High-Temperature Heat Pumps Used for Low-Grade Industrial Process and Waste Heat Recovery

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Abstract: Industrial low-grade process and/or waste heat sources are abundant Canada as well as in many other countries. Such sources, especially those available at low-temperatures going from -5 °C to about 45 °C, are rarely recovered in order to improve the overall efficiency of industrial energy intensity and productivity, and to reduce the global greenhouse gas emissions. This paper summarizes a part of Canada's contributions to Tasks 3 and 4 of the IEA-IETS Annex 13/IEA-HPP Annex 35 project. It succinctly presents a number of heat pump systems able to supply process or domestic hot water at temperatures up to 85 °C or hot air up to 95 °C (in the case of high-temperature drying heat pumps) by recovering low-grade energy from various process and waste heat sources. Small-scale prototypes have been laboratory designed, built and experienced in order to validate the refrigerant choice and simulated performances, and to improve the systems' control sequences and reliability. In the case of natural refrigerants as ammonia and CO_2, additional objectives aimed at forming and training local expertises, testing and improving the safety measures according to the Canadian national refrigeration code and increasing the confidence of refrigeration specialists, industrial plant owners and general public. A part of laboratory and in-field, or simulated results obtained, intending at preparing and encouraging future implementations in various industrial and institutional facilities, are also succinctly presented.

Key Words: high-temperature heat pumps, industrial waste heat recovery, energy efficiency

1 INTRODUCTION

Canadian industry requires about 48% of the total country primary energy input (Canadian National Energy Board, 2008). However, in most of manufacturing industries, more than 50% of it is released to the environment via waste heat streams such as stack emissions (combustion gases and hot air), steam, process gases and liquid effluents (Stricker & Ass,, 2006). Industrial heat pumps (IHP) may contribute to energy conservation and productivity improvement, and to the global reduction of greenhouse gas emissions (IEA Annex 13/35, 2010). They are most cost effective in countries with low electrical energy prices compared to those of certain fossil fuels. This is not the case of Canada where, for example, the price of natural gas is still relatively low compared to the average cost of electrical energy. There are also some technical (e.g. absence of users for the heat generated) and legal (e.g. lack of public or private incentives) barriers to the widespread application of IHPs. In spite of this particular context, during the last few years, Canada has initiated a number of R&D studies followed by in-field applications. They mainly focused on high-temperature heat pumps able to recover heat at relatively low temperatures and produce hot water at temperatures between 45 and 85 °C or hot air up to 95 °C for high-temperature drying processes. Mechanical vapour recompression and ejector cooling systems powered by industrial waste heat at higher temperatures have been analysed and/or in-field applied, but not presented in this paper (Figure 1). As it can be seen in Figure 1, natural refrigerants, such as carbon dioxide and ammonia, and more or less known but simple cycles, have been targeted (Minea, 2013a, 2013b). Some applications in medium-sized industries, as food processing plants (Agriculture Canada, 2010), as well as in large institutional buildings, where simultaneous heating and cooling processes are generally required (Energy, Mines and Resource Canada, 2010), have been performed in the past, or are still under way. The scope was to demonstrate their energetic and environmental benefits, and prepare the industry for future implementations.

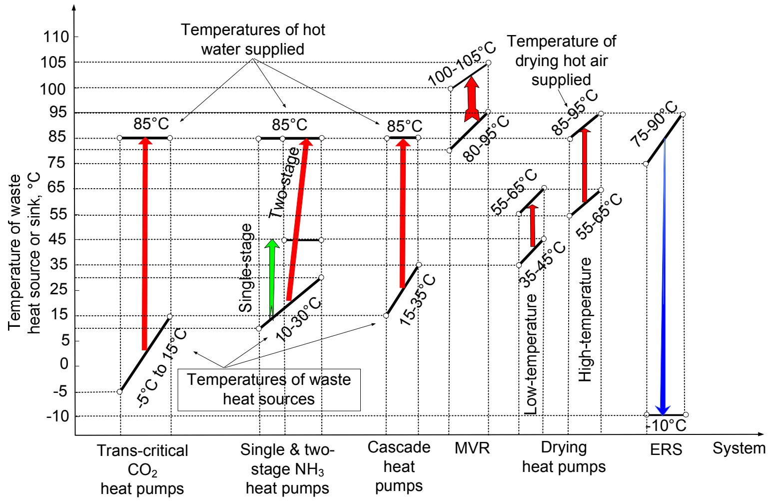


Figure 1: Industrial Heat Recovery Systems Studied in the Frame of IEA HPP Annex 13/35 (2010). ERS: ejector refrigeration system; Input: inlet temperature; MVR: mechanical vapour recompression; Output: output temperature (Note: MVR and ERS not presented in this paper) (Minea, 2013a, 2013b).

