DISTRICT HEATING SYSTEMS WITH CO_2 AS REFRIGERANT

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ABSTRACT

Carbon dioxide is very effective as the refrigerant for instantaneous hot water heating, because the low incoming water temperature provides the opportunity to achieve low gas cooler outlet temperatures and hence efficient operation. In a district heating system this is not possible because the water return temperature is typically in the range 60°C to 70°C. This paper presents an alternative design for district heating systems which uses carbon dioxide as the primary refrigerant and offers efficiencies above the figures achieved by typical R-134a systems.

1. INTRODUCTION

Transcritical operation presents great opportunities for heat recovery to high temperatures. Whereas in a condenser it is necessary to raise the compressor discharge pressure to achieve high temperatures in a water or air heating system, supercritical carbon dioxide will allow high temperatures at any discharge pressure across the compressor's operating range. The heat recovery is most effective where the temperature range on the process fluid is high; the efficiency of a heat pump taking water from 10°C to 80°C will compare favourably with a traditional system, but if the application is for building heating and the range is only 65°C to 80°C then the performance will compare less favourably. Performance can only be improved for this type of heating system by spreading the compression over several stages and dividing the process stream into several parallel flows, each of which is heated by one stage of compression (Lorentzen, 1994). As the pressure ratio of carbon dioxide is low even for single stage compression this is not a particularly attractive solution.

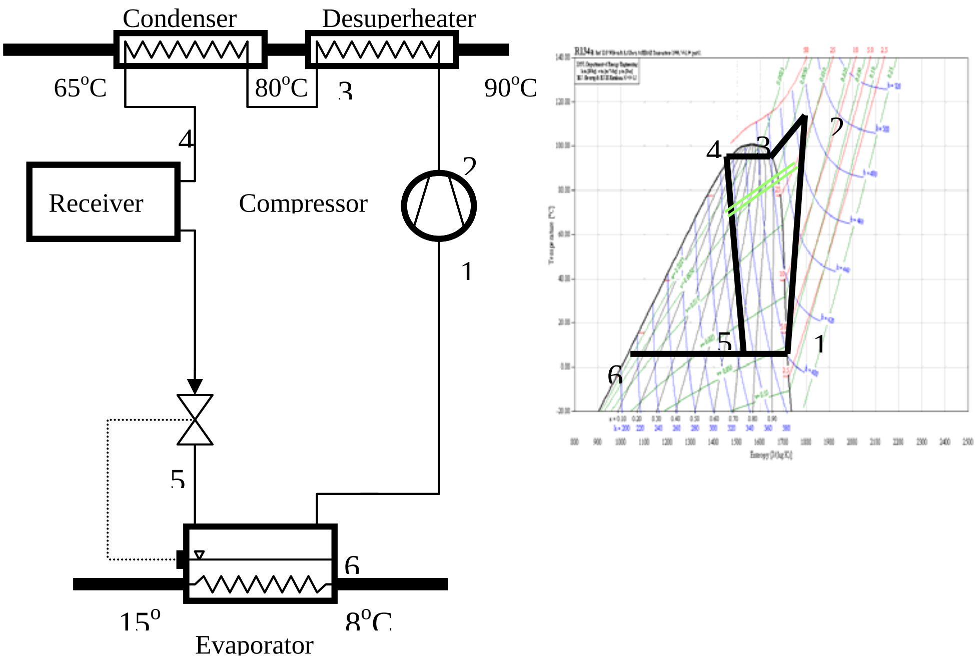

Figure 1 A simple large R-134a heat pump system

Consider the example of a district heating system, raising water in the heating loop from 65°C to 90°C. Traditionally this would have been done using a simple R-12 system condensing well below its critical temperature of 112°C. The critical temperature of R-134a is 102°C, which makes the heating requirement much more difficult. Simple analysis of a basic system evaporating at 5°C and condensing at 95°C (with compressor η_I of 75%), as shown in Fig 1, shows a work input of 62.9 kJ/kg for a heat output of 109.2kJ/kg, a heating CoP of 1.73. The green lines in the figure show the rise in water temperature in the condenser and desuperheater, assuming countercurrent flow. The pinch at point 3 (inlet to the condenser) is evident.

