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Parametric analysis of air–water heat recovery concept applied to HVAC systems: Effect of mass flow rates

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ABSTRACT

In the last three decades, the world has experienced enormous increases in energy and fuel consumption as a consequence of the economic and population growth. This causes renewable energy and energy recovery to become a requirement in building designs rather than option. The present work concerns a coupling between energy recovery and Heating, Ventilating and Air Conditioning HVAC domains and aims to apply heat recovery concepts to HVAC applications working on refrigeration cycles. It particularly uses the waste energy of the condenser hot air to heat/preheat domestic water. The heat exchanger considered in the recovery system is concentric tube heat exchanger. A thermal modeling of the complete system as well as a corresponding iterative code are developed and presented. Calculations with the code are performed and give pertinent magnitude orders of energy saving and management in HVAC applications. A parametric analysis based on several water and air flow rates is carried out. It was shown that water can be heated from 25 to 70 °C depending on the mass flow rates and cooling loads of the HVAC system. The most efficient configurations are obtained by lowering the air flow rate of the condenser fan.

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1. Introduction

The necessity to reduce fuel consumption while producing energy has escalated the efforts toward developing renewable energy and energy management concepts [1–3]. Heat recovery is a particular route in energy management where one tends to mainly capture heat generated from various applications, contrary to solar and wind energies where natural energies are invested [3]. Consequently, heat recovery or capturing heat has been receiving a special attention, namely, when it comes to heat water in without suffering from high costs of using electricity, fuels, and solar energy [4–5].

Heat recovery is defined as the process of recovering energy (or heat) from a stream at a high temperature to a low temperature stream that is effective and economical to run [6]. Depending on the system in which hot fluids or gases are contained, heat recovery/capturing might take place from applications; such as, internal combustion engines, heat pumps, chillers, chimneys, shower water and power generators [7–11].

On the other hand, Heating, Ventilating and Air Conditioning HVAC systems has changed from being luxury to essential need for people and thus involve many energy components which need to be more and more managed [12–14]. Passing

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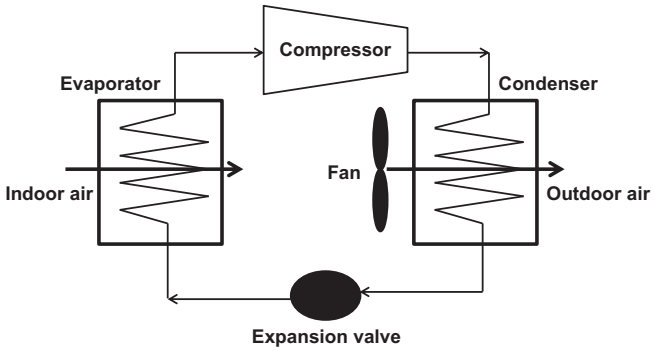

Fig. 1. Schematic of the operational mode of refrigeration cycle.

through the existent art in the literature, it is found that large amounts of energy (heat) are lost in Heating, Ventilating, and Air Conditioning systems [15–16]. Moreover, heat recovery technology will offer an optimal solution in terms of fresh air, better climate control and energy efficiency [17]. In this context, the present work suggests a new concept that permits to couple between the two energy axes described above: energy recovery and HVAC. It particularly use the heat wasted in the condenser air to heat and/or preheat residential water. For this purpose, a complete thermal modeling of the system as well as a corresponding iterative code are developed. A parametric analysis taking into account the water flow rate, air flow rate and the capacity of the refrigeration cycle of the HVAC system is performed.

This paper is organized such that Section 2 presents the thermal modeling as well as the iterative procedure of performance calculations followed by results and analysis Section 3. Finally, the main conclusions are drawn in Section 4.

2. Materials and methods

The refrigeration cycle is significantly used in HVAC systems (Fig. 1).

During this cycle, the cold low pressure refrigerant vapor enters the compressor and is pumped into the high pressure side of the system. The hot high pressure gas passes through the condenser which is a coil placed outside the cooled space. A fan is used to cool the refrigerant and thus the refrigerant undergoes a change of state from gas to liquid. The refrigerant passes then through an expansion valve, where it undergoes a pressure decrease necessary to the evaporation process to occur in the evaporator. The heat required to the evaporation is removed from the indoor air resulting in decreasing its temperature.

