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This paper investigates the use of several zero-ozone depleting potential (zero-ODP) HFC refrigerants, including, HFC-134a, HFC-227ca, HFC-227ea, HFC-236ea, HFC-236cb, HFC-236fa, HFC-245cb, and HFC-254cb, for centrifugal chiller applications. We took into account the thermodynamic properties of the refrigerant and aerodynamic properties of the impeller compression process to this evaluation. For a given operating temperature lift, there are significant differences in the pressure ratio required by each refrigerant and this variation in pressure ratio directly affects compressor size, efficiency, and performance. A comparison of the HFC refrigerant candidates with CFC-114 shows that HFC-236ea, HFC-227ca, and HFC-227ea are viable alternatives for centrifugal water chillers. HFC-227ea results in a significantly lower enthalpy rise requirement, potentially allowing single-stage compression, however, wet compression could be a problem. Single-stage compression gives an overall performance advantage over CFC-114 and when considering thermodynamics and aerodynamics, as is necessary in centrifugal applications, we find that HFC-227ca and HFC-227ea have additional advantages over HFC-236ea and CFC-114.

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ABSTRACT

This paper investigates the use of several zero-ozone depleting potential (zero-ODP) HFC refrigerants, including HFC-134a, HFC-227ca, HFC-227ea, HFC-236ea, HFC-236cb, HFC-236fa, HFC-245cb, and HFC-254cb, for centrifugal chiller applications. We took into account the thermodynamic properties of the refrigerant and aerodynamic characteristics of the impeller compression process in this evaluation.. For a given operating temperature lift, there are significant differences in the pressure ratio required by each refrigerant and this variation in pressure ratio directly affects compressor size, efficiency, and performance. A comparison of the HFC refrigerant candidates with CFC-114 shows that HFC-236ea, HFC-227ca and HFC-227ea are viable alternatives for centrifugal water chillers. HFC-236ea has properties closest to CFC-114, and will result in comparible performance, but will require a slightly larger impeller and a purge system. Using HFC-227ca or HFC-227ea results in a significantly lower enthalpy rise requirement, potentially allowing single-stage compression, however, wet compression could be a problem. Single-stage compression gives an overall performance advantage over CFC-114 (operating with 3-5 °C of liquid subcooling), and when considering thermodynamics and aerodynamics, as is necessary in centrifugal applications, we find that HFC-227ca and HFC-227ea have additional advantages over HFC-236ea and CFC-114.

INTRODUCTION

Since the manufacture of CFCs has been banned and HCFCs are due to be phased out soon, there has been considerable focus in the past decade on finding zero-ODP replacement refrigerants. There are approximately 110,000 centrifugal water chillers in the world and 70,000 in the U.S. using CFC-11 or CFC-114 [Sand and Fisher, 1994]. HCFC-123 has been used for centrifugal chiller applications, and as an HCFC, is less damaging to the ozone layer than CFC refrigerants. However, HCFC-123 still has an ODP and will face future phase-out. Hughes [1992] recommended HFC-245ca as a long term replacement for CFC-11 and its current alternative, HFC-123. Sand and Fisher [1994] modeled 16 chlorine-free alternatives and found several CFC-11 alternatives, including HFC-143 and HFC-152, with performance comparable to CFC-11. Since these alternatives are flammable, blends were also studied and formulated. Fewer alternatives were identified for CFC-114. Recently, Smith et al.[1993], Sand and Fisher [1994] suggested HFC-236ea as an alternative for CFC-114. Kazachki and Gage [1993], and Kazachki and Hendriks [1993], performed a thermodynamic analysis and obtained similar conclusions. Chen and Kruse [1994] tested a reciprocating heat pump using HFC-227 and found that HFC-227 has a better or equivalent heating COP.

To avoid significant changes to existing hardware, the focus of finding replacement refrigerants has typically been on volumetric flow rates and operating pressures. In these replacement studies, the compressor efficiency was normally assumed constant in the evaluation of the system's COP. However, the efficiency of centrifugal compressors are a function of the chosen refrigerant, with particular sensitivity to the required **enthalpy rise** and the **specific speed.** It is therefore crucial to allow for this effect in any serious performance assessment of replacement refrigerants. This is our chosen approach for this paper - *a full assessment of replacement refrigerants for the current CFC compressors.* This paper consists of a series of comparisons of HFC replacements with specific emphasis on pressure ratio, compression efficiency, specific speed, cooling capacity, enthalpy rise, and overall system COP.

