

PERFORMANCE CHARACTERISATION OF A REVERSIBLE WATER TO WATER HEAT PUMP

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Abstract: This paper presents a detailed performance characterisation of a reversible water to water heat pump operating in cooling mode (SHERHPA 2007). This was designed to be integrated in a building air conditioning system in order to produce cold or hot water as required (Urchueguia et. al. 2006, GEOCOOL 2005). The characterisation of the heat pump behaviour with regard to both cooling capacity and the coefficient of performance (COP) under variable load conditions on secondary fluid external loops is obtained through a laboratory experimental test campaign. It was possible to simulate the conditions of the real application based on: i) variations of water inlet temperatures on both evaporator and condenser sides and ii) variations of flow rates on the water loops. This experimental performance study is obtained for water flow rates between 2000 and 4200 kg/h at 10.8°C and 13°C water temperatures at the inlet of evaporator. For further analysis IMST - ART simulation software is used to extend the range of input parameters up to a minimum water mass flow rate of 1000 kg/h and water inlet temperatures on evaporator side between 8°C up to 18°C as opposed with experimental tests that have restricted possibilities. The consistent understanding of the system response under different load conditions has as a direct result an improved system efficiency obtained based on the optimum control of the water mass flow rates on both evaporator and condenser side.

Key Words: *heat pump, water mass flow rate, optimum, cooling capacity, COP*

1 INTRODUCTION

Sustainable energy systems such as heat pump technologies provide an efficient use of renewable energy from the ambient. Typical applications of a heat pump are: space heating, domestic hot water and processes with combined heating and cooling. In the last decade a significant research effort for optimised system design with direct application in industry was undertaken (Jacobi et. al. 2005, Corberan et. al. 2006, Primal et. al. 2006). It is aimed to develop heat pumps that are cost – energy efficient and in compliance with the environmental regulations that lead to the phase out of conventional refrigerants due to the protection of the ozone layer. As a result of Kyoto agreement a series of refrigerants like R22 are scheduled to be replaced until 2010 (Kyoto protocol 1997). Therefore “natural refrigerants” like CO₂, ammonia and hydrocarbons have to be adopted even if this implies a change in the components technology and control [Pearson 2006, Jakobsen et. al. 2006, Huff et. al. 2006].

The 6th European Framework Programme research project entitled: “Sustainable Heat and Energy Research for Heat Pump Applications – SHERHPA” aimed to address these challenges and solutions with immediate application in industry. The work presented in this paper is part of the mentioned project and the main objective is to offer a complete experimental characterisation of a prototype heat pump (Figure 1) working with propane

(R290). It is wished to determine the optimum operating parameters for increased efficiency and to prepare the first steps in the control strategy. The main components of the prototype are as follows: i) semi hermetical compressor Bitzer 4EC–6 type (1), ii) a four-way valve to switch between heating and cooling modes (2), iii) electronic expansion valve - Carel EV 24 (3), and iv) SWEV V80, B80 evaporator and condenser (4).

The heat pump is designed to provide a nominal cooling capacity of 15 kW and heating capacity of 17 kW. The final purpose of the project is to commission and install the prototype as part of a geothermal building air conditioning system (Figure 1) used to provide the desired temperature control based on both cooling and heating as necessary.

The building air conditioning system (GEOCOOL 2005) has the following components: i) an external loop that consists in a ground heat exchanger (GSHX), a circulating pump, a storage expansion tank and the corresponding piping; this loop takes the heat exchanged at the condenser side of the heat pump (while working in cooling mode) and dissipates it to the ground, ii) an internal loop that distributes the chilled water (while working in cool mode) produced at the evaporator side of the heat pump to a series of 12 fan coils installed in the building and iii) the “engine” of the whole system that is the water to water reversible heat pump working with propane (R290).