The principle of the recovery system suggested (Fig. 2) is to capture the heat rejected from the condenser of the HVAC system and use it to heat water.

The air flow produced by the condenser fan and heated by the heat released from the condenser is oriented to a concentric tube heat exchanger. When flowing in the heat exchanger, the hot air will heat the cold water supply.

The heat exchanger considered in the present study is counter flow concentric tube heat exchanger with inner diameter D_i, outer diameter D_o and length L. In the thermal modeling presented below and for presentation clarity, water is considered flowing in the inner tube of the exchanger and condenser hot air flowing in the annulus of the exchanger. Later in the calculations, the configurations can be inversed and the governing equations inversed accordingly. In this type of exchanger and neglecting the heat transfer between air in the annulus and the ambient air, the energy balance can be written

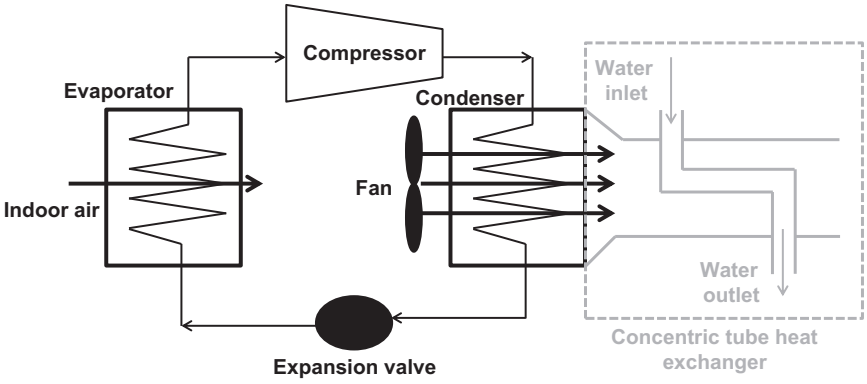

Fig. 2. Schematic of the heat recovery system proposed.

as [18]:

$$\dot{m}_w C_{p,w}\left(T_{w,out} - T_{w,in}\right) = \dot{m}_a C_{p,a}\left(T_{a,in} - T_{a,out}\right) = UA\Delta T_{\ln} \tag{1}$$

Where $\dot{m}_w$ and $\dot{m}_a$ are respectively the water and air mass flow rates, $C_{p,w}$ and $C_{p,a}$ are respectively the water and air specific heats, $T_{w,in}$ and $T_{a,in}$ are respectively the water and air inlet temperatures, $T_{w,out}$ and $T_{a,out}$ are respectively the water and air outlet temperatures, U the overall heat transfer coefficient between the two fluid streams, A the area of heat transfer between the two fluids, and $\Delta T_{\ln}$ the logarithmic mean temperature difference between the two fluids calculated from:

$$\Delta T_{\ln} = \frac{\left(T_{a,out} - T_{w,in}\right) - \left(T_{a,in} - T_{w,out}\right)}{\ln\left[\frac{\left(T_{a,out} - T_{w,in}\right)}{\left(T_{a,in} - T_{w,out}\right)}\right]} \tag{2}$$

In most heat exchanger applications, a thin wall of large conductivity is generally used between the two fluid streams and then the overall heat transfer coefficient and the heat transfer area can be calculated from:

$$U = \frac{1}{\frac{1}{h_w} + \frac{1}{h_a}} \tag{3}$$

$$A = \pi D_i L \tag{4}$$

Where h_w and h_a are the convective heat transfer coefficients of respectively the water and air flows calculated from heat transfer correlations found in [18]. Below these calculations are detailed.

Two types of flows are considered the laminar and turbulent flows. The physical parameters of laminar and turbulent inner and outer flows are detailed below:

1. Inner flow:
When the inner flow is laminar, the convective transfer coefficient can be calculated from the following correlation [18]:

$$Nu_i = \frac{h_i D_i}{k_i} = 3.66 + \frac{0.0668(D_i/L)\mathrm{Re}_i\mathrm{Pr}_i}{1 + 0.04\left[(D_i/L)\mathrm{Re}_i\mathrm{Pr}_i\right]^{2/3}} \tag{5}$$

