Techno-economic evaluation of combining heat pump and mechanical steam compression for the production of low pressure steam from waste heat

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Abstract: Most of the industrial use of energy is for thermal applications such as drying and heating. A large percentage of the energy used is lost or dissipated into the environment as heat through warm liquid effluents and gaseous effluents. We evaluated an approach relying on combined use of a mechanical closed-cycle heat pump and mechanical steam compression. The heat pump produces steam at moderate vacuum conditions which is afterward compressed above atmospheric pressure. This sequence is to be used for upgrading of waste heat at temperatures of 60 °C, with the subsequent production of low pressure steam (110 °C or above) as a hot utility. A techno-economic evaluation of such a sequence is presented in the context of heat recovery from wet air of a paper drying process. A COP higher than three is achievable. Although such a sequence is currently hardly economically justifiable in a context of low natural gas prices, it could eventually be beneficial, especially where renewable energy is available and where the price of electricity compares to fuel prices.

Key Words: heat pump, mechanical steam compression, waste heat, wet air, heat recovery

1 INTRODUCTION

Most of the industrial use of energy is for thermal applications such as drying and heating of reactors. In the Canadian industry, about 70 % of the energy used is simply lost or dissipated into the environment as heat (Stricker & Associates Inc. 2006). Namely, heat is lost though warm liquid effluents (40 °C or above) which have to be cooled with a cooling tower or other heat dissipating device. Heat is also lost through gaseous effluents such as flue gas from combustion processes and wet air from air drying processes. In many cases, the quality of lost heat is poor because its temperature is too low for practical use or economic recovery by means of conventional heat recovery devices such as heat exchangers. There is a need for a flexible, efficient and economic approach, specifically for applications requiring heat at temperature above 100 °C.

The use of closed-cycle heat pumps (HP) for heat valuation is well known but there are limitations related to the achievable final temperature (below 120 °C) and the economically achievable temperature lift (typically below 50 °C). Furthermore, the performance of a heat pump is strongly related to the composition and conditions of each of the streams serving as heat source and heat sink, respectively. The idea of using HP for heat recovery in air drying process is not new and there are many commercial applications relying on this technology (Prasertsan and Saen-saby 1998, Morris 1993, Colak and Hepbasli 2009a and 2009b). Nevertheless, the use of HP for the production of steam (as a hot utility) is less considered, although this application of HP had been evaluated in the past (Binet et Frote 1987, Abrahamsson et al. 1997). The use of two stages HP or a combination of HP with mechanical steam compressors or thermo compression device such as ejectors had also

received some attention. Mechanical steam compressors alone have also been examined (Mujumdar 2006).

In this paper, we briefly present an approach relying on combined use of mechanical heat pump and mechanical steam compression. This sequence is to be used for valorisation of heat from waste heat down to temperatures of 50 °C with the production of low pressure steam (117 °C or above) as a hot utility. It is considered that such a utility can fulfill a large part of the energy requirements, namely in the pulp and paper industry and the agro-food industry. A basic techno-economic evaluation of such a sequence is presented, in the context of heat recovery from wet air from drying process.

2 SYSTEM PROPOSED

2.1 Description of the System

We present here the techno-economic evaluation of a heat recovery system for the valorisation of heat from a typical paper air drying process. Characteristics of the waste heat effluent (wet air from paper drying process) that we consider for the purpose of techno-economic evaluation are shown in Table 1. The heat valorisation system is to recover about 60 % of heat in a wet air stream having a 60 °C dew point. Recovered heat is to be upgraded as low pressure steam at 130 kPa and at a temperature of 117 °C. Water at 40 °C is used for the production of this utility steam.

Table 1 : Parameters considered for heat valorisation system

Wet air (waste heat)		**Outlet air (saturated)**		**Vapour produced (utility)**	
Flow	25 000 Nm^3/h	Temperature	47 °C	Pressure	130 kPa
Temperature	85 °C	Humidity	0.07 $kg_{water}/kg_{dry\ air}$	Superheat	10 °C
Pressure	101.3 kPa			Temperature	117.1 °C
Dew point	60 °C (0.15 $kg_{water}/kg_{dry\ air}$)				

Figure 1 shows the sequence proposed.
a) Direct-contact heat exchanger device for the recovery of heat from wet air
b) HP system comprising evaporator, compressor, and condenser and throttling valve
c) Steam compressor.

Heat from wet air is recovered as hot water in a direct-contact heat exchanger instead of feeding it directly to the evaporator of the HP system. The use of such auxiliary equipment allows a more versatile use of HP system as well as a better control. This type of exchanger allows good heat transfer coefficients and is already used in drying hoods for the production of hot water from hot wet air (Thermal energy international Inc, Newswire 2010). In such a system, outlet air is at a temperature closed to the temperature of the inlet water (temperature difference of 5 °C or below) (Sofame Technologies inc. 2009). As shown in the figure, hot water exiting the direct heat exchanger is directed to the evaporator (hot side) of the HP system. Excess water coming from condensation of the wet air stream is evacuated and can be reuse for another application.

In the evaporator, heat is transferred to an evaporating HP fluid or refrigerant (cold side of the evaporator). At the exit of the evaporator, vaporised HP fluid is compressed before being condensed in the condenser (hot side of the condenser). Liquid water maintained at moderate vacuum condition (50 kPa) is fed in the cold side of the condenser for the purpose of producing steam. The produced steam is then compressed at the desired pressure (130

kPa), and some water is added to the steam at the exit of the steam compressor, for adjustment of the superheating of the steam.

