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ABSTRACT

Currently, improving energy efficiency becomes a main challenge for all industrial energy systems. This challenge involves an improved recovery of wasted heat generated by several industrial processes. Large energy savings and potential environmental benefits are associated with the use of industrial heat pump mainly at high temperature levels. A laboratory flexible industrial scale heat recovery system is able to reproduce the operating conditions of real case simulating energetic losses and requirements in high temperature industrial applications. The integrated heat pump is an electrically-driven vapor compression using a twin screw compressor. Water vapor has been adopted as a working fluid using a modified screw compressor to carry out a dry compression process at high temperature levels. This heat pump generates vapor using flash evaporation. A purging valve is implemented in order to eliminate non-condensable gases present in system. Experimental simulation of the start-up phase has been presented showing the non-condensable purging process and the evolution of some parameters of the heat pump. Several scenarios of industrial processes for high-temperature heat recovery (heat sources between 80°C and 90°C) and heat upgrading are numerically simulated. The presented results show the global energy savings and the environmental benefits of using water as refrigerant at high temperature levels.

1. INTRODUCTION

Currently, improving energy efficiency becomes a main challenge for all energy systems due to the sharp increase of fuel prices (FY, 2010). This challenge involves a reduction of industrial processes fuel consumption by a better recovery of process waste heat and using it to meet heating requirements.

In addition to energy savings, the environmental context is introduced by further international and local agreements that involve a stronger awareness of environmental concerns especially by emission reduction. Large energy savings and potential environmental benefits are associated with the use of industrial heat recovery systems mainly at high temperature levels.

Several market studies assess a large amount of waste heat rejections in the range 80°C-90°C in different industrial sectors like chemistry, paper drying, cleaning in place, … However, several applications need heat at higher range of temperature, typically 120°C-130°C (EDF R&D, 2009, Dupont and Sapora, 2009, US DoE, 2003). A heat pump technology is able to upgrade these wastes to make it useful in higher temperature applications.

Presently, implemented heat pumps in industries are limited to heat production at a temperature level of 100 °C. Several studies try the development of new high temperature heat pumps using newly developed synthetic refrigerant which still harmful to the environment. Therefore the development of a new industrial high temperature heat pump using a natural refrigerant becomes a step to improve industrial energy efficiency while reducing environmental impact which is the goal of the present work. One of the major problems with the development of this high temperature compression heat pump is the choice of the working fluid and corresponding compressor that can be used for these high ranges of temperatures.

In this work, experimental simulations reproducing industrial scale operating conditions are carried out to evaluate the potential of an electrically-driven vapor compression heat pump using water as refrigerant. Technological choices, test bench and experimental simulation results have been presented for the start-up phase of the machine. The calculated performance of this machine shows the global energy savings and the environmental benefits of using water as refrigerant at high temperature levels.

2. TECHNOLOGICAL CHOICES FOR THE HIGH TEMPERATURE HEAT PUMP

2.1 Working fluid

The choice of the working fluid for this high temperature heat pump should takes into account several considerations like environmental, economical, safety, efficiency, thermodynamic properties. In general, synthetic refrigerants (R134a, R245fa, R1234yf …) dominate vapor compression refrigeration systems. Calm (2008) breaks history into four refrigerant generations based on defining selection criteria. He discusses the succession of criteria that involved a displacement of earlier working fluids and the renewed interest in natural refrigerants. Several studies (Chamoun *et al.,* 2011, EDF R&D, 2009) selected water vapor as refrigerant for high temperature heat pump applications.

In refrigeration applications, the use of water has been limited to absorption systems around a binary fluid. This natural fluid is a very attractive refrigerant, lots of studies compared water to other refrigerants from different point of views (Chamoun *et al.*, 2011, Kilicarslan and Muller, 2005). Others made investigations and discussed the feasibility of a refrigerating machine using water vapor as refrigerant (Van Orshoven, 1991, Lachner *et al.*, 2007, Wight, 2000).

Water vapor is an environmental friendly refrigerant with no Ozone Depletion Potential (ODP) and a highly reduced Total Equivalent Warming Impact comparing to synthetic refrigerants. It is non-toxic, non-flammable, chemically inert at high temperature and needs low compressing pressures. Water has a high critical temperature therefore its thermodynamic properties suites high temperature ranges. Otherwise, it is easily available with low costs and presenting high theoretical coefficient of performance (COP) due to its high latent heat of vaporization compared to other traditional refrigerants. Hence, the use of water as refrigerant in this high temperature heat pump offer several potentially significant advantages but involves technical and feasibility difficulties especially related to the compressor technology (Chamoun *et al.,* 2012).

