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ABSTRACT

Recently, several promising candidates for replacing R-502 and R-22 in commercial refrigeration and residential heat pump applications have been proposed and tested. Among them, the alternatives, SUVA® HP62 for replacing R-502 and SUVA® AC9000 for replacing R-22, have been found acceptable for many existing and new systems requiring little to no equipment modifications. There is, however, potential to further improve the performance of these alternatives for both existing and new equipment. Understanding the compressor characteristics of these compounds is one of the most important key factors for this purpose. Here, we have performed compressor calorimeter tests for these alternatives as well as with R-502 and R-22. The experimental data have been analyzed with theoretical cycle calculations, and several implications of the present findings will be discussed.

INTRODUCTION

Impending reductions in chlorofluorocarbons (CFC) production, with an accelerated phase-out by January 1, 1996, have made it necessary to develop replacement refrigerants for R-502[1]. R-502 (a mixture of R-22 and CFC-115) is widely used for low and medium temperature commercial refrigeration. Furthermore, R-22 (hydrochlorofluorocarbon, HCFC), which is commonly used for heat pump and air-conditioning applications, is scheduled for phase-out according to the Montreal Protocol[2]. We have recently developed two hydrofluorocarbon alternatives called HP62 for R-502[1] applications and AC9000 for R-22[2] applications. HP62 is a near-azeotropic mixture of pentafluoroethane (HFC-125), 1,1,1-trifluoroethane (HFC-143a), and 1,1,1,2-tetrafluoroethane (HFC-134a) with a composition of 44/52/4 wt%, respectively, and AC9000 is a ternary mixture of difluoromethane (HFC-32), HFC-125, and HFC-134a with a composition of 23/25/52 wt%, respectively.

These alternatives were initially developed based on a thermodynamic cycle model[1] as well as flammability considerations, and have been tested in existing and new equipment without requiring significant modifications. Their performance (capacity, energy efficiency, etc.) w.r.t. R-502 and R-22 have been found to be similar. Additional enhancements may be possible to further improve new equipment and system performance. For this purpose, we must understand the performance and characteristics of each part of the integrated system using these alternatives: heat exchanger, compressor, expansion device, etc. This report will focus on the compressor performance of these alternatives w.r.t. R-502 and R-22, using a compressor calorimeter. For the R-502 case, a semi-hermetic reciprocating compressor was tested, while for the R-22 application, a scroll compressor was tested The experimental results are compared and discussed with thermodynamic cycle calculations, and a possible simulation model for the calorimeter will also be mentioned.

EXPERIMENTAL

A commercial calorimeter [3] with a nominal capacity of 30 to 130 kBTU/hr (8.8 to 38.1 kW) was used with two types of compressors: a semi-hermetic reciprocating compressor designed for R-502 and R-22 refrigeration[4] and a scroll compressor designed for R-22 heat pump and air-conditioning applications[5]. A polyol ester lubricant was used for both compressors. A coriolis mass flow meter was installed to check the heat balance between the boiler and the refrigerant. A sight glass was also installed just after the flow meter to ensure single phase flow. In addition, intrusive thermocouples and pressure taps were installed within 6" (15 cm) of the suction and discharge connections on the compressor. The compressor was mounted inside a temperature controlled cabinet and the temperature was maintained at 95°F (35°C) for all tests. The reciprocating compressor had a displacement of 60.05 cft/min. and was operated at 500 volts. The compressor was equipped with a top mounted cooling fan which was operated at 230 volts. The scroll compressor had a displacement of 10.7 cft/min and was operated at 230 volts. The power consumption was measured using a multiphase digital power meter and the energy efficiency for both compressors was based on the power required to operate the compressor only.

Overall accuracy in the present experiment were examined by the energy balances between the boiler side and the refrigerant side. Typically they were in excellent agreement within 1-3% over a wide range of operating conditions from 60 to 130 kBTU/hr (17.5 to 38.1 kW). At the lower capacity region, the energy balances were accurate within 1 to 5%. The calorimeter test is usually run by setting various temperatures in the evaporator and condenser with a specified amount of subcooling and superheat. In general this creates no difficulty or confusion with pure refrigerants. However, it is not an obvious matter, when we compare refrigerant mixtures, since their condensing and evaporating temperatures are no longer constant at a given pressure. The condenser temperature for the mixtures was set by taking an arithmetic average of the dew and bubble point temperatures at a given pressure, while the evaporator temperature was set similarly but with the mean between the evaporator inlet (instead of bubble) and dew point temperatures. The desired temperature settings were maintained using a pressure controller. The amount of subcooling and superheat were set relative to the mean temperatures mentioned above[6]. The proper pressures for the desired temperature settings and the subcooling correction were calculated using a thermodynamic cycle model[1].

