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A Novel Centrifugal Air Conditioning System

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A NOVEL CENTRIFUGAL AIR CONDITIONING SYSTEM

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ABSTRACT

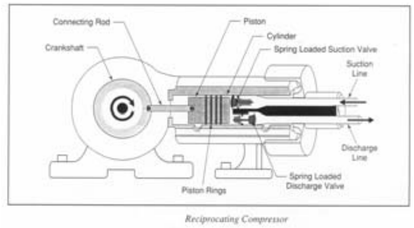
Conventional centrifugal compressors sized for residential heating and cooling heat pumps are not available. In this study, we examine the potential of a new type of centrifugal heat pump in which the refrigerant compression, condensation, expansion and evaporation processes are contained in a rotating loop component that is directly coupled to the fan blades. In this paper, we discuss the concept and analyze its theoretical thermodynamic performance with several refrigerants, including pure refrigerants and mixtures.

1. INTRODUCTION TO CONVENTIONAL COMPRESSORS

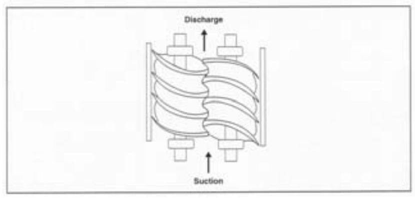
The compressor is one of the four essential components of a vapor compression refrigeration system. The compressor circulates refrigerant through the system in a continuous cycle and raises the temperature and pressure of the refrigerant vapor. Currently there are two main types of compressors: positive displacement and dynamic [1, 2]. Positive-displacement compressors increase the pressure of refrigerant vapor by mechanically reducing the volume of the compression chamber. Positive-displacement compressors include reciprocating (crank and piston) and rotary types. Dynamic compressors increase the pressure of refrigerant vapor by a continuous transfer of angular momentum from the rotating member to the vapor followed by the conversion of this momentum into a pressure rise.

Figure 1 illustrates differences between compressor types. Reciprocating compressors are the most common type of compressors. Most reciprocating compressors are single-acting, using pistons that are driven directly through a pin and connecting rod from the crankshaft. One type of rotary compressor is the rotary-screw compressor shown in Fig. 1(b). This type of compressor is widely used in industry and commercial building cooling systems. Two-screw compressors use two helical rotors that rotate toward each other inside of a volute causing the lobes to mesh. As the left rotor turns clockwise, the right rotor rotates counterclockwise. This forces the gases to become trapped in the central cavity formed between the rotors and the volute. At least one of the rotors is attached to a drive shaft that transmits the mechanical energy needed to operate the compressor. This type of compressor in a single stage has the capability to operate at pressure ratios (ratio of absolute discharge pressure to absolute suction pressure) above 20:1. The capacity range currently available is from 70 to 4600 kW. Centrifugal compressors are commonly used for large systems, including chillers. In a centrifugal compressor, gas enters near the axis of a spinning impeller and is accelerated tangentially and radially outward to the periphery of the wheel. In this process, the vapor velocity increases, and the increase in kinetic energy of the gas is converted into pressure as the gas leaves the volute and enters the discharge header pipe. Typically, centrifugal compressors deliver much higher flow rates than positive displacement compressors. Usually centrifugal compressors operate at speeds more than 3,000 rpm.

To protect against abnormal conditions the conventional centrifugal compressors must be provided with several devices, such as high-pressure protection, high-temperature control, low-pressure protection, low voltage or reversal protection, and a suction line strainer. Lubrication, prescribed by the compressor manufacturers in accordance with the many different operating levels and the specific refrigerant being used is a requirement [4].



(a)



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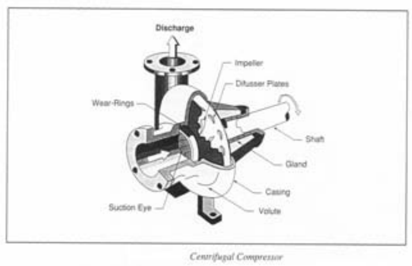

(c)

Figure 1.1 Compressor types; (a) reciprocating; (b) rotary-screw; (c) conventional centrifugal.

There are two other major types of compressors: the scroll and the rotary. Although the principle of the scroll compressor was known for almost 100 years, it was not until about 1980 when precision design and manufacturing was sufficiently advanced to achieve the close tolerances required for the scroll compressor. This type of compressor uses two identical scroll pieces that are meshed together to form crescent-shaped pockets that carry the refrigerant gas. One scroll is stationary, and the other orbits (not rotates) compressing the gas taken in at the periphery of the scroll and exiting at the center. The rotary compressor, a simpler technology, uses a rotor that rotated inside of a cylinder. The center of the bore of the cylinder is offset from the center of the rotor, and this creates a refrigerant volume that changes as the rotor rotates.

Although some compressor types are more tolerant than others, all compressors available are designed to handle gas only, and for this reason, means for providing some superheat to the suction gas are usually employed. If these means were not used, the compressor could be damaged by liquid refrigerant that causes slugging, floodback and flooded starts. The degree of damage depends on the quantity of liquid, the frequency and duration of liquid (or wet gases) entering the compressor and the type of compressor. Generally, reciprocating compressors, because they are equipped with discharge valves, are more susceptible to damage from slugging than rotary and scroll compressors [4].

2. CONCEPT OF THE ROTATING HEAT PUMP

As mentioned above, a conventional compressor is somewhat complicated and all require tight tolerances between moving and stationary parts. Moreover, lubrication is required to prevent wear and cool the compressor. Since the lubrication oil is mixed with the refrigerant, it is carried to the heat exchangers where it inhibits heat transfer between the tubing inside the condenser and evaporator and the refrigerant itself. This fouling of the heat exchangers reduces the performance (capacity and efficiency) of the compressor. The complexity of conventional compressors and their need for lubrication makes it interesting to explore an alternative type of integrated compressor system that combines the compressor function with the heat exchangers in a way that is simple, does not have metal-to-metal wear (no lubrication requirement), and likely to be a low-cost option.

ORNL conceived of a novel concept for an integrated approach to a centrifugal heat pump [3]. The concept adopts a closed loop to serve as a refrigerating circuit instead of heat exchangers and a conventional compressor. The design is such that the refrigerant stays in the loop rather than being drawn to or from the compressor into stationery heat exchangers. In this concept, as the loop is rotated, the working fluid undergoes an acceleration process, which can be transferred into pressure rise. Figure 2.1 shows one potential configuration for accomplishing the concept.

