

2010

# Energy Efficiency of Air-to-Air Mini Heat Pump

Sorina Mortada

*Center for Energy & Process/ EDF R&D*

Assaad Zoughaib

*Center for Energy & Process/ EDF R&D*

Christine Arzano-Daurelle

*Center for Energy & Process/ EDF R&D*

Denis Clodic

*Center for Energy & Process/ EDF R&D*

Follow this and additional works at: <http://docs.lib.purdue.edu/iracc>

---

Mortada, Sorina; Zoughaib, Assaad; Arzano-Daurelle, Christine; and Clodic, Denis, "Energy Efficiency of Air-to-Air Mini Heat Pump" (2010). *International Refrigeration and Air Conditioning Conference*. Paper 1078.

<http://docs.lib.purdue.edu/iracc/1078>

This document has been made available through Purdue e-Pubs, a service of the Purdue University Libraries. Please contact [epubs@purdue.edu](mailto:epubs@purdue.edu) for additional information.

Complete proceedings may be acquired in print and on CD-ROM directly from the Ray W. Herrick Laboratories at <https://engineering.purdue.edu/Herrick/Events/orderlit.html>

## ENERGY EFFICIENCY OF AIR-TO-AIR MINI HEAT PUMP

Sorina MORTADA<sup>1\*</sup>, Assaad ZOUGHAIB<sup>2</sup>, Christine ARZANO-DAURELLE<sup>3</sup>, Denis CLODIC<sup>4</sup>

<sup>1,2,4</sup> Center of Energy and Processes, Ecole des Mines de Paris,  
5 Rue Leon Blum 91120 Palaiseau, France

<sup>1,3</sup> EDF R&D Avenue des Renardières  
Ecuelles 77818 MORET-SUR-LOING, France

<sup>1</sup> Phone: +33-1-69194233, Fax: +33-1-69194501

Email address: [sorina.mortada@mines-paristech.fr](mailto:sorina.mortada@mines-paristech.fr)

<sup>2</sup> Phone: +33-1-69194518, Fax: +33-1-69194501

Email address: [assaad.zoughaib@mines-paristech.fr](mailto:assaad.zoughaib@mines-paristech.fr)

<sup>3</sup> Phone: +33-1-60737181, Fax: +33-1-60736560

Email address: [christine.arzano-daurelle@edf.fr](mailto:christine.arzano-daurelle@edf.fr)

<sup>4</sup> Phone: +33-1-69194502, Fax: +33-1-69194501

Email address: [denis.clodic@mines-paristech.fr](mailto:denis.clodic@mines-paristech.fr)

### ABSTRACT

Mini-channel heat exchangers (MCHE) are used in automobile applications due to their low weight and high compactness. Those MCHEs have just gained interest in stationary applications 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 the current new individual houses built in European countries.

This paper compares performance and efficiency of a low-capacity heat pump (< 2 kW) with mini-channels as evaporator and condenser (mini-heat pump MHP) and an existing heat pump (EHP). This MHP is based on a third heat exchanger that will ensure an important part of a passive house heating demand.

A numerical model of a heat pump is presented in order to achieve this comparison.

### 1 INTRODUCTION

Climate change affecting the world pushed the governments to take actions to reduce greenhouse gases and air pollution. Global emissions of greenhouse gases should be divided by 2 before 2050. At the European level, the energy-climate package is a plan 3 x 20% by 2020. This plan is based on:

- 20% energy saving,
- 20% renewable energy, and
- 20% reduction of green house gases emissions

To reduce energy consumption, standards are reinforced in the building sector because it represents 47% of the energy consumption in France (industry, agriculture 28%, transportation 25%).

A low-energy consumption building, as defined in the RT 2005, is a new building that consumes less than 50 kWh/m<sup>2</sup>.yr of primary energy (for heating, hot water, ventilation, lightening and cooling). The greenhouse gases (GHGs) emitted by those buildings are less than 5 kg CO<sub>2</sub>.m<sup>2</sup>/yr. For these buildings, the heating needs are estimated at 15 kWh/m<sup>2</sup>.yr.

