EXPERIMENTAL PERFORMANCE OF OZONE-SAFE ALTERNATIVE REFRIGERANTS

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ABSTRACT

Several compounds proposed as near-term or longer range substitutes for the regulated chlorofluorocarbon (CFC) refrigerants were tested in a breadboard vapor-compression circuit, and their performance was evaluated relative to more commonly used refrigerants. The limited physical property information available in the literature for these alternative compounds was used to fit an equation of state so coefficients of performance (COP) and capacities calculated from refrigerant property subroutines could be compared to those obtained experimentally.

Comparisons of measured and modeled performance are given for 11 alternatives and for R22, R12, and R114. Estimates of compressor efficiency with each refrigerant are provided. Several of the alternatives exhibited better performance than the more widely used refrigerants at some or all of the conditions tested. Ozone-safe, alternative refrigerants that performed better than CFC counterparts at selected conditions are R152a, R143a, R134a, R134, and R142b.

INTRODUCTION

Restrictions on the production of chlorine-containing, fully halogenated refrigerants and the likely prospect that these restrictions will become more limiting have prompted a search for environmentally acceptable alternatives. Many early substitute compounds were identified by similarities in normal boiling points and corresponding, saturated vapor pressure characteristics.

Additional concern about warming of the global environment through the "greenhouse effect" has necessitated selection of compounds with shorter atmospheric lifetimes and higher energy efficiencies as substitute refrigerants for heating, air-conditioning, and refrigerating equipment.

Several ozone-safe alternatives for chlorofluorocarbons (CFCs) were identified as potential components for a nonazeotropic refrigerant mixture (NARM) to be used for heat pump applications (Vineyard et al. 1989). The CFC refrigerants and HFC (hydrofluorocarbon: hydrogen-containing fluorocarbons), HCFC (hydrochlorofluorocarbon: alkylhalides with substituted hydrogen, chlorine, and fluorine), and FC (fluorocarbon: fully fluorinated compounds) alternatives are listed in Table 1. Refrigerants R32, R125, R143a, R134a, R152a, R134, R124, R142b, and R143 were identified as the most attractive alternative candidates. Performance data are available for some of these compounds (ASHRAE 1985), but the open literature does not contain a concise comparison of modeled and experimentally measured data because of insufficient physical property information and because the materials are not generally available for system testing. Sample quantities were obtained from chemical producers and specialty chemical supply houses. These were experimentally tested in a breadboard refrigeration test loop at operating conditions typical of a residential heat pump.

TEST FACILITY

A schematic diagram of the alternative refrigerants calorimeter (ARC) test rig is shown in Figure 1. The refrigerant circuit contains the following major components:

Compressor
—Hermetic/reciprocating
—1.247 in.^{3} (0.0204 L) displacement
—11,000 Btu/h (3224 W) rated for R22 at
—130°F (54.4°C) condensing temperature
—45°F (7.2°C) evaporating temperature
—95°F (35°C) ambient temperature

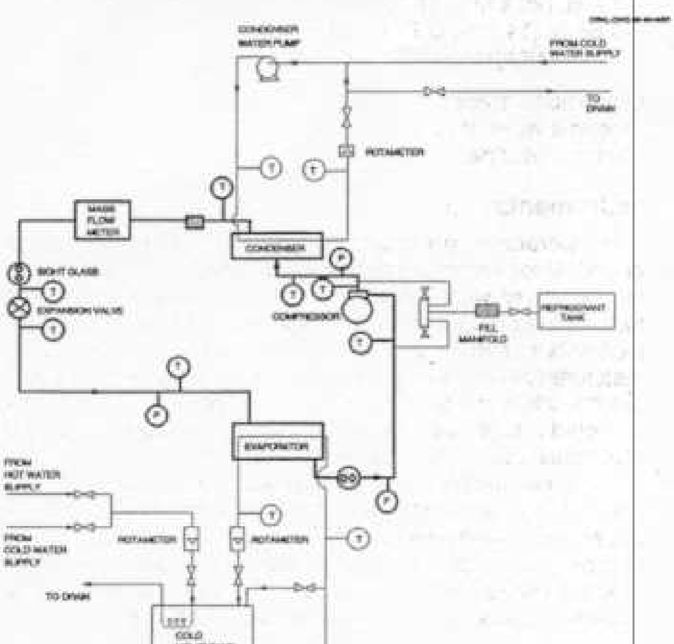

Figure 1 Schematic diagram of the alternative refrigerants test loop

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TABLE 1
CFC Refrigerants and Alternatives

