CONF-981107--

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AUG 13 1998

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This work was supported by the Environmental Technology & Education Center (ETEC) under User Agreement UR-97-108 with Lockheed Martin Energy Research Corporation (LMER). The work was performed at Oak Ridge National Laboratory, managed by LMER under contract DE-AC05-96OR22464 with the U. S. Department of Energy.

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EXPERIMENTAL COMPARISON OF A NEARLY AZEOTROPIC REFRIGERANT BLEND AND R-22 IN A WATER-TO-WATER LABORATORY TEST RIG

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ABSTRACT

A newly developed, non-ozone-depleting refrigerant blend containing a fluoroiodocarbon compound was compared with R-22 in a laboratory-scale, water-to-water refrigeration cycle test loop. The loop is equipped with a coaxial tube-in-tube condenser, a baffled shell-in-tube evaporator, and a variable-speed compressor. The blend and R-22 were tested at saturated evaporator temperature conditions of about -10 °F to -20 °F and saturated condenser temperature conditions from about 80 °F to 100 °F. The compressor speed was varied such that evaporator (cooling) capacity was held constant at approximately the same value for both refrigerants for each test condition. To maintain equal capacity, results showed the compressor speed for the blend to be at least 53% greater than that for R-22. Measured system efficiency with the blend was at least 25% better than for R-22 for near equal operating temperature and capacity conditions. When allowing for maximum impact of the experimental uncertainties, the minimum COP improvement observed for the blend was about 10%. Ideal cycle calculations using the best available property estimates for the blend and the compressor efficiencies observed during the tests also suggested a 10% improvement for the blend over R-22 (almost all due to better isentropic efficiency of the compressor with the blend). Better thermodynamic property data are needed for the blend and further investigations are needed to fully assess its potential for refrigeration equipment applications.

INTRODUCTION

In the early 1970s, scientists first presented the theory that tropospherically stable chlorofluorocarbons (CFCs) slowly migrate into the stratosphere where sunlight can split off free chlorine atoms that react with ozone and reduce its concentration (Molina and Rowland 1974). This theory coupled with atmospheric measurements of reduced stratospheric ozone levels, led to ratification of the Montreal Protocol (UNEP, 1987) and subsequent amendments agreed to in London, in 1991, and Copenhagen, in 1995. The Protocol called for cessation of

CFC production for almost all applications (including refrigeration and air-conditioning) in developed countries by January 1, 1996 (developing nations have until 2005 to cease production). In its current form it also calls for cessation of production of hydrochlorofluorocarbons (HCFCs) between 2003 and 2030.

More recently, the issue of rising atmospheric temperatures due to increased concentration of greenhouse gases has gained the attention of international policymakers. The principal greenhouse gas of concern is carbon dioxide but hydrofluorocarbons (HFCs), the leading replacement alternatives for CFCs and HCFCs in most applications, are also under scrutiny. Atmospheric measurements have shown that concentrations of these gases have been increasing. There are concerns that increased levels of greenhouse gases will lead to increases in global average temperatures with potentially severe downside risks for parts of the globe. A new international agreement, the Kyoto Protocol, was negotiated in December, 1997, to rollback emissions of CO₂, and five other greenhouse gases (including HFCs) by 6 to 8% below 1990 levels by 2012 (ACHRN 1997). With CO₂ emissions tied directly to energy use, pressures for improved air-conditioning and refrigeration equipment efficiency will continue to increase.

The primary fluids used as refrigerants worldwide have been R-11, R-12, R-502 (all CFCs), and R-22 (an HCFC). Replacements for these fluids have been investigated for the past decade by industry, government and private labs, and academia. Industry is now producing new and replacement equipment using R-123 (for R-11), R-134a (for R-12), and R-404A and R-507A (for R-502). R-22 is scheduled for phaseout in new equipment beginning in 2010. The leading replacement candidates for R-22 at this time are R-410A and R-407C. All of the foregoing alternatives are HFCs except R-123 (an HCFC now scheduled for phaseout beginning in 2020).

