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Propositions of improvement of the cross-flow M-Cycle heat exchangers in different air-conditioning applications

Demis Pandelidis[1,*], *Sergey* Anisimov[1], *Krzysztof* Rajski[1], and *Ewa* Brychcy[2]

[1]Wroclaw University of Science and Technology, Department of Environmental Engineering, Wybrzeże Wyspianskiego 27, 50-370, Wroclaw, Poland
[2]Consultant, Researcher of Aurae Cooling Technologies Limited, Hong Kong

Abstract. This paper presents results of mathematical simulation of the heat and mass transfer in the two different Maisotsenko Cycle (M-Cycle) heat and mass exchangers used for the indirect evaporative cooling in different air-conditioning systems. A two-dimensional heat and mass transfer model is developed to perform the thermal calculations of the indirect evaporative cooling process, thus quantifying the overall heat exchangers' performance. The mathematical model was validated against the experimental data. Numerical simulations reveal many unique features of the considered units, enabling an accurate prediction of their performance. Results of the model allow for comparison of the two types of heat exchangers in different applications for air conditioning systems in order to obtain optimal efficiency.

1 Introduction

In recent years, the increase in summer temperatures, improved insulation of the buildings, and a growth of indoor facilities have led to an increased requirement for air conditioning in buildings. Conventional mechanical vapor-compression air-conditioning systems consume a large amount of the electrical energy that is largely dependent upon a fossil fuel. This mode of air conditioning is, therefore, neither sustainable nor environmentally-friendly. Due to the increasing need for air conditioning and the growing interest in energy savings, seeking ways to reduce fossil fuel consumption and to increase usage of the renewable energy during air-conditioning process in building sector is a matter of great importance. Due to this fact, this paper focuses on analysis of two types of Maisotsenko cycle heat and mass exchangers (HMXs), which are able to generate significant savings in the air conditioning systems and therefore reduce the energy consumption. First unit is a currently produced device, while the second one is the proposition of an improvement proposed by authors.

* Corresponding author: demis.pandelidis@pwr.edu.pl

(a) (b)

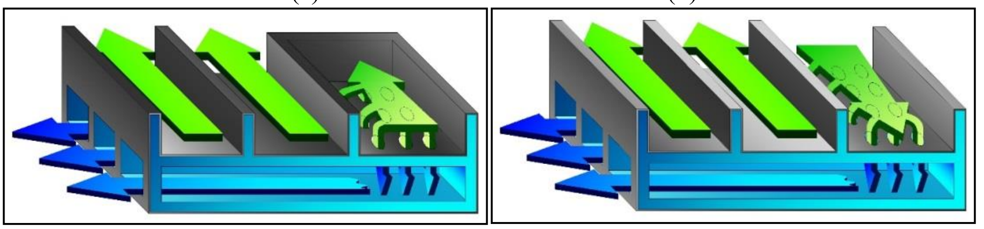


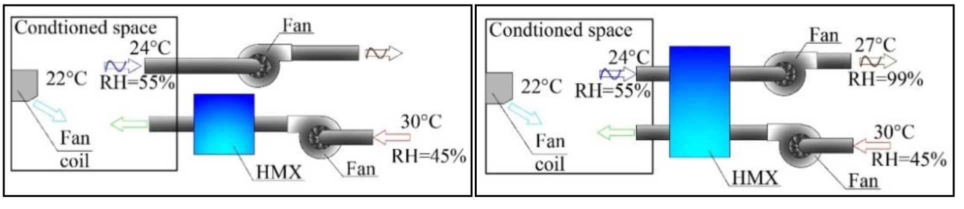
Fig. 1. Analyzed HMXs: (a) original cross-flow HMX (HMX1); (b) modified cross-flow HMX (HMX2).