CFC Refrigerants					CFC Alternatives				
Name	Formula	NBP°F (°C)[a]	ODP[b]	GWP[c]	Name	Formula	NBP°F (°C)[a]	ODP[b]	GWP[c]
R113	$CFCl_2CF_2Cl$	118 (48)	0.9	1.4	R141b	CCl_2FCH_3	90 (32)	0.08	0.09
R11	CCl_3F	75 (24)	1.0	1.0	R123/123a	C_2HCl_2	81 (27)	0.02	0.02
					R143	CHF_2CH_2F	41 (5)	0	—
					RC318	$\overline{CF_2CF_2CF_2CF_2}$	21 (−5)	0	—
					R142b	$CClF_2CH_3$	16 (−9)	0.06	0.39
R114	$CClF_2CClF_2$	39 (4)	0.8	3.9	R124	$CHClFCF_3$	10 (−12)	0.02	0.10
					R134	CHF_2CHF_2	−4 (−20)	0	—
					R152a	CHF_2CH_3	−13 (−25)	0	0.03
R12	CCl_2F_2	−22 (−30)	1.0	3.1	R134a	CF_3CH_2F	−17 (−27)	0	0.30
R500	CCl_2F_2/CHF_2CH_3	−27 (−33)	<0.8	<2.4	R218	$CF_3CF_2CF_3$	−35 (37)	0	—
R115	$CClF_2CF_3$	−38 (−39)	0.5	7.6	R22	$CHClF_2$	−42 (−41)	0.05	0.35
					R143a	CF_3CH_3	−54 (−48)	0	0.75
					R125	CF_3CHF_2	−54 (−48)	0	0.60
R502	$CHClF_2/CClF_2CF_3$	−49 (−45)	<0.3	<4.1	R32	CH_2F_2	−62 (−52)	0	—
					R23	CHF_3	−116 (−82)	0	—

a. Normal boiling point
b. Ozone depletion potential relative to R11 = 1.0, scientific assessment of ozone: 1989
c. Global warming potential relative to R11 = 1.0, scientific assessment of ozone: 1989

Condenser and Evaporator
—Coaxial, tube-in-tube, turbulators on refrigerant side
—10 ft (3 m) total tube length
—Counterflow arrangement
—1.0 in. (25.4 mm) O.D. refrigerant tube
—0.56 in. (14 mm) O.D. water tube
—15,000 Btu/h (4395 W) nominal rating

Expansion Valves
—needle valve type
—micrometer handle

Instrumentation

Temperatures, pressures, and flow rates were measured at critical locations in the system. In the refrigerant circuit, temperatures were measured at the inlet and outlet of the four major components: the compressor, condenser, expansion valve, and evaporator. Refrigerant pressures were measured between each of these major components. A coriolis mass flow meter was used at the condenser exit to monitor liquid refrigerant flow rate. Compressor electrical power was measured using a watt transducer.

In the secondary liquid circuits (water or water-glycol, as appropriate), temperatures were measured at the inlet and outlet of the evaporator and condenser. Rotameters (a total of three) were used to monitor secondary liquid flow rates. Figure 1 shows the locations of all sensors. All control valves and the expansion valve were manually operated.

Data Acquisition System

The data acquisition system consisted of a desktop computer, digital voltmeter, multiprogrammer, multiprogrammer interface, digital clock, printer, and scanner. Voltage signals were read from the instrumentation by the digital voltmeter. The multiprogrammer was the master control unit for input/output cards. Bidirectional communication between the multiprogrammer and the computer was performed through a multiprogrammer interface. The scanner was a channel selector that housed input connectors for analog transducers.

REFRIGERANTS

Whenever possible, refrigerant samples used for the test program were obtained from commercial refrigerant manufacturers. Where this was not possible, small (4- to 5-lb [2.25 kg]) samples were ordered from a specialty chemical supply house.

Fluids obtained from specialty houses are much more expensive and may contain two orders of magnitude more impurities than refrigerants available through refrigeration supply stores. Small levels of impurities would be a much more important consideration if these fluids were to be used for precise thermodynamic or thermophysical property measurements, material comparability evaluations, or toxicity testing. Compounds obtained in this manner were charged into the test rig through a new filter dryer core to help remove acidic impurities and excess moisture that would be detrimental to the hermetic refrigerant circuit.

TEST PROCEDURES

Operating conditions simulating the Department of Energy's 17°F/47°F (-8.3°C/8.3°C) heating and 82°F/95°F (28°C/ 35°C) cooling heat pump rating conditions were chosen to evaluate these alternative fluids (Vineyard et al. 1989). Entering temperatures for the secondary heat transfer liquids were adjusted to give discharge and suction saturation pressures with R22 equivalent to those measured by Miller (1982) for an R22 air-source system. Liquid-refrigerant heat exchangers were used on the breadboard rig to maintain better temperature control and to permit more precise measurement of condenser and evaporator heat transfer loads. Temperature differences across the inlet and outlet for the sink and source fluids were chosen to be representative of typical air-side temperature differences for air-source heat pumps. Secondary fluid operating conditions for alternative refrigerant testing are summarized in Table 2.

System tests were performed with R22 to establish secondary fluid entering temperatures. Additional runs were performed with R12 and R114 to provide reference data for known refrigerants.

