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ABSTRACT

This paper discusses the findings of an experimental study designed to show the effect of changing refrigerants on the performance of a refrigeration system. The findings are based on a total of 88 independent tests using two refrigerants, namely R-12 and R-134a. The test equipment consists of a fully instrumented, 3-ton refrigeration system which provides values for the system COP, cooling capacity, mass flow rate, and thermodynamic states of the refrigerant at the inlet and exit of each component in the system. The parameters varied in the study are the evaporator and condenser sink temperatures and the amount of refrigerant used in the system. The evaporator exit superheat was maintained constant at 13.5 F (7.5°C) for all tests.

The experimental results show that for almost all test conditions the R-134a operates with greater COP and cooling capacity. This is especially true of the system when charged with R-134a where the condenser subcooling is 10 to 15 F (5.6 to 7.5°C). The effects of operating conditions on system performance are presented in detail. Several questions are considered through examination of the thermodynamic states at key points in the system. For example, it is shown how the system can operate at a higher pressure ratio with R-134a and yet have a greater COP than for the system charged with R-12.

INTRODUCTION

In a time where many air conditioning systems are being converted from R-12 in an effort to reduce the use of CFCs, an experimental study on system performance for different refrigerants is appropriate. Several studies have been reported in literature related to this issue (Petersson 1990, Snelson 1992, Snelson 1990, Vinyard 1991, Vinyard 1989) which can be categorized as drop-in or thermodynamic studies. Drop-in studies (Vinyard 1989) give a good representation of the expected change in performance for a particular application. Thermodynamic studies that focus on the thermodynamic states of the refrigerant (Snelson 1992) help explain the change in performance. The study presented here is a first step in reconciling the drop-in and thermodynamic approaches. Comparisons based on equivalent sink temperatures, as is done in typical drop-in studies, combined with the variation of those temperatures, as is done in studies based on equivalent thermodynamic states, gives a realistic and comprehensive look at system performance. Questions that may be addressed with this combined approach are:

- How does system performance for an alternative refrigerant compare with that of R-12 when the system is operated over a range of operating conditions?
- What is the effect of refrigerant charge on system performance for the two refrigerants?
- How do the thermodynamic states of the refrigerants in the individual components (ie. compressor, evaporator, and condenser) relate to the total system performance?

To address these questions a project was undertaken to study the behavior of a fully instrumented air conditioning system operating with two different refrigerants (CFC-12, and HFC-134a) over a wide range of operating conditions.

EQUIPMENT AND CONTROL

The test equipment consists of a 3 ton (11 kW) refrigeration system and an air flow loop which provides controllable air-side conditions for the evaporator. The air flow loop consists of several controllable heat exchangers, a squirrel cage fan, and a spray humidifier. In addition to the evaporator, the refrigeration system consists of a water-cooled shell-and-tube condenser, an expansion valve, and a hermetically sealed piston compressor.

The air flow loop is automatically controlled to maintain air flow rate, humidity and temperature at their desired values. A schematic of the air flow loop is shown in Figure 1. The air is recirculated by a variable speed fan in an insulated duct at flow rates up to 2500 cfm (71 m^3/min). Energy is input to the air stream by a steam coil (cross-flow heat exchanger) and an electric duct heater. The flow rate through the steam coil is manually adjusted with a gate valve such that only a portion of the heating load is met by the steam coil. The electric duct heater, which is controlled with a PID SCR controller, is used to meet the balance of the heating load so that the inlet air temperature at the evaporator remains constant at a set value. The inlet air temperature can be maintained at temperatures in the range of 55 to 95 F (13 to 35°C). The humidity is controlled from 0 to 50% relative humidity by the adjustment of a gate valve on a steam spray humidifier.

The refrigeration system being studied is shown schematically in Figure 2. The system is controlled by a thermostatic expansion valve or a hand expansion valve. The hand expansion valve can be controlled manually or by a servo-motor connected to a computer so as to maintain a set superheat. In the latter case, the superheat is continually monitored and the motor is controlled through a digital control scheme.

