Efficient Process Integration and Cooling & Heating Energy Performance of Supercritical CO_2 Heat Pumps

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Abstract: Low-grade heat losses in the form of liquid effluents at temperatures of up to 45°C represent more than 24% of total heat losses in the Canadian manufacturing industry. On the other hand, process or domestic hot water up to 85°C is simultaneously or intermittently required in many industrial plants. By recovering low-grade heat from process fluids or waste liquid or gas effluents, incoming fresh or recycled water can be heated up to relatively high-temperatures by using supercritical CO_2 heat pumps. This paper presents the process integration procedure and the main energy performance achieved with two supercritical CO_2 heat pumps. The first one has been implemented in a Canadian dairy plant to provide process cooling and heating simultaneously. The second one has been laboratory tested over an extensive range of waste heat temperatures to heat cold city water directly. Both the cooling and heating thermodynamic parameters as well as the overall energy performance achieved are presented and briefly discussed.

Key Words: supercritical CO_2 heat pumps, process integration, industrial heat recovery, energy efficiency

1 INTRODUCTION

Most Canadian manufacturing industries release, via waste heat streams such as combustion gases, hot air, steam, process gases and liquid effluents (Stricker & Associates, 2006), more than 50% of their energy consumption into the environment which represents about 48% of the total country annual energy input (National Energy Board, 2008). On the other hand, process or domestic hot water up to 85°C is simultaneously or intermittently required, for example, in dairy plants and slaughterhouses. By recovering energy from such low-temperature waste heat effluents, the incoming cold water can be heated with single- or two-stage heat recovery systems using (if required) preheating heat exchangers and supercritical CO_2 heat pumps as the first and second stages, respectively. This paper presents a number of key operating parameters and the energy performance of two such systems. The first one is an industrial-scale unit implemented in a Canadian dairy plant to provide process cooling and heating simultaneously. The second one is a laboratory-scale two-stage heat recovery system, including a passive heat recovery heat exchanger (as the first stage) and a supercritical CO_2 heat pump (as the second stage).

2 BACKGROUND

Identified as a natural refrigerant in 1850, carbon dioxide (CO_2) has been widely used since 1900, mainly in marine refrigeration and air-conditioning (Neksa et al., 2010). However, it more or less disappeared when artificial refrigerants (CFC) were launched in the 1930's. Research on CO_2 applications in heat pumps was initiated in the late 1980's (Lorentzen, 1989, 1993) in order to replace environmentally harmful chemical refrigerants, and the CO_2 technology has undergone tremendous development since this revival (Neksa et al., 2010). CO_2 is a natural, low-cost, non-flammable and, at concentrations below 5% by volume in air, a non-toxic fluid. It has high specific heat and thermal conductivity, and low viscosity. Its Global Warming Potential (over a 100 year integration period) is negligible (GWP = 1) when used as a confined refrigerant. Compared to conventional refrigerants, its critical temperature

is very low (31.1°C), a remarkable property allowing CO_2 heat pumps to work above the critical pressure (7.3773 MPa) and reject heat over large temperature glides (Neksa, 2002; Kim et al., 2004). Gustav Lorentzen (1989) was the first to suggest a supercritical CO_2 cycle for automobile air-conditioning, and Lorentzen and Pettersen (1992) published experimental results in this area. Later on, the supercritical CO_2 heat pump cycle generated widespread interest as an efficient and environmentally attractive technology for heating water (Neksa et al., 1994, 1997, 1998a, 1999; Hwang and Radermacher, 1998; Yarrall et al., 1998). Since 2001, air source heat pumps using CO_2 as a refrigerant have been successfully marketed in Japan to produce hot water using cheap late-night electric power and store hot water in tanks for daytime use. White et al. (2002) studied a 115 kW (thermal capacity) supercritical CO_2 heat pump prototype used to produce hot water at a temperature of 77.5°C and refrigeration at temperature below 2°C with a heating coefficient of performance (COP) of 3.4. Another supercritical CO_2 heat pump prototype was simultaneously used to provide cooling at 5°C and (water) heating up to 90°C (Yarrall et al., 1999). A 50 kW supercritical CO_2 heat pump prototype used for water heating was built in 1996 (Pettersen and Neksa, 2002) and another 25 kW pilot plant was installed in a food-processing factory in 1999 using industrial waste heat as a heat source (Kim et al., 2004). Stene (2007) investigated the application of supercritical CO_2 heat pumps in low-energy houses concluding that they can be used for simultaneous or alternate space and hot water heating. Finally, over the last few years, air and water source supercritical CO_2 heat pumps used for water heating began to be implemented in the European and North American markets for domestic (Fornasieri et al., 2008) and industrial/commercial applications.