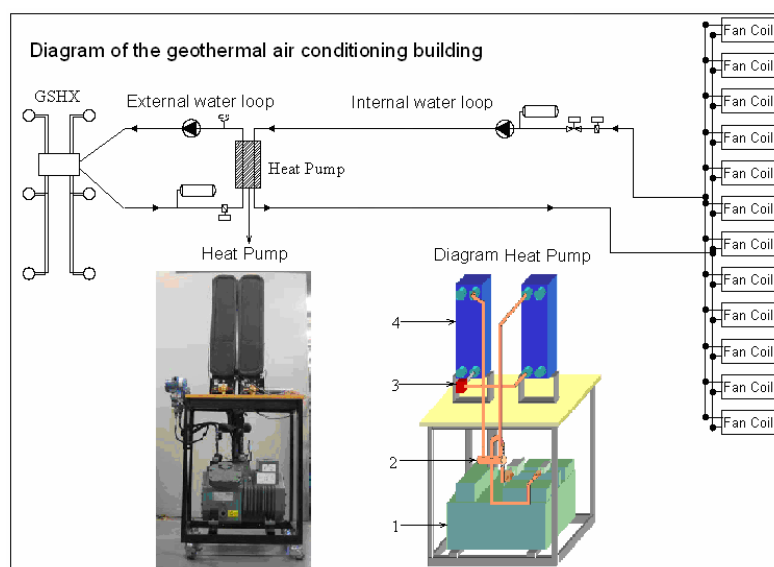


Figure 1: Building air conditioning system

The preliminary stage of the project consisted in an extended experimental test plan of the prototype in a dedicated test facility (Figure 2) capable to reproduce in the laboratory conditions the demands of the real installation. Considering that the development of highly efficient units is one of the crucial challenges for the future success of refrigeration and air conditioning equipment, the conclusions of the experimental results wish to provide a clear understanding for the optimum response of the unit for specific operating conditions and to present an empirical correlation that could be used for future analysis in the development of improved control algorithms.

2 EXPERIMENTAL SET - UP

Considering that the main demand for the heat pump is to work as a chiller, the laboratory experimental stage is focused on efficiency maximisation during the cool mode.

The dedicated test rig was designed to be used for both cooling and heating performance characterisation of reversible water to water heat pumps in conformity with the existing standards EN-255 (1998) and EN-14511 (2004). Figure 2 shows the schematic design of the

test rig that consists of two water hydraulic groups (loops) capable to maintain the desired set point temperatures at both evaporator and condenser inlet based on mass flow rate control. The test rig was designed in order to simulate on both sides of the unit the load conditions encountered during the typical field operation. Each water loop was equipped with the corresponding secondary plate heat exchanger (PHE) (1), pump (2), by-pass valve (3), inertia tank (4), expansion chamber (5), and pressure valves and gauges.

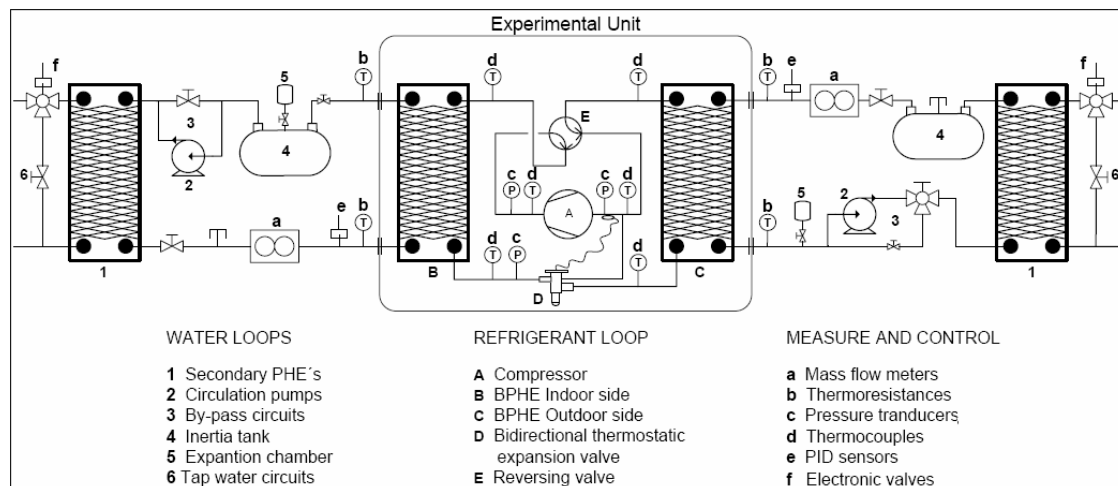


Figure 2: Heat Pump laboratory experimental test rig – Technical University Valencia (UPV)

The heat pump operation conditions are obtained based on control of the outlet or inlet water temperatures in the inner and outer heat exchangers (evaporator and condenser respectively). This control strategy is based on variations of the flow rate of the city water (6) through the secondary heat exchangers (1). This is accomplished with a three-way regulating valve (f) controlled by a PID that receives the signal of the water inlet temperatures (e). During the experimental tests the inlet temperatures of the secondary fluid were controlled within ± 0.2 K from the set point. The temperature difference of the water between the inlet and outlet of the primary heat exchangers was maintained at 5K, controlling the water flow rates by means of a by-pass circuit (3).