This could be improved by designing a system with an R-134a system using a desuperheater, a condenser and a suction/liquid heat exchanger, as shown in Figure 2.

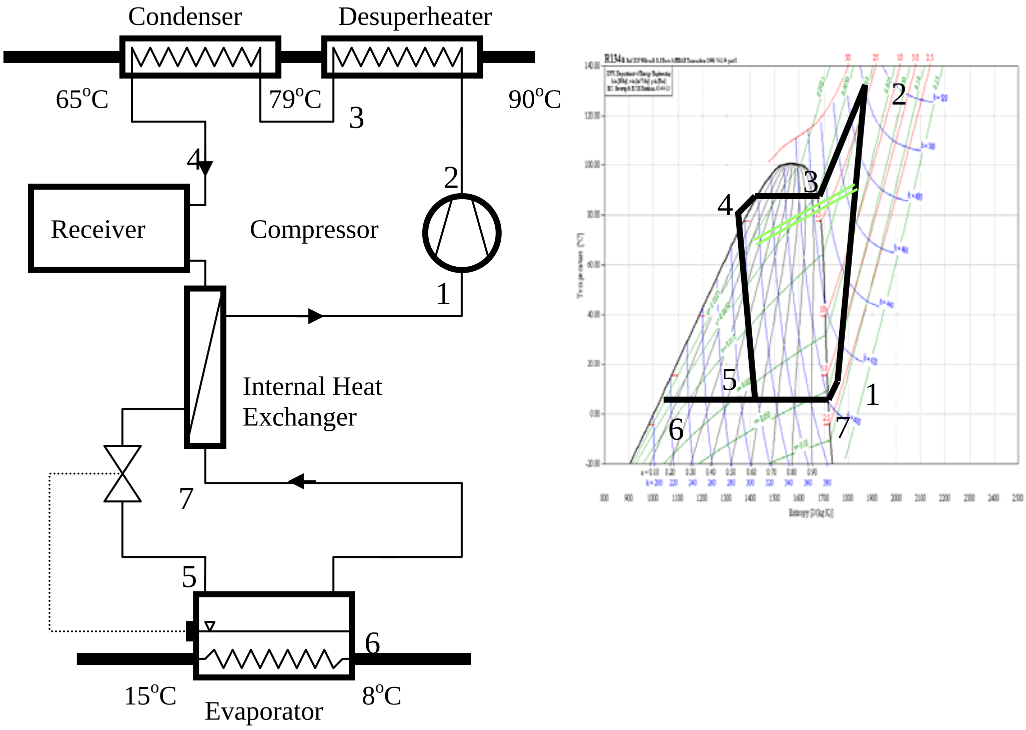

Figure 2 An enhanced R-134a heat pump system

The system would be condensing at say 85°C, and the water would be heated from 65°C to about 80°C in the condenser and from about 80°C to 90°C in the desuperheater. Thus 60% of the heat recovered comes from condensing and 40% from desuperheating. This proportion is only possible if there is a large heat exchanger to subcool the liquid by adding superheat to the compressor suction. The total heat recovered is (500-332)=168kJ/kg. Desuperheating accounts for (500-428)=72kJ/kg which is 43% of the total, so heat rejection by condensing makes up the remaining 57%. This requires a desuperheater to cool R-134a gas from 134°C to 85°C while heating water from 80°C to 90°C. This could be a counterflow plate heat exchanger with a total duty of 4.2MW, operating at a pressure of 29.3 bar abs. The condenser must do 5.8MW cooling while heating water from 65°C to 80°C, condensing R-134a at 85°C. The suction – liquid heat exchanger must transfer 1.74MW cooling R-134a liquid at 29.3 bar abs. from 85°C to 70°C while heating R-134a gas at 3.5 bar abs. from 5°C to 37°C. The water temperatures in figure 2 are again indicated by the green line, and the pinch at point 3 is still the main constraint.

The power input to this system is (500-431)=69kJ/kg, equivalent to 4.13MW, giving a heating coefficient of performance of 2.42. This is an improvement of 40% over the simple system. The variation in CoP_h is shown in Figure 3, which indicates that the conditions of 83 bar abs, 32K suction superheat) are the most favourable of the conditions shown, with a maximum CoP_h of 2.51.