Where Nu_i is the Nusselt number, k_i the thermal conductivity, Pr_i the Prandtl number and Re_i the Reynolds number of the inner flow. The thermo-physical properties of the fluid are determined at the average temperature between the inlet and outlet sections.
The Reynolds number can be calculated from:

$$Re_i = \frac{4\dot{m}_i}{\pi D_i \mu_i} \tag{6}$$

Where μ_i is the dynamic viscosity of the inner fluid.
Furthermore, if the inner flow is turbulent [18]:

$$Nu_i = \frac{\left(f_i/8\right)(Re_i - 1000)Pr_i}{1 + 12.7\left(f_i/8\right)^{1/2}\left(Pr_i^{2/3} - 1\right)} \text{for } 0.5 \le Pr_i \le 2000,\ 3000 \le Re_i \le 5{*}10^6 \text{ and } \left(L/D_i\right) \ge 10 \tag{7}$$

$$Nu_i = 0.027 Re_i^{4/5} Pr_i^{1/3}\left(\frac{\mu_i}{\mu_{s,i}}\right)^{0.14} \text{ for } 0.7 \le Pr_i \le 16700,\ Re_i \ge 10\,000 \text{ and } \left(L/D_i\right) \ge 10 \tag{8}$$

Where $\mu_{s,i}$ is the dynamic viscosity of the inner fluid at the temperature of the inner surface of the inner tube and f_i the friction coefficient with the inner surface of the inner tube calculated from the following relations [18]:

$$f_i = 0.316 Re_i^{-1/4} \text{ for } Re_i \le 2{*}10^4 \tag{9}$$

$$f_i = 0.184 Re_i^{-1/5} \text{ for } Re_i \ge 2{*}10^4 \tag{10}$$

$$f_i = \left(0.79 \ln Re_i - 1.64\right)^{-2} \text{ for } 3000 \le Re_i \le 5{*}10^6 \tag{11}$$

2. Outer flow

If the outer flow is laminar, the convective heat transfer coefficient can be calculated from Table 1. In Table 1, Nu_o is the

Table 1
Nusselt number for fully developed laminar flow in a circular annulus [18].

Di/Do	0.05	0.1	0.25	0.5	1
Nuo	17.46	11.56	7.37	5.74	4.86

Nusselt number of the outer flow defined as:

$$Nu_o = \frac{h_o(D_o - D_i)}{k_o} \tag{12}$$

Where k_o is the conductivity of the outer fluid.

If the outer flow is turbulent [18]:

$$Nu_o = \frac{(f_o/8)(Re_o - 1000)Pr_o}{1 + 12.7(f_o/8)^{1/2}\left(Pr_o^{2/3} - 1\right)} \text{ for } 0.5 \le Pr_o \le 2000,\ 3000 \le Re_o \le 5{*}10^6 \text{ and } \left(L/(D_o - D_i)\right) \ge 10 \tag{13}$$

$$Nu_o = 0.027Re_o^{4/5}Pr_o^{1/3}\left(\frac{\mu_o}{\mu_{s,o}}\right)^{0.14} \text{ for } 0.7 \le Pr_o \le 16700,\ Re_o \ge 10\,000 \text{ and } \left(L/D_o - D_i\right) \ge 10 \tag{14}$$

Where Pr_o is the Prandtl number of the outer fluid, μ_o its dynamic viscosity at the average fluid temperature and $\mu_{s,o}$ its dynamic viscosity at the surface temperature.

Re_o is the outer Reynolds of the outer flow calculated as:

$$Re_o = \frac{4\dot{m}_o}{\pi(D_o - D_i)\mu_o} \tag{15}$$

Where $\dot{m}_o$ is the mass flow rate of the outer fluid.

f_o is the friction coefficient with the outer surface of the inner tube calculated from the following relations [18]:

$$f_o = 0.316Re_o^{-1/4} \text{ for } Re_o \le 2{*}10^4 \tag{16}$$

$$f_o = 0.184Re_o^{-1/5} \text{ for } Re_o \ge 2{*}10^4 \tag{17}$$

$$f_o = (0.79 \ln Re_o - 1.64)^{-2} \text{ for } 3000 \le Re_o \le 5{*}10^6 \tag{18}$$