The micrometer expansion valve settings and system charge were adjusted to obtain 1-2°F (0.5-1.0°C) superheat out of the evaporator and 4-8°F (2-4°C) refrigerant subcooling out of the condenser. Conditions of low system superheat and subcooling were verified by checking saturated refrigerant property tables when these were available. When reliable refrigerant property data were not published, the system expansion valve and charge size were controlled to sustain

TABLE 2
Operating Conditions for Alternative Refrigerant Testing Alternative Refrigerants Calorimeter

17 °F (-8.3 °C) Heating Condition	Evaporator [°F(°C)]	Condenser [°F(°C)]
Entering Glycol/Water Temperature	30 (-1.1)	62 (16.7)
Glycol/Water ΔT Across Heat Exchanger	10 (5.6)	20 (11.1)
47 °F (8.3 °C) Heating Condition		
Entering Water Temperature	51 (10.6)	68 (20)
Water ΔT Across Heat Exchanger	10 (5.6)	20 (11.1)
82 °F (27.8 °C) Cooling Condition		
Entering Water Temperature	77 (25)	84 (28.9)
Water ΔT Across Heat Exchanger	20 (11.1)	20 (11.1)
95 °F (35 °C) Cooling Conditions		
Entering Water Temperature	80 (26.7)	98 (36.7)
Water ΔT Across Heat Exchanger	20 (11.1)	20 (11.1)

saturated refrigerant gas at the evaporator exit sight glass and saturated refrigerant liquid at the condenser exit sight glass. This was fairly easy to do if rough estimates of the saturated vapor pressure could be made from the normal boiling point of a compound.

DATA ANALYSIS

System performance values were obtained by measuring the volumetric flow rate and the inlet and outlet temperatures of secondary heat transfer liquids through the evaporator and condenser. Experimental heating and cooling COPs were calculated by

$$COPH_e = \frac{(\dot{m}C_p\Delta T)_{cn}}{P_{cm}} \tag{1}$$

$$COPR_e = \frac{(\dot{m}C_p\Delta T)_{ev}}{P_{cm}} \tag{2}$$

where

$COPH_e$ = experimentally measured heating COP
$COPR_e$ = experimentally measured cooling COP
$\dot{m}$ = mass flow rate of the secondary fluid
C_p = specific heat of the secondary fluid
ΔT = temperature difference across the heat exchanger
P_{cm} = compressor power
cn = condenser
ev = evaporator

Modeled COPs were obtained through refrigerant property data available in the literature. Accessible property data were used to estimate Carnahan-Starling-DeSantis (CSD) equation of state coefficients (Morrison 1986). Enthalpy changes of the refrigerant across the appropriate heat exchanger were calculated from measured temperatures and pressures using the CSD refrigerant property routines. Modeled COP values were then calculated by

$$COPH_m = \frac{\Delta h_{cn}\dot{m}_r}{P_{cm}} \tag{3}$$

$$COPR_m = \frac{\Delta h_{ev}\dot{m}_r}{P_{cm}} \tag{4}$$

where

$COPH_m$ = modeled heating coefficient of performance
$COPR_m$ = modeled cooling coefficient of performance
Δh_{cn} = refrigerant enthalpy change across condenser
Δh_{ev} = refrigerant enthalpy change across evaporator

TABLE 3
Comparable Refrigerant Performance: Experimental Results (at ARI 17°F Heat Pump Rating Condition)[1]

	COMPRESSOR SUCTION TEMPERATURE (°F)	EVAPORATING PRESSURE (psia)	CONDENSING PRESSURE (psia)	COMPRESSION RATIO	NET HEATING EFFECT (Btu/lb)	REFRIGERANT CIRCULATED (lb/min/kW)	COEFFICIENT OF PERFORMANCE (HEATING)
R32	14.1	81.7	311.7	3.97	124.1 [2]	0.459 [2]	2.38
R125	13.7	64.7	247.7	4.11	56.6 [2]	1.00 [2]	2.40
R143a	14.0	60.8	223.4	3.89	78.4 [2]	0.726 [2]	2.48
R22	11.8	48.9	183.1	3.97	95.3 [3]	0.597 [3]	2.46
R218	16.7	38.1	149.0	3.90	36.5 [2]	1.56 [2]	2.03
R12	14.2	31.9	111.4	3.84	73.8 [2]	0.771 [2]	2.32
R134a	17.6	29.2	113.6	4.22	94.7 [2]	0.601 [2]	2.30
R152a	16.5	27.1	101.1	3.73	135.8 [3]	0.419 [3]	2.38
R134	17.6	22.8	86.8	3.39	94.7 [3]	0.601 [3]	2.23
R142b	18.5	15.7	55.2	3.98	104.0 [3]	0.547 [3]	2.25
R114	24.0	11.87	30.7	3.36	101.9 [2]	0.559 [2]	1.59
R143	20.4	9.9	39.0	4.59	123.9 [3]	0.459 [3]	1.71