The condenser water inlet temperature and flow rate are controlled with the use of a manually operated gate valve near the condenser inlet. The condenser water is pumped through the condenser by a three-stage constant-speed centrifugal pump. Adjustment of a gate valve provides a range of water flow rates from 20 to 30 gpm (76 to 110 L/min). The water inlet temperature is controlled between 70 and 100 F (21 and 38°C) by partially recirculating water through the condenser. Water from the condenser exit is returned to the pump inlet holding tank and mixed with fresh cold water. A constant head is maintained in this tank with use of a stand pipe. A constant flow rate of the cold water is provided through the use of a second tank and standpipe. The flow rate is regulated by the adjustment of a ball valve.

INSTRUMENTATION

The refrigeration system is instrumented so that its performance can be evaluated at various operating conditions. Instrumentation was selected so that the state of the refrigerant at the inlet and exit of each of the components in the system can be determined. This instrumentation includes pressure transducers, thermocouple probes, and flow sensors as shown in Figure 2. The installed instrumentation can be used to perform energy balances on each component in the system.

Additional instrumentation on the non-refrigerant side of the condenser, evaporator, and compressor is also installed, as shown in Figure 1. This additional instrumentation can be used to perform energy balances which can then be compared to refrigerant-side energy balances. The condenser water inlet and exit temperature are measured with thermocouple probes, and the water flow rate is measured with a drag flow meter. The air temperatures at the inlet and exit of the evaporator are measured with a grid of eight thermocouples. The humidity is measured upstream and downstream of the evaporator with dry and wicked thermocouple probes. The flow rate of air through the evaporator is measured with a pitot tube arrangement and a differential pressure transducer. The power used by the compressor is measured by a watt transducer.

DATA ACQUISITION

Each of the instruments used has a voltage output which is channeled to a computer through a computer-addressable scanner and voltmeter using a programmable input bus. As the data are collected, refrigerant properties are then calculated for each state point in the system. Key variables are plotted during system operation for the purpose of controlling and monitoring the transient behavior of the system. The property data is also presented on a pressure-enthalpy diagram on the computer monitor as data is being taken during system operation. The constant feedback of displayed data and plots during system operation allows system conditions to be monitored while experiments are in progress so that inputs can be adjusted to obtain the desired conditions.

TEST CONDITIONS

To compare the performance of the two refrigerants under varying operating conditions the evaporator air temperature and the condenser water temperature were varied over a range of values. A third independent variable, refrigerant charge was also considered in looking at the effect on system performance. Varying these three parameters provides a basis of comparison valid for a whole range of operating conditions rather than for one specific condition. In addition, several tests were repeated at various conditions to demonstrate the repeatability of the experimental results. A total of 28 independent tests for the R12 and 60 independent tests for the R134a were performed.

Variable parameters

In order to simulate actual air conditioning conditions, the inlet temperature was set at 55, 60 and 65 F (13, 15.5, and 18°C) for each setting of the other independent variables. The condenser water temperature was set at values of 70, 80 and 90 F (21, 27, and 32°C) which represents typical values of heat sinks available for refrigeration and air conditioning systems. The minimum refrigerant charge of the system was dictated by the condition that subcooling must occur in the condenser. The system was run at three different charges for the R12 tests: 8.86 lb (4.02 kg), 9.27 lb (4.20 kg), and 9.77 lb (4.43 kg) and run at five different levels of charge for the R134a tests: 7.42 lb (3.37 kg), 7.98 lb (3.62 kg), 8.52 lb (3.86 kg), 9.05 lb (4.10 kg), and 9.60 lb (4.35 kg).

Constant parameters

As the condenser water flow rate is increased, its effect on system performance decreases. The reason is that the water flow rate becomes high enough that the temperature change of the water from inlet to exit is small. Thus, the condenser water flow rate was set near its maximum value (29 gpm or 110 L/min) and held constant for all tests. For the same reason, a large evaporator air flow rate of 2400 cfm (68 m^3/min) was chosen and held constant for all tests. The air in the flow loop is recirculated in an effort to reduce the effects of condensation on the coil surface. Thus, the humidity ratio is constant throughout the flow loop.

