A heat pump for simultaneous refrigeration and water heating

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A novel heat pump has recently been proposed which provides both refrigeration and water heating using a transcritical CO_2 cycle. Applications in the food processing industry have been identified where the transcritical CO_2 heat pump could provide large energy cost savings with reduced environmental impact compared with conventional systems. In this paper, the energy efficiency of the heat pump is quantified and the potential economic and environmental benefits are discussed.

***Keywords:** heat pump – refrigeration - water heating - CO_2 - transcritical cycle*

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1. Introduction

Many food processing industries in New Zealand require both refrigeration and water heating. Refrigeration at temperatures below 10°C is employed for product preservation, while hot water at temperatures up to 90°C is employed for cleaning, sterilisation or process heating. The NZ Energy End-Use Database[1] estimates that the New Zealand food industry consumes approximately 4200 TJ/yr and 2500 TJ/yr of energy for water heating and refrigeration respectively. It is possible to conserve a large fraction of this energy.

It is common for the refrigeration and water heating systems to be separate and unconnected, each consuming purchased energy. This approach wastes considerable energy, contributing to the depletion of fossil fuel reserves and the release of greenhouse gases. In addition, many refrigeration systems employ ozone depleting refrigerants. There are strong international moves to use naturally occurring and ecologically safe working fluids rather than man-made chemicals, in order to minimise the impact on the environment. Alternative technologies are required to limit the economic cost and environmental impact of these applications. Two technologies which could significantly reduce energy consumption are waste heat recovery and heat pumps.

Waste heat recovery is feasible when an existing process stream is available at sufficiently high temperature to heat water in a heat exchanger. One example is the desuperheating of refrigerant in a refrigeration system. In this application, heat is obtained by cooling hot refrigerant exiting the compressor down to its dew point. Unfortunately, the available heat is usually less than 15% of the total heat which must be rejected from the refrigeration system. At most sites, the demand for hot water can not be met by such waste heat recovery opportunities.

Heat pumps absorb heat from the environment at low temperature and supply it at high temperature for process heating. Conventional heat pumps employ fluorocarbons or ammonia as the working fluid, in a cycle closely resembling a refrigeration cycle. Commercially available single stage heat pumps are limited to delivering most of the heat at temperatures below 75°C and the evaporation temperature is too high to be of value for refrigeration.

In applications where both refrigeration and water heating are required, it is technically feasible to use the refrigeration system as both a refrigerator and heat pump, by increasing the pressure lift across the compressor or by employing a two stage cycle. Loss of refrigeration capacity and energy efficiency often makes this option unattractive. More widespread application of heat pump technology will only be possible if higher heat rejection temperatures can be achieved with improved energy efficiency.

A combined refrigeration/heat pump system that could overcome these difficulties has recently been proposed.[2] The system utilises the heat rejected from the refrigeration process to heat water. However, in contrast to conventional technology, high heat rejection temperatures and high efficiencies are maintained by employing CO_2 as the refrigerant in a transcritical refrigeration cycle. CO_2 offers additional advantages because it is inexpensive, non-flammable and environmentally benign.

The object of this paper is to discuss the potential applications and benefits of this technology in the New Zealand food processing industry, and quantify the theoretical performance of the transcritical heat pump system.

2. Transcritical CO_2 heat pump description

A simple, single stage transcritical CO_2 heat pump cycle is illustrated in Figure 1. Low-pressure CO_2 vapour exiting the evaporator, is compressed to a pressure higher than the critical pressure of CO_2 (73.8 bar, 31°C). The resulting high pressure, high temperature CO_2 vapour is then cooled by counter-current heat exchange with water, both to reject heat from the system and to heat water to the desired temperature. As the pressure is above the critical pressure, the CO_2 can not condense, and sensible heat is removed rather than latent heat. The cool high pressure CO_2 is then expanded to a pressure below the critical pressure, where the CO_2 turns to a saturated vapour-liquid mixture. The refrigerant can then provide useful cooling by boiling in the evaporator.