(1) ROUGHLY 11°F EVAPORATION AND 86°F CONDENSATION
(2) MEASURED REFRIGERANT FLOW RATE
(3) REFRIGERANT FLOW RATE CALCULATED FROM EQUATION OF STATE

TABLE 3A
Comparable Refrigerant Performance: Experimental Results
(At ARI −8.3°C Heat Pump Rating Condition)[1]

	COMPRESSOR SUCTION TEMPERATURE (°C)	EVAPORATING PRESSURE (kPa)	CONDENSING PRESSURE (kPa)	COMPRESSION RATIO	NET HEATING EFFECT (kJ/kg)	REFRIGERANT CIRCULATED (g/s/kw)	COEFFICIENT OF PERFORMANCE (HEATING)
R32	-9.9	563	2149	3.97	268.5 [2]	3.47 [2]	2.38
R125	-10.2	446	1708	4.11	132.0 [2]	7.56 [2]	2.40
R143a	-10.0	419	1540	3.89	182.0 [2]	5.49 [2]	2.48
R22	-11.2	337	1262	3.97	222.0 [3]	4.51 [3]	2.48
R218	-8.5	194	1027	3.90	84.8 [2]	11.80 [2]	2.03
R12	-9.9	220	768	3.84	172.0 [2]	5.83 [2]	2.32
R134a	-8.0	201	783	4.22	220.0 [2]	4.54 [2]	2.30
R152a	-8.6	287	697	3.73	316.0 [3]	3.17 [3]	2.38
R134	-8.0	157	598	3.39	220.0 [3]	4.54 [3]	2.23
R142b	-7.5	108	380	3.98	242.0 [3]	4.13 [3]	2.25
R114	-4.4	82.0	212	3.36	237.0 [2]	4.23 [2]	1.59
R143	-6.4	68	269	4.59	268.0 [3]	3.47 [3]	1.71

(1) ROUGHLY -11.7°C EVAPORATION AND 30°C CONDENSATION
(2) MEASURED REFRIGERANT FLOW RATE
(3) REFRIGERANT FLOW RATE CALCULATED FROM EQUATION OF STATE

$\dot{m}_r$ = measured refrigerant mass flow rate
P_{cm} = compressor power

All of these COP results are affected by the efficiency of the compressor used in the test loop. Compressor isentropic efficiency for each refrigerant was calculated as follows:

$$\eta_i = \frac{\dot{m}_r \Delta h_i}{P_{cm}} \tag{5}$$

where

η_i = compressor isentropic efficiency
$\dot{m}_r$ = measured refrigerant mass flow rate
Δh_i = isentropic enthalpy rise across compressor calculated by refrigerant property routines
P_{cm} = compressor power.

RESULTS

Tables 3 through 6 summarize experimentally measured temperatures and pressures in the test loop, experimental compression ratios, net heating or cooling effects, refrigerant mass flow rates needed to achieve a stated amount of heating or cooling, and COPs computed as indicated in Equations 1 and 2. These tables are similar in format to those indicating comparative refrigerant performance in Chapter 16 of *ASHRAE Fundamentals* (ASHRAE 1985).

Compression ratio values are calculated from actual compressor suction and discharge pressures, so they may not correlate with the ratio of condensing and evaporating pressures. Net refrigerant heating and cooling effects in Tables 3 through 6 are calculated by dividing measured or calculated refrigerant mass flow rates into the heat transferred by the condenser or evaporator secondary fluid, respectively.

Table 3 contains no data for R124 and RC318 because different evaporator inlet conditions were used for the 17°F (−8.3°C) runs with these two refrigerants and performance comparisons would be inappropriate. Tables 5 and 6 contain no experimental data for R32, R125, and R143a because these refrigerants produce excessively high condensing pressures under operating conditions summarized in these tables.

A comparison between "experimental" COPs calculated by Equation 1 or 2 and "modeled" COPs calculated by Equation 3 or 4 is shown in Table 7. COPH and COPR refer to heating and cooling COPs calculated from condenser or evaporator data, respectively. The original intent of this comparison was to check on the validity of CSD equation-of-state coefficients calculated from sketchy physical property data. Large discrepancies between experimental and modeled values for alternative refrigerants (such as the 95°F cooling condition for R134a) can be attributed to poor estimates for the equation-of-state coefficients.

Differences between the modeled and experimental values for the more common refrigerants such as R22, R12, and R114, which have more reliable CSD coefficients, are probably due to inaccuracies in the refrigerant mass flow rate measurements. Small variations in refrigerant flow rate have a relatively large effect on the latent heat contibution to this calculation, and the meter used for these measurements was particularly difficult to zero with each new fluid. The experimental values in Table 7 are considered more reliable.

Table 8 lists the isentropic compressor efficiencies calculated for alternative and known refrigerants in these system property data. Uncertainties and inaccuracies discussed above contribute to the scatter seen in these points, but they are presented here to give an indication of relative compressor efficiencies for each fluid.