A COMPARISON OF ENTHALPY RISE

The **enthalpy rise** across the compressor and the compressor's **specific speed** are two critical fluid-dependent aerodynamic variables in the design of a centrifugal compressor. The enthalpy rise is related to the required operating pressure difference and represents the specific energy required for a specific operating condition and refrigerant. Although the enthalpy rise is an energy variable, it is related to the pressure rise across the compressor and is therefore directly related to the impeller's tip speed (which is the product of the impeller radius and its angular speed). The required pressure rise affects many aspects of the compressor design. A small compressor pressure ratio requires a relatively low compressor rotating speed and/or a smaller impeller diameter. Refrigerants such as CFC-11 and HFC-134a typically require relatively high pressure ratios, thereby forcing two stages of compression to reduce the per stage pressure ratio. Reducing the required pressure ratio allows a reduced impeller diameter, a reduced speed, or even a single compression stage.

A unique feature of HFC-227ca and HFC-227ea is the low compressor enthalpy rise relative to other refrigerants. Table 1 summarizes several critical properties of HFC-227ea, HFC-227ca, and 114. To analyze the characteristics and performances of HFC-227ea and other refrigerants, we developed a thermal cycle model which generates thermodynamic and aerodynamic properties for the cycle. In this model, we neglected the pressure drop between the evaporator and the compressor inlet, and the pressure difference between the compressor discharge and the condenser. Because our primary interest was in relatively small centrifugal applications, a conservative compressor efficiency of 75% was assumed (actually the compressor efficiency would increase as the enthalpy rise decreases). Unless specified otherwise, a common compressor rotating speed and loading coefficient was assumed in the dimensionless comparison of refrigerants. We computed thermodynamic properties not available in the literature using the generalized three-parameter corresponding states correlation of Lee and Kesler [1975].

Tables 2a and 2b lists the isentropic enthalpy rises, $h_{ad,s}$, along with other key parameters, for eight HFC refrigerants as well as HCFC-123 and CFC's 11 and 114. This data was evaluated for an evaporator temperature of 4.4°C (40°F) and a condenser temperature of 37.8°C (100°F). Clearly, , there are significant differences in the required enthalpy rise among refrigerants, but that the required pressure ratio does not vary significantly (Table 2). However, the refrigerants with the lowest enthalpy rise, HFC-227ea and HFC-227ca, also exhibit the lowest pressure ratio. For comparison, we normalized the enthalpy rise by the enthalpy rise of CFC-114. Clearly HFC-227ea has the lowest required enthalpy rise. In fact, the enthalpy rise required for HFC-227ea is 16% lower than that of CFC-114. The relatively low enthalpy rise stems from a reduction in the displacement work during an adiabatic compression process. A reduction in displacement work assists the compression process and reduces the total energy required for a given pressure ratio. For all operating conditions of interest, the required enthalpy rise of HFC-227ea is the lowest among the refrigerants discussed in this paper.

Figure 1 shows the variation of the enthalpy rise (of refrigerants CFC-114, HFC-227ea, and HFC-236ea) as a function of condensing temperature for an evaporator temperature of 4.44°C (40°F). Similar curves can be drawn for other evaporator temperatures, and in conclusion, HFC-227ea consistently maintains an enthalpy rise which is lowest among CFC and HFC refrigerants.

BENEFITS OF LOW ENTHALPY RISE

Many advantages result from a low enthalpy rise. These advantages include a reduced impeller tip speed, a smaller impeller diameter, and a potential for higher compressor efficiency.