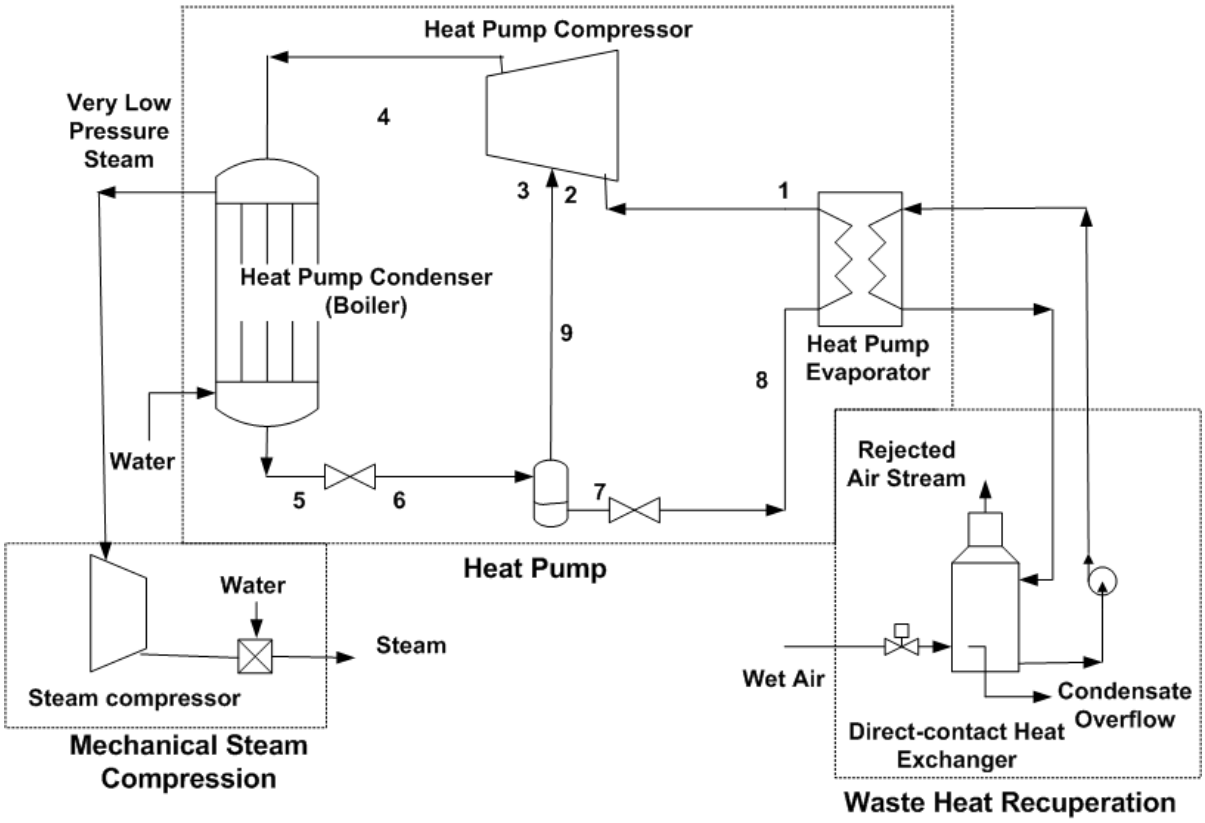


Figure 1: Schematic of system for the valorisation of heat from wet air from a drying process

In this study, heat pump fluids R-245fa and R-134a are considered. The fluid R-245fa, with a critical temperature of 154 °C presents interesting characteristics for high temperature applications although slightly toxic (ASHRAE Std 34 safety group B1, cannot be used for human comfort). The fluid R-134a, with a critical temperature of 101 °C, is a HFC traditionally used for medium temperature HP (generally with condensation below 80 °C). More recently developed fluids with low Global Warming Potential (GWP) index , R-1234ze and R-1234yf, were also modeled. In principle, HP using one of these fluids could easily be used to recover heat from a heat source of 40 °C or below for generation of heat at 80 °C.

One of the critical parts of HP is the compressor. Screw compressors appear to be the most appropriate for the application considered, because of their versatility regarding the span of flow rates and compression ratios. Furthermore, the presence of some liquid can be tolerated. Compressors without oil injection, used for steam compression, can support elevated temperature (225 °C) but are limited to compression ratio below 7.

2.2 Mass and Energy Balance

EES (Engineering Equation Solver from F-Chart Software) was used for a complete calculation of the mass and energy balance of such a system. Thermo physical properties required for the calculations are available and can be recovered by the software solver (properties for steam, R-245fa, for R-134a, for R-1234yf and for R-1234ze).

2.2.1 Waste Heat Recuperation

Direct-contact heat exchanger is simulated as a process involving direct contact of hot water and hot wet air with counter current flow. Condensed water is evacuated and the balance of

the water outlet from the direct heat exchanger goes to the evaporator of the HP system. The temperature difference between dew point of the inlet air and outlet water temperature is specified as 5 °C (which is the inlet water of the hot side of the evaporator). Similarly, the temperature difference between the exit saturated air and inlet water temperature is specified as 5 °C (which is the outlet water of the hot side of the evaporator). The temperature of the outlet saturated air is specified to be 47 °C for the recovery of 60 % of the heat stream (reference at 25 °C). The flow rate of the water fed into the direct exchanger is obtained from the energy balance.