Now to calculate the heat transfer rate that can be recovered from the condenser hot air, one of the outlet temperatures should be calculated. To proceed, Eqs. (1)–(4) are rearranged in order to obtain one equation with only one unknown the water outlet temperature $T_{w,out}$:

$$\dot{m}_w C_{p,w}\left(T_{w,out} - T_{w,in}\right) = UA\frac{\left[T_{a,in} - \frac{\dot{m}_w C_{p,w}}{\dot{m}_a C_{p,a}}\left(T_{w,out} - T_{w,in}\right) - T_{w,in}\right] - \left(T_{a,in} - T_{w,out}\right)}{\ln\left[\frac{\left(T_{a,in} - \frac{\dot{m}_w C_{p,w}}{\dot{m}_a C_{p,a}}\left(T_{w,out} - T_{w,in}\right) - T_{w,in}\right)}{\left(T_{a,in} - T_{w,out}\right)}\right]} \tag{19}$$

Eq. (19) is solved by iteration to obtain the water outlet temperature $T_{w,out}$ for given flow rates, inlet temperatures and specific heats of water and air. Then, the heat rate recovered from condenser air can be determined:

$$P_{exchanger} = \dot{m}_w C_{p,w}\left(T_{w,out} - T_{w,in}\right) \tag{20}$$

Now, to calculate the air flow rate $\dot{m}_a$ and inlet temperature $T_{a,in}$ that are input to the heat exchanger calculations presented above, the HVAC system side is considered. The heat rate $\dot{Q}_h$released by the refrigerant (gained by air) in the condenser and the heat rate $\dot{Q}_C$ absorbed by the refrigerant (lost by air) in the evaporator (cooling load) are related through the following equation:

$$\dot{Q}_h = \dot{Q}_C + \dot{W}_C \tag{21}$$

Where $\dot{W}_C$ is the work required to drive the compressor of the system.

$\dot{Q}_h$, $\dot{Q}_C$, and $\dot{W}_C$ are related by the coefficient of performance COP as follows:

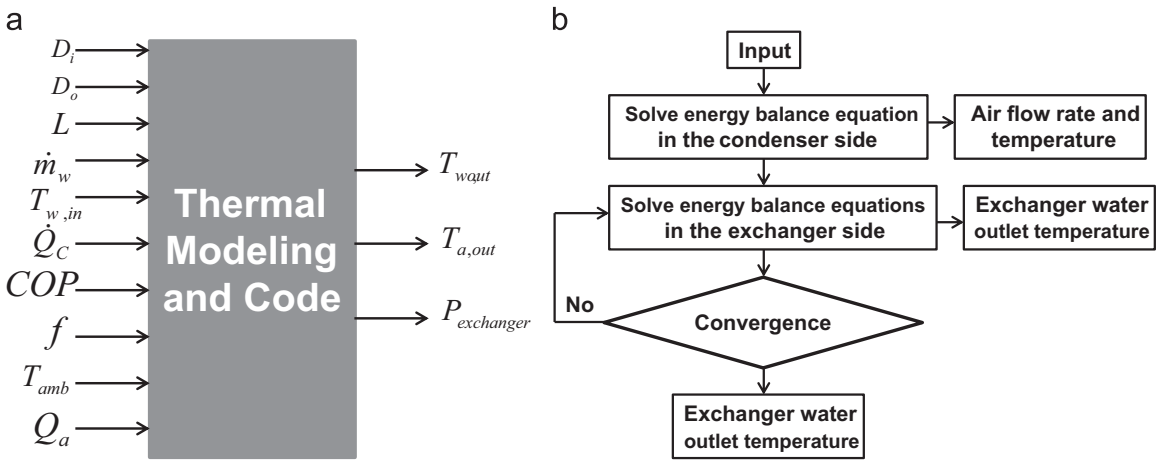

Fig. 3. Schematic of the (a) overall scheme and (b) operational mode of the code developed.

$$COP = \frac{\dot{Q}_C}{\dot{W}_C} = \frac{\dot{Q}_C}{\dot{Q}_h - \dot{Q}_C} \tag{22}$$

The heat balance at the condenser side can be written as follows:

$$\dot{Q}_{a,C} = f\dot{Q}_h = \rho_a Q_a C_{p,a}\left(T_{a,in} - T_{amb}\right) \tag{23}$$

Where $\dot{Q}_{a,C}$ is the heat gain of air as it passes through the condenser, f a correction factor which takes into account the heat losses in the condenser side, ρ_a the air density, Q_a the air volumetric flow rate, $C_{p,a}$ the air specific heat, $T_{a,in}$ is the air temperature after passing through the condenser and entering the exchanger of the recovery system and T_{amb} is the air ambient temperature.