2. BACKGROUND

Prior to 1995, several hundred of IHPs were developed and implemented in Canada, especially in the lumber drying sector, food industry, sugar refining, breweries and fish processing (IEA HPP Annex 21, 1995). For example, toward the end of 1993, 17% of 14 investigated processes in more than 1900 individual plants were using IHPs based on the closed-vapor compression cycle, more than 90% of which were used for lumber drying. At the end of 2010, in 339 plants surveyed in Québec (Eastern Canada), Ontario and Manitoba (Central Canada), and British Columbia (Western Canada), 31% of existing industrial heat pumps (26) were used for drying, 27% for waste heat recovery, and 8% for evaporation processes with cooling capacities varying between 14 and 1050 kW. Today, the number of IHPs installed in Canada is still relatively low compared to the number of technically and economically viable opportunities. Higher capital costs and low energy prices, as well as a lack of knowledge on the potential benefits and/or experience with industrial heat pump technology may explain this situation. The future market penetration rate of IHPs was estimated at 5% per year until 2030 of which about 80% could be closed vapour compression cycles and up to 20% mechanical vapour recompression systems (Stricker & Ass., 2006).

3 CASCADE HEAT PUMPS

High-temperature cascade heat pump systems have the advantage of lower pressure ratios and higher isentropic efficiencies for each stage compressor. Different combinations of working fluids can also be used according to the temperature ranges of both the waste heat and heat sink sources. On the other hand, cascade heat pump systems introduce extra temperature differences, a greater complexity and extra control problems, and slightly reduce the overall system coefficients of performance. However, this performance reduction seems less critical in the context of heat pumps recovering large quantities of *free* industrial waste heat. Two cascade heat pump cycles have been studied in order to find the best working fluid combination (Minea, 2013a). The first concept (Figure 2) was an optimized cascade system including two vapour compression cycles with intermediate heat exchanger. Compared to a standard cascade cycle, this configuration included a liquid refrigerant pump (LP) and a vapour injection solenoid valve (ISV) on the second heat pump cycle aiming at facilitating the system start-up by removing the refrigerant stored inside liquid receiver LR2. The second concept consisted of two vapour compression cycles coupled by an intermediate liquid closed-loop. Both experimental set-ups have been sized to recover waste heat at temperatures varying from 15 °C to 35 °C and supply process or domestic hot water at temperatures up to 85 °C. The new fluid HFO-1234yf as well as HFC-245fa have been chosen as refrigerants on the first and the second stage, respectively. Both electrically-driven reciprocating compressors were equipped with variable speed controllers, and the electronic expansion valves kept the superheating at between 5 °C and 15 °C, depending on the thermal properties of each working fluid.

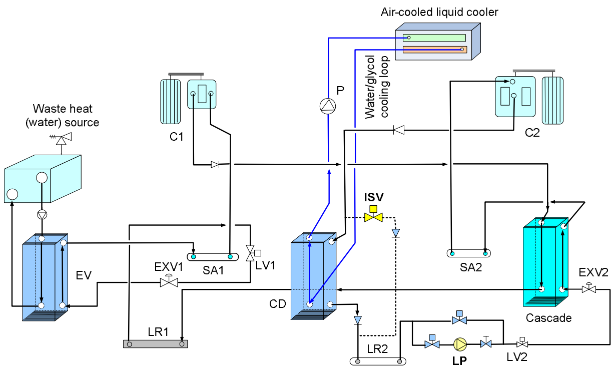


Figure 2: Optimized Cascade Heat Pump Prototype; EXV: expansion valve (Minea, 2013a, 2013b); C: compressor; EV: evaporator; EXV: expansion valve; ISV: injection solenoid valve; LP: liquid pump; LR: liquid receiver; LV: liquid valve; P: circulating pump; SA: suction accumulator

Simulations achieved for both experimental setups showed that, for example, by using waste heat at a 25 °C inlet temperature, hot water ca be provided at 84.5 °C with an overall coefficient of performance of 2.6 (Richard, 2013). The industrial implementation of a cascade heat pump system in a poultry processing plant is schematically shown in Figure 3 (CADDET, 1990). The first stage of this system recovered waste heat rejected by an industrial ice machine. The cascade heat pump acted as the second stage of the heat recovery system. Heat was also recovered from an existing refrigeration plant by the aid of an intermediate closed-loop. Cold water entering the system at 12 °C was heated up to 25 °C inside a pre-heating heat exchanger and, then, up to 63 °C within the cascade heat pump, prior to being stored inside a storage tank and/or directly supplied to industrial processes or other consumers. The total investment cost of this heat recovery system (engineering, equipment, installation) was US$ 165 000 (1990) while natural gas annual savings of 330 m^3 have been achieved. The system overall COP has been estimated at 10.7, and the simple pay back period at 2.7 years.

4 AMMONIA HEAT PUMPS

Among other candidates, ammonia, an energy efficient and relatively cheap refrigerant, having zero Ozone Depleting (ODP) and Global Warming (GWP) Potentials, was focused to replace the current synthetic refrigerants. In spite of its known good thermo-physical properties, the present restrictive legislation limits the ammonia utilization in heat pumps, mainly because of its toxicity and flammability at high concentrations in the air (Pearson, 2007). However, an ammonia residential heat pump operated in Norway, first with ambient air, then with ground as *green* energy sources (Johassen and Hardarson, 2006). By using compact mini-channel heat exchangers, Palm (2008) and Monfared and Palm (2011) tested a heat pump containing 110 grams of ammonia in order to produce domestic hot water.

