EPA/600/A-96/114

ALTERNATIVES TO CFC-114 IN HIGH-TEMPERATURE HEAT PUMPS: COMPRESSOR PERFORMANCE WITH HFC-236EA AND HFC-236FA

Georgi S. Kazachki
Acurex Environmental Corporation
P.O. Box 13109
Research Triangle Park, NC 27709

Cynthia L. Gage and Robert V. Hendriks
U.S. Environmental Protection Agency
National Risk Management Research Laboratory
Research Triangle Park, NC 27711

ABSTRACT

Comprehensive calorimeter tests on a semi-hermetic compressor were performed with CFC-114, HFC-236ea, and HFC-236fa over a wide range of temperature test conditions: evaporating temperatures from 0 to 35°C and condensing temperatures from 40 to 110°C. More than 600 tests were run at 66 test conditions with the three refrigerants. The following parameters were assessed as criteria for performance evaluation and for reliable performance: cooling capacity; electric power input, current, and voltage; coefficient of performance; compressor volumetric and isentropic efficiencies; and discharge and oil temperatures. Polyolester oil was used as lubricant in the compressor. The oil charge was unchanged when switching refrigerants. With all three refrigerants, the compressor ran 1,800 hours without failure and without indication of excessive noise or vibration.

INTRODUCTION

For decades, chlorofluorocarbon (CFC)-114 has been the refrigerant of choice in high-temperature, high-lift vapor-compression heat pumps. Its thermal stability and wet isentropic compression allowed for operation at condensing temperatures of up to 120°C and in some instances even higher without refrigerant decomposition or chemical reactions with the oil or with construction materials. The wet isentropic compression of its saturated vapor meant that the discharge temperatures did not significantly exceed the condensing temperatures even at high suction superheat. As a result, the compressor was able to handle high condensing temperatures without risk of damage. After the ban on the production of chlorine-containing refrigerants, a replacement for CFC-114 was needed. Since this refrigerant was also the refrigerant of choice in surface craft and submarine air-conditioning systems, the need for alternatives to CFC-114 became even more urgent. Under these circumstances, the Environmental Protection Agency (EPA) in cooperation with the Electric Power Research Institute (EPRI) initiated a search for new compounds to be identified and synthesized as replacements for all chlorine-containing refrigerants (1,2,3). From this work, two isomers were recognized as primary candidates for CFC-114 alternatives: hydrofluorocarbon (HFC)-236ea and HFC-236fa.

Early analyses focused on HFC-236ea as a replacement for CFC-114 in chiller applications (4,5,6). Calorimeter tests with a semi-hermetic compressor were also reported at chiller conditions for this refrigerant (7). Later, thermodynamic evaluations were performed on both HFC-236fa and HFC-236ea at high-temperature heat pump conditions (8). These evaluations indicated that both refrigerants would have wet isentropic compression similar to CFC-114. However, HFC-236fa had cooling capacities from 0 to 20% higher than CFC-114, while HFC-236ea was predicted to have capacities ±6% of those for CFC-114 at all but the highest condensing temperature. The COPs (coefficients of performance) for both HFC-236fa and HFC-236ea were found to be close to those for CFC-114 except at condensing temperatures greater than 80°C.

Atmospheric lifetimes of the two isomers have been measured(9). HFC-236fa has a relatively long atmospheric lifetime of 194 years, while the lifetime for HFC-236ea is only 10 years. Recent material compatibility studies show that both isomers have acceptable performance with several different elastomers and desiccant materials(10).

TEST METHOD AND PROCEDURE

Experimental evaluation of the CFC-114 alternatives was done on a compressor calorimeter test rig with a semi-hermetic compressor. The original calorimeter was modified for low-pressure refrigerants to eliminate excessive pressure drops. These modifications were detailed in earlier publications(7). The semi-hermetic compressor had a 0.56 kW motor and delivered 1.329 L/s at 1750 RPM. The compressor was designed for use with HFC refrigerants and was lubricated by polyolester oil.

