Energy and Exergy analysis of an HVAC system

Thesis submitted for the degree of

Master in Energy

Author :
Alberto Martin QUINTANILLA MUÑOZ

Advisors :
PhD. Alexandre VAUDREY
from ECAM Lyon, School of Engineering

PhD. Luis CHIRINOS
from PUCP Graduate School

Jury :

PhD. Alexandre VAUDREY	ECAM Lyon, School of Engineering
PhD. Vincent CAILLÉ	ECAM Lyon, School of Engineering
PhD. Julio CUISANO	PUCP

Lima – Peru

2017

Abstract

The efficient use of energy is a major issue nowadays. Environmental and economic purposes push various investigations to focus on the performances of energy systems and equipment. In the context of the coming energy transition, Heat, Ventilation and Air Conditioning (HVAC) systems will certainly take an increasing and worldwide importance. In this work, energy and exergy analysis are used to assess the performances of each component of an air treatment station. Results of energy and exergy analysis for each process are presented. The most important result is that simple heating and cooling processes with deshumidification have the worst exergy efficiencies; and that both processes represent almost all the exergy losses of the studied HVAC system.

Acknowledgements

I would first like to thank my project advisor PhD. Alexandre VAUDREY of the Energy Department at ECAM Lyon. The door to Prof. VAUDREY office was always open whenever I ran into a trouble spot or had a question about my project.

I would also like to thank the Director of the Master of Energy at Pontifical Catholic University of Peru, PhD. Luis Chirinos; the Director of the Energy Department at ECAM Lyon, PhD. Vincent CAILLÉ; and the Director of International Relations at ECAM Lyon, Edith FREY for giving me the opportunity to continue my studies at the PUCP and the ECAM Lyon in the framework of a double degree.

Finally, I must express my very profound gratitude to my parents for providing me with unfailing support and continuous encouragement throughout my years of study. This accomplishment would not have been possible without them. Thank you.

Author,

Martin QUINTANILLA

Index

Table of figures

Nomenclature

Nomenclature

Cp_a : Specific heat of air (kJ/kg.K)

Cp_v : Specific heat of vapour

ex : Specific exergy

$\dot{E}x_{\dot{Q}}$: Thermal or work exergy

$\dot{E}x_{dest}$: Exergy destroyed

h : Specific enthalpy

L_v : Enthalpy of vaporization

p : Pressure

p_{atm} : Atmospheric pressure

p_{sat} : Saturation pressure

p_v : Partial vapour pressure

$\dot{Q}$: Heat flow

$\dot{Q}_{loss}$: Heat loss

r_a : Gas constant for the dry air

s : Entropy

T_0 : Ambient temperature

T : Absolute temperature in Kelvin

X : Quality

Greek letters

α : Recirculation rate

η : Energy efficiency

θ : Temperature in degrees Celsius (°C)

$\emptyset$: Relative humidity

ψ : Exergy efficiency

ω : Specific humidity

$\tilde{\omega}$: Specific humidity on a molal basis

A. Introduction

1. Context

The efficient use of energy is a major issue nowadays. Environmental and economic reasons make various investigations focus on the performance of energy systems and equipment. *Heat, Ventilation and Air Conditioning (HVAC)* systems are one of the most used in the world.

In the year 2000 for example, HVAC systems represented 69% of the final energy consumption in the French residential sector. The residential and tertiary sectors represent 44% of the final energy consumption, which amounted for 1 796 TWh in 2012. For renovated low-consumption building, HVAC represents 48% of energy consumption (55%for a house) and 34% for a new low-energy building (42% for a house).

Even if we consider 50% of consumption for HVAC in buildings (all renovated old buildings) in 2012, the final energy consumption was 898 TWh. In France, the housing stock is old (67% was built before 1975) and a complete renovation of the old housing stock will be possible in 2050.

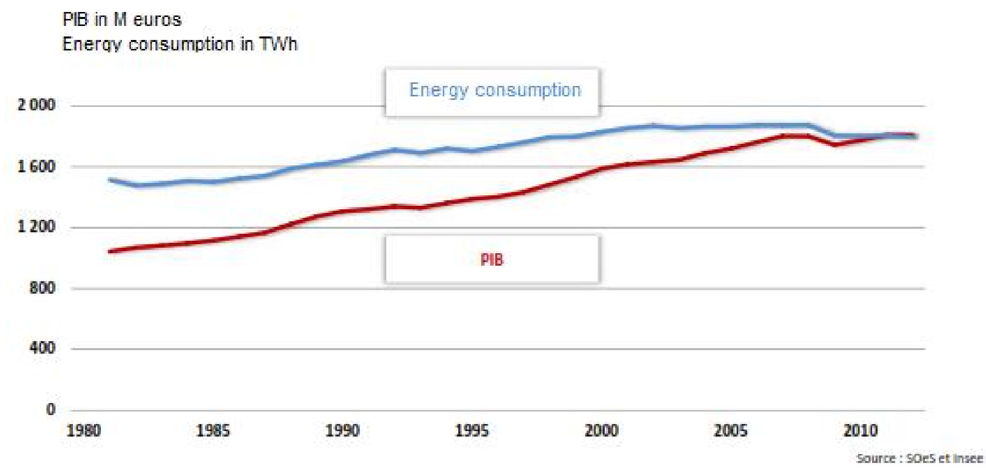

Figure 1 : Evolution of the French final energy consumption and GDP (PIB in French) [1].

In order to reduce energy consumption of HVAC systems, it is necessary to improve their *efficiency*[1]. Thermodynamics gives to engineers the necessary tools to analyse such systems. Thus, it is possible to make an *energy* and *exergy* analysis. The energy analysis is based on the sole First Law of Thermodynamics and is the most common approach to evaluate the performance of HVAC systems [2]. Nowadays, engineers and researchers also use the Second Law of Thermodynamics to make an analysis of efficiency. The *exergy analysis* identifies the locations and the causes of dissipations in a thermodynamic system. As a result, this approach helps to improve and optimize such systems [3].