Air-to-air heat pumps are installed in apartment buildings and individual houses. They were introduced in the 90s for heating buildings and have benefited from their introduction of significant improvements in performance; some of them are equipped with variable speed compressor.

Existing air-to-air heat pumps have a high heating capacity compared to the heating needs of a low energy building. Heating systems adapted to the low demand of low energy buildings are under development and MCHE seems to be a solution for low heating capacity heat pumps.

The purpose of this study is to describe a model of a low heating capacity heat pump (MHP) adapted for low energy buildings.

### 2 SYSTEM DESCRIPTION

#### 2.1 Mini-heat pump

An air-to-air heat pump extracts heat from the outside air and transfers it to the air inside the house via a refrigeration loop. Air-to-air heat pumps (EHP system) are usually in Europe split systems.

A new type of HP is developed in Europe, installed in the air supply exhaust system, allowing the recovery from a recovery heat exchanger (RHX); see Figure 1(a). This additional heat exchanger transfers the heat between the exhaust air and the fresh air. Both sensible and latent heats may be transferred. Latent heat is transferred when moisture in the exhaust air condensates in the RHX. The power recovered may reach 30% of the heating demand when it is a high demand and sometimes 100% if the outside temperature exceeds 9°C.

The integrated MHP is used to complement the heating demand. 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 RHX is mixed with 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, compressors will be variable-speed compressors. The condenser and the evaporator are mini-channel heat exchangers (MCHE). They are light, compact, and more efficient than conventional heat exchangers. For the same temperature, the MCHE presents a capacity higher by about 30% than the conventional heat exchanger and a lower pressure drop of about 33 % (Mortada 2010). Figure 1 shows a schematic description of the studied systems.

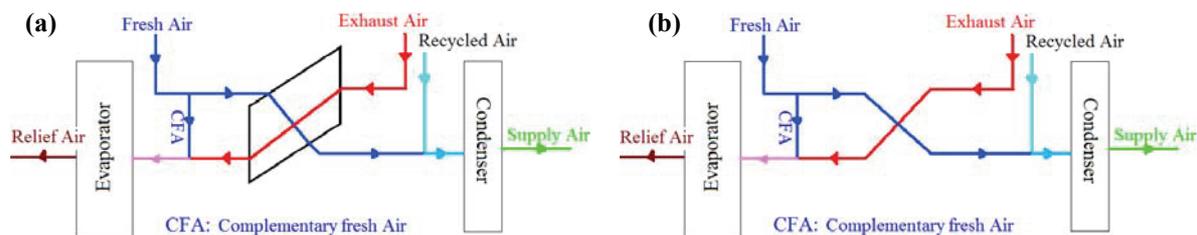


Figure 1: Schematic description of mini-heat pump (MHP).

## 2.2 Performance and control

The COP level of the studied MHP is directly affected by the RHX performance and the control system.

Based on the French standards, the ventilation must be permanent. The minimum fresh airflow rate is defined depending on the building type and the occupancy. This airflow rate is capable of covering the heating demand of a low energy building by passing it through a high efficiency recovery heat exchanger.

The control parameters are the airflow rate related to the fan speed and the temperature of the air supply related to a comfort temperature inside the house. The temperature of the supply air affects the condensation temperature. A higher supply air temperature involves a higher condensation temperature. The COP is then reduced due to the increase in the condensing pressure, which is due to:

- A greater compression ratio, the isentropic efficiency decreases
- A higher discharge temperature
- Degradation of the  $COP_{cycle}$  then degradation of the  $COP_{system}$ .

On the other hand, choosing a low condensing temperature leads to higher airflow rates then to higher fan electric consumption. This results in the degradation of the  $COP_{system}$ .

For optimizing  $COP_{system}$ , the control system has to find an agreement between the air supply temperature and the airflow rate.

In this study, the importance of the additional heat exchanger and the control system are presented via a model developed with an equation-based object-oriented modeling language Modelica.

## 3 MODEL

Dymola /Modelica is a solver able to solve numerical systems in a dynamic way. It supports hierarchical model composition, libraries components, connectors, and composite acausal connections.