2.2 Compressor

Except expensive compression machinery, EDF R&D (2009) shows that currently, water vapor compression at high temperature levels can be performed by several industrial compressors:

- blowers allow compression of high mass flow rates but under low compression ratios (corresponding to saturated temperature difference ΔT of 5 to 7 K).
- multi-stages blowers allow an increase of ΔT to 10 to 12 K.
- lobe compressors are adapted to low flow rates but with higher ΔT to 20 K. These compressors have a poor isentropic efficiency.
- classical centrifugal compressors (mechanical bearings) can be used in double stage compression process to attain a maximum ΔT of 40 K. These machines are very expensive and work at high rotational speeds at high compression ratio. At these conditions, high level of maintenance is required. In addition, reduced reliability of these compressors is due to the use of mechanical bearings and imperfect sealing at these conditions.

Therefore, the largest obstacle is associated with the compressor technology. Adapted compressor should simultaneously satisfy different requirements at high temperature levels like:

- high compression ratio corresponding to ΔT of 40 K
- high volumetric flow capacity
- high isentropic efficiency

To overcome this compression problem, an ANR (Association Nationale de la Recherche) project named PACO launched the development of a new twin screw compressor. This compressor is an air compressor specially adapted to water vapor which could satisfy this heat pump demand with some modifications (water injection, sealing …). This type of volumetric compressor presents high compression ratio (corresponding to 40 - 50 K), sufficient isentropic and high volumetric efficiencies. This electrically-driven vapor compressor provides a dry compression in the heat pumping application by a single stage compression at variable speed.

2.3 Other components

At these temperature levels, water evaporating pressure is below atmospheric pressure which implies high specific volume in the evaporation process. Hence, the heat exchangers should be large enough to allow phase change. To avoid large dimensions of heat exchangers, a coupled liquid-liquid plate heat exchanger with a flash evaporation system is implemented to assure evaporation process. In addition, the condenser is a plate and gasket heat exchanger with reduced pinch.

A substantial amount of air will have to be removed upon start-up phase of the machine. An additional amount of incoming air depends on many parameters related to the specifics and the hermeticity of the overall system. The presence of air degrades heat exchanger performance and places an additional load on the compressor. Therefore, a significant amount of non-condensable gas in the refrigerant results in a loss of cycle performance and so the non-condensable gases must be purged. An accumulator with a purging valve is implemented at the end of the condensing process to eliminate non-condensable gases existing at the start-up of the machine.

3. TEST BENCH DESCRIPTION

The experimental set-up is composed of two main parts: the heat recovery heat pump system and the hydraulic circuit. The designed heat pump is implemented to be tested in an industrial environment imposed by the hydraulic system which simulates industrial wastes and requirements.

3.4 Heat pump circuit

In this refrigerating cycle, the evaporative system generates vapor (state 1) into the inlet of the double screw compressor, this vapor flow is mixed to an injected water flow (state 6) in the suction chamber of the compressor. This injected mass flow rate is necessary to avoid high superheating in the compression process risky for the compressor. The biphasic mixture is compressed and discharged at higher pressure level at state 2 in a superheated state. Afterward, vapor condenses by releasing heat to the external sink via the condenser to be collected as condensate (state 3) at the inlet of the accumulator.

Then, water (state 4) flows through a ball float steam trap to be expanded into the flash reservoir (state 5) at low pressure. A re-circulating mass flow rate of water (state 6) leaves the tank to absorb heat through the evaporator and re-enters the tank at state 7 to be flashed generating vapor at state 1. Another mass flow rate at state 6 leaves this tank to be injected in the suction chamber of the compressor. A controlled purging valve is implemented at the top of the accumulator in order to eliminate existing air upon start-up phase and air leakages that will escape to low pressure branch of the machine at normal operating conditions.

