1994

HCFC-22 Alternatives for Air Conditioners and Heat Pumps

M. B. Shiflett
DuPont Fluoroproducts

HCFC-22 ALTERNATIVES FOR AIR CONDITIONERS AND HEAT PUMPS

M.B. Shiflett

DuPont Fluoroproducts
Chestnut Run Plaza
Wilmington, DE 19880

ABSTRACT

Hydrochlorofluorocarbons (HCFCs) have been added to the list of chlorine containing refrigerants scheduled for phase-out by the Montreal Protocol. Several alternative refrigerants have been proposed for replacing HCFC-22 with interest focused on HFC-134a, HFC-32/HFC-125, and HFC-32/HFC-125/HFC-134a. This paper will compare experimental performance data for HFC-32/HFC-125/HFC-134a in a split system residential heat pump. In addition, equipment modifications for improving heating capacity and energy efficiency will be discussed.

INTRODUCTION

HCFC-22 has been widely used in the air conditioning and heat pump industry, especially in residential unitary and central air conditioning systems, for many years. Because HCFC-22 has been readily available, inexpensive, and less harmful to the environment than CFC's, it has become the alternative of choice for many new applications. Concern over the long-term effects of HCFCs such as HCFC-22 on atmospheric ozone concentrations led DuPont to announce intentions to discontinue sale of HCFC-22 for all but service applications by January 1, 2005 and for all applications by January 1, 2020. Because HCFC-22 is used in a large variety of applications, more than one alternative may be needed to provide optimum performance for all applications.

DuPont has been involved in several initiatives for evaluating HCFC-22 alternatives. One such industry program is the Alternative Refrigerants Evaluation Program (AREP) initiated by the Air Conditioning & Refrigeration Institute (ARI). The objective of this program is to provide performance data on replacement refrigerants in compressors, system components, and/or systems by conducting tests with participating member companies. Several HCFC-22 alternative candidates have been proposed. Among those, interest has focused on three candidates: HFC-134a, and two mixtures HFC-32/HFC-125/HFC-134a (23/25/52 wt.%), and HFC-32/HFC-125 (50/50 wt.%). HFC-134a and HFC-32/HFC-125 are not equivalent pressure and capacity matches with respect to HCFC-22 and would require system redesign for use in new equipment. Both appear promising candidates to replace HCFC-22 in certain new design applications. HFC-32/HFC-125/HFC-134a offers the closest match to HCFC-22 in existing equipment with respect to both capacity and energy efficiency in addition to other performance measures such as compressor discharge temperature and pressure /1/. This paper will compare experimental performance data for HFC-32/HFC-125/HFC-134a versus HCFC-22 in a split system residential heat pump /2/. In addition, equipment modifications for improving heating capacity and energy efficiency will be discussed.

The mixture of HFC-32/HFC-125/HFC-134a was formulated to meet many criteria, three of which were: (1) to provide similar capacity as HCFC-22 in air conditioners and heat pumps, (2) to have maximum possible energy efficiency, and (3) to be nonflammable and remain nonflammable under normal and abnormal operating conditions. These conditions include leaks and the case of charging systems from the vapor space of cylinders. Although mixtures must be liquid charged into systems to maintain the desired refrigerant composition, it is inevitable that vapor charging will occasionally occur due to mistakes, misunderstandings, or inadequate service person training. Product stewardship risk analysis requires that refrigerants be formulated to remain nonflammable during such product mishandling. The 23 wt% HFC-32, 25 wt% HFC-125, 52 wt% HFC-134a mixture was selected as the nonflammable HCFC-22 alternative which met all of these criteria /3,4/.