### 3.1 Description

Each MHP component is integrated as a model in a package named PAC. Connectors are created to make the links between these components. Two types of connectors exist in this model, the air connector and the refrigerant connector.

Moist air thermodynamic properties are integrated as equations in a single package called HumidAir. A model is developed in PAC package dedicated to call HumidAir every time needed to calculate the air properties in the RHX, the condenser, and the evaporator. This model takes in consideration the air state (dry or wet) in the RHX and the evaporator. It calculates the latent and the sensible heats, and the condensed water amount.

The thermodynamic properties of the fluid (temperature, enthalpy, pressure, and vapor quality) are calculated using Refprop 8, which is interfaced with Dymola.

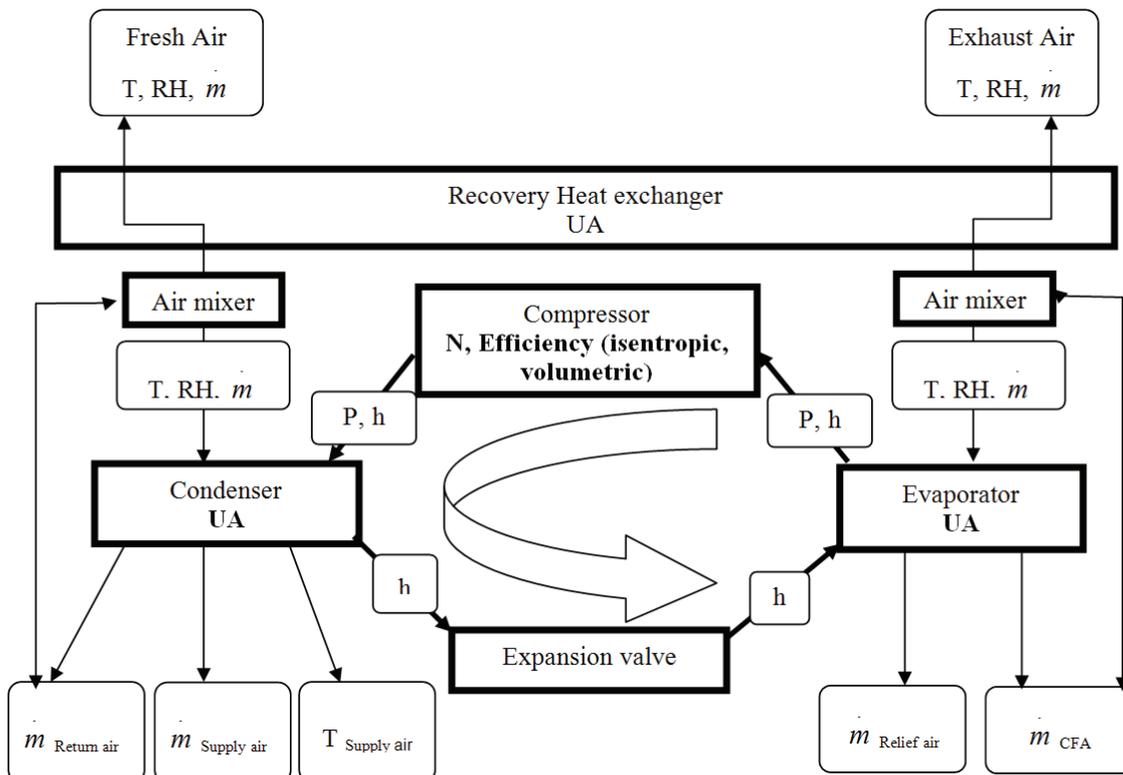


Figure 2: MHP system diagram.