Tests were performed using ASHRAE Standard 23-1993(11) as a basis. The cooling capacity was determined by two methods: a primary method based on the quantity of electrically supplied heat to the calorimeter boiler and a secondary method based on the heat balance of the water-cooled condenser. The secondary method was not available at the highest condensing temperature because the water flashed to steam. Agreement between the two methods was better than 3%. Each test condition was evaluated three times, and the resulting values were averaged. The standard deviation of the average value for any given parameter was never greater than 2%. The primary method was also used to determine the refrigerant mass flowrate. This value was used, along with the enthalpy difference between the compressor discharge at the measured conditions and the condition of saturated liquid at the condensing pressure, to determine the heating capacities. With all three refrigerants, the compressor ran 1,800 hours without failure and without indication of excessive noise or vibration, a positive indicator for field performance.

Tests were performed at evaporating temperatures from 0 to 35°C and condensing temperatures from 40 to 110°C. A few degrees of subcooling was achieved in the condenser to ensure liquid feeding to the expansion valve. Similarly some superheating was achieved in the evaporator to ensure that no liquid reached the compressor to avoid wet compression. All results for the cooling capacities were corrected back to conditions of saturated liquid leaving the condenser and saturated vapor leaving the evaporator.

RESULTS

Table 1 presents an example table for HFC-236ea at 32.2°C evaporating and 65.6°C condensing temperatures. Similar tables were generated for all refrigerants at all test conditions.

Due to the significant number of tests performed (more than 600 tests at 66 test conditions with the three refrigerants), representative results at 10 and 32.2°C evaporating temperatures are presented graphically. As a result of the upper capacity limit of the calorimeter, data could not be collected for HFC-236fa at the 32.2°C evaporating and 51.7°C condensing temperatures. The lower critical temperature for HFC-236fa (130.65°C) compared to HFC-236ea(141.15°C) and CFC-114 (145.65°C) also prevented measurements at 107.2°C for this chemical.

Figures 1 and 2 present the cooling capacities of all three refrigerants at 10°C and 32.2°C evaporating temperature, respectively. As predicted, cooling capacities for HFC-236fa were from 0 to 25% higher than for CFC-114. For HFC-236ea, the cooling capacities as compared to CFC-114 were slightly lower than predicted. Heating capacities for the three refrigerants at both temperatures are shown in Figures 3 and 4. Again, HFC-236fa had capacities up to 25% higher than CFC-114, while HFC-236ea has heating capacities slightly less.

Electrical power input to the compressor is shown in Figures 5 and 6. At 10°C evaporating, the power input requirements for HFC-236fa were about 10% more than for CFC-114. This requirement increased to about 14% at 32.2°C evaporating. HFC-236ea required about 5% less electrical input at the lower evaporating temperature and had almost identical power input as CFC-114 at the higher temperature. The cooling COP is shown in Figures 7 and 8. For HFC-236fa, the COP is similar to higher than CFC-114 at condensing temperatures lower than 79.4°C. This performance was better than predicted. The performance becomes up to 10% poorer at the higher condensing temperatures. For HFC-236ea, the COPs are similar to lower than CFC-114 at all conditions. This agrees with the theoretical expectations.

Compressor isentropic energy efficiency is shown in Figures 9 and 10. For HFC-236fa, this efficiency is higher than CFC-114 at all conditions. This explains why the measured COPs were higher than expected. The energy efficiency for HFC-236ea is similar to CFC-114 at condensing temperatures less than 79.4°C, but is poorer at higher condensing temperature. Compressor volumetric efficiencies are presented in Figures 11 and 12. For HFC-236fa, these efficiencies are very close to those of CFC-114. For HFC-236ea, the efficiencies are lower than for CFC-114, explaining why measured cooling capacities were lower than predicted.