2.2.2 Heat Pump Evaporation and Condensation

The heat pump evaporator is assumed to be a plate-and-frame heat exchanger. The evaporation temperature is 5 °C less then the outlet water at the hot side of the evaporator. Flow rate of the HP fluid is calculated according to the heat transfer requirement for waste heat recuperation (1 736 kW) and a 10 °C of superheating.

The heat pump condenser is assumed to be a shell-and-tube heat exchanger. The cold side is fed with liquid water at 40 °C and at 50 kPa absolute pressure (which is vapour pressure at a saturation temperature of 81 °C) and its flow is adjusted for a temperature pinch of 7 °C.

The heat exchange surface, used to estimate heat exchangers cost, is calculated from the global heat transfer coefficient (U, in kW/m²°C) and the heat duty. The global heat transfer coefficient (U), respectively for the evaporator and the condenser, is calculated from:

$$\frac{1}{U_{Evap}} = \frac{1}{h_{fluid,evap}} + R''_{wall} + \frac{1}{h_{water}}, \quad (1)$$

$$\frac{1}{U_{Cond}} = \frac{1}{h_{fluid,cond}} + R''_{wall} + \frac{1}{h_{water\ boiling}}, \quad (2)$$

Where the heat transfer coefficients (h, in kW/m² °C) are obtained from correlation for plate and shell-and-tube heat exchangers. The calculations procedure is not detailed here but the correlation can be found in the literature (for fluid boiling in a plate heat exchanger ($h_{fluid,evap}$) : Ayub 2003; for water circulation in a plate heat exchanger (h_{Water}) : Kays and London 1984; for fluid condensation in an horizontal tube ($h_{fluid,cond}$): Dobson and Chato 1998 ; and for water boiling (nucleate) (h_{Water}): Rohsenow 1952). To take into account wall and surface fouling, the term $1/R''_{wall}$ is assumed to be approximately 7.5 kW/m² °C.

2.2.3 Heat Pump Compression

Heat pump compressor is used to compress fluid vapour up to a pressure corresponding to a saturation temperature close to 90 °C. The isentropic efficiency ($\eta_{isen,HP,}$) of the compression process is assumed to be 75 % and about 20% of the enthalpy increase is assumed to be dissipated to the environment through oil cooling and various heat losses ($f_{Heat\ Loss}$).The electrical efficiency ($\eta_{el,HP}$) is assumed to be 90%. As illustrated in Figure 1, 2 and 3, the heat pump uses an economizer cycle. The economizer port in the screw compressor is assumed be located where the volume ratio is 1.5. The point 2 is located just before the economizer port and the point 3 is located just after the port. The pressure drop of the stream passing through the economizer port is assumed to represent about 30% of the difference between pressure at point 6 and at point 2. Equations 3 to 8 were used to calculate the compressor power. The enthalpy (h, in kJ/kg) in the first step of the compression process is calculated using:

$$h_{2,no\ heat\ loss} = h_1 + \frac{(h_{2s} - h_1)}{\eta_{isen,HP}}, \quad (3)$$

$$h_2 = h_1 + (1 - f_{Heat\ Loss,HP})(h_{2,\ noheat\ loss} - h_1), \quad (4)$$

where h_{2s} is the enthalpy obtained from an isentropic compression process and $h_{2,no\ heat\ loss}$ is the enthalpy at point 2 without considering the heat losses. The fluid state after the economizer port (point 3) is calculated using:

$$u_3 \dot{m}_{fluid,3-4} = u_2 * \dot{m}_{fluid,1-2} + h_9 * (\dot{m}_{fluid,3-4} - \dot{m}_{fluid,1-2}), \quad (5)$$

$$P_3 = P_2 + (P_9 - P_2) * 0.7, \quad (6)$$

$$\dot{m}_{fluid,3-4} / \rho_3 = \dot{m}_{fluid,1-2} / \rho_2, \quad (7)$$

where u is the internal energy (in kJ/kg), $\dot{m}_{fluid,1-2}$ is the fluid mass flow rate (in kg/s) before the compressor port, $\dot{m}_{fluid,3-4}$ is the fluid mass flow rate after the port, ρ is the density (in kg/m^3) and P is the pressure. The enthalpy (h) in the second step of the compression process is obtained from equations analogous to equations 3 and 4. Finally, the compressor power is calculated using:

$$\dot{W}_{HP} = \frac{\dot{m}_{fluid,1-2}(h_{2,no\ heat\ loss} - h_1) + \dot{m}_{fluid,3-4}(h_{4,no\ heat\ loss} - h_3)}{\eta_{el,HP}}, \quad (8)$$

Expansion processes (5 to 6 and 7 to 8) are assumed to be isenthalpic expansion processes.

2.2.4 Steam compression

Steam at the outlet of the HP condenser is compressed from 50 to 130 kPa. The isentropic efficiency of the compression process is assumed to be 75 % and about 10% of the enthalpy increase is assumed to be dissipated to the environment. The electrical efficiency is assumed to be 90%. Liquid water at 40 °C is mixed with the pressurized water vapour in order to achieve 10 °C of superheat.

