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## Performance comparison of round tubes finned heat exchangers and macro micro-channel heat exchangers in a low capacity heat pump

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### ABSTRACT

Micro-channel heat exchangers (MCHE) are used in automobile applications due to their low weight and high compactness. Those MCHE are just gaining interest in stationary application and they have a great potential for low heating capacity heat pumps to be installed in “passive houses” where the heating demand is 3 to 5 times lower than in the current new individual houses built in European countries.

In this paper, a low capacity integrated air to air heat pump prototype is used to perform an experimental comparison of both a round finned tube and a MCHE used as an evaporator. The MCHE is a prototype one using flat fins. The comparison is based on a constant face surface criterion and an equivalent cooling capacity for different inlet temperature and air flow rate conditions. R-134a is the refrigerant used in the heat pump.

First both evaporators' performances are compared experimentally and the results show a clear advantage for the MCHE. In a second step, the behavior of the MCHE evaporator is studied under frosting conditions. The frosting kinetics and the defrosting behavior are analyzed. The results show a behavior similar to round tube finned heat exchangers with frosting cycle higher than 45mn and a defrosting using ventilation exhaust air in less than 10mn.

### 1. INTRODUCTION

The global political agenda is loaded by a new series of negotiations on climate change and improving the energy efficiency is one of the major issues. Low-energy buildings are a key to reduce energy consumptions. Their heating demand is very low (15-50 kWh/m<sup>2</sup>.yr). Heating systems adapted to this low demand are under development and MCHE seems to be a solution for low heating capacity heat pumps.

The publications agree on the fact that the internal heat transfer coefficient (HTC) in MCHE increases with the heat flux. The effect of several other parameters such as the vapor quality and the mass flux received less attention, and showed in some cases opposed tendencies in the literature. For example, Hamdar et al (2010), Bao et al. (2000) and Lazarek and Black (1982) suppose a heat transfer coefficient nearly constant and independent of the vapor quality and the flow regime in the micro-channels. On the other hand, Saitoh et al. (2005) found a meaningful increase in the heat transfer coefficient for high vapor quality, as it is generally observed in tubes of large diameter. Bertsch et al. (2008a), Yan and Lin (1998), Yun et al. (2006), and Lin et al. (2001) showed a peak in the heat transfer coefficient for vapor qualities between 0.1 and 0.6, then this coefficient drops sharply while the vapor quality increases. All these results permit to conclude on high internal HTC in the MCHE even with parallel refrigerant distribution. This leads to high compactness heat exchangers with very low pressure drop on the refrigerant side. Such characteristics (in addition to refrigerant charge reduction, weight and cost) led to prefer MCHE on conventional round tube heat exchangers first in the automotive air conditioning systems and more recently in stationary applications.

These studies are usually based on tests for high mass fluxes ( $G > 150 \text{ kg/m}^2\cdot\text{s}$ ) which are usually taking place in automotive air conditioning systems and large stationary equipments. Low capacity heat pumps presents low refrigerant mass flow rate leading to low mass fluxes. Mortada et al.(2012) evaluated the evaporation HTC for low mass fluxes ( $G < 40 \text{ kg/m}^2\cdot\text{s}$ ). Their results showed a clear dependence of the HTC on the flow regime and a small dependence of the vapor quality. This behavior is close to large diameter tubes and one may reconsider the

actual conclusions on the MCHE performances and advantage on the conventional round tubes finned heat exchangers.

In this paper, a low capacity integrated air to air heat pump prototype is used to perform an experimental comparison of both a round finned tube and a MCHE used as an evaporator. The comparison is based on a constant face surface criterion and an equivalent cooling capacity for different inlet temperature and air flow rate conditions. R-134a is the refrigerant used in the heat pump. First both evaporators were designed using a tube by tube evaporator model. Both evaporators' performances are compared experimentally and the results show a clear advantage for the MCHE. In a second step, the behavior of the MCHE evaporator is studied under frosting conditions.