The effect of superheat on system performance was also studied and found to be insignificant relative to the effect of other variables. A 5 F (2.8°C) increase in superheat caused only a 1% increase in COP and a 2% decrease in capacity for tests with R12. Similar behavior was observed with the R134a. Operation of the system at low superheats (below 13 F or 7°C) frequently resulted in uncontrollable fluctuations in superheat. Such fluctuations might lead to two-phase conditions at the evaporator exit, thus damaging the compressor. Even though the effect of superheat on system performance is measurable, its effect is much smaller than the effect of other independent variables and was, therefore, held constant at 13.5 F (7.5°C) for all tests. This value of superheat is consistent with those found in actual operating refrigeration and air conditioning systems.

DATA ANALYSIS

A curve fit of cooling capacity (based on enthalpy change of the refrigerant) as a function of evaporator air inlet temperature and condenser water inlet temperature is shown in Figure 3 for R134a. The lines of lower evaporator air temperature correspond to lower capacity. The capacity increases approximately 14% for a 10 F (5.6°C) increase in evaporator air temperature. In contrast, a 10 F (5.6°C) increase in condenser water temperature causes a decrease in capacity of approximately 5%. Ninety-five percent confidence intervals are also shown on the figure. These lines account for uncertainty in the data as well as lack of fit of the data to the mathematical model selected.

A decrease in cooling capacity is caused by either of two conditions: an increase in the condenser water temperature or a decrease in the evaporator air temperature. The decrease in capacity can be explained by considering mass flow rates and thermodynamic states. An increase in condenser water temperature causes an increase in condenser pressure with limited effect on subcooling and mass flow rate. For these conditions the enthalpy change in the evaporator is reduced which decreases the cooling capacity. Decreasing the evaporator air temperature causes an drop in the evaporator pressure and refrigerant mass flow rate, but has limited effect on the enthalpy change in the evaporator. The reduced refrigerant mass flow rate causes a drop in capacity.

Figure 4 shows the results of a similar curve fit for COP, where the COP is calculated as the ratio of cooling capacity to compressor power. The behavior of COP is similar to that of capacity, in that the COP increases with increasing evaporator air temperature and decreases with increasing condenser water temperature. An increase in the evaporator air temperature of 10 F (5.6°C) causes an increase in the COP of approximately 7%. A 10 F (5.6°C) increase in the condenser water temperature causes a decrease in the COP of approximately 8%.

From a statistical analysis it was determined that charge had little effect on system performance for R12, but was significant for R134a. This conclusion appears to be consistent with the results of Snelson (1991). Therefore, in comparing the R12 and R134a data, the minimum R12 charge was used, since this level of charge is likely to be used in practice. Individual comparisons are then made based on each charge of R134a.

All of the curve fits used to represent the data were selected based on an analysis of the regression coefficients, standard deviation, and maximum errors associated with each possible model. The condenser water temperature has a slight second order effect on system performance while the effect of evaporator air temperature on system performance is nearly linear. The introduction of interactive terms between the two variables greatly improved the accuracy of the model. Over 99% of the observed variation in the system performance was explained with each set of data by using models with only three or four variables leaving several degrees of freedom as a check on the error. The regression coefficient, standard deviation, and maximum error for each of the curve fits presented are given in Table 1 and 2.

The behavior of the experimental COP is consistent with trends observed in a Carnot refrigeration cycle where the COP of a Carnot cycle is defined as:

$$\text{COP} = \frac{T_{cold}}{T_{hot} - T_{cold}} \tag{1}$$

For the Carnot cycle, the COP is increased by either an increase in T_{cold} or a decrease in T_{hot}, which is the same behavior that was observed in experiments.

The behavior of the cooling capacity as a function of evaporator air temperature and condenser water temperature is more difficult to predict. The capacity may be expressed as follows:

$$\text{Capacity} = \text{COP} * \dot{m}_{refg} * \Delta h_{compressor} \tag{2}$$

since the compressor power varies proportionately with refrigerant mass flow rate and change in enthalpy across the compressor. The expected behavior of the refrigerant mass flow rate is similar to that of the COP in that it is increased by either an increase in T_{cold} or a decrease in T_{hot}. Such behavior is expected since an increase in T_{cold} greatly increases the vapor density at the compressor inlet. A higher vapor density translates into a higher flow rate for a constant volume stroke in the compressor. A decrease in T_{hot} corresponds to a lower discharge pressure. Therefore, a smaller percentage of the vapor is held up in the compressor pistons giving a greater mass flow rate. Since Δh_{cmp} is fairly constant, the above equation predicts that cooling capacity, like the COP and mass flow rate, increases with an increase in T_{cold} or a decrease in T_{hot}. The similarity of curve fits for the cooling capacity and COP, shown in Figures 3 and 4, reveals that the experimental results follow the predicted behavior.