The HMXs considered are shown schematically in Figure 1. The first device (HMX1 – Fig. 1(a)) is currently produced by Coolerado Corporation, where the working and the primary air flows in the dry channels are flowing parallel to each other. The second unit (HMX2- Fig. 1(b)) is the modification of the first cross-flow HMX, where entrance to the dry channel for the working and the product air flow are placed on the opposite sides of the HMX, which allows for implementation in different configurations in air-conditioning systems. Both units can be used as an individual cooling coils, which are the only source of cooling power in the system. However, in many air conditioning systems, especially in the public buildings and offices, the individual rooms typically use fan coil units to provide individual comfort for the occupants (Fig. 2(a), (b), (d)). The exhaust air for the systems with the original HMX is removed to from the outside or it is partly recirculated to the primary air flow (Fig. 2(a)). The modified HMX allows for the system to be implemented in the supply-exhaust flow stream in the air-handling unit (Fig. 2(b)). Therefore it can operate on the exhaust air, which is colder than the ambient air. The system presented in Figure 2(b) can be replaced with a system having a heat pump, presented in Figure 2(c). The operation base in this case is the same as in supply-exhaust system with the fan coil units. In more humid climates, there is a possibility to use the original HMX with the desiccant wheel (Fig. 2(d)), which dries the air flow and increases its temperature [1]. The arrangements presented require different operational conditions for the presented HMXs, with different temperatures and relative humidities of the primary and the working air-flow entering each exchanger. It is essential to establish the conditions for which it is more reasonable to use one HMX instead of another.

2 Methods

All the analysis is based on the numerical model based on the modified ε-NTU method. All the details of the model are presented in [2], along with the validation, transformation of the heat balance equations and detail analysis of the algorithm. Due to the fact that the model was presented in the past, it will be omitted in this paper.

(a) (b)



(c) (d)

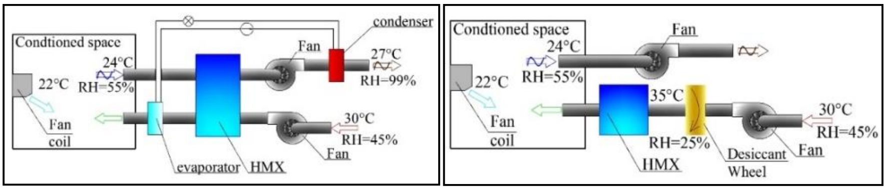


Fig. 2. Analyzed heat exchangers in different applications for air conditioning systems: (a) HMX1 in air conditioning system with fan coils; (b) HMX2 in air conditioning system with fan coils; (c) HMX2 in air conditioning system with heat pump; (d) HMX1 in air conditioning system with desiccant wheel.

3 Results and discussion

The reference operating conditions for the analyzed exchangers are presented in Table 1. The considered exchangers will be compared for the three hypothetical inlet conditions, representing different operational possibilities for the air conditioning system:

1. Both exchangers have the same inlet parameters for the working and the product air channels: $t_{1i}= t_{3i}$; $RH_{1i}=RH_{3i}$; (the purpose of this study is the general comparison between the two exchangers for the same operational conditions),

2. For HMX1 the working and the product air streams' inlet temperature and humidity are the same, while HMX2 have different values of the primary and the working air inlet temperature and humidity (comparison between the HMX1 working in the supply air system presented in Fig. 2(a) and HMX2 working in the supply-exhaust air system shown in Fig. 2(b) and (c)). In this case, the inlet temperature of the working air flow is lower than the inlet temperature of the product air flow for HMX2,

3. HMX1 has hot and dry inlet air conditions (the same for the product and the working air flow), while HMX2 has different values for the primary and the working air inlet temperature and humidity (comparison between HMX1 working in the desiccant system presented in Fig. 2(d) and HMX2 working in the supply-exhaust air system visible in Fig. 2(b) and (c)).

Table 1. The reference operating conditions for the analyzed exchangers.

Length, m	Width, m	Channel height, mm	Working to primary air ratio, -	Primary air stream velocity, m/s
0.5	0.5	3.0	1.0	3.0

Three main parameters (indices) have been selected to study the operational performance of the investigated HMX configurations:
• temperature of outlet primary air flow;
• the wet bulb thermal effectiveness, which is defined as the ratio of the difference between intake and outlet process air temperature to the difference between intake process air temperature and its wet bulb temperature [3, 4];

$$\varepsilon_{WB} = \frac{t_{1i} - \overline{t}_{1o}}{t_{1i} - t_{1i}^{WB}} \tag{1}$$

• the specific cooling capacity per cubic meter of the HMX structure

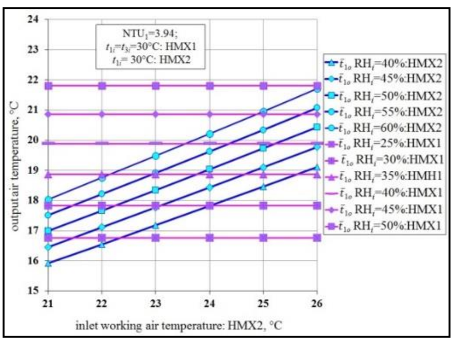
$$\hat{Q} = Q_1 / V_{HMX} \tag{2}$$

where $Q_1 = G_1 c_{p1} \left(t_{1i} - \overline{t}_{1o} \right)$ is the cooling capacity of the HMX and V_{HMX} is the volume of the HMX structure, m^3.