2.3 Economic evaluation

Economic evaluation of the system is essentially based on calculating the cost of the compressors and of the heat exchangers. For each of these equipments, the delivered equipment cost (C) is calculated and adjusted using:

$$C = C_{Base} f_{mat} \frac{CE_{Study}}{CE_{Equation}} f_{delivery}, \quad (9)$$

where C_{Base} is a calculated equipment cost and f_{mat} is a material factor. CE_{Study} is the Chemical Engineering (CE) index used for this study (CE_{Study} value is set to 600, approximately the CE index in 2012 and 2013) and $CE_{Equation}$ is CE index related to the empirical equation used for C_{Base}. The value of $f_{delivery}$, a factor used to obtain the delivered cost from the FOB cost, is assumed to be 1.1 (Silla 2003).

For the heat pump screw compressor, the following expression was used for C_{Base} (Walas 1988, p. 667):

$$C_{Base,HPCompressor} = 1490 * W_{HPComp}^{0.71}, \quad (10)$$

where $W_{HP\ Comp}$ is the heat pump compressor power (in HP). The CE index is 325. For the steam screw compressor (functioning in vacuum), the following expression was used for C_{Base} (Seider 2009, p. 595):

$$C_{Base,SteamComp} = 7840.5 * Flow_{Steam\ Comp}^{0.38}, \quad (11)$$

where $Flow_{Steam\ Comp}$ is the actual flow at suction (in m^3/h) . The CE index is 500 and f_{mat} is 1. For the heat pump evaporator, a plate-and-frame heat exchanger, the following expression was used (Seider 2009, p. 592):

$$C_{Base,Evap} = 24\ 090 * A_{Evap}^{0,42}, \quad (12)$$

where A_{Evap} is the heat-transfer area (in m^2). The CE index is 500 and f_{mat} is 1(Stainless steel). For the heat pump condenser, a kettle shell-and-tube heat exchanger, the following expressions were used (Seider 2009, p. 571):

$$C_{Base,Cond} = \exp\left\{9.89757 - 0.8709\left(\ln\left(A_{Cond}\right)\right) + 0.09005\left(\ln\left(10.7634 * A_{Cond}\right)\right)^2\right\}, \quad (13)$$

$$f_{mat} = a + 1.18\left(A_{Cond}/100\right)^b, \quad (14)$$

where A_{Cond} is the heat-transfer area (in m^2). For Cr-Mo steel both sides, a is 1.7 and b is 0.07.The CE index is 500.

The cost for direct-contact heat exchanger was assumed to be 175 000 $ for the capacity of the system under study (Chabot 2012). When the wet air stream flow rate is modified, the following equation was used:

$$C = 175\,000 * \left(\frac{Flow_{wet\ air}}{25\ 000}\right)^{0.7}, \quad (15)$$

where the flow rate of wet air is in normal cubic meters per hour.

The capital cost of the project is obtained using:

$$Capital\ Cost = \sum f_{DC} f_{IC} f_{CCF} C, \quad (16)$$

where f_{DC} is equipment direct installation cost factor (assumed to be 1.2 for compressors, 2 for the condenser, 1.5 for the evaporator and 1.5 for the direct-HX), f_{IC} is the indirect cost factor (assumed to be 1.34) and f_{CCF} is the construction and contractor fees factor (assumed to be 1.18) (Silla 2003).

Amortization cost was calculated for a period of 5 years. Income tax and interest were not considered in the economic calculations. The annual operation and maintenance cost is calculated assuming 5 % of the total capital cost. The electricity cost is calculated assuming a price of 0.05 $/kWh. In the next section, Table 2 and 3 summarize the economic evaluations procedure and the results for the case studied.

3 RESULTS AND DISCUSSION

3.1 Base Case Results

Figures 2 and 3 show detailed results from simulation calculations for a heat source with a dew point of 60 °C and the production of steam at 130 kPa and 117 °C (parameters in Table 1) respectively using HP fluid R-245fa and HP fluid R-134a.

In the first case, the system allows the production of 2.3 MW of heat as steam at 130 kPa and 117 °C (flow rate of 0.90 kg/s), with an electrical consumption of 734 kW for compressors. Global coefficient of performance (COP), the ratio of heat produces on electricity consumed, is 3.13. It is approximately 50 % of the ideal Carnot COP based on the saturation temperature of the steam produced and the temperature of the air exiting the direct-contact heat exchanger (47 °C). The heat pump itself presents a COP of 4.13 or 60 % of ideal Carnot COP (based on HP fluid evaporation and condensation temperatures). The heat pump compressor power is 513 kW for a pressure ratio of 4.45. Flow rate of the HP fluid R-245fa at the exit of the compressor is 14.5 kg/s, requiring a compressor capacity of about 3600 m^3/h (assuming a volumetric efficiency 80 %). Discharge temperature at the exit of the fluid compressor is 91 °C which is a moderate temperature. Energy consumption of the steam compressor (working at vacuum conditions at the inlet) is 221 kW for a compression ratio of 2.6. The steam compressor volumetric capacity is high, about 12 500 m^3/h. Globally, we can expect a good performance for such a system.