DISCUSSION

Many of the compounds selected for this study do not possess all of the attributes desirable for an ideal refrigeration

TABLE 4
Comparable Refrigerant Performance: Experimental Results (at ARI 47°F Heat Pump Rating Condition)[1]

	COMPRESSOR SUCTION TEMPERATURE (°F)	EVAPORATING PRESSURE (psia)	CONDENSING PRESSURE (psia)	COMPRESSION RATIO	NET HEATING EFFECT (Btu/lb)	REFRIGERANT CIRCULATED (lb/min/kW)	COEFFICIENT OF PERFORMANCE (HEATING)
R32	29.6	109.0	346.9	3.26	125.8 [2]	0.452 [2]	2.76
R125	41.7	93.2	279.3	3.25	50.4 [2]	1.13 [2]	2.72
R143a	37.5	85.1	252.0	3.15	71.3 [2]	0.799 [2]	2.90
R22	32.0	73.3	210.3	3.03	89.1 [3]	0.639 [3]	3.17
R218	49.5	60.1	169.0	3.18	37.6 [2]	1.51 [2]	2.67
R12	32.4	47.5	124.8	2.86	64.3 [2]	0.886 [2]	2.94
R134a	36.9	45.7	133.8	3.16	88.1 [2]	0.646 [2]	3.26
R152a	39.4	41.2	117.4	2.72	132.2 [3]	0.431 [3]	3.40
R134	45.7	34.0	98.9	3.19	93.1 [3]	0.611 [3]	3.15
R124	38.9	23.9	73.1	3.43	69.3 [3]	0.821 [3]	2.16
R142b	47.3	23.4	61.5	2.77	104.5 [3]	0.544 [3]	2.98
RC318	45.2	27.0	57.0	2.50	58.6 [2]	0.971 [2]	1.76
R114	34.9	14.8	36.3	3.00	78.5 [2]	0.725 [2]	1.83
R143	36.7	14.7	43.7	3.28	128.8 [3]	0.442 [3]	2.34

(1) ROUGHLY 28°F EVAPORATION AND 94°F CONDENSATION
(2) MEASURED REFRIGERANT FLOW RATE
(3) REFRIGERANT FLOW RATE CALCULATED FROM EQUATION OF STATE

fluid (nonflammability, low greenhouse potential, etc.). However, combinations of them may be blended together to form azeotropes, near azeotropes, or NARMs that match the intended application. An undesirable characteristic of one component in a mixture may be overcome by sensible selection of the other constituents. NARMs also offer the potential of improved thermodynamic cycle efficiencies.

R218 (perfluoropropane) and RC318 (perfluorocyclobutane) have been proposed as chlorine-free alternatives for R22 and R114, respectively, based on the proximity of their normal boiling points (see Table 1). The relatively poor performance measured for both of these fully fluorinated molecules is consistent with the concept that larger molecules have more degrees of internal vibrational freedom that result in larger volumetric heat capacities and higher system throttling losses in simple vapor compression systems. Also, less polarizable molecules, such as those that are fully fluorinated, have smaller inter-molecular forces that reduce the latent heat of vaporization and their net refrigerating (or heating) efficiencies (Eiseman 1968; Downing 1988). This fully fluorinated structure also favors long atmospheric lifetimes, which increases their greenhouse potential compared to other alternative refrigerants.

For R134 and R143, estimates of saturated vapor pressures needed to set up experimental runs and calculate CSD equation-of-state coefficients were estimated from their normal boiling points. Log-pressure vs. reciprocal-temperature straight-line plots were constructed that passed through the boiling points for R134 and R143 at atmospheric pressure and ran parallel to plotted vapor pressure data for their structural isomers, R134a and R143a, respectively.

The SUPERTRAPP® properties estimation program (McLinden 1989), which is available through the Office of Standards and Reference Data at the National Institute of Standards and Technology (NIST, formerly NBS), was later used to supplement data generated for R143. It remains to be seen how accurate these initial estimates are.

Results for R134 and R143 in Tables 3 through 6 support some of the favorable performance predictions made by Hodgett during the CFC forum at the 1988 IIR Conference at a U.S. university (Kuijpers et al. 1989). The R143 sample tested gave better COPs than R114 at every test condition. However, definitive comparisons between these two refrigerants are not possible here because of the purity of the R143 sample. Gas chromatographic analysis indicated that it contained 10 to 15 vol.% of a more volatile component. Relative retention times indicate this impurity is R143a. Refrigerant samples of higher purity and more reliable physical property information are needed for more accurate evaluations of R143.

CONCLUSIONS

For the following conclusions it must be stressed that the experimental results are for tests done on one specific piece of hardware, and they may not be generalized to all situations. In addition, the tests reported in this paper were for simulated heat pump conditions. Thus the relative rankings of alternatives may not hold true for conditions typical of other applications, such as household or commercial refrigeration, commercial chillers, etc. Similar tests should be conducted for conditions simulating those other applications.