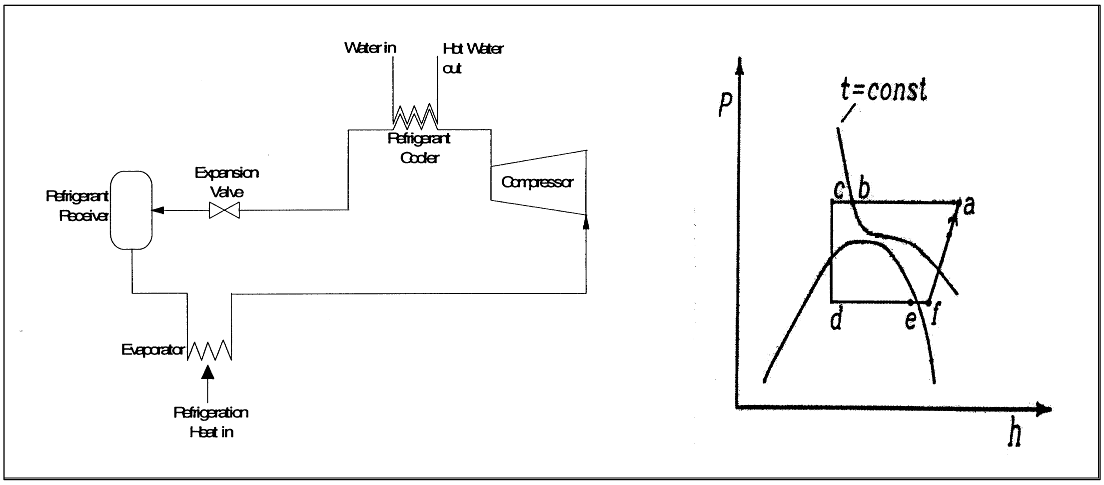

FIGURE 1: Flow diagram and corresponding pressure enthalpy diagram of the transcritical CO_2 heat pump cycle.

The principal difference between this cycle and that of a conventional cycle is the high-pressure cooling step. In a conventional cycle, most of the heat is removed to condense the refrigerant at constant (near ambient) temperature. In the transcritical cycle, only sensible heat is removed to lower the temperature of the refrigerant. As a consequence, a larger fraction of the heat is removed at a higher temperature in the transcritical cycle. This difference between the refrigerant condenser in the conventional cycle, and the refrigerant cooler in the transcritical cycle, is illustrated in Figure 2. The temperature profile of the supercritical refrigerant cooler is ideal for counter-current process heating.

3. Preliminary economic analysis

A preliminary economic analysis was performed to compare an integrated transcritical CO_2 heat pump (providing both water chilling and water heating) with a system consisting of a separate water chiller and gas hot water boiler. In both cases, 400 kW of water chilling was provided at an evaporator temperature of –5°C, and 600 kW of water heating was provided to heat water from 15°C to 90°C. This was considered typical of medium sized heating and cooling loads.

Supplier quotations indicated that the conventional refrigeration package and gas boiler would cost approximately $190,000 and $20,000 respectively, giving a total system cost of $210,000.

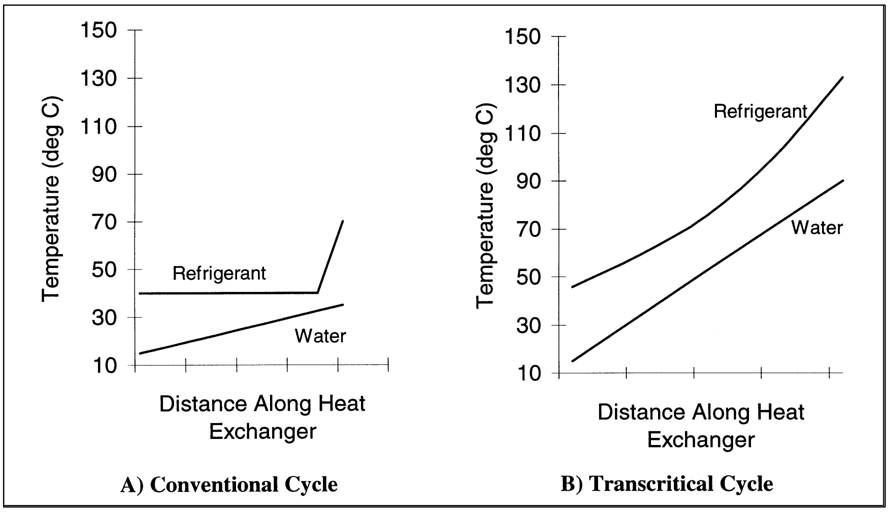

FIGURE 2: Comparison of typical refrigerant and coolant temperature profiles in a conventional cycle condenser and a transcritical refrigerant cooler.