## 2. TUBE BY TUBE EVAPORATOR MODEL

### 2.1 Governing equations

This model is developed to allow numerical simulation of various types of air source heat exchangers. The model calculates the thermodynamic properties of the fluid (temperature, enthalpy, pressure, and vapor quality). Programmed in C++, it integrates Refprop module to calculate thermodynamic properties.

Having chosen the heat exchanger type, different elements are described in a (.txt) file: working fluids and their mass flow rates, geometry description, number of rows and tubes, air and refrigerant inlet conditions, selected correlations as well as the circuit.

Conservation of mass, momentum, and energy are the leading equations for resolution. They are applied on each tube.

Having fixed geometry parameters of the tubes and knowing the characteristics of the fluid and air at the inlet of each tube, the model calculates the fluid and air outlet conditions thanks to the energy balance. Thanks to the circuit description, the outlet conditions of a tube are the inlet conditions of the next tube. When the circuit presents a counter current tendency, the resolution is iterative.

The refrigerant input properties are:

- The mass flow rate,
- The inlet pressure of the heat exchanger, which is equal to the first tube pressure,
- And, the inlet enthalpy of the heat exchanger, which is equal to the inlet enthalpy of the first tube.

Air input properties are:

- Air speed,
- Air temperature and relative humidity.

### 2.2 Resolution method

Energy and mass are balance written on both sides (air and refrigerant) in steady state.

*Refrigerant side*

Equation of energy of energy conservation on the refrigerant side is written in (Eq.1).

$$\dot{m}_{In} * i_{In} - \dot{m}_{Out} * i_{Out} - Q_{ref} = 0 \quad \text{Eq. 1}$$

The internal heat transfer coefficient is calculated as a function of the fluid state. In fact, the vapor quality is calculated indicating the fluid state: liquid, vapor or in two-phase. Then an adapted correlation is used. The heat exchange between the refrigerant and the tube wall is written in equation 2.

$$Q_{ref} = h_{int} * S_{int} * (T_f - T_w) \quad \text{Eq. 2}$$

*Air side*

Equation 3 is the energy conservation on the air side.

$$Q_{ref} + \dot{m}_{In,a} * i_{In,a} - \dot{m}_{Out,a} * i_{Out,a} = 0 \quad \text{Eq. 3}$$

The sensible and latent heat capacities are calculated. The external heat-transfer coefficient is calculated using an adapted correlation.

The wall temperature is compared to the dew point in order to detect air condensation. Equations (4 to 8) are solved for the air side. The sensible and the latent capacities are calculated using Equations 4 and 5. The analogy between heat and mass transfer is assumed.

$$Q_s = h_{ext} * S_{ext} * (T_a - T_w) * \eta_{fin} \quad \text{Eq. 4}$$

$$Q_l = \dot{m}_{In,a} * L_v * (w_{In,a} - w_{Out,a}) \quad \text{Eq. 5}$$

$$Q_l = h_{ext,wet} * S_{ext} * L_v * (w_{In,a} - w_w) * \eta_{fin} \quad \text{Eq. 6}$$

$$h_{ext,wet} = \frac{h_{ext}}{Le * cp_a} \quad \text{Eq. 7}$$

$$Q_{ref} = Q_l + Q_s \quad \text{Eq. 8}$$

$w_w$  corresponds to the saturated air absolute humidity at the wall temperature. The Lewis number is assumed to be equal to 1.

In addition to these equations, the momentum equation is solved permitting to determine the pressure variation through the tube.

### 2.3 Correlations

The micro channel evaporator studied uses plain fins in order to avoid frost formation problems. For the air side, the existing correlations that define the heat transfer coefficient and the pressure drop for finned-tube heat exchangers are restricted to round finned tubes with all types of fins and flat tubes with louvered fins and slit fins.

There were no studies in the literature on the air thermal and hydraulic behavior of plain-fin flat-tube heat exchangers. The airflow pattern in this type of heat exchanger is similar to that of a channel flow with simultaneously growing thermal and hydraulic boundary layers (Jacobi et al. 2001).

The first study about this technology was recently published in Sweden by Stignor (2009). Two different types of plain-fin flat-tube heat exchangers were considered, one with a serpentine fin and one with continuous plate fins. The model developed is based on a previous paper not published yet but in which the airflow is assumed to be fully developed. Corrections were done in order to reflect results from the experimental study presented by the same author. This recently developed correlation will be used in this study to calculate the air-side heat transfer coefficient.