Since HFCs are greenhouse gases, research into alternative fluids is continuing. Much of this work is focused on non-fluorocarbon compounds like ammonia, hydrocarbons, and CO₂ which have much lower global warming potential (GWP). However, work on non-HFC fluorocarbon alternatives is underway also. This paper discusses results

of initial experimental comparisons between a non-ozone-depleting blend containing a fluoriodocarbon compound (the Blend) and R-22. The 100-year GWP of the Blend is about 500 (relative to CO_2 @ GWP=1).

TEST APPARATUS

The Vapor Compression Test Loop (or VCTL), illustrated schematically in Figure 1, consists of a vapor compression refrigerant circuit, evaporator and condenser secondary heat transfer fluid circuits, controls, and instrumentation. A dry cooling tower is used for condenser circuit heat rejection and both secondary circuits use an ethylene glycol and water mixture. The refrigerant circuit compressor was a single-cylinder reciprocating model having a variable speed range of 500 to 3000 rpm. It is driven by an electric motor with adjustable-speed controller. The condenser was of coaxial, tube-in-tube design with refrigerant and secondary circuits in counterflow arrangement. A shell-and-tube heat exchanger was used for the evaporator with refrigerant and secondary circuits in an approximate counterflow arrangement with refrigerant inside the tubes. A pneumatically controlled expansion valve was used for refrigerant flow control. The suction-to-liquid heat exchanger was not used in any of the tests.

Refrigerant and secondary fluid temperatures and flow rates and refrigerant pressures were measured at critical locations in the system. Refrigerant temperatures were measured at entrance and exit of all major components (compressor, evaporator, condenser, and expansion valve). Pressures were measured at compressor suction and discharge, and at the entrance to the evaporator, condenser, and expansion valve. In addition, differential pressures were measured across each heat exchanger. A coriolis mass flow meter in the liquid line was used to monitor refrigerant flow. Power input to the open-drive compressor was measured with a torque/rpm meter mounted on the motor drive shaft.

In the secondary liquid circuits, temperatures were measured at the entrance and exit of each heat exchanger. Turbine flowmeters were used to measure liquid flow. All temperatures in both liquid and refrigerant circuits were measured using type-T thermocouples inserted into the fluid stream.

The output of the various sensors was processed through a data logger and transmitted via a RS-232/422 interface to a desktop computer. A commercial data acquisition and processing program was used to interrogate each sensor at six second intervals. The data are averaged over one minute intervals and converted to engineering units for storage and display on a monitoring screen.

TEST PROCEDURE

The VCTL was set up to run tests approximating saturated condensing temperatures of 80 to 100°F and saturated evaporating temperature conditions of -20 to -10°F as specified by the work sponsor. In addition we tried to maintain minimum levels of refrigerant subcooling leaving the condenser and superheat leaving the evaporator - about 10°F was desired. Due to equipment limitations it was not possible to meet both superheat and subcooling conditions so tests were run with minimal subcooling levels and superheat was allowed to float. Compressor speed was set so that capacity was maintained at a relatively constant value for both refrigerants at each test condition. Pneumatic pressure was used to adjust the expansion valve to the desired evaporator temperature and pressure condition. A Polyol Ester (POE) oil was used for compressor lubrication for both refrigerants tested.

Two refrigerants were tested under this project. The VCTL was

operated with R-22 to provide baseline performance values for the test conditions specified. Subsequent tests were performed with the Blend for comparison. The Blend exhibits nearly azeotropic behavior in the two-phase region. At the condensing conditions tested the saturated liquid and vapor temperatures (bubble and dew points) were within about 0.5°F. At the evaporating conditions there was about a 2-3°F difference (glide) between liquid and vapor temperatures as computed with the NIST REFPROP refrigerant properties program (NIST 1996).