Stene (2008) discussed the main characteristics of ammonia heat pump systems for heating and cooling of non-residential buildings. In Canada, a single-stage 7.5 kW (compressor power input) water-to-water NH_3 heat pump has been designed, built and laboratory tested accordance with the national refrigeration code (Figure 4) (Minea and Richard, 2011). An electrical boiler supplied hot water simulating the waste heat, and the condenser excess heat was discharged outside through an air-cooled liquid cooler.

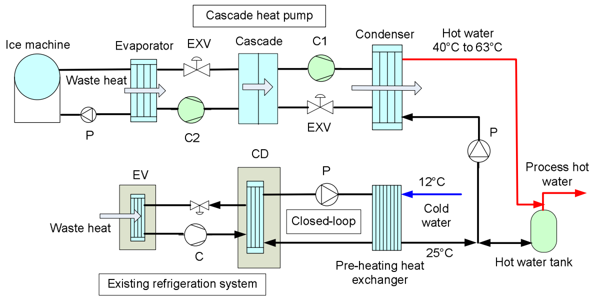


Figure 3: Schematic Diagram of a Cascade Heat Pump Implemented in a Canadian Poultry Processing Plant (**CADDET, 1990**)

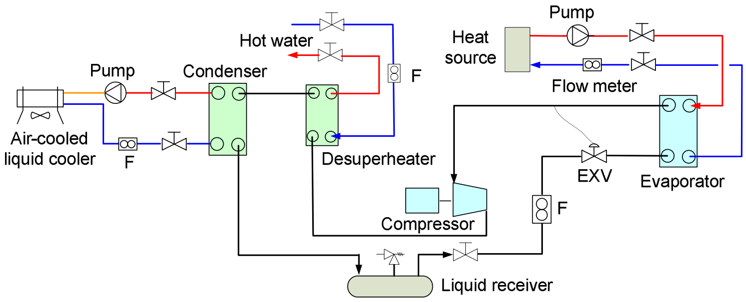


Figure 4: Experimental Setup of the Single-Stage Ammonia Heat Pump (**Minea, 2013a**)

With waste heat carrier fluid (water) entering the heat pump evaporator at 15 °C, the heat pump supplied hot water at 42 °C. At the same time, the desuperheater heated domestic hot water from 25.5 °C to 44 °C. Based on the compressor energy consumption, the heat pump coefficient of performance was 3.84, but it dropped to 2.85 when the energy consumption of the compressor and waste heat and hot water circulating pumps were taken in consideration. Figure 5a schematically represents the single-stage ammonia heat pump recently implemented in a new Canadian dairy plant (Gosselin, 2013). This system recovers heat from the superheated ammonia vapor coming from the plant's existing ammonia compressors and heats process water. Since hot water isn't produced and consumed simultaneously, a large hot water storage system was provided. In the past, the maximum supply water temperature from single-stage ammonia heat pumps using standard (25 bar) compressors was about 48 °C. However, ammonia screw compressors available today on the market can work at higher pressures up to 45 bars and condensing temperatures as high as 75 °C. Consequently, double-stage ammonia heat pumps equipped with desuperheaters are technically ready for industrial applications (www.vuilter-emerson.com). Simulations shown that such systems are able to heat cold water from 10 °C up to 85 °C by recovering heat, for example, from the discharge ammonia vapour coming from existing ammonia refrigeration systems (Figure 5b) (Richard, 2013).

5 CARBONE DIOXYDE HEAT PUMPS

Industrial process fluids in a liquid form at temperatures between -5 °C and 35 °C are valuable heat sources for CO_2 supercritical heat pumps in order to produce hot water at temperatures as high as 85 °C (Lorentzen, 1993). A two-stage heat recovery system (Figure 6), including a pre-heating heat exchanger (as the first stage) and a 7 kW (shaft power input)

CO_2 water-to-water supercritical heat pump (as the heat recovery second stage), has been in laboratory tested (Minea, 2012; 2013a). The pre-heating heat exchanger shown in Figure 6 is required when the temperature of the heat source is higher than the temperature of the cold water to be heated. However, when the heat source inlet temperature isn't high enough to pre-heat the cold water, the pre-heating heat exchanger must be bypassed. After passing through or by-passing the pre-heating heat exchanger, the cold water is supplied to the once-through gas cooler at constant flow rate, temperature and pressure. In actual industrial applications, the gas cooler is connected to a closed loop where a variable speed pump circulates the water between it and the storage tanks.