COMPARISON OF REFRIGERANTS

Mapping the performance of each refrigerant over a range of operating conditions (e.g. refrigerant charge, evaporator temperature, condenser temperature, etc.) is crucial for the following reasons:

- Since the capacity and COP cannot both be optimized simultaneously, their values must be determined for all possible operating conditions in order to make accurate comparisons.
- Changing seasons and load conditions cause the evaporator and condenser temperatures to vary, which in turn affect system performance. Varying the evaporator and condenser inlet temperatures, as was done for the data presented here, accounts for these changes in operating conditions.

In short, it is more useful and accurate to make comparisons that take all system variables into account.

For the present study comparisons are made based on air side and water side temperatures rather than on refrigerant temperatures, since local temperature differences in the heat exchangers are likely to vary for the two refrigerants and from test to test. For R12 the average temperature differences were 26 F (14°C) and 12 F (6.7°C) for the evaporator and condenser, respectively. For the R134a, the average temperature difference was 26 F (14°C) in the evaporator and 10 F (5.6°C) in the condenser.

Comparison of COP

The ratio of the COP of R134a over that of R12 for the same operating conditions shows the fractional change in COP for the system.

$$\text{COP Ratio} = \frac{\text{COP}_{\text{R134a}}}{\text{COP}_{\text{R12}}} \tag{3}$$

Values greater than one represent an improvement in performance by using R134a in the system. Since identical operating conditions were difficult to achieve, the ratios are calculated based on the curve fits for each refrigerant. The uncertainty in the original curve fits, which accounts for all random error and any lack of fit, must then be carried over to the uncertainty in the calculation of ratios. By representing the ratio with a Taylor series expansion and neglecting second order and higher terms, the variance of the ratio may be determined. By this method of analysis it was determined that the differences in system performance for the various nominal test conditions were valid statistically. A representative value of the standard deviation of the ratio is 0.0012.

Plots of the COP ratio for three different charges of R134a are shown in Figures 5 through 7. At the lowest of the three charges shown, 7.98 lb (3.62 kg) of R134a, the greatest difference in performance occurs at the highest condenser water temperatures. This peak shifts to the lower water temperatures for each successive increase in charge. A comparison of the performance of different charges shows that the charge of 8.52 lb (3.86 kg) of R134a displayed the greatest improvement in COP over the range of operating conditions with the exception of at the highest condenser to evaporator temperature differences. At this charge, of 8.52 lb (3.86 kg), which is approximately 20% higher than the charge required to produce minimal subcooling, the subcooling varied from 10 to 15 F (5.6 to 8.3°C). The effects of condenser exit subcooling on system performance have been reported in literature on a theoretical (Petersson 1990) and experimental basis (Snelson 1992). At this charge the COP of R134a varies between 0 to 5% above that of R12 for the same conditions. The greatest differences occur at the highest evaporator air temperatures.

Comparison of cooling capacity

The cooling capacities of the two refrigerants were compared in the same manner as the COPs by using ratios to evaluate differences in performance.

$$\text{Capacity Ratio} = \frac{\text{Capacity}_{\text{R134a}}}{\text{Capacity}_{\text{R12}}} \tag{4}$$

Calculation of the variance in the capacity ratios shows that the differences in capacity for the various operating conditions are statisically significant. A representative value of the standard deviation of the ratio is 0.004.

Plots of the cooling capacity for the three different charges of R134a are shown in Figures 8 through 10. The same shift in peak values that is observed with COP is seen with capacity. The peak in capacity relative to R12 shifts toward higher condenser water temperatures with increasing charge. Also, the greatest differences occur at the highest evaporator air temperatures. Again the charge of 8.52 lb (3.86 kg) of R134a shows the greatest increase in performance over the entire range of conditions, however, every other charge showed higher capacities at some condition than with R12. For the charge of 8.52 lb (3.86 kg) of R134a, the ratio of capacities varied between .98 and 1.07 with a majority of the curve showing a greater capacity with R134a.