3 PROCESS COOLING & HEATING

Industrial applications of CO_2 heat pumps are still rare in the Canadian energetic context where the current prices of fossil and electrical energy are relatively low compared to other countries. This section summarizes in comprehensible, simple terms the main features of one of the first industrial-scale supercritical CO_2 heat pumps implemented in a Canadian dairy plant (Marchand, 2011; Minea, 2013a; 2013b).

3.1 Process integration

An industrial-scale supercritical CO_2 heat pump, including a 25 kW compressor (nominal power input), has been integrated between two of the plant's industrial processes (Figure 1a) (Marchand, 2011). It recovers energy from one process at temperatures below 0°C and provides heat to another process at temperatures above 85°C (Figure 1b). Due to the low critical temperature of CO_2, these boundary thermal conditions allow the heat pump to operate in the so-called *supercritical* cycle with evaporation at subcritical pressures and heat rejection at pressures above the CO_2 critical pressure (7.377 MPa-a). So, unlike conventional subcritical heat pump cycles, heat is not supplied by means of refrigerant condensation, but by cooling compressed high-pressure CO_2 inside a special heat exchanger called *gas cooler*. As can be seen in Figure 1a, no internal heat exchanger is included in the heat pump thermodynamic cycle. Because both the cooling and heating thermal effects of the unit can be used by the industrial facility, the overall system energy efficiency could be determined.

The superheated refrigerant (state 1) flows into the compressor where it is brought to higher pressures and temperatures (state 2) by a non-isentropic compression process (1-2) (Note: the theoretical isentropic process is shown as 1-2s.) (Figure 1b). The discharge pressure is controlled based on the varying amount of refrigerant inside the high-pressure side of the system and not by the saturation pressure as is the case with conventional HFC refrigerants (Montagner and Melo, 2010). By reducing the valve opening, the CO_2 mass flow rate decreases, the CO_2 accumulates in the high-pressure side and its pressure thus rises. Conversely, by increasing the valve opening, the high-side pressure decreases while the excess charge is stored as a liquid inside the low-pressure side of the refrigeration circuit (Kim et al., 2004). This particular control strategy requires special algorithms to adjust the adiabatic expansion process in order to keep the evaporation almost constant and optimize

the compressor discharge pressure and the gas cooler outlet temperature (Kauf, 1999; Kim et al., 2004; Chen and Gu, 2005).

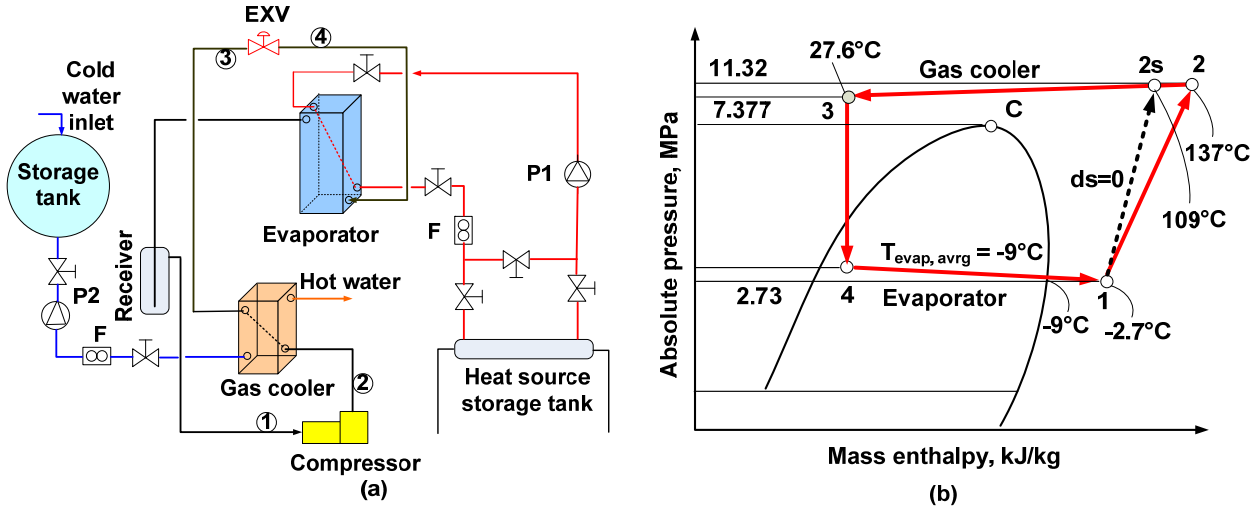


Figure 1: (a) Schematic Diagram of the Industrial-Scale Supercritical CO_2 Heat Pump Implemented in a Canadian Dairy Plant; (b) Average thermodynamic cycle of the one-week running test in the pressure-enthalpy diagram; C: critical point; P1: hot brine pump; P2: cold water pump; EXV: expansion valve; F: flow meter; (Minea, 2013b)