Citing [3]: "Exergy analysis yields efficiencies that provide a true measure of how nearly actual performance approaches the ideal".

Studying the efficiency of HVAC systems and how to increase it is the purpose of different works. In 2002, Chengqin et al. [2] made a review of principles of exergy analysis in HVAC and proposed a different selection of dead-state[2]. In 2010, Sakulpipatsin et al. presented a method for exergy analysis of HVAC systems and buildings [4], Tolga et al. [5] compared four options of heating applications for a

---

[1] *By **efficiency**, we mean the ratio of the useful energy provided by a system, over the required energy it consumes.*

[2] *The concept of exergy, which is related to the « usefulness » of energy, is always defined relatively to a "dead state" or reference state, usually the one of the system's surroundings.*

building under both the energy and exergy approaches; finally, the same year, Luigi Marletta [6] made an exergy analysis in three plant schemes of air conditioning: all-air, dual-duct and fan-coil systems.

In 2013, Goncalves et al. [7] compared different heating systems for buildings under different outdoor conditions.

2. System under consideration

The system under consideration is the A660 Base Unit with option A661B Recirculation Duct Upgrade manufactured by P. A. Hilton Ltd. The system is composed by an axial flow fan discharging into a 250 mm square duct. Inside the duct, there are two electrical pre-heaters, a direct expansion cooling coil, a steam humidifier and two electrical re-heaters. Through a valve, the recirculation duct allows for variations in the proportions of fresh air. The air treatment station is showed in Figure 2.

The use of measuring instruments makes possible the statements of energy and mass balances across each *psychrometric*[3] *process*.

Figure 2 : Air treatment station A660 Base Unit with A661B Recirculation Duct Upgrade [8].

[3] *The word « **psychrometric** » is related to the physical properties of gas-vapor mixtures, usually of air and water vapor.*

3. Energy and Exergy

In order to solve energy problems, the simultaneous use of both the first and the second law of thermodynamics is required. The first law of thermodynamics allows for a balance of energy, based on the law of energy conservation, but it says nothing about the "quality" of this energy.

Here, the word "quality" is linked to the utility that can be given to a particular form and amount of energy. For example, the mechanical energy can be spontaneously and entirely converted for instance into thermal energy or into electrical energy. At the contrary, any amount of heat, whatever its specific temperature, can never be completely converted into mechanical or electrical energy. Its usefulness is thus lower.

Energy is by definition equal to the sum of *exergy* and *anergy*. Energy represents the sum of all forms of energy present in a process or cycle. In detail, exergy represents the useful part of the energy and anergy is the energy lost or destroyed exergy. This equivalence is shown in Figure 3.

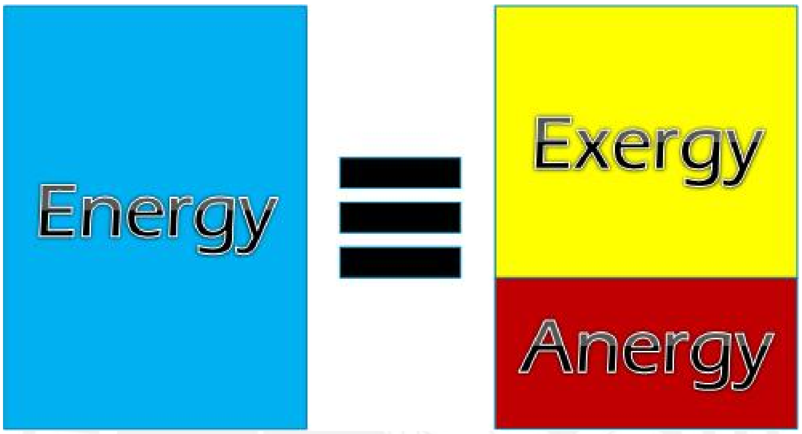

Figure 3 : Relationship between energy and exergy – anergy.

B. Formulas for Air Properties and Exergy Analysis of Psychrometric Processes

In this section, expressions used to calculate air properties, and formulas used for exergy analysis are going to be presented.

1. Air Properties

Required most air properties, so usually its "water content" and its specific energy, can be practically obtained from just two different temperature measurements: the dry-bulb and the wet-bulb one, and the value of the atmospheric pressure.

1.1. Saturation pressure

The following empirical expression was developed by Cadiergues for the calculation of the *saturation pressure*[4] at a given temperature, over liquid water[5]. In this case the concerned temperature is above 0°C.

$$p_{sat} = 10^{2.7877+\frac{7.625\,\theta}{241.00+\theta}} \tag{1}$$

With θ the relative temperature, expressed in °C.

1.2. Vapour pressure

According to the following expression (Eq. 2), the actual air *specific humidity*[6] ω can be measured by dry and wet bulb thermometers in supposing that the evaporation process of water from the wet bulb is entirely responsible for the drop of the wet temperature :

$$Cp_a \cdot (\theta - \theta_w) = L_v \cdot (\omega_{sat(T_w)} - \omega) \tag{2}$$

Cp_a is the specific heat at constant pressure of dry air, θ and θ the dry and wet temperatures, respectively, L_v the water latent heat of vaporisation and $\omega_{sat(T_w)}$ the specific humidity in the saturated stated. Humidity ratio cleared and solved to obtain:

$$\omega = \omega_{sat(T_w)} - Cp_a \cdot \frac{T - T_w}{L_v} \tag{3}$$

$$\alpha \cdot \frac{p_v}{p - p_v} = \alpha \cdot \frac{p_{sat}(\theta_w)}{p - p_{sat}(\theta_w)} - Cp_a \cdot \frac{T - T_w}{L_v} \tag{4}$$

The atmospheric pressure p is much higher than the vapour partial pressure $p_v \ll p$, and the saturation pressure is much lower than atmospheric pressure: $p_{sat} \ll p$. Thus, it is possible to make the following approximations (Eq. 5 and 6):

$$p_v \ll p \Rightarrow p - p_v \approx p \tag{5}$$

---

[4] *The saturation pressure is the maximum pressure that a given gas can achieve before being condensed as liquid. Its value depends only on temperature. Beyond this saturation pressure, liquid and vapor phases of the same fluid coexist.*

