | Uncertain | Uncertain |
| Evaporator Pressure (psia) | 7.03 | 50.25 | 13.61 | 13.98 | 14.37 | 14.38 | 13.89 | 5.64 | 5.32 | 39.37 |
| Condenser Pressure (psia) | 23.52 | 140.57 | 44.32 | 43.76 | 46.12 | 44.68 | 45.85 | 21.15 | 20.26 | 112.62 |
| Pressure Ratio | 3.35 | 2.80 | 3.25 | 3.13 | 3.21 | 3.11 | 3.30 | 3.75 | 3.81 | 2.86 |
| Net Refrigerating Effect (Btu/lb) | 68.23 | 64.89 | 60.59 | 76.30 | 59.55 | 75.59 | 80.83 | 78.89 | 86.62 | 77.43 |
| Isentropic Compressor Work (Btu/lb) | 9.04 | 9.30 | 8.43 | 10.39 | 8.31 | 10.31 | 10.98 | 10.77 | 11.74 | 10.79 |
| Equivalent Isentropic Head (ft·lb/lb) | 7029 | 7235 | 6552 | 8079 | 6459 | 8015 | 8536 | 8379 | 9128 | 8392 |
| Isentropic COP | 7.545 | 6.972 | 7.187 | 7.340 | 7.168 | 7.330 | 7.360 | 7.319 | 7.376 | 7.171 |
| Mass Flow Rate (lb/min/ton) | 2.93 | 3.08 | 3.30 | 2.62 | 3.56 | 2.65 | 2.47 | 2.53 | 2.31 | 2.58 |
| Suction Vapor Density ($ft^3$/lb) | 5.44 | 0.95 | 2.77 | 3.39 | 2.62 | 3.29 | 3.42 | 7.50 | 8.52 | 1.37 |
| Suction Flow Rate (cfm/ton) | 15.94 | 2.94 | 9.13 | 8.88 | 8.79 | 8.71 | 8.46 | 19.01 | 19.68 | 3.54 |
| Impeller Tip Speed (with 0.59 Head Coefficient) (ft/s) | 619.1 | 628.1 | 597.7 | 663.7 | 593.4 | 661.1 | 682.2 | 675.9 | 705.5 | 676.5 |
| Refrigerant Sonic Velocity (ft/s) | 443.2 | 484.3 | 428.9 | 487.3 | 428.4 | 486.8 | 489.8 | 458.5 | 476.0 | 517.2 |
| Rotational Mach Number | 1.397 | 1.297 | 1.394 | 1.362 | 1.385 | 1.358 | 1.393 | 1.474 | 1.482 | 1.308 |
| Relative Efficiency Due to Mach Number | 0.9486 | 0.9643 | 0.9491 | 0.9543 | 0.9505 | 0.9550 | 0.9493 | 0.9350 | 0.9335 | 0.9627 |
| Stage Efficiency (Assumed 80% Mechanical Efficiency) | 0.7589 | 0.7715 | 0.7593 | 0.7634 | 0.7604 | 0.7640 | 0.7594 | 0.7480 | 0.7468 | 0.7702 |
| Discharge Temperature (°F) | 112.4 | 105.7 | 100.0 | 100.6 | 100.0 | 100.4 | 107.8 | 100.3 | 100.7 | 106.1 |
| Kilowatt hours/ton | 0.6136 | 0.6532 | 0.6437 | 0.6269 | 0.6447 | 0.6273 | 0.6285 | 0.6417 | 0.6377 | 0.6360 |
| RPM for 20-in. Impeller | 7094 | 7197 | 6849 | 7606 | 6800 | 7575 | 7817 | 7745 | 8084 | 7752 |
| Tons for 20-in. Impeller | 371.0 | 2038.6 | 625.2 | 713.4 | 644.7 | 724.7 | 770.4 | 339.5 | 342.3 | 1824.5 |
| Tonnage Compared with R-11 (%) | 100.0 | 549.5 | 168.5 | 192.3 | 173.8 | 195.3 | 207.6 | 91.51 | 92.3 | 491.8 |
| Tonnage Compared with R-134a (%) | 18.2 | 100.0 | 30.7 | 35.0 | 31.6 | 35.5 | 37.8 | 16.6 | 16.8 | 89.5 |

[a] Enough suction superheat added to avoid "wet compression" where applicable; cooling effect of superheating applied to performance and COP.