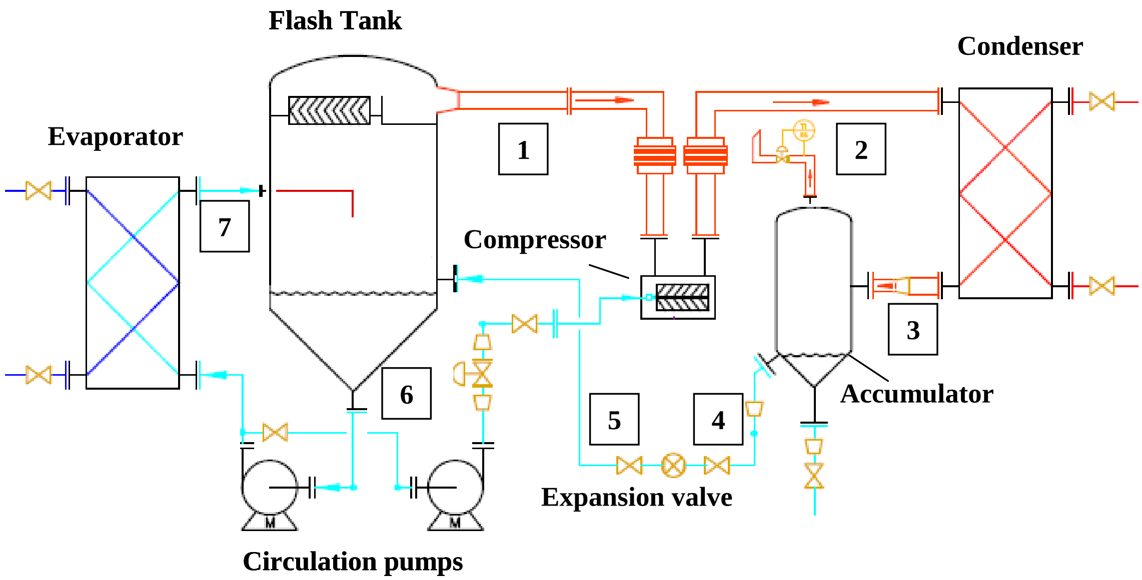

Figure 1. Scheme of the heat pump

3.5 Hydraulic circuit

The industrial operating conditions are simulated using the hydraulic circuit (Khattar *et al.*, 2010). This circuit is composed of two separate loops (figure 2), the first one ''low temperature'' represents industrial waste which delivers energy to the heat pump evaporator. This loop includes a water tank, a controlled electric heater, and a water pump with adjustable volume flow rate. Including the condenser, the second hydraulic loop ''high temperature'' simulates the user's heat requirement that the heat pump has to satisfy. This loop includes a water tank, a variable capacity dry cooler, a controlled electric heater, and a water pump with adjustable volume flow rate.

