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Bruce P. Biederman
United Technologies Research Center

Jarso Mulugeta
United Technologies Research Center

Lili Zhang
United Technologies Research Center

Joost J. Brasz
Carrier Corporation

CYCLE ANALYSIS AND TURBO COMPRESSOR SIZING WITH KETONE C6F AS WORKING FLUID FOR WATER-COOLED CHILLER APPLICATIONS

Bruce P. Biederman[1], Jarso Mulugeta[1], Lili Zhang[1], Yu Chen[1], Joost J. Brasz[2]

[1]United Technologies Research Center
East Hartford, CT 06108
biederbp@utrc.utc.com
mulugej@utrc.utc.com
zhangl1@utrc.utc.com
cheny@utrc.utc.com
[2]Carrier Corporation
Syracuse, NY 13221
joost.j.brasz@carrier.utc.com

ABSTRACT

Ketone C6F, which has the chemical formula: $CF_3CF_2C(O)CF(CF_3)_2$, has been introduced recently as a zero-ODP, zero-GWP fire extinguishing fluid. Its non-flammability and non-toxicity combined with its excellent environmental properties make it an attractive fluid for HVAC applications. Its low density limits its use to centrifugal water-cooled chiller applications. The relatively high critical temperature of the fluid promises good thermal cycle efficiency. However, the slope of the saturation dome on the temperature-entropy diagram forces the use of a vapor suction / condensed liquid heat exchanger to prevent wet compression and reduce the throttling losses. The low speed of sound of the fluid allows direct drive single stage compressor operation at relatively small tonnages. The sub-atmospheric evaporator (0.162 bar) and condenser (0.604 bar) pressures necessitate the use of a purge but eliminate the need for pressure vessel code certification.

1. INTRODUCTION

Flammable and/or toxic natural refrigerants are unacceptable as working fluid for large air conditioning and refrigeration systems due to the risk posed by the substantial charge of such systems. Efforts to reduce the refrigeration charge by applying technologies such as falling film evaporation reduce the charge level but still leave charge levels too large to allow from a safety point of view, use of these systems in the built environment. The only two natural refrigerants left for large HVAC systems are water and CO_2.

1.1 Water as refrigerant in chillers

From a safety and efficiency point of view water as refrigerant is a very attractive choice. However, the low density of water at refrigeration temperature levels results in large volumetric flow rates requires extremely large machinery with low power density. The lowest cost compressor solution will be a turbomachine, of either the centrifugal or the axial type. The high sonic velocity of water puts special demands on the strength/weight ratio of the materials to be used in the turbo-machinery. It is therefore questionable whether a cost effective chillers using water as refrigerant are possible.

1.2 CO_2 as refrigerant in chillers

The very high pressure of CO_2 as refrigerant has just the opposite drawback of that of water. Its very high density forms a safety risk. A system leak could cause asphyxiation. Also, the trans-critical cycle conditions of CO_2 result in poor system efficiencies due to the large throttling losses, which can only be overcome by the use of power recovery during the expansion process.

1.3 Are there man-made fluids with natural refrigerant GWP values?

The main driver for the use of natural refrigerants is not the fact that they happen to occur in nature but that these fluids have a very low direct-effect global warming potential. All commonly used fluorocarbon refrigerants have global warming potentials orders of magnitude larger than those of the natural refrigerants. Ketone C6F - chemical formula $CF_3CF_2C(O)CF(CF_3)_2$ - has been introduced recently as a zero-ODP, GWP=1 fire extinguishing fluid [1,2]. It has also been considered as a refrigerant for centrifugal chillers but was dismissed based on its poor cycle efficiency. This low cycle efficiency is somewhat surprising because the fluid has a fairly high critical temperature (168.66 ^{0}C in between the critical temperatures of HFC245fa 154.05 ^{0}C and HCFC123 183.68 ^{0}C), which typically indicates good thermodynamic cycle efficiency. Figure 1 shows the COP of the for a number of CFC, HCFC and HFC refrigerants used on centrifugal chillers as a function of the critical temperature of the refrigerant. The general trend for fluorocarbon-based refrigerants is an increase in COP with critical temperature. The critical temperature of ketone C6F suggested a COP in the neighborhood of that of HFC245fa and HCFC123. However, the cycle calculations resulted in a much lower COP.