The behaviour of the heat pump under test (experimental unit) is monitored using dedicated sensors placed on the refrigerant and water circuits (Table 1). Two Coriolis mass flow meters were used to measure the water flow (a). The temperature difference through the brazed plate heat exchangers (BPHEs) was monitored with calibrated PT-100 RTDs (b). Inside the refrigerant loop, pressure transducers (c), located on the suction and discharge line of the compressor, measured the evaporation and condensation pressures. The temperatures along the refrigerant circuit for the evaluation of the superheat and subcooling were obtained with T-type calibrated thermocouples (d). An energy counter was used to measure the electrical consumption of the compressor in order to evaluate the COP.

Table 1; Specifications of the measuring sensors

Instrument	Type	Range	Precision
Thermocouple	T	-270 – 400C	± 0.5 K
Thermoresistance	PT – 100	-220 – 850C	± 0.1 K
Masss Flow Meter (Coriolis)	Danfoss	0 – 5500 kg/h	$\pm 0.05\%$
	Danfoss	0 – 5500 kg/h	$\pm 0.05\%$
Pressure Transducer	Fisher Rosemount	1000 kPa	$\pm 0.2\%$
	Fisher Rosemount	1000 – 2500 kPa	$\pm 0.2\%$
	Fisher Rosemount	0 – 5500 kPa	$\pm 0.2\%$
Power Meter	Quantum D - 200	30 kW	$\pm 0.1\%$

Data acquisition is performed using a Hewlett Packard datalogger and the measurements are registered in a continuous, cyclic way, acquiring a complete set of all parameters every 10 seconds, once that steady state conditions are reached. It is considered that this condition is obtained when all the measured magnitudes stay constant, without set points modifications. The periodic fluctuations of these magnitudes due to the performance of the control system are allowed under the condition that the average values of these fluctuations do not exceed the permissible deviations between the set-points and the arithmetic average of the real values as indicated: i) water inlet-outlet temperatures between $\pm 0.3\text{K}$, and ii) volumetric flow $\pm 2\%$. As far as the capacities are concerned, the EN-14511 (2004) standard establishes that these must be determined with a maximum uncertainty of 5% independently of the uncertainties of measurement instruments used, including the uncertainties due to the properties of the fluids. The calculation procedure for both Cooling Capacity ($Q_{cooling}$) and COP ($COP_{cooling}$) is based on the heat balance applied to the heat exchangers, throughout the measurement of the mass flow rate (m_{evap}), and the inlet and outlet temperatures (ΔT_{water}), as shown in Eq. 1. The coefficient of performance (COP) was obtained as a calculation (Eq. 2) from the cooling capacity and compressor power consumption - P (Incropera and deWitt 2000) that was measured directly by the data acquisition whilst the test duration estimated at 30 minutes for each test.

$$Q_{cooling} = m_{evap} c_p \Delta T_{water} \quad Eq.1$$

$$COP_{cooling} = \frac{Q_{cooling}}{P} \quad Eq. 2$$

3 VALIDATION IMST – ART SOFTWARE

IMST-ART[®] is an advanced computer-aided engineering design system that combines accurate and fast algorithms, easy-to-use graphical interface and powerful analysis capabilities into a software package suitable for modelling any vapour-compression system operating with several types of refrigerants and secondary fluids. The software is based on a global numerical model that consists of individual mathematical equations for each component of the heat pump. The solution is given in a steady state format through the Newton Raphson algorithm (Corberan and Gonzalvez 2002). The technical characteristics of each component together with information regarding the operation parameters for the desired regime are introduced as an input and the software is capable to provide information regarding the detailed operating parameters for each component including the pipes and fittings together with information about capacity, COP, compressor efficiency, superheat, subcooling and heat transfer data for the heat exchangers. The comparison between the experimental results and the predictions of IMST – ART obtained for the test conditions listed in Table 2 are presented in Figure 3.