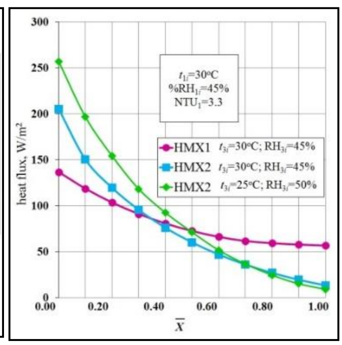
3.1 Analysis at different operational conditions

A set of simulations was performed in order to compare the effectiveness of the considered exchangers in the different operational conditions in the air conditioning systems. Figure 3(a) presents comparison of systems visible in Figure 2(a)–(c). For the purpose of this analysis, HMX1 is using only on ambient air, while in HMX2 the ambient air enters only the product air channels and the working air channels are operating on the exhaust air from the conditioned space. Depending on the type of conditioned space, the exhaust air may have a different temperature and relative humidity; however the temperature inside the conditioned space is colder than the ambient temperature, while its relative humidity is usually a little higher. Simulations were conducted for an ambient air temperature equal 30°C, and the relative humidity of the ambient air was varied from 25 to 50%. The exhaust air temperature was changed from 21 to 26°C, with 1°C increments, while its relative humidity was changed from 40 to 60%. The charts (Fig. 3(a), (c) and (d)) are presented for constant levels of inlet temperatures of the primary air flow (t_{1i}), which are listed in frames in the upper part of the charts. For HMX1 the inlet temperatures represent different operation in air conditioning system (operation only on ambient air or in a system with a desiccant wheel. HMX2 operates on the ambient air flow in the primary air channels and exhaust air from the conditioned spaces in the working air channels. The X axis in Figs. 3(a), (c) and (d) represents the temperature of the exhaust air, which enters the channels of the HMX2. It's important to mention, that the parameters of the exhaust air don't affect the outlet air temperatures obtained by HMX1 (for this unit $t_{1i}= t_{2i}$), therefore values of t_{1o} for this unit are presented as a constant lines. Blue lines represent the outlet air temperatures obtained by the HMX1 for constant inlet air temperature ($t_{1i}= t_{2i}$) and different inlet relative humidities of the working air flow (listed in the frame on the right side of each chart). The black lines represent the outlet temperatures obtained by the HMX2 for constant level of inlet primary air temperature (t_{1i}), variable inlet working air temperature (presented on the X axis) and variable inlet working air relative humidity (listed in frame on the right side of each chart). It can be seen, that the inlet parameters have significant impact on the efficiency of considered systems. For moderate climate conditions (RH_i = 45 to 50%) outlet temperatures obtained by HMX1 were equal 21.8°C and 20.8°C. At the same time, for the typical indoor conditions (t = 24 to 26°C; RH_i = 50 to 60%) HMX2 achieved outlet temperatures varying from 19.0°C (t_{3i} = 24°C and RH_{3i} = 50%) to 21.75°C (t_{3i} = 26°C and RH_{3i} = 60%). This shows that HMX2 operating on the exhaust air stream at the entrance to the wet channel has more effective performance than the HMX1, which operates only on the ambient air. This phenomenon can be explained by analyzing the heat flux profiles (Fig. 3(b)). Although the heat flux profile for the HMX2 operating on the exhaust air is similar to the profiles characteristic for the parallel-flow exchangers, the heat transfer rate is higher than the heat transfer rate of the same unit using ambient air. The higher value of the heat transfer rate, caused by the lower temperature of the working air, results in more efficient cooling of the primary air. As a result, HMX2 can achieve higher effectiveness than the HMX1 operating on outside air in moderate climates (Fig. 3(a)). However, in more dry climatic conditions (RH_i = 35 to 40%) HMX1 achieves higher efficiency than HMX2 using exhaust air (for most of the typical indoor conditions). It can be seen that the detail analysis of the ambient and indoor air parameters is essential to achieve the highest efficiency of the designed air-conditioning system. The next set of simulations was performed in order to compare the HMX1 operating in system with the desiccant wheel (see Fig. 2(d)) with HMX2 operating on the exhaust air (see Fig. 2(b) and (c)). The desiccant wheel dehumidifies the ambient air and it additionally increases its temperature. Depending on the material used for the wheel structure and the range of the dehumidification process, the air temperature may increase or decrease depending on the conditions. For HMX2, the inlet temperature of the primary stream was assumed to be 30°C, while for HMX1 two types of the inlet conditions were selected for this study: dehumidification with a small increase in inlet temperature ($t_{1i} = t_{3i}$ = 3 2.5°C; RH_i = varied) and dehumidification with a high increase in inlet temperature $t_{1i} = t_{3i}$ = 35.0°C; RH_i = varied). The results of the comparison are shown in Figure 3(c) and (d), respectively. It can be observed, that for the inlet temperature equal to 32.5°C and $RH_i \leq 25\%$, HMX1 achieves lower outlet temperatures than HMX2 operating on the exhaust air with the typical indoor conditions (Fig. 3(c)).