The pressure drop on the air side is essentially the one corresponding to developing duct flows. The physics of channel flows will be applied to calculate the air pressure drop.

For the refrigerant side, the heat transfer coefficient is calculated depending on the refrigerant quality. If the fluid is single-phase (liquid or vapor), the heat transfer coefficient is calculated using Dittus-Boelter correlation. The pressure drop in this case is calculated using Blasius equation.

When the refrigerant is in two-phase state, the heat transfer coefficient is calculated using Mortada et al (2012) correlation.

The fluid pressure loss is calculated using the Friedel (1987) correlation and the Hughmark model for the void fraction calculation.

**Table 1:** Summary of correlations used in this study.

		Micro-channel evaporator	Conventional evaporator
Air side	Heat transfer coefficient	Stignor (2009)	Wang & Chi (1999)
	Pressure drop	Jacobi et al.(2001)	Wang & Chi (1999)
Refrigerant side	Heat transfer coefficient	Mortada et al (2012)	Bivens (1994)
	Pressure drop	Friedel	Friedel
	Void fraction	Hughmark	Hughmark

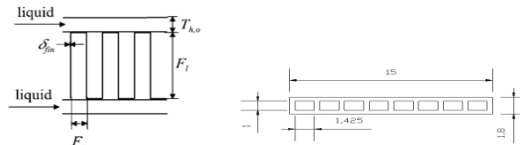
## 2.4 Results

The heat exchangers design is done in a parametric study in order to select for each technology, the best possible design. The main constraint is the prototype duct geometry which imposes the face area ( $0.105 \text{ m}^2$ ).

For the round heat exchanger technology (only one circuit), the varied parameters are:

- Tube external diameter within the standard values: 7.94 mm, 9.52 mm
- The fin pitch: 2 mm, 3.2 mm
- Number of rows: 1, 2 and 3

This MCHE has 2 passes in each of its two rows. Each pass has 16 parallel tubes. Figure 1 presents a schematic description of the micro-channel evaporator (MCHC).



**Figure 1:** Schematic description.

This prototype MCHE is numerically compared to several round tube finned heat exchangers. The selection criteria are based on the cooling capacity (targeting equivalent cooling capacity) but also refrigerant side and air side pressure drop.

The parametric study has shown that only a 2 rows round tube heat exchanger is feasible and comparable to the MCHE. Indeed, with only 1 row, the cooling capacity generated is not sufficient while when using three rows, the pressure losses on both air side and refrigerant side become very important.

When considering 2 rows, the best solution is found with the smallest tube diameter and the smallest fin pitch.

Table 2 presents the geometry details of the designed heat exchangers. Both, the MCHE and the round tube heat exchanger use plain fins. The fin spacing for the two evaporators is identical at 2 mm.

**Table 2:** Geometry characteristics

	MCHE	Round
Tube		
Length (m)	0.2	0.315
Width (mm)	15	
Height (mm)	1.8	
Tube Thick (mm)	0.4	0.3
Number of channels	8	
Hydraulic Diameter (mm)	1.1	
Tube OD (mm)		7.93
Number	64	18
Material	Aluminium	Copper
Fins	Aluminium	Aluminium
Pt (mm)	10	22
Number of rows	2	2
Fin pitch (mm)	2	2

Table 3 presents the numerical comparison of the performance of both heat exchangers operating in three conditions.