CONCLUSIONS

1. Both HFC-236fa and HFC-236ea are viable alternatives for CFC-114 in high temperature heat pumps. The shorter atmospheric lifetime for HFC-236ea is also favorable.

2. HFC-236fa was found to have better cooling and heating capacities than both HFC-236ea and

CFC-114 at almost all conditions tested. Capacities for HFC-236ea were similar to those of CFC-114.

3. Electric power input for HFC-236fa was higher than for CFC-114, while input for HFC-236ea was lower. When combined with cooling capacities, this resulted in COPs similar to CFC-114 for both alternatives.

4. The low critical temperature for HFC-236fa may be a limiting factor in some heat pump applications.

REFERENCES

1. Adcock, J.L., S.B. Mathur, W.A.Van Hook, H.Q.Huang, M. Narkharde, and B.H.Wang, "Fluorinated Ethers: A New Series of CFC Substitutes," Proceedings of the 1991 International CFC and Halon Alternatives Conference, Baltimore, MD, 1991, 386-395.
2. Beyerlein, A.L., D.D. Des Marteau, S.H. Hwang, N.D. Smith, and P. Joyner, "Physical Property Data on Fluorinated Propanes and Butanes as CFC and HCFC Alternatives," Proceedings of the 1991 International CFC and Halon Alternatives Conference, Baltimore, MD, December 1991, 396-405.
3. Smith, N.D. "New Chemical Alternatives for CFCs and HCFCs," EPA-600/F-92-012, March 1992.
4. Smith, N.D., K. Ratanaphruks, M.W. Tufts, and A.S. Ng, "HFC-236ea: A Potential Alternative for CFC-114," Proceedings of the 1993 International CFC and Halon Alternatives Conference, Washington DC, October 1993, 150-157.
5. Kazachki, G.S., and C.L. Gage, "Thermodynamic Evaluation and Compressor Characteristics of HFC-236ea and HFC-245ca as CFC-114 and CFC-11 Replacements in Chillers," Proceedings of the 1993 International CFC and Halon Alternatives Conference, Washington, DC, October 1993, 167-176.
6. Kazachki, G.S., and R.V. Hendriks, "Calorimeter Tests of HFC-236ea as a CFC-114 Alternative and HFC-245ca as a CFC-11 Alternative," Proceedings of the 1993 International CFC and Halon Alternatives Conference, Washington, DC, October 1993, 158-166.
7. Kazachki, G.S., and R.V. Hendriks, "Performance Testing of a Semi-hermetic Compressor with HFC-236ea at CFC-114 Chiller Conditions," Proceedings of the 1994 International Refrigeration Conference at Purdue University, West Lafayette, IN, July 1994, 407-411.
8. Kazachki, G.S., C.L. Gage, R.V. Hendriks, and W.J. Rhodes, "Investigation of HFC-236ea and HFC-236fa as CFC-114 Replacements in High-Temperature Heat Pumps," CFCs: The Day After, Padua, Italy, September 1994, 155-162.
9. DeMore, W.B., S.P. Sander, C.J. Howard, A.R. Ravishankara, D.M. Golden, C.E. Kolb, R.F. Hampson, M.J. Kurylo, and M.J. Molina, "Chemical Kinetics and Photochemical Data for Use in Stratospheric Modeling, Evaluation Number 11," Jet Propulsion Laboratory Publication 94-26, 1994.
10. Ratanaphruks, K., M.W. Tufts, A.S. Ng, and N.D. Smith,"Material Compatibility Evaluations of HFC-245ca, HFC-245fa, HFE-125, HFC-236ea, and HFC-236fa,"For presentation at the International Conference On Ozone Protection Technologies, Washington, DC, October 1996.

11. ASHRAE Standard 23-1993, "Methods of Testing for Rating Positive Displacement Refrigerant Compressors and Condensing Units," American Society of Heating, Refrigerating and Air-Conditioning Engineers, Atlanta, GA, 1993.