(a)



(b)



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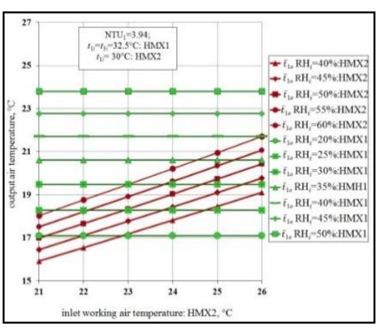

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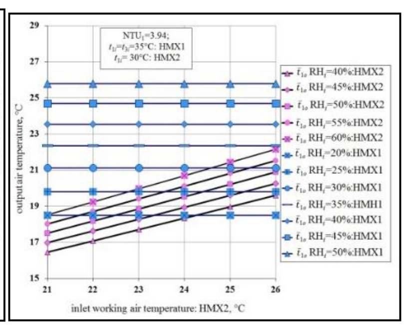

Fig. 3. Simulation results for HMXes operating in different inlet conditions: (a) outlet temperatures HMX1: t_{1i} = 30°C; RH_i = 25 to 50%; HMX2: t_{1i} = 30°C; t_{3i} = 21 to 26°C; RH_i = 40 to 60%; (b) heat flux distribution inside the primary air channels; (c) outlet temperatures: HMX1: t_{1i} = 32.5°C; RH_i = 20 to 50%; HMX2: t_{1i} = 30°C; t_{3i} = 21 to 26°C; RH_i = 40 to 60%; (d) outlet temperatures: HMX1: t_{1i} = 35°C; RH_i = 20 to 50%; HMX2: t_{1i}=30°C; t_{3i} = 21 to 26°C; RH_i = 40 to 60%.

