Experimental Results of a Newly Developed Very High Temperature Industrial Heat Pump (140°C) Equipped With Scroll Compressors and Working With a New Blend Refrigerant

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ABSTRACT

During last decades more and more attention has been paid on CO_2 emissions. One of the solutions for decreasing CO_2 emissions concerns the substitution of fossil fuels industrial boilers by the use of very high temperature electrical heat pumps. Moreover, according to the Kyoto and Montreal protocols, the CFC and HCFC are or will be forbidden. In that context, the developments of industrial heat pumps (HP) and new working fluids with high critical temperatures are necessary. In this paper, the main refrigerants types are considered for very high temperature heat pump applications (i.e. natural fluids, HFC and HFO). Performance calculations and CO_2 emissions impact are presented for some potential interesting fluids for high temperature HP applications. In a second time, the paper demonstrates the technological feasibility and reliability of a newly developed very high temperature heat pump using a new blend as working fluid. This industrial heat pump can supply hot water up to 140°C, or low-pressure steam. The performances of the machine are characterized and reliability is demonstrated by an ageing test campaign.

1. INTRODUCTION

As a result of the worldwide increased energy consumption, energy efficiency becomes more and more important for industrial sector. In industry, the heat consumption is the first type of final energy consumption. Particularly, in the French industry, about 70 % of final energy use is for thermal purposes (furnaces, reactors, boilers, dryers, etc.). The major part of that heat is produced by combustion of fossil fuels which generates large CO_2 emissions and a large part of this energy is wasted through losses. The utilization of waste heat is an additional possibility of saving energy, leading to enhance the efficiency of industrial processes. Substitution of boilers by industrial heat pumps offers a double benefit: the reduction of the energy consumption and the reduction from 60 to 95 % of CO_2 emissions. The amount of avoided CO_2 will depend on the considered country kWh CO_2 content and the fed boiler combustible type (gas, fuel or coal).
A first experimental development of high temperature industrial heat pump has been performed by Bobelin and Sapora (2011) [1]. Experiments have been conducted up to 100 °C using R245fa as working fluid. In order to reach higher energy efficiency potential, in the framework of the present paper, condensation temperature up to 140 °C is aimed.
The chemistry and Environment competitiveness pole Axelera allowed to federate different partners around a collaborative project named ALTER ECO (Analysis Low Temperature Energy Recovering / ECOnomy). This project aimed to develop clean and\or energy-efficient technologies for industrial low-grade waste heat recovery. The paper will describe the newly developed very high industrial heat pump (i.e. T_{cond} = 140 °C) using a new blend refrigerant which fulfils the requirements on very high temperatures and working with adapted compressors. The HP has been characterized in a laboratory flexible industrial scale heat recovery system designed and built to carry out experimental simulations by reproducing the operating conditions of real industrial case applications. Results of experimental test campaign for the characterization of the machine are presented.

2. HEAT MARKET IN THE FRENCH INDUSTRY

In 2009, the final energy consumption of the French industries represents 23 % of the total part of energy consumed in France. This energy arises from a third of electricity and two thirds from fossil fuels. Furthermore, Around 70 % (i.e. ~300 TWh) of the energy used in the French industry is for thermal purposes. Regarding CO_2 emissions, France rejected 365 megatons and 24 % of these emissions are due to industrial activities (Berail and Sapora 2010)[2].
With a good energy management and by using the best available technologies, the energy savings potential are distributed according to figure 1. We can observe that 28 % of the energy savings potential is linked to the waste heat recovery (i.e. boilers, chillers, dryers, etc.).

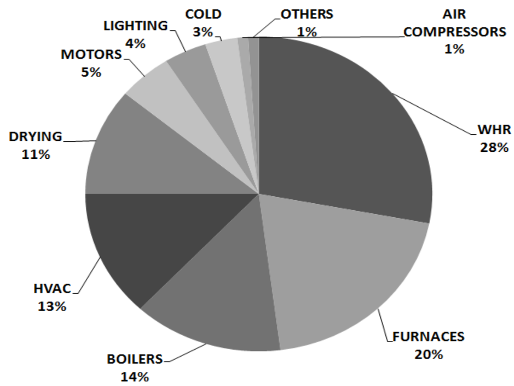

Figure 1: Distribution of the energy saving potential in the French industry [2]

The temperature range in industry is large so that we considered only the following temperature range: 60 – 140 °C. Indeed, the opportunities and the potential of recovering the heat especially at low-temperatures to produce high temperatures, by heat pump is the target of our study.
Dupont and al.(2009) showed that in France pulp and paper and food industries are the largest consumers of low-temperature industrial heat [3]. The distribution of heat needs at temperatures in the 60 – 140 °C range for eight main industrial branches and six usages is presented in figures 2 and 3.