The annual energy consumption, energy cost and greenhouse CO_2 emissions were calculated for each of the two options using the assumptions summarised in Appendix 1. The results of this analysis are presented in Table 1. They highlight the potential to reduce the cost of supplied energy by approximately 30% and total CO_2 emissions by approximately 50% using the new technology.

TABLE 1: Annual energy costs and greenhouse gas emissions for the transcritical CO_2 heat pump and the conventional refrigeration/hot water boiler system.

System	***Electricity consumption (GJ)***	***Gas consumption (GJ)***	***Total energy cost ($)***	***Total CO_2 emissions (tonne)***
Separate refrigeration plant and gas boiler	1598	12 300	129,600	713
Transcritical CO_2 heat pump	3197	-	86,400	340

The reduction in energy costs can be used to justify an increase in the capital cost of the transcritical CO_2 heat pump over the conventional system. The justifiable capital cost of the CO_2 heat pump is illustrated in Figure 3 as a function of the desired investment internal rate of return. The cost of manufacturing the CO_2 heat pump is not yet known, but preliminary costings indicate that it should be well below that required to achieve an acceptable internal rate of return for most food processing companies.

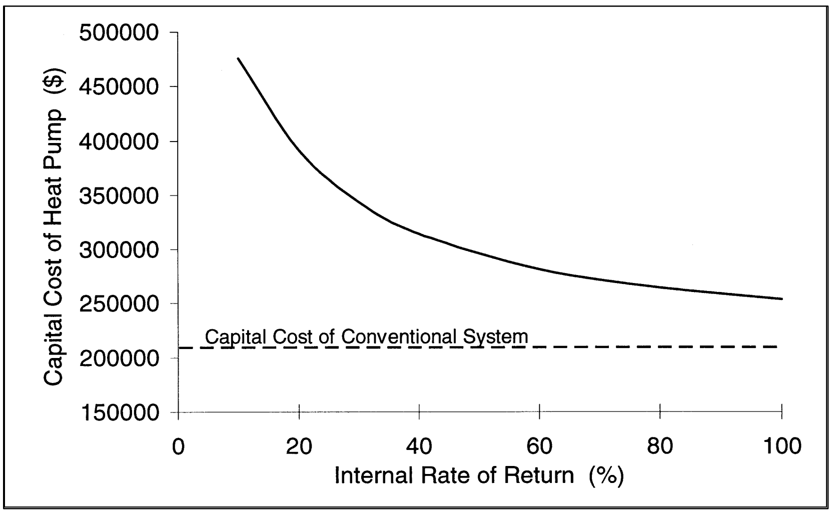

FIGURE 3: Influence of desired internal rate of return on the allowable capital cost for the transcritical CO_2 heat pump

The analysis suggests that the transcritical CO_2 heat pump offers great potential for reducing the economic cost and environmental impact of providing refrigeration and water heating. A joint venture between Massey University, ECNZ, and Flotech has been established to construct a prototype system and to demonstrate the likely benefits and applications of this technology for the New Zealand food processing industry.

4. Optimising the theoretical performance of the CO_2 heat pump

4.1 Methodology

The thermodynamic performance of the transcritical CO_2 heat pump was investigated by computer simulation. Calculations were performed for the application discussed above (-5°C evaporation temperature and heating water from 15°C to 90°C). In addition, the performance of the cycle was evaluated at a hot water outlet temperature of 65°C rather than 90°C.

The computer simulations involved detailed mass and energy balances around the cycle using the Peng-Robinson equation of state for CO_2 thermodynamic property estimation. Table 2 lists the performance constraints employed throughout the study.