Specific conclusions drawn from the data presented in Tables 3 through 6 are given below:

—R152a yielded the best measured performance at all test conditions except the lowest evaporator temperature.

TABLE 4A
Comparable Refrigerant Performance: Experimental Results
(at ARI 8.3°C Heat Pump Rating Condition)[1]

	COMPRESSOR SUCTION TEMPERATURE (°C)	EVAPORATING PRESSURE (kPa)	CONDENSING PRESSURE (kPa)	COMPRESSION RATIO	NET HEATING EFFECT (kJ/kg)	REFRIGERANT CIRCULATED (g/s/kW)	COEFFICIENT OF PERFORMANCE (HEATING)
R32	-1.3	751	2392	3.26	292.0 (2)	3.42 (2)	2.76
R125	5.4	642	1926	3.25	117.0 (2)	8.54 (2)	2.72
R143a	3.0	567	1737	3.15	166.0 (2)	6.04 (2)	2.90
R22	0.0	505	1450	3.03	207.0 (3)	4.83 (3)	3.17
R218	9.7	414	1165	3.18	87.4 (2)	11.4 (2)	2.67
R12	0.2	327	860	2.86	149.0 (2)	6.78 (2)	2.94
R134a	2.7	315	922	3.16	205.0 (2)	4.88 (2)	3.26
R152a	4.1	284	809	2.72	307.0 (3)	3.26 (3)	3.40
R134	7.6	234	682	3.19	216.0 (3)	4.62 (3)	3.15
R124	3.8	165	504	3.43	161.0 (3)	6.21 (3)	2.16
R142b	8.5	161	424	2.77	243.0 (3)	4.11 (3)	2.96
RC318	7.3	166	393	2.50	136.0 (2)	7.34 (2)	1.76
R114	1.6	102	250	3.00	182.0 (2)	5.48 (2)	1.83
R143	2.6	101	301	3.28	299.0 (3)	3.34 (3)	2.34

(1) ROUGHLY -2.2°C EVAPORATION AND 34.4°C CONDENSATION
(2) MEASURED REFRIGERANT FLOW RATE
(3) REFRIGERANT FLOW RATE CALCULATED FROM EQUATION OF STATE

—R143a was the best performing alternative at the lowest evaporator temperature.
—R134a outperformed R12 at all but the lowest temperature conditions.
—R134 outperformed R134a at the two highest temperature conditions.

Both R152a and R143a are classified as flammable. This characteristic could be ameliorated by combining them with a nonflammable fluid (or fluids) to produce a nonflammable mixture.

The thermodynamic performance of promising alternatives (pure fluids and mixtures) can be estimated from a minimal amount of physical property information. In cases where good performance is indicated, small portions of the most promising materials can be tested at standard conditions and compared to established refrigerants and modeled predictions.

Research efforts are required to measure the physical, chemical, and thermodynamic properties of compounds and mixtures of compounds that demonstrate efficient performance as vapor-compression refrigerants.

Beyond thermodynamic considerations, thermophysical, toxicological, and extensive material compatibility information is necessary to fully determine the utility of new refrigerants or refrigerant mixtures in refrigerating or air-conditioning equipment.

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TABLE 5
Comparable Refrigerant Performance: Experimental Results
(at ARI 82°F Heat Pump Rating Condition)[1]

	COMPRESSOR SUCTION TEMPERATURE (°F)	EVAPORATING PRESSURE (psia)	CONDENSING PRESSURE (psia)	COMPRESSION RATIO	NET COOLING EFFECT (Btu/lb)	REFRIGERANT CIRCULATED (lb/min/ton)	COEFFICIENT OF PERFORMANCE (COOLING)
R22	50.1	94.3	264.3	2.97	65.6 [3]	3.05 [3]	2.50
R218	64.0	84.7	220.7	2.94	23.8 [2]	8.42 [2]	2.02
R12	57.7	61.0	156.7	2.78	49.9 [2]	4.00 [2]	2.32
R134a	50.7	61.0	171.9	3.02	56.9 [2]	3.51 [2]	2.51
R152a	53.4	56.2	152.0	2.71	108.9 [3]	1.84 [3]	2.93
R134	64.5	46.6	131.8	2.95	67.9 [3]	2.94 [3]	2.75
R124	57.1	39.7	100.7	2.71	54.0 [2]	3.71 [2]	2.25
R142b	64.8	32.5	88.4	2.78	81.1 [3]	2.47 [3]	2.64
RC318	58.7	31.7	82.9	3.00	36.9 [2]	5.43 [2]	2.04
R114	52.0	18.6	50.3	2.70	46.7 [3]	4.28 [3]	1.42
R143	55.3	22.2	59.7	2.84	103.9 [3]	1.93 [3]	2.22

(1) ROUGHLY 48°F EVAPORATION AND 105°F CONDENSATION
(2) MEASURED REFRIGERANT FLOW RATE
(3) REFRIGERANT FLOW RATE CALCULATED FROM EQUATION OF STATE