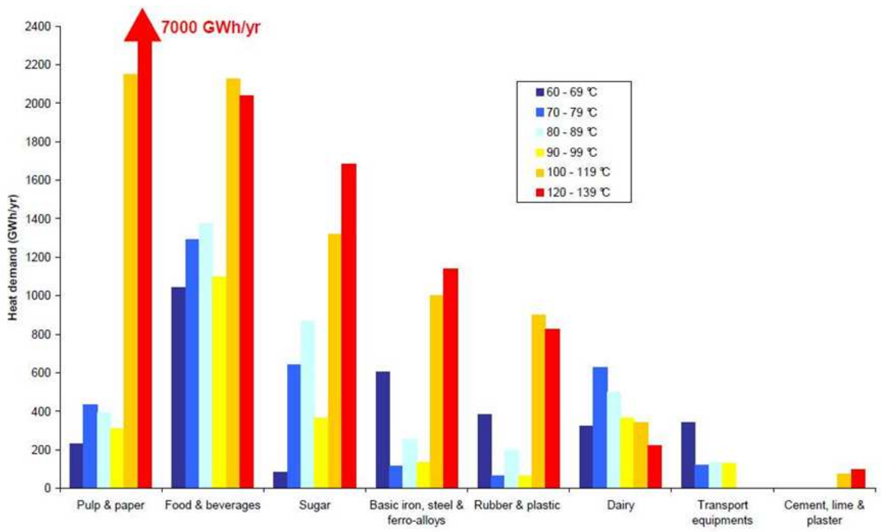

Figure 2 : Distribution of the heat demand per industrial branch [3]

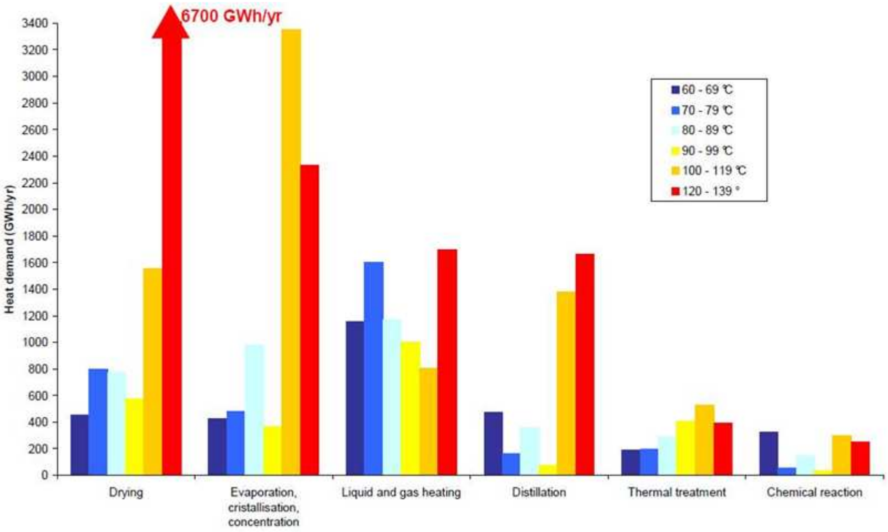

Figure 3 : Distribution of the heat demand per industrial usage [3]

Both graphs show that the total demand of heat at 60 - 140 °C range for the considered 8 industrial branches and the 6 usages is estimated at around 33 TWh/year.

At these temperature levels, many opportunities for heat pump technology exist in the industry. Nevertheless, the rise in temperature is necessary to reach high-temperature process needs (i.e. process steam between 2 and 4 bars). The objective of the present work is the development and characterization of a very high temperature industrial heat pump.