TABLE 2: Transcritical CO_2 heat pump computer model constraints

High pressure cooler pressure drop	5 kPa
Evaporator pressure drop	8 kPa
Refrigerant exit condition from the evaporator	Saturated vapour
Compressor suction swept volume	21.5 m^3/s
Compressor isentropic efficiency	72.5%

For a given compressor discharge pressure, the duty of the transcritical refrigerant cooler was varied. For each value of the refrigerant cooler duty, the calculated refrigeration duty and compressor power were recorded. The calculated size of the refrigerant cooler (heat exchange surface area multiplied by the overall heat transfer coefficient) was also recorded. Simulations were repeated over a range of compressor discharge pressures.

Compressor size was held constant by maintaining a constant compressor swept volume in all simulations. This was done in a crude attempt to minimise variations in compressor cost. By eliminating variations in compressor cost, the relative capital cost of the overall cycle becomes dependent on the size of the refrigerant cooler. This enables the size of the refrigerant cooler to be used for comparing the relative capital cost of each cycle.

The cycle coefficient of performance (COP) for heating was used to compare cycle thermodynamic performance. As heat losses will be small, and all of the heat rejected from the cycle is used to heat water, the refrigeration COP can be obtained from the heating COP using:

$$COP_{refrig} = COP_{heating} - 1$$

4.2 Results and discussion

4.2.1 Single stage cycle, refrigeration at –5°C and heating water from 15°C to 90°C

Simulation results are presented in Figures 4 and 5 for a cycle providing refrigeration at –5°C and heating water from 15°C to 90°C. For a given refrigerant cooler size, the refrigeration capacity increased with increasing compressor discharge pressure. For compressor discharge pressures below 130 bar, increased capacity was accompanied by an improvement in COP. Above 130 bar, the COP did not vary significantly with increasing discharge pressure.

The observed poor thermodynamic performance at low compressor discharge pressures can be explained by the location of the minimum temperature difference (pinch point) between the refrigerant (hot) and the water (cold), in the refrigerant cooler heat exchanger. The location of this pinch point was found to vary, depending on the discharge pressure from the compressor. At a discharge pressure below 130 bar, the pinch point was predicted to be in the middle of the refrigerant cooler, i.e. the shapes of the temperature profiles in the refrigerant cooler were similar to that illustrated in Figure 2b. Under these conditions, poor performance is obtained because the CO_2 vapour can not be cooled below approximately 40°C. In contrast, for discharge pressures above 130 bar, the pinch point was found at the cold end of the refrigerant cooler. At high pressures, the required size of the refrigerant cooler does not vary dramatically for a given COP.

The optimum design of the CO_2 heat pump will obviously involve a trade off between capital and operating cost. Examination of Figures 4 and 5 suggests that the optimum design for a single stage cycle would probably have a compressor discharge pressure between 130 and 150 bar with sufficient refrigerant cooler heat transfer area to give a heating COP slightly in excess of 3.

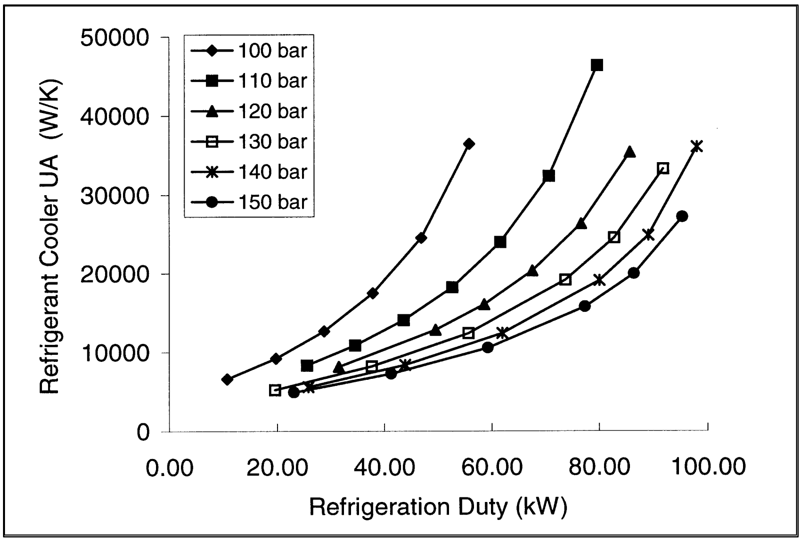

FIGURE 4: Required refrigerant cooler size as a function of refrigeration duty and compressor discharge duty.