It was fairly easy to set up the tests with R-22 given the large amount of published property information for this refrigerant. The expansion control valve was set to obtain a pressure corresponding to the desired evaporator saturation temperature. Condenser secondary fluid entering temperature and flow were controlled to obtain the condensing temperatures and pressures desired. Very little saturation temperature and pressure data was available for the Blend. The project sponsor provided a table of saturation properties and we supplemented that with computations using REFPROP. Initial test runs attempted to match the measured blend and R-22 entering evaporator temperatures. However, we observed that the measured blend pressure entering the evaporator corresponded to a saturation temperature from the sponsor's property estimates that was about 5 to 6°F higher than the measured temperature. Saturated liquid and vapor temperatures computed by REFPROP for the measured entering evaporator pressure were 2 to 5°F lower than the actual measured temperature. Figure 2 illustrates these observations for one test condition.

We decided to run the Blend tests for two cases. First, an evaporator entering pressure was chosen to yield a saturation temperature from the sponsor's estimated property data that matched the entering evaporator refrigerant temperature from the corresponding R-22 test. For the second case, tests were run where the measured entering evaporator temperature for the Blend matched the corresponding measured R-22 temperature (as closely as practical). In both cases, condenser and evaporator entering secondary fluid temperature and flow were maintained equal to the values used in the R-22 tests.

Due to the uncertainties about the available Blend property estimates, all capacity and efficiency comparisons reported in this paper are based on the measured evaporator secondary fluid side capacity and the compressor shaft power input. We were able to check the secondary-side vs refrigerant-side evaporator and condenser capacity heat balances for R-22 (using REFPROP for R-22 properties). The two measures matched within about $\pm 2\%$ for the condenser and within about 2-5% (refrigerant-side high) for the evaporator. The thermocouples for the secondary fluid measurements were calibrated against a precision thermometer (traceable to NIST) in a constant temperature bath. Results showed that the thermocouples reproduced the thermometer readings within $\pm 0.3^\circ\text{F}$.

TEST RESULTS AND DISCUSSION

After achieving steady operating conditions at each test condition, data were taken over a 30 minute test run. Most of the data signals held very steady during the tests. One significant exception was the evaporator secondary fluid exiting temperature. Figure 3 is a plot of evaporator secondary fluid temperature difference and flow rate for one of the test runs for the Blend indicating the periodic nature typically observed for this quantity. Data were averaged over 2 complete temperature difference cycles for the blend cases. For the R-22 cases, the temperature difference was much steadier and averages were taken over the full 30-minute runs.

Figure 1. Vapor compression test loop (VCTL) schematic.

Blend Temperature at Evaporator Inlet:

Measured Temperature	= -13.1°F
Saturation Temperature @ 16.7 Psia from Sponsor's Data	= -7.2°F
Liquid Saturation Temperature from REFPROP @ 16.7 Psia	= -17.8°F
Vapor Saturation Temperature from REFPROP @ 16.7 Psia	= -15.3°F

Figure 2. Comparison of measured Blend evaporator temperature to available property estimates.

Figure 3. Representative plot of evaporator secondary fluid (glycol) temperature difference and flow from Blend test run.

Table 1 presents test results for the baseline R-22 tests for condenser secondary fluid entering temperatures of 70, 80, and 90°F (yielding saturated condensing temperatures of approximately 80, 90, and 100°F) and one evaporating condition (approximately -12°F). Corresponding data from the Blend tests are also presented in Table 1 for the two test control cases noted above. Subcooling levels at the condenser exit were about 5-8°F for both refrigerants. Superheat levels exiting the evaporator were about 33°F for R-22 and about 40°F for the Blend. Sight glassess on the VCTL loop showed no indication of two phase refrigerant flowing in either the liquid line (entering the expansion valve) or the vapor line exiting the evaporator.