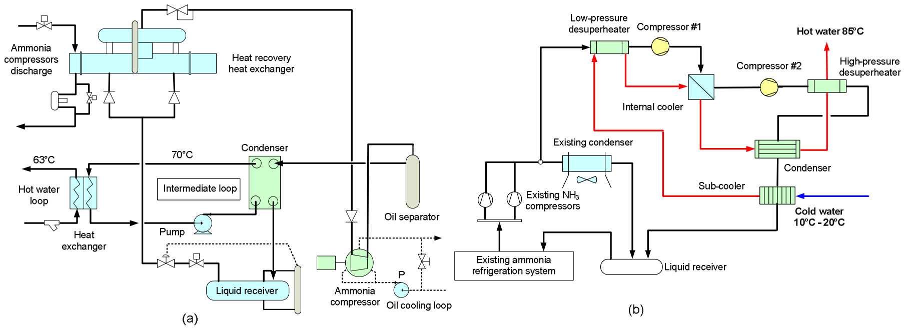


Figure 5: Applications of Ammonia Industrial Heat Pumps in Existing Refrigeration Systems; (a) Single-Stage Heat Pump Implemented in a Canadian Diary Plant (Gosselin, 2013); (b) Double-Stage Ammonia Heat Pump (www.vuilter-emerson.com)

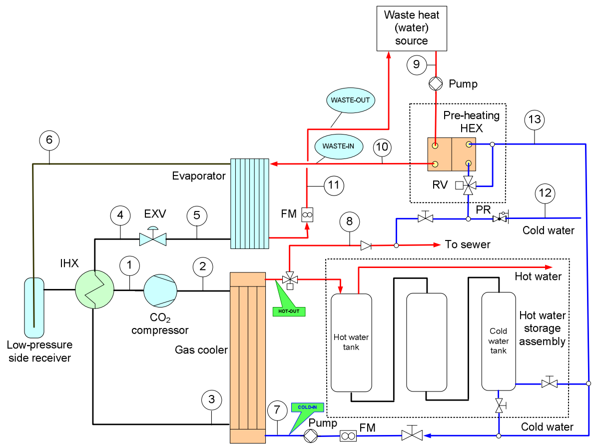
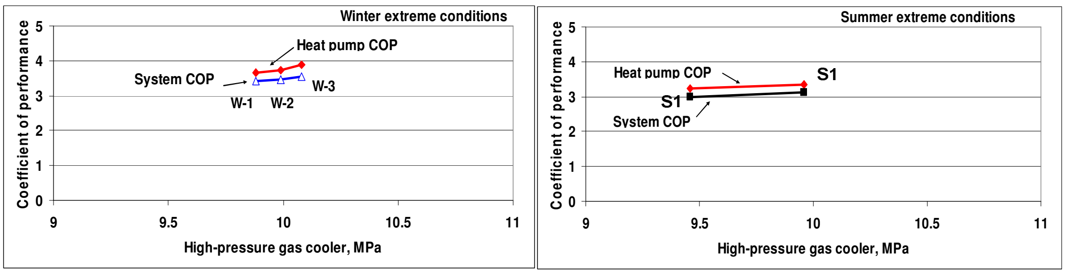


Figure 6: Experimental Setup of the Laboratory Two-Stage Heat Recovery System with a CO_2 Supercritical Heat Pump as the Second Stage. EXV: expansion valve; FM: flow meter; HEX: heat exchanger; IHE: heat exchanger; PR: pressure regulator; RV: 3-way regulating valve; 1 to 13: measurement points (temperatures, pressures, flow rates) (Minea, 2012; 2013a)

Several tests have been done under the following experimental conditions: (i) both waste heat source and heat sink fluids entered the heat pump at constant flow rates; (ii) the waste heat source fluid entered the heat pump at 7 °C (test W-1), 10 °C (test W-2) and 12 °C (test W-3) in the winter, and at 7 °C (test S-1) and 15 °C (test S-2) in the summer (see Figure 7); (iii) under these thermal conditions, the preheating heat exchanger and the hot water storage tank assembly were by-passed (Minea, 2012). During these *extreme* winter operating conditions, with cold water entering the heat pump at 7 °C, process hot water has been supplied at average temperatures of 67 °C, 69 °C and 71 °C by using waste heat water entering the heat pump evaporator at 7 °C, 10 °C and 12 °C, respectively. The winter *heat pump* coefficient of performance (COP) (defined as the gas cooler thermal power divided by the compressor electrical power input) and the *system* COP (defined as the gas cooler thermal power divided by the electrical input power of the compressor and waste water circulating pump) slightly increased with the gas cooler high-pressures (Figure 7a). As for the winter *extreme* operating conditions (see Figure 7a), under the summer *extreme* operating conditions, both measured *heat pump* and *system* COPs were over 3, but the last one was about 8.2% lower (Figure 7b). Figure 8 schematically represents a 100 kW_{th} CO_2 supercritical industrial heat pump recently implemented in a Canadian dairy plant (Marchand, 2011; Minea, 2013a). In this application, hot water is provided at temperatures varying between 60 and 75 °C by recovering process waste heat. Many other implementation options exist for similar supercritical CO_2 heat pumps, for example, for recovering heat from groundwater, industrial cooling systems, including ice storage tanks, fish farming processes or industrial pasteurization processes in order to produce process or domestic hot water (www.mycomcanada.com). A second Canadian dairy plant integrates a 100 kW_{th} supercritical CO_2 heat pump between a process cooling loop and the plant's washing water closed-loop (Minea, 2013b). In this application, process water at 12 °C (heat source) is circulated within a closed loop between a buffer tank and the heat pump evaporator in order to produce cold water at 7 °C, able to cool a process fluid from 16 °C to 12 °C through a heat recovery heat exchanger. The cooling process provides additional energy savings to the heat recovery system and thus increases the system overall COP. On the other hand, cold city water is heated inside the heat pump gas cooler up to 85 °C, prior to being supplied to the plant's washing loop via a storage tank. The supercritical CO_2 heat pumps can also be implemented in hospitals requiring large quantities of process and domestic hot water. In such a Canadian large hospital, city cold water at 7 °C in the winter and 12 °C in the summer enters the heat pump gas cooler and leaves it at temperatures varying between 70 °C and 75 °C. The hot water storage capacity exceeds 5.5 m^3 while the instantaneous hot water demand is around 212 L/min, which is higher than what the CO_2 heat pump can produce, thus ensuring continuous water consumption for more than 15 hours (Minea, 2013a).