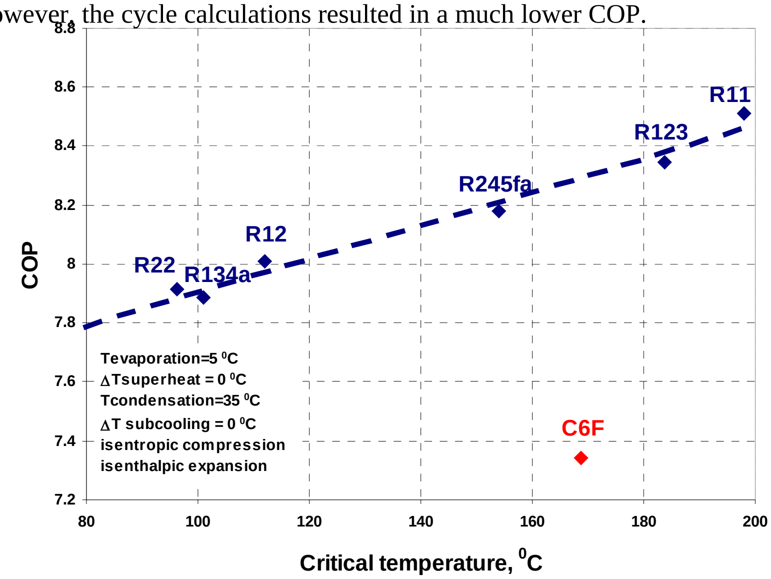


Figure 1. Coefficient of performance (COP) of various fluorocarbon refrigerants and ketone C6F as a function of critical temperature

The reason for this deviation can be explained by comparing the TS diagrams of HFC245fa, CF6 and HCFC123. The saturation lines for C6F are leaned over heavily. The small positive slope of the liquid saturation line results in large throttling losses. Figure 2 shows the TS diagrams for HCFC123, HFC245fa

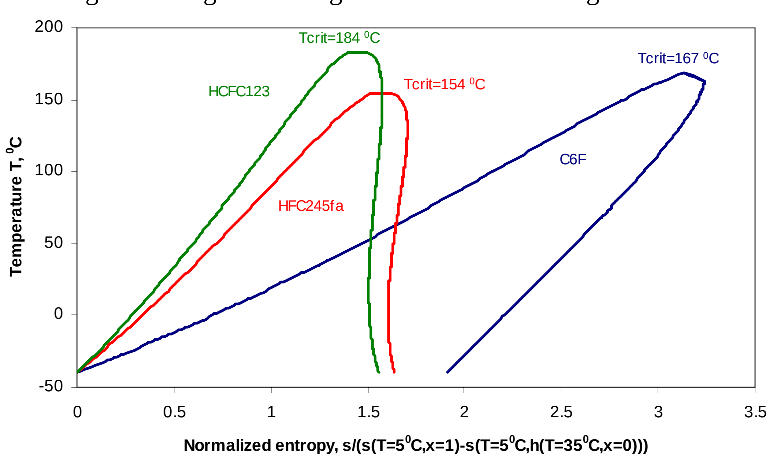


Figure 2. TS diagrams with two-phase flow domes of HCFC123, HFC245fa and C6F on a normalized entropy basis

and C6F. In order to graphically compare the throttling losses the horizontal entropy axis has to be normalized based on the refrigeration effect by dividing the actual entropy by the difference in entropy experienced by the working fluid in the evaporator:

$$s_{normalized} = \frac{s}{s_{evap,out} - s_{evap,in}} = \frac{s}{s(T = 5^0C, x = 1) - s(T = 5^0C, h(T = 35^0C, x = 0))}$$

The difference in the slopes of the saturation lines of the Ts diagrams explains the low COP of C6F. Figures 3a, b and c show side by side the TS diagrams of HFC245fa, C6F and HCFC123, while maintaining the saturation lines of the other two refrigerants. Since the throttle loss is equal to the triangle in the TS diagram, it can be seen why the COP of ketone C6F is so much lower than that of HFC123 or HFC245fa.

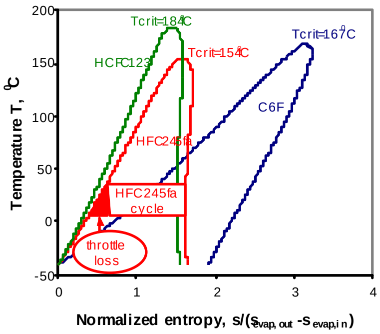


Figure 3a. TS diagram of HFC245fa.