[5] *Such expression is indeed slightly modified when the water vapor coexists with its ice.*

[6] *The specific humidity ω is defined as the ratio of mass of water vapor content in air, over the sole dry air :* $\omega = \frac{m_{water}}{m_{dry\ air}} \ll 1.$

$$p_{sat} \ll p \Rightarrow p - p_{sat} \approx p \qquad (6)$$

Given Eq. 5 and Eq. 6, an expression calculating vapour pressure is obtained (Eq.7):

$$p_v = p_{sat}(T_w) - p \cdot \frac{c_{p_a}}{\alpha} \cdot \frac{(T - T_w)}{L_v} \qquad (7)$$

By replacing the various constants, an equation indicating the approximate value of vapour pressure is obtained (Eq.8):

$$p_v \approx p_{sat}(T_w) - 0.000666 \; p_{atm}(T - T_w) \qquad (8)$$

1.3. Relative humidity

Once the partial pressure of water is known, it is easy to compute the other specific properties of moist air, as the relative humidity for instance:

$$\phi = \frac{p_v}{p_{sat}} \qquad (9)$$

1.4. Humidity ratio

The humidity ratio/specific humidity is done by:

$$\omega = 0.622 \times \frac{p_v}{p_{atm} - p_v} \qquad (10)$$

1.5. Specific volume

The specific volume, corresponding to the inverse of the air density, is obtained by:

$$v = \frac{r_a \times (0.622 + \omega) \times T}{(1 + \omega) \times p_{atm}} \qquad (11)$$

1.6. Specific enthalpy

And the specific enthalpy, corresponding the specific energy of moist air:

$$h = c_{pa} \cdot \theta + \omega \cdot (L_v + c_{pv} \cdot \theta) \qquad (12)$$

2. Formulas for Exergy Analysis

In order to perform the exergy analysis, the exergy balance is necessary. The two first equations show the exergy balance (Eq. 13 and 14) of a stationary system. The general expression to obtain the specific exergy (ex) is presented in the third equation (Eq. 15).

$$\sum_{in} \dot{E}x_{\dot{Q}} + \sum_{in} \dot{m}(ex) - \sum_{out} \dot{E}x_{\dot{Q}} - \sum_{out} \dot{m}(ex) - \dot{E}x_{dest} = 0 \qquad (13)$$

$$\sum_{in} \dot{Q}\left(1 - \frac{T_0}{T}\right) + \sum_{in} \dot{m}\,(ex) - \sum_{out} \dot{Q}\left(1 - \frac{T_0}{T}\right) - \sum_{out} \dot{m}\,(ex) - \dot{E}x_{dest} = 0 \qquad (14)$$

where:

$$ex = h - h_0 - T_0 (s - s_0) \tag{15}$$

Wepfer et al. [9] proposed an expression to calculate the specific total exergy considering air (mixture of dry air and water vapour) as an ideal gas. The first term is the thermal exergy, the middle one corresponds to the mechanical exergy and the last term is the chemical exergy.

$$ex = (Cp_a + \omega\, Cp_v)\, T_0 \left(\frac{T}{T_0} - 1 - \ln\frac{T}{T_0}\right) + (1 + \tilde{\omega})\, r_a\, T_0 \ln\frac{p}{p_0} + r_a\, T_0 \left[(1 + \tilde{\omega}) \ln\frac{1 + \tilde{\omega}_0}{1 + \tilde{\omega}} + \tilde{\omega} \ln\frac{\tilde{\omega}}{\tilde{\omega}_0}\right] \tag{16}$$

On the previous expressions, the proportionality between specific humidity ratio ω and specific humidity ratio on a molal basis $\tilde{\omega}$ is given by:

$$\tilde{\omega} = 1.608\, \omega \tag{17}$$

C. Energy and Exergy Analysis

In the previous section, the equations used to calculate exergy and perform exergy analysis have been presented. The energy balance equations used are the known expressions of thermodynamics. In this section the results from energy and exergy analysis will be presented.

1. Definition of efficiency

An important step in energy and exergy analysis is the evaluation of efficiency. This study uses the ratio between the obtained output and the expended input as the efficiency definition. Efficiency definition is applied for energy and exergy concepts, Equation 18 shows this definition:

$$Efficiency = \frac{Obtained\ output}{Expended\ input} \tag{18}$$

2. Case study

In this analysis, an HVAC system with air recirculation is being studied. A variable speed drive allows for the regulation of different flow rates (3 in the study). Recirculated airflow is regulated with a valve positioned after the air outlet (5 valve position).

The conceptual lay-out is given in Figure 4, examples of experimental data and thermal cycles for specific fan speeds and recirculation rates are collected in Table 1 and presented in Figure 5 respectively.

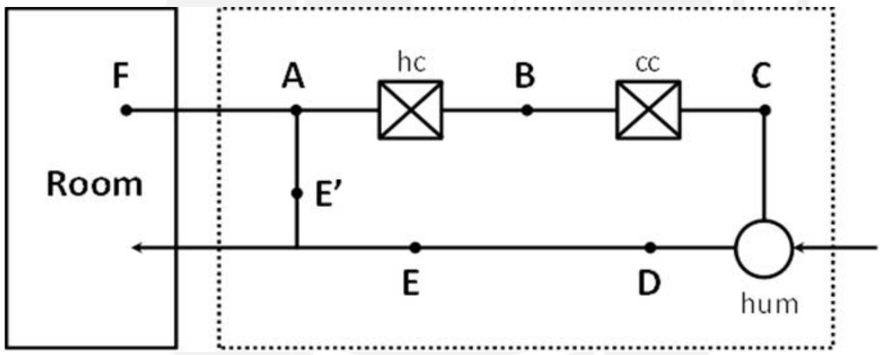

Figure 4 : HVAC Conceptual lay-out.