Comparisons based on pressure ratio

In Figures 5-10, the COPs of the two refrigerants were compared based on identical test conditions. The COP may also be compared based on the pressure ratio (condenser inlet pressure/evaporator inlet pressure). A plot of COP versus pressure ratio is shown in Figure 11. The curves show that the COP decreases with increasing pressure ratio for both refrigerants which is consistent with the trends predicted by theory. Since the pressure ratio of R134a is higher than that for R12, one might conclude from theoretical considerations that the COP for R134a is lower. However, such a conclusion appears to be inaccurate based on observations of the experimental data. Figure 11 shows that in fact the pressure ratio of R134a would have to be 25% greater than that of R12 to give a lower COP. For all test conditions, the pressure ratio was much less than 25% leading to the conclusion that the COP of R134a is greater than that of R12.

CONCLUSIONS

The COP and cooling capacity of R134a proved to exceed that of R12 for the majority of test conditions. The values of COP and capacity for R134a are a maximum of 5.4% and 6.8% higher than those for R12, respectively where these differences were determined to be statistically significant. The greatest differences between the two refrigerants were observed at the highest evaporator air temperatures.

The amount of refrigerant used in the system had little effect on system performance with R12 but, had a significant effect with R134a. The charge that gave the greatest increase in performance over the range of operating conditions was at a charge 20% higher than that which first showed subcooling. The amount of superheat at the evaporator exit has a fairly insignificant effect on the total performance of the system. A superheat of 13.5 F (7.5°C) proved to be a reasonable choice for the comparison tests.

The comparison of refrigerants presented in this paper demonstrates the importance of a study which allows for variation of several independent variables. The effects of refrigerant charge, evaporator air temperature, and condenser water temperature on system performance and on comparisons of the two refrigerants were shown to be statistically significant. The data base generated by the variation of these parameters allows for studies based on other independent variables such as system pressures. Such comparisons will add in the understanding and modeling of systems and will be the topic future papers.

ACKNOWLEDGEMENTS

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Table 1: Statistical data for curve fits of COP.

	7.98lb R134a	8.52lb R134a	9.05lb R134a	8.86 lb R12
Regression Coefficient	0.997	0.998	0.996	0.996
Standard Deviation	0.011	0.016	0.005	0.006
Maximum Error	0.019	0.021	0.007	0.008

Table 2: Statistical data for curve fits of cooling capacity (BTU/min).

	7.98lb R134a	8.52lb R134a	9.05lb R134a	8.86 lb R12
Regression Coefficient	0.997	0.998	0.996	0.996
Standard Deviation	4.2	3.7	2.9	2.3
Maximum Error	7.0	6.7	5.3	2.6

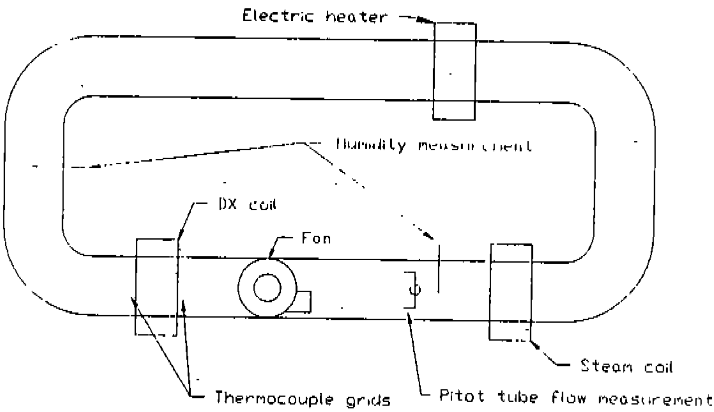

Figure 1: Schematic of air flow loop.