3. EXPERIMENTAL TEST BENCH

3.1 Experimental hydraulic loops

In perspective to test the developed industrial heat pump, two hydraulic loops have been constructed at EDF R&D laboratory. The high temperature loop allows simulating the process heat requirement. This circuit is equipped with a pump and variable capacity dry cooler. The low temperature hydraulic loop simulates the process waste heat. Both loops include a water tank, a controlled electric heater, and a water pump with adjustable volume flow rate. Furthermore, the system includes a counter-current plate type heat exchanger for primary heat recovery before the heat pump. This test bench is briefly presented in figure 4 and more detailed by Assaf and al. (2010) [4].

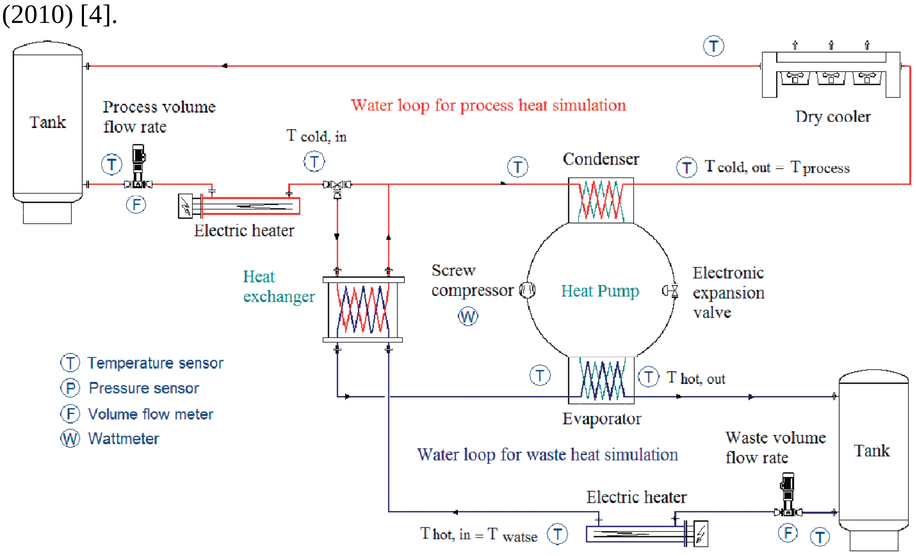

Figure 4: Scheme of the experimental set-up

These hydraulic loops are composed of several sensors: temperature transducers PT100 (0 – 200 °C range ±0.5 K), electromagnetic flow meters (± 0.25 % in the operating range of the experimental conditions). Their locations are shown in figure 4. All sensor measurements are collected at steady state conditions using a dedicated PC via convenient data acquisition software.

3.2 Very high temperature industrial heat pump

An electrically-driven vapor compression heat pump has been developed in the framework of a collaborative project ALTERECO. Figure 5 shows a schematic diagram of the experimental apparatus. The heat pump technology was designed for very high temperature applications.

The main components of the heat pump, provided by project partners, are listed below:

- A new blend refrigerant (non-toxic, non-flammable, environmentally safe) as working fluid for very high-temperature applications called ECO3^{TM}.
- AON scroll compressors.
- Brazed plate heat evaporator.
- Brazed plate heat condenser.
- Brazed plate heat subcooler.
- An electronic expansion valve.

In order to allow an easy industrial implementation and in-situ field tests, the heat pump has its own measurement points (temperatures, mass flow rates) and acquisition data system. Moreover, a wattmeter to measure the compressor input power with an accuracy of ±0.5% in the operating range is implemented.