A) HP62/R-502 (Refrigeration)

The calorimeter tests for HP62 and R-502 were conducted at condenser temperatures from 70 to 130°F (21 to 54°C) and evaporator temperatures from -40 to 0°F (-40 to -18°C) with compressor suction temperature maintained at 65°F (18°C) with about 5°F (2.8°C) subcooling which was corrected to zero using the thermodynamic model.

B) AC9000/R-22 (Heat Pump/Air-Conditioning)

The experiments for AC9000 and R-22 were performed at typical residential heat pump conditions: condenser temperatures from 90 to 130°F (32 to 54°C), and evaporator temperatures from 10 to 50°C), with the amounts of subcooling and superheat of 15°F (8°C) and 20°F (11°C), respectively.

RESULTS AND DISCUSSIONS

A) HP62/R-502 (Reciprocating Compressor)

The experimental results are summarized in Figs. 1-3 for cooling capacity, energy efficiency (EER), and discharge temperature, respectively, relative to that of R-502. The hatched bars are the experimental data, while the dotted bars are calculated results based on a standard thermodynamic cycle model, which assumes isentropic compression, adiabatic liquid expansion, volumetric efficiency of 1, and mean temperatures for evaporator and condenser as described in the previous section. Twelve experimental conditions are denoted as T°F (condenser)/T°F (evaporator) on the horizontal axis; they are arranged in increasing order of compression pressure ratio, which ranges from about 5 to 18 for these experiments. In general., HP62 had the same to better capacity, within +/- 5% in energy efficiency, and about 15°F (9.4°C) lower compressor discharge temperatures compared with R-502.

A-1) Cooling Capacity

The ideal cycle calculation explains the general trend of observed data (Fig.1), and agrees with the measured data within about 5%. However, it is clearly seen that the measured values are systematically 2-5% higher than the calculated. This is due to the difference in the volumetric efficiency (η_V) for both refrigerants; η_V for HP62 must be higher than that for R-502. If we assume η_V is purely due to the clearance volume effect, then it is described by[7]:

$$\eta_V = 100 - Vc\,\{(Dd/Ds) - 1\} \tag{1}$$

Vc: clearance volume %; Dd: discharge gas density; Ds: suction gas density

The actual cooling capacity (Q) is given by:

$$Q = \Delta He * \dot{m} \tag{2}$$

$$m = \eta_v * Ds * Vdis \qquad (3)$$

ΔHe: enthalpy change in evaporator; m: actual mass flow rate; Ds: suction gas density; Vdis: compressor displacement volume rate.

To estimate this effect, we used Vc of 4%, and calculated η_v with the experimental data. Eq. (1) gave, however, the opposite result; η_v for HP62 was lower, on the contrary. It is known that the actual efficiency depends on several other factors occurring in the compressor, such as reduction of suction gas density due to heating, gas leakage during the compression, oil/refrigerant solubility degassing, etc. All these effects will act in a way similar to the clearance volume, and may be lumped together in Eq.(1), where Vc is then treated as an effective value. We have analyzed the experimental data in such a model, and found the effective Vc for R-502 is indeed consistently larger than HP62 by a few percent. The effective Vc was well characterized by an empirical function of compression ratio and suction gas density. The apparently large Vc for R-502 relative to HP62 may be due to oil/refrigerant solubility effects. In order to investigate such an idea, we have performed similar experiments using a different oil. Our preliminary experimental results seem to support this hypothesis. This suggests the importance of choosing the correct lubricant oil for the compressor.

A-2) Energy Efficiency

Theoretical calculations agree with the observed data within about 5%, but again the measured values are systematically larger by 2-5% (Fig.2). This is similar to the case of the capacity comparison above. Before discussing the results further, we shall define the energy efficiency, since there are many ways to measure the energy efficiency in general.

Thermodynamic COP (coefficient of performance) is defined by:

$$COP = (\text{Cooling Capacity}) / (\text{Isentropic Compression Work}) = \Delta He / \Delta Hi \qquad (4)$$

ΔHe: enthalpy change in evaporator; ΔHi: enthalpy change in isentropic compression

In practice, we are often interested in an actual energy efficiency, which is cooling capacity relative to the total input electricity to operate the compressor. This is called EER (energy efficiency ratio); it should be dimension less, but commonly it is expressed by (BTU/hr)/W in the United States.

$$\begin{aligned} EER &= (\text{Cooling Capacity}) / (\text{Input Electricity to Compressor}) = (\Delta He * m) / Wtotal \\ &= (\Delta He * m) / (Wloss + Wisen) \\ &= (\Delta He * m) / (Wloss + \Delta Hi * m) \end{aligned} \qquad (5)$$