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Table B-10
NEARMS FROM SELECTED REFRIGERANTS
(based on a cycle at 40°F evaporation/100°F condensation, no superheat [a]/no liquid subcooling)

| Refrigerant | R-11 | R-134a | R-143/ R-236ea | R-143/ R-236ea | R-143/ R-236ca | R-143/ R-236ca | R-143/ E-134 | E-143/ R-245ca | E-143/ R-245ca | R-152a/ R-134 |
|---|---|---|---|---|---|---|---|---|---|---|
| Mass Fraction | 1.00 | 1.00 | 0.108/ 0.892 | 0.485/ 0.515 | 0.108/ 0.892 | 0.485/ 0.515 | 0.20/ 0.80 | 0.208/ 0.792 | 0.45/ 0.55 | 0.19/ 0.81 |
| Flammability Index Classification | Nonflam. | Nonflam. | Nonflam. | Uncertain | Nonflam. | Uncertain | Uncertain | Uncertain | Uncertain | Uncertain |
| Evaporator Pressure (psia) | 7.03 | 50.25 | 13.61 | 13.98 | 14.37 | 14.38 | 13.89 | 5.64 | 5.32 | 39.37 |
| Condenser Pressure (psia) | 23.52 | 140.57 | 44.32 | 43.76 | 46.12 | 44.68 | 45.85 | 21.15 | 20.26 | 112.82 |
| Pressure Ratio | 3.35 | 2.80 | 3.25 | 3.13 | 3.21 | 3.11 | 3.30 | 3.75 | 3.81 | 2.86 |
| Net Refrigerating Effect (Btu/lb) | 68.23 | 64.89 | 58.89 | 76.30 | 57.78 | 75.59 | 80.83 | 78.20 | 86.62 | 77.43 |
| Isentropic Compressor Work (Btu/lb) | 9.04 | 9.30 | 10.12 | 10.39 | 10.08 | 10.31 | 10.98 | 11.47 | 11.74 | 10.79 |
| Equivalent Isentropic Head (ft·lb/lb) | 7029 | 7235 | 7870 | 8079 | 7836 | 8015 | 8536 | 8918 | 9128 | 8392 |
| Isentropic COP | 7.545 | 6.972 | 5.816 | 7.340 | 5.731 | 7.330 | 7.360 | 6.816 | 7.376 | 7.171 |
| Mass Flow Rate (lb/min/ton) | 2.93 | 3.08 | 3.40 | 2.62 | 3.46 | 2.65 | 2.47 | 2.56 | 2.31 | 2.58 |
| Suction Vapor Density (ft³/lb) | 5.44 | 0.95 | 2.71 | 3.39 | 2.56 | 3.29 | 3.42 | 7.45 | 8.52 | 1.37 |
| Suction Flow Rate (cfm/ton) | 15.94 | 2.94 | 9.21 | 8.88 | 8.87 | 8.71 | 8.46 | 19.04 | 19.68 | 3.54 |
| Impeller Tip Speed (with 0.59 Head Coefficient) (ft/s) | 619.1 | 628.1 | 655.1 | 663.7 | 653.7 | 661.1 | 682.2 | 697.3 | 705.5 | 676.5 |
| Refrigerant Sonic Velocity (ft/s) | 443.2 | 484.3 | 424.6 | 487.3 | 423.8 | 486.8 | 489.8 | 456.9 | 476.0 | 517.2 |
| Rotational Mach Number | 1.397 | 1.297 | 1.543 | 1.362 | 1.542 | 1.358 | 1.393 | 1.526 | 1.482 | 1.308 |
| Relative Efficiency Due to Mach Number | 0.9486 | 0.9643 | 0.9217 | 0.9543 | 0.9218 | 0.9550 | 0.9493 | 0.9257 | 0.9335 | 0.9627 |
| Stage Efficiency (Assumed 80% Mechanical Efficiency) | 0.7589 | 0.7715 | 0.7374 | 0.7634 | 0.7375 | 0.7640 | 0.7594 | 0.7400 | 0.7468 | 0.7702 |
| Discharge Temperature (°F) | 112.4 | 105.7 | 100.0 | 100.6 | 100.0 | 100.4 | 107.8 | 100.3 | 100.7 | 106.1 |
| Kilowatt hours/ton | 0.6136 | 0.6532 | 0.8191 | 0.6269 | 0.8312 | 0.6273 | 0.6285 | 0.6965 | 0.6377 | 0.6360 |
| RPM for 20-in. Impeller | 7094 | 7197 | 7507 | 7606 | 7490 | 7575 | 7817 | 7991 | 8084 | 7752 |
| Tons for 20-in. Impeller | 371.0 | 2038.6 | 679.4 | 713.4 | 703.5 | 724.7 | 770.4 | 349.7 | 342.3 | 1824.5 |
| Tonnage Compared with R-11 (%) | 100.0 | 549.5 | 183.1 | 192.3 | 189.6 | 195.3 | 207.8 | 94.3 | 92.3 | 491.8 |
| Tonnage Compared with R-134a (%) | 18.2 | 100.0 | 33.3 | 35.0 | 34.5 | 35.5 | 37.8 | 17.2 | 16.8 | 89.5 |

a "Wet" isentropic compression allowed.