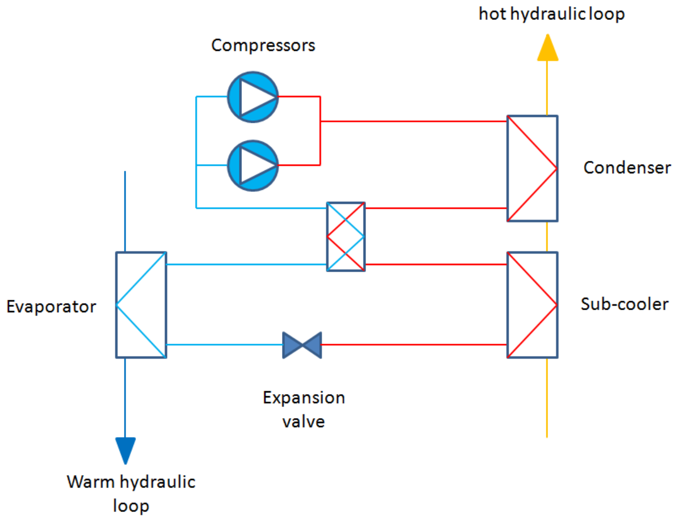

Figure 5: Scheme of the ALTER ECO heat pump

3.3 Working fluids overview

Working fluids have for role to assure the energy transfer from the heat source (by evaporating) to the heat sink (by condensing). They are chosen according to the condensation temperature. For medium-temperature HP (i.e. lower than 80°C), the traditional used fluid is the R134a, which presents a GWP of 1430 kg eq. The new fluid R1234yf which has a GWP of 4, is considered as the direct replacement of the R134a (i.e. mobile air conditioning applications). For high temperature HP, the existing fluids R227ea, R236fa, R245fa, R365mfc, R717 (i.e. NH_3), etc. can be used.

Table 1 below raises a list of potential working fluids for high temperatures HP applications.

Table 1: Working fluids properties for high temperature heat pumps

Fluids	**Chemical composition**	**$T_{critical}$ (°C)**	**$P_{critical}$ (bars)**	**GWP**	**Safety group**
ECO3™	-	-	-	980	B1[1]
R1234ze	C3H2F4	109	36.4	6	A2
R245fa	C3H3F5	154.1	36.5	950	B1
R365mfc	C4H5F5	186.9	32.7	910	A2
R236fa	C3H2F6	124.9	32	9600	A1
R717	NH_3	129.2	113.33	0	B2

The critical temperature is not the only criterion to estimate the capacity of a fluid to work with high temperatures of condensation. The pressure of condensation is also a criterion of selection of a high-temperature fluid because of the limitations of pressure.

The REFPROP software (Lemon et al., 2010)[5] is used for fluids properties calculations. The calculation hypotheses are presented in the following table 2:

Table 2 : Calculation hypotheses

Modules	**Characteritics**	**Values**
T_{evap}	Temperature	50 °C
T_{cond}	Temperature	100 °C
Evaporator	Undercooling	2 °C
Condenser	Superheating	10 °C
Compressor	$\eta_{isentropic}$	0.7

Simulations according to the scenario of table 2 are performed with five working fluids: R1234ze, R245fa, R236fa, R365mfc, ECO3™ and R717.

For the heat pump, greenhouse gas emissions are estimated by the calculation of TEWI (Equivalent Total Warming Impact). The TEWI is defined as being the sum of the direct incidence of the emissions of fluids in equivalent CO_2, and of the indirect incidence of the emissions of CO_2 (due to the primary energy used for the functioning of the heat pump). It allows calculating globally the greenhouse gas emissions of a heat pump on a whole life cycle. TEWI expression is:

$$TEWI = (GWP \cdot \tau \cdot m \cdot n) + (GWP \cdot m \cdot (1 - \alpha_{recovery})) \qquad \textbf{Direct emissions}$$

$$+ \left(n \cdot \frac{H \cdot P}{COP} \cdot m_{elec} \right) \qquad \textbf{Indirect emissions} \qquad (1)$$

For calculations, the following hypotheses are used: τ is the annual leakage rate (2 % per year); n is the heat pump life time (20 years); m is the fluid charge (27 kg); $\alpha_{recovery}$ is the fluid rate that will be recovered after the HP dismantling; H is the annual operating time (8000 hours); P is the considered thermal power (200 KW); m_{elec} is the equivalent CO_2 emissions per kWh (55 gCO_2/kWh[2]). Table 3 below summarizes the main results of COP, working pressures and TEWI.

[1] Not yet standardized by ASHRAE.

[2] This value corresponds to a French industrial use except lighting. This value will be different for other countries.

Table 3 : Calculation results

Characteristics	P_{evap} (bar)	P_{cond} (bar)	COP real	TEWI : direct (tons of CO_2)	TEWI : indirect (tons of CO_2)	TEWI total (tons of CO_2)
ECO3™	3	11.41	4.43	11.11	397.29	408.4
R245fa	3.43	12.60	4.35	10.77	404.6	415.37
R1234ze	9.94	30.17	3.57	0.07	493	493.07
R365mfc	1.42	5.85	4.45	10.32	395.5	405.82
R236fa	5.82	19.33	3.8	108.86	463.16	572.02
R717	20.26	62.37	4.32	0	407.4	407.4