HEAT PUMP AND INSTRUMENTATION

The heat pump was designed to operate with HCFC-22 and had a rated capacity of about 30,000 Btu/hr (8.8 kW). The unit was equipped with a reciprocating compressor, a fixed orifice for cooling, and expansion valve for heating, a fin and tube indoor coil with four circuits, and a spined fin outdoor coil with five circuits and one subcooling circuit. The heat pump was set up in two environmentally controlled chambers so the dry and wet bulb temperatures for the indoor and outdoor coil could be maintained at test conditions according to ARI standards. The 95°F (35°C) and 82°F (27.8°C) cooling and 47°F (8.3°C) and 17°F (-8.3°C) heating tests were selected in order to verify steady state performance over a wide range of operating conditions /5/.

The cooling and heating capacity was measured on both the air and refrigerant side of the indoor coil. The air side sensible heat difference was measured using thermopiles. The air side latent heat difference was measured using two techniques. The first technique involved using dew point meters while the second technique involved collecting the condensate from the indoor coil. Both techniques worked very well and were typically within 1%. Energy balances between the air side and refrigerant side were within 1 to 3%. The capacity data reported is based on the air side measurements. Refrigerant temperatures were measured using T type (copper-constantan) thermocouples mounted on the surface of the copper tubes well insulated from the ambient. System pressures were measured using electronic transducers. A coriolis mass flow meter was installed in order to measure refrigerant flow rate. Using a compressor calorimeter map the mass flow meter was calibrated for the range of flow rates expected in order to improve accuracy. The flow meter was installed in the liquid line just before the fixed orifice tubes feeding the indoor coil. Sight glasses were mounted on both ends of the flow meter to ensure single liquid phase flow and a four way valve was installed to maintain unidirectional flow through the meter. A schematic of the instrumentation used on the refrigerant side of the system is provided in Figure 1.

A blower was located at the outlet of the air tunnel with a damper in order to maintain 1000 scfm of air across the indoor coil. The voltage to the compressor contacts was maintained at 230 volts for all tests using a automatic voltage regulator. Power consumption was measured using a single phase digital power meter and the energy efficiency ratio was based on the power required to operate the compressor and outdoor fan. The original fixed orifice tubes and an expansion valve were used for testing both HCFC-22 and HFC-32/HFC-125/HFC-134a. The same two piston reciprocating compressor was also used for testing HCFC-22 and HFC-32/HFC-125/HFC-134a and contained a polyol ester lubricant which was used for both refrigerants.

TEST RESULTS

The first set of experiments were conducted at the 95°F (35°C) cooling test condition to determine the optimum charge size for HCFC-22 and HFC-32/HFC-125/HFC-134a (23/25/52%). The charge sizes for HCFC-22 and HFC-32/HFC-125/HFC-134a were selected based on maintaining 10°F (5.6°C) superheat. Superheat was determined by measuring the suction line pressure and temperature at the inlet to the outdoor unit. Superheat was defined as the difference between the saturation temperature calculated from the pressure and the actual superheated gas temperature measured. The saturation temperature for the mixtures was defined by taking the average between the bubble point and dew point temperatures. The charge sizes selected for HCFC-22 and HFC-32/HFC-125/HFC-134a were 9.8 and 9.2, respectively. The same charge size for each refrigerant was used for the remaining steady state tests.

Figure 2 compares the capacity ratio for the HFC-32/HFC-125/HFC-134a mixture versus HCFC-22 at the four test conditions. HFC-32/HFC-125/HFC-134a provided essentially the same cooling capacity as compared with HCFC-22 with no equipment modifications. During heating the HFC-32/HFC-125/HFC-134a mixture capacity decreased 2 to 4% versus HCFC-22 because the expansion valve used for heating was not optimized for the mixture. Additional work has shown that HFC-32/HFC-125/HFC-134a will provide the same heating capacity as HCFC-22 and depending on equipment some minor adjustments to expansion valves (EV) may be necessary to optimize performance.