In the second case (Figure 3) where the HP fluid is R-134a, global coefficient of performance (COP) is 2.84 which is 9% lower then calculated global COP for the system using R-245fa. HP itself presents a COP of 3.6 or 52 % of ideal Carnot COP. Flow rate of the HP fluid is R-134a at the exit of the HP compressor is 19.9 kg/s, requiring a compressor capacity of about 1300 m^3/h (assuming a volumetric efficiency 80 %). The R-134a condensation pressure of 3200 kPa is high, but compressors that withstand this pressure are available.

Table 2 and 3 present the details of the economic evaluation. The cost of the heat produced is 0.053 $ and 0.055 $ per kWh_{th} for, respectively, R-245fa and R-134a (14.7 and 15.3 $/GJ). This is equivalent to 10.3 and 10.7 $/GJ of combustible (ex: natural gas) used at 70 % efficiency (0.39 and 0.41 $/$m^3$). However, after the 5 year simple amortization period considered in this study, the price would drop by more than 50 % and be of 0.023 and 0.025 $/$kWh_{th.}$ Considering the context of low natural gas prices, such a system is currently hardly economically justifiable for the base case presented in this study.

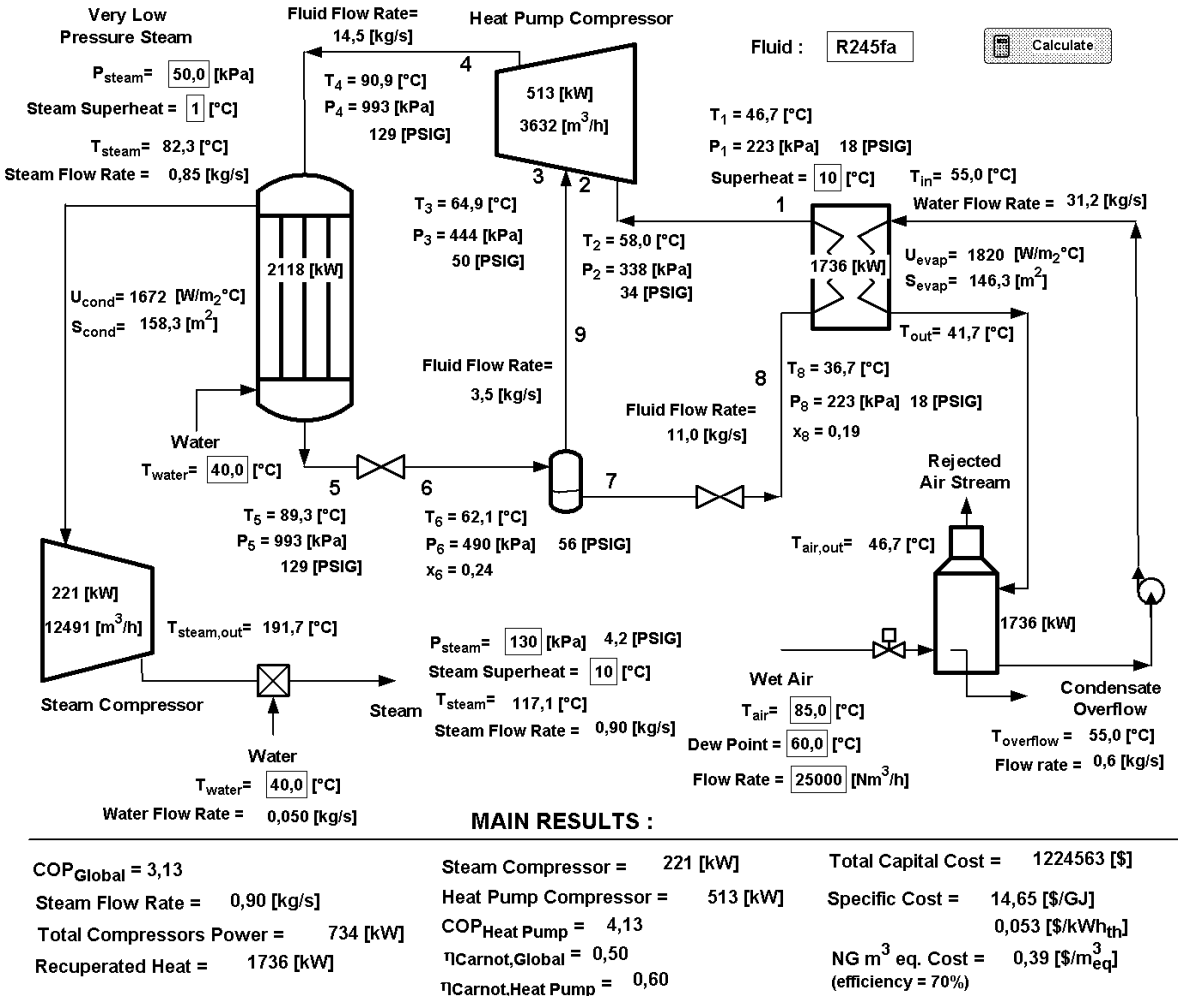


Figure 2: Schematic of system based on HP and mechanical steam compression with R-245fa for production of 2.3 MW of heat (steam at 130 kPa and 117 °C)

Table 2: Economic evaluation for a system based on HP and mechanical steam compression with R-245fa for production of 2.3 MW of heat (steam at 130 kPa and 117 °C)