Table 1 : Experimental data

	t [°C]	w [g/kg]
A	20.4	10.2
B	30.5	10.8
C	17.1	10.3
D	18.1	13.0
E	18.1	11.4
F	23.4	8.6

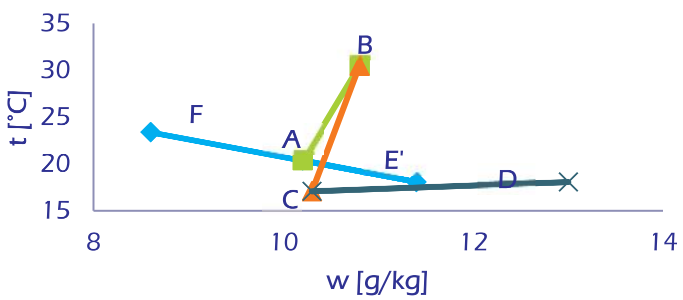

Figure 5 : Thermal cycle of the HVAC system

The experimental HVAC system allows for the investigation of basic air conditioning processes. In this study mixing of air streams (C.3), simple heating (C.4), cooling with dehumidification (C.5) and heating with humidification (C.6) have been evaluated taking energy and exergy efficiencies into consideration.

The atmospheric temperature is considered to be the temperature of the dead state. In the HVAC system the ambient temperature corresponds to the temperature measured at the air inlet to the lower recirculation rate.

The complete HVAC system is showed in Figure 6.

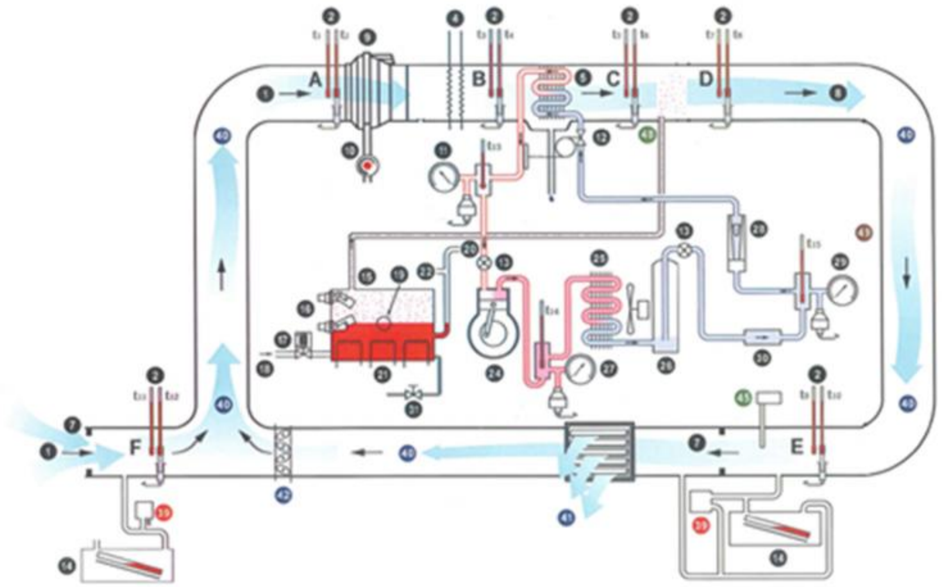

Figure 6 : HVAC system studied

3. Mixing of air streams

Mixing of air streams is a very common process, many air-conditioning applications require the mixing of two airstreams. This process is usually assumed to be adiabatic because the heat transfer with the surroundings is small.

In the case of study, no work is involved in the process. Furthermore, kinetic and potential energy are negligible. In order to perform the energy and exergy analysis, it is important to define the system boundary, and the mass and energy flows crossing the border, Figure 7 shows the system studied.

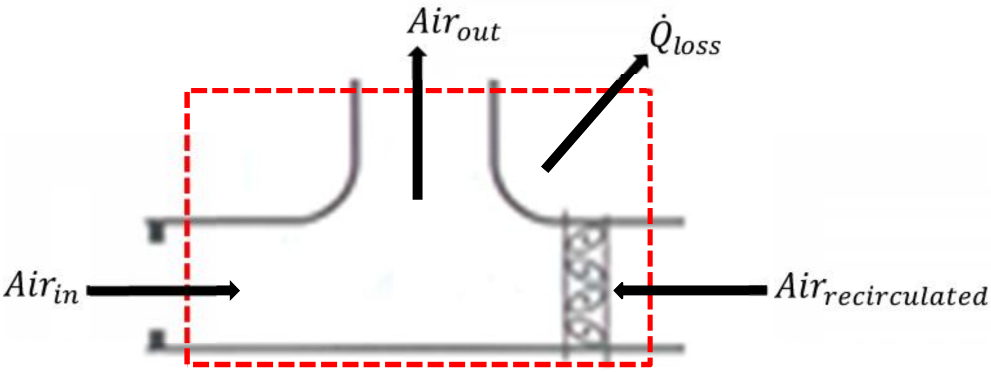

Figure 7 : Mixing system, boundary and flows across the border

The energy balance for the mixing of two airstreams reduces to the following:

$$\dot{Q}_{loss} = \dot{m}_F h_F + \dot{m}_{E'} h_{E'} - \dot{m}_A h_A \quad (19)$$

The energy efficiency is expressed as following:

$$\eta = \frac{\dot{m}_A h_A}{\dot{m}_F h_F + \dot{m}_{E'} h_{E'}} \quad (20)$$

Energy efficiency results obtained are around 100% (between 99.22% and 101.78%). As expected heat losses are small, for this reason this process is usually considered adiabatic. Results of energy efficiency as function of flow rate for different recirculation rates are shown in Figure 8.