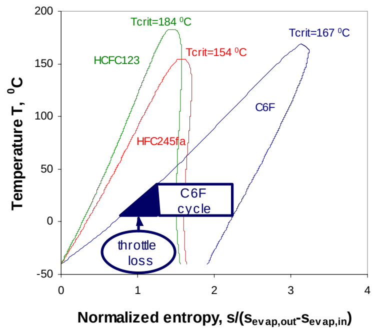


Figure 3b. TS diagram of C6F.

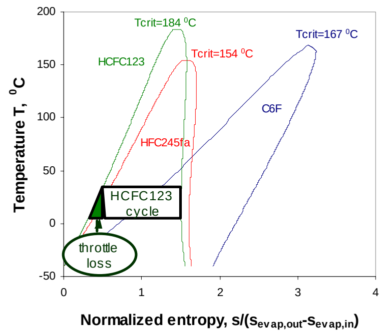


Figure 3c. TS diagram of HCFC123

Replacing the throttle valve in the cycle with a two-phase flow expansion turbine is one way of recovering the throttle losses. However, this method adds substantial complexity and cost to the system due to the additional rotating equipment and its controls.

3. IMPROVING THE C6F REFRIGERATION CYCLE WITH A SUCTION LINE HEAT EXCHANGER

When the TS diagram has so much lean to it that the vapor saturation line becomes positive, isentropic compression results in wet compressor exit conditions. Under those circumstances the refrigeration cycle does not suffer from a de-superheating loss and throttling is the only fundamental irreversibility of the cycle. Slightly wet isentropic compressor exit conditions do not result in compressor erosion since actual compressor inefficiencies will cause additional heating of the working fluid resulting in dry compressor exit conditions. Both HCFC123 and HFC245fa compress wet isentropically but exhibit superheated compressor discharge conditions at aerodynamic efficiencies in the high nineties, much higher than what is technically achievable. C6F compressor exit conditions are still wet at 35% compressor aerodynamic efficiency. Since actual compressor efficiencies will be higher, compressor erosion becomes a concern for C6F. A suction line heat exchanger has to be used to superheat the vapor before entering the compressor to prevent compressor erosion. When the heat required for superheating the vapor leaving the evaporator is extracted from the liquid leaving the condenser as shown in Figure 4b, a boost overall cycle efficiency is possible. Sub-cooling the liquid refrigerant leaving the condenser reduces the throttling loss of the cycle.

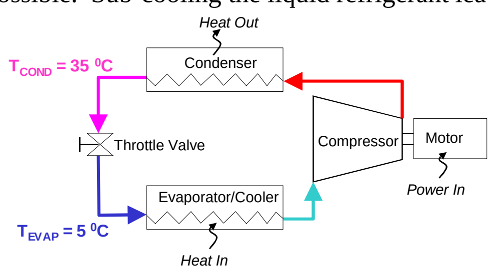


Figure 4a. Simple vapor compression cycle

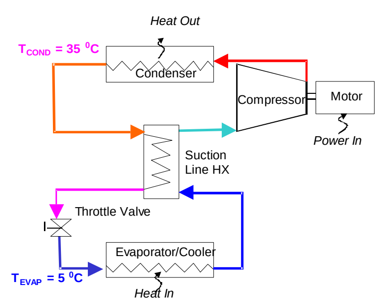


Figure 4b. Vapor compression cycle with suction line heat exchanger

The suction line heat exchanger introduces a thermodynamic irreversibility, since transferring heat from hot liquid to cold vapor requires a temperature difference ΔT as driving force, which is a thermodynamic loss. With increased suction superheat from the suction line heat exchanger this thermodynamic irreversibility is reduced. Eventually another thermodynamic cycle irreversibility, the condenser de-superheat loss will arise. A small suction-line heat exchanger with a large log-mean-temperature difference (LMTD), although good from a cost point of view, can actually worsen the system COP. Such is the case for C6F. Its reduction in throttling loss irreversibility is initially more than offset by the increased irreversibility in the suction line heat exchanger. For larger heat exchangers with smaller LMTD's the process reverses and an overall system COP improvement can be achieved. This trend is shown in Figure 5. A suction line heat exchanger with an effectiveness of 80% corresponding to an LMTD of 8 ^{0}C, results in a cycle efficiency slightly better than that of HFC245fa.