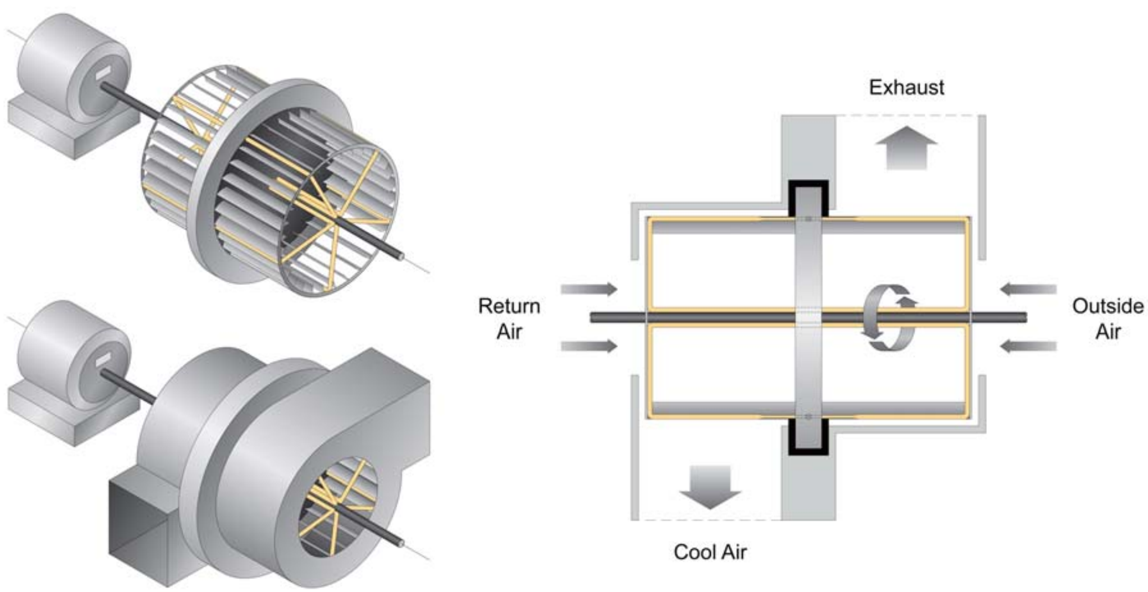

Figure 2.1. Novel conceptual centrifugal heat pump

A section of the concept shown at the right of Figure 2.1 details fan blades that have a "compressor/condenser" section and an "evaporator" section. An expansion device is located at the junction of two sections. Although more loops could be included, the left side of Fig. 2.1 shows five of these loops and each one of them is an independent circuit, i.e. closed. The assembly of blades with "compressor/condenser" and "evaporator" sections rotates inside of a two-chamber fan volute with the "compressor/condenser" side for rejection of heat to an air stream and the "evaporator" side for the removal of heat from a second air stream as shown.

A simplified schematic of one of the loops is illustrated in Figure 2.2. A vertical axis of rotation for the loop is at the left. Each leg of the loop constitutes one or more processes needed for a vapor compression system. Process (1-2) is assumed to be an isentropic compression where the working fluid enters point 1 as a saturated vapor (quality ≤ 1). For the current analysis, the entire compression process is assumed to be fully in the refrigerant superheat region, although it is possible that the compressor could also operate in the saturated region as well. The process (2-3) is a condensation process. The process (3-4) is a throttling process. The process (4-6) is an evaporative process. As in the case of a conventional compressor, both evaporation and

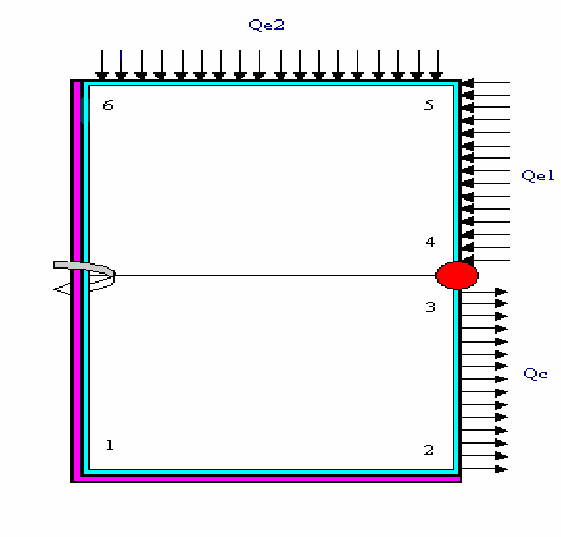

Figure 2.2 Loop section (vertical axis of rotation at LH side).

condensation are two-phase flow. Finally, the process (6-1) is considered as an insulated process, which means that the thermodynamic state at point 1 is equal to that at point 6 (T1=T6, P1=P6, h1=h6). Each of these processes is mathematically described in detail in Section 3.

The advantages of the proposed centrifugal heat pump would appear to be (1) simplicity, (2) no requirement for lubrication since there are no wear surfaces in contact with the refrigerant, and (3) a high tolerance for wet gas application.

3. MATHEMATICAL ANALYSIS

The conservation of energy principle for a control volume is put on a rate basis by dividing the energy variation with the mass by the time interval, hence

$$d\dot{Q} + d\dot{W} = d\dot{H} + d\dot{K}e + d\dot{P}e \tag{1}$$

In the above equation, $d\dot{K}e$ is change in kinetic energy, including two parts – rotational kinetic energy and translational kinetic energy. $d\dot{P}e$ is change in potential energy. In the case of centrifugal compression, the translational kinetic and potential energies are significantly less than rotational kinetic energy. The terms $d\dot{Q}, d\dot{W}$ and $d\dot{H}$ are changes in heat flux, compression work and enthalpy variation, respectively. For a steady-state control volume, the properties at a given position do not change with time, and this means that $\dot{m} = \dot{m}_i = \dot{m}_e$. Therefore, the above conservation of energy equation for a control volume can be written on a unit-mass basis in the form

$$dq + dw = \left(h + \frac{V_{Tran}^2}{2} + \frac{V_{Rot}^2}{2} + gz \right)_e - \left(h + \frac{V_{Tran}^2}{2} + \frac{V_{Rot}^2}{2} + gz \right)_i \tag{2}$$

Based on Eq. 2, we can analyze each process along the loop.

<u>Centrifugal compression process</u>

As mentioned earlier, the compression process is considered as an isentropic process. In Eq. (2), mechanical energy includes kinetic energy and potential energy in a control volume while the kinetic energy consists of translational kinetic energy and rotational kinetic energy. Usually translational kinetic energy and potential energy are much smaller than rotational kinetic energy, and ignoring them won't affect the accuracy of analysis. Based on the fact that $V_R=\omega R$ with ω the angular velocity of rotation and, on the basis of the preceding discussion, Eq (2) may be rewritten as

$$dw = dh + d\left(\frac{V_R^2}{2} \right) = dh + \omega^2 r dr \tag{3}$$

The compression work is generated due to centrifugal force, and centrifugal force can be given by $F = ma = m\omega^2 r$. Therefore, the compression work during the process of centrifugal compression can be expressed by $d\dot{W} = d(\dot{m}a \cdot r) = \dot{m}adr + \dot{m}rda + ard\dot{m} + = \dot{m}\omega^2 rdr + \dot{m}\omega^2 rdr + \omega^2 r^2 d\dot{m}$.