Outlet air properties in the RHX are calculated by the effectiveness-NTU method. The overall thermal conductance of the heat exchanger is experimentally determined. Heat exchangers have been tested and a database of their global thermal conductance has been built. Equations (1) to (5) are used in this model.

$$NTU = \frac{(UA)}{C_{\min}} \quad (1)$$

$$\varepsilon = 1 - \exp \left[ \left( \frac{1}{C_r} \right) * NTU^{0.22} \left\{ \exp \left[ -C_r * NTU^{0.78} \right] - 1 \right\} \right] \quad (2)$$

$$Q = \varepsilon * C_{\min} * (T_{inExhaustAir} - T_{inFreshAir}) \quad (3)$$

$$Q = m_{Air} * (h_{out} - h_{in}) \quad (4)$$

$$Q_{ExhaustAir} = Q_{FreshAir} = Q \quad (5)$$

The recovery heat exchanger model calculates the mass flow of condensate water on the heat exchanger surface. The surface temperature is calculated and compared to the air dew point. When that surface is cooler than the dew point temperature, the condensation is observed. In this case, Q is calculated by Equation (6).

$$Q = \varepsilon * C_{\min} * (T_{inExhaustAir} - T_{inFreshAir}) + \text{Condensate} * L_v \quad (6)$$

The air mixer model is used in the MHP twice; it calculates the air properties of the air entering the evaporator and the condenser. Equations of energy and mass conservation are used.

$$m_{Air1} + m_{Air2} = m_{Airmix} \quad (7)$$

$$m_{Air1} h_{Air1} + m_{Air2} h_{Air2} = m_{Airmix} h_{Airmix} \quad (8)$$

$$m_{Air1} W_{Air1} + m_{Air2} W_{Air2} = m_{Airmix} W_{Airmix} \quad (9)$$

Depending on the heating demand, the compressor speed is controlled. Using the manufacturer database, the isentropic and volumetric efficiencies are computed as functions of the compression ratio and the compressor

speed. The compressor determines the refrigerant mass flow rate circulating in the system. Equations (11) to (15) are used to model the compressor behavior.

$$\eta_{is} = F4 * \tau^4 + F3 * \tau^3 + F2 * \tau^2 + F1 * \tau + F \quad (11)$$

$$\eta_{vol} = G4 * \tau^4 + G3 * \tau^3 + G2 * \tau^2 + G1 * \tau + G \quad (12)$$

$$m_{ref} = \frac{\eta_{vol} * N}{100 * 60} * D * \rho_{ref} \quad (13)$$

$$\eta_{is} = \frac{h_{out,is} - h_{in}}{h_{out} - h_{in}} \quad (14)$$

$$Q_{compressor} = m_{ref} * (h_{out} - h_{in}) \quad (15)$$

$\tau$  : compression ratio; F4,F3,F2,F1,F,G4,G3,G2,G1,G are functions of N (the compressor speed); D: displacement (m<sup>3</sup>)

For the condenser and the evaporator the Log Mean Temperature Difference (LMTD) method is used. Their UA (global conductance) is calculated with a detailed model called "Air-Hex" developed at the Center for Energy and Processes (Bigot 2001). "Air-Hex" is developed to perform numerical simulations of various types of heat exchangers. Having chosen the type of heat exchanger, different elements are described in a (.txt) file: working fluids and their mass flow rates, geometric description, number of rows and tubes, entry conditions, adapted correlations as well as the circuit. The resolution method is iterative: evaporation pressure varies until the air conditions (temperature and humidity) are constant. Conservation of mass and energy and momentum equations are solved under a tube-by-tube calculation scheme.

Based on Air\_Hex simulations, UA is defined for the condenser and the evaporator. They are integrated as a parameter in Dymola. The evaporation and the condensation temperatures are considered constant in the heat exchanger. The LMTD method is used to define these temperatures.

$$Q = UA * LMTD \quad (16)$$

$$Q = m_{ref} * (h_{ref,out} - h_{ref,in}) \quad (17)$$

$$Q = m_{Air} * (h_{Air,out} - h_{Air,in}) \quad (18)$$

An iterative calculation takes place to fix the condensation and evaporation pressures depending on the compressor behavior. The air moisture condensation is also taken into account in the evaporator model.

The model of the expansion valve is based on one equation.

$$h_{ref,out} = h_{ref,in} \quad (19)$$

### 3.2 Control scenario

The control system depends on the house heating demand. Therefore, a controller model is created. In the simulated scenario, fans are at constant speed. The variable speed fans will be considered for future simulations. The controller compares the load to the sum of the condenser and the air to air heat exchanger capacities. Three inputs are needed for the controller model, the outdoor temperature, the condenser capacity, and the air-to-air heat exchanger capacity. An output signal is sent to a first order controller controlling the variable speed compressor.