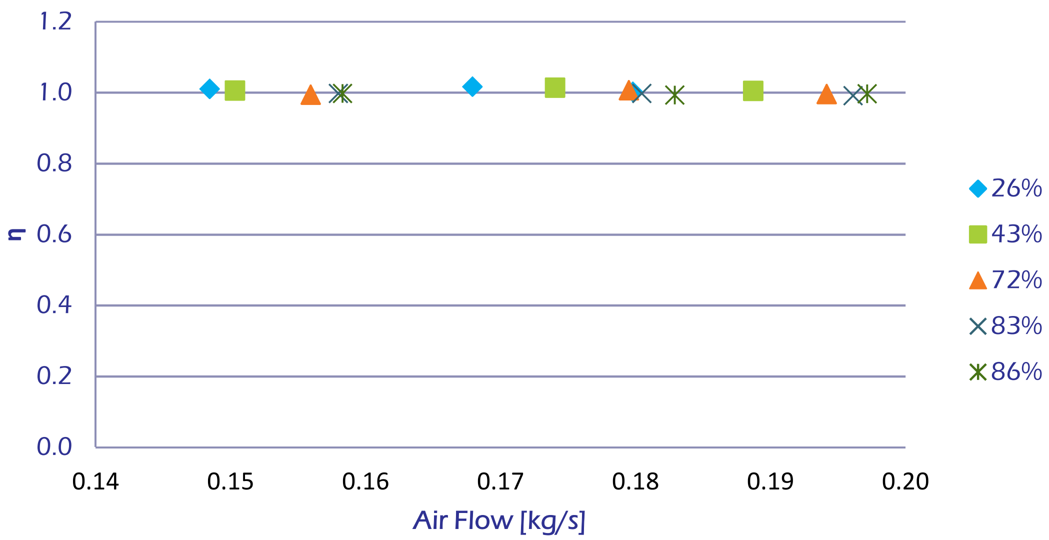

Figure 8 : Energy efficiencies for adiabatic mixing of air streams at different recirculation rates

Complete exergy balance has to take into consideration heat loss funded in energy balance. Heat loss is modified by a Carnot factor, $1 - T_0/T$, where T_0 represent temperature of dead state and T the temperature of the air flow. For this study, T_0 was considered as the ambient temperature measured at the air inlet (point F). In a psychrometric process is an option to consider a logarithmic mean temperature (T_{ml}) between inlet and outlet process in order to represent the temperature of air flow. The mixing process brings the problem of identifying which temperatures used to calculate T_{ml}, it has two inlet temperatures.

Energy efficiency results show that is reasonable to consider mixing process as an adiabatic process, for this reason and given the problem to identify which inlet temperature to use, heat loss multiplied by Carnot factor is considered negligible.

The exergy balance for the mixing of two airstreams reduces to the following:

$$\dot{Ex}_d = \dot{m}_F ex_F + \dot{m}_{E'} ex_{E'} - \dot{m}_A ex_A - \dot{Q}_{loss}\left(1 - \frac{T_0}{T}\right)$$

$$\dot{Q}_{loss}\left(1 - \frac{T_0}{T}\right) \approx 0$$

The exergy efficiency is expressed as following:

$$\psi = \frac{\dot{m}_A ex_A}{\dot{m}_F ex_F + \dot{m}_{E'} ex_{E'}}$$

Exergy efficiency as function of air flow results obtained shows a constant behaviour for each air recirculation rate. The most important efficiency is obtained is for an air recirculate rate around 70%. Results are show in Figure 9.

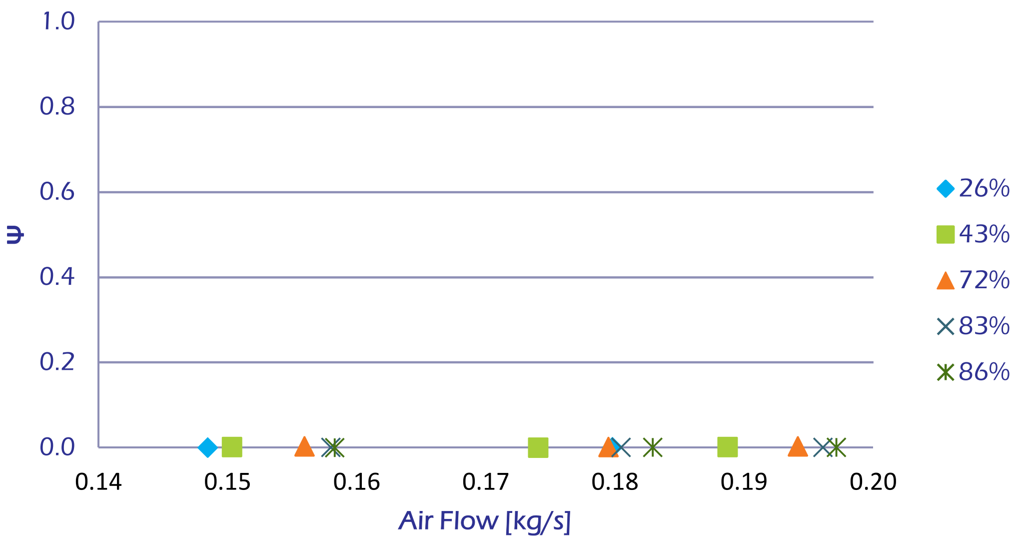

Figure 9 : Exergy efficiencies for adiabatic mixing of air streams for different recirculation rates

If exergy efficiency results are presented as function of air recirculation rates for each fan tension (fan speed-air flow), it is interesting to note the same behaviour for all different air flows. Experimental points show a growing behaviour up to around 50% at around 70% of air recirculation rate, and then exergy efficiency decreases. This behaviour is show in Figure 10.