**Table 3:** Numerical comparison

	Round			MCHE		
	Cooling capacity (W)	Internal pressure losses (Pa)	Air side pressure losses (Pa)	Cooling capacity (W)	Internal pressure losses (Pa)	Air side pressure losses (Pa)
Condition 1 (air inlet temp. 14.5°C)	569.9	2419.8	1.9	667.7	824.9	1,42
Condition 2 (air inlet temp. 7°C)	618	2681.5	3.9	681.5	876.8	3,3
Condition 3 (air inlet temp. -2.4°C)	731.6	5555.4	7	812.5	1882.8	5,9

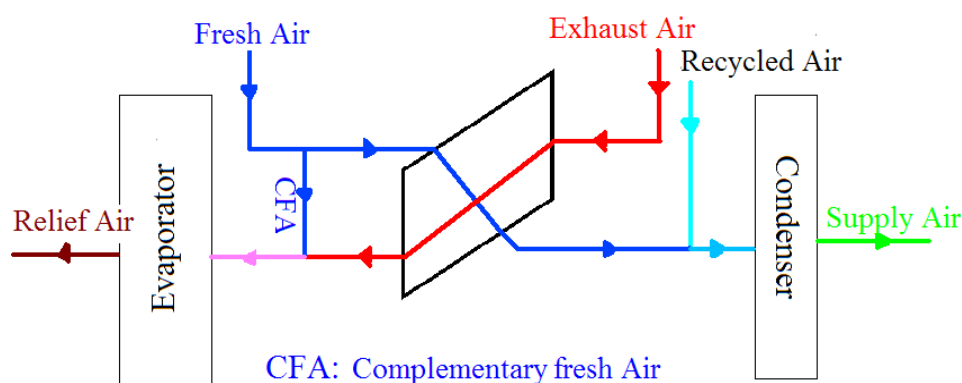
Table 3 shows that the cooling capacity of the round tube heat exchanger is 10 to 15% lower than the one obtained with the MCHE. In addition, the air side pressure losses are lower in the case of the MCHE. The refrigerant side pressure losses are much higher in the case of the round tubes heat exchanger. This is mainly due to the circuitry. Indeed, the round tube heat exchanger contains 18 tubes that are connected in series while the MCHE presents 2 passes per row with 16 tubes per pass alimented in parallel.

### 3. EXPERIMENTAL STUDY

#### 3.1 Prototype heat pump description

The air-to-air prototype heat pump extracts heat from the outside air and transfers it to the air inside the house via a refrigeration loop. It is a low capacity heat pump integrated in the ventilation system (Mortada et al, 2010). The studied system Mini Heat Pump (MHP) is based on the same principle of an air to air heat pump (EHP). The difference in this system is the use of a recovery heat exchanger (RHX). This additional heat exchanger will permit the heat transfer between the exhaust air and the fresh air through the separating walls of the heat exchanger. Both sensible and latent heat may be transferred. Latent heat is transferred when moisture in the exhaust air condensates on the heat exchanger. Moisture is not transferred. The power recovered may reach 30% of heating demand when it is a high demand and sometimes 100% if the outside temperature exceeds 9 °C. According to the heating demand, the integrated MHP is used to complete the heating of a mixed air before passing it on the condenser. This mixing is created from the fresh air preheated by the recovery heat exchanger and a recycled airflow. Depending on this demand a variable speed fan is regulated to provide variable airflow rates. On the other hand, the exhaust air passing through the additional heat exchanger is mixed to a complementary fresh airflow before passing on the evaporator. This complementary fresh air is also regulated by a variable speed fan.

To adapt to the needs, the compressors are variable-speed compressors. The condenser is a mini-channel heat exchanger (MCHE). On the evaporator side, both designed evaporators can be mounted and tested. Figure 2 shows a schematic description of the studied systems.



**Figure 2:** Schematic description of the used heat pump prototype.

### 3.2 Test bench

The MHP is a low-capacity heating system. The low ranges of airflow rates make their measurements difficult and involve a precise metrology. Two small climatic rooms were built, tested, and calibrated in order to measure heat losses through their walls. These climatic rooms handle air flows: from one room the air is exhausted and fresh air is blown in the other. The two rooms communicate via dampers ensuring the pressure balance. A small chiller is used to generate the heating needs in each climatic room. The cooling capacity is adjusted for each outdoor temperature. The MHP is designed to deliver the required heating demand.

Pitot tubes installed inside the ducts are used to measure the airflow rates. This system measures the total and static pressures and calculates the dynamic pressure then the air velocity. Dry and wet bulb temperatures are measured for each air flow rate, especially at the evaporator inlet and outlet. Pressure transducers and thermocouples are positioned at the inlet and the outlet of each component in the refrigerant loop. Compressor and fan speeds are controlled via field-point modules and their input powers are measured. Figure 3 shows a lay-out of the test bench and connections between the climate rooms and the MHP.