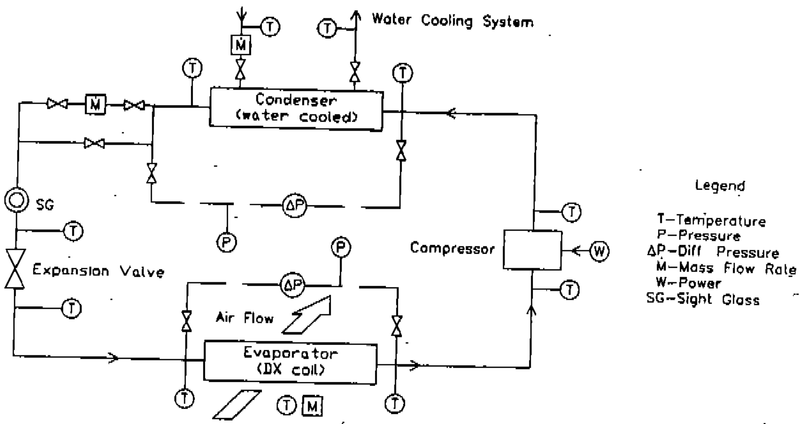

Figure 2: Schematic of vapor compression refrigeration system.

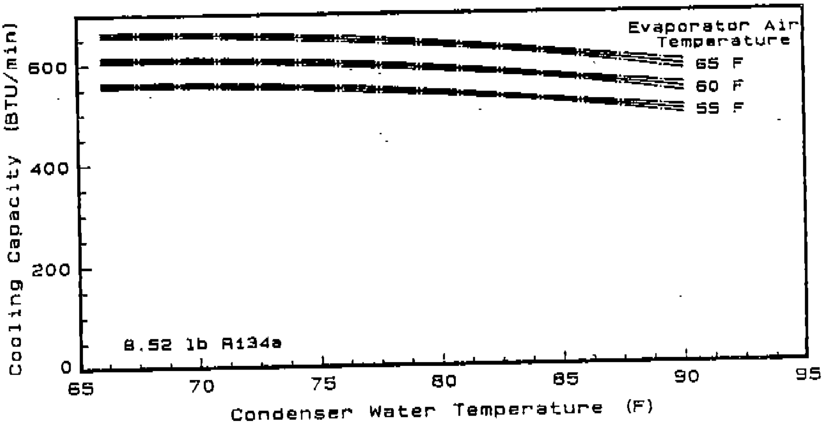

Figure 3: Evaporator capacity with 8.52 lb R134a.

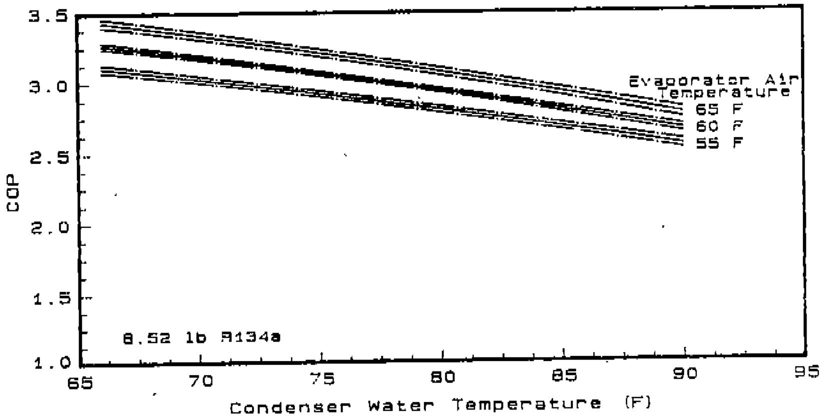

Figure 4: Coefficient of performance with 8.52 lb R134a.

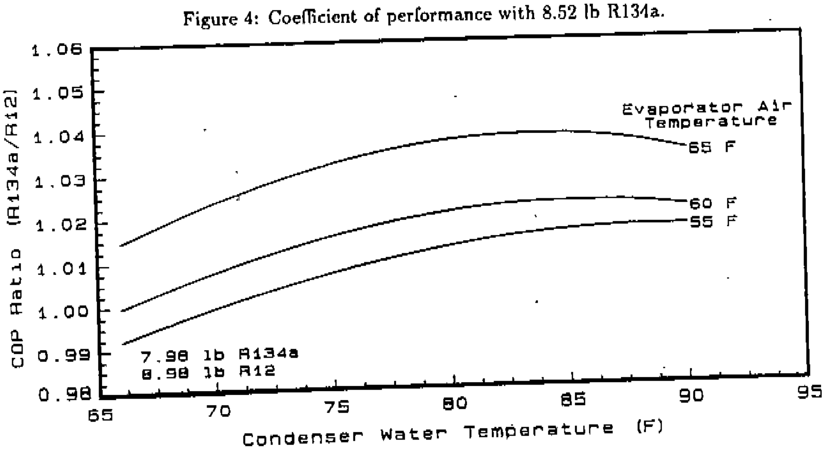

Figure 5: COP ratio with 7.98 lb R134a and 8.86 lb R12.