The operational mode of the code developed for the present calculations consists on an iterative procedure manipulating the different equations presented in the thermal modeling above in order to have a set of input parameters and calculating the main output: the heat exchanger performance. Fig. 3(a) shows a schematic of the overall scheme of the code.

The code solves the problem iteratively using a partitioned algorithm allowing coupling the two physical problems: cooling the condenser by the air and heating the water by the heated air. The inputs of the developed code are: the operational parameters of the HVAC system, the characteristics of the heat exchanger, and the physical properties of air and water.

Fig. 3(b) shows schematically the operational mode of the code. In the first part of the solver the energy balance equation applied to the condenser side is solved, which gives the value of the air mass flow rate and temperature. In the second part the energy balance equation applied to the heat exchanger is solved. The temperature and mass flow rate obtained from the first part are input for the heat exchanger calculation part, whereas the temperature of the water is the inlet cold temperature. The solver iterates until reaching a convergence.

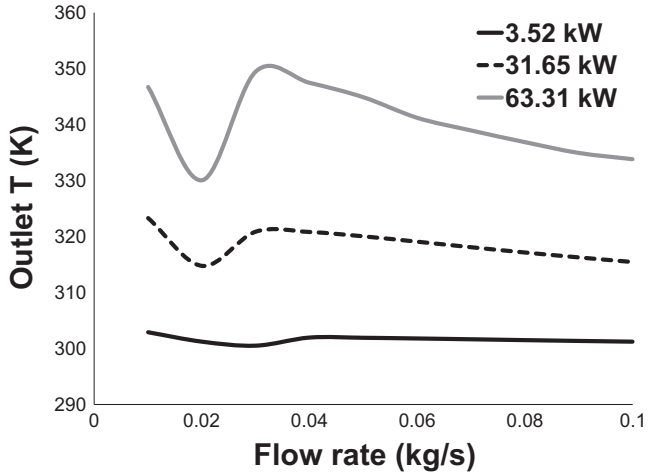

Fig. 4. Variation of the water outlet temperature in function of the water flow rate for different cooling loads.

3. Results and analysis

The thermal modeling presented in the previous section is employed to perform parametric analysis on the thermal performance of the HVAC heat recovery concept proposed. The performance of the heat exchanger of the system is explored when the water and air flow rates are varied. A counter-flow concentric tube heat exchanger of three meters length, 0.02 m inner diameter and 0.04 m outer diameter used to heat water by the condenser air is considered. The ambient temperature is considered equal to 30 °C and the water inlet temperature is fixed at 25 °C. The coefficient of performance *COP* of the HVAC system is considered equal to 3.

3.1. Effect of water mass flow rate

The water mass flow rate affects the heat transfer inside heat exchangers. Indeed, when the mass flow rate increases the heat rate changes. However, the outlet temperature does not change proportionally to the mass flow rate because it depends on other parameters such as the flow type and the physical parameters that generally changes with the temperature. To explore explicitly the relation between the exchanger (heat recovery concept) performance and the water flow rate, a first set of calculations is performed for the water flow rate varying from 0.01 to 0.1 kg/s for 3.52 kW, 31.65 kW and 63.31 kW cases. For this set of calculations the air volumetric flow rate is fixed at 0.8 m^3/s. Fig. 4 shows the water outlet temperature in function of the water mass flow rate for the different cooling load $\dot{Q}_c$ values.