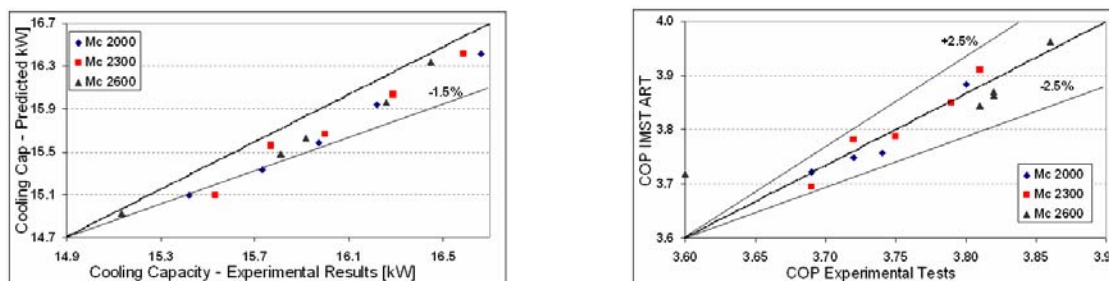


Figure 3: Validation of the IMST – ART predictions versus experimental results

It is noticed that a very good agreement of $\pm 2.5\%$ is obtained for COP of the heat pump when compared with the experimental results. Even a better prediction of only -1.5% is obtained for the cooling capacity of the heat pump under consideration. The validation of the IMST – ART predictions was detailed investigated (Gonzalvez et. al. 2007) and a maximum average difference of only 4% is obtained for all operating parameters.

4 RESULTS AND DISSCUSION

The main objectives of the experimental work was to determine the optimum response of the system based on variations of the water flow rates as follows: i) test condition I: it was achieved based on control of evaporator water flow rates $M_e = 2000, 2300, 2600, 3100$ and 4200 kg/h for constant condenser water flow rates at three levels $M_c = 2000, 2300$ and 2600 kg/h and ii) test condition II: the water flow rates on evaporator side were maintained constant at $2000, 2300$ and 2600 kg/h while variations were imposed on the condenser side (Table 2). The experimental tests were performed based on the control at constant set-point of the water temperatures at the evaporator and condenser inlet as required during field operating conditions. Considering that the heat pump was primarily designed to operate in cooling as part of the building air conditioning system, the aim of the test matrices was to determine the conditions for the optimum cooling capacity and coefficient of the performance (COP).

Table 2; Test conditions – variations of water flow rates.

Evaporator water inlet temperature	Condenser water inlet temperature	Condenser water flow rate Condition I	Evaporator water flow rate Condition I	Evaporator flow rate Condition II	Condenser flow rate Condition II
°C	°C	kg/h	kg/h	kg/h	kg/h
10.8±0.7	22±0.7	2000, 2300, and 2600	2000	2000, 2300, and 2600	2000
			2300		2300
			2600		2600
			3100		3100
			4200		4200

For each experimental condition the cooling capacity is calculated based on the heat balance applied for the heat pump evaporator (*Eq. 1*). The coefficient of performance (COP) is obtained as the ratio between the cooling capacity and the compressor electric consumption (*Eq. 2*).

Experimental Results

This section presents the most significant results obtained during the experimental campaign together with an extended analysis of the system behaviour based on IMST – ART predictions.

4.1.1 Test condition I

Figure 4 shows the results obtained for Test Condition I. The condenser water flow rate is maintained constant at $2000, 2300$ and 2600 kg/h while a controlled variation of the evaporator water flow rate is imposed at $2000, 2300, 2600, 3100$ and 4200 kg/h.

It is shown that higher evaporator water flow rate has as a result a similar trend in the cooling capacity variation. An increase of 100kg/h in the secondary fluid mass flow rate provides an average 2.5% higher cooling capacity. It is also noticed that the magnitude of the condenser water flow rate has a smaller impact on the cooling capacity especially when on evaporator side the water flow rate is higher than an approximate value of $M_e=3100$ kg/h.

The combined effect of the evaporator inlet water temperature and the condenser flow rate is also presented in Figure 4. While the set - point temperature was 10.8°C for both $M_c = 2000$ and 2300 kg/h the increase by 1% in the cooling capacity is the result of 300 kg/h higher water flow rate on condenser side. For the case $M_c = 2600$ kg/h the evaporator water inlet temperature was maintained at 0.4°C lower value. For this condition it is concluded that the effect of increased water flow rate with 300 kg/h is overcome by the lower water temperature as the cooling capacity for this condition is with an average 1% lower than at 2300kg/h mass flow rate. Further analysis of the combined influence of both temperature set point and flow rate for COP is presented in Figure 6.

It is concluded that the minimum and maximum cooling performance of the heat pump is between 14.9 kW and 16.4 kW respectively, depending strongly on the variations of water mass flow rates on evaporator side.

Figure 4 (b) shows that the coefficient of performance (COP) of the heat pump also experiences a uniform increase with both evaporator and condenser water flow rates. The maximum COP of the unit is obtained for high evaporator flow rates ($M_e > 3100 \text{ kg/h}$). However, it is difficult to take into consideration this region as during the heat pump operation in the dedicated building air conditioning system the consumption of the recirculation pumps is included in the COP of the entire system (Figure 1). This overall COP is decreasing at high evaporator and condenser flow rates ($> 3100 \text{ kg/h}$) as this regime also implies higher power consumption of the recirculation pumps. As a result, for further detailed analysis of the optimum operating parameters, it is taken into consideration the region with lower water flow rates on evaporator side, between 2000 and 3100 kg/h . A more detailed review of the test results for this region (Figure 4 b) shows that it is possible to obtain an optimum for COP when both M_c and M_e are controlled between 2300 and 2600 kg/h .