Impeller Diameter

For a centrifugal compressor with a fixed loading coefficient, the impeller tip speed is approximately proportional to the square root of the required enthalpy rise. In addition, for a constant rotating speed, the impeller diameter is proportional to its tip speed. Using these two proportionality relations, a dimensionless impeller diameter d_2^+ (non-dimensionalized by the impeller diameter of CFC-114), can be calculated from following equation:

$$d_2^+ = \frac{d_2}{d_{2,R-114}} = \sqrt{\frac{H_{ad}}{H_{ad,R-114}}} \quad [1]$$

Equation 1 has been used to calculate the non-dimensional impeller diameter for several candidate refrigerants as shown in Table 3. Note that the impeller diameter of HFC-227ea is 8% smaller than that of CFC-114 while the impeller diameters of HFC-236ea and HFC-134a are 11% and 20% larger respectively. More than a 20% difference in diameter exists between HFC-227ea and HFC-236ea. Note also that the normalized axial F_a^+ and centrifugal loads F_c^+ vary from those of CFC-114, necessitating different bearing designs.

Compressor Efficiency

There are also direct benefits of low enthalpy rise on the compressor efficiency. A potential for higher compressor efficiency for HFC-227ea is derived from the following flow properties:

- Smaller flow losses due to lower operational speeds,
- Reduced degree of diffusion, and
- Lower frictional losses.

Lower vapor velocities in the impeller and diffuser reduce the flow losses. Most losses in centrifugal compressors are turbulent flow losses and flow separation losses which vary exponentially with the flow velocity (although the exact values are hard to predict). Low flow velocity can effectively avoid high turbulent and flow separation losses occurring in a compression process. The maximum absolute vapor velocity in a HFC-227ea compressor is less than 110m/s, which is smaller than the flow velocities in current centrifugal compressors. The diffusion process is also reduced because the flow velocity is reduced. Flow velocities for a typical HFC-227ea diffuser enter at 105 m/s and reduce to 20-30m/s.

In short, the frictional losses for HFC-227ea are reduced over CFC-114 or HFC-236ea. This is due to the impeller's friction loss being proportional to the cubic power of the impeller diameter, which is smaller for HFC-227ea.

SPECIFIC SPEED AND MINIMUM COOLING CAPACITY

There is an inherent relationship between flow rate (compressor capacity), impeller rotating speed, refrigerant enthalpy rise, and compressor efficiency. Gui et al. [1994, 1995] and Biancardi et al. [1994] have suggested utilizing higher rotating speeds (which increases tip speed without increasing impeller diameter), to achieve high performance low-tonnage centrifugal units. The criterion to determine the system's minimum cooling capacity is to keep the specific speed at an appropriate value. A dimensionless variable $Q^+_{e,min}$ can be defined (Equation 2) to compare the minimal cooling capacity $Q_{e,min}$ to the minimum cooling capacity of a CFC-114 operating at the same speed.

$$Q^+_{e,\min} = \frac{Q_{e,\min}}{Q_{e,R-114,\min}} = \frac{v_{1,R-114}}{v_1} \frac{\Delta h_e}{\Delta h_{e,R-114}} \left(\frac{h_{ad}}{h_{ad,R-114}}\right)^{3/2} \quad [2]$$

where the $Q_{e,min}$ is the minimum cooling capacity corresponding to the smallest specific speed possible for a single-stage unit at a given rotating speed. The normalized minimal cooling capacity has been calculated for the several HFC refrigerants as well as CFC-114, and the results are presented in Table 3. The disadvantage of the alternative HFC refrigerants is that all of them have significantly higher minimal cooling capacities compared to CFC-114. That is the minimal cooling capacity $Q_{e,min}$ must increase by 47% for HFC-227ea and 35% for HFC-236ea. For example, suppose the minimum cooling capacity ($Q_{e,min}$) for CFC-114 at a given rotating speed is 125 tons, then the $Q_{e,min}$ is 184 tons for HFC-227ea and 169 tons for HFC-236ea. The minimal cooling capacity is about 5 times higher for HFC-134a compared to CFC-114, clearly HFC-134a is not suitable for low-tonnage single-stage centrifugal systems.