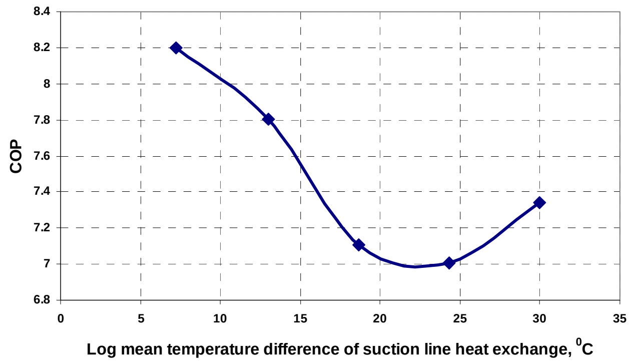


Figure 5. COP of the ideal C6F refrigeration cycle with a suction line heat exchanger as a function of heat exchanger log mean temperature difference.

The suction line heat exchanger adds cost to the system but since it also increases the system capacity per unit volume passing though the compressor, its net effect in terms of increased system cost for a given cooling capacity is moderate. The suction line heat exchanger brings the cycle efficiency of C6F at an attractive level. This combined with its negligible GWP makes C6F an intriguing refrigerant for future chillers. Obviously, the suction line heat exchanger has to be designed with minimal vapor side pressure drop since this pressure drop will increase the required compressor head and therefore its corresponding power requirement.

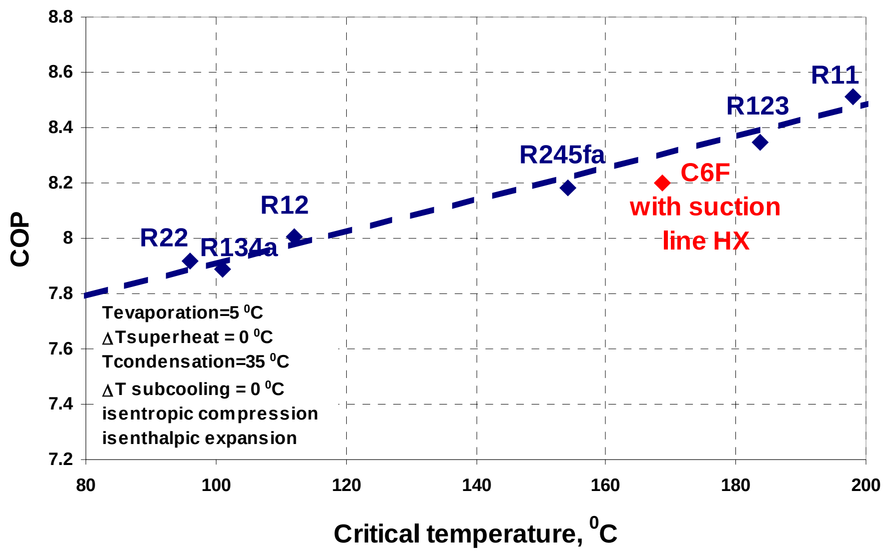


Figure 6. COP as a function of critical temperature showing that C6F matches the traditional correlation when equipped with a suction line heat exchanger

Figure 7 illustrates how for C6F the improvement in cycle efficiency is obtained. The large throttle loss of the simple refrigeration cycle is reduced dramatically and being offset by a much smaller suction line heat exchange loss and a miniscule de-superheat loss. As a result the cycle efficiency has improved from a COP of 7.34 to 8.20.