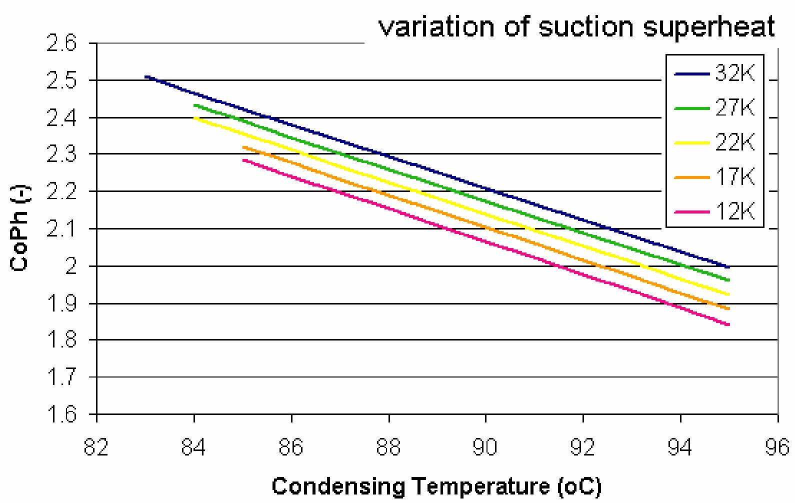

Figure 3 Enhanced performance from an R-134a heat pump

The condition selected gave a compressor discharge temperature of 132°C, which is approaching the limit for this type of equipment, so combinations of higher suction superheat or higher discharge pressures were not considered. It is not possible to drop the condensing temperature lower than the values indicated to the left of Figure 3, because there is insufficient superheat in the discharge gas to raise the water temperature to 90°C.

2. COMBINATION R-744/R-134a SYSTEM

Consider now a combination system, where some of the water heating is provided directly by a carbon dioxide heat pump but the remainder is provided by an R-134a liquid/liquid heat pump which cools the outlet from the transcritical gas cooler. This is shown in Figure 4

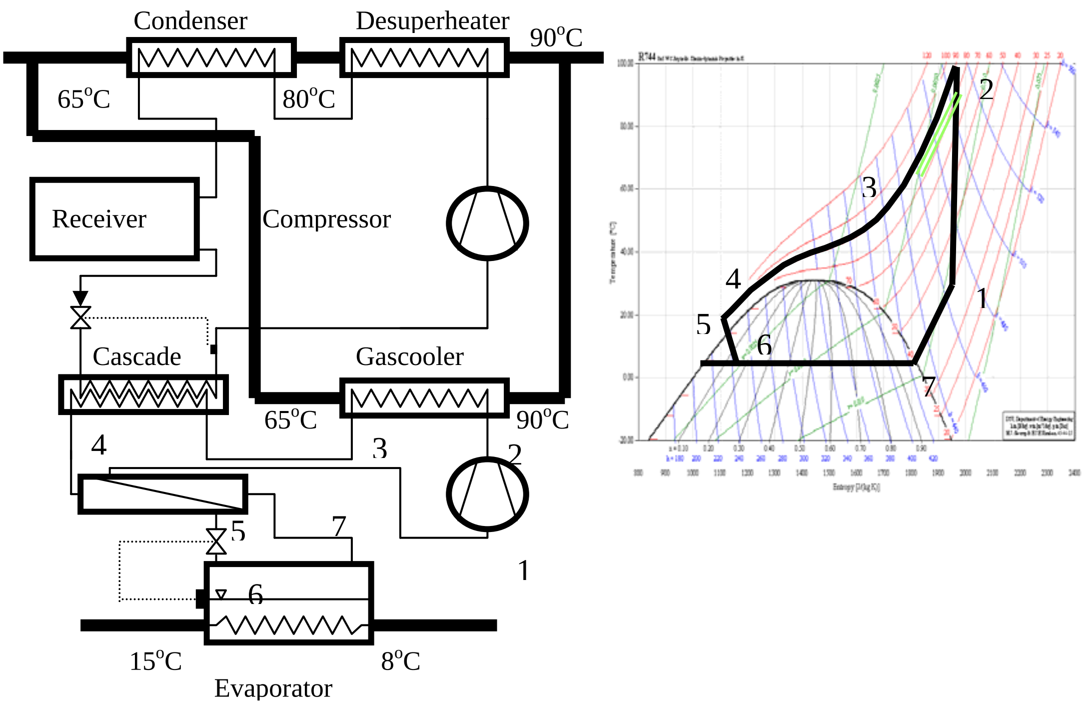

Figure 4 the combination R-134a/Carbon Dioxide Heat Pump, with T-s chart for the carbon dioxide cycle only