In terms of TEWI, we can see that R365mfc and R717 exhibit better results than ECO3™. However, R717, as natural refrigerant, having a dominant role in industrial refrigeration, has safety risks (toxicity) and higher evaporation and condensation pressures.
R365mfc, although giving the best performance, due to its safety class A2, has also not been retained.
R236fa exhibits the lowest performance and its TEWI is the highest one. It is therefore not retained here.
Furthermore, R1234ze characterized by a very low GWP (i.e. 6), due to its relatively low thermodynamic performance leads to a high indirect equivalent warming impact (i.e. due to the heat pump energy consumption), allowing to disqualify it.
Finally, even if the GWP of the ECO3™ is slightly higher to that of the R245fa, the improvement of the performance by around 2 % within these conditions allows obtaining a lower TEWI with ECO3™.
For the same calculation scenario, this first comparison between calculated HP performances using different working fluids shows that the choice of the ECO3™ is therefore completely justified.
However, these theoretical simulations should be experimentally tested to confirm these conclusions.

3.4 Heat pump operating field

The scroll compressors are hermetically sealed. Due to this configuration, the maximal authorized evaporation temperature is 60 °C. This limitation is due to the indispensable cooling of the electrical motor, which is done by the low temperature gas at the compressor inlet. Furthermore, the low limit for evaporation temperature has been fixed at 35 °C. This is explained by the possibility to use other working fluid under this temperature. In addition, the ECO3™ has a saturation temperature of 17 °C at atmospheric pressure. Therefore, to avoid any air entrance in the circuit, this limitation at 35 °C is justified.
The condensation temperature is limited by the working pressure and the pressure ratio. In accordance with the compressor manufacturer, we agreed that we can work up to 140 °C with at least 30 °C of difference between evaporating and condensing temperature. The upper ΔT (i.e. $T_{cond} - T_{evap}$) limit is 80 °C. Within the present experimental test campaign, except for the point T_{cond} = 140 °C, experiments with ΔT up to 75 K have been carried out.

Finally, another limitation is given by the actual expansion valve commercial offer. Currently, the highest acceptable temperature at the expansion valve inlet is 120 °C. For high condensation temperatures, this limitation imposes a high subcooling after condensation. That explains the interest of the second exchanger on the high pressure part of the HP circuit (see figure 5).

4. RESULTS AND DISCUSSION

4.1 Test program

The aim of the test campaign was to demonstrate the technical feasibility of a very high temperature heat pump. To reach this goal, two different tests have been carried out: performances mapping and endurance tests.
The aim of the performances mapping tests is to identify the characteristics of the heat pump in the considered operating field. For each evaporation temperature, the condensation temperature is increased by step of 5 °C. Once all the reachable condensation temperatures are characterized, the evaporation temperature is then

increased by step of 5 °C. In order to avoid a transitory effect on our measurements, a stabilization time of 90 minutes is made between two points. During the last ten minutes of each step, the measurements are recorded each second.
In order to demonstrate the reliability of the machine, endurance tests have been performed in industrial-like conditions. These tests have been conducted for an evaporation temperature of 60 °C and condensation temperature of 120 °C. This condensation temperature allows producing low pressure steam (i.e. 2 bars gages) which is commonly used in food and paper industries.
The test program mapping is presented in figure 6. The endurance experimental point (60 °C → 120 °C) and the simulation point presented in section 3.3 (50 °C → 100 °C) are shown.