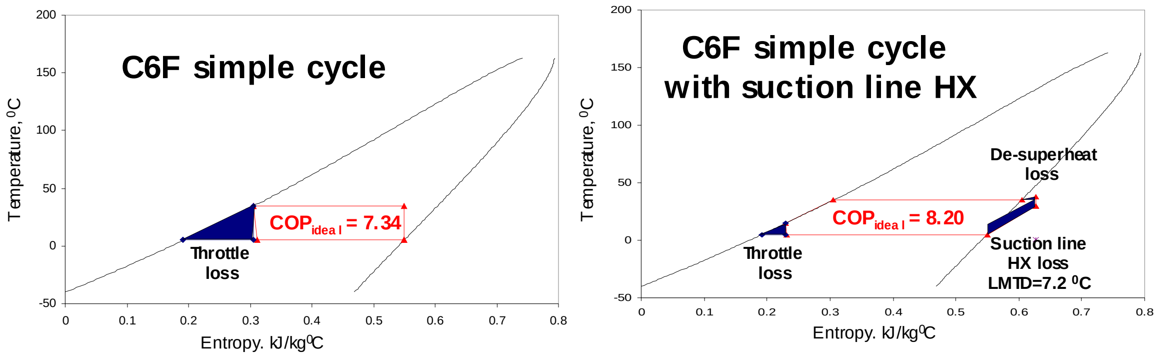


Figure 7. Temperature entropy diagrams with losses of C6F in the simple cycle and the modified cycle with a suction line heat exchanger

4. CENTRIFUGAL COMPRESSOR DESIGN FOR C6F

In this section we will size a single–stage centrifugal compressors for ketone C6F, the non-flammable natural refrigerants water and CO_2 as well as the commonly used refrigerants HCFC123 and HFC134a. For each of these fluids a compressor will be designed for the same specific speed and diameter. The concept of specific speed comes from similarity considerations and has been promoted by Balje, 1981. Compressor rotational speed, volumetric flow rate and head (or isentropic enthalpy rise) van be combined in a non-dimensional group. Isentropic enthalpy rise has the dimension of J/kg or m^2/s^2 while volumetric flow rate has the dimension of m^3/s. Rotational speed, which has a dimension of s^{-1}, can be obtained taking the ratio of the head to the power ¾ and the flow to the power ½:

$$\frac{[H]^{3/4}}{[Q]^{1/2}} = \frac{\left(\frac{m^2}{s^2}\right)^{3/4}}{\left(\frac{m^3}{s}\right)^{1/2}} = \frac{\frac{m^{3/2}}{s^{3/2}}}{\frac{m^{3/2}}{s^{1/2}}} = s^{-1} = [N] \tag{1}$$

Following this approach a non-dimensional specific speed n_s can be defined as

$$n_s = \frac{N\sqrt{Q}}{H^{.75}} \tag{2}$$

Similarly, a non-dimensional specific diameter d_s can be defined as:

$$d_s = \frac{DH^{.25}}{\sqrt{Q}} \tag{3}$$

The concept of specific speed and diameter is useful for preliminary sizing of turbo machinery equipment. Optimum efficiency of centrifugal compressors is obtained at a specific speed between 0.6 and 0.9 and a specific diameter between 3 and 5 as shown in Figure 8. We will size single stage centrifugal compressors for a specific speed on 0.76 and a specific diameter of 3.4, resulting in the following equations defining compressor speed N and diameter D:

$$N = 0.76\frac{H^{0.75}}{\sqrt{Q}} \tag{4a}$$

$$D = 3.4\frac{\sqrt{Q}}{H^{.25}} \tag{4b}$$

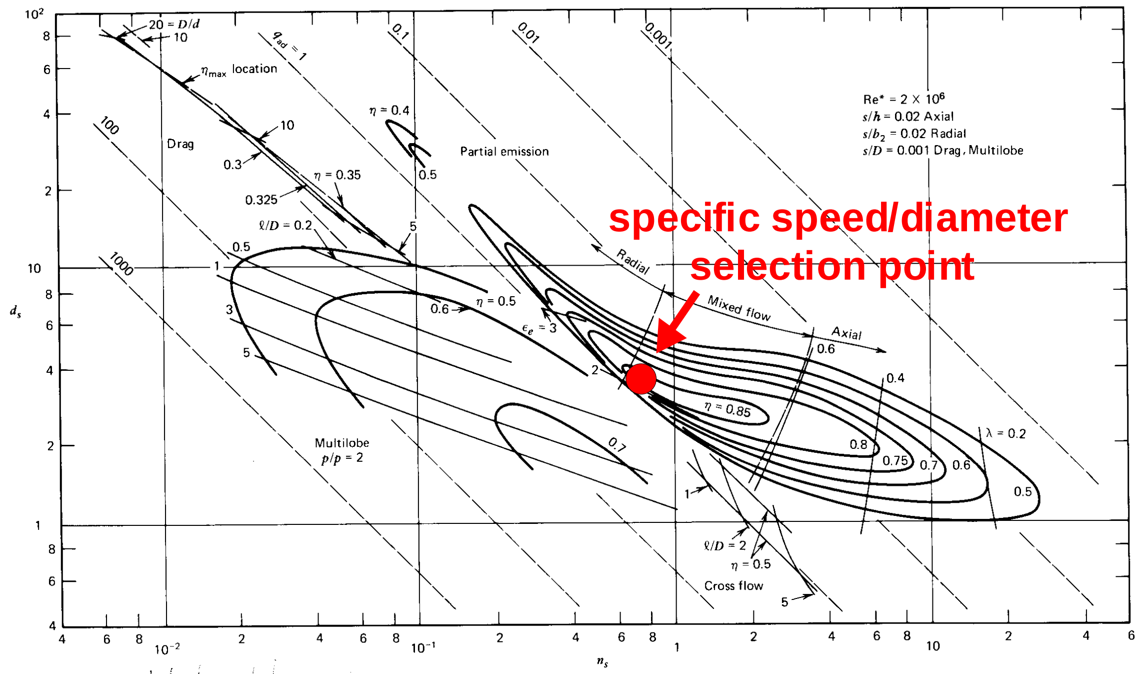


Figure 8. Selection of specific speed n_s and diameter d_s on the Balje diagram for centrifugal compressors, reproduced with permission from [3].

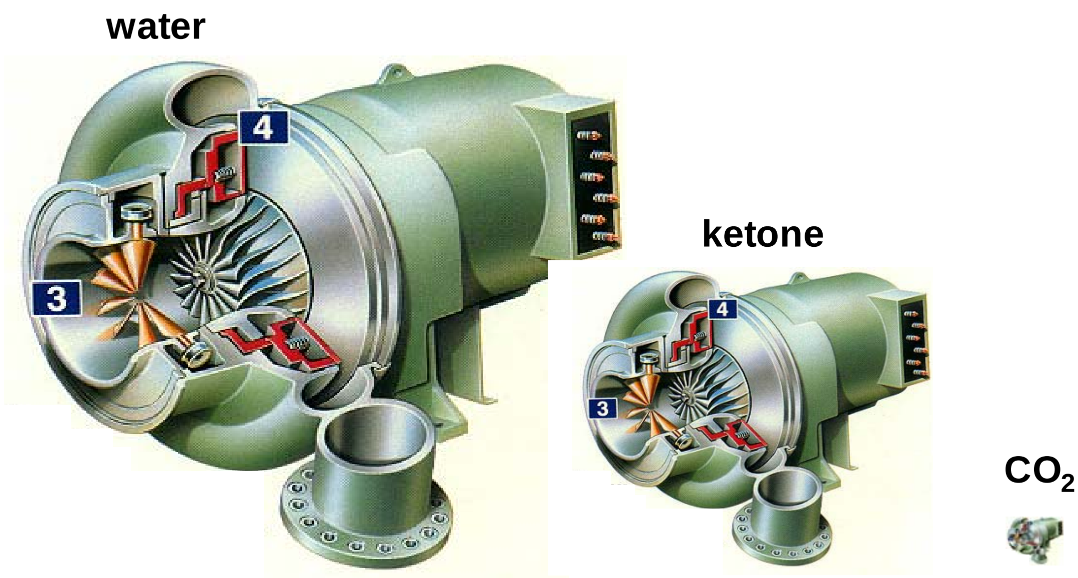
The compressor sizing will be done for a refrigeration duty of 500 tons (1760 kW) assuming identical evaporation and condensation temperatures of 5 and 35 ^{0}C, respectively. Cycle analysis will then give the values for head and volumetric flow rate. Using equations 4a and 4b will then give the speed and impeller diameter of the centrifugal compressor for each of these working fluids. The results of these calculations are summarized in Table 1.