TABLE 1: TEST CONDITIONS AND MAIN RESULTS FROM THE CALORIMETER TESTS OF THE SEMI-HERMETIC COMPRESSOR WITH HFC-236ea:

Symbols: $\mathbf{p_1}$ - suction pressure, kPa; $\mathbf{p_2}$ - discharge pressure, kPa; $\mathbf{T_{Ei}}$ - temperature at the evaporator inlet, °C; $\mathbf{T_{EST}}$ - saturated vapor temperature at suction pressure, °C; $\mathbf{T_{CST}}$ - saturated temperature at discharge pressure, °C; $\mathbf{T_{Ci}}$ - temperature at the condenser inlet, °C; $\mathbf{T_{SH}}$ - temperature at the evaporator exit, °C; $\mathbf{T_{SC}}$ - temperature before the expansion valve, °C; $\mathbf{T_1}$ - temperature at the compressor inlet, °C; $\mathbf{T_2}$ - temperature at the compressor outlet, °C; $\mathbf{T_{OIL}}$ - oil temperature at the sump bottom, °C; $\mathbf{Q_E^P}$ - cooling capacity from the primary method, W; $\mathbf{Q_E^{SC}}$ - cooling capacity from the condenser secondary method, W; $\mathbf{\delta Q}$- relative deviation of the secondary condenser method from the primary method, %; $\mathbf{P_{el}}$ - electrical input power into the compressor at the real conditions, W; **V** - Voltage, V; $\mathbf{I_C}$ - compressor motor current, A; **COP** - coefficient of performance; $\mathbf{Q_H}$ - heating capacity (heat transferred to the water), W; $\mathbf{COP_H}$ - heating coefficient of performance; λ_C - compressor volumetric efficiency, %; η_C - compressor energy efficiency, %; **SD** - standard deviation.

Remarks: 1. Test conditions: $T_E = 32.2°C$ $T_C = 65.6°C$

Test	p_1	p_2	T_{Ei}	T_{EST}	T_{Ci}	T_{CST}	T_{SH}	T_{SC}	T_1	T_2	T_{OIL}	Q_E^P	Q_E^{SC}	δQ	P_{el}	V	I_C	COP	Q_H	COP_H	λ_C	η_C
325	260.9	696.9	32.3	32.2	71.4	65.6	42.1	60.1	41.6	72.4	51.3	1951	1971	1.0	611	115	7.11	3.2	2553	4.18	82.1	45.5
326	260.3	696.8	32.4	32.2	71.7	65.6	42.1	60.2	41.6	72.8	52.2	1951	1975	1.2	612	115	7.13	3.2	2561	4.18	82.4	45.5
327	260.8	696.7	32.4	32.2	71.7	65.6	42.1	60.3	41.6	72.8	52.2	1956	1979	1.1	612	115	7.12	3.2	2565	4.19	82.4	45.6
Avg	260.7	696.8	32.4	32.2	71.6	65.6	42.1	60.2	41.6	72.7	51.9	1953	1975	1.1	612	115	7.1	3.2	2560	4.18	82.3	45.5
SD	0.32	0.1	0.1	0.0	0.2	0.0	0.0	0.1	0.0	0.2	0.5	3.2	3.8	NA	0.5	0.0	0.0	0.0	6.1	0.01	0.15	0.03