Wloss: all irreversible losses in electricity (no-load power, friction, etc.);
m: actual mass flow rate;
Wisen: isentropic work: Wtotal: total input electricity to operate compressor

This can be rearranged by algebraic manipulations using Eq.(4):

$$EER = \eta_i * COP \qquad (6)$$

$$\eta_i = Wisen / Wtotal = \Delta Hi * m / Wtotal \qquad (7)$$

Eq.(7) may be suitably called the isentropic efficiency for the compressor. In order to have EER in units of (Btu/hr/W) in Eq. (6), it should be multiplied by a constant of 3.414.

In Fig.2, the measured data is based on Eq. (5) or (6), while the calculated is based on Eq.(4). The difference of 2-5% mentioned above is due to the η_i effect, which may be largely due to the actual mass flow rate "m" (or η_v) effect in the present case, since the results in Figs. 1 and 2 are very similar.

An actual simulation model for EER can be made based on Eqs. 5-7, where all realistic irreversible losses in the compression process may be lumped together in Wloss. Then, Wloss is correlated with the observable quantities such as compression ratio, gas density, etc. This has been done for the present system. Preliminary results are encouraging.

A-3) Discharge Temperature

Compressor discharge temperature is an important factor for choosing refrigerants; lower temperatures would have positive effects on lubricant stability and compressor life. Fig. 3 shows HP62 has lower discharge temperatures by 10-15°F (6 - 8°C) than those of R-502, as expected by thermodynamic theory. The observed values are in close agreement with the ideal model (isentropic compression) within 3°F (1.7°C), when they are compared with the relative magnitudes. This suggests the possibility to make a reasonable simulation model for calculating the difference in compressor discharge temperature. In fact, we have developed a generalized empirical model base on the present experiment. The model is still being tested for its validity.

B) AC9000/R-22 (Scroll Compressor)

The experimental results of AC9000 are summarized in Figs. 4-6 for cooling capacity, energy efficiency (EER), and discharge temperature, respectively, relative to that of R-22. The hatched bars are the experimental data, while the dotted bars are calculated results based on the standard thermodynamic cycle model as described in the previous section. Seven experimental conditions are denoted as T°F(condenser)/T°F(evaporator) along the horizontal axis; and are arranged according to increasing order of compression pressure ratio, which ranges from about 2 to 4 for these experiments. In general, AC9000 had similar capacity, slightly lower energy efficiency, and about 15°F (9.4°C) lower compressor discharge temperatures compared with R-22.

B-1) Cooling Capacity

In contrast to the HP62/R-502 case, rather striking agreement is seen between the theoretical predictions and the experimental data (Fig. 4). Only a slight reduction (1 - 2 %) for AC9000 was observed, which is well within experimental uncertainties for the measured data. This is expected according to Eq. (1), if we assume a small clearance volume for the scroll compressor; the clearance volume effect indicates that AC9000 should be slightly lower for these experimental conditions. However, it may be safely said that no significant difference exists between AC9000 and R-22 using this compressor; effects such as oil/refrigerant interactions mentioned earlier are not apparent here. This may be due to the fact that the compression ratios are much smaller than those in the previous case.

B-2) Energy Efficiency

As shown in Fig.5, the agreement between the theoretical and measured data is within about 5 %. However, with this case, the behavior is quite different compared with the case of HP62/R-502. At high compression ratios the observed values for AC9000 are 4-5% lower, while at low compression ranges they are 0-3% higher relative to R-22. Since in this case the capacity is essentially the same, it means the volumetric efficiency is the same as seen in Fig.4 The observed behavior must be due to the effect of the isentropic efficiency (Eq. 7). In order to examine this effect, we have analyzed the experimental data based on Eq. (7). The result is shown in Fig. 7; the symbols are the analyzed data, and the smooth curves are fitted to the data. It is clearly seen that this compressor efficiency is optimized for R-22 with a flat maximum around the compression range for typical heat pump and air-conditioning applications. AC9000 has slightly higher compression ratios relative to R-22 at the same condenser/evaporator temperatures. This explains the fact observed in Fig.5, but also suggests an important matter for compressor engineers. That is, the EER for AC9000 can be improved compared with R-22 if the compressor efficiency optimum (maximum) can be shifted slightly toward higher compression ratios; see Fig.7.