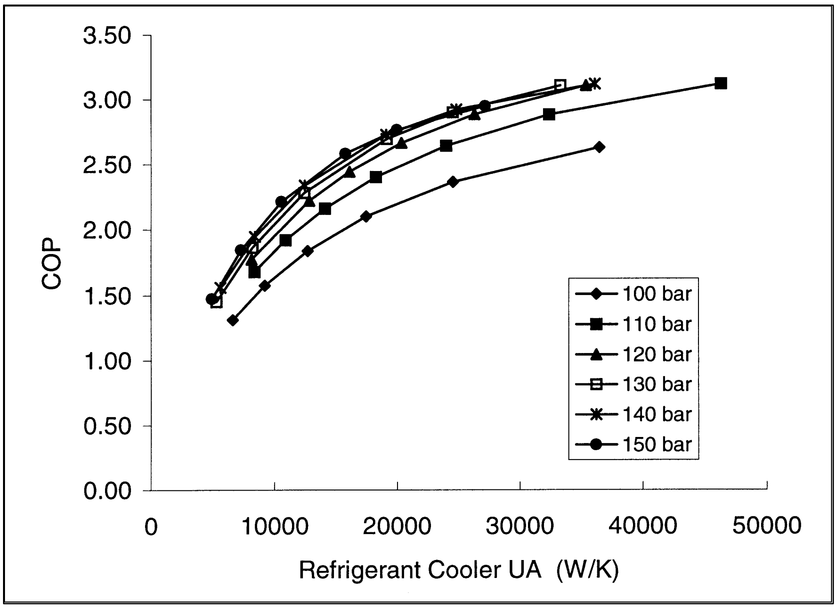

FIGURE 5: Heating COP as a function of refrigerant cooler size and compressor discharge pressure.

4.2.2 Single stage cycle with compressor suction preheater

A number of authors [2,3] have suggested improving the performance of the simple single-stage cycle by inserting a preheater in the compressor suction. The preheater also acts as a precooler for the refrigerant, before it is expanded into the evaporator. The modified cycle is illustrated in Figure 6. One of the benefits of increasing the compressor suction temperature with the preheater, is an increase in the compressor discharge temperature. This can be used to reduce the effect of the internal pinch in the refrigerant cooler.

Simulations were performed on this modified cycle with the compressor discharge pressure set at 130 bar. To ensure a consistent comparison when increasing compressor suction temperature, the vapour volume passing through the compressor was held constant by reducing the mass flow rate of CO_2. The required size of the suction preheater and refrigerant cooler were added to enable comparison with the required size of the original cycle refrigerant cooler.

The cycle employing the compressor suction preheater is compared with the original cycle in Figures 7 and 8. Increasing the compressor suction temperature, with the suction preheater, gave only a small increase in refrigeration capacity and COP. A much greater improvement in refrigeration capacity would be obtained by increasing the compressor discharge pressure without a significant change in COP.