The only way to build a efficient low-tonnage centrifugal compressor using HFC-227ea, HFC-227ca, or HFC-236ea is to increase the compressor rotating speed. Figure 2 displays the reduction in the minimum cooling capacity which results from increased compressor speed. By using magnetic bearings instead of conventional bearings, it is practical to raise the compressor speed significantly and thereby reduce the minimum cooling capacity. The rotating speeds as a function of minimum cooling capacity can be calculated from equation 3, and a specific speed of 0.8 is assumed.

$$n = \frac{0.8 \times 30}{\pi} h_{ad,s}^{3/4} \sqrt{\frac{Q_e}{\Delta h_e} v_i} \quad [3]$$

The specific speed, which characterizes the compressor geometry and implies the compressor efficiency, is a dimensionless variable determined by the compressor volumetric flow rate, enthalpy rise, and compressor rotating speed. There is a correlation between the compressor efficiency and the compressor specific speed [Balje, 1981]: *Compressor efficiencies are high when the specific speed falls into the range from 0.8 to 1.2.*

COEFFICIENT OF PERFORMANCE

As shown in Figure 3, HFC-236ea has a COP which is similar to that of CFC-114 when the compressor efficiency is assumed constant. However, as mentioned earlier, the actual COP of HFC-236ea would be slightly lower because of the lower compressor efficiency for HFC-236ea. Unfortunately as shown in Figure 3, HFC-227ea exhibits a lower COP compared to CFC-114, if the compressor efficiency is assumed to be the same. This is because the lower heat of vaporization of HFC-227ea increases the refrigerant circulation, resulting in larger flow losses. A high liquid heat capacity further deteriorates HFC-227ea's cooling performance. However this conclusion uses the erroneous constant compressor efficiency assumption. We expect higher compressor efficiencies with HFC-227ea.

There are two approaches that can be used to improve the COP of HFC-227ea, namely:

- Apply liquid subcooling to alleviate the thermal flow losses, and/or
- Design a high efficiency compressor.

Subcooling is effective in reducing thermal flow loss, especially for high heat capacity fluids. With a 5°C subcooling, the COP can be improved by 7.70% for HFC-227ea while the improvement is only about 5% for CFC-114 and HFC-236ea and only 2.91% for CFC-11. Due to this improvement, the COP difference between HFC-227ea and HFC-236ea or CFC-114 can be reduced from 7.3% to 5.0%.

In terms of improved compressor efficiency, if a HFC-227ea compressor can produce a efficiency which is 4% higher than that of HFC-236ea, then the COP of HFC-227ea would exceed that of HFC-236ea. Considering the 48% difference in the enthalpy rise between HFC-227ea and HFC-236ea, it is reasonable to expect at least a 4% improvement in compressor efficiency. A COP curve of HFC-227ea with a compressor efficiency of 0.80 has been shown in Figure 5 along with the COP curves for CFC-114 and HFC-235 (at a compressor efficiency of 75%). The COP of the HFC-227ea system begins to exceed the COP of the CFC-114 system at a subcooling of about 3 °C. At a subcooling of 5 °C, the COP of HFC-227ea is 5.55, compared to 5.45 for CFC-114 (at the same subcooling but a compressor efficiency of 0.75).

OTHER ADVANTAGES OF HFC-227ea

There are other advantages of HFC-227ea as a zero-ODP replacement for centrifugal chiller applications. The centrifugal compressor will have a greater stable operating range, because most of the pressure rise occurs in the impeller where the flow is more stable rather than in the diffuser.

The higher operating pressure of a HFC-227ea system will reduce the significance of the flow resistance (pressure drop) in the system when compared with HFC-236ea. For the same frictional pressure drop, the performance drops more for HFC-236ea than for HFC-227ea.

DISADVANTAGES OF HFC-227ea

HFC-227ea has a low acoustic speed which limits the maximum energy imparted to the refrigerant. If this maximum energy is lower than the required enthalpy rise the compressor can not reach the pressure rise required without additional stages of compression. The acoustic speed of HFC-227ea is about 110 m/s, the lowest among all refrigerants considered here. However the acoustic speed is sufficient for single-stage water chiller applications as long as the compressor's efficiency is above 0.80.

There is a slight, but acceptable, wet compression for HFC-227ea operating at typical chiller conditions. The vapor leaving the impeller would have a vapor quality of 0.96~0.97. Since liquid drops usually will not form until the vapor quality drops below 0.95 [Balje, 1981] this slightly wet compression should not be a problem (the refrigerant will still be homogenous and the centrifugal compression stable). In addition, because the HFC-227ea centrifugal compressor impeller operates at low discharge velocities droplet impact on the diffuser surface, if droplets were to form, would be minimal. The wet compression does avoid vapor exit superheat reductions in compression efficiency and keeps the compression at a lower operating temperature.