For a representative one-week running test, Figure 1b shows that the high discharge pressure reached 11.22 MPa-a (112.2 bar-a), a value about 6-7 times higher than that of an equivalent HFC-based water heater heat pump. Such a high discharge pressure automatically results in high volumetric capacities leading to small pipe diameters and inner volumes. On the other hand, for a given capacity, compressor displacement is much smaller, as well as its compression ratios. Also, due to the smaller size of the piping and components, the explosion energy stored in a CO_2 system is not much different from that of conventional heat pump units (Neksa et al., 2010). After compression, the superheated vapour at state 2 enters the once-through gas cooler where the non-isobaric heat rejection (cooling) process (2-3) takes place. The gas cooler is different from the conventional diphase condensers because only sensible heat is transferred to the constant flow cold water. As noted, the single-phase heat transfer process occurs at supercritical high pressures, while the CO_2 temperature is gliding. This means that sensible heat is transferred over a large temperature range and the CO_2 temperature drops during heat rejection, resulting in relatively low CO_2 outlet temperatures from the gas cooler. The vapour leaving the gas cooler at state 3 flows through the expansion valve where it is adiabatically throttled to lower pressures and temperatures (state 4) while controlling the compressor discharge pressure. The electronic expansion valve provides a great advantage in terms of performance, even when thermodynamic losses are present (Cho et al., 2007). The evaporation pressure, much higher than that of HFC-based heat pumps, is controlled by regulating the refrigerant flow through the expansion valve. Relatively high operating pressures within the evaporator and gas cooler lead to efficient heat transfer processes enabling these devices to be more compact than those of standard heat pumps. Instead of an expansion valve, fixed geometry capillary tubes could also be selected, but they operate with specific refrigerant charges aimed at maximizing system performance, and consequently their efficiency tends to decrease when differing from the design point that corresponds to the fixed charge, being unable to self-regulate with operating condition variations (Madsen et al., 2005). Finally, the two-phase refrigerant enters the plate-type evaporator at state 4 where it absorbs heat, evaporates and is superheated up to state 1 through the non-isobaric evaporation process (4-1). Superheated CO_2 vapour at state 1 enters the compressor suction line and the heat pump cycle restarts.

3.2 Operating parameters

The industrial-scale supercritical CO_2 heat pump was fully equipped with +/-0.3°C accurate thermocouples, and compressor (Ohio SEMITRONICS W4-063E 80 kW, accuracy ± 0.5) and pump electrical power (Ohio SEMITRONICS PC20-006D 4 kW, accuracy ± 0.25) transducers. The compressor suction and discharge pressures, as well as the heat source and sink thermal carrier flow rates, were measured with high resolution pressure transducers (DANFOSS MSB-300, accuracy ± 1%) and flow meters (ONICON type F, accuracy ± 0.5%), respectively. A Wi-Fi data transmission system and analysis software were set up to monitor system operation. Sixteen thermal (temperatures), hydraulic (flow rates) and electrical (power) parameters were scanned at 15 second intervals, then averaged and saved every 2 minutes to help determine the cycle's instantaneous and overall thermodynamic performance. A processing program was created to compute the cycle instantaneous and overall energy performance.

The one-week running test was performed under constant heat source (water/glycol brine at 35% by volume) and heat sink (water) flow rates and temperatures (Table 1). Based on these average thermodynamic parameters, Figure 1b presents in the pressure-enthalpy diagram the corresponding supercritical CO_2 heat pump thermodynamic cycle. It can be seen that the heat pump operated at a relatively low evaporating temperature (-9°C) with a compression ratio of 4.26 and 6.3°C superheating at the evaporator outlet (Table 2). To optimally match the CO_2 vapour temperature glide inside the gas cooler at the actual discharge pressure with the heated water temperature increase, the ideal gas cooler CO_2 outlet temperature must be 109°C exactly. However, the recorded compressor discharge temperature was 137°C, i.e. about 28°C higher than the final temperature of the ideal isentropic compression process. It shows that vapour superheating during the non-adiabatic compression process is relatively excessive, probably due to the inefficient control of the expansion valve at relatively low evaporating pressures and temperatures. However, since the compressor discharge gas temperature was relatively high, the heat pump was able to heat the cold water up to the temperature required (>85°C). On the other hand, these parameters led to a lower than expected isentropic compression efficiency (i.e. 62.6% vs. 70-75%), a compressor performance parameter defined as follows (Figure 1b):

$$\eta_{is} = \frac{h_{2s} - h_1}{h_2 - h_1} \tag{1}$$

where h is the CO_2 vapour mass enthalpy (kJ/kg).