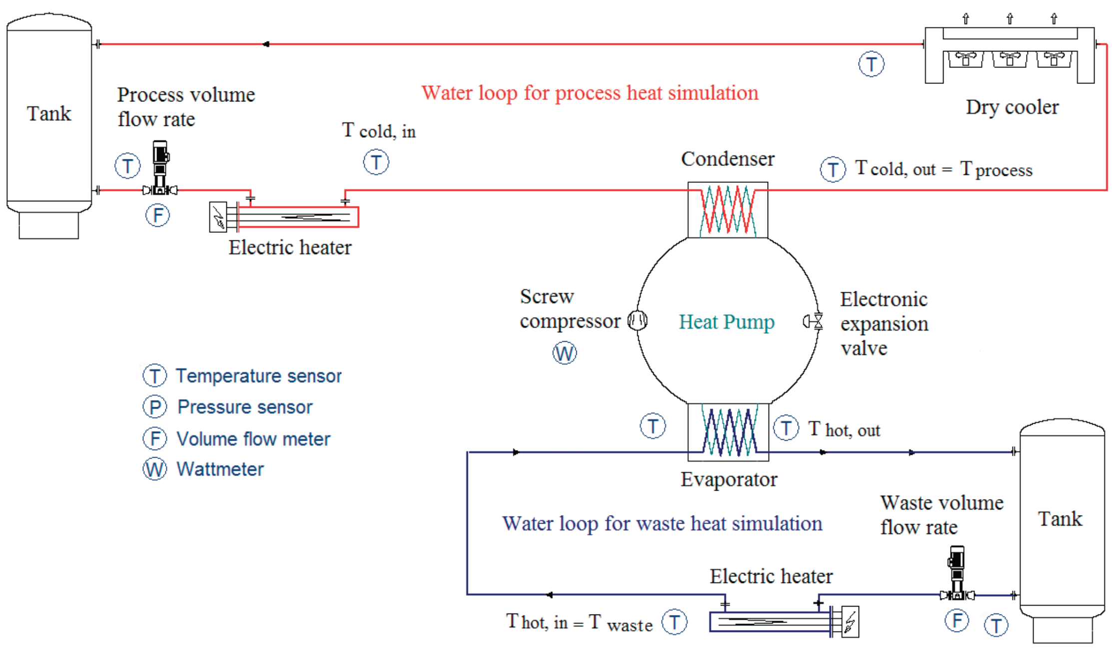

Figure 2. Scheme of the hydraulic loops

4. RESULTS AND DISCUSSION

4.1 Test campaign

The preliminary test was carried out at EDF R&D - EPI Department Renardières laboratory. The presented experimental test is the start-up phase of the heat pump at partial load conditions. In this heat pump configuration, a substantial amount of air will have to be removed upon start-up phase before attaining steady state conditions. This phase was experimentally simulated imposing a constant motor frequency of 500 RPM. Water flow rates on the hydraulic circuit were set to 61 $m^3.h^{-1}$ at 85°C at the inlet of the evaporator and 56 $m^3.h^{-1}$ at 110°C at the inlet of the condenser. These parameters can be adjusted at any desired level by the control features of heat pump and hydraulic circuits.

The dynamic evolution of condensing pressure and temperature are disturbed by the presence of non-condensable gases in the machine. Figure 3 shows measured temperature evolution of air and water vapor mixture in the accumulator. The difference between saturated temperature at condensing pressure and condensing temperature is a function of partial added pressure of air in the system. After 6 minutes, this difference tends to disappear while eliminating air through the air purging valve at the top of the accumulator.

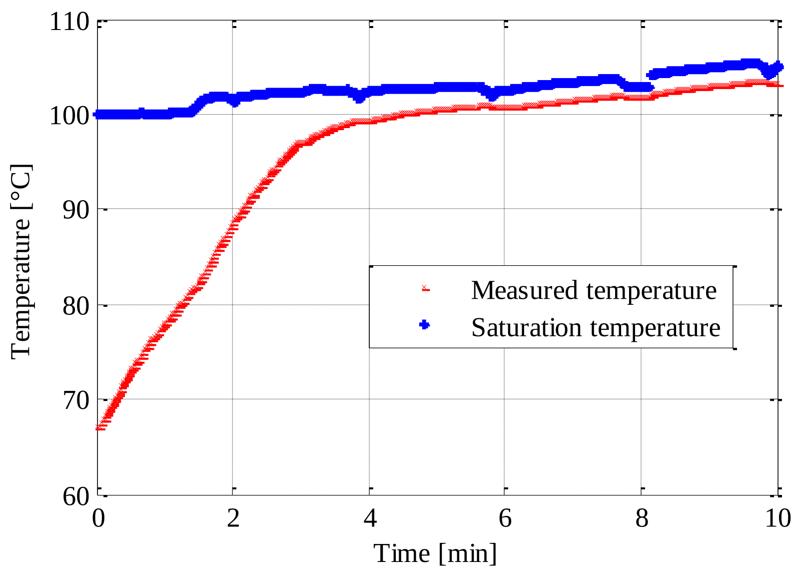

Figure 3. Temperature evolution in high pressure branch of the heat pump

In the first part of this start-up phase, the process simulation circuit adds heat to heat pump system via the condenser and absorbs heat in the second part where heat pumping function of the heat pump begins. Figure 4 presents evaporation temperature which stills constant in the flash vessel while the discharge temperature of the compressor increases to attain a constant temperature of 130°C in 35 min. The increase of this temperature is due to the pressure evolution while a constant injected mass flow rate is imposed at the suction of the compressor.