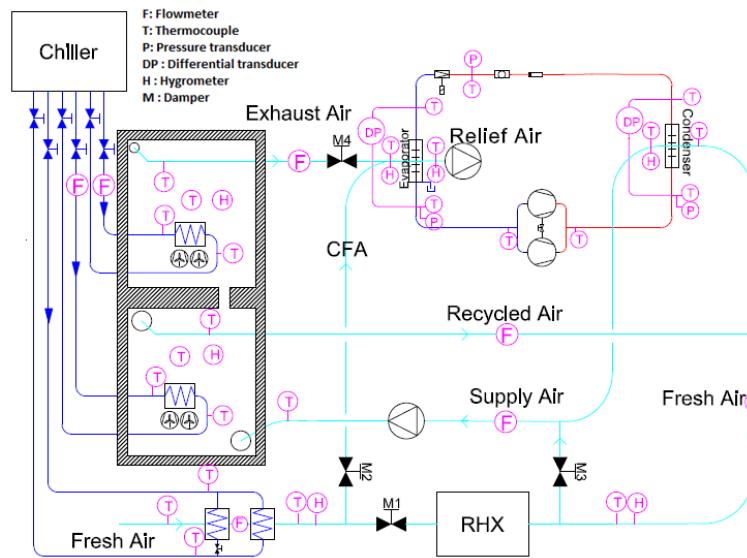


Figure 3: Schematic lay-out of the test bench

### 3.3 Experimental evaporator comparison results

Tests were performed with both designed evaporators. The working refrigerant is the R-134a. Three operating conditions were tested representing three ambient conditions. For each test condition, the fixed parameters were:

- Heat load to be tackled by the heat pump;
- Air mass flowrate and inlet temperatures at evaporator and condenser.

The heat pump controlled the electronic expansion valve to settle the superheat, the compressor frequency to match the heat capacity and the fans frequencies to match the air mass flowrate.

The test results with both evaporators for the three testing conditions are given in table 4.

Table 4: Experimental results

Evaporator type	Evaporator inlet air temp. (°C)	Heating cap. (W)	Cooling cap. (W)	Air side pressure losses (Pa)	HP COP	System COP
MCHE	15.43	831.95	791.1	1.34	6.93	4.75
Round	15.1	821.3	771.2	4.95	5.87	4.19
MCHE	7.7	744	704.7	2.9	6.62	3.62
Round	7.02	711.8	676.4	9.63	5.35	3.19
MCHE	-1.1	822.4	763.8	12.6	6.33	2.91
Round	-0.73	827.5	753.2	21.3	5.75	2.87

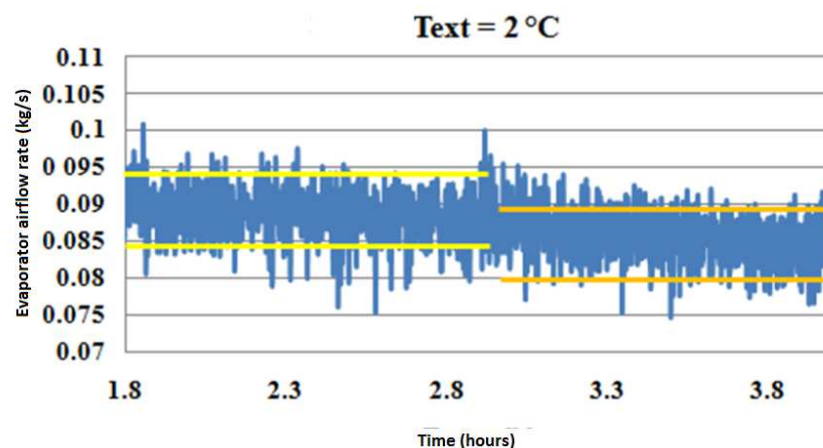
In the experimental situation, since the heat load is matched (less than 5% difference), and since the expansion valve is controlling the superheat, the evaporation pressure is not identical for both evaporator and the best comparison is done on the heat pump COP (defined as the heating capacity divided by the compressor power input) and also the system COP (defined as the heating capacity divided by the compressor and fans power input).