	HP Comp.	Steam Comp.	Direct-HX	Evapora-tor	Conden-sor	Total
Key Parameter	513 kW	221 kW	1736 kW	146 m^2	158 m^2	
Delivered Equipment Cost ($)	312703	342 606	175 000	258 103	136 161	1 224 573
Installation Factor	1,2	1,2	1,5	1,5	2	
Indirect Cost Factor	1.34	1.34	1.34	1.34	1.34	
Contingency and Contractor Fees Factor	1.18	1.18	1.18	1.18	1.18	
Capital Cost ($)	593 335	650 075	415 065	612 168	430 595	2 701238
Operation Cost Factor	0.05	0.05	0.05	0.05	0.05	
Amortization Period (y)	5	5	5	5	5	
Utilisation (h/y)	8000	8000	8000	8000	8000	
Annual Amortization Cost ($/y)	118 667	130 015	83 013	122 434	86 119	540 248
Annual Electricity Cost ($/y)	205 031	88 409				293 440
Annual Operation Cost ($/y)	29 667	32 504	20 753	30 608	21 530	135 062
Total Annual Cost ($/y)	353 365	250 927	103 766	153 042	107 649	968 750
Specific.Cost ($/$kWh_{th}$)	0.0192	0.0137	0.0056	0.0083	0.0059	0.053
Spec. Cost ($/GJ)	5.34	3.79	1.57	2.31	1.63	14.65
Spec. Cost (GJ eff. 70%)						10.25
Spec. Cost ($/NG m^3 eff. 70%)						0.39