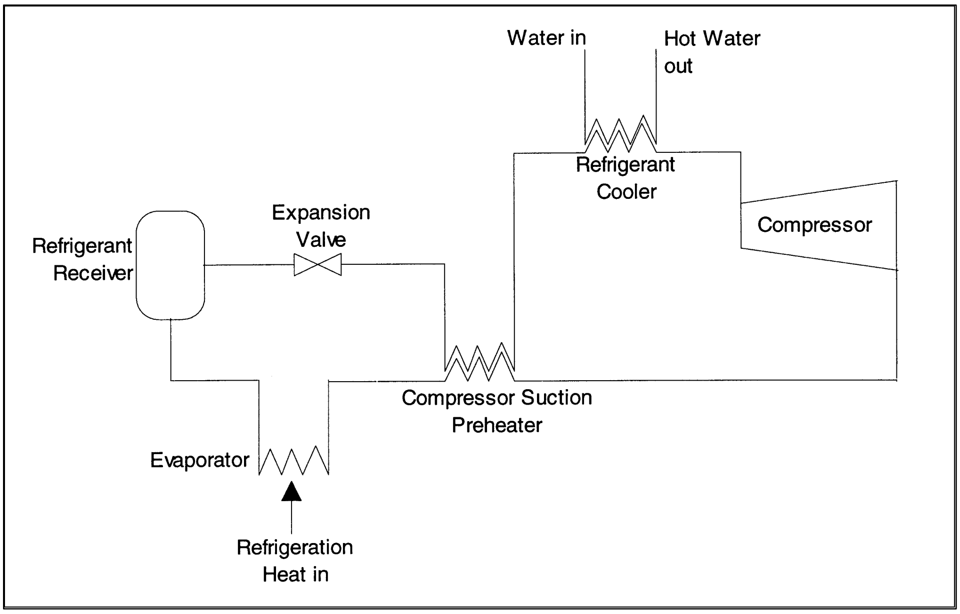

FIGURE 6: Schematic of the transcritical heat pump with compressor suction preheater

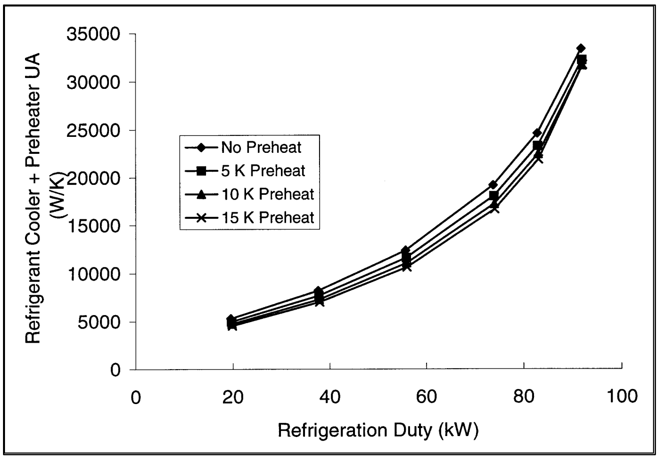

FIGURE 7: Required heat exchange size as a function of refrigerant duty and temperature rise in the compression suction preheater.

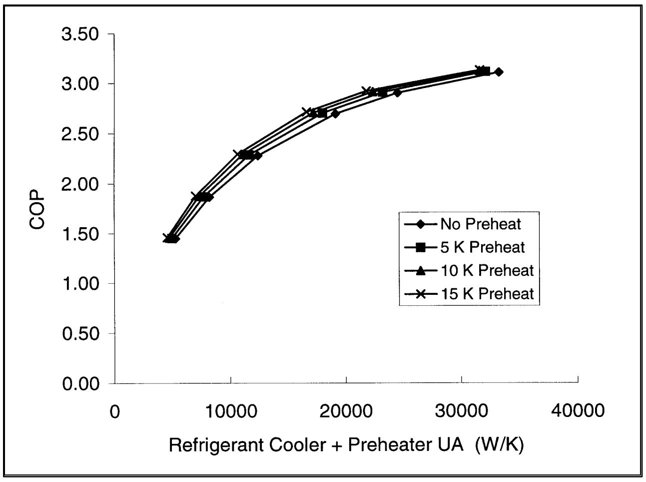

FIGURE 8: Heating COP as a function of refrigerant cooler size and temperature rise in the compressor suction preheater.

The compressor suction preheater does not appear to be justified on the grounds of improved COP when compressor discharge pressures are well above the critical pressure. Possibly, the performance enhancement reported previously was obtained when operating at low compressor discharge pressures, where performance is limited by the internal pinch in the refrigerant cooler. The small cost of including the suction preheater may be justified by the flexibility of operating over a wider range of compressor discharge pressures.

4.2.3 Single stage cycle, refrigeration at –5°C and heating water from 15°C to 65°C
The influence of reducing the hot water exit temperature from 90°C to 65°C is illustrated in Figures 9 and 10. To enable comparison, the compressor discharge pressure was set to 130 bar in both cases.