If the process is steady state, $d\dot{m} = 0$. Thus we get

$$dw = \frac{d\dot{W}}{\dot{m}} = 2\omega^2 rdr \,. \tag{4a}$$

Equating Eq.(4a) and Eq (3), we can attain the enthalpy variation, as is described by

$$dh = \omega^2 rdr \tag{4b}$$

At the radius R of the periphery, the compression work and enthalpy variation on a unit-mass basis are achieved after the integration of the control volume energy term along the compression route.

$$\Delta h_{1-2} = \frac{1}{2}\omega^2 R^2 \tag{4c}$$

and

$$\Delta w_{1-2} = \omega^2 R^2 \tag{4d}$$

The centrifugal stage pressure ratio can be expressed as a function of the Mach number at the periphery derived from Eq (4). The Mach number is described as the form of $M = U/\sqrt{kRT}$ [2]. The equation (4d) can be rewritten as follows

$$(p_2 v_2 - p_1 v_1) + (u_2 - u_1) = \frac{U_R^2}{2} \tag{5}$$

For analytical convenience, we considered the working fluid to be an ideal gas. Thus we get the following equations:

$$\frac{p_2}{p_1} = \left(\frac{v_1}{v_2}\right)^k,\ \frac{T_2}{T_1} = \left(\frac{p_2}{p_1}\right)^{\frac{k-1}{k}},\ u_1 = \frac{RT_1}{k-1},\ u_2 = \frac{RT_2}{k-1},\ p_1 v_1 = RT_1, \text{ and } c_v = \frac{R}{k-1}$$

The equation (5) is divided by $p_1 v_1$ on left side and RT_1 on right side. Then we get the pressure ratio of a centrifugal compression process from the equation (5), as follows

$$\mathrm{Pr}_{1-2} = \left(1 + M_R^2 (k-1)/2\right)^{k/k-1} \tag{6}$$

The above analysis is on the basis of a compressible flow with energy exchange. The equation illustrates that the pressure ratio of compression is strongly dependent of peripheral Mach number. There is no direct effect of density of working fluid on pressure ratio. If we consider the flow to be incompressible flow ($\rho = const$) without the variation of internal energy, a completely different form is attained from the equation (5): $dpv = \omega^2 rdr$ and $v = 1/\rho = const$

Then, we obtain

$$dp = p_e - p_i = \rho\omega^2 rdr \tag{7}$$

Equation (7) shows that the pressure variation of incompressible flow depends strongly on the density of the working fluid as well as rotational speed and geometry. Unfortunately, Eq. (7) can only be suitable for the case of incompressible flow. This equation considers the force balance while ignoring the variation of internal energy. Therefore, it is incorrect to analyze the phenomenon of compressible flow accompanying the variation of internal energy.

Condensation process

Since the rotational radius of the condensation process is constant, the variation of kinetic energy and potential energy are zero in accordance with the preceding discussion. Also the variation of radius is zero. Therefore, the energy conservation becomes

$$dw = \omega^2 r dr = 0 \tag{8a}$$

$$dq = h_e - h_i \tag{8b}$$

From Eq. (8a), we could find there is no pressure variation during the process of condensation (2-3). The integration of the control-volume energy will lead to Eq. (8) becoming

$$\Delta h_{2-3} = \Delta q_{cond} \tag{8c}$$

where Δq_{cond} is negative.

Evaporation process

The evaporation process includes two parts: a peripheral zone and an axial zone. In accordance to the same principle, we can derive the energy equations. For the peripheral zone (4-5), the rotational radius and speed of condensation process is constant. Therefore, the variation of kinetic energy and radius is zero in accordance with the preceding discussion. The energy conservation becomes

$$dw = \omega^2 r dr = 0 \tag{9a}$$

$$dq = dh \tag{9b}$$

For the axial zone (5-6), if we consider expansion work and ignore translational kinetic energy and potential energy, the Equation (2b) becomes

$$dw = \frac{d\dot{W}}{\dot{m}} = dw_R + dw_{\exp ansion} = 2\omega^2 r dr + dw_{\exp ansion} \tag{9c}$$

$$dq + dw = dh + \omega^2 r dr \tag{9d}$$

where Δq_{evap} is positive; w_R is the centrifugal compression work; $w_{expansion}$ is the possible expansion work done by working fluid. Integrating to determine the control-volume energy for processes (4-5) and (5-6) leads to the following equations

$$\Delta w_{4-5} = 0 \tag{10a}$$

$$\Delta h_{4-5} = \Delta q_{evap1} \tag{10b}$$

$$\Delta w_{5-6} = w_{\exp ansion} - \omega^2 R^2 \tag{10c}$$

$$\Delta h_{5-6} = \Delta q_{evap2} + w_{\exp ansion} - \omega^2 R^2 / 2 \tag{10d}$$

and

$$\Delta h_{4-6} = \Delta h_{4-5} + \Delta h_{5-6} = \Delta q_{evap1} + \Delta q_{evap2} + w_{\exp ansion} - \omega^2 R^2 / 2 \tag{10e}$$

From the equation (10a), we see that there is no pressure variation during the process of evaporation (4-5). However, there is pressure variation during the process of (5-6). Let's derive the form of pressure ratio (p_5/p_6) from the equation (9d).

$$\left(p_6 v_6 - p_5 v_5\right) + \left(u_6 - u_5\right) = \Delta q_{evap2} + w_{\exp ansion} - \frac{U_R^2}{2} \tag{11a}$$

or

$$\left(p_5 v_5 - p_6 v_6\right) + \left(u_5 - u_6\right) = \frac{U_R^2}{2} - \Delta q_{evap2} - w_{\exp ansion}\,. \tag{11b}$$

If we employ the perfect gas relation, we obtain

$$\Pr_{6-5} = \frac{p_5}{p_6} = \left(1 + \left(\frac{U_R^2 - 2\left(\Delta q_{evap2} + w_{\exp ansion}\right)}{k p_6 v_6}\right)(k-1)/2\right)^{k/k-1} \tag{11c}$$

If we ignore heat flux and expansion work, the above equation becomes equation (6d). Therefore, allowing evaporation to take place between points 5 and 6 would reduce the pressure ratio between these points. We also assumed that $P_1=P_6$, that is we assumed that there was no flow restriction between these two points along the axis of rotation. Based on Eq. (11c), we see that p_5 is always less than p_2. This means that flow will occur in the direction needed for the cycle to operate. This supports the contention that the centrifugal thermal cycle is possible. However, if $2\left(\Delta q_{evap2} + w_{\exp ansion}\right)$ is larger than U_R^2, $\Pr_{6-5}$ will be less than 1 (that is, P_5 is less than P_6). Normally P_5 is equal to P_6.

<u>Throttling process and COP calculation</u>

The throttling process (3-4) is considered to be an isenthalpic process. Therefore $h_3 = h_4$. The calculation of COP is given as follows

$$COP = \frac{\Delta q_e}{\Delta q_c - \Delta q_e} \tag{12}$$

4. PERFORMANCE ANALYSIS

In this report describing a theoretical analysis of the novel compressor, the cooling mode is considered. We examined the performance of several refrigerants and mixtures that may offer a boost in the performance of centrifugal air conditioning system. The screening range of these refrigerants covers CFC, HCFC, HFC, and natural refrigerants. Some of the CFCs and HCFCs were chosen simply as a benchmark in the analysis and not to suggest their use since they contain chlorine, an ozone depleter. The pure refrigerants considered in the analysis include R11, R123, R245CA, R134A, R22, CO2, R141B, R218, R227EA, R245FA, R236EA, and R236FA. The mixtures were chosen on the basis of available information from properties of the pure refrigerant constituents. The first simulation condition is single stage system where the condensation temperature is 40°C and the saturated evaporation temperature is 10°C. A second, follow-on simulation explored the potential for improving the performance of the system using a multistage system. In the multistage analysis, the condensation temperature of the first stage was 25°C and the suction (evaporating) temperature was 10°C saturated vapor; the condensation temperature of the second stage is 40°C and the suction line of both cases is at 25°C saturated vapor. We assumed a radius of rotation to be 0.6m in each case.