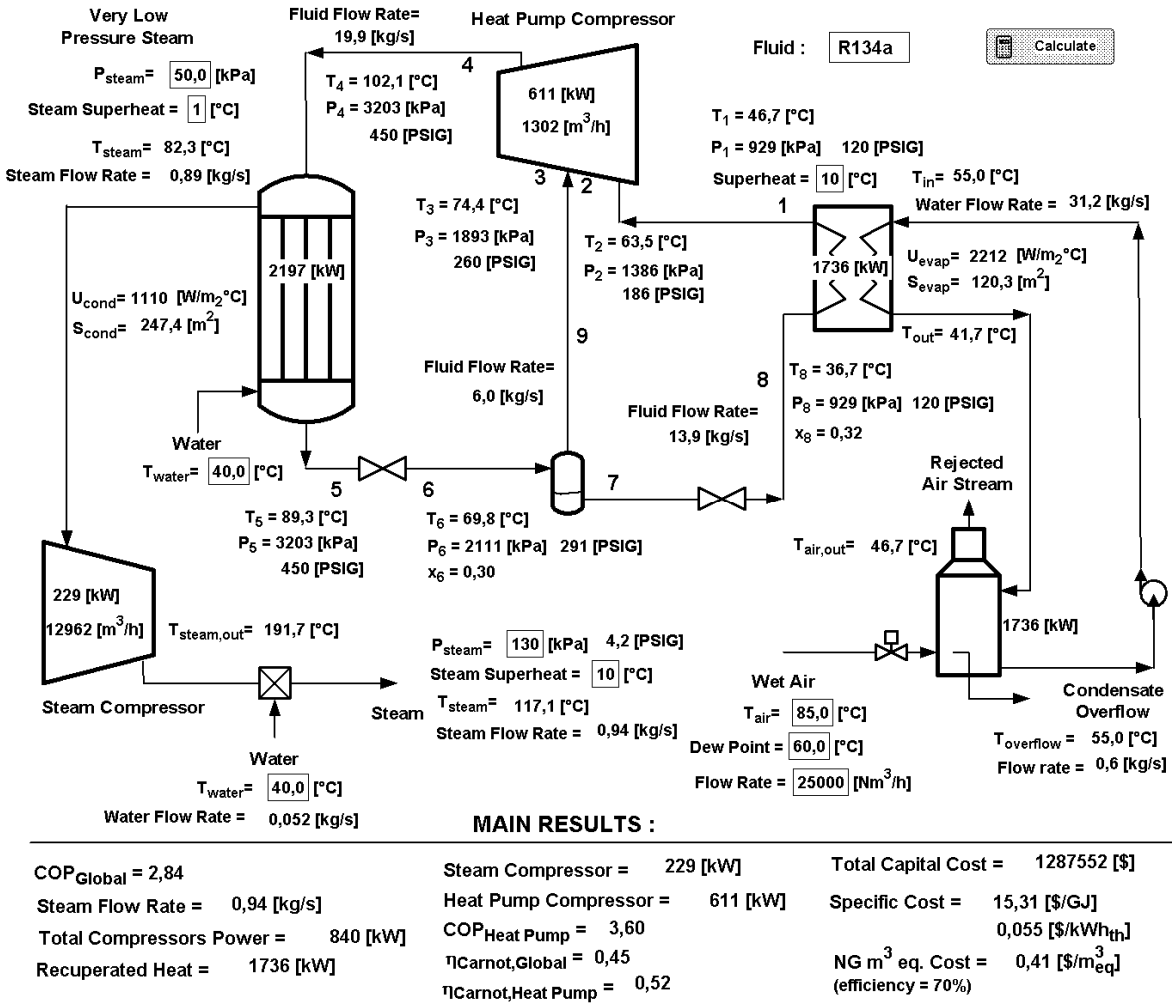


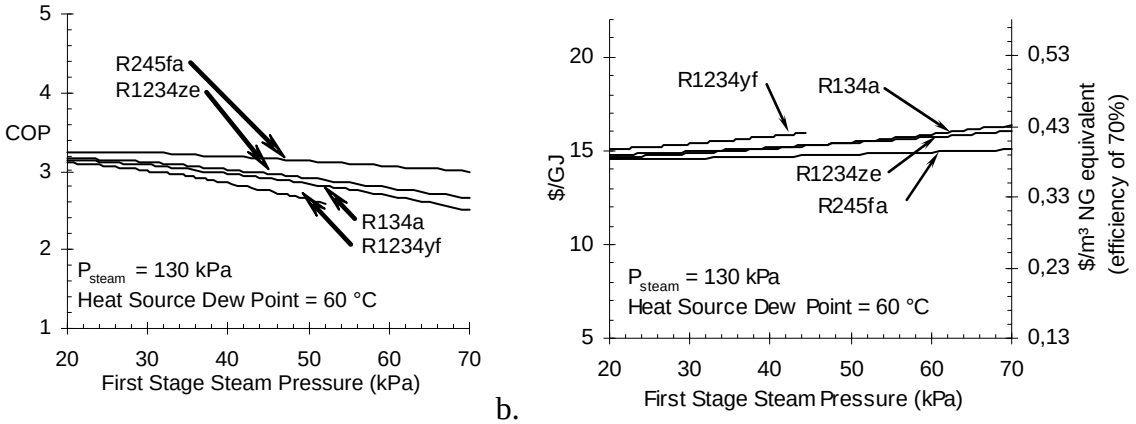
Figure 3: Schematic of system based on HP and mechanical steam compression with R-134a for production of 2.4 MW of heat (steam at 130 kPa and 117 °C)

Table 3: Economic evaluation for a system based on HP and mechanical steam compression with R-134a for production of 2.4 MW of heat (steam at 130 kPa and 117 °C)

	HP Comp.	Steam Comp.	Direct-HX	Evapora-tor	Conden-sor	Total
Key Parameter	611 kW	229 kW	1736 kW	120 m^2	247 m^2	
Delivered Equipment Cost ($)	354 132	347 461	175 000	237 783	173 177	1 287 553
Installation Factor	1.2	1.2	1.5	1.5	2.0	
Indirect Cost Factor	1.34	1.34	1.34	1.34	1.34	
Contingency and Contractor Fees Factor	1.18	1.18	1.18	1.18	1.18	
Capital Cost ($)	671 944	659 286	415 065	563 974	547 655	2 857 925
Operation Cost Factor	0.05	0.05	0.05	0.05	0.05	
Amortization Period (y)	5	5	5	5	5	
Utilisation (h/y)	8000	8000	8000	8000	8000	
Annual Amortization Cost ($/y)	134 389	131 857	83 013	112 795	109 531	571 585
Annual Electricity Cost ($/y)	244 300	91 744				336 044
Annual Operation Cost ($/y)	33 597	32 964	20 753	28 199	27 383	142 896
Total Annual Cost ($/y)	412 286	256 565	103 766	140 993	136 914	1 050 525
Specific.Cost ($/$kWh_{th}$)	0.0216	0.0135	0.0054	0.0074	0.0072	0.055
Spec. Cost ($/GJ)	6.01	3.74	1.51	2.05	2.00	15.31
Spec. Cost (GJ eff. 70%)						10.71
Spec. Cost ($/NG m^3 eff. 70%)						0.41

3.2 Other parameters

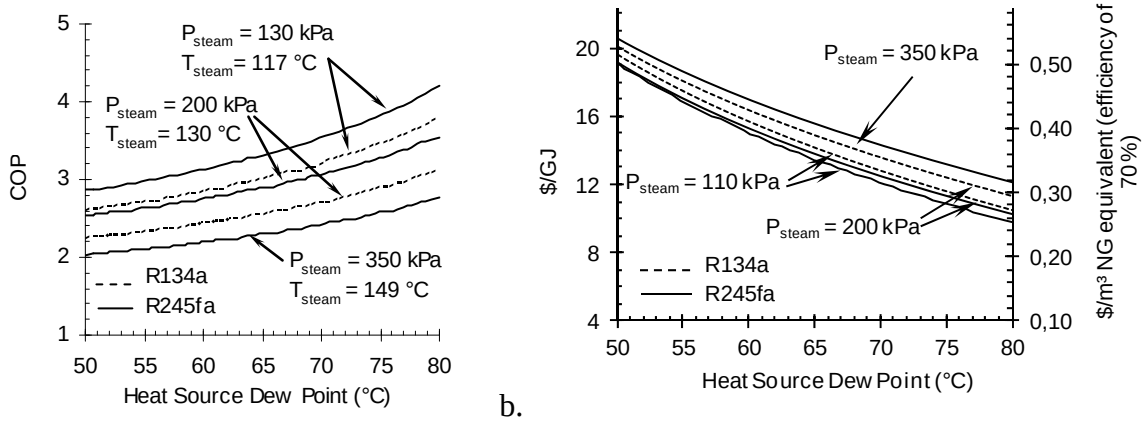
The fist stage steam pressure of the system presented above was set to 50 kPa (82 °C). Figure 4 shows the impact of this parameter on the system efficiency and economics for various HP fluids. The repartition of the load on either the HP or the steam compressor does not affect significantly the efficiency or the cost of the system for R-245fa. Technical considerations on the available equipments (size, pressure ratio, max. pressure, resistance to vacuum, etc.) should determine the final choice. R-245fa shows the best performances, but the other fluids (R-134a or low GWP index fluids such as R-1234ze and R-1234yf) constitute valid options.



a. b.