When the temperature of the inlet air is raised to the higher level after dehumidification, the HMX1 obtains higher outlet temperatures (Fig. 3(d)). In this case, for the inlet temperature equal to 35.0°C and RH_i = 25%, HMX1 achieves higher outlet temperatures than HMX2 operating using exhaust air with typical indoor conditions. However, for relative humidities equal to 20%, HMX1 shows better performance than HMX2 for the most of the typical indoor conditions (Fig. 3(d)). This shows that the inlet air stream for HMX1 has to be dehumidified to very low values of the relative humidity to overcome the performance of HMX2 operating on the exhaust air. Table 2 presents the outlet temperature, specific cooling capacity and the wet bulb effectiveness obtained by HMX1 and HMX2 for the selected inlet parameters. The analysis of Table 2 illustrates another interesting observation: HMX1 for the inlet conditions of t_{1i} = 32.5°C; RH_i = 20% obtained lower outlet temperature than HMX2 (for t_{1i} = 30°C; t_{3i} = 24°C and RH_{3i} = 40%), however its wet bulb effectiveness is 32.5% lower. HMX2 (for t_{1i} = 30°C; t_{3i} = 24°C and RH_{3i} = 40%) obtained a lower outlet temperature than HMX1 (for t_{1i} = 35.0°C and RH_i = 20%), however its specific cooling capacity is 4.4 kW/m^3 lower. This shows that the wet bulb effectiveness and the specific cooling capacity aren't adequate to describe the performance of the HMXs operating in different arrangements in the air-conditioning system. The higher specific cooling capacity for the air with higher inlet temperature is caused by the higher difference between inlet and outlet product air temperatures. Greater temperature difference, for the higher temperature of the inlet air flow causes a very intensive evaporation process in the wet channels, which improves the efficiency of cooling process and results in greater cooling capacity. However, the additional cooling capacity can be used only for the reduction of the primary air flow temperature to level before the air stream is heated in dehumidification process and thus it doesn't give any energy benefit. An explanation of the trends in the wet bulb effectiveness presented in Table 2 is that the wet bulb effectiveness refers to the wet bulb temperature of the product air. In case of HMX2, the air stream entering the wet channel has a different temperature, relative humidity and therefore different wet bulb temperature. In this case, the efficiency factor based on the primary air's wet bulb temperature is not informative and can lead to an incorrect conclusions. Therefore it shouldn't be used for the description of indirect evaporative exchangers with different primary and working air inlet parameters.

Table 2. Outlet temperature, specific cooling capacity and wet bulb effectiveness of HMX1 and HMX2.

	HMX1: t_{1i} = 32.5°C; RH_i = 20%;	**HMX1: t_{1i} = 35.0°C; RH_i = 20%;**	**HMX2: t_{1i} = 30°C; t_{3i} = 24°C; RH_{3i} = 40%;**	**HMX2: t_{1i} = 30°C; t_{3i} = 24°C; RH_i = 50%;**
$\bar{t}_{1o}$, °C	17.1	18.5	18.17	19.35
$\hat{Q}$, kW/m^3	33.4	35.5	31.1	28.0
ε_{WB}, %	114.5	115.0	147.0	132.6

It can be seen on the basis of Figure 3, that in many cases the HMX2 can show a better performance than HMX1, especially for the colder indoor parameters.

4 Summary and conclusions

This study presents the theoretical analyses of the M-Cycle cross-flow heat exchangers used for the indirect evaporative cooling in air-conditioning systems. The units were compared in different operational conditions, representing potential different applications in air-conditioning systems. Modified ε–NTU-model based on the accurate assumptions was proposed for the mathematical analysis of the heat and mass transfer processes inside the exchangers. The basic operating parameters were given in Table 1. The discrepancies in effectiveness characterizing the considered HMXes were explained on the basis of the unique features of heat and mass transfer phenomenon occurring in each of the units. It was established that: The energy efficiency of the considered HMXs strongly depends on the inlet air parameters. HMX2 operating on the exhaust air with the typical indoor parameters in the working air channels shows higher effectiveness than HMX1 operating on ambient air in moderate climate conditions; HMX1 operating in the system with the desiccant wheel shows higher efficiency than HMX2 operating on indoor air, when air stream is highly dehumidified (RH*i* = 25% and lower). The material used for the desiccant wheel construction should allow for intense dehumidification of the air stream and a relatively small temperature increase in order to obtain the lowest outlet temperatures; In systems, where the air stream is additionally cooled by a mechanical compression system, it is reasonable to partly recirculate the exhaust air stream in systems with HMX1. The wet bulb effectiveness and the specific cooling capacity are not informative when comparing the indirect evaporative exchangers operating on the different primary and working air inlet parameters.

Nomenclature

c_p – specific heat capacity of moist air, J/(kg·K); G – moist air mass flow rate, kg/s; NTU – number of transfer units, NTU=$\alpha F/(Gc_p)$; $\hat{Q}$ – specific cooling capacity respected to 1 m^3 of the exchanger structure, kW/m^3, W; RH - relative humidity, %; t – temperature, $\bar{t}$ – average temperature, °C; 1 – main (primary) air flow; 2 – working air flow in the wet channels (product part of exchanger); 3 - working air flow in the dry channels (pre-cooling part of exchanger); 4 – working air flow in the wet channels (pre-cooling part of exchanger); i – inlet; o – output; WB - wet bulb temperature; $\overline{X}$ – relative X coordinate, dimensionless, $\overline{X} = X/l$;

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