(a) (b)

Figure 7: *Heat Pump* and *System* Coefficients of Performance (COP) in *Extreme* Operating Conditions; (a) Winter (tests W); (b) Summer (tests S) (Minea, 2012; 2013a)

6. HEAT PUMP-ASSISTED DRYING SYSTEMS

About 45% of the Canadian territory is wooded, representing 10% of the planet's forests. Hardwood, such as hard maple, yellow birch, oak, and white walnut is usually dried at low temperatures (maximum 55 °C), a process that consumes up to 70% of the total energy required for primary wood transformation. On the other hand, softwood drying at temperatures higher than 75 °C is essential to prevent warping and cracking. Most Canadian conventional hardwood and softwood drying kilns use fossil fuels (oil, propane, natural gas) or biomass (bark) as primary energy sources (Canada Statistics, 2002). However, electrically-driven low- and high-temperature heat pumps can be used in combination with fossil fuels or electricity as back-up energy sources. To save energy, these systems recover heat from the dryer hot and humid air, and the recovered sensible and latent energy is used to reheat the drying air.

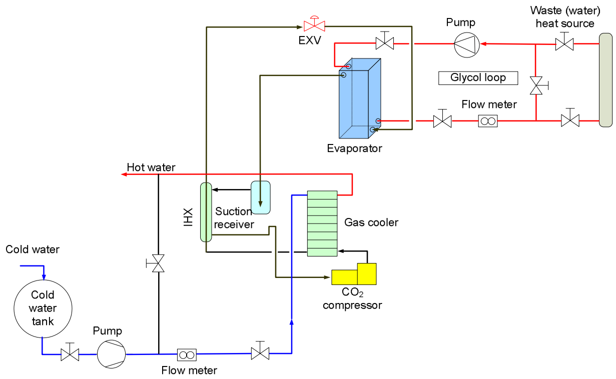


Figure 8: Schematic Diagram of the CO_2 Supercritical Industrial Heat Pump Implemented in a Canadian Dairy Plant (Marchand, 2011; Minea, 2013a)

6.1 Low-temperature drying heat pump

A low-temperature heat pump consisting of a 13 m^3 forced-air dryer coupled to a 5.6 kW (compressor power input) has been applied for drying hardwood species such as sugar maple and white and yellow birch through dehumidification (Figure 9a). The average green moisture content of these species generally varies between 65% and 72%, so drying is an essential step in the manufacturing process (furniture, etc.) (Minea, 2006a). The dryer was equipped with steam and electrical backup heating coils and the heat pump was installed in a mechanical room next to the dryer. Operating according to an intermittent drying schedule, the heat pump has extracted 1935 litres of water from a yellow birch stack with initial and final moisture contents of 75.9% and 7.6% (dry basis), respectively, during a 220-hour drying cycle (Figure 9b). The average dehumidification efficiency of such a cycle, expressed in terms of the specific moisture extraction rate (SMER), which represents the ratio between the mass of water extracted and the heat pump total electrical energy consumption (compressor - C and blower - B), was $2.5\text{kg}_{\text{water}}/\text{kWh}_{\text{C+B}}$ above the wood fibre saturation point (FSP). For hardwood from Eastern Canada, the FSP, defined as the physical state where the wood cell cavities are completely devoid of free water and their walls are still completely saturated, is of about 25% (dry-basis). The natural gas consumption and the total energy costs (electricity plus natural gas) of almost all drying cycles decreased by 23% and 57.5%, respectively, as compared to the natural gas consumption of equivalent *conventional* drying cycles. At the same time, for initial moisture contents above 41%, the total water quantity extracted above the FSP were up to 2.9 times higher than that removed below this point. Consequently, in these cases, the dehumidification efficiency of the low-temperature drying heat pump (SMER) was up to 3 times higher above the FSP than below it.