Uncertainties for capacity and power measurements were

determined using an error propagation analysis as outlined in Holman (1971). For R-22, capacity and power uncertainties are about ±11% and ±3%, respectively, based on accuracies of ±2 in-lb for torque, ±1 rpm for shaft speed, ±1% for fluid flow, and ±0.3°F for the thermocouples as noted above. These power and capacity uncertainties led to an uncertainty in COP or efficiency (capacity/power) of about ±11%. The primary driver in these uncertainties was the relatively low secondary fluid temperature difference measured across the evaporator. For the Blend cases, power, capacity, and COP uncertainties were ±7.5%, ±11%, and ±13%, respectively. The higher power uncertainty for the blend tests was caused by the lower average torque levels (about half the values for R-22).

Evaporator cooling capacity (from secondary fluid measurement) is plotted against average condensing temperature in Figure 4. The condensing temperatures for both refrigerants are those computed using the REFPROP code. No direct measurement of the two-phase condensing temperature was made. For the Blend at -13°F saturated refrigerant temperature (as measured at entrance to evaporator) and 1250 rpm compressor speed, capacity is about 6-7% greater than that of R-22 (-12°F and 750 rpm). At 1370 rpm and -19°F, the blend capacity is about 3-4% lower than that of R-22. It can be estimated that for a compressor speed of about 1150 rpm at -13°F refrigerant temperature, the Blend capacity would have been approximately equal to that measured for R-22 in these tests. This implies that, to achieve equivalent capacity, this blend would require a compressor speed (or displacement) about 53% higher than required for R-22. However, it must be noted that the capacity differences noted above are well within the uncertainty limits for this measurement. More definitive investigations at wider ranges of condensing and evaporating conditions and capacity levels are needed in order to precisely determine compressor displacement or speed required for this refrigerant.

Cooling COP, defined as

$$\text{Evaporator capacity} \div \text{compressor shaft power},$$

is plotted against cooling capacity in Figure 5. For the Blend test results at 1370 rpm and -19°F (closest match to R-22 capacities at equal condenser entering glycol temperatures), system cooling COP ranged from 25% to 40% greater than that measured for R-22. This is a significantly higher efficiency but we must be reminded that the measured COPs are subject to uncertainty limits of 11% (for R-22) and 13% (for the Blend). Figure 6 illustrates the efficiency vs capacity comparison above with the uncertainty limits shown.

Even considering the uncertainties embedded in these results, the efficiency measured for the Blend relative to R-22 is surprisingly greater (the minimum gap between the envelopes for the Blend and R-22 in Figure 6 is about 10%). At this point, we can offer no definitive explanation as to why the Blend's performance should be so much better. The estimated isentropic efficiency of the test stand compressor (using REFPROP property estimates) was about 9% higher with the Blend (80-81% vs 71-75% with R-22). This could account for most of the minimum improvement noted above but not for the 25%-40% gain level exhibited by the numbers in Table 1. Ideal cycle calculations made using REFPROP estimates for enthalpies are summarized in Table 2. They indicate no significant cycle efficiency difference between these two refrigerants assuming equal compressor efficiency. When considering the compressor efficiencies observed in the tests (Case 3), the calculated Blend COP is about 10% higher than that of R-22.