NA - not applicable

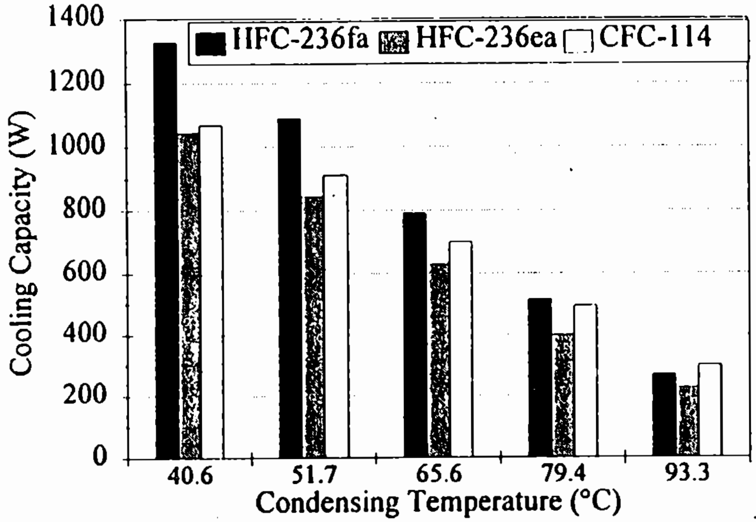

Figure 1: Cooling Capacity at 10°C Evaporating.

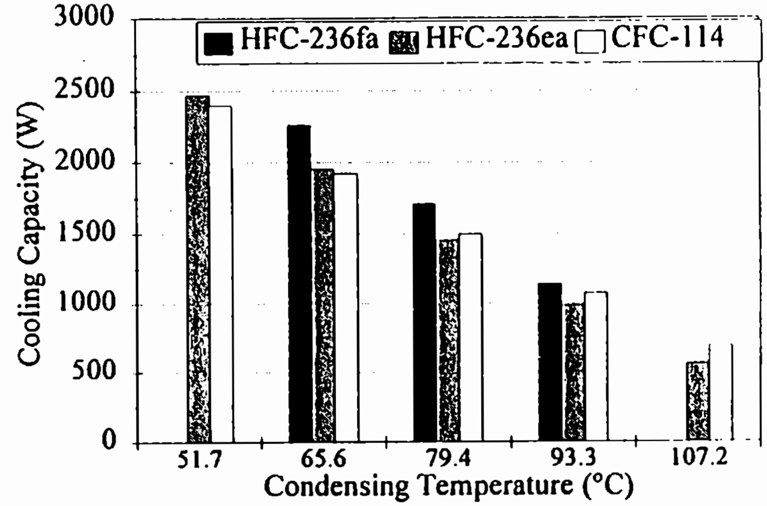

Figure 2: Cooling Capacity at 32.2°C Evaporating.

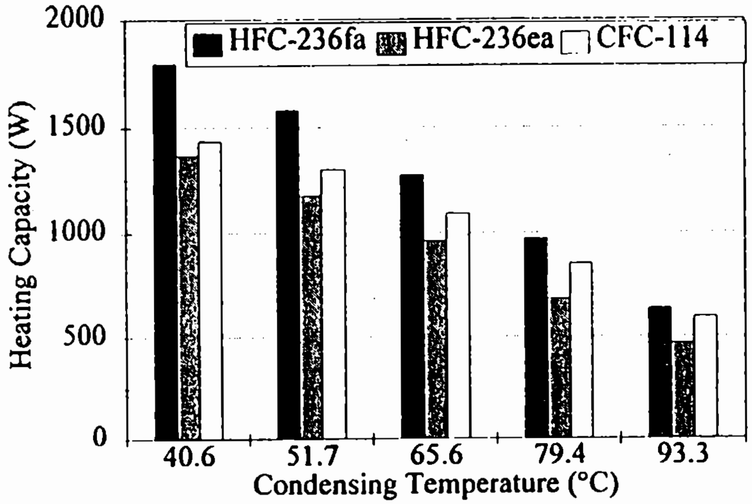

Figure 3: Heating Capacity at 10°C Evaporating.