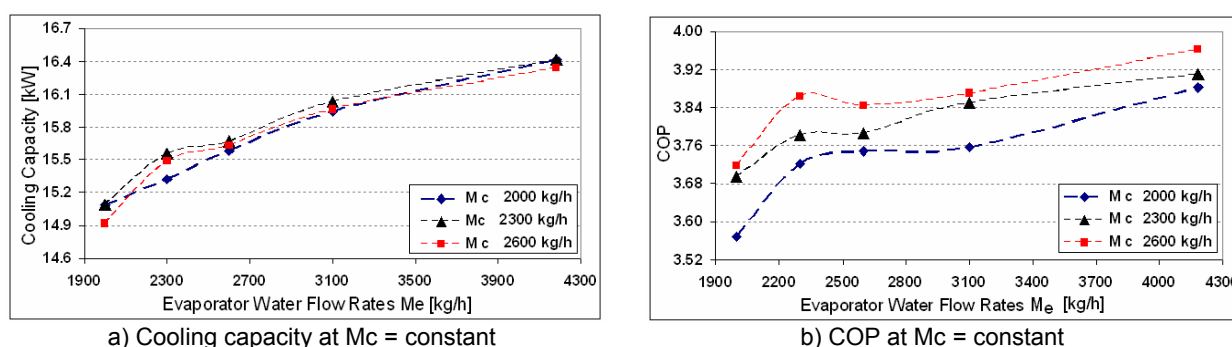


Figure 4: Experimental results obtained for Test Condition I

Regarding the optimized control of the unit it is wished an algorithm capable to decide between periods of time when the heat load is very high and it is necessary to operate at maximum cooling capacity (maintain water flow rates $> 3100 \text{ kg/h}$) even if it means less efficiency of the system and periods of time when it is wished to operate efficient and the maximized cooling capacity is no longer a priority (maintain water flow rates between 2300 and 2600 kg/h).

4.1.2 Test condition II

Figure 5 shows the experimental results obtained for both cooling capacity and COP of the system during Test Condition II when the evaporator water flow rates are maintained constant at 2000, 2300 and 2600 kg/h while the condenser flow rate was controlled at 2000, 2300, 2600, 3100 and 4200 kg/h .

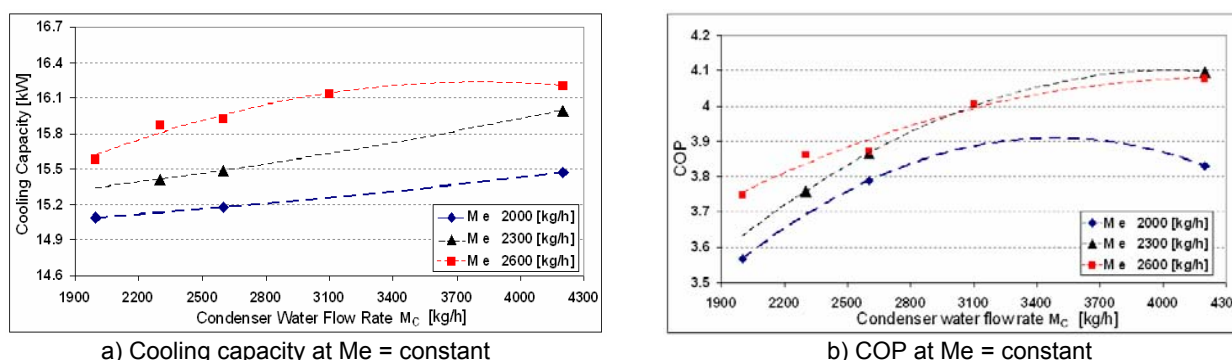


Figure 5: Experimental results obtained for Test Condition II

The results presented in Figure 5 confirm the conclusions stated in the previous section. The general trend of both cooling capacity and COP reflects a uniform increase once higher water flow rates are imposed on both evaporator and condenser sides. At variation of M_c between 2000 ÷ 4200 kg/h and M_e between 2000 ÷ 4200 kg/h the maximum and minimum limits of the

heat pump coefficient of performance are 4.1 and 3.58 respectively. An average increase by 300kg/h in M_c has as a result only an average 1.2% higher cooling capacity. It is concluded that the variations of the secondary fluid mass flow rate on condenser side has a significant effect only on the COP while the cooling capacity is influenced significantly by the evaporator side water flow rates.