B-3) Discharge Temperature

Fig. 6 compares the calculated and the measured compressor discharge temperatures for AC9000 relative to R-22. The measured values agree within 1 to 7°F (.5 to 3°C) with the thermodynamic calculations, however it should be noted that there exists a systematic difference at low compression ratios. This may be due to the difference between the reciprocating and scroll compressors. The result is different from that observed in the HP62/R-502 comparison (Fig. 3).

CONCLUSIONS

Compressor calorimeter tests for SUVA® HP62 and SUVA® AC9000, have been performed in comparison to R-502 and R-22. In general., HP62 had the same to better capacity, within +/- 5% in energy efficiency, and about 15°F (9.4°C) lower compressor discharge temperatures compared with R-502 in a reciprocating compressor tested at twelve low temperature refrigeration conditions. AC9000 had similar capacity, slightly lower energy efficiency, and about 15°F (9.4°C) lower compressor discharge temperatures compared with R-22 in a scroll compressor tested at seven air conditioning and heat pump conditions. The experimental results have been analyzed based on a theoretical thermodynamic cycle model. It has been found that the ideal thermodynamic cycle model is reasonably accurate and useful for evaluating actual compressor performance, when it is used on a relative basis such as described in the present study. Several new findings for the performance of two types of compressors (reciprocating and scroll) have been discussed in terms of volumetric and isentropic efficiencies as well as compressor discharge temperature. A possible simple model for calorimeter tests has also been suggested.

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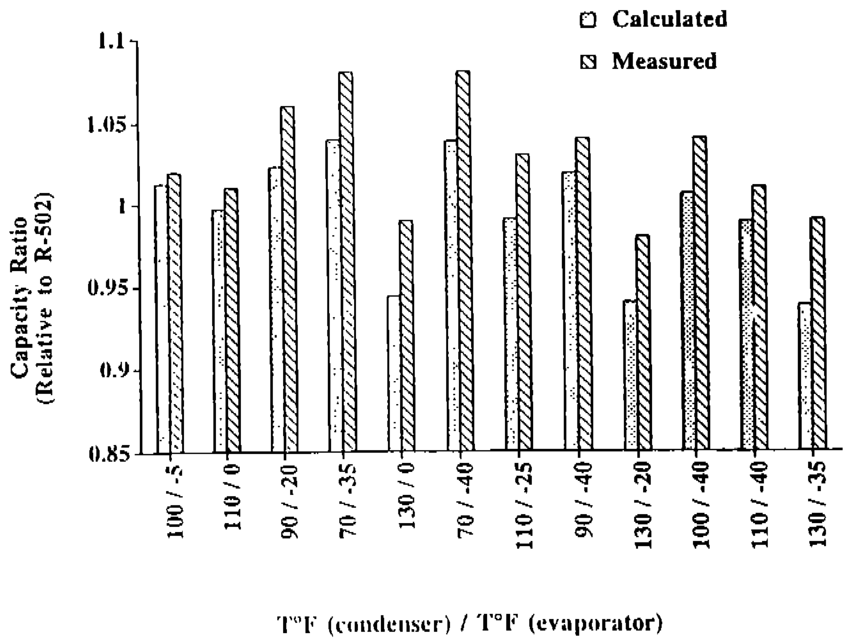