Pure Refrigerant Evaluation

Figure 4 shows plots of the COP evaluation of the discussed refrigerants at the condition of a single stage. Figure 5 illustrates the rotation speed required for each case. Detailed data for refrigerant state points in the cycle are provided in Table 1. From the view of Fig. 4, the COP of the centrifugal heat pump is significantly less than that of traditional refrigerating cycle, as caused by the compression effect existing during the evaporative process. This compression effect along the evaporative process could cost more energy consumption. The conventional refrigerants, such as R11, R123 and R141B, possess slightly higher COP than others. However, these refrigerants are either CFC or HCFC, which is a prohibited refrigerant because of its high ozone depletion index. On the other hand, their rotational speeds are more than 3000rpm, which is negative because such kind of rotational speed is beyond the current fan motor technology. Of the remaining refrigerants evaluated, we can't find one which achieves both higher COP and lower rotational speed, simultaneously.

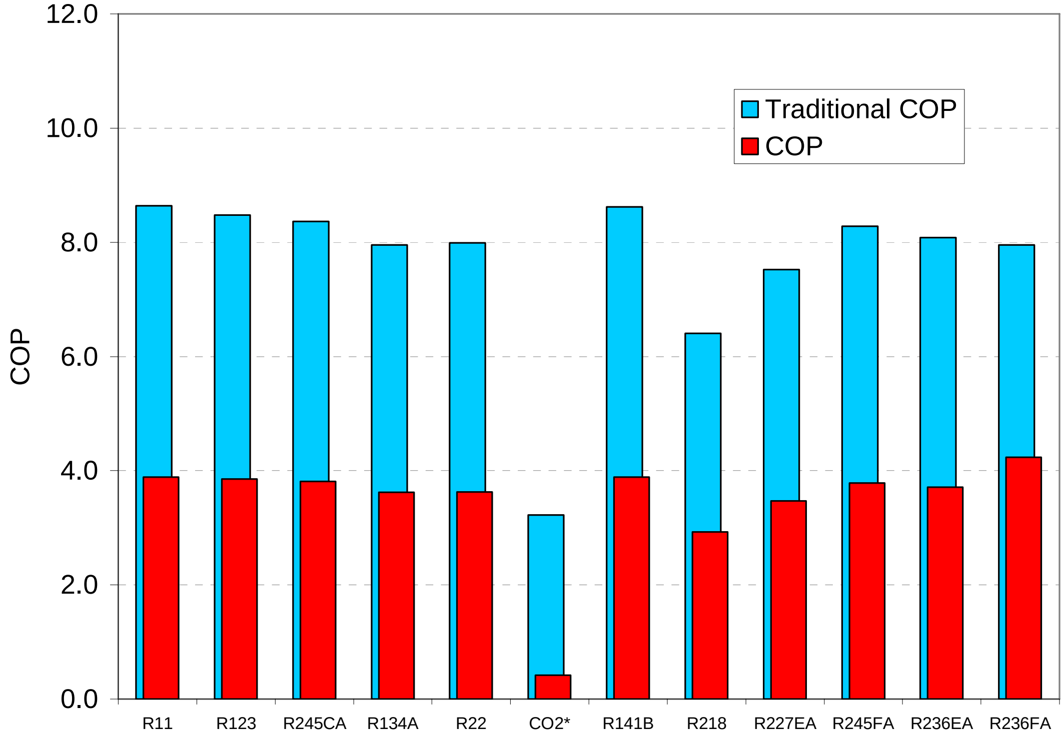

Figure 4.1. The cooling mode COP of selected pure refrigerants in the novel centrifugal heat pump

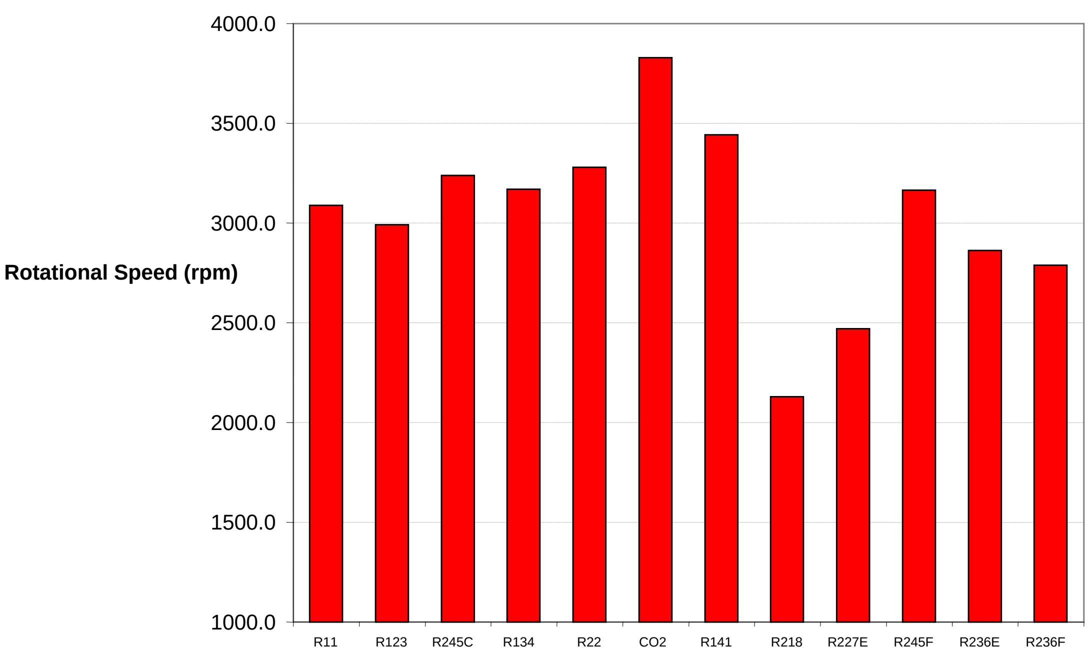

Figure 4.2 The rotational speed of selected pure refrigerants in the novel centrifugal heat pump.

Figure 4.2 The rotational speed of selected pure refrigerants in the novel centrifugal heat pump

Table 4.1 Performance of selected pure refrigerants in the novel system.