Table 1: Heat source and heat sink fluid parameters over the one-week running test

Heat source (water-glycol 35%)			Heat sink (water)		
Temperature Inlet/outlet	Flow rate	Pump input power	Temperature Inlet/outlet	Flow rate	Pump input power
°C	L/min (GPM)	kW	°C	L/min (GPM)	kW
-1.5/-3.9	159.7 (41)	1.7	22.3/87.5	11.7 (3)	0.64

The cold water (heat sink) entered the gas cooler at 22.3°C and the useful heat was supplied by the gas cooler over a wide (109°C) temperature glide resulting in a large enthalpy difference for CO_2 in the gas cooler (285 kJ/kg) and a relatively low CO_2 temperature prior to entering the expansion valve (27.6°C). Thus, within the gas cooler the refrigerant was cooled down to about 5°C above the cold water entering temperature, i.e. an optimum temperature approach.

Table 2: Average CO_2 parameters over the one-week running test

CO_2 compressor					T_{EVAP}	Hot water outlet T
Suction		Discharge		π_C	-	-
P	T	P	T	-	-	-
MPa-a (bar-a)	°C	MPa-a (bar-a)	°C	-	°C	°C
2.73 (27.3)	-2.7	11.3 (113.2)	137	4.26	-9	87.5

P: pressure (a - absolute); π_C: compression ratio; T: temperature; EVAP: evaporation

The gas cooler thermal efficiency is defined using the following equation (Figure 1b):

$$\eta_{GC} = \frac{\dot{m}_{GC}(h_2 - h_3)}{\min(\dot{m}_{GC}, \dot{m}_{water}) * (h_2 - c_p T_{SINK}^{IN})} \quad (2)$$

where c_p is the average specific heat of the CO_2 superheated vapour flowing at a constant pressure within the gas cooler (kJ/kgK); $\dot{m}_{GC}$ - the CO_2 vapour mass flow rate (kg/s); $\dot{m}_{water}$ - the water (heat sink) mass flow rate (kg/s); T_{SINK}^{IN} - the water (heat sink) inlet temperature to the gas cooler (°C). According to equation 2, the measured gas cooler thermal efficiency was 63.1%, a relatively poor performance due to a high compression ratio (4.26) followed by excessive vapour superheating during the non-isentropic compression process.

3.3 Energy performance

The supercritical CO_2 industrial heat pump energy performance was determined with the compressor operating at full speed using the constant operating parameters shown in Tables 1 and 2. As can be seen in Table 3, the machine provided 52 kW_{th} (thermal power) by recovering 25.2 kW_{th} from the heat source.

The overall *system* coefficient of performance can be defined as follows:

$$System\ \ COP = \frac{Q_{EV} + Q_{GC}}{E_C + E_{P1} + E_{P2}} \quad (3)$$

where Q_{EV} is the thermal energy recovered by the evaporator (i.e. the cooling effect) (kWh); Q_{GC} - the thermal energy supplied by the gas cooler (kWh); E_C, E_{P1} and E_{P2} - the electrical energy consumed by the compressor, and the heat source (P_1) and heat sink (P_2) pumps, respectively (kWh). Based on the overall cycle energy balance (Figure 2a), with an average error margin of 8% (Table 3), the *system* coefficient of performance was 3.21. If in equation 3 only the gas cooler thermal energy supplied and the compressor electrical energy consumed are taken into consideration, the resulting *heat pump* coefficient of performance ($COP_{HP} = \frac{Q_{GC}}{E_C} = \frac{h_2 - h_3}{h_2 - h_4}$) drops to 2.29 (Table 4). Such moderate energy performance suggests that the expansion valve high-side pressure and refrigerant charge control algorithm must cover an extensive range of negative evaporating temperatures providing lower evaporating pressures compared to those of air-source supercritical CO_2 heat pumps designed for temperate climates. Improving control to recover heat at temperatures below 0°C will provide appropriate evaporator and compressor superheating, as well as lower compression ratios and much higher *heat pump* and *system* COPs.