System modifications for improving the heating capacity for the ternary mixture were considered. Based on experimental data the addition of a suction line accumulator (AC) allows mixture composition shifting during the heating cycle. HFC-32 and HFC-125 concentrations increase from 1 to 6 and 1 to 3 wt%, respectively depending on the amount of HFC-134a stored in the accumulator. The net result is the heating capacity can be increased 3% during the 47°F (8.3°C) heating test and 6% during the 17°F (-8.3°C) heating test. The additional capacity will help reduce the amount of supplemental heat required during the heating season.

Figure 3 compares the energy efficiency ratio for the ternary mixture at the four test conditions. The energy efficiency ratio for HFC-32/HFC-125/HFC-134a versus HCFC-22 during cooling and heating ranged from 0.95 to 0.97. System modifications for improving cooling cycle energy efficiency were also considered. The benefits of counterflow evaporators and condensers (XC) and liquid line/suction line heat exchange (LSHX) were investigated in cooperation with the National Institute of Standards and Technology (NIST) using their CYCLE-11 computer model /6,7/. Results from the computer model calculations are shown in Figure 3. The calculated data indicate that utilizing counterflow evaporator and condenser operation can increase the ternary mixture energy efficiency by 6 to 7%. The counterflow heat exchangers were only considered for air conditioning due to additional complexity in heat pump applications. Calculated data indicate that the use of a LSHX could increase the HFC-32/HFC-125/HFC-134a mixture energy efficiency an additional 2%. The net result indicates that improvements of 3 to 5% in energy efficiency versus HCFC-22 may be obtainable for cooling with the HFC-32/HFC-125/HFC-134a mixture. In addition, the HFC-32/HFC-125/HFC-134a cooling capacity utilizing these equipment changes would be increased by 10% relative to HCFC-22.

HFC-32/HFC-125/HFC-134a had 9 to 12°F (5 to 6.7°C) lower compressor discharge temperatures compared with HCFC-22 depending on operating condition (see Figure 4). The lower temperatures during operation should provide a positive effect on life expectancy of the compressor and lubricant. Figure 5 compares the compressor discharge pressures. HFC-32/HFC-125/HFC-134a had a 4 to 14% higher discharge pressure compared with HCFC-22 depending on operating condition.

CONCLUSIONS

A mixture of 23 wt% HFC-32, 25 wt% HFC-125, and 52 wt% HFC-134a provided similar cooling and heating capacity with a 3 to 4% reduction in energy efficiency compared with HCFC-22 in an unmodified split system heat pump. The addition of a suction line accumulator can shift the circulating HFC-32 concentration by 1 to 6 wt% which can provide up to 6% additional heating capacity. In addition, the use of cross flow heat exchangers and a liquid line/suction line heat exchanger should increase energy efficiency and cooling capacity by as much as 8 to 10%. SUVA® AC9000 provides both original equipment manufactures and service groups the closest match in performance to HCFC-22 with no significant equipment modifications required. In addition, further opportunities for improving performance can be achieved by optimizing equipment.

ACKNOWLEDGMENTS

The author would like to thank the following people who helped design and construct equipment, measured experimental data, and provided valuable discussions: D.B. Bivens, L. Bracale, H.A. Hammond, K. Hogg, T.M. Luko, R. Parkell, D.M. Patron, and A. Yokozeki. The author would also like to thank the engineers in the Trane Unitary Products Group for providing assistance with equipment selection and test methodology.

REFERENCES

1. Patron, D.M. 1994. Performance Test Results. Seminar Presentation at ASHRAE winter meeting, New Orleans, LA, January 23.

2. The Trane Company, Tyler Texas, private communications.

3. Bivens, D.B., et al. 1994. HCFC-22 Alternative for Air Conditioners and Heat Pumps. Symposium paper presented at ASHRAE annual meeting, Orlando, FL, June 26.