As expected, lowering the hot water exit temperature provided a significant improvement in thermodynamic efficiency and refrigeration capacity. For a given heat exchanger size the COP and refrigeration capacity was increased by approximately 10% and 15% respectively.

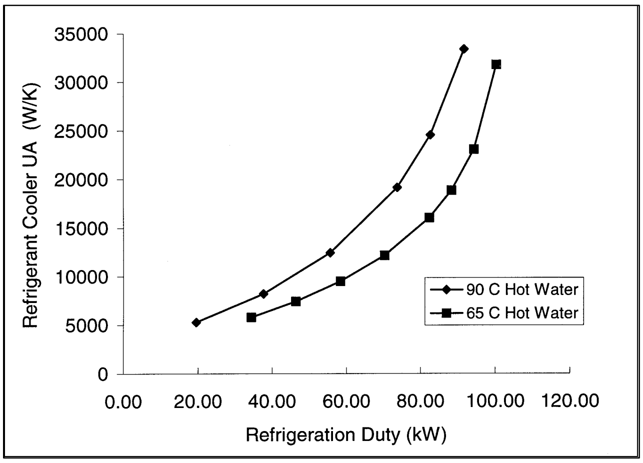

FIGURE 9: Comparison of refrigerant cooler size as a function of refrigerant duty when heating water to either 90°C or 65°C.

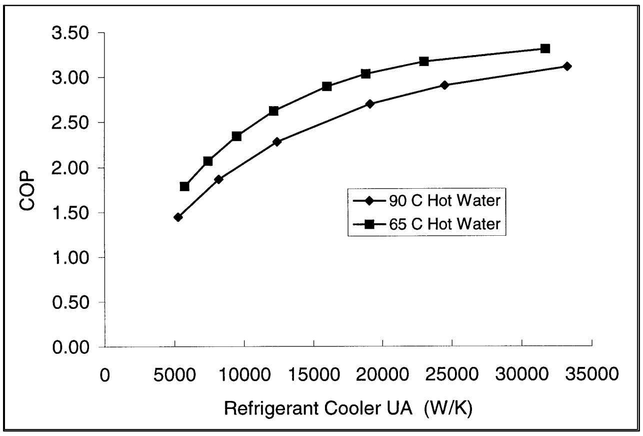

FIGURE 10: Comparison of heating COP as a function of refrigerant cooler size when heating water to either 90°C or 65°C.

5. Conclusions

A transcritical heat pump cycle utilising the natural refrigerant CO_2 as the working fluid appears to be ideal for simultaneous refrigeration and water heating. It offers the potential for significantly reduced energy costs and greenhouse gas emissions.

Heat pump performance was modelled on a computer for a hypothetical meat plant application. An economic optimum heating COP in excess of 3.0 was obtained with a compressor discharge pressure between 130 and 150 bar. At lower discharge pressures, cycle performance was shown to be limited by an internal pinch in the refrigerant cooler. The total energy cost was predicted to reduce by 33% and CO_2 emissions by 52%, compared with a conventional refrigeration system and separate gas hot water boiler system.

Capacity and COP were shown to increase by 15% and 10% respectively when the hot water exit temperature was reduced from 90°C to 65°C. The use of a compressor suction preheater was found to make only minor improvements to system performance.

6. References

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7. Appendix 1

Basis for the economic analysis of the transcritical CO_2 heat pump

Equipment life	10 years
Plant operating time	4000 hours/yr
Electricity price	\$0.097 /kWh
Electric motor efficiency	90%
Gas price	\$7 /GJ
Gas boiler efficiency	70%
Conventional system refrigeration COP	4.0
Transcritical CO_2 heat pump refrigeration COP	2.0
Average CO_2 emissions for electricity[4] (for the initial 100 kW of electricity required by both systems)	140 t CO_2/GWh
Marginal CO_2 emissions for electricity[4] (for extra 100 kW of electricity required by the transcritical heat pump)	624 t CO_2/GWh
Average CO_2 emissions for gas[4]	52.7 t CO_2/TJ