The impact of the evaporator technology on the heat pump COP is very important leading to a difference ranging between 10 to 19%. The COP difference is mainly explained by the evaporating pressure difference due to higher heat transfer capacity of the MCHE compared to the round tube evaporator. The internal pressure losses influence also the compressor power input and affects the heat pump COP. When introducing the fans power input, in that case we compare the system COP, the difference is less important due to the high level of fans power input compared to the compressor. This is a particularity of this heat pump because of its low capacity and the fact that it is integrated in the ventilation system. The system COP difference ranges between 1.5 to 12%. The last comparison parameter is the air side pressure losses. The experimental results show an important difference between the air side pressure losses of the MCHE and the round tube heat exchanger. The observed difference is much higher than the simulated one. Indeed, the model seems to underestimate the air side pressure losses for the round tube heat exchanger.

### 3.4 MCHE behavior in frosting conditions

Due to the presence of the recovery heat exchanger (RHX), the average air temperature at the evaporator inlet is higher than the external temperature. For that reason frost appears on the evaporator of the prototype heat pump at lower external air temperatures than the classical air to air heat pumps.

In the performed tests, frost was observed at external temperatures lower than 2°C. In this section, we focus on the behavior of the MCHE during frost build up. Figure 4 shows the evolution of the air mass flowrate at the evaporator.



**Figure 4:** Evaporator air mass flowrate during frost build up

From figure 4, two periods can be identified:

- The first three hours of operations: the frost formation causes a slight air flow reduction without having an impact on the heat pump performance
- The last 45 minutes: the air flow decreases by 10% starting to influence the low pressure of the heat pump and its capacity.

This behavior is comparable to round tube heat exchanger where cycles of frost/defrost are in the same order of magnitude.

Figure 5 shows a photography of the evaporator just before the defrost cycle begins. As it can be observed on the photography, frost formation is not homogeneous on the evaporator surface.





**Figure 5:** Photography of the MCHE

This behavior is typical of misdistribution. One drawback of MCHE is the difficulty to distribute the refrigerant between the tubes; this can lead to non homogeneous frost formation. However, another parameter of influence is the air temperature homogeneity. This seems to be the case, for the heat pump prototype, as shown in the figure 5, frost is more formed in the area where the complementary air arrives while no frost is observed on the area where the relief air arrives.

## CONCLUSIONS

In this paper, a numerical and experimental comparison of two evaporators showed that the micro-channel parallel flow evaporator is more efficient than a conventional round finned tube evaporator. The particularity of this comparison is that it concerns a very low capacity heat pump. For a given face area, MCHE has a capacity higher than the round tube evaporator. This leads for the heat pump to operate at lower evaporation pressure with round tubes leading to lower COP. Another feature of the MCHE is the reduced air side and refrigerant side pressure losses. In addition to the performance difference, the advantage of using a MCHE is its small internal volume and then the possibility of lower refrigerant charge.

One usually cited drawback of MCHE is their behavior in frosting conditions. Standard MCHE are known to have a fast frost formation cycle with dramatic performance degradation. In this paper, the usage of flat fins on the MCHE evaporator permitted to obtain similar performances to round tubes heat exchangers under frosting conditions.

## NOMENCLATURE

h	heat transfer coefficient	$W.K^{-1}.m^{-2}$
i	enthalpy	$kJ.kg^{-1}$
$L_v$	latent heat	$kJ.kg^{-1}$
$\dot{m}$	mass flow rate	$kg.s^{-1}$
P	pressure	Pa
Q	capacity	W
RH	relative humidity	%
S	surface	$m^2$
T	temperature	$^{\circ}C, K$
UA	thermal conductance	$W.K^{-1}$

x	vapor quality	$\text{kg.kg}^{-1}$
w	absolute humidity	$\text{kg.kg}^{-1}_{\text{da}}$
ref	refrigerant	

**Subscript**

n	nights
p	participants

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