Table 3: Power balance of supercritical CO_2 heat pump over the one-week running test

Duration	Cooling capacity provided	Compressor power input	Thermal power supplied	Energy balance average error margin	Pump power input	
-	-	-	-	-	Brine (heat source)	Water (heat sink)
Hours	kW	kW	kW	%	kW	kW
192	25.2	22.7	52.5	8.06	0.98	0.36

Table 4: System and heat pump average coefficients of performance

Coefficient of performance	
System COP	*Heat pump* COP
3.21	2.29

3.4 Annual overall performance

The annual energy and cost savings were estimated by assuming that the supercritical CO_2 heat pump operates continuously throughout the year. Table 5 shows the annual energy and cost savings achieved when only the process *cooling* effect is taken into consideration. If the plant's existing electrically driven refrigeration system operates with an average annual cooling coefficient of performance of 4, the industrial facility will save Can$4,529/year (based on an electrical energy price of Can$0.08206/kWh – 2014) only by reducing the electrical (compressor) energy consumption (55.188 MWH/year) of its own refrigeration system.

Table 5: Annual savings related to process fluid cooling (52 week period) (2014)

Process cooling energy supplied (or waste heat recovered)	Electrical (compressor) energy saved	
-	Amount	Cost
MWh/year	MWh/year	Can$/year
220.752	55.188	4,529

The supercritical CO_2 heat pump heats process water simultaneously. By assuming the annual efficiency of the plant's propane boiler is 80% and the price of propane is Can$0.5614/L (2014), we may reduce the annual consumption of propane by 81,000 litres and its cost by Can$45,473 (2014) (Table 6).

However, to achieve these annual energy and cost savings, the supercritical CO_2 heat pump (compressor) as well as the heat source and heat sink circulating pumps have to consume electrical energy. The amount of electrical energy consumed by the heat pump compressor and both circulating pumps will be 198.852 MWh/year and 11.7 MWh/year, respectively. Based on the price of electricity (Can$0.08206/kWh - 2014), the cost of the electrical energy consumed by the compressor and both circulating pumps will be Can$16,317/year and Can$963.2/year, respectively. Consequently, the net annual savings achieved by reducing the annual consumption of propane will be Can$28,193/year (i.e. $45,473 - $17,280), and the *total* net annual cost savings for both the cooling and heating processes could be estimated at Can$32,722/year (i.e. $28,193 + $4,529) (Tables 5 and 6). Consequently, based on the price of the CO_2 heat pump (Can$60,000$ - 2014) only, the simple payback period could be estimated at 1.83 years (i.e. Can$60,000/Can$33,858), without taking into consideration the additional costs incurred for heat pump integration (engineering, circulation pumps, piping, valves, thermal insulation and controls).

Table 6: Annual energy and cost savings for process water heating (52 week period) (2014)

Process heating supplied	Boiler propane inlet energy	Propane saved		Electrical energy consumed (compressor + pumps) by the supercritical CO_2 heat pump	
-	-	Quantity	Cost	Amount	Cost
MWh/year	MWh/year	L/year	Can$/year	MWh/year	Can$/year
524,264	655,330	81,000	45,473	198.852	16,317

On the other hand, reducing the propane annual consumption by 81,000 L, equivalent to 655,330 MWh/year (Table 6), helps decrease the plant's annual CO_2 emissions by 153.74 tons per year (Table 7). This amount was estimated based on the propane equivalent carbon dioxide emission (0.2346 kg of CO_2/kWh). In addition, the electrical energy consumed by the heat pump compressor and both circulating pumps (208.9 MWh/year) (Table 6) increases the annual CO_2 emissions by 254.8 kg because in Québec (Eastern Canada), where almost 98% of the electric production is hydraulic, the average conversion factor is 0.00122 kg CO_2/kWh. Consequently, the net reduction in CO_2 emissions will reach 153.48 tons per year (Table 7). If, for example, 100 similar supercritical CO_2 industrial heat pumps were implemented in Québec, the CO_2 emissions could be reduced by 15 348 tons/year.