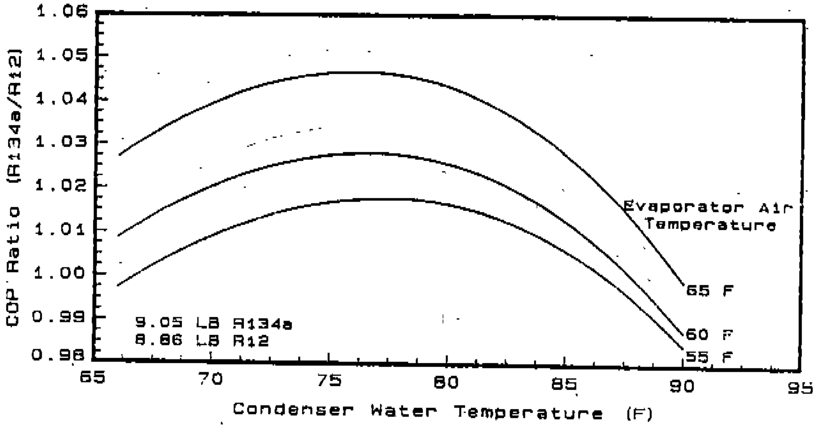

Figure 6: COP ratio with 8.52 lb R134a and 8.86 lb R12.

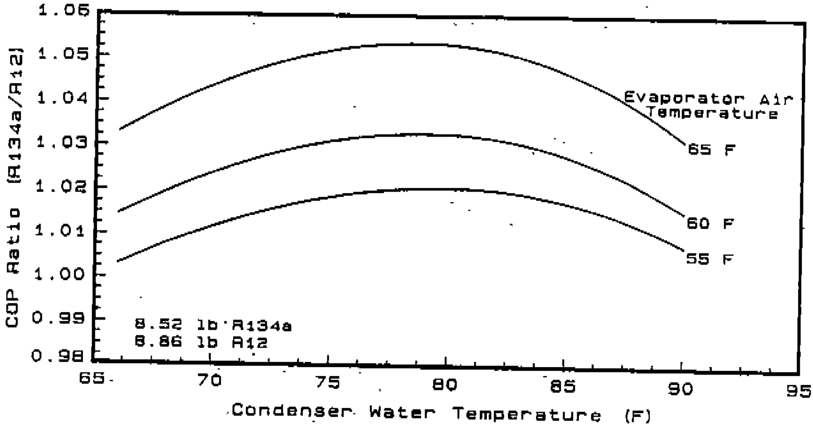

Figure 7: COP ratio with 9.05 lb R134a and 8.86 lb R12.

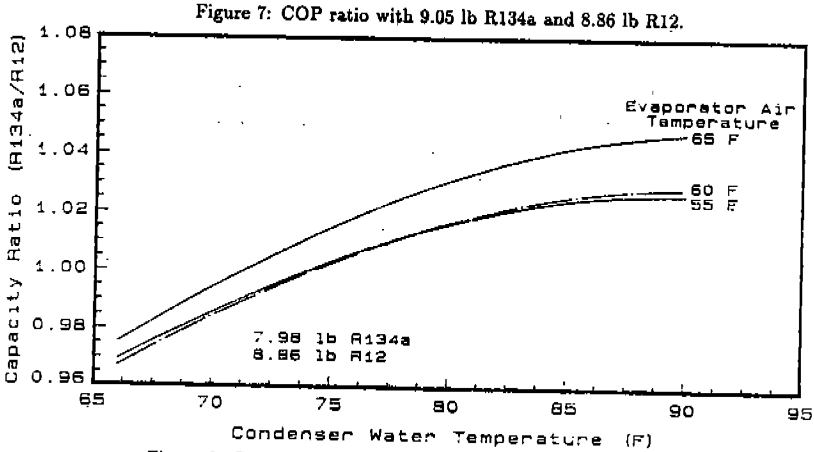

Figure 8: Capacity ratio with 7.98 lb R134a and 8.86 lb R12.