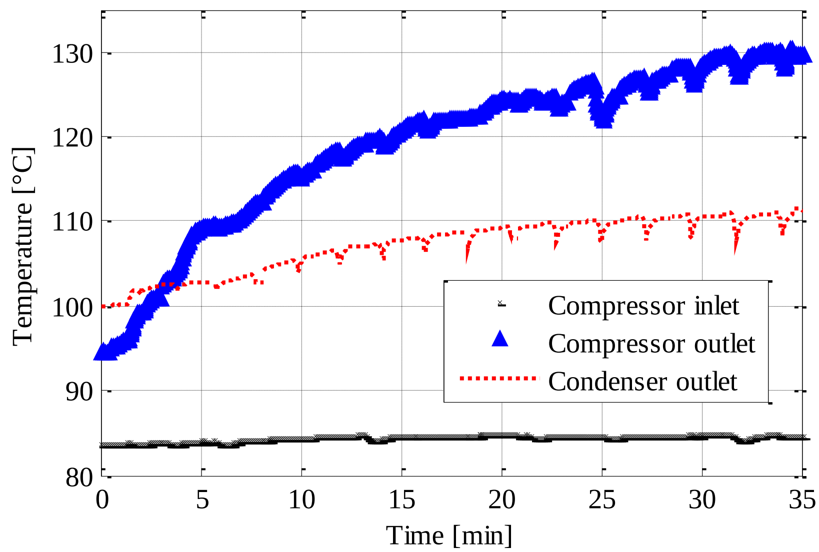

Figure 4. Inlet and outlet temperature of the compressor and the condenser at start-up

At constant evaporation pressure 0.6 bars, figure 5 presents the condensing pressure evolution in function of time to attain steady state conditions at 1.5 bars which corresponds to an superheating of the compressor outlet of 20K in 35 minutes. In this figure, we observe a lot of sudden pressure drops due to the opening of the ball float steam trap which control the water level in the accumulator. The amplitude of these pressure drops are function of the difference between mass flow rates imposed by the compressor and the expansion valve. These drops disappear at higher rotational speed of the compressor.

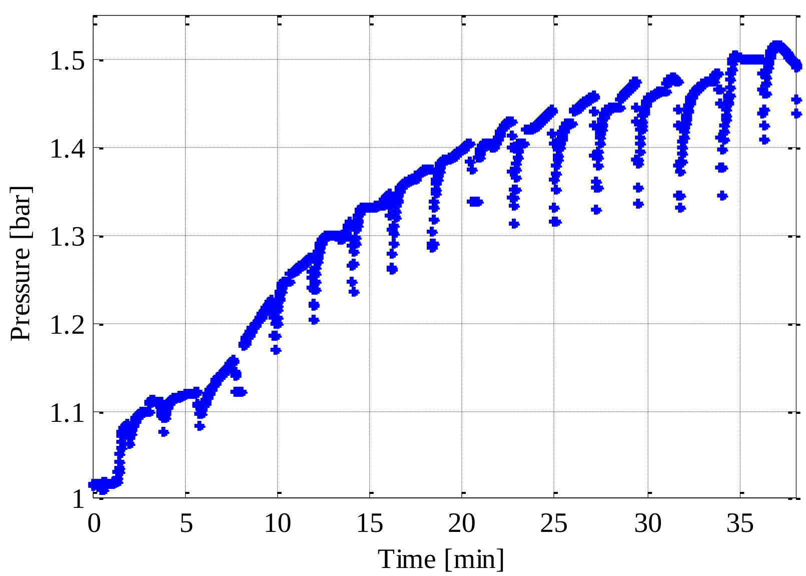

Figure 5. High pressure evolution of the heat pump at start-up phase

Furthermore, performances of the present heat pump are presented using the numerical model established by Chamoun *et al*., (2012). Different high temperature processes were simulated for different waste heat scenarios.

At full load operating conditions of the compressor (motor frequency of 4700 RPM), process volume flow rate was set to 30 $m^3.h^{-1}$ at 110°C at the inlet of the condenser. Figure 6 shows the evolution of process temperature and heating power supplied by the heat pump in several process scenarios for waste heat temperature levels varying between 85°C and 95°C, with different waste volume flow rates (VFR). The increase of waste heat temperature or waste volume flow rate increases heating power. This figure shows that increasing waste temperature from 88°C to 92°C at constant waste VFR of 36$m^3.h^{-1}$ increases heating power from 300 to 335 kW and from 328 to 368 kW while the VFR is doubled.