6.2 High-temperature drying heat pump

On the other hand, coniferous (also known as softwood) species such as pine, spruce and fir are generally dried at temperatures higher than 75 °C. An industrial-scale, high-temperature drying heat pump prototype, including one 354 m^3 forced-air wood dryer with steam heating coils and two high-temperature drying heat pumps (Figure 10a) has been implemented in Canada (Minea, 2004). Each heat pump, using as refrigerant the HFC-236fa, a non-toxic and non-flammable fluid, having a relatively high critical temperature, included a 65 kW (nominal power input) compressor, an evaporator, a variable speed blower and electronic controls located in an adjacent mechanical room. Both remote-type condensers were installed inside the drying chamber and operated at condensing temperatures as high as 100 °C in order to provide hot air up to 95 °C. Based on the softwood initial moisture content, generally in the range of 35% to 45% (dry basis), optimum drying schedules were developed for each species. The average heat pump COPs, defined as useful thermal power output divided by compressor electrical power input, varied from 4.6 at the beginning to 3 at the end of each drying cycles.

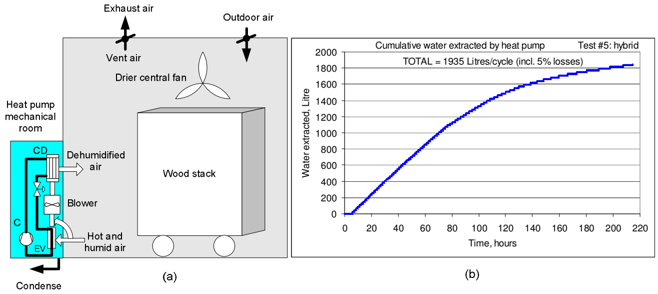


Figure 9: (a) Schematic Diagram of the Laboratory-Scale Hardwood Drying Heat Pump Prototype; (b) Cumulative Amount of Water Extracted [29, 30]; C: compressor; CD: condenser; EV: evaporator (**Minea, 2006a**)

The drying time to deliver white spruce with an approximate final moisture content of 18% was about 2.5 days, while, for balsam fir, it averaged 6.3 days. Total amounts of water extracted by each of the two heat pumps exceeded 9 600 litres (Figure 11b) for dried white spruce and mote than 13 500 litres in the case of dried balsam fir. The Specific Moisture Extraction Rate (SMER) ranged from $1.46\,\mathrm{kg_{water}/kWh}$ (with balsam fir) to $2.52\,\mathrm{kg_{water}/kWh}$ (with white spruce). By excluding the energy consumed for kiln preheating and for running the central fans, and the venting moisture losses, the energy consumed for softwood drying was between 27% and 57% lower compared to that of *conventional* cycles using oil as the sole source of energy.

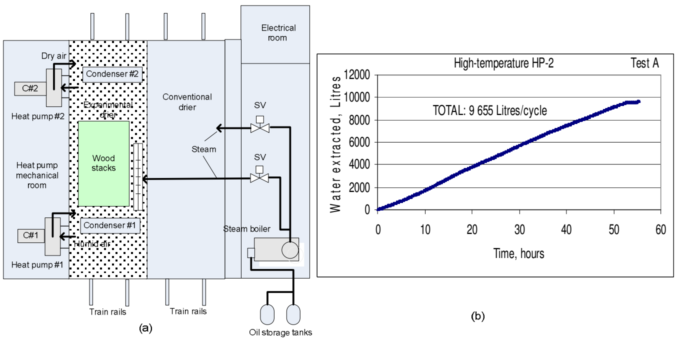


Figure 10: (a) Site of the Experimental Industrial-Scale Softwood Drying Heat Pump System; (b) Cumulative Volume of Water Extracted by Heat Pump #2; C: compressor; HP: heat pump; SV: vapour solenoid valve (**Minea, 2004**)