From Figure 5 (b) it is concluded that for controlled condenser water flow rates in the range $2000 \leq M_c \leq 4200$ kg/h the optimum system COP is obtained while maintaining a constant evaporator mass flow rate at $M_e = 2300$ kg/h. This effect is more pronounced at condenser water flow rates higher than 3100 kg/h. However, taking into account the overall air conditioning system COP it is concluded that it is indicated to maintain the condenser flow rate between 3100 and 3300 kg/h to overcome the negative impact resulted from the high power consumption of the water recirculation pump installed on the external ground loop.

Further analysis of the COP is shown in Figure 6 (a, b) at both test conditions I and II. Several tests were performed in order to quantify the combined influence of water temperature at the inlet of evaporator and condenser with the variations of the mass flow rate. It is noted that during experiments it is difficult to quantify separately the influence on the cooling capacity and COP of these two parameters due to the feedback between them. For the evaporator operating under cooling mode the increase in the mass flow rate results in higher heat transfer coefficient and therefore an increase for both capacity and COP as the temperature difference between the refrigerant and the secondary fluid approach each other (countercurrent configuration). However if we keep constant the evaporator water inlet temperature, as an increase in the flow rate also leads to a decrease in temperature drop across the heat exchanger, then the mean water temperature increases and therefore the refrigerant senses a higher secondary temperature. This effect is beneficial for the COP. Inversely if the outlet temperature is maintained constant then the mean water temperature will decrease and therefore the refrigerant senses a lower secondary temperature which will be transformed in a loss of COP.

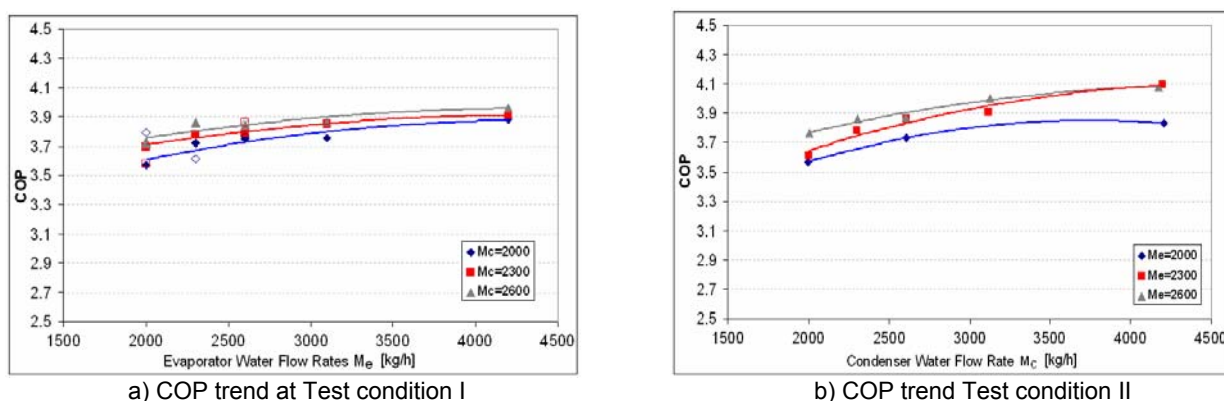


Figure 6: General trend of COP for test conditions I and II

The combined water temperature and flow rate interaction is highlighted during the test campaign for $M_e = 2600$ kg/h and $M_c = 4200$ kg/h. While maintaining these parameters constant two tests were performed at different condenser water inlet temperature that was initially maintained at 22°C and for the second test this temperature was 5% higher. The increase in this parameter had a significant effect on COP as a loss of 9% was experienced. It is concluded that the heat pump COP appears to be very sensitive at water temperature variations on condenser side while operating in cooling mode. The cooling capacity is less influenced by the condenser inlet temperature as it experiences only a 3% decrease for the same test conditions. Figure 6 shows a similar test comparison for $M_{\text{evaporator}} = 2000$ kg/h and $M_{\text{condenser}} = 2000$ kg/h. It is noted that at higher condenser inlet temperature with the same ratio of 5.4% the COP experiences a loss by 3.8%.

In the real application of the heat pump (Figure 1), the water temperature is highly dependent on the ground loop. The closer the ground and condensation temperatures are the higher is the increase obtained for COP.

IMST – ART Results

In this section an extended analysis of the heat pump behaviour is presented based on supplementary information offered by the IMST – ART software predictions. The test range was extended starting with a minimum of 1000 kg/h water flow rates with plotted data at each 500 kg/h increase. As a novelty the influence of evaporator water inlet temperature is quantified for a variation range between 8.1°C and 18.9°C.