## 4 RESULTS

### 4.1 Model parameters

The air source variables used in this model are defined in Table 1.

UA of heat exchangers needed for the problem resolution and the compressor parameters for both cases MHP and EHP are shown in Table 2.

Table 1: Air source parameters

	Temperature (°C)	Relative humidity %	Flow rate (m <sup>3</sup> /h)
<b>Fresh air</b>	[-7;14]	50	105
<b>CFA</b>	[-7;14]	50	300
<b>Exhaust air</b>	20	60	105
<b>Recycled air</b>	20	60	300

Table 2: Calculation parameters

	RHX	Evaporator	Condenser	Compressor	
	UA (W/K)			Maximal Swept volume (m <sup>3</sup> /s)	Refrigerant
MHP	55	82	92	0.00122	R-134a
EHP		111	124	0.001788	R-134a

## 4.2 Results

### 4.2.1 Recovery heat exchanger (RHX)

Heating demand was previously calculated using the building simulation software COMFIE. Simulations have been done in different French regions and for the whole year. The maximal heating demand is 2 kW for a 112 m<sup>2</sup> house in the Trappes region (continental climate). This heating demand is compared to the heating capacity recovered by the RHX simulated in Dymola.

Figure 3 shows the heating capacity recovered by the RHX for an outdoor temperature higher than 5°C. RHX can ensure the total heating demand 43% of the time all over the heating season. In these conditions, the electrical consumption is restricted to the fans consumption.

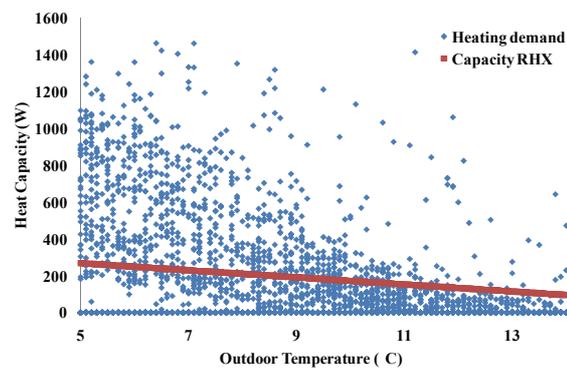


Figure 3: Heating demand and RHX as functions of the outdoor temperature.

### 4.2.2 System

When the RHX cannot ensure the total heating demand, the heat pump is used to cover the load. In this case, the compressor and the recycling ventilators are running. In order to compare the MHP (Mini heat pump) and the EHP (existing heat pump) the  $COP_{Thermodynamic}$  and the  $COP_{System}$  are calculated.

$$COP_{Thermodynamic} = \frac{Q_{Condenser}}{P_{Compressor}} \quad (20)$$

$$COP_{System\_MHP} = \frac{Q_{Condenser} + Q_{recoveryHX}}{P_{Compressor} + P_{Fans}} \quad (21)$$

$$COP_{System\_EHP} = \frac{Q_{Condenser}}{P_{Compressor} + P_{Fans}} \quad (22)$$

Three outdoor temperatures were chosen for this comparison: -7 °C an extreme case, -5°C and 5°C a mid case.

The  $COP_{Thermodynamic}$  is degraded when the outdoor temperature decreases. This is due to a lower evaporation temperature then a higher compression ratio. Increasing the low pressure implies a reduction of the compressor input power. For 5°C outdoor temperature, it is obvious that HEP  $COP_{Thermodynamic}$  is lower than that of the MHP. This is related to the UA of the heat exchangers in the EHP that becomes limiting.