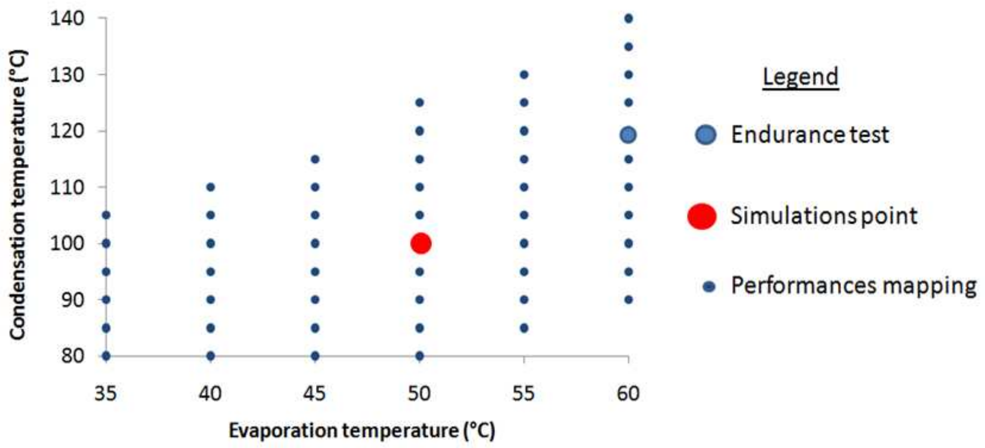

Figure 6 : Test program

4.2 Experimental results

The HP has been characterized according to the calorific power delivered at the condenser, Q_{cond}, A particular attention are paid on some fundamental parameters. Those parameters are listed bellow with their formulae. The measured COP is given by the formula according to both powers:

$$COP_{experimental} = \frac{Q_{cond}}{P_{elec}} \tag{2}$$

The expression of the Carnot cycle (isentropic compression and expansion, isothermal condensation and evaporation) for a pure body is given by the following classic formula:

$$COP_{carnot} = \frac{T_{cond}}{T_{cond} - T_{evap}} \tag{3}$$

$$Q_{cond} = \dot{m} * C_p * (T_{outlet} - T_{inlet}) \tag{4}$$

$$\eta_{exergetic} = \frac{COP_{experimental}}{COP_{carnot}} \tag{5}$$

The tests results are shown in figures 7. The figure 7 (right) shows the delivered calorific power at the condenser and subcooler according to the evaporation and condensation temperatures. The corresponding heat pump $COP_{experimental}$ are presented in figure 7 (left).

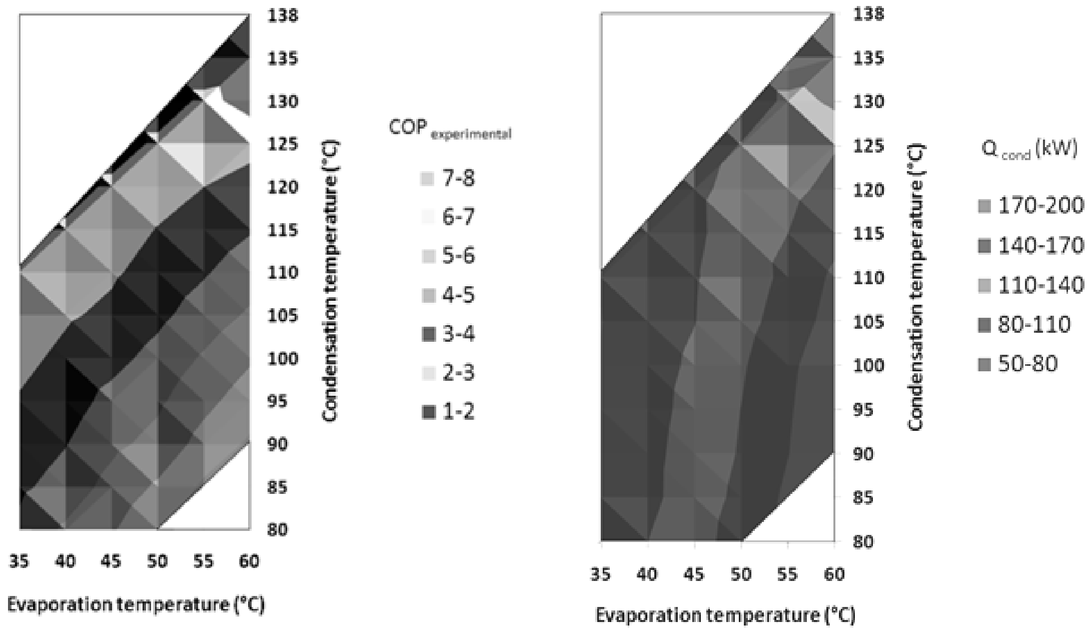

Figure 7 : (Left): HP COP and (right): delivered power at condenser (in kW) depending on evaporation and condensation temperature (in °C)

Good heat pump performances are obtained. For the following conditions: T_{evap} = 50 °C and T_{cond} = 100 °C, a COP of 4.48 is obtained. This measured $COP_{experimental}$ corroborates the calculated COP presented in section 3.3.
A loss of efficiency can be pointed out for the high condensation temperatures. The compressor being designed for a given pressure ratio, that phenomenon is due to a high pressure ratio between high and low pressure parts of the circuit.
The figure 8 shows the exegetic efficiency according to the condensation and evaporation temperatures.