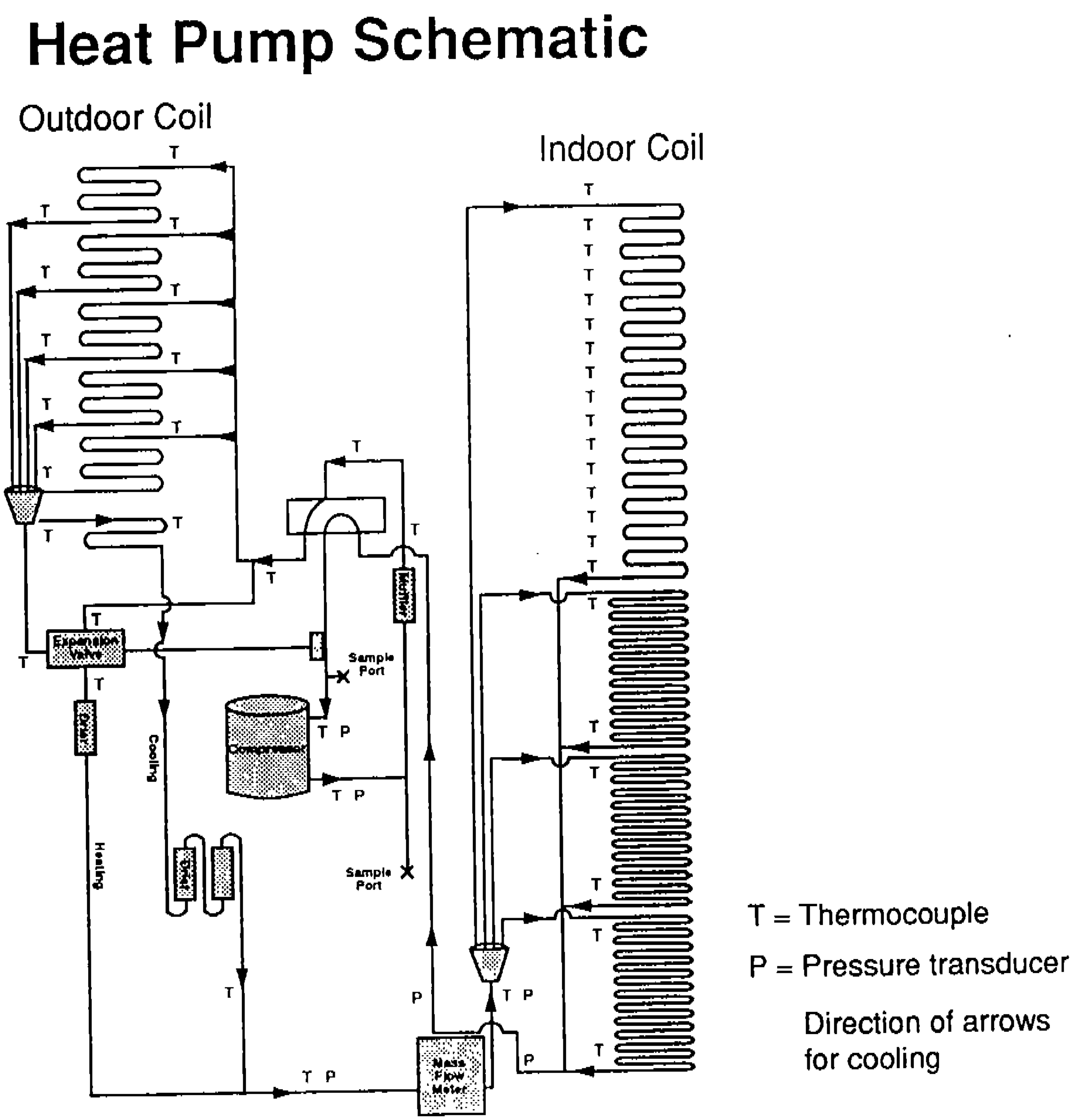
4. ASTM 1985. Standard Test Method for Concentration Limits of Flammability of Chemicals, ASTM Designation: E681-85, American Society for Testing and Materials.