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TABLE 5A
Comparable Refrigerant Performance: Experimental Results
(at ARI 27.8°C Heat Pump Rating Condition)[1]

	COMPRESSOR SUCTION TEMPERATURE (°C)	EVAPORATING PRESSURE (kPa)	CONDENSING PRESSURE (kPa)	COMPRESSION RATIO	NET COOLING EFFECT (kJ/kg)	REFRIGERANT CIRCULATED (g/s/kW)	COEFFICIENT OF PERFORMANCE (COOLING)
R22	10.0	650	1822	2.97	152.0 [3]	6.56 [3]	2.50
R218	17.8	584	1522	2.94	55.3 [2]	18.10 [2]	2.02
R12	14.3	420	1080	2.78	116.0 [2]	8.60 [2]	2.32
R134a	10.4	420	1185	3.02	132.0 [2]	7.55 [2]	2.51
R152a	11.9	387	1048	2.71	253.0 [3]	3.96 [3]	2.93
R134	18.0	335	909	2.95	158.0 [3]	6.32 [3]	2.75
R124	13.9	274	694	2.71	126.0 [2]	7.96 [2]	2.25
R142b	18.2	224	609	2.78	189.0 [3]	5.31 [3]	2.64
RC318	14.8	218	571	3.00	85.8 [2]	11.70 [2]	2.04
R114	11.1	128	347	2.70	109.0 [3]	10.40 [3]	1.42
R143	12.9	153	412	2.84	242.0 [3]	4.15 [3]	2.22

(1) ROUGHLY 8.9°C EVAPORATION AND 40.6°C CONDENSATION
(2) MEASURED REFRIGERANT FLOW RATE
(3) REFRIGERANT FLOW RATE CALCULATED FROM EQUATION OF STATE

ORNL-DWG 89-15308D

TABLE 6
Comparable Refrigerant Performance: Experimental Results (at ARI 95°F Heat Pump Rating Condition)[1]

	COMPRESSOR SUCTION TEMPERATURE (°F)	EVAPORATING PRESSURE (psia)	CONDENSING PRESSURE (psia)	COMPRESSION RATIO	NET COOLING EFFECT (Btu/lb)	REFRIGERANT CIRCULATED (lb/min/ton)	COEFFICIENT OF PERFORMANCE (COOLING)
R22	52.2	103.4	313.8	3.20	62.1 [3]	3.22 [3]	2.07
R218	60.1	93.9	257.0	3.06	17.7 [2]	11.32 [2]	1.52
R12	55.2	66.1	187.9	3.05	46.6 [2]	4.29 [2]	2.26
R134a	57.5	65.3	202.7	3.33	44.5 [2]	4.50 [2]	2.39
R152a	56.3	58.0	181.7	3.03	104.0 [3]	1.92 [3]	2.60
R134	67.4	54.3	159.4	3.35	64.3 [3]	3.11 [3]	2.45
R124	57.5	40.8	118.3	3.19	41.9 [2]	4.77 [2]	1.94
R142b	58.1	34.9	100.2	3.09	77.8 [3]	2.57 [3]	2.40
RC318	59.9	35.6	96.6	3.09	28.9 [2]	6.93 [2]	1.40
R114	57.3	23.0	61.4	2.69	42.6 [2]	4.70 [2]	1.58
R143	57.5	23.6	74.7	3.34	99.5 [3]	2.01 [3]	1.91

(1) ROUGHLY 52°F EVAPORATION AND 122°F CONDENSATION
(2) MEASURED REFRIGERANT FLOW RATE
(3) REFRIGERANT FLOW RATE CALCULATED FROM EQUATION OF STATE

ORNL-DWG 89-15430D

TABLE 6A
Comparable Refrigerant Performance: Experimental Results (at ARI 35°C Heat Pump Rating Condition)[1]

	COMPRESSOR SUCTION TEMPERATURE (°C)	EVAPORATING PRESSURE (KPa)	CONDENSING PRESSURE (KPa)	COMPRESSION RATIO	NET COOLING EFFECT (kJ/kg)	REFRIGERANT CIRCULATED (g/s/kW)	COEFFICIENT OF PERFORMANCE (COOLING)
R22	11.2	713	2163	3.20	144.0 [3]	6.92 [3]	2.07
R218	15.6	647	1772	3.06	41.1 [2]	24.3 [2]	1.52
R12	12.9	456	1295	3.05	108.0 [2]	9.22 [2]	2.26
R134a	14.2	450	1397	3.33	103.0 [2]	9.68 [2]	2.39
R152a	13.5	400	1253	3.03	242.0 [3]	4.13 [3]	2.60
R134	19.7	374	1099	3.35	149.0 [3]	6.69 [3]	2.45
R124	14.2	281	816	3.19	97.4 [2]	11.1 [2]	1.94
R142b	14.5	241	691	3.09	181.0 [3]	5.53 [3]	2.40
RC318	15.5	245	666	3.09	67.2 [2]	14.9 [2]	1.40
R114	14.0	158	423	2.69	99.0 [2]	10.1 [2]	1.58
R143	14.2	163	515	3.34	231.0 [3]	4.32 [3]	1.91