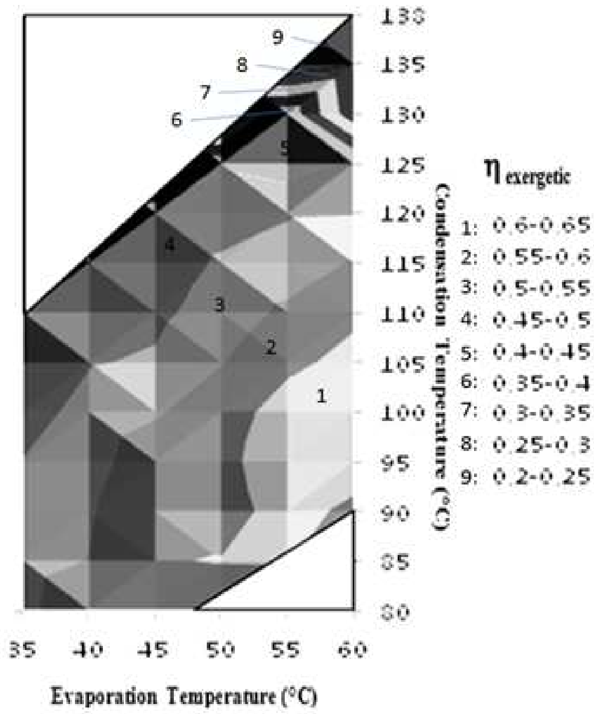

Figure 8 : diagram exegetic efficiency depending evaporation and condensation temperature (in °C)

For temperature differences between evaporation and condensation up to $\Delta T = 60$ K, the diagram shows good heat pump efficiencies. For higher temperature differences $\Delta T > 60$ K, the heat pump efficiency decreases. This can be explained by the fact that the pressure ratio between evaporation and condensation increases significantly and Scroll compressors can't maintain a good efficiency in those conditions.
Furthermore, it can be easily observed in figure 8 that the heat pump performances drastically decrease for condensation temperatures above 125 °C. This effect is due to the fluid properties and particularly due to the decreasing latent heat at higher temperatures. For example, at a pressure of 26.694 MPa, the latent heat represents only 58.7 % of that at 1.202 MPa. Both combined phenomenon explain the machine performances deterioration.
According to the test program defined in section 4.1, the ageing tests were conducted during 1000 hours at full load. The tests do not show any significant variation during heat pump operation and the performances at the beginning and at the end of the test campaign are similar. This demonstrates the prototype reliability and the capacity to use this newly developed machine for industrial purposes.

5. CONCLUSIONS

Regarding the huge industrial heat market, the attention paid on CO_2 emissions and the energy price increase, waste heat recovery by industrial heat pumps presents great energy efficiency potential. In this context, a new very high temperature industrial heat pump has been designed, constructed and characterized. This machine is able to provide efficiently heat up to 125 °C. The technological reliability has been demonstrated.
For higher temperatures, between 125 °C and 140 °C, the technological feasibility is demonstrated and objectives are reached. For this high temperature range (i.e. 125 – 140 °C), some further developments have to be carried out in order to increase the efficiency of the technologies, and consequently a better economical viability.
The needed developments are:

- Compressors with better efficiency for high pressure ratios.
- Expansion valve allowing working at higher temperatures (i.e. T > 120 °C).

These developments will be conducted in a near future.

NOMENCLATURE

AON	All or None regulation
COP	Coefficient of performance
CFC	Chloroflurocarbons
Cp	Specific heat of water
HCFC	Hydrochlorofluorocarbons
HFC	Hydrofluorocarbons
HFO	hydrofluoroolefins
HP	Heat pump
m	Water mass flow rate (kg/s)
P_{elec}	Electrical power to the compressor (W)
Q_{cond}	Thermal power delivered at the condenser (W)
T_{cond}	Temperature of condensation (°C)
T_{evap}	Temperature of evaporation (°C)
T_{inlet}	Water temperature entering the heat exchanger (°C)
T_{outlet}	Water temperature leaving the heat exchanger (°C)
TEWI	Total Equivalent Warming Impact
WHR	Waste heat recovery
$\eta_{exergetic}$	Exegetic efficiency
$\eta_{isentropic}$	isentropic efficiency

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