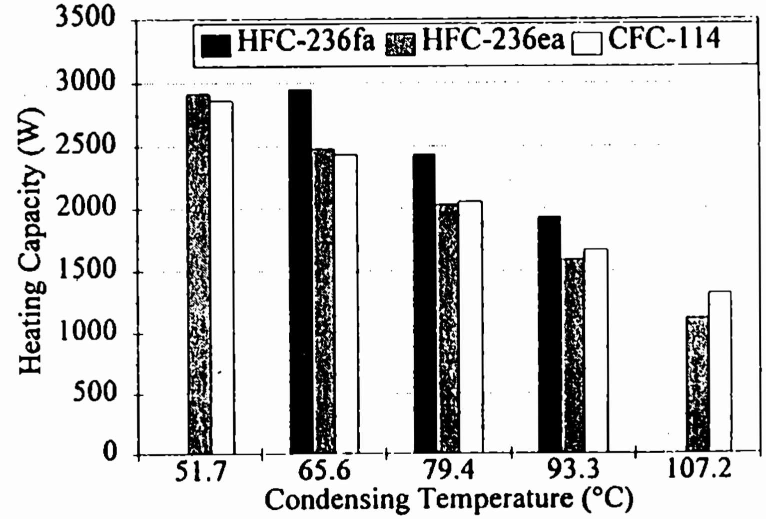

Figure 4: Heating Capacity at 32.2°C Evaporating.

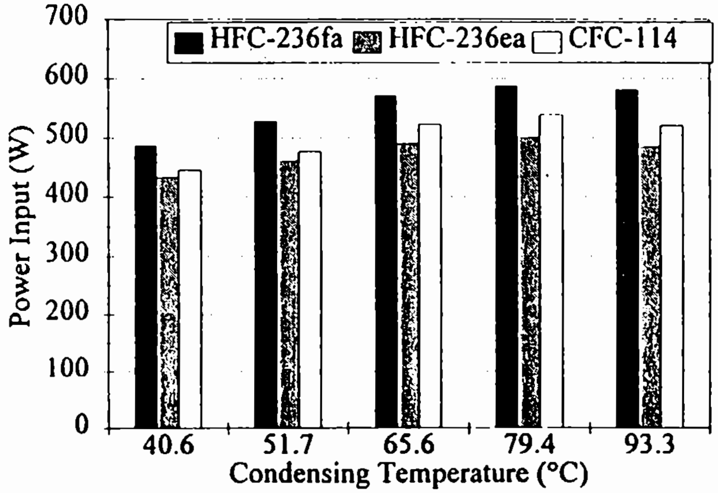

Figure 5: Compressor Electrical Power Input at 10°C Evaporating.

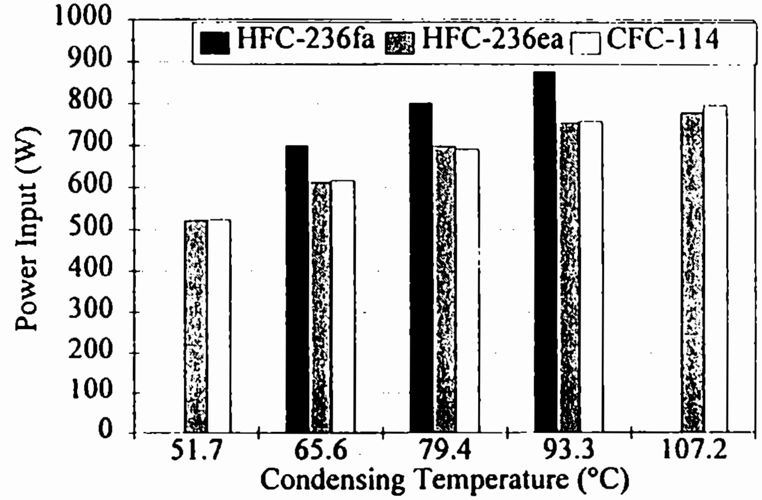

Figure 6: Compressor Electrical Power Input at 32.2°C Evaporating.