The water outlet temperature increases with the cooling load $\dot{Q}_c$ which is due to a higher inlet air temperature caused by the higher heat rate $\dot{Q}_h$ exchanged at the condenser side. For a water flow rate of 0.01 kg/s, the water outlet temperature is equal to 304 K for $\dot{Q}_c$ equal to 3.52 kW; it increases with $\dot{Q}_c$ to reach 346.7 K for $\dot{Q}_c$ of 63.31 kW. At a water flow rate of 0.1 kg/s, the water outlet temperature increases from 301 K to 334 K with the value of $\dot{Q}_c$ increasing from 3.52 to 63.31 kW. On the other hand, the curves show variable monotonies in the water outlet temperature variations when increasing the water flow rate. One can notice two monotony changes in each variation. The first monotony is due to the transition from laminar flow regime to turbulent flow. The second change can be explained by two competing effects. Indeed when the mass flow rate increases, the heat transfer coefficient and the heat transfer rate increases. However, this transfer rate will be gained by a greater quantity per second of water. When the mass flow rate increase dominates on the heat transfer rate increase the water outlet temperature decrease as is the case in the laminar region and the second part of the turbulent region. If the increase in water heat rate however dominates on the water flow rate increase, this results on an increase in the water outlet temperature as is the case in the first part of the turbulent region. It is then at the limiting mass flow rates between the two situations described above that occurs the second change of monotonies.

Now, to well understand the first monotonies change, the Reynolds number in the inlet pipe (carrying water) is presented in terms of the mass flow rate in Fig. 5.

It can be shown that the transition from laminar to turbulent flow which occurs at a Reynolds number of 2300 is affected by the cooling load $\dot{Q}_c$. In other terms, the water flow rate at which the transition occurs is not the same between the different cases of cooling loads. The transition flow rate is decreased by $\dot{Q}_c$ increase. As illustration, the mass flow rate of transition is 0.033 kg/s for 3.52 kW, 0.026 kg/s for 31.65 kW and 0.022 kg/s for 63.31 kW.

3.2. Effect of air mass flow rate

To explore now the relation between the exchanger (heat recovery concept) performance and the air flow rate, a second

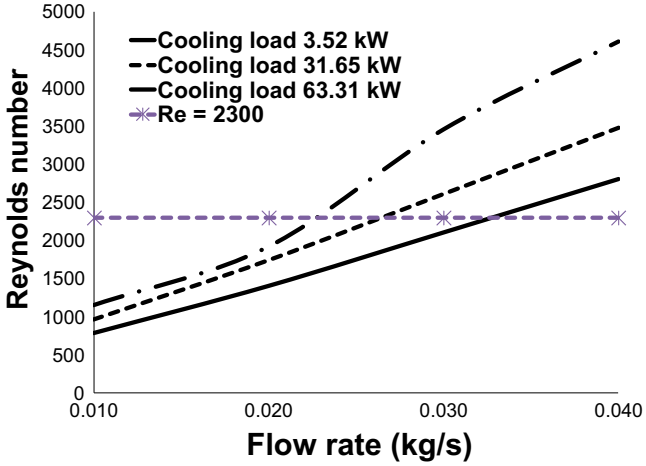

Fig. 5. Variation of the Reynolds number of the water flow in function of the mass flow rate for different cooling loads.

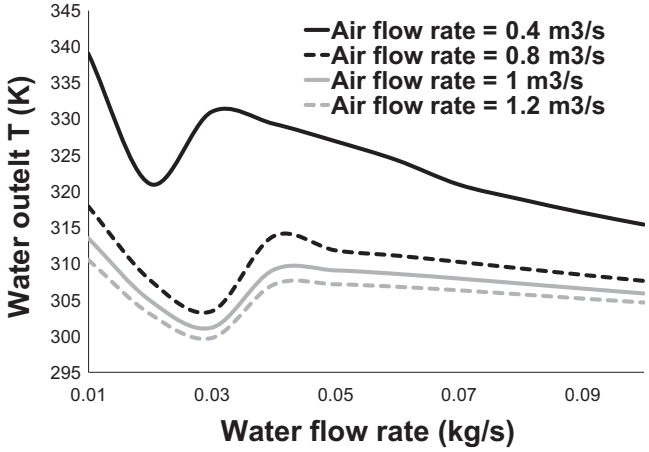

Fig. 6. Water outlet temperatures in function of the water flow rate for different air flow rates.

set of calculations is performed for the air volumetric flow rates of 0.4, 0.8, 1 and 1.2 m^3/s for the water mass flow rate varying from 0.01 to 0.1 kg/s. For this set of calculations the cooling load value is fixed at 31.65 kW. Fig. 6 shows the water outlet temperature in function of the water mass flow rate for the different air flow rates at constant cooling load equal to 31.65 kW.