	C6F	water	CO_2	HCFC123	HFC134a	
ODP	0	0	0	0.02	0	
GWP	1	1	1	120	1300	
Δh_{evap}	88.63	2363.47	139.11	149.03	152.48	kJ/kg
$\Delta h_{s,comp}$	10.81	303.30	31.11	17.86	19.33	kJ/kg
Head	1103	30949	3174	1822	1972	m
mdot	19.86	0.74	12.65	11.81	11.54	kg/s
ρ	2.24	0.01	114.62	2.76	17.13	kg/m^3
Vdot	8.85	109.48	0.11	4.28	0.67	m^3/s
N	2587	8966	51177	5423	14497	rpm
D	0.991	1.515	0.085	0.608	0.237	m
u_2	134	711	228	173	180	m/s
a_0	96	413	209	126	147	m/s
u_2/a_0	1.40	1.72	1.09	1.37	1.22	-
Pr	3.73	6.45	2.27	3.20	2.54	-
COP	8.20	7.79	4.47	8.34	7.89	-

Table 1. Calculation of impeller speed and diameter of a 500 ton (1760 kW) centrifugal compressor for ketone C6F and other working fluids

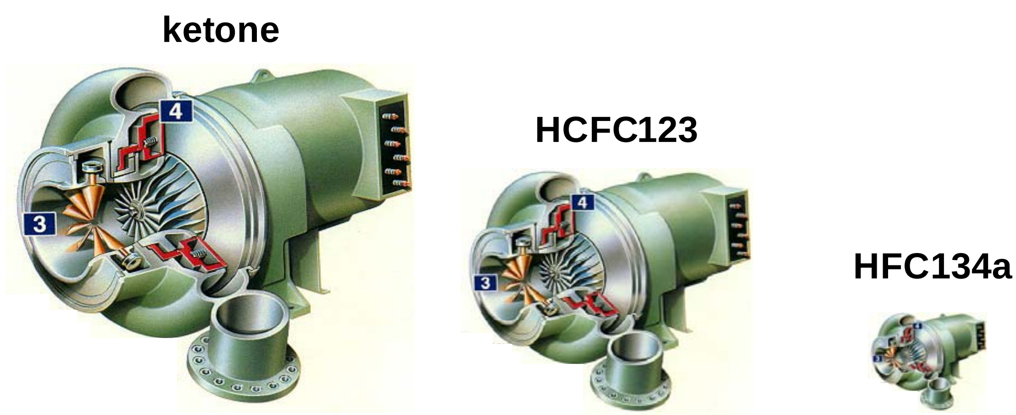
These calculations show that C6F with a suction line heat exchanger results in a low rpm N, a low impeller tip speed u_2 and a large impeller diameter D compressor design with a COP approaching that of HCFC123. The low isentropic enthalpy rise of the compressor corresponds to a low head and results in a low tip speed

u_2. Combined with the low density of C6F – comparable to, but lower than that of HCFC123 – it results in a compressor rotational speed close to 50/60 Hz allowing single stage direct drive operation at the appropriate specific speed selection. To illustrate the relative size of a C6F compressor Figure 9 shows its size in relation centrifugal compressors with the natural refrigerants water and CO_2 while Figure 10 makes a compressor size comparison with the two currently most popular centrifugal chiller refrigerants HCFC123 and HFC134a.



Impeller diameter:	1.515 m	.991 m	.085 m
Impeller rotational speed:	8,966 rpm	2,587 rpm	51,177 rpm
Impeller tip speed:	711 m/s	134 m/s	228 m/s

Figure 9. Illustration of relative, size and speed of a 500 ton centrifugal compressor with ketone C6F, water and CO_2 as refrigerants



Impeller diameter:	.991 m	.608 m	.203 m
Impeller rotational speed:	2,587 rpm	5,423 rpm	14,497 rpm
Impeller tip speed:	134 m/s	173 m/s	180 m/s

Figure 10. Illustration of relative, size and speed of a 500 ton centrifugal compressor with ketone C6F, HCFC123 and HFC134a as refrigerants

5. CONCLUSIONS

1. Ketone C6F shares the environmental advantages of natural refrigerants (zero ODP and negligible GWP) with the engineering advantages of man-made fluids (reasonable power density and non-flammable and non-toxic).
2. The low cycle efficiency of ketone C6F in the simple vapor compression cycle is dramatically improved by the use of a suction line heat exchanger, resulting in COP values in line with expectations based on its critical temperature.
3. The low head requirement for ketone C6F combined with its large volumetric flow rate requirement enable gearless single-stage direct-drive 50/60 Hz compressor operation for compressors serving chillers in the 100 to 500 ton capacity range.
4. The low impeller tip speeds allow the use of alternate materials for impeller fabrication
5. The sub-atmospheric evaporator (0.162 bar) and condenser (0.604 bar) pressures necessitate the use of a purge but eliminate the need for pressure vessel code certification.

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