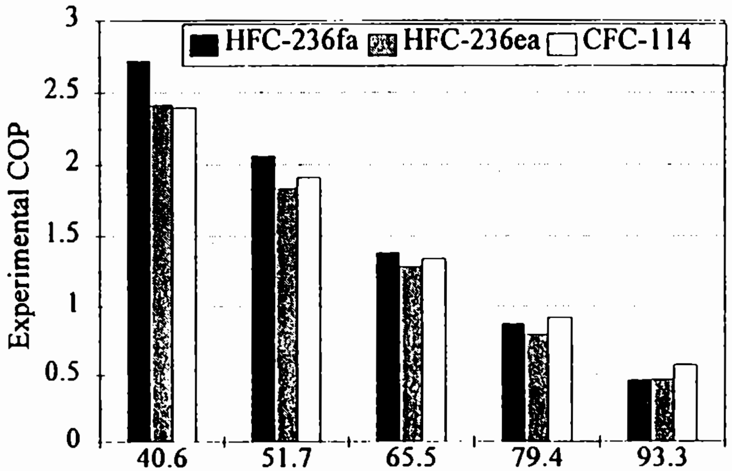

Figure 7: Cooling Coefficient of Performance at 10°C Evaporating

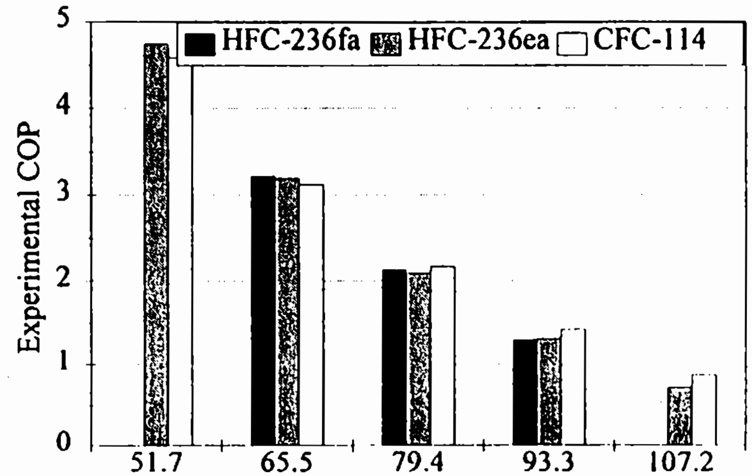

Figure 8: Cooling Coefficient of Performance at 32.2°C Evaporating.

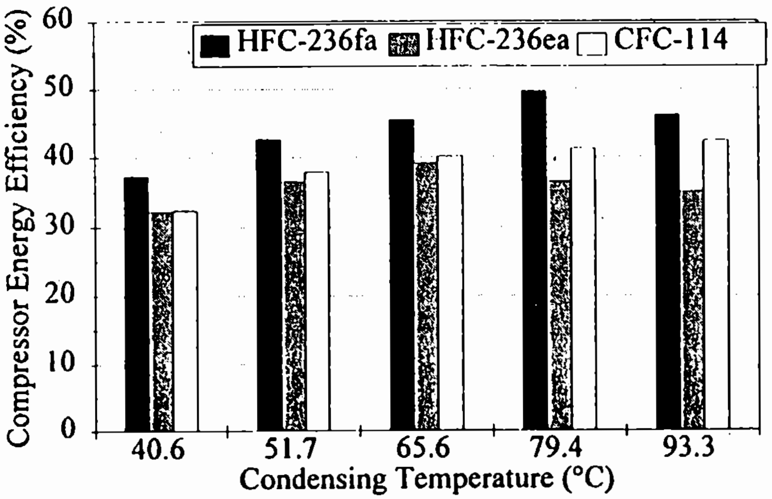

Figure 9: Compressor Isentropic Energy Efficiency at 10°C Evaporating.