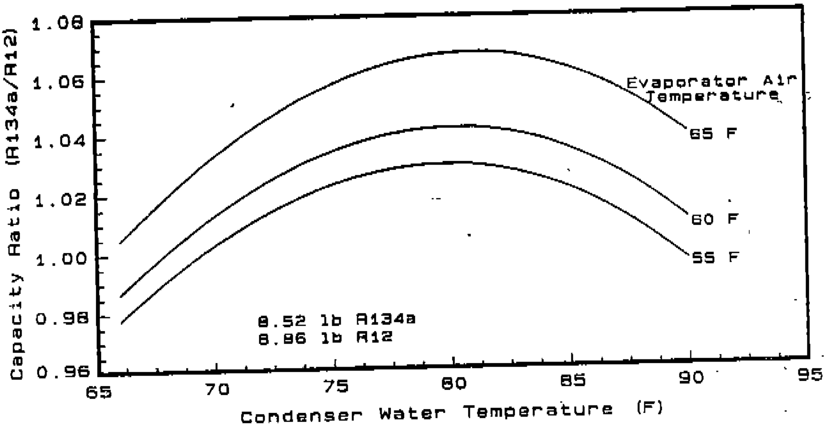

Figure 9: Capacity ratio with 8.52 lb R134a and 8.86 lb R12.

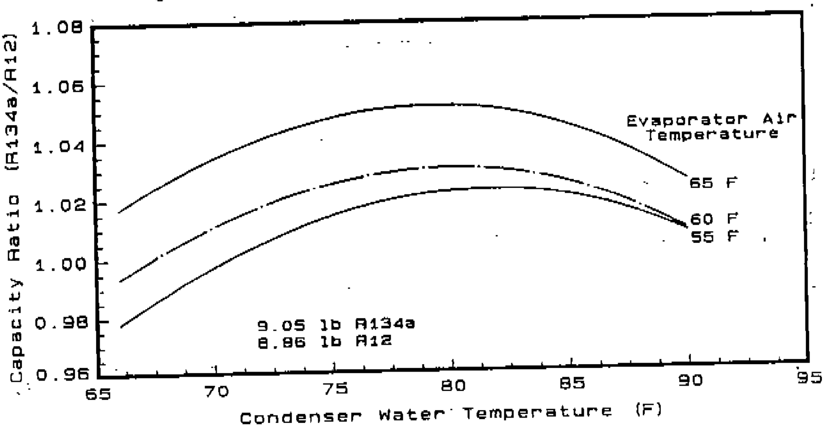

Figure 10: Capacity ratio with 9.05 lb R134a and 8.86 lb R12.

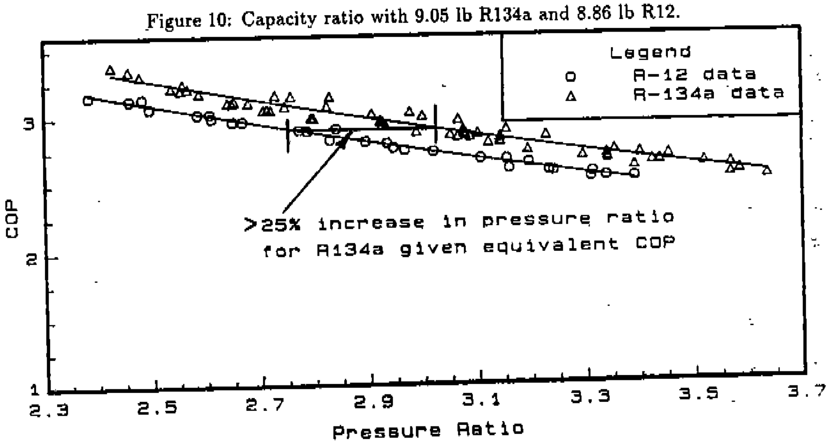

Figure 11: COP versus pressure ratio for all data.