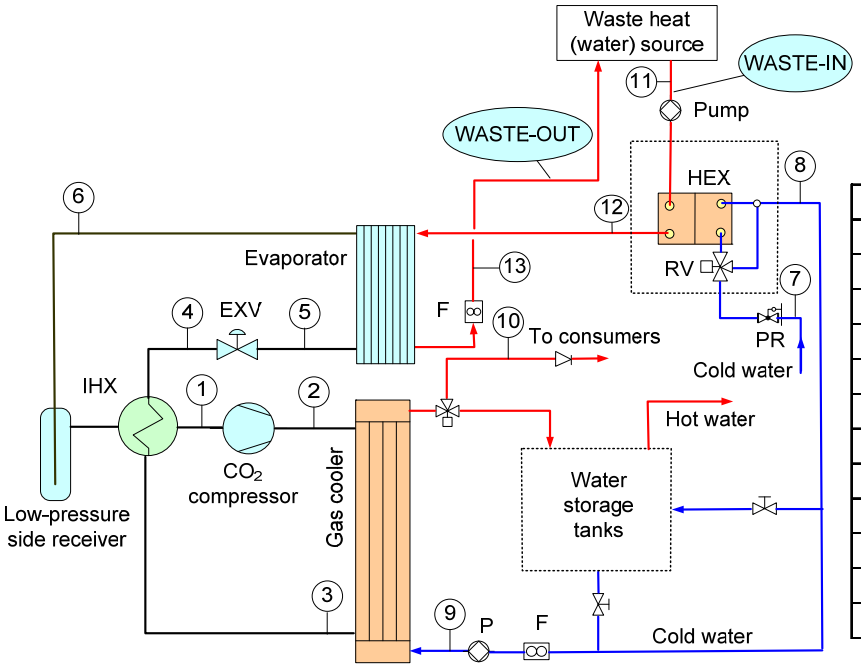
Table 7: Annual CO_2 gas emission reduction (52 week period) (2014)

Reduction of annual CO_2 emissions due to reduction of propane consumption		Increase of annual CO_2 emissions due to additional electrical energy consumed		Net reduction of CO_2 annual emissions
Equivalent CO_2 emissions	Total	Equivalent CO_2 emissions (Québec)	Total	-
kg of CO_2/kWh	Tons of CO_2/year	kg of CO_2/kWh	kg of CO_2/year	Tons of CO_2/year
0.2346	153.74	0.00122	254.8	153.48

4 WASTE HEAT RECOVERY

In the agro-food and other similar medium-sized industries, as well as in large institutional buildings (hospitals, etc.), process and/or domestic hot water is required at temperatures above 60°C for process, cleaning or washing purposes. Simultaneously, in such facilities, low-enthalpy waste heat in a liquid (or air) form is often available at temperatures of up to 45°C, while the cold water to be heated comes from municipal water systems at relatively low temperatures depending on seasonal weather. To efficiently recover heat from waste heat effluents at temperatures of up to 45°C, a two-stage heat recovery system, using a preheating heat exchanger and a CO_2 supercritical heat pump as the first and second stages, respectively, was experimentally studied (Figure 2) (Minea, 2012). The supercritical CO_2 heat pump included a 7 kW (nominal power input) single-stage, constant-speed compressor, three plate-type heat exchangers (evaporator, gas cooler and internal heat exchanger), a low-pressure side receiver and two parallel electronic expansion valves (EXV) that control the high-side pressure of the thermodynamic cycle. The low-pressure side receiver manages the refrigerant charge variation without flooding or drying up the evaporator. The internal heat exchanger is used for sub-cooling the high pressure superheated CO_2 with the compressor suction gas in order to improve overall cycle efficiency. An electrical boiler supplies (waste) hot water to the evaporator at different preset temperatures. As can be seen in Figure 3a, if the waste heat fluid enters the heat recovery system at temperatures between 15°C and 45°C it can be cooled down to between 10°C and 38°C, depending on the actual inlet temperature of the cold water. In cold climates, it generally varies from a minimum of 7°C (in the winter) to a maximum of 17°C (in the

summer). By bypassing (or not) the preheating heat exchanger, the cold city water can be heated up to 38°C before entering the supercritical CO_2 heat pump. The process is controlled by the regulating valve RV according to the relative magnitude of the inlet temperatures of the heat source and heat sink thermal carriers.