CONCLUSIONS

HFC-227ca, HFC-227ea, and HFC-236ea can be used for single-stage centrifugal systems with COP's comparable to CFC-114. In regards to a thermodynamic parameters, HFC-236ea has many properties similar to CFC-114, however, the operating pressure ratio is at least 20% greater than CFC-114. While this is not significant in a positive displacement compressor application, it is very significant in a centrifugal compressor application (where pressure ratio is created from tip speed). This increased pressure ratio translates into a 10% increase in the impeller diameter for HFC-236ea when compared to CFC-114 (assuming the speed is unchanged). The minimal cooling capacities for HFC-227ea and HFC-236ea are approximately 47% and 35% larger that CFC-114. HFC-236ea also requires a non-condensable purge system because of the subambient operation.

From this detailed thermodynamic and aerodynamic analysis, we conclude that for single stage centrifugal chiller applications, HFC-227ea is the best suited refrigerant. HFC-227ea is commercially available as a fire suppressant. Clearly compact high-efficiency compressors can be designed and manufactured for HFC-227ea.

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Nomenclature

a	acoustic speed, m/s
c_p	constant heat capacity, kJ/kg,K
COP	coefficient of performance
d_2	impeller exit diameter,m
F_a	compressor axial load, N
F_c	compressor centrifugal load, N
$h_{ad,s}$	isentropic enthalpy rise, kJ/kg
Δh_e	refrigerant unit cooling, kJ/kg
$\dot{m}$	system mass flow rate, kg/s
n	compressor rotating speed, RPM
Q_e	system cooling capacity, kW, 1 Ton=3.517 kW
v_i	specific volume at compressor inlet, m^3/kg
η_c	compressor efficiency

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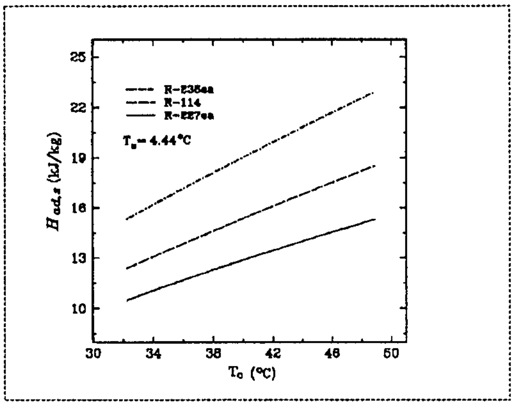