(1) ROUGHLY 11.1°C EVAPORATION AND 50°C CONDENSATION
(2) MEASURED REFRIGERANT FLOW RATE
(3) REFRIGERANT FLOW RATE CALCULATED FROM EQUATION OF STATE

TABLE 7
COP Comparisons for Modeled and Experimental Performance Refrigerant Data from ARC Rig

	17°F (-8.3°C) HEATING		47°F (8.3°C) HEATING		82°F (27.8°C) COOLING		95°F (35°C) COOLING	
	COPH	(COPR)	COPH	(COPR)	(COPH)	COPR	(COPH)	COPR
R32								
MODELED	2.8	2.3	3.2	2.5	—	—	—	—
EXPERIMENTAL	2.4	1.7	2.8	2.2	—	—	—	—
R125								
MODELED	2.4	1.6	2.8	2.0	—	—	—	—
EXPERIMENTAL	2.4	1.7	2.7	2.1	—	—	—	—
R143A								
MODELED	2.8	2.1	3.3	2.8	—	—	—	—
EXPERIMENTAL	2.5	1.7	2.9	2.3	—	—	—	—
R22								
MODELED	2.3	1.7	3.3	2.6	3.7	2.9	3.0	2.3
EXPERIMENTAL	2.5	1.8	3.2	2.5	3.2	2.5	2.8	2.1
R218								
MODELED	2.1	1.3	2.7	2.0	2.9	2.2	2.5	1.7
EXPERIMENTAL	2.0	1.0	2.7	2.0	2.8	2.0	2.3	1.5
R12								
MODELED	2.0	1.5	2.9	2.4	3.2	2.5	2.9	2.3
EXPERIMENTAL	2.3	1.4	2.9	2.1	3.3	2.3	2.9	2.3
R134a								
MODELED	2.1	1.6	3.1	2.5	3.4	2.8	4.2	3.2
EXPERIMENTAL	2.3	1.4	3.3	2.4	3.3	2.5	3.1	2.4
R152a								
MODELED	2.0	1.6	3.1	2.6	3.8	2.9	3.3	2.8
EXPERIMENTAL	2.4	1.4	3.4	2.5	3.7	2.9	3.3	2.6
R134								
MODELED	2.0	1.5	2.9	2.2	3.4	2.6	2.1	1.6
EXPERIMENTAL	2.2	1.4	3.2	2.5	3.5	2.6	3.2	2.5
R124								
MODELED	2.7	2.4	1.8	1.4	2.7	2.3	2.6	2.1
EXPERIMENTAL	—	—	2.2	1.2	2.5	2.2	2.6	1.9
R142b								
MODELED	2.0	1.4	3.0	2.3	3.7	3.0	3.5	2.8
EXPERIMENTAL	2.2	1.3	3.0	2.2	3.5	2.6	3.3	2.4
RC318								
MODELED	1.6	1.6	1.2	1.1	2.3	1.9	2.2	1.6
EXPERIMENTAL	—	—	1.6	1.4	2.6	2.0	2.3	1.4
R114								
MODELED	1.1	0.7	1.5	1.1	2.2	1.6	2.2	1.7
EXPERIMENTAL	1.6	0.8	1.8	1.1	2.3	1.4	2.3	1.6
R143								
MODELED	1.6	1.3	2.4	2.0	2.9	2.4	2.8	2.2
EXPERIMENTAL	1.7	1.3	2.3	1.5	3.2	2.2	2.9	1.9

TABLE 8
Compressor Isentropic Efficiency Calculated from Refrigerant Properties at Various Rating Points

Refrigerant	Calculated Isentropic Efficiencies (%)			
	17°F (-8.3°C) Heating	47°F (8.3°C) Heating	82°F (27.8°C) Cooling	95°F (35°C) Cooling
R32	40	41	—	—
R125	44	45	—	—
R143a	39	40	—	—
R22	37	45	44	43
R218	45	38	38	37
R12	25	35	39	45
R134a	31	39	45	44
R152a	30	37	42	44
R134	29	37	41	43
R124	—	23	33	38
R142b	25	28	35	37
RC318	—	16	31	38
R114	15	17	28	33
R143	19	23	30	31

DISCUSSION

David Zietlow, Research Assistant, University of Illinois, Urbana: How did you clean the system when changing from one refrigerant to another?

J.R. Sand: Refrigerant from the completed test was drawn out of the system into containers cooled with dry ice. The test loop was then held under pumped vacuum overnight before a new refrigerant charge was added.

A 150-second (SUS), synthetic, alkylbenzene oil was used in the compressor for these alternate refrigerant tests to help minimize miscibility problems.