The combi system would have a carbon dioxide compressor at 5°C saturated suction, discharging at 94 bar abs. Assuming an isentropic efficiency of 85% gives a discharge condition of h'_2=480kJ/kg and hence t_2=86°C. This must be raised to over 100°C to provide the required water heating (note that the R-134a system had to run at higher temperature to get the ratio of desuperheating to condensing correct) so this system also requires a suction superheater to add 25K to the suction gas, raising h_1 from 427kJ/kg to 468kJ/kg, and hence h'_2 from 480kJ/kg to 517kJ/kg. The carbon dioxide can be cooled from 105°C to 70°C in heat exchange with a portion of the process water. This drops the high side enthalpy from h'_2 of 517kJ/kg to h_3 of 459kJ/kg. Now consider the addition of an R-134a system to chill the carbon dioxide from 70°C to a lower temperature. If the direct heat rejection from the carbon dioxide is 31% of the total heat rejected by the carbon dioxide then a further 126kJ/kg can be removed, lowering the enthalpy to 328kJ/kg, at a temperature of 40°C. Finally the suction superheat is provided by further cooling the carbon dioxide, reducing the enthalpy to 288kJ/kg at t=33.5°C. The heat pump coefficient of performance for the carbon dioxide loop is:

$$CoP_{hC} = (\dot{Q}_{CO2} + \dot{W}_{CO2}) / \dot{W}_{CO2} = (517 - 328)/(517 - 459) = 3.80 \qquad (1)$$

The R-134a system also requires power input. In this case the carbon dioxide is cooled from 70°C to 40°C by evaporating R-134a at 35°C and condensing at 85°C, with some desuperheating. Thus the heat pump coefficient of performance for the R-134a loop is:

$$CoP_{hR} = (\dot{Q}_{R134a} + \dot{W}_{R134a}) / \dot{W}_{R134a} = (445 - 332)/(445 - 417) = 4.10 \qquad (2)$$

The total heat supplied to the process is given by the summation:

$$\dot{Q}_{hp} = \dot{Q}_{CO2} + \dot{W}_{CO2} + \dot{W}_{R134a} \qquad (3)$$

where $0.69(\dot{Q}_{CO2} + \dot{W}_{CO2}) = (CoP_{hR} - 1)\dot{W}_{R134a}$ (4)

and $\dot{Q}_{CO2} = (CoP_{hC} - 1)\dot{W}_{CO2}$ (5)

Therefore $\dot{Q}_{hp} = \left(\frac{CoP_{hR} - 1}{0.69}\right)\dot{W}_{R134a} + \dot{W}_{R134a}$ (6)

and hence $\dot{W}_{R134a} = \frac{0.69\dot{Q}_{hp}}{(CoP_{hR} - 0.3)}$ (7)

Hence $\dot{W}_{R134a} = \frac{6{,}900}{3.80}$ =1820kW (8)

and $\dot{W}_{CO2} = \frac{\dot{Q}_{hp} - \dot{W}_{R134a}}{CoP_{hC}}$ so $\dot{W}_{CO2} = \frac{8{,}180}{3.80}$ =2138kW (9)

Therefore the power input is 1.82MW to the R-134a circuit and 2.15MW to the carbon dioxide circuit, giving a total power of 3.97MW and CoP_h of 2.52 This represents an improvement of 46% over the simple R-134a circuit and an improvement of 4% over the enhanced R-134a circuit.

The power input to the CO2 circuit suggests a massflow of 2154/50=43kg/s. The heat extracted from the cold water is (427-288)=139kJ/kg, which for this mass flow is 139*43=6.03MW.