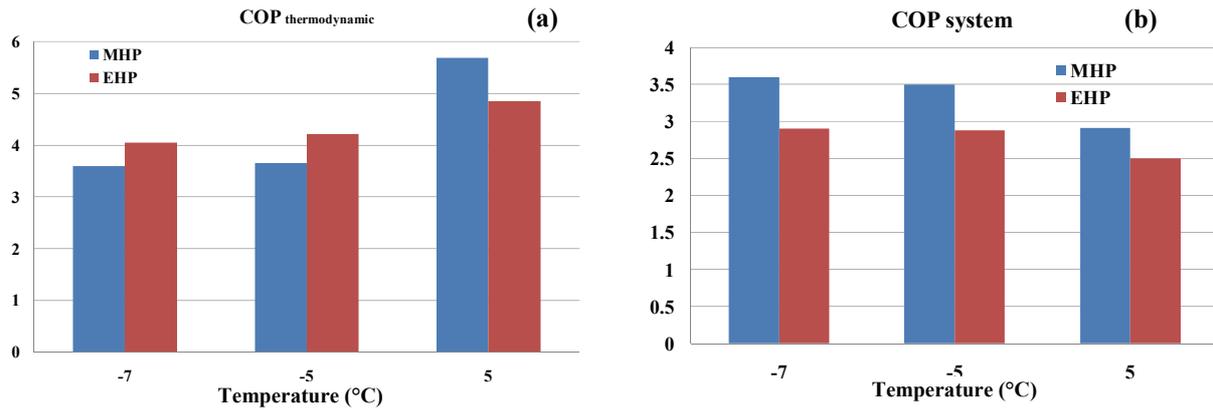


Figure 4: COP<sub>Thermodynamic</sub> (a) and (b) COP<sub>System</sub> as functions of the outdoor temperature.

The MHP COP<sub>System</sub> is higher than that of the EHP (see Figure 4 (b)). This is related to the heat capacity recovered by the RHX. It is also important to note that the COP<sub>System</sub> is degraded when the outdoor temperature increases due to the use of a constant air flow rate.

In fact, a higher outdoor temperature implies lower heating demand, then lower compressor speed and lower air flow rate may be used, which is not the case in the simulated scenario. French standards impose permanent ventilation with a minimum air flow rate. Therefore, minimum consumption of electrical fans is imposed. Figure 5 (a) shows that the electrical consumption of fans can be higher than that of compressors. In this case, the control of the variable speed fans according to the heating demand will permit the consumption reduction of fans and then improve the COP<sub>System</sub>.

$$\%_{Fans} = \frac{P_{Fans}}{P_{consumed}} * 100 \tag{23}$$

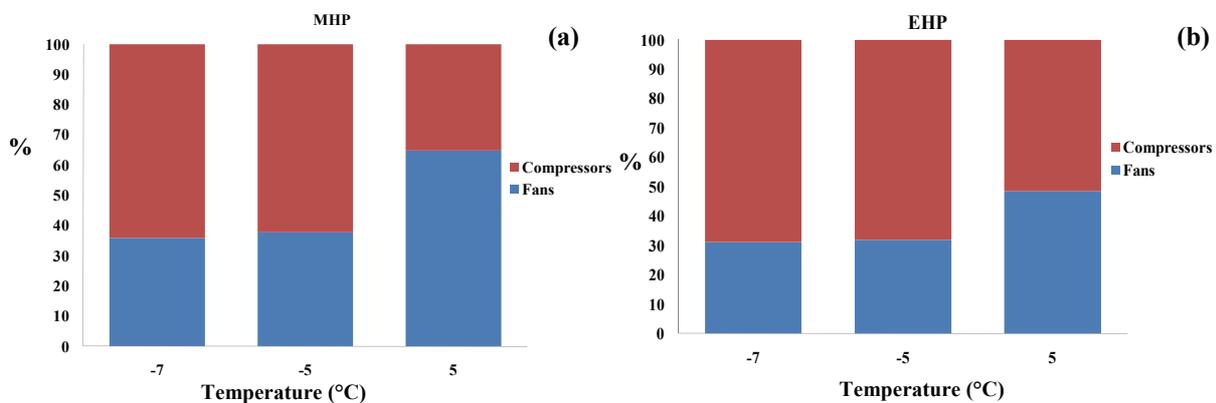


Figure 5: % of power consumed by the fans (MHP (a) & EHP (b)).

$$\%_{RHX} = \frac{Q_{RHX}}{Heatingdemand} * 100 \tag{24}$$

The RHX ensures about 35% of the heating demand for the extreme case -7°C as outdoor temperature (see Figure 6). Its presence is necessary for a higher COP<sub>System</sub>. The energy recovered decreases for a high outdoor temperature but it can be sometimes sufficient for covering the total heating demand.