Figure 7 shows the evolution for the cooling capacity and COP at Test condition I while at constant condenser water flow rates the variations are imposed on the evaporator side. As a supplementary information opposed with the experimental results it is noted (Figure 7b) that a slightly higher increase in COP takes place at the lower range of the condenser water flow rates (<2500 kg/h) while the influence on the cooling capacity is less significant at the same rate.

With regard to evaporator mass flow rates Figure 7 shows that the control of this parameter in the low variation range $M_e < 2000$ kg/h has greater impact on the increase of both COP and cooling capacity by 4.1% and 7.1% respectively at a 300kg/h evaporator flow rate variation.

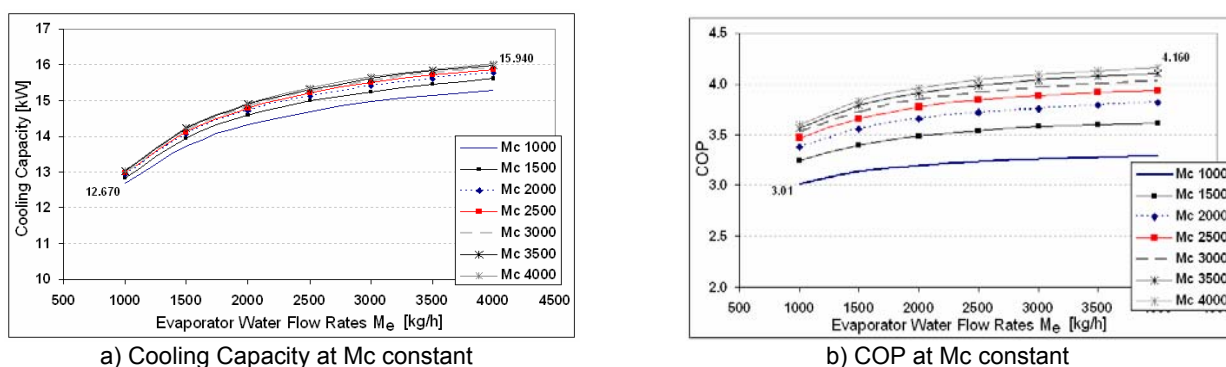


Figure 7: IMST – ART predictions for Test Condition I

Figure 8 shows the evolution for both cooling capacity and COP for Test condition II. The evaporator mass flow rate is maintained constant at 6 levels between 1000 and 4000 kg/h while variations of 500 kg/h for each data plotted are imposed for the condenser water flow rate. It can be concluded that for both cooling capacity and COP the significant increase takes place at high water mass flow rates on evaporator side (>2500kg/h).

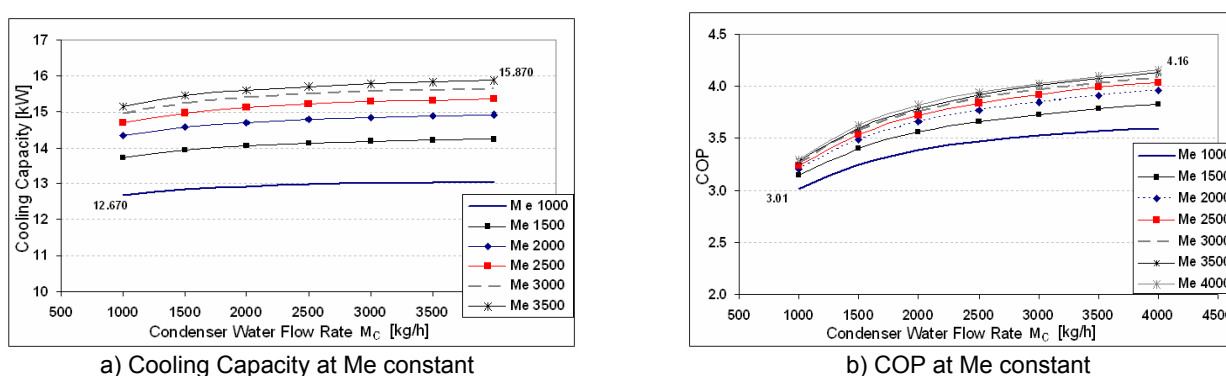


Figure 8: IMST – ART predictions for Test Condition II

A comparison between the results presented in Figures 7 and 8 confirms that the variations of the condenser mass flow rates have stronger impact on the COP while the effect on the cooling capacity is diminished as it is more strongly influenced by changes in the evaporator flow rates. The results are consistent with the experimental data presented in the previous section. It has to be mentioned that in the real application the increased flow rate on condenser side has as a result a higher heat transfer coefficient on the water side of the ground heat exchanger.