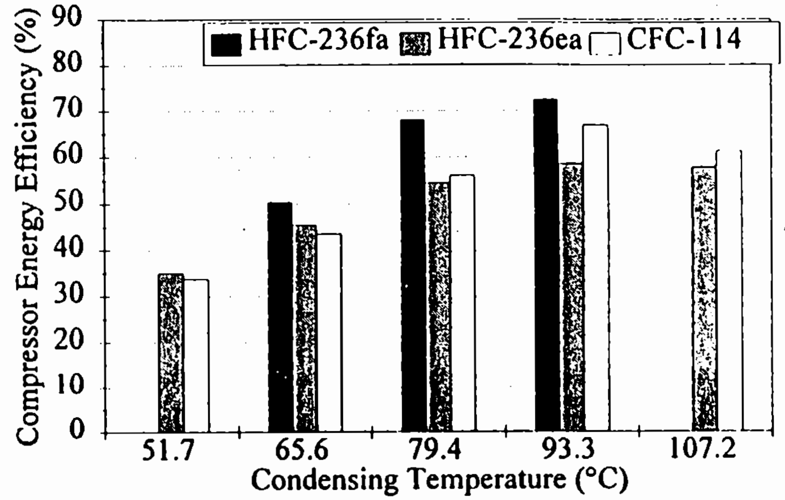

Figure 10: Compressor Isentropic Energy Efficiency at 32.2°C Evaporating.

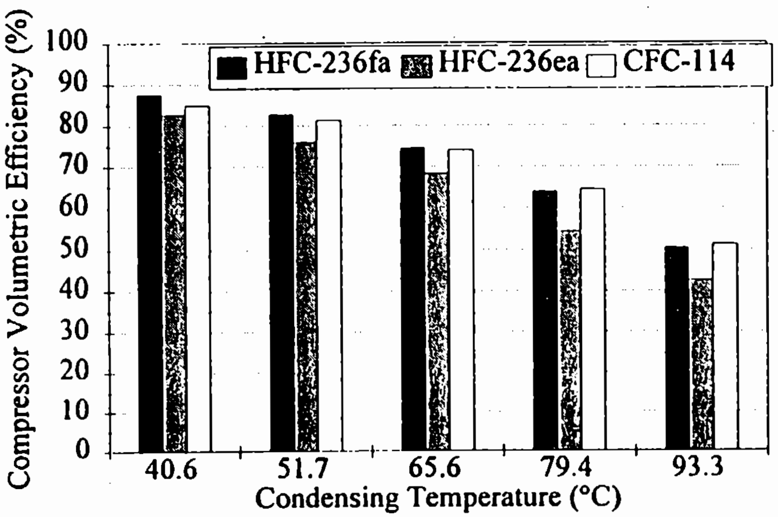

Figure 11: Compressor Volumetric Energy Efficiency at 10°C Evaporating.

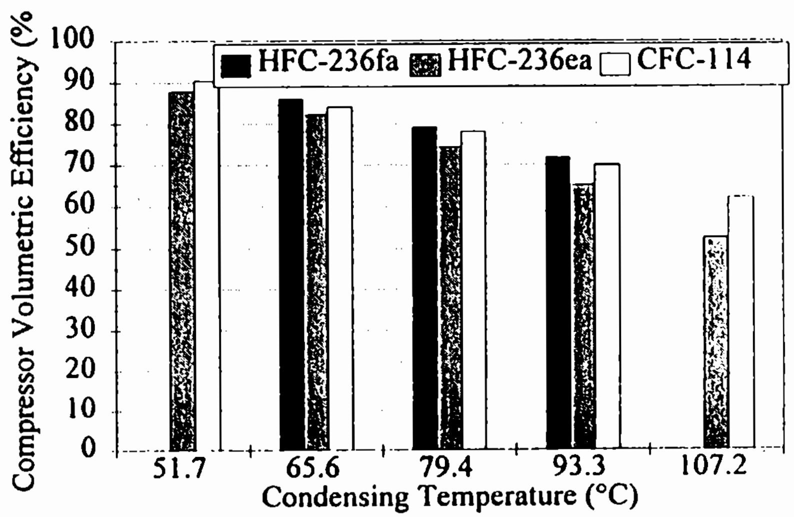

Figure 12: Compressor Volumetric Energy Efficiency at 32.2°C Evaporating.