Point	R11	R123	R245CA	R134A	R22	CO2*	R141B	R218	R227EA	R245FA	R236EA	R236FA
Pressure (Kpa) at point 1	60.677	50.567	54.677	414.610	680.950	4502.200	43.492	565.940	280.130	82.925	118.020	159.720
Pressure (Kpa) at point 2	174.430	154.470	173.470	1016.600	1533.600	9000.000	132.820	1278.400	702.350	251.790	337.650	437.770
Pressure (Kpa) at point 2'	174.430	154.470	173.470	1016.600	1533.600	9000.000	132.820	1278.400	702.350	251.790	337.650	437.770
Pressure (Kpa) at point 3	174.430	154.470	173.470	1016.600	1533.600	9000.000	132.820	1278.400	702.350	251.790	337.650	437.770
Pressure (Kpa) at point 4	64.293	54.858	60.598	482.682	781.720	7257.220	46.248	759.376	341.913	92.753	135.576	138.766
Pressure (Kpa) at point 5	64.293	54.858	60.598	482.682	781.720	7257.220	46.248	759.376	341.913	92.753	135.576	138.766
Pressure (Kpa) at point 6	60.677	50.567	54.677	414.610	680.950	4502.200	43.492	565.940	280.130	82.925	118.020	159.720
Pressure ratio at compression	2.875	3.055	3.173	2.452	2.252	1.999	3.054	2.259	2.507	3.036	2.861	2.741
Temperature (C) at point 1	10.000	10.000	10.000	10.000	10.000	10.000	10.000	10.000	10.000	10.000	10.000	10.000
Temperature (C) at point 2	44.960	40.000	40.000	43.073	52.930	62.000	43.018	40.000	40.000	40.000	40.000	40.000
Temperature (C) at point 2'	40.000	40.000	40.000	40.000	40.000	40.000	40.000	40.000	40.000	40.000	40.000	40.000
Temperature (C) at point 3	40.000	40.000	40.000	40.000	40.000	40.000	40.000	40.000	40.000	40.000	40.000	40.000
Temperature (C) at point 4	11.395	11.862	12.287	14.614	14.686	28.376	10.926	18.145	14.390	11.746	12.398	7.739
Temperature (C) at point 5	11.395	11.862	12.287	14.614	14.686	28.376	10.926	18.145	14.390	11.746	12.398	7.739
Temperature (C) at point 6	10.000	10.000	10.000	10.000	10.000	10.000	10.000	10.000	10.000	10.000	10.000	10.000
Enthalpy (kJ/kg) at point 1	174.300	159.430	189.000	238.510	254.560	422.880	207.790	133.530	148.490	193.010	167.950	168.270
Enthalpy (kJ/kg) at point 2	192.800	176.750	209.190	257.100	274.450	447.400	230.820	141.390	159.650	212.230	183.490	182.890
Enthalpy (kJ/kg) at point 2'	189.670	176.750	209.190	253.620	262.240	343.780	228.350	141.390	159.650	212.230	183.490	182.890
Enthalpy (kJ/kg) at point 3	14.488	12.567	20.086	90.599	95.645	343.780	9.286	83.195	64.546	33.799	42.379	9.286
Enthalpy (kJ/kg) at point 4	14.488	12.567	20.086	90.599	95.645	343.780	9.286	83.195	64.546	33.799	42.379	9.286
Enthalpy (kJ/kg) at point 5	173.029	157.881	186.760	234.139	249.938	399.946	206.237	129.329	145.117	190.579	165.278	169.586
Enthalpy (kJ/kg) at point 6	174.300	159.430	189.000	238.510	254.560	422.880	207.790	133.530	148.490	193.010	167.950	168.270
Mach number at 1	0.000	0.000	0.000	0.000	0.000	0.000	0.000	0.000	0.000	0.000	0.000	0.000
Mach number at 2	1.407	1.461	1.490	1.271	1.166	0.803	1.455	1.225	1.315	1.458	1.418	1.385
Mach number at 2'	1.407	1.461	1.490	1.271	1.166	0.803	1.455	1.225	1.315	1.458	1.418	1.385
Mach number at 3	1.407	1.461	1.490	1.271	1.166	0.803	1.455	1.225	1.315	1.458	1.418	1.385
Mach number at 4	1.407	1.461	1.490	1.271	1.166	0.803	1.455	1.225	1.315	1.458	1.418	1.385
Mach number at 5	1.407	1.461	1.490	1.271	1.166	0.803	1.455	1.225	1.315	1.458	1.418	1.385
Mach number at 6	0.000	0.000	0.000	0.000	0.000	0.000	0.000	0.000	0.000	0.000	0.000	0.000
Compression work (kJ/kg) (1-2)	37.602	35.287	41.341	39.611	42.393	57.816	46.737	17.880	24.044	39.494	32.323	30.665
kinetic energy variation (kJ/kg) (1-2)	19.102	17.967	21.151	21.021	22.503	33.296	23.707	10.020	12.884	20.274	16.783	16.045
Compression work (kJ/kg) (4-5)	-1.940	-2.451	-3.520	-6.311	-6.632	-37.092	-2.430	-6.241	-5.023	-3.808	-4.101	4.056
kinetic energy variation (kJ/kg) (4-5)	-19.102	-17.967	-21.151	-21.021	-22.503	-33.296	-23.707	-10.020	-12.884	-20.274	-16.783	-16.045
Rotation speed (rpm):	3087.785	2991.227	3237.642	3169.189	3278.581	3828.817	3442.461	2129.245	2469.112	3164.511	2862.825	2788.425
COP	3.891	3.851	3.814	3.621	3.629	0.420	3.890	2.928	3.472	3.787	3.709	4.234
COP of traditional refrigerating cycle	8.638	8.479	8.366	7.956	7.990	3.226	8.619	6.404	7.522	8.284	8.081	7.957
Carnot COP	8.713	8.573	8.467	8.097	8.119	3.749	8.697	6.694	7.706	8.394	8.211	8.100

In the theoretical analysis, several new HFC refrigerants, such as R245FA, R236EA, and R236FA, showed performance similar to that of R11 and R22 in the novel centrifugal air conditioning system. Their COPs varied over the range of 40-50% of that of a conventional system. Rotational speeds for these refrigerants was about 3000 rpm – slightly lower than those of R11 and R12. Detailed information for these refrigerants is shown in Table 1. Based on our theoretical analysis, CO_2 seems to have no advantage in achieving better COP and lower rotational speeds than the other refrigerants analyzed. Importantly, the critical temperature of CO_2 is low - around 31^{o}C - and this is lower than the typical condensation temperature (40^{o}C). Therefore, the CO2 cycle operates in the transcritical regime so that the discharge temperature is higher than the sink temperature. Therefore, the condensing process (at a subcritical temperature) is replaced by a supercritical gas cooling process, and this leads to a wide glide temperature across the gas cooler. Although the high glide temperature of the CO2 is an advantage for a countercurrent heat exchanger, e.g. for heating water in a single pass heat exchanger, it is of no particular advantage in an air-cooled gas cooler. Therefore, temperature glide in a gas cooler for the high pressure side of the CO2 cycle is similar to what happens in a conventional vapor compression cycle using a high-glide refrigerant mixture. Better temperature matching across the heat exchanger and improved effectiveness is the outcome. In the next section, we will discuss work done in FY05 to examine the performance of the novel centrifugal cycle using selected, near-azeotropic refrigerant mixtures.

It can be seen from Figure 4.4, that of the mixtures examined, R218 and R277EA possess significantly lower rotational speeds (RPM) with a slightly penalty of COP as shown in Figure 4.3. In the case of a 0.6m radius or rotation, the rotational speeds of R218 and R227ea are 2100 rpm and 2400 rpm, respectively. The other refrigerants we analyzed typically required rotational speeds of nearly 3000 rpm or more for the same condition. A rotational speed between 2100 rpm and 2400 rpm is easily approached with current fan motor technology Therefore, of the mixtures evaluated, R218 and R277EA seem to be the only choice for the current centrifugal heat pump. Compared to the other single refrigerants, R218 and R227EA have a more significant advantage in lower rotational speed although their COP is a little bit less than the others. Although we did not intend to examine other effects and considerations of alternative refrigerant mixtures, care must be taken. For example, limited data suggests that R218 could cause health effects in low concentrations. The symptoms may include dizziness, headache, nausea and loss of co-ordination. In high concentrations, like many refrigerants, R218 may cause asphyxiation and loss of mobility /consciousness [5]. Hence, R218 might be serve as one of the components in a refrigerant mixture, it may not be suitable as a single component working fluid on the basis of safety. Unlike R218, R227ea is classified as an A1 safety group by ASHRAE [4]. This means that R227ea shows no evidence of toxicity below 400ppm and will not propagate a flame under normal conditions in open air. Currently, R227ea has been proposed as replacement for R114 in high temperature heat pumps, and should be acceptable for the novel centrifugal system from the standpoint of flammability and safety.