However, the heat transfer to the ground is dominated by the heat transfer through the soil so water side variations will have negligible influence on the global heat transfer to the soil.

As a novelty, Figure 9 shows the results for both COP and cooling capacity when it takes place at several set points for evaporator inlet water temperatures: 8.1°C, 10.8°C, 13.5°C, 16°C and 18.9°C. It is noted that while the variation of evaporator water flow rate is plotted for the whole range examined, the condenser flow rate is maintained constant at 2600 kg/h.

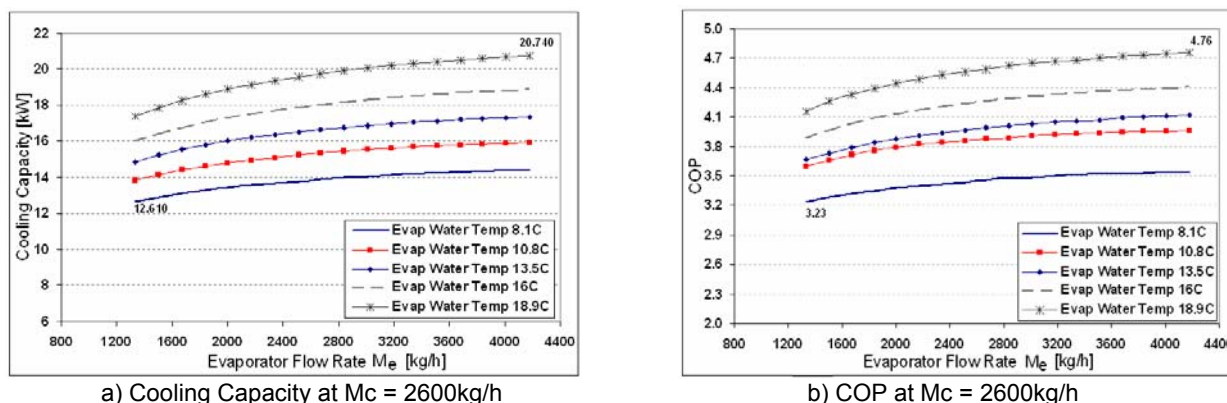


Figure 9: Experimental and IMST – ART results for Evaporator water Inlet Temperatures

A uniform increase in the cooling capacity by an average 8% and 6.8% for COP is obtained for 2.7°C temperature variation. The optimum evaporator inlet temperature obtained mainly based on the response of COP is considered to be at 10.8°C.

The cooling performance and COP operating limits are quantified. The cooling capacity has a minimum of 12.6kW at 1000kg/h on both condenser and evaporator side while the maximum cooling capacity is 20.74kW at 4200 kg/h and 18.9°C water temperature.

5 CONCLUSIONS

This work presents a detailed experimental performance characterisation of a reversible water to water heat pump operating in cooling mode. This was designed to be integrated in a building air conditioning system mainly in order to produce cold water (Geocool 2005, Urchueguia et. al. 2006). The characterisation of the heat pump behaviour with respect to the cooling capacity and COP is obtained under controlled variations of water flow rates on the secondary external loops. It is aimed to conclude the optimum water flow ranges in order to obtain increased efficiency of the system. The experimental test range is extended and the influence of water temperature at the evaporator and condenser inlet is quantified through the predictions of the IMST – ART simulation software. The general evolution obtained for both cooling capacity and COP reflect a uniform increase once higher water flow rates are imposed. The evaporator flow rates have a strong impact on the cooling capacity increase while the condenser flow rates have a more pronounced influence on the system COP. It is quantified that an increase by 100 kg/h in the evaporator mass flow rate generates an average 2.5% higher cooling capacity while the same increase in the condenser flow rate has a stronger impact on the COP that experiences an average 5% higher value. The optimum COP of the heat pump is established to be obtained when both condenser and evaporator mass flow rates are between 2300 and 2600 kg/h. This range involves not only the heat pump response but also the requirements and conditions of the real air conditioning system application. The potential for improved control is based on the increase in secondary fluid mass flow rates that lead to higher COP while taking into account the power consumption of the water recirculation pumps from the external loops.

The preliminary experimental test campaign presented in this paper is important as it provides the results for the optimum operating conditions. At present the heat pump is installed in the building air conditioning system in order to assess the real advantage of controlling the water flow rates on external loops that leads at minimum consumption of the entire system.

6 ACKNOWLEDGEMENTS

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