The mass flow in the R-134a circuit of the combi system is 1820/(445-417)=66.3kg/s. In the enhanced R-134a system it is 4360/(502-431)=61.3kg/s

However in the enhanced system case the swept volume of the R-134a compressor required, assuming volumetric efficiency of 80% and specific volume of 0.069m^3/kg is

$$V = \frac{v\dot{m}}{0.8} = \frac{0.068x61.3}{0.8} = 18\ 658\text{m}^3/\text{h} \tag{10}$$

In the combination circuit the R-134a specific volume is 0.023m^3/kg, so V= 6 830m^3/h. The carbon dioxide compressor, again assuming volumetric efficiency of 0.8, requires a swept volume of 0.011x43.1/0.8 = 2 185m^3/h. The heat rejected directly from the carbon dioxide is 2.54MW, so the heat transferred to the R-134a loop is 5.64MW and the heat rejected from the R-134a loop is 7.46MW.

3. COMPARISON OF EQUIPMENT REQUIREMENTS AND PERFORMANCE

A summary of the equipment required, with its relative size is given in Table 1. This table uses assumed heat transfer coefficients to assess the relative size of the heat exchangers with slightly higher values assumed for the carbon dioxide performance in the evaporator, as indicated by Stoecker (2000). In this case the assumed values are 1000 W/m^{2}K for the R-134a evaporator and 1500W/m^{2}K for the carbon dioxide one . Similar differences are applied for the condenser, suction-liquid heat exchanger and gas cooler. The motor sizing assumes a tolerance of 5% above the design shaft power input to the compressor. The type of compressor and type of heat exchanger are not specified. It is probable that centrifugal machines would be used for this size of plant on R-134a, and it is possible that a similar machine could handle the carbon dioxide requirement very economically. The heat exchanger types are not considered, but the requirements could be met by plate and frame heat exchangers for the R-134a and by printed circuit heat exchangers for the carbon dioxide. These would have the further advantage that the heat recovery gas cooler, the R-134a evaporator and the aftercooler could all be included in a single, multi-path PCHE suitable for the operating pressure of 100 bar.

Component	**Simple R-134a**	**Enhanced R-134a**	**Combination CO2/R-134a**
Main compressor	24 058m^3/h	18 658m^3/h	1 860m^3/h
Aux. compressor	-----	-----	7 450m^3/h
Main drive motor	6050kW	4200kW	2050kW
Aux drive motor	-----	-----	2050kW
Evaporator	554m^2	777m^2	475m^2
SLHX	-----	11.4m^2	8.2m^2
Desuperheater	-----	53m^2	34m^2
Condenser	238m^2	185m^2	-----
Gas cooler	-----	-----	33m^2
CoP_h	1.8	2.51	2.57
Energy cost	£1 111k	£797k	£778k

Note: Energy cost is based on running 4 000 hours per year and 5p/kWh

Table 1 Heat Pump system comparison

It is evident that the simple R-134a heat pump system pays a penalty in capital and operating cost. The enhanced R-134a system has a smaller compressor, but the evaporator is 40% larger. The carbon dioxide combination system has the added complexity of two compressors, but they are much smaller and the drive motors are significantly smaller. The combination system's evaporator is the smallest of the three, and the other heat exchangers, although adding complexity, are significantly smaller.

The performance of the carbon dioxide transcritical system prompted an investigation of an all-R-134a system running at transcritical conditions. It was found that such a system is feasible, but would be more expensive and less efficient than the combi system. Unlike carbon dioxide, the R-134a system did not offer increased efficiency as the discharge pressure was raised above the critical point: the maximum coefficient of heating performance was 2.4 at a discharge pressure of 41 bar abs, falling to 2.29 at a discharge pressure of 54 bar abs. The compressor swept volume was only slightly smaller than the enhanced R-134a system, at about 16 000m^3/h compared with 18,560 m^3/h, but the compressor would need to be designed for at least 50 bar abs allowable pressure and would operate at a pressure ratio of at least 11.7 The discharge temperature at these conditions is 125°C, compared with 115°C for the basic system, 132°C for the enhanced system and 95°C for the R-134a circuit in the combi-system.