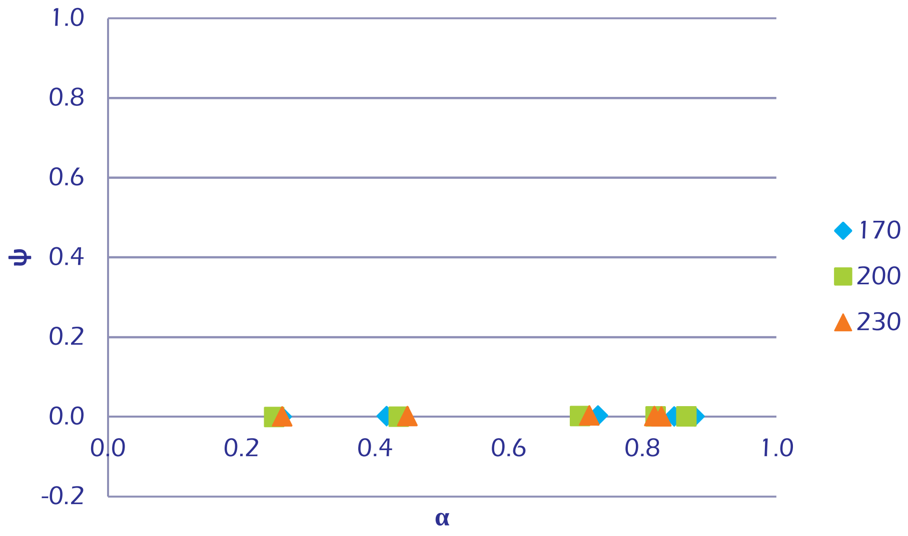

Figure 10 : Exergy efficiencies for adiabatic mixing of air streams at different fan speed

4. Simple heating process

In simple heating process, the air circulating through a duct is heated. No moisture is added, resulting in a constant specific humidity. In psychometric chart, simple heating is represented by a horizontal line in the direction of increasing dry-bulb temperature.

Relative humidity is modified during simple heating process and it could reach bellow-comfortable levels causing dry skin, respiratory difficulties and an increase in static electricity.

In this study, electric power at fan is considered as part of input power. The thermometer position is before the fan, as a result, measured property include fan and electric heater effect, Figure 11 shows the scheme of process.

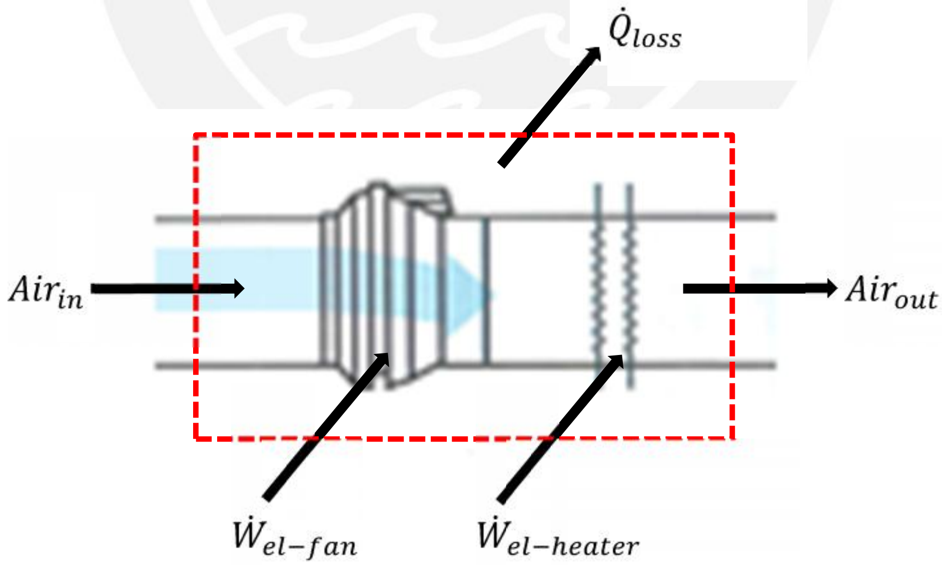

Figure 11 : Simple heating process

The energy balance for the simple heating process includes two sources of heat/power. It reduces to the following:

$$\dot{Q}_{loss} = \dot{m}_A h_A - \dot{m}_B h_B + \dot{W}_{el-fan} + \dot{W}_{el-heater} \quad (21)$$

The energy efficiency is expressed as following:

$$\eta = \frac{\dot{m}_B h_B - \dot{m}_A h_A}{\dot{W}_{el-fan} + \dot{W}_{el-heater}} \quad (22)$$

Efficiencies results are not according with expectations. Values greater than 100% efficiency are obtained (between 125% and 140%). Even if surrounding temperature is higher than temperature at input air stream, the difference would not be that important as to explain these results.

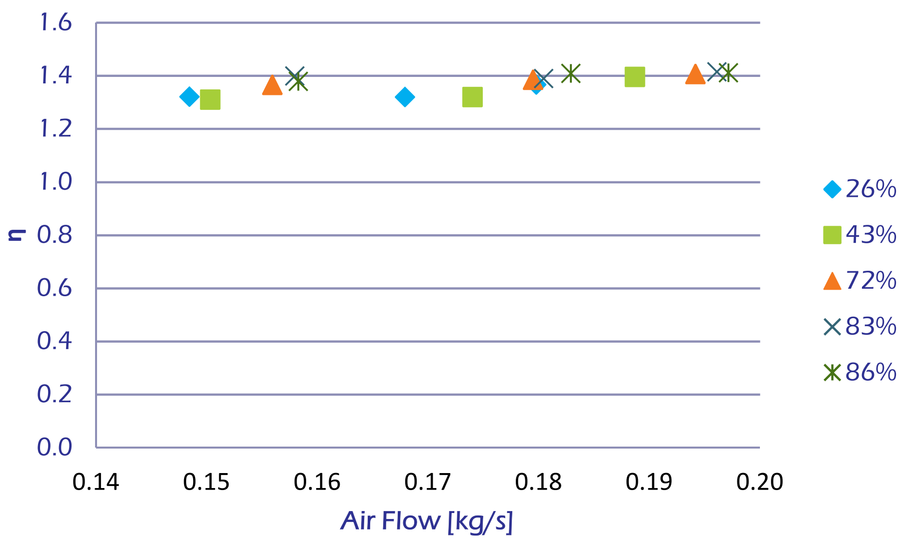

Figure 12 : Energy efficiencies for simple heating at different recirculation rates

In exergy analysis electrical work is equal to the work W. Carnot factor no modify value of work as different of heat power. The exergy balance for the simple heating process reduces to the following:

$$\dot{E}x_d = \dot{Q}_{in}\left(1 - \frac{T_0}{T}\right) + \dot{m}_A ex_A - \dot{m}_B ex_B \quad (23)$$

$$\dot{E}x_d = \dot{W}_{el-fan} + \dot{W}_{el-heater} + \dot{m}_A ex_A - \dot{m}_B ex_B \quad (24)$$

The exergy efficiency is expressed as following:

$$\psi = \frac{\dot{m}_B ex_B - \dot{m}_A ex_A}{\dot{W}_{el-fan} + \dot{W}_{el-heater}} \quad (25)$$

Exergy efficiencies values obtained are very low (less than 3%). Unfortunately, results are not reliable as a result of energy efficiencies results.