Table 1. Test Results for the Blend and R-22

Compressor Speed (rpm)	Evaporator capacity (10 ³ Btu/h)	Fluid temperature entering condenser (°F)	Average saturated refrigerant condensing temperature (°F)*	Fluid temperature entering evaporator (°F)	Refrigerant temperature entering evaporator (°F)	Refrigerant pressure entering evaporator (psia)	Cooling COP
R-22	750	3.93	89.9	99.9	19.2	-12.2	29.98
	750	4.47	80.2	91.6	19.4	-12.1	30.08
	750	4.73	71.5	81.7	19.8	-12.0	30.08
Blend	1250	3.64	90.0	96.1	19.7	-18.8	14.67
	1250	4.14	79.3	87.2	19.3	-19.3	14.42
	1250	4.34	70.5	79.4	19.8	-19.4	14.42
	1370	3.99	90.1	96.7	19.0	-20.6	13.98
	1370	4.41	80.8	89.2	19.1	-19.3	14.36
	1370	4.67	70.5	79.0	19.0	-19.2	14.54
	1250	4.40	89.9	95.4	19.2	-12.7	16.87
	1250	5.06	80.2	84.7	19.3	-13.1	16.72
	1250	5.35	70.2	75.9	19.1	-12.9	16.78
	1370	4.85	90.0	96.6	19.5	-12.3	17.08
	1370	5.36	80.3	86.0	18.9	-12.8	16.84
	1370	5.74	70.4	78.6	19.1	-12.9	16.82

* Computed using REFPROP for measured condenser pressure.

Figure 4. Comparison of cooling capacity between the Blend and R-22.

Figure 5. Comparison of COP between the Blend and R-22.

Table 2. Ideal "Reverse Rankine" cycle COP calculations for Blend and R-22

Mean evaporating and condensing temperatures: -12°F and 95°F

Calculation case	R-22 COP	Blend COP
1 No superheat; No subcooling; 100% compressor efficiency	3.17	3.07
2 5°F subcooling for both; Superheat of 40°F for Blend; 33°F for R-22; 100% compressor efficiency	3.34	3.37
3 Subcooling and superheat same as Case 2; 81% compressor efficiency for Blend; 74% compressor efficiency for R-22	2.47	2.73

Figure 6. Uncertainty envelopes for COP vs cooling capacity comparison between the Blend and R-22.

CONCLUSIONS AND OBSERVATIONS

For the following conclusions it must be stressed that the experimental results reported in this paper are for tests done on one specific piece of hardware, and they may not be generalizable to all situations. Further, the results herein are from initial comparisons between the Blend and R-22 and are based on tests conducted at specific condensing and evaporating conditions.

Specific conclusions drawn from the data and results presented in Tables 1 and 2 and Figures 4-6 are given below:

- * The Blend has a much lower specific volumetric capacity than R-22. A compressor speed or displacement at least 53% greater than that required for R-22 would be needed to meet the same refrigeration load in a given application.

- * For nearly equivalent capacity and evaporating/condensing temperature conditions, the Blend exhibited measured efficiency levels 25% to 40% greater than those of R-22.

- * Thermodynamic cycle calculations using REFPROP for properties for both R-22 and the Blend (the best property estimates available for the Blend) show about a 10% improvement for the Blend with the compressor efficiencies observed during these tests.

These experimental results must be interpreted in light of the rather significant uncertainties in the capacity and efficiency measurements (11% to 13%). The COP gains observed for the Blend over R-22 are well outside these uncertainty limits. They are also much higher than the levels the ideal cycle calculations suggest (about 10%). Further experimental and analytical investigations of the Blend, considering a wider range of condensing/evaporating conditions, are warranted to more precisely evaluate the efficiency improvement potential of this blend for refrigeration equipment applications. It is recommended that experimental conditions be adjusted to reduce the uncertainty limits in future test programs. Running tests at higher capacity levels, thus forcing greater temperature differences, would be a simple means to reduce measurement uncertainty.

More detailed thermodynamic property measurements are needed for the Blend to facilitate better property estimates using REFPROP or other techniques. For the evaporator pressures observed during our tests, the available property estimates gave saturation temperatures from about 5°F lower to 6°F higher than that which was measured. More accurate properties are essential in order to enable more precise comparisons between this blend and existing HFC refrigerants. This would permit refrigerant-side capacity and COP calculations to be made as a check on the secondary-side values. Beyond thermodynamic considerations, thermophysical, toxicological, and material compatibility information will be necessary to fully determine the Blend's utility in refrigeration and air-conditioning equipment.

ACKNOWLEDGEMENT

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