4. SYSTEM MODELLING

To check the optimum operating conditions for the combination system heat pump a model was created in Excel, using Refprop 7.0 values for R-134a and carbon dioxide. The model does not simulate the operation of a given set of components, rather it assesses the optimum design point for a given set of operating circumstances and then indicates what size of heat exchangers would need to be selected to achieve that level of performance. The model confirmed that the values used for the simple and enhanced R-134a heat pumps were correct, and it explored the performance of the combination circuit under conditions of varying the carbon dioxide compressor discharge pressure, varying the suction superheat and varying the proportion of heat rejection from the carbon dioxide system which went to the hot water system. There is a limit to the amount of suction superheating that can be achieved; an approach temperature difference of 5K in the aftercooler, which requires to be true counterflow, was used. With a discharge pressure of 94 bar abs and suction superheat of 25K the maximum CoP_h was found to be 2.56 when the proportion of carbon dioxide heat rejected to the heated water was reduced to 28% from the original estimate of 31%. This is shown in Figure 5

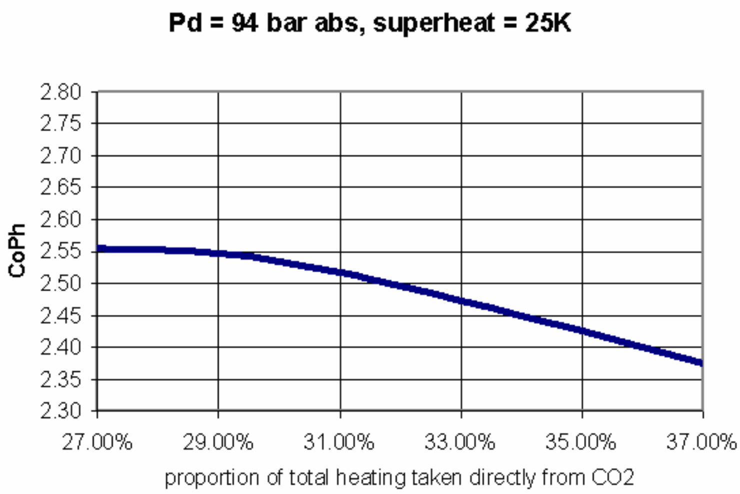

Figure 5 the effect of varying the proportion of carbon dioxide waste heat

With the proportion fixed at the benchmark value of 31%, and the discharge pressure held at 94 bar abs the maximum CoP_h was found to be 2.57 at higher values of suction superheat, as shown in Figure 6 The maximum superheat used in the model was 28K because this was the maximum that could be achieved in the internal heat exchanger. This gives a carbon dioxide discharge temperature of 110^{o}C which is moderate in comparison with ammonia reciprocating compressors.

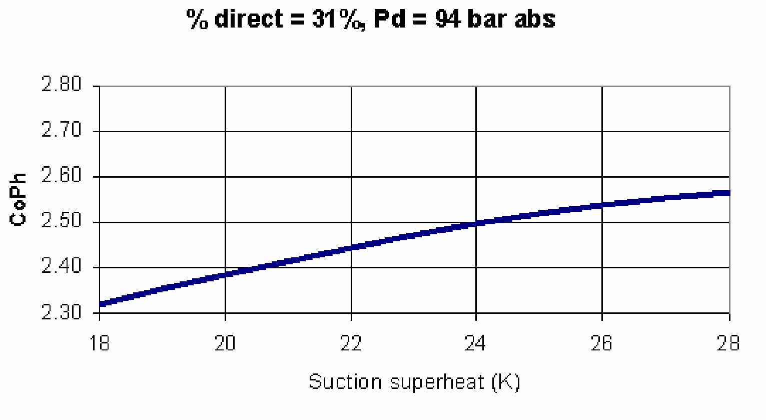

Figure 6 The effect of increasing suction superheat (shown in K)

In the third test the suction superheat was held at 25K and the proportion of heat rejected directly was held at 31%. The discharge pressure was varied from 85 bar abs to 95 bar abs. The outcome is shown in Figure 7, with a CoP_h of 2.54 at 95 bar abs. Figures 6 and 7 do not show the optimum value because variation in the control parameters was restricted by the operating limits.