Figure 4: Impact of First Stage Steam Pressure on Efficiency (a) and Economics (b)

Figure 5 shows the impact of heat source dew point on the production of various steam pressures. The specific cost of the 350 kPa steam is only about 15 % higher than for 130 kPa steam. However, the wet air stream dew points have a significant impact on the system efficiency and steam production cost. A higher dew point does not only provide a higher temperature, but a significantly higher humidity ratio (i.e. energy content). The specific cost of 130 kPa steam decreases by 28 % with a heat source dew point at 75°C instead of 60°C (10.6 $/GJ instead of 14.7 $/GJ) and the COP rises from 3.1 to 3.8. If hot water is used directly as the heat source, the cost would decrease by 10% and it should be considered 5 °C higher to fit with Figure 5.

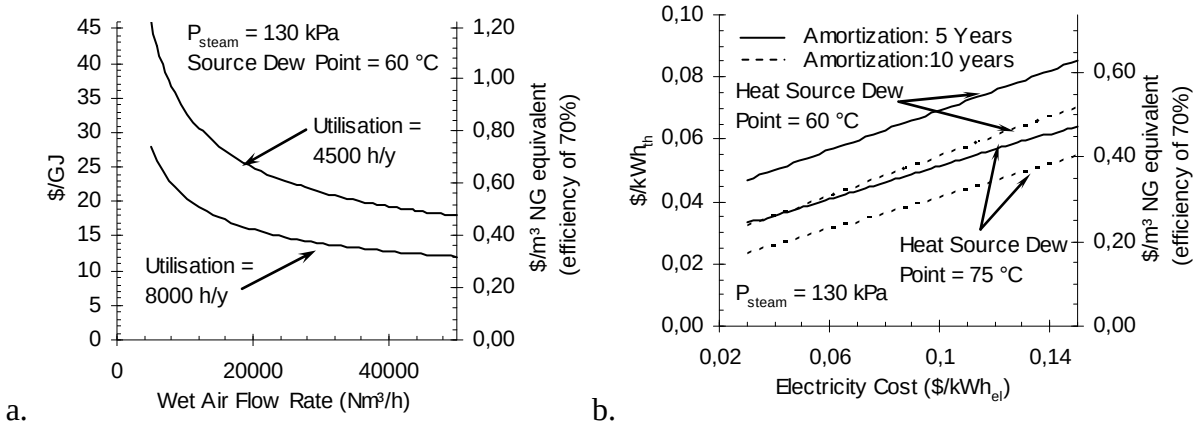


a. b.

Figure 5: Impact of Heat Source Dew Point on Efficiency (a) and Economics (b)

As shown in Figure 6a, the capacity and the utilisation have a significant impact on the specific cost of the steam produced. The wet air stream flow rate should not be much lower than 25 000 Nm^3/h, the flow rate used in this study. Ideally, the steam production process should also be continuous (8000 hours per year). Figure 6b presents the impact of electricity

rates and amortization period on the specific cost of 130 kPa steam. Steam specific cost is not much affected by electricity rate fluctuations. When the amortization period is 10 years instead of 5 and the air stream dew point is 75 °C, the system is more economically interesting with a specific cost of less than 8 $/GJ (0.21 $/m³, natural gas at 70 % efficiency).



a. b.

Figure 6: Impact of Wet Air Flow Rate and Utilisation (a) and Electricity Cost (b)

4 CONCLUSIONS

This preliminary techno-economic evaluation suggests that 130 kPa steam (117°C) can be produced from a wet air stream (dew point 60°C) by the combined use of a mechanical heat pump and mechanical steam compression with a COP higher than three and at a cost of 15 $/GJ. This is equivalent to 10.5 $/GJ of combustible (ex: natural gas) used at 70 % efficiency (0.40 $/m^3). After the 5 year simple amortization period, the price would decrease by more than 50 %.The study suggests that the steam pressure could be raised to 350 kPa without much affecting the steam specific cost (15% raise). The waste wet air stream dew point has a significant impact on the system efficiency and economics: at 75°C instead of 60 °C , the COP rises to 3.8 and the specific cost decreases to 10.5 $/GJ. When an amortization period of 10 years instead of 5 is used, the specific cost decreases to 8 $/GJ (equivalent to 0.21 $/m^3 natural gas at 70 % efficiency). Steam cost is not much affected by electricity rate fluctuations. The capacity and the utilisation have a significant impact on steam cost: the wet air stream flow rate should ideally be higher than 20 000 Nm^3/h and the steam production process should be continuous (8000 hours per year). The authors would like to underline that, given the lack of precision of such techno-economic evaluation, the data presented here should only be interpreted as rough approximations and tendencies. For a better cost estimate, a study with actual cases and actual costs for specific project is necessary.

Although a sequence combining a HP and mechanical steam compression for the production of low pressure steam as a hot utility from waste wet air stream is currently hardly economically justifiable in a context of low natural gas prices, it could eventually be beneficial, especially where renewable energy is available and where the price of electricity compares to fuel prices.

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