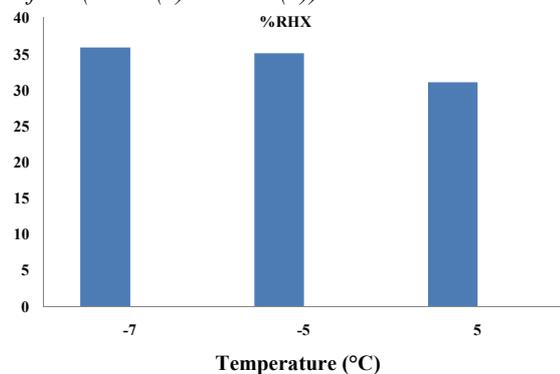


Figure 6: % of capacity generated by the recovery heat exchanger in MHP.

## 5 CONCLUSIONS AND PERSPECTIVES

A detailed model of a heat pump with an integrated third heat exchanger was presented. The MHP presented a higher COP<sub>System</sub> even with a non-optimized control. This is due to the essential role of the RHX that can ensure about 30% of the heating demand in the extreme climate conditions and 100% of the heating demand for 43% of the time during the heating season. The electrical consumption of fans affects remarkably the COP<sub>System</sub>.

The improved version of this model will include the control of the variable speed fans and the study of frost formation on the evaporator and the air to air heat exchanger.

### NOMENCLATURE

$C_{\min}$	minimum heat capacity	$W K^{-1}$	<b>Greek letters</b>	
D	displacement	$m^3$	$\rho$	density $kg m^{-3}$
Q	capacity	W	$\varepsilon$	effectiveness
h	enthalpy	$kJ kg^{-1}$	$\eta$	efficiency
Lv	latent heat of vaporization	$kJ kg^{-1}$	$\tau$	compression ratio
$\dot{m}$	mass flow rate	$kg s^{-1}$	<b>Subscripts</b>	
N	compressor speed	RPM	CFA	complementary fresh air
P	pressure	bar	comp	compressor
	power	W	COP	coefficient of performance
RH	relative humidity	%	EHP	existing heat pump
T	temperature	K	is	isentropic
UA	overall conductance for heat transfer	$W K^{-1}$	min	minimal
			MHP	mini heat pump
			ref	refrigerant
			RHX	recovery heat exchanger
			vol	volumetric

### REFERENCES

1. ASHRAE Handbook, 2005, *Fundamentals*, Chapter 6: *Psychometrics*,
2. Bigot G., 2001, *Étude et conception de systèmes air/air inversables utilisant des mélanges à glissement de température*, Center for Energy and Processes, Paris
3. COMFIE, Center for Energy and Processes, France. <http://www.izuba.fr/logiciel/pleiadescomfie>
4. Dynasim AB. Dymola—dynamic modeling laboratory. Lund, Sweden. <http://www.dynasim.se/dymola.htm>
5. Incropera F., Dewitt D., 1996, *Fundamentals of Heat and Mass Transfer*, Wiley, 886 p.
6. Mortada S., 2010, Heat transfer performance of a mini-channel evaporator for heat pump application, *Sustainable Refrigeration and Heat Pump Technology, KTH Stockholm, Sweden*.
7. RT 2005, *Décret relatif aux caractéristiques thermiques et à la performance des constructions*, (Order of 8 may 2007)
8. Slim, R. et al., 2008, Modeling of a solar and heat pump sludge drying system, *Int. J. Refrig*, doi:10.1016/j.ijrefrig.2008.03.003
9. The Modelica Association. Modelica –A unified object-oriented language for physical systems modeling. Language Specification Version 3.1, 2009. [www.modelica.org](http://www.modelica.org).