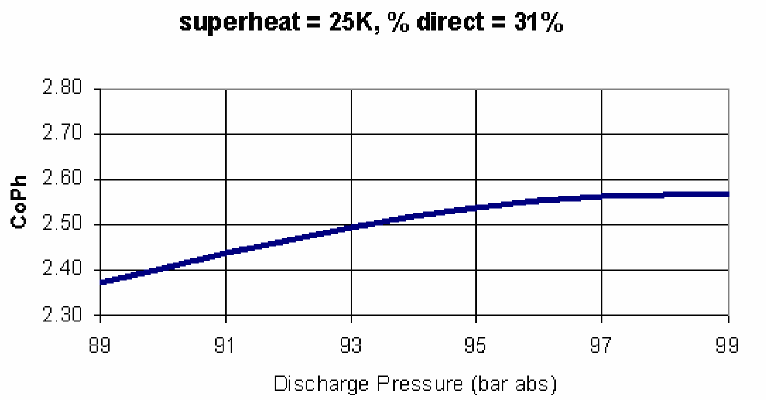

Figure 7 Variation of discharge pressure (in bar abs)

To determine the optimal coefficient of performance a wider range of discharge pressures was investigated, allowing the proportion of heat rejected directly and the suction superheat to float at each pressure, within the limits set by the aftercooler/suction superheater performance. This showed that the performance was optimal over all parameters at a discharge pressure ranging from 94 to 102 bar abs, with suction superheat of about 27K and a direct heat rejection proportion of 29%-33%. The highest CoP_h achieved was 2.57, which is 2.4% higher than the maximum R-134a enhanced figure.

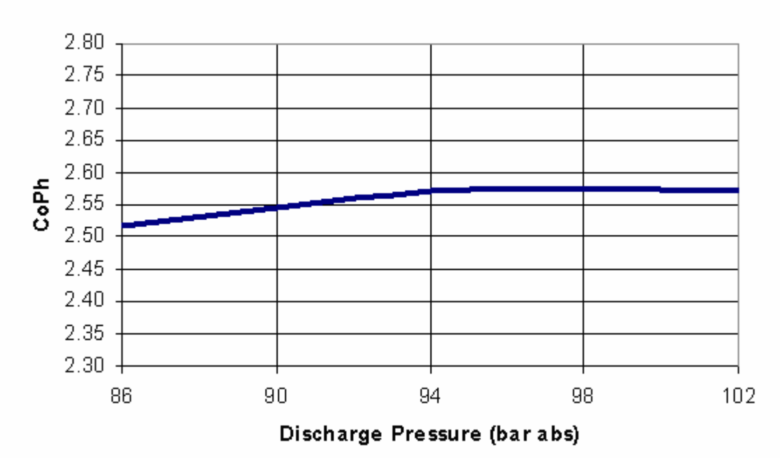

Figure 8 Optimal values for combi heat pump performance

It is interesting to note that the optimal CoP_h characteristic in Figure 8 is very flat across a wide range of discharge pressures. This is evidence of a system that will not require tight pressure control to keep it operating close to its peak performance. The optimal carbon dioxide suction superheat across this wide temperature range was consistently 26K or 27K, and the maximum discharge temperature was in the range 99 ^oC to 115 ^oC. The discharge temperature of the R-134a system was consistently 94 ^oC for all conditions. Further small improvements to the overall CoP_h could be achieved by enhancing the R-134a system in the same way as described above, but at the cost of greater complexity.

5. CONCLUSIONS

Carbon dioxide can be used for district heating heat pumps provided that the gascooler outlet temperature is as low as possible. In instantaneous hot water heaters this is easy as the incoming mains cold water is typically at about 10^oC. However for a district heating system a more complex system is required to achieve this, perhaps by using a secondary heat pump in combination with the main carbon dioxide gas cooler. With this type of system it is possible to match or even exceed the performance levels achieved in traditional heat pump systems. Standard and near-standard components can be used, but care must always be taken to ensure that the unusual transcritical properties are considered in the design arrangement. A further advantage of this system is that the discharge temperatures are relatively low, suggesting superior exergetic performance compared with simpler designs. High temperatures are a particular concern as air and water may be heated to levels not encountered in traditional refrigeration systems.

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