7 OTHER HEAT RECOVERY SYSTEMS

7.1 Fish Farming

Salmonid culture intended for the breeding, consumption and sport-fishing markets is growing in Canada. Fishery companies use surface or groundwater directly in incubation and fish nursery units, without any preheating. However, preheating the water would accelerate alevin growth. In order to reduce and/or eliminate fossil fuel consumption and, thus, improve environmental performance, a six-month field study of a two-stage heat recovery system with passive heat exchanger (as a first stage) and water-to-water heat pump (as a second stage) has been conducted in a Canadian fish farming facility (Figure 11a) (Minea, 2002). The cold groundwater was heated with heat recovered from the process waste water in order to reduce the alevin growth cycle. The plate heat exchanger raised its temperature by recovering energy from the waste water coming from the fish breeding pools. Then, it entered the heat pump condenser where it was heated once again by the refrigerant, which recovered heat from the waste water leaving the passive heat recovery heat exchanger. Finally, the warmer fresh water was directed through an oxygenation system (not shown in Figure 11a) where it mixed with the re-circulated water flow, prior to returning to the alevin basins. The heat pump's (compressors only) monthly average coefficient of performance (COP), defined as the thermal energy provided to the process water supplied to the alevin basins divided by the total electrical energy consumed by the heat pump compressors, was about 6.1. The average thermal efficiency of the first-stage heat recovery passive heat exchanger varied around 66%, and it provided more than 109 kW of thermal power to the groundwater. In terms of energy, the fresh groundwater was first heated by the passive heat exchanger, which provided 64% of the energy, and then by the heat pump, which provided 30.5% over a six month production period (Figure 11b). Heat pump heat recovery was made possible with a net power consumption of 45.8%, while the water circulation pumps took 54.2% of the total energy input. Consequently, the overall average coefficient of performance, calculated for the whole system, including both the passive heat exchanger and the heat pump, was 7.9. As a direct consequence, the alevin growth rates with heated (10-12 °C) and unheated (7-8 °C) fresh water showed that the annual production of 5 gram rainbow trout increased by approximately 50% using 4°C warmer water. Moreover, the time period required for the rainbow trout to grow was 65% shorter than that of conventional processes, leading to substantial improvements in the company's overall efficiency. Finally, the system simple pay-back period was estimated at 1.28 years, without taking the increase in fish production into consideration.

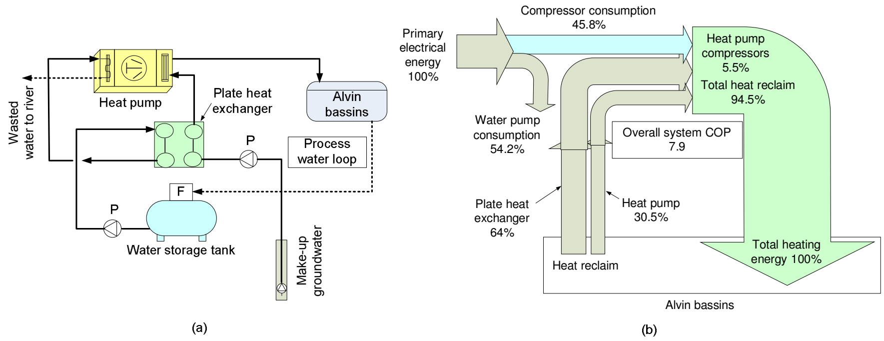


Figure 11: (a) Two-Stage Heat Recovery System Implemented in a Canadian Fish Farming Facility; (b) 6-Month Global Energy Balance of the Two-Stage Heat Recovery System Implemented in a Canadian Fish Farming Facility (Minea, 2002); COP: coefficient of performance; F: filter; P: water pump

7.2 Cold Warehouses

Ammonia refrigeration systems in large cold storage warehouses operate with discharge and condensing temperatures that make it possible to efficiently recover heat normally rejected in the atmosphere by dry or evaporative condensers. Heat recovered could be used for space (e.g. offices) and/or industrial hot water heating (e.g. neighbouring food processing plants). Such a heat recovery system was developed to recover heat from new retrofit ammonia compressors installed in a cold warehouse (Minea, 2007). Six new single-stage screw compressors having the total nominal refrigeration capacity of 1600 kW operated at evaporating temperatures between -33 °C and -44 °C. Oil cooling and injection processes helped keeping the common discharge header at a relatively low temperature (around 52 °C). A shell-and-tube heat recovery heat exchanger (desuperheater) was installed on the common discharge header of the six new ammonia compressors. To recover part of the available heat rejected (i.e. 2361 kW), from the common discharge header, a two-stage heat recovery system with a desuperheater (as a first stage) and water-to-air heat pumps (as a second stage) was designed (Figure 12). About 33.5% of the available thermal power was recovered and used for heating a building service and office spaces with a total area of 12

250 m^2 via 21 reversible water-to-air heat pumps with nominal capacities varying between 2 and 8 refrigeration tons (1 ton = 3.517 kW) connected to the building reverse closed-loop. Each of these heat pumps used as a heat or sink source the brine flowing from the building closed loop at temperatures ranging between 15 °C (in the winter) and 25 °C (in the summer). When at least two compressors operated simultaneously, all the available sensible heat and up to 93.9% of the condensing enthalpy were recovered to meet 100% of the building peak heating demand. In addition, two 22.3 kW (heating capacity) brine-to-water heat pumps have been installed to the same brine closed-loop in order to heat industrial hot water up to maximum 65 °C for a neighbouring food processing plant (Figure 13).