It can be shown that the trends of Fig. 5 for varying the water flow rate are reproduced when the air flow rate is varied at constant cooling load value. Water outlet temperatures and monotonies changes in the curves are more significant at low air flow rates. As illustration for an air flow rate of 0.8 m^3/s, the outlet temperature decreases from 318 K to 303 K as the water flow rate increases from 0.01 to 0.03 kg/s at which the transition to turbulence in the water flow occurs and from which the outlet temperature start to increase. It increases then from 303 to 314 K when the flow rate increases to 0.04 kg/s and then decreases to 308 K as the flow rate remain increasing to 0.1 kg/s. At an air flow rate of 0.4 m^3/s, the outlet temperature decreases from 339 K to 321 K as the water flow rate increases from 0.01 to 0.02 kg/s at which the transition to turbulence in the water flow occurs and from which the outlet temperature start to increase to 331 K at 0.03 kg/s of water and then decreases to 315 K when the water flow rate remain increasing to 0.1 kg/s.

It should be noticed that the first change in monotonies of the curves corresponding to the transition to turbulence of the water flow is fastened when the air flow rate decreases. In other terms, when the air flow rate decrease for the same cooling load, the transition to turbulence in the water flow occurs at lower flow rates and the minimum in the curves is fastened.

Fig. 7 shows the variations of the water outlet temperature in terms of the air flow rate for different air flow rates. The cooling load value is fixed at 31.65 kW.

It can be noticed that the water outlet temperature decreases when the air flow rate increases for a constant cooling load of the HVAC system. As illustration for a water flow rate of 0.02 kg/s (laminar flow), the water outlet temperature decreases from 321 K to 303 K when the air flow rate increases from 0.4 to 1.2 m^3/s, let a decrease of 18 K. The water outlet temperature decrease with the air flow rate is much more sensible to the air flow rate in turbulent water flow cases. To exemplify for a water flow rate of 0.04 kg/s (turbulent flow), the water outlet temperature decreases from 329 K to 307 K, let a decrease of 22 K, when the air flow rate increases from 0.4 to 1.2 m^3/s.

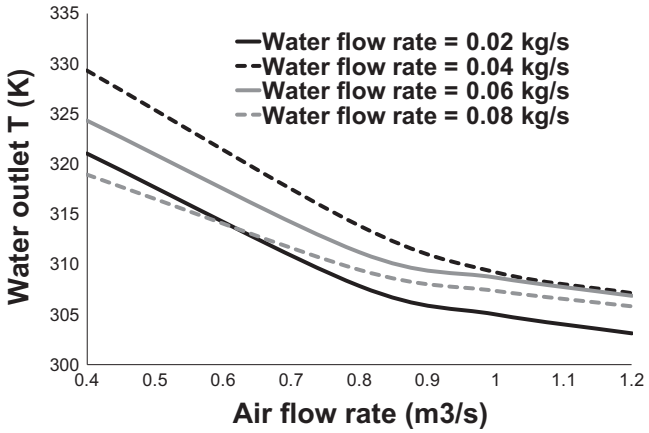

Fig. 7. Water outlet temperatures in function of the air flow rate for different water flow rates.

Indeed, when the air flow rate increases for a constant cooling load value of the refrigeration cycle, the air temperature downstream of the condenser decreases. This air temperature decrease induces a decrease in the thermal performance of the heat exchanger of the recovery system. At the same time, the increase in air flow rate induces an increase in the exchanger performance. The effect of the air temperature is dominant on that of the air flow rate in such a manner that the water outlet temperature decreases with the air flow rate. It should be noticed also that starting from an air flow rate of almost 1–1.2 m^3/s, curves reach asymptotic values corresponding to cases where the effects of air flow rate and temperature become in the same order of magnitudes.

4. Conclusions

In the present paper, a heat recovery system is presented. It allows taking advantage of the heat released by the condenser of HVAC systems working on the refrigeration cycle to heat/preheat water. A complete modeling of the design is presented and an iterative code is developed.

It is found that the water outlet temperature increases with the cooling load of the HVAC system. For a mass flow rate equal to 0.01 Kg/s, the water outlet temperature increases from 304 K to 347 K when the cooling load increases from 3.52 to 63.31 kW.

For a fixed cooling load and water flow rate, the outlet temperature of water decreases when the air flow rate passing through the condenser is increased.

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