FIGURE 1. ENTHALPY RISE AS A FUNCTION OF CONDENSER TEMPERATURE FOR SEVERAL REFRIGERANTS

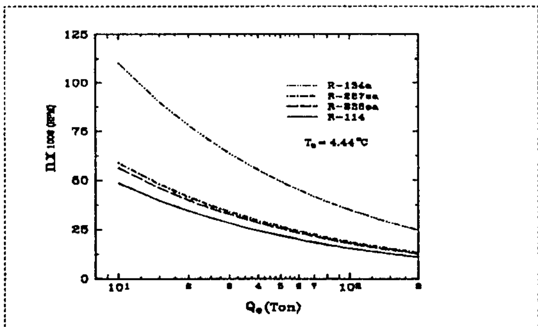

FIGURE 2. REQUIRED COMPRESSOR SPEED AS A FUNCTION OF MINIMUM COOLING CAPACITY FOR A SPECIFIC SPEED OF 0.8

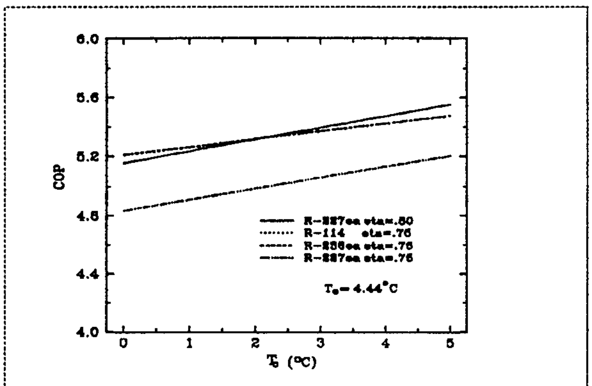

FIGURE 3. THE EFFECT OF SUBCOOLING ON COP

TABLE 1. THE PHYSICAL PROPERTIES OF HFC-227ea

	HFC-227ca	HFC-227ea	CFC-114
Molecular weight kg/kmol	170	170	171
Boiling pt, °C	-17.3	-16.4	3.8
Tc (K)	379	376	419
Pc (atm)	28.3	29.0	32.3
Heat capacity of liquid at 37.8 °C (kJ/kg-K)	1.26	1.27	1.06

TABLE 2A CRITICAL COMPRESSOR PARAMETERS

Refrigerant	p_e (kPa)	p_c/p_e	v_i×100 (m^3/kg)	a (m/s)
CFC-11	47.22	3.4	34.83	135
CFC-114	103.6	3.06	12.43	115
HFC-123	40.93	3.59	36.01	126
HFC-134a	344.2	2.79	5.958	147
HFC-227ca	223.7	2.78	5.531	109
HFC-227ea	243.4	2.78	5.064	109
HFC-236cb	127.0	3.07	11.28	120
HFC-236ea	92.51	3.38	15.74	121
HFC-236fa	125.6	3.09	11.42	120
HFC-245cb	243.5	2.79	6.645	124
HFC-254cb	123.6	3.01	15.31	139

TABLE 2B CRITICAL COMPRESSOR PARAMETERS *

Refrigerant	Δh_e (kJ/kg)	$h_{ad,e}$ (kJ/kg)	$h^+_{ad,e}$	γ (kJ/kg)
CFC-11	158.4	20.73	1.43	188.6
CFC-114	100.8	14.51	1.00	134.3
HFC-123	142.0	19.59	1.35	176.6
HFC-134a	146.2	20.74	1.43	195.0
HFC-227ca	79.28	12.25	0.84	117.2
HFC-227ea	78.56	12.19	0.84	116.8
HFC-236cb	111.4	16.15	1.11	149.5
HFC-236ea	124.9	17.98	1.24	165.5
HFC-236fa	112.5	16.31	1.12	150.9
HFC-245cb	105.0	15.83	1.09	149.4
HFC-254cb	148.3	21.15	1.46	194.5

TABLE 3 DIMENSIONLESS DIAMETER OF CANDIDATE REFRIGERANTS*

Refrigerant	d_2^+	F_o^+	F_e^+	$Q^+_{e,min}$
HFC-227ca	0.92	0.78	1.54	1.37
HFC-227ea	0.92	0.77	1.67	1.47
CFC-114	1.00	1.00	1.00	1.00
HFC-236cb	1.05	1.17	1.38	1.43
HFC-236fa	1.06	1.19	1.40	1.45
HFC-236ea	1.11	1.38	1.30	1.35
HFC-134a	1.20	1.71	4.25	5.17

* Data applies to situations where temperature lift from 4.44 °C to 37.78 °C (40 °F to 100 °F)

TABLE 4 THE COP'S OF DIFFERENT REFRIGERANTS FOR CENTRIFUGAL CHILLERS*

Refrigerant	COP ΔT_{sub} =0, η=.75	COP ΔT_{sub} =5°C, η=.75	Improvement (%)
CFC-11	5.730	5.897	2.91
CFC-114	5.208	5.476	5.15
HFC-123	5.473	5.642	3.09
HFC-134a	5.298	5.576	5.25
HFC-227ca	4.856	5.223	7.56
HFC-227ea	4.832	5.204	7.70
HFC-236 cb	5.171	5.447	5.34
HFC-236ea	5.212	5.475	5.05
HFC-236eafa	5.176	5.451	5.31
HFC-245cb	4.974	5.308	6.71
HFC-254cb	5.260	5.517	4.88

* For a cycle of temperature lift from 4.44 °C to 37.8 °C (40 °F to 100 °C)
The compressor efficiency is assumed to be 0.75 for all refrigerants.