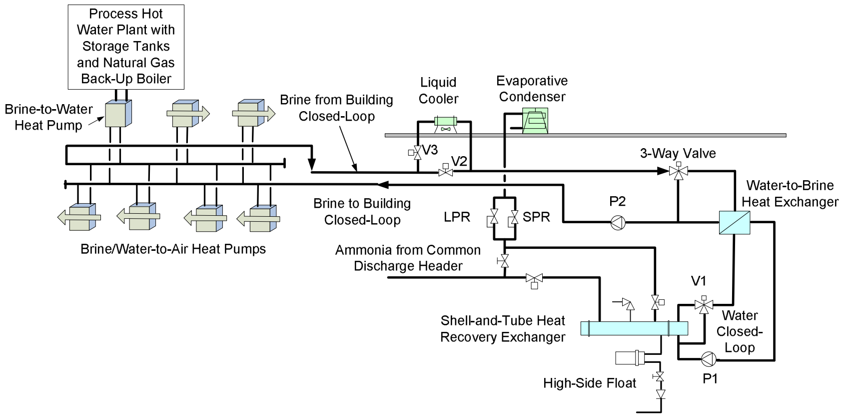


Figure 12: Two-Stage Heat Recovery System [24] and Configuration of the Building Brine Closed-Loop Distribution System; P: water or brine circulating pump; NO: normally open; NC: normally closed; LPR: large size pressure regulator; SPR: small size pressure regulator; 3-way motorized valve; P: brine circulating pump (Minea, 2007)

7.3 Cooling towers

Relatively large amounts of low-grade waste heat are also available in industrial processes using cooling towers (or condensers). Cooling water entering the cooling towers at around 30 °C year-around represents one of the most promising fields of application for industrial heat pumps. Several heat recovery options have been studied and designed for implantation in a large Canadian metallurgical plant where exothermal chemical reactions occur within eight electrolytic reactors (Minea, 2006b). The associated evaporators and crystallizers were initially cooled by a water closed loop linked to three forced air-cooled cooling towers where the cooling water entered at temperatures varying between 30 and 40 °C with flow rate of maximum 2220 m^3/h. The heat recovery options were based on the assumption that 1066 m^3/h of cooling water, i.e. 48% of the total flow rate available on the site, would be used as a waste heat source for one or several water-to-water IHPs combined (or not) with passive plate heat recovery heat exchangers (Figure 13). By using about 4 MW of electrical input power, the thermal power recovered was 13.6 MW and the total thermal power delivered by the heat pump facility, 17.6 MW. As a result, the total hot water flow rate supplied to the agro-cultural complex reached 1536 m^3/h at a temperature of about 70 °C. The temperature of the return water was about 60 °C, and the heat recovery system overall coefficient of performance, of 4.33.

CONCLUSIONS

This paper resumes a part of Canada's country contributions to the IEA-IETS Annex 13 / IEA-HPP Annex 35 project that aimed at developing, improving and promoting heat pump concepts in order to encourage future industrial implementations. Because low-grade waste heat rejections at temperatures lower than 45 °C represent about 25% of the total primary energy input of Canadian manufacturing industries and, simultaneously, industrial processes and domestic consumers need hot water at temperatures up to 85 ℃ or drying hot air up to 95 ℃, Canada's past (and under way) projects focused on high- and low-temperature heat pump applications in medium-size industries and large institutional or industrial buildings.

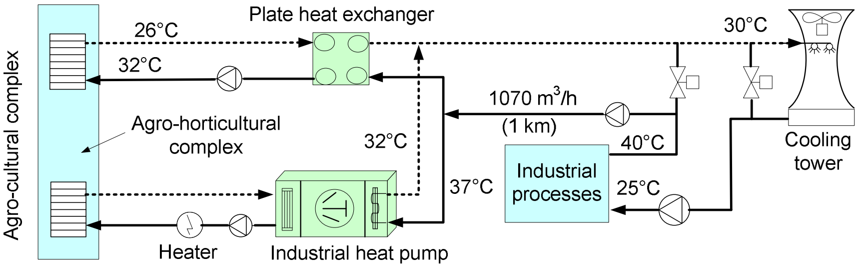


Figure 13: Combined Heat Recovery System with Intermediate Heat Exchanger and Industrial Heat Pump at a Canadian Agro-Cultural Complex (Minea, 2006b)

Among the selected concepts there were cascade, single- and double-stage ammonia, and supercritical CO_2 heat pumps. Improved high-temperature cascade heat pumps, including or not intermediate closed-loops, may recover heat from liquid effluents at 15-35 ℃ in order to heat cold water up to 85 ℃ by using new environmental friendly refrigerants as HFO-1234yf. Ammonia double-stage industrial heat pumps could also accomplish this task by using high-pressure screw compressors, while most of manufacturers are able today to provide hermetic ammonia refrigeration circuits, as well as reliable detection and evacuation systems in case of massive leakage. CO_2 supercritical heat pumps proved having efficiency at least comparable to those of other heat recovery systems, being cost effective alternatives for simultaneous process cooling and water heating in most of agro-food industries and large institutional buildings. Other presented applications of IHPs, as for industrial drying and heat recovery from ammonia refrigeration plants and cooling towers are valuable methods to save energy and reduce the global greenhouse gas emissions. In conclusion, this succinct review of a part of Canada's contributions to the IEA-IETS Annex 13 / IEA-HPP Annex 35 project shows that there are promising application opportunities for IHPs because large amounts of low-grade waste heat are available in many industrial sectors, especially in form of process liquid effluents.

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