Table B-11
NEARMS FROM SELECTED REFRIGERANTS
(based on a cycle at 40°F evaporation/100°F condensation,
suction-to-liquid line heat exchanger to avoid "wet compression")

| Refrigerant | R-11 | R-134a | R-143/ R-236ea | R-143/ R-236ea | R-143/ R-236ca | R-143/ R-236ca | R-143/ E-134 | E-143/ R-245ca | E-143/ R-245ca | R-152a/ R-134 |
|---|---|---|---|---|---|---|---|---|---|---|
| Mass Fraction | 1.00 | 1.00 | 0.108/ 0.892 | 0.485/ 0.515 | 0.108/ 0.892 | 0.485/ 0.515 | 0.20/ 0.80 | 0.208/ 0.792 | 0.45/ 0.55 | 0.19/ 0.81 |
| Flammability Index Classification | Nonflam. | Nonflam. | Nonflam. | Uncertain | Nonflam. | Uncertain | Uncertain | Uncertain | Uncertain | Uncertain |
| Evaporator Pressure (psia) | 7.03 | 50.25 | 13.61 | 13.98 | 14.37 | 14.38 | 13.89 | 5.64 | 5.32 | 39.37 |
| Condenser Pressure (psia) | 23.52 | 140.57 | 44.32 | 43.76 | 46.12 | 44.68 | 45.85 | 21.15 | 20.26 | 112.62 |
| Pressure Ratio | 3.35 | 2.80 | 3.25 | 3.13 | 3.21 | 3.11 | 3.30 | 3.75 • | 3.81 | 2.86 |
| Net Refrigerating Effect (Btu/lb) | 68.23 | 64.89 | 62.08 | 76.30 | 60.94 | 75.59 | 80.83 | 79.92 | 86.62 | 77.43 |
| Isentropic Compressor Work (Btu/lb) | 9.04 | 9.30 | 8.55 | 10.39 | 8.41 | 10.31 | 10.98 | 10.77 | 11.74 | 10.79 |
| Equivalent Isentropic Head (ft·lb/lb) | 7029 | 7235 | 6649 | 8079 | 6541 | 8015 | 8536 | 8376 | 9128 | 8392 |
| Isentropic COP | 7.545 | 6.972 | 7.261 | 7.340 | 7.245 | 7.330 | 7.360 | 7.421 | 7.376 | 7.171 |
| Mass Flow Rate (lb/min/ton) | 2.93 | 3.08 | 3.22 | 2.62 | 3.243 | 2.65 | 2.47 | 2.50 | 2.31 | 2.58 |
| Suction Vapor Density ($ft^3$/lb) | 5.44 | 0.95 | 2.77 | 3.39 | 2.62 | 3.29 | 3.42 | 7.33 | 8.52 | 1.37 |
| Suction Flow Rate (cfm/ton) | 15.94 | 2.94 | 8.93 | 8.88 | 8.58 | 8.71 | 8.46 | 18.35 | 19.68 | 3.54 |
| Impeller Tip Speed (with 0.59 Head Coefficient) (ft/s) | 619.1 | 628.1 | 602.1 | 663.7 | 597.2 | 661.1 | 682.2 | 675.8 | 705.5 | 676.5 |
| Refrigerant Sonic Velocity (ft/s) | 443.2 | 484.3 | 430.0 | 487.3 | 431.5 | 486.8 | 489.8 | 461.8 | 476.0 | 517.2 |
| Rotational Mach Number | 1.397 | 1.297 | 1.400 | 1.362 | 1.384 | 1.358 | 1.393 | 1.463 | 1.482 | 1.308 |
| Relative Efficiency Due to Mach Number | 0.9486 | 0.9643 | 0.9480 | 0.9543 | 0.9507 | 0.9550 | 0.9493 | 0.9369 | 0.9335 | 0.9627 |
| Stage Efficiency (Assumed 80% Mechanical Efficiency) | 0.7589 | 0.7715 | 0.7584 | 0.7634 | 0.7606 | 0.7640 | 0.7594 | 0.7495 | 0.7468 | 0.7702 |
| Discharge Temperature (°F) | 112.4 | 105.7 | 108.4 | 100.6 | 108.4 | 100.4 | 107.8 | 107.1 | 100.7 | 106.1 |
| Kilowatt hours/ton | 0.6136 | 0.6532 | 0.6382 | 0.6269 | 0.6378 | 0.6273 | 0.6285 | 0.6319 | 0.6377 | 0.6360 |
| RPM for 20-in. Impeller | 7094 | 7197 | 6899 | 7606 | 6844 | 7575 | 7817 | 7444 | 8084 | 7752 |
| Tons for 20-in. Impeller | 371.0 | 2038.6 | 643.6 | 713.4 | 664.3 | 724.7 | 770.4 | 351.7 | 342.3 | 1824.5 |
| Tonnage Compared with R-11 (%) | 100.0 | 549.5 | 173.5 | 192.3 | 179.1 | 195.3 | 207.6 | 94.8 | 92.3 | 491.8 |
| Tonnage Compared with R-134a (%) | 18.2 | 100.0 | 31.6 | 35.0 | 32.6 | 35.5 | 37.8 | 17.2 | 16.8 | 89.5 |