Mixed Refrigerant Evaluation

The performance and thermal properties of a mixed refrigerant are determined by components and their fraction in the mixture. In this study, the focus is refrigerant performance in the novel cycle as defined by COP and rotational speed. Thus, when we choose one component with higher COP and higher rotational speed in a mixture, we needed also to examine the other components with lower rotational speed. Finally, a suitable mixture might be achieved by optimizing its composition based on the weight fraction of the individual components. Based on the foregoing, two groups of refrigerants were considered: the first is the group of well-known refrigerants such as R32, R134A, R125, R142B, and R22, are considered as the components with higher COP and higher rotational speed in the mixtures, and the second group are less well-known refrigerants such as R218 and R227EA that have smaller rotational speeds. We ran the model with mixtures based on property data that was weighted according to the weight percent of the constituents. The results of these runs is shown in Figures 6 and 7, and detailed performance information for each section of the loop is listed in Table 2.

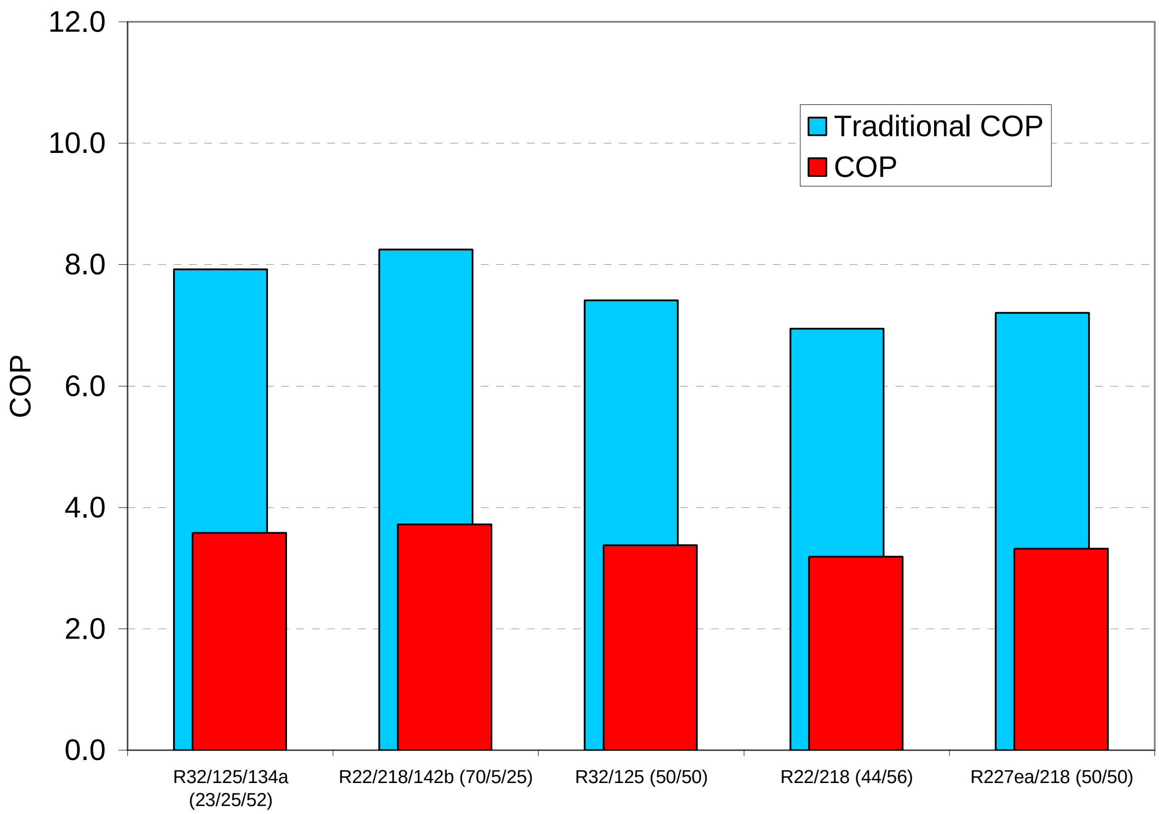

Figure 4.3 The cooling-mode COP of selected refrigerant mixtures in the novel system.

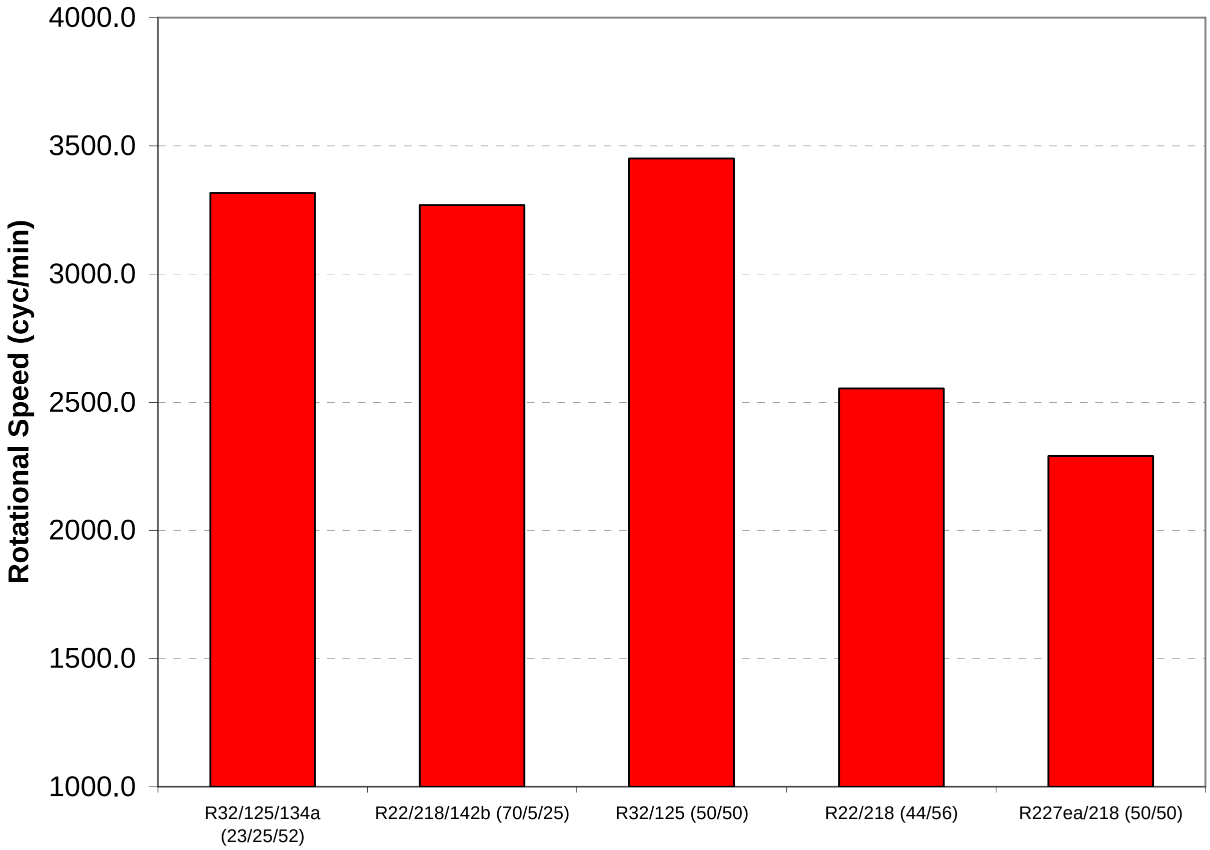

Figure 4.4 The rotational speed of selected refrigerant mixtures in the novel system.