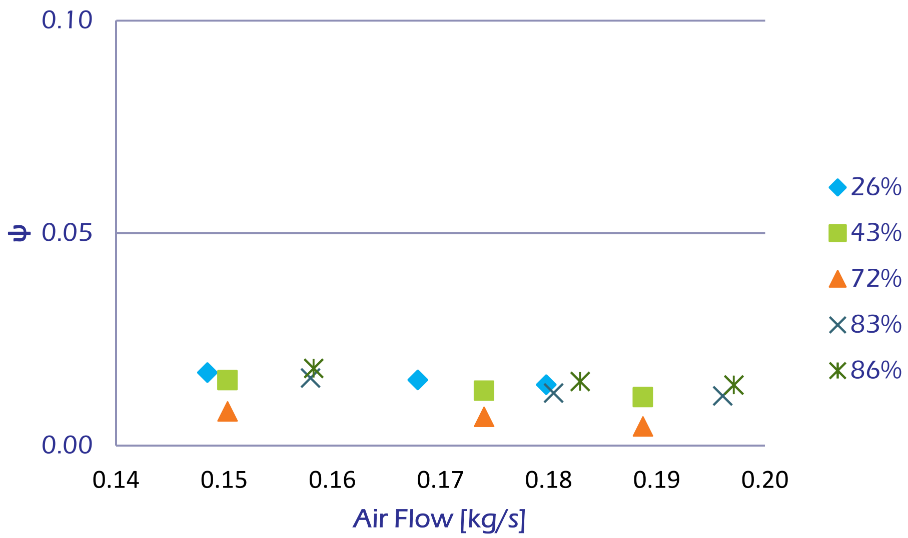

Figure 13 : Exergy efficiencies for simple heating at different recirculation rates

If exergy efficiency results are presented as function of air recirculation rates for each fan tension (fan speed-air flow), it is interesting to note the same behaviour for all different air flows. Experimental points show a decreasing behaviour between around 50% at around 70% of air recirculation rate. Outside this range, exergy efficiency is higher. This behaviour is opposite to the behaviour found in the mixing process. Behaviour founded in simple heating process is show in Figure 14.

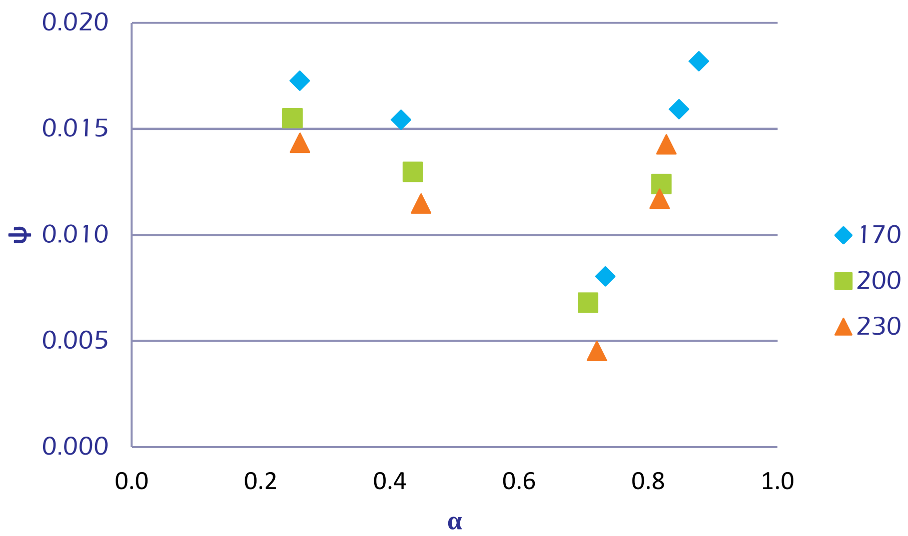

Figure 14 : Exergy efficiencies for simple heating at different fan speed

5. Cooling with dehumidification

In cooling with dehumidification process, the beginning dry-bulb temperature decrease (horizontal line, similar as simple cooling). If relative humidity reaches 100% some moisture is removed from the air. This process required an under dew-point temperature.

Cooling process is performed by an evaporator of a refrigeration system. Energy and exergy analysis include duct of air and evaporator as show Figure 15.

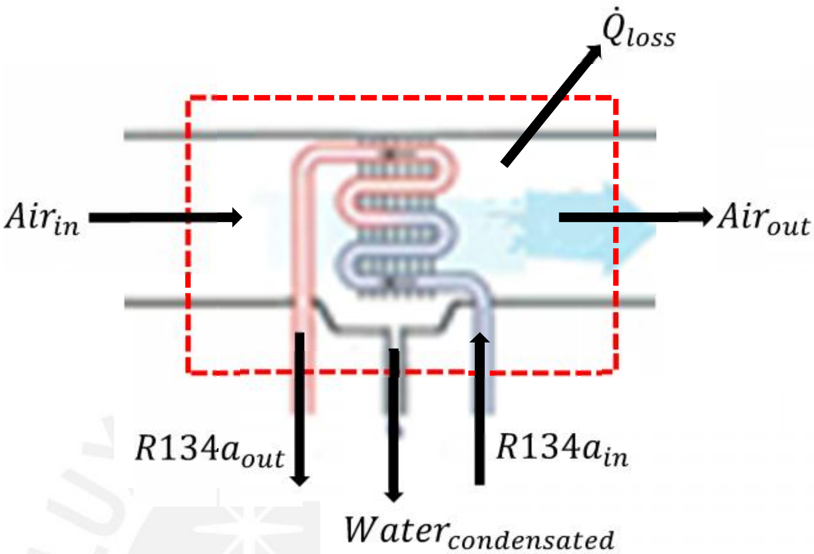

Figure 15 : Cooling with dehumidification process

The energy balance for cooling with dehumidification process reduces to the following:

$$\dot{Q}_{loss} = \dot{m}_B h_B - \dot{m}_C h_C + \dot{m}_{15'} h_{15'} - \dot{m}_{13} h_{13} - \dot{m}_W h_W \quad (26)$$