5. Air-Conditioning and Refrigeration Institute (ARI). 1989. Standard for Unitary Air-Conditioning and Air-Source Heat Pump Equipment, Standard 210-240.

6. Domanski, P.A., and D.A. Didion. 1993. Thermodynamic Evaluation of R-22 Alternative Refrigerants and Refrigerant Mixtures. ASHRAE Transactions, Vol. 99, Part 2, DE-93-6-2.

7. Mulroy, W.J. 1993. Private communication. National Institute of Standards and Technology, Gaithersburg, MD, November 19.

Figure 1


Heat Pump Schematic
Outdoor Coil
Indoor Coil
Expansion Valve
Dryer
Cooling
Heating
Sample Port
Compressor
Muffler
Sample Port
Mass Flow Meter
T = Thermocouple
P = Pressure transducer
Direction of arrows
for cooling

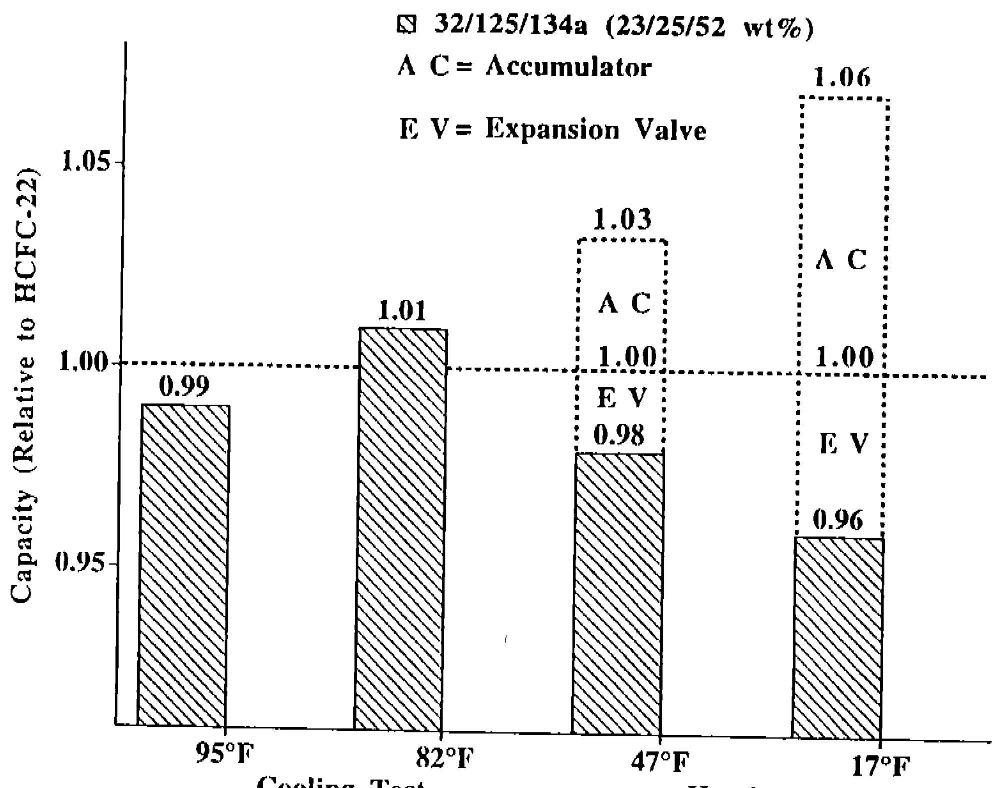


Figure 2

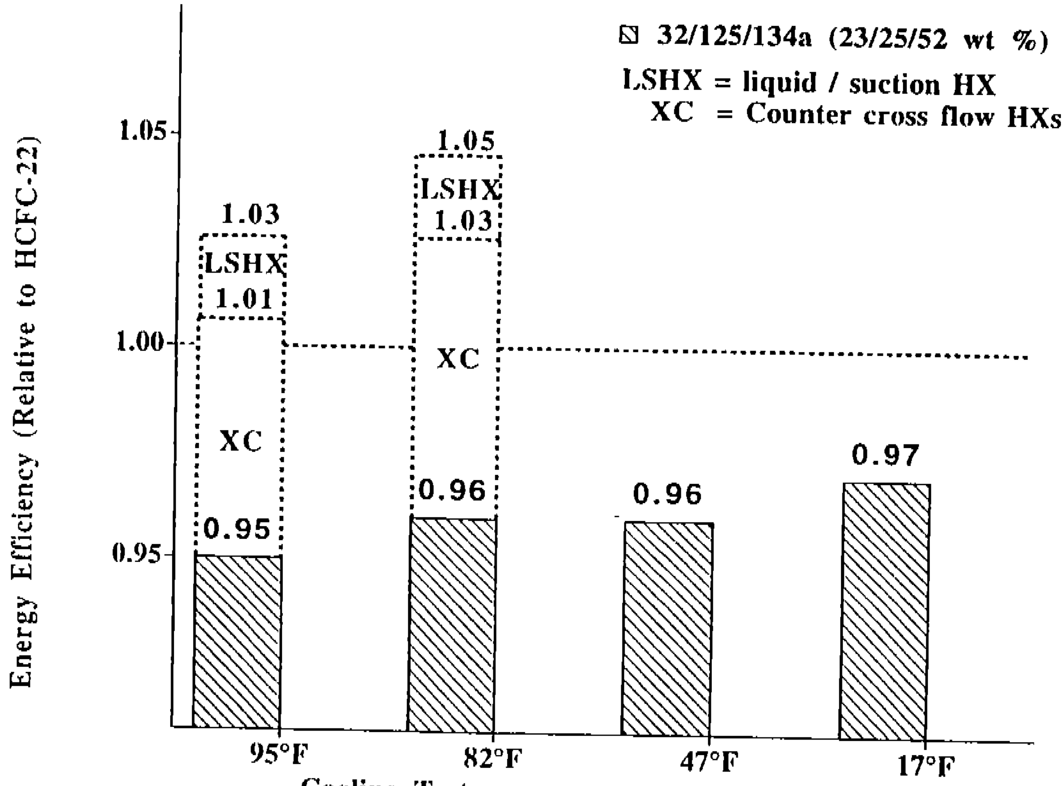


Figure 3

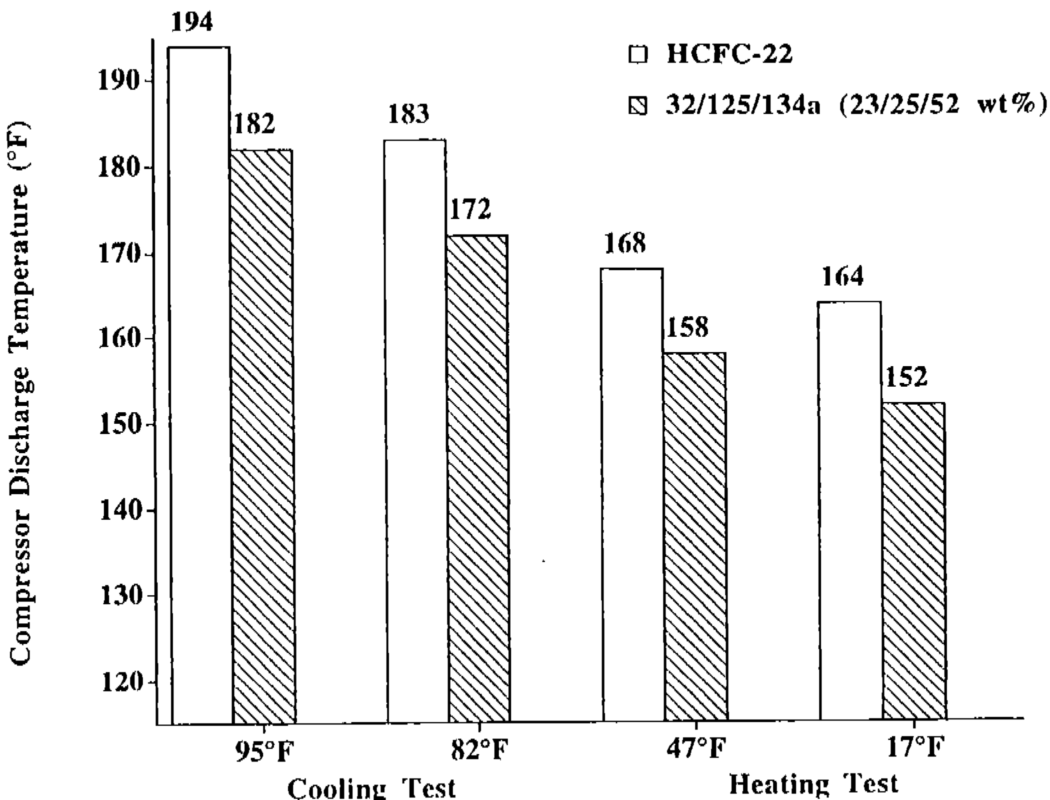


Figure 4

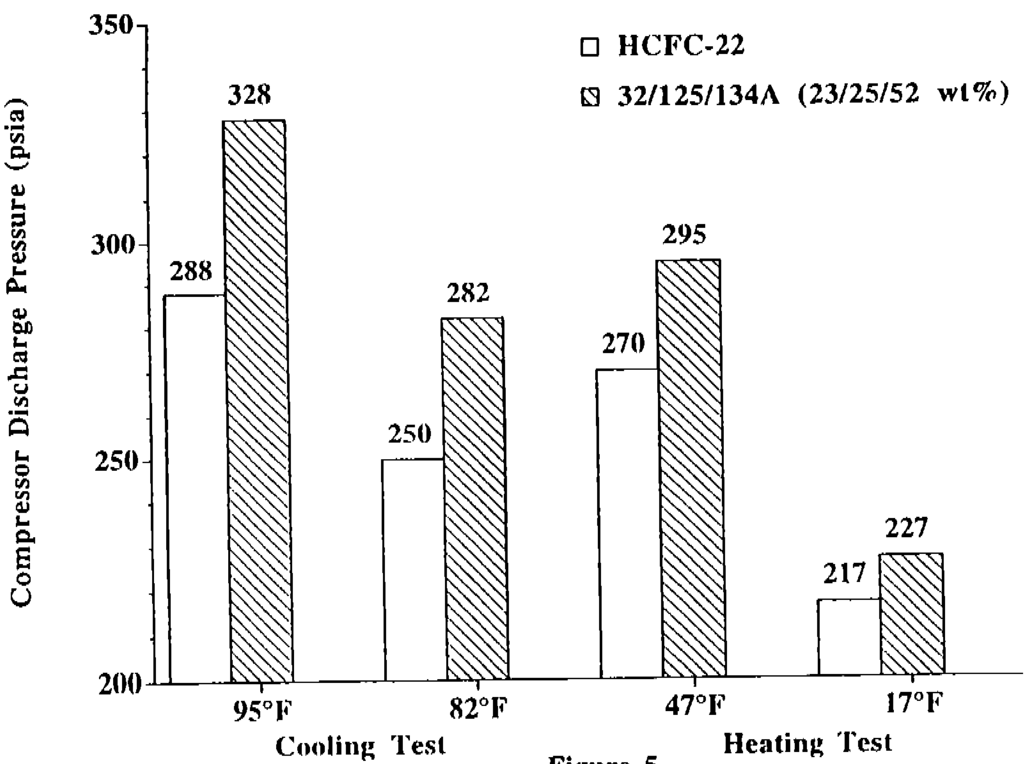


Figure 5