Earlier we discussed the penalty associated with temperature glide for air cooled (or air heated) cross flow heat exchangers. Because of this penalty, we selected mixed refrigerants that are azeotropic or near azetropic. Figure 6 shows that the performance of several of these mixtures is about the same; however, mixtures comprised of R218 or R227ea have significantly lower rotational rates. Notable in this regard is the 50/50 misture of R227ea and R218. As compared to pure refrigerants (e.g. R218), the mixture of R227ea and R218 has a better performance (COP = 3.32) and rotational speed (2290 rpm).

Figure 4.5 shows T-S diagram of phase equilibrium data for the 50/50 mixture of RR227ea and 218. The gliding temperature of the mixture is around 3-4 K. Although the figure is complex, we determined from it that the R218/R227ea mixture can undergo a wet compression process, and in the novel centrifugal system, this may be an advantage from the standpoint of refrigerant control and possibly cycle performance.

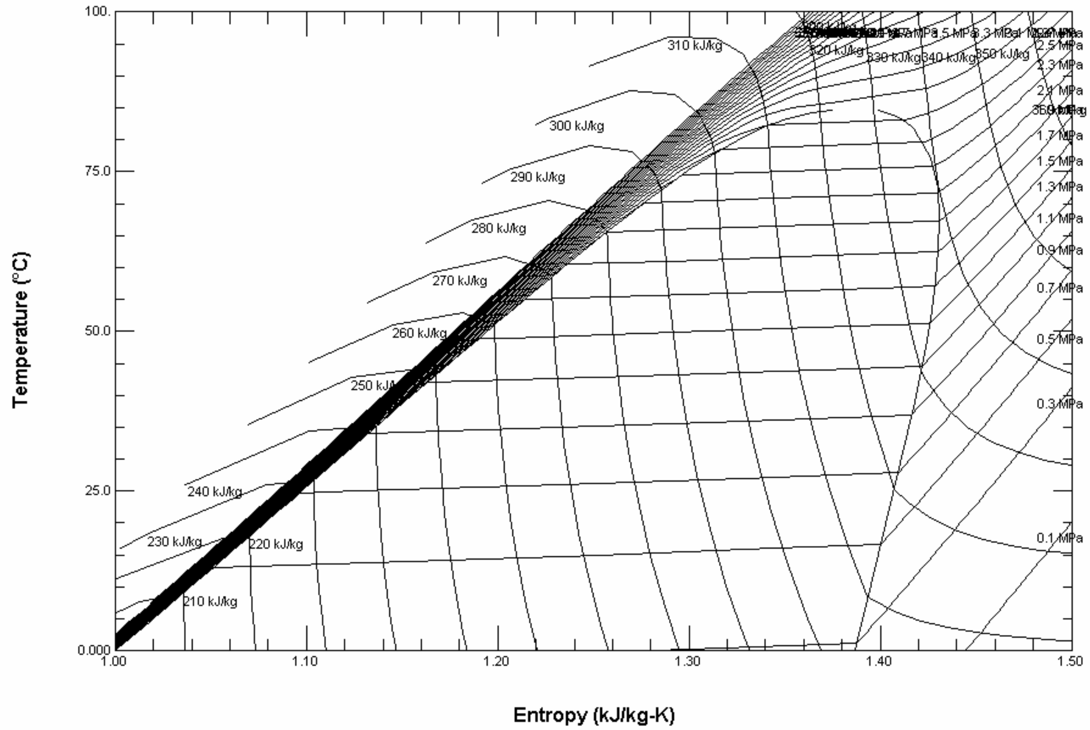

Figure 4.5 The T-S diagram of a mixture of R218 and R227ea (50/50 mass%)

Table 4.2 Performance of selected mixed refrigerants in the novel system.

Point	R32/125/134a	R22/218/142b	R32/125	R22/218	R227ea/218
Composition (%)	23/25/52	70/5/25	50/50	44/56	50/50
Pressure (Kpa) at point 1	710	516	1085	892	430
Pressure (Kpa) at point 2	1645	1193	2419	1931	1012
Pressure (Kpa) at point 3	1645	1193	2419	1931	1012
Pressure (Kpa) at point 4	831	576	1337	1130	539
Pressure (Kpa) at point 5	831	576	1337	1130	539
Pressure (Kpa) at point 6	710	516	1085	892	430
Pressure ratio at compression	2.318	2.312	2.230	2.166	2.354
Temperature (C) at point 1	13.0	14.3	10.0	10.3	11.8
Temperature (C) at point 2	50.7	52.8	52.6	42.4	41.1
Temperature (C) at point 3	37.5	36.2	40.0	39.8	38.6
Temperature (C) at point 4	13.3	10.6	17.2	18.4	16.4
Temperature (C) at point 5	17.9	17.5	17.2	18.7	18.7
Temperature (C) at point 6	13.0	14.3	10.0	10.3	11.8
Enthalpy (kJ/kg) at point 1	274.8	252.9	297.6	182.0	144.5
Enthalpy (kJ/kg) at point 2	294.8	272.9	318.9	193.6	153.9
Enthalpy (kJ/kg) at point 3	115.7	88.2	140.0	102.1	76.8
Enthalpy (kJ/kg) at point 4	115.7	88.2	140.0	102.1	76.8

Enthalpy (kJ/kg) at point 5	269.4	249.2	289.8	177.0	140.9
Enthalpy (kJ/kg) at point 6	274.8	252.9	297.6	182.0	144.5
Mach number at 1	0.000	0.000	0.000	0.000	0.000
Mach number at 2	1.201	1.210	1.126	1.152	1.263
Mach number at 3	1.201	1.210	1.126	1.152	1.263
Mach number at 4	1.201	1.210	1.126	1.152	1.263
Mach number at 5	1.201	1.210	1.126	1.152	1.263
Mach number at 6	0.000	0.000	0.000	0.000	0.000
Compression work (kJ/kg) (1-2)	43.360	42.121	46.943	25.726	20.689
kinetic energy variation (kJ/kg) (1-2)	23.280	22.151	25.673	14.216	11.299
Compression work (kJ/kg) (4-5)	-7.530	-5.181	-11.163	-7.436	-5.275
kinetic energy variation (kJ/kg) (4-5)	-23.280	-22.151	-25.673	-14.216	-11.299
Rotation speed (rpm):	3316	3268	3450	2554	2290
COP	3.58	3.72	3.38	3.19	3.32
COP of traditional refrigerating cycle	7.92	8.25	7.41	6.95	7.21
Carnot COP	8.08	8.37	7.59	7.18	7.44

Cascade System Evaluation

As was discussed earlier, most of refrigerants require nearly 3000 rpm for a loop rotational speed even at a radius of rotation of 0.6 m. It would be difficult to develop a more compact system with a rotational speed less than 1800-2000 rpm. Therefore, a cascade system may be an option to overcome this limitation. Two small centrifugal heat pumps with low-speed fans in series usually achieve a greater combined efficiency thatn one large, high-speed fan that covers the entire pressure range from the evaporator to the condenser.