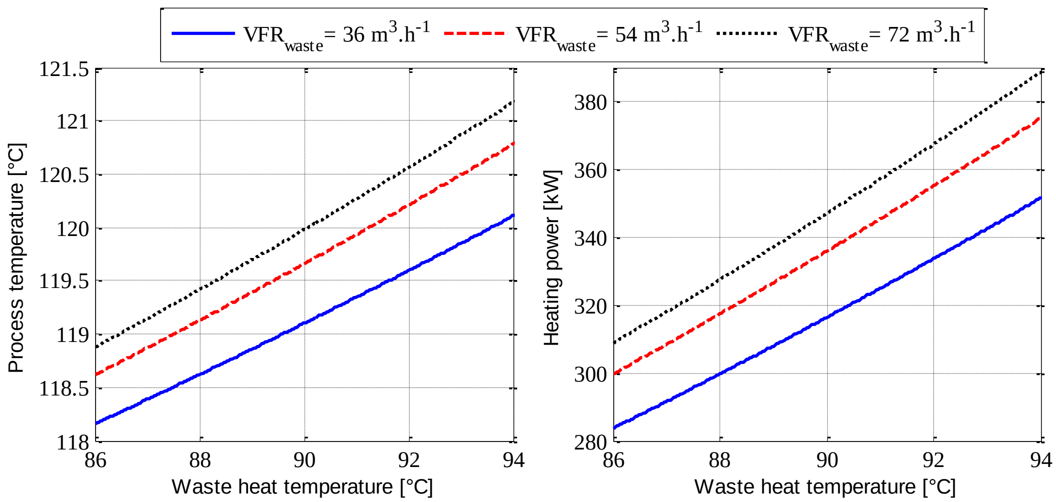

Figure 6. Condensing temperature and heating power versus VFR and temperature of waste heat

Figure 7 shows the heat pump performance in the process scenarios mentioned above. This figure illustrates the sensitivity of the global COP to the waste heat VFR and temperature. We can conclude that the heat pump COP is not heavily affected by the VFR variation (maximum observed increase about 5 %) but an increase of 11% is observed with the rising of the waste heat temperature.

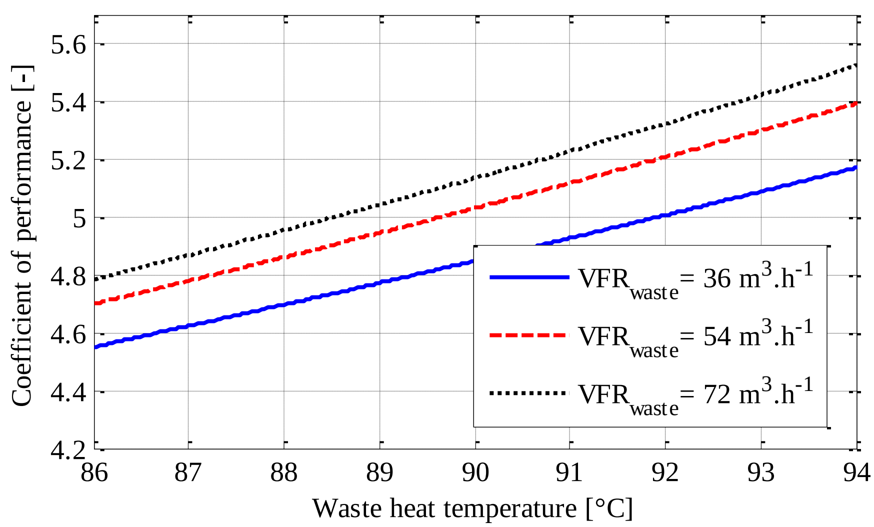

Figure 7. COP of the heat pump versus VFR and temperature of waste heat

The present heat pump for industrial heat recovery presents high performance highly depending on waste heat temperature and process temperature to satisfy. The presented heat pump can recover waste heat from 85°C to 95°C to satisfy heat demand at higher temperature level attaining a condensing temperature of 145°C.

5. CONCLUSION

High temperature heat pumps offer a great potential for recovering waste heat generated by several industrial applications. A high temperature heat pump using water as refrigerant is designed and built to be tested on a laboratory test bench that reproduce the operating conditions of real case industrial applications. This heat pump is an electrically-driven vapor compression closed-cycle type, using a newly developed twin screw compressor. This new heat pump generates vapor by flash evaporation. A purging valve is implemented at the top of the accumulator in order to eliminate non-condensable gases from the installation. Since a substantial amount of air will have to be removed upon start-up phase of the heat pump, preliminary experimental results has been presented showing the uniqueness of using this design of heat pumping specific for water vapor. COP is calculated showing generally good energy performance and high environmental benefits.

NOMENCLATURE

COP	Coefficient of performance	(-)
T	Temperature	(°C)
ΔT	Temperature difference	(K)
VFR	Volume flow rate	($m^3.h^{-1}$)

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