The energy efficiency is expressed as following:

$$\eta = \frac{\dot{m}_B h_B - \dot{m}_C h_C}{\dot{m}_W (h_{13} - h_{15'})} \quad (27)$$

Energy efficiencies are higher than 100%. To explain these results it is important to analyse the ambient temperature. In this case air stream inlet temperature is higher than ambient temperature; and at the other end, air stream outlet temperature is less than ambient temperature. Heat transfer is receive by the HVAC system in the first case and rejected by the HVAC system to the ambient in the second case.

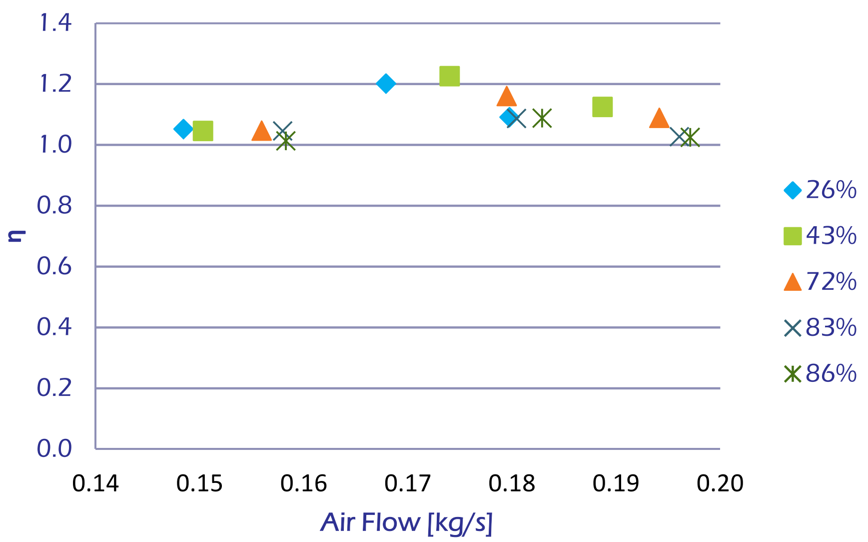

Figure 16 : Energy efficiencies for cooling with dehumidification at different recirculation rates

The energy balance for cooling with dehumidification process reduces to the following:

$$\dot{E}x_d = \dot{m}_B h_B - \dot{m}_C h_C + \dot{m}_{15'} h_{15'} - \dot{m}_{13} h_{13} - \dot{m}_W h_W - \dot{Q}_{loss}\left(1 - \frac{T_0}{T}\right) \quad (28)$$

The exergy efficiency is expressed as following:

$$\psi = \frac{\dot{m}_B h_B - \dot{m}_C h_C}{\dot{m}_W (ex_{13} - ex_{15'})} \quad (29)$$

The HVAC system used in this study has a refrigeration system; for this reason it is necessary to calculate R-134a properties. In refrigeration cycles states at high pressure are completely defined; at low pressure state obtained after compressor is also completely defined but conditions' state after expansion are in saturated liquid-vapour region, so it is necessary identify the quality X of fluid. Expression 30 shows how to obtain quality X.

$$X = \frac{h - h_f}{h_g - h_f} \quad (30)$$

Value of quality X allows to calculated entropy of the fluid. Entropy formula is shown in expression 31.

$$s_{15'} = s_f + X \cdot (s_g - s_f) \quad (31)$$

Exergy efficiencies obtained are under 13%. For the 3 larger recirculation ratios, a downward trend is observed when the air flow increases. Results are presented in Figure 17.

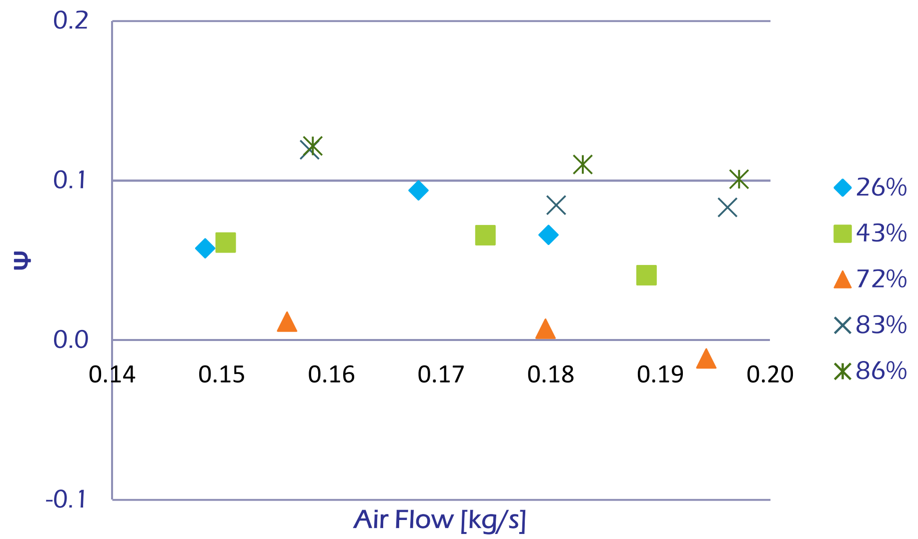

Figure 17 : Exergy efficiencies for cooling with dehumidification at different recirculation rates

If exergy efficiency results are presented as function of air recirculation rates for each fan tension (fan speed-air flow), it is interesting to note the same behaviour for all different air flows. Experimental points show a decreasing behaviour between around 50% at around 70% of air recirculation rate, outside this range exergy efficiency in higher. This behaviour is similar as results obtained at simple heater process and is opposite to the