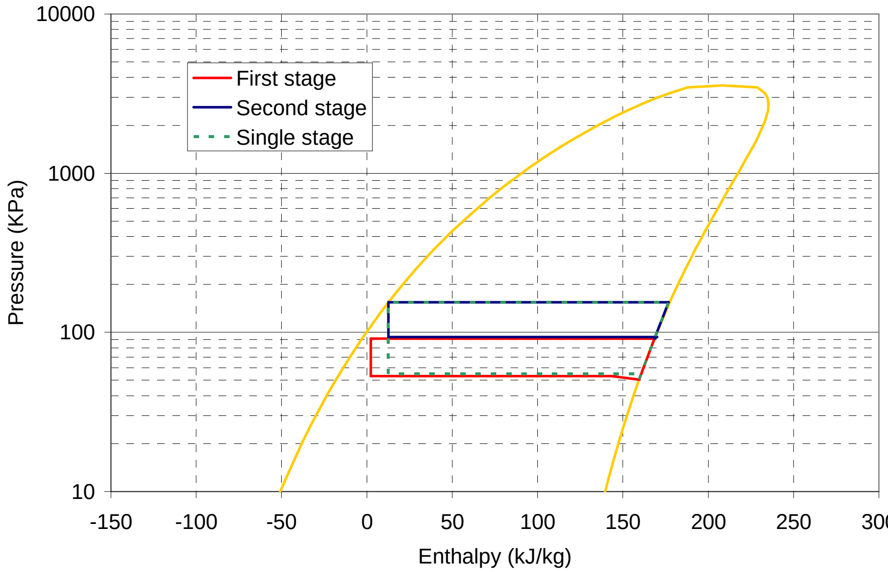

Figure 4.6 P – h diagram for the cascade case; refrigerant is R22.

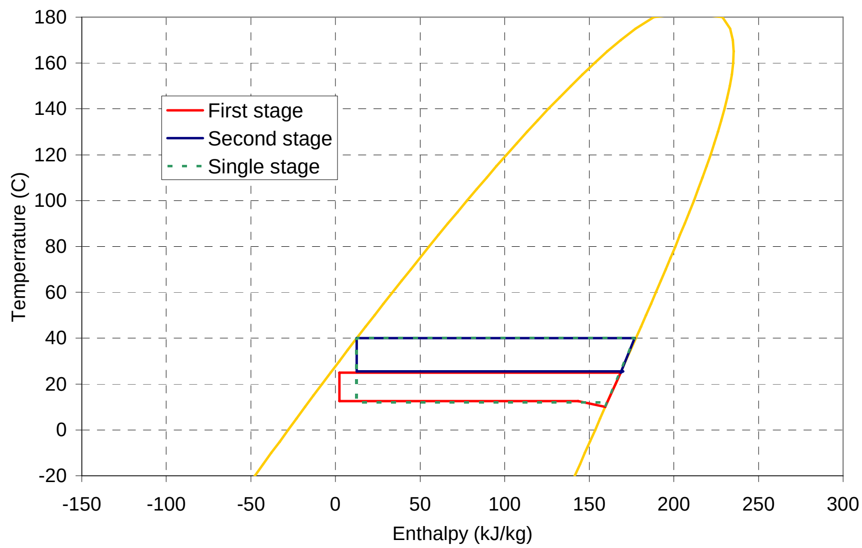

Figure 4.7 T – h diagram for the cascade case; refrigerant is R22.

Figures 4.6 and 4.7 are diagrams of enthalpy-pressure or enthalpy-temperature for an example of multistage case. The working fluid for these plots is R22. The dashed line in the figures indicates the single-stage case. In the multistage case, the pressure ratio of each stage is much less than in a single-stage system and would provide an advantage in reducing the necessary rotational speed. The rotational speed of each stage is only 2060 rpm as opposed to nearly 3000 rpm for a single-stage system. Meanwhile, the COP of a multistage system seems higher than that of a single-stage system. The theoretical analysis indicates that the COP of a multistage system is 4.36 and the COP of a single-stage system is 3.89. This difference is due to the fact that the multistage system leads to the higher capacity than a single-stage system as can be seen particularly in Figure 4.7.

5. TECHNOLOGY ASSESSMENT

The new centrifugal cycle is a simplified and potentially low-cost system, consisting of a loop with integral heat exchangers. The new system also does not need any refrigerant control protection nor does it have a need for refrigerant lubrication. Morover, there should not be a problem with a “wet” gas from the evaporator entering the compressor section of the device. The entire working fluid can participate in both condensing and evaporating process.

The analysis showed the theoretical performance of the novel centrifugal heat pump to be about 40% of Carnot. A comparative analysis of a traditional heat pump (not including heat exchangers) puts the COP at about 7.99 for R22 whereas the Carnot COP is 8.12. This points out that the traditional vapor-compression cycle with fixed heat exchangers is significantly more efficient than the centrifugal heat pump at this point in the study. However, heat exchanger performance is the major factor in the reduction from Carnot to true operating performance, and the centrifugal design in which the condensing/evaporating surfaces become part of the fan blade itself could provide a significant performance advantage.

A wide range of refrigerants have been evaluated, including CO2, CFC, HCFC, and HFCs. None were able to simultaneously improve COP and reduce the needed rotational speed of the refrigerant loop. However, R218 and R227ea could significantly reduce rotational speed with only a slight COP penalty. Their rotational speed is around 2100- 2400 rpm. The further evaluation of the mixing refrigerant based on the analysis of single refrigerant indicated that the mixture consisted of R218/R227EA (50%/50%) could achieve lower rotational speed and a reasonable COP.

We found that a multistage centrifugal system such as a cascade refrigeration system may provide a size and speed advantage, but further work to develop a loop concept that could take advantage of cascading needs to be done prior to the initiation of laboratory studies

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APPENDIX A

NOMENCLATURE

APPENDIX A. NOMENCLATURE

a	:Acceleration (m/s2)
F	:centrifugal force (N)
g	: Gravity (m/s2)
h	:Enthalpy on a unit-mass basis (J/kg)
k	:Polytropic coefficient or specific-heat ratio (c_p/c_v)
$\dot{m}$	:Mass rate (kg/s)
m	:Mass (kg)
M	:Mach number ($U/\sqrt{kRT}$)
p	:Pressure (Pa)
Pr	:Pressure ratio
q	:Heat flux on a unit-mass basis (J/kg)
R	:Peripheral radius (m) or gas constant
T	:Temperature (K)
r	:Radius (m)
u	:Internal energy on a unit-mass basis (J/kg)
U_R	:Peripheral velocity ($U_R = \omega R$) (m/s)
v	:Specific volume (m3/kg)
V_{Tran}	:Translational velocity (m/s)
V_{Rot}	:Rotational velocity (m/s)
w	:Work on a unit-mass basis (J/kg)
ω	:Rotational speed (rpm)

Subscript

i	:Inlet
e	:Exit
expansion	: Expansion work
evap	:Evaporation
cond	:Condensation
R	:Peripheral radius location

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