Temperatures and pressures from test W-3 (see Tables 8 and 9)

1	10.6 °C (3.97 MPa)
2	91.2 °C (10.06 MPa)
3	29 °C
4	21.2°C
5	5.9°C
6	5.8 °C
7	7.54 °C
8	7.54 °C
9	7.54 °C
10	69.9 °C
11	12.4 °C
12	12.4 °C
13	8.4 °C

Figure 2: Experimental Setup of the Laboratory-Scale Heat Recovery System. EXV: expansion valve; F: flow meter; HEX: preheating heat exchanger; IHE: internal heat exchanger; PR: pressure regulator; RV: 3-way regulating valve; 1 to 13: measurement points (Minea, 2012)

Several tests were conducted using the following assumptions: (i) both the waste heat source and heat sink fluids enter the heat pump at constant flow rates, i.e. 60 L/min for the waste heat source water and 6.6 L/min for the cold water; (ii) the waste heat source fluid enters the heat pump at approximately 7°C, 10°C and 12°C, representing the lowest values expected during the colder seasons; (iii) the cold city water enters the CO_2 heat pump at between 6.5°C and 7.5°C, representing the average municipal water temperature in the winter in cold climates; (iv) under these boundary thermal conditions, the cold city water by-passes the preheating heat exchanger as well as the hot water storage tank assembly; (v) for waste heat inlet temperatures above 15°C (in the winter) and 25°C (in the summer), the preheating heat exchanger must operate in order to recover more heat and, therefore, preheat the cold water before it enters the heat pump evaporator. This way, the preheating heat exchanger will improve the overall energy efficiency of the entire heat recovery system. Figure 3b shows the energy balance of the supercritical CO_2 heat pump during a 6 hour running test (W-3). Results from two other similar tests (W-1 and W-2) are shown in Tables 8 and 9. In these tables there is only one difference between the three tests shown, i.e. the waste heat inlet temperature was set at 7°C, 10°C and 12°C respectively, while the cooling fluid inlet temperature was set at 7°C. It can be seen in Table 9 that the evaporator superheating figures (5.6°C to 7°C) were in the same order of magnitude as the ones achieved with the industrial-scale CO_2 heat pump, i.e. ~ 6.3°C (see Table 2). Also, the compressor isentropic compression efficiency varied around 70% under the average experimental parameters shown in Tables 8 and 9 (Minea, 2012).

This important performance indicator, with evaporating temperatures varying from 0°C to 5°C and compression ratios around 2.6 (Table 9), was 10.6% higher than the performance indicator of the industrial-scale supercritical CO_2 heat pump operating at a much lower

evaporating temperature (-9°C) (see section 3.2) and with compression ratios about 39% higher (Table 2).

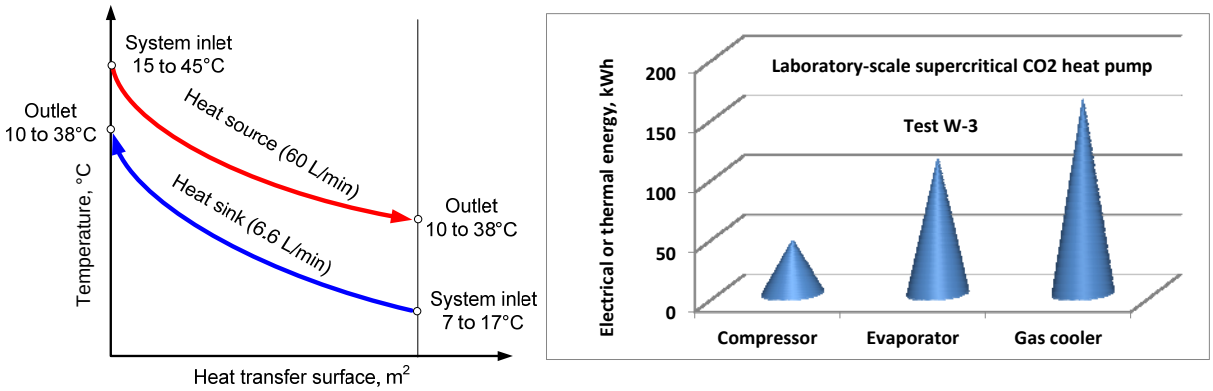


Figure 3: (a) Extended Temperature Ranges of Heat Source and Heat Sink Thermal Carriers Entering and Leaving the Two-Stage Heat Recovery sSstem; (b) Overall Heat Pump Energy Balance of a Typical Running Test (W-3) (Minea, 2012)

Table 8: Average temperatures of heat source and heat sink thermal carriers entering and leaving the laboratory-scale supercritical CO_2 heat pump (Minea, 2012)

Test #	Heat source (water: 60 L/min)		Heat sink (water: 6.6 L/min)	
-	Inlet	Outlet	Inlet	Outlet
-	°C	°C	°C	°C
W-1	7.5	4.3	6.5	67.6
W-2	10.1	6.4	7.3	69.5
W-3	12.4	8.4	7.5	69.9