Figure 1

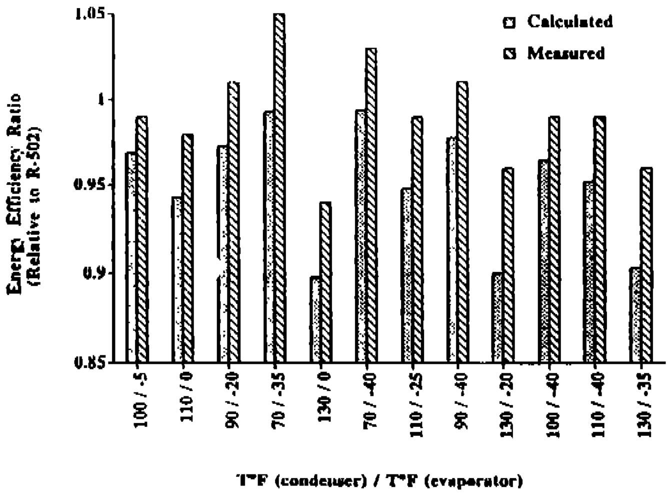


Figure 2

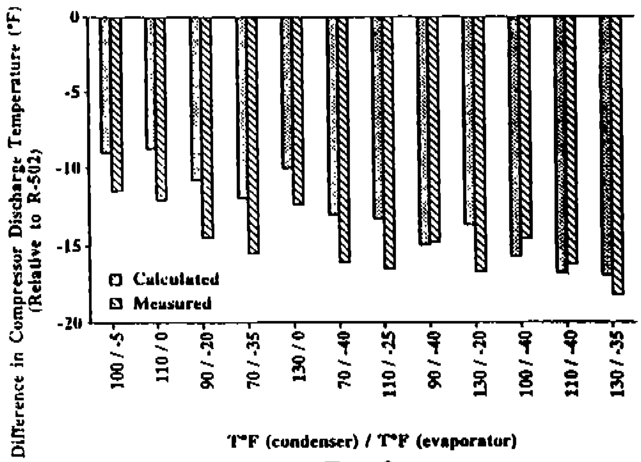
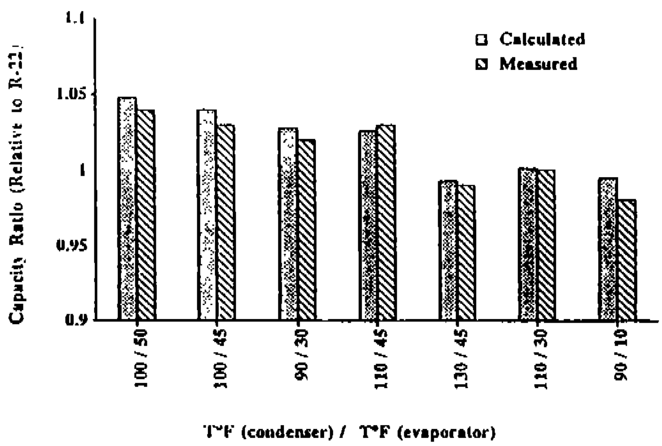


Figure 3



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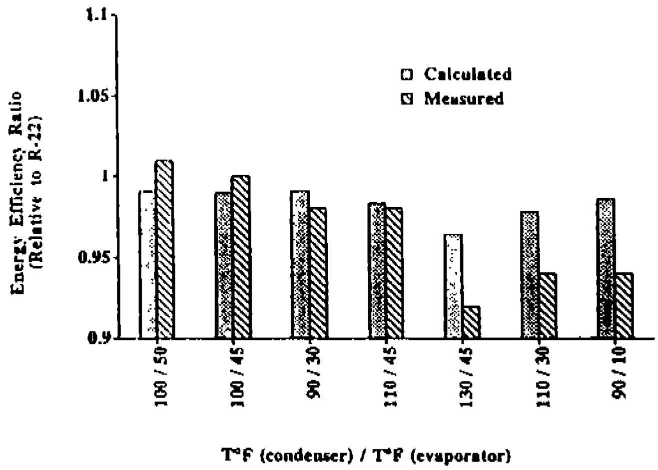

Figure 5

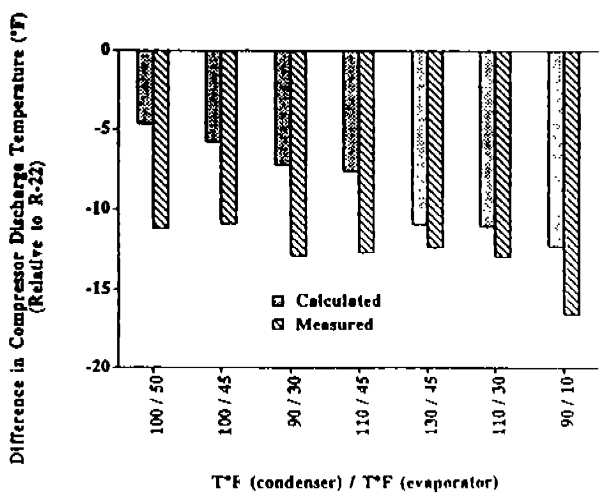

Figure 6

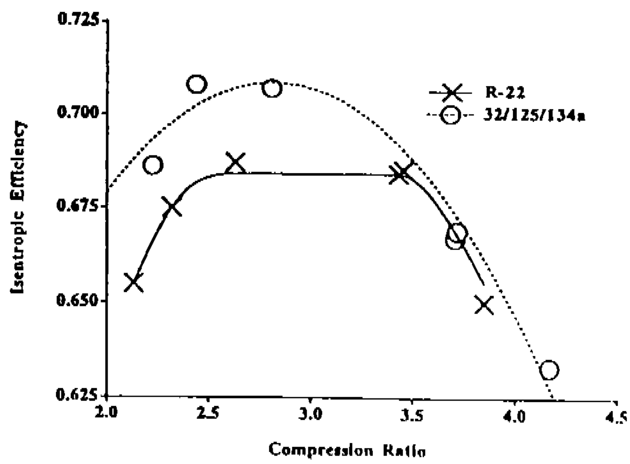

Figure 7