Table 9: CO_2 average operating parameters of the laboratory-scale supercritical CO_2 heat pump (Minea, 2012)

Test #	Compressor					T_{EVAP}
-	Suction		Discharge		π_C	-
-	P	T	P	T		-
-	MPa-a (bar-a)	°C	MPa-a (bar-a)	°C		°C
W-1	3.49 (34.9)	5.7	9.60 (96)	93	2.75	0
W-2	3.67 (36.7)	9.0	9.76 (97.6)	92	2.66	2
W-3	3.97 (39.7)	10.6	10.06 (100.6)	91.2	2.54	5

P: pressure (a - absolute); π_C: compression ratio; T: temperature; EVAP: evaporation

In this case, according to equation 2, the *heat pump* coefficient of performance is defined as the gas cooler thermal energy supplied divided by the compressor electrical energy consumed, and the *system* heating coefficient of performance as the gas cooler thermal energy supplied plus the cooling energy recovered divided by the electrical energy consumed by the compressor and the heat source (water) circulating pump. It can be seen that the average *system COP* (5.93) was 37.8% higher than the *heat pump COP* (3.69), both displaying excellent energy performance ratios (Figure 4).

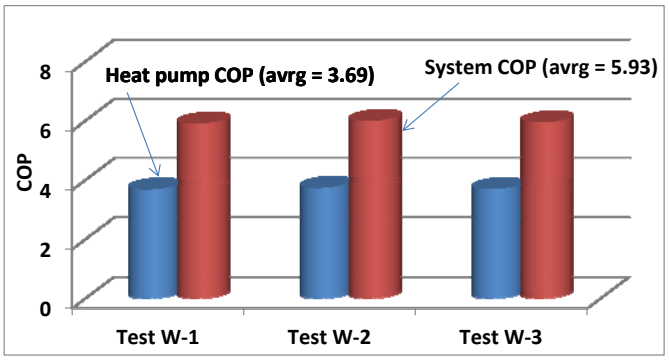


Figure 4: *Heat Pump* and *System* Coefficients of Performance (COP) of the Laboratory-Scale Supercritical CO_2 Heat Pump

5 CONCLUSIONS

This paper presents a number of experimental results achieved with two supercritical CO_2 heat pumps. The scope is to help promote the CO_2 heat pump technology in industrial settings (e.g. agro-food processing plants) and large institutional buildings (e.g. hospitals).

The first one is an industrial-scale supercritical CO_2 heat pump implemented in a Canadian dairy plant to provide process cooling and heating simultaneously. It recovers energy from one process at temperatures around -1.5°C and provides heat to another process at temperatures above 85°C. This industrial-scale CO_2 heat pump operates with relatively low isentropic compression (62.6%) and gas cooler thermal efficiencies (63.1%), and high compression ratios (4.26). These parameters, which are lower than expected, could be attributed to a faulty high-side pressure control by the expansion valve operating at a relatively low evaporating temperature (-9°C). As a consequence, the *system* coefficient of performance, including both the useful cooling and heating thermal effects and total energy consumption (CO_2 compressor and circulating pumps) was 3.21. This lower than expected performance suggests that the expansion valve algorithm controlling the high-side pressure and refrigerant charge must cover an extended evaporating temperature range below 0°C. An improved control algorithm may provide appropriate evaporator and compressor superheating, as well as lower compression ratios and much higher *heat pump* and *system* COPs. However, by eliminating the use of 81,000 litres of propane annually, the plant's *total* cost saving is estimated at Can$32,722/year and the simple payback period (based on the CO_2 heat pump acquisition cost only) at 1.83 years. In addition, the plant's carbon dioxide emissions are reduced by more than 153 tons per year.

The second system is a laboratory-scale two-stage heat recovery system, including a passive heat recovery heat exchanger (as the first stage) and a supercritical CO_2 heat pump (as the second stage). It is sized to recover heat from low-grade waste heat effluents at temperatures of up to 45°C in order to heat cold city water entering the system at temperatures between 6.5°C and 7.5°C. The evaporator superheating figures (5.6°C to 7°C) are in the same order of magnitude as the ones achieved with the industrial-scale CO_2 heat pump (6.3°C). However, the isentropic compression efficiency (70%) is achieved at higher evaporating temperatures (i.e. 0 to 5°C) than those of the industrial-scale supercritical CO_2 heat pump, while the compression ratios (2.6) are about 39% lower. The average *system* COP is 45.8% higher than that of the industrial-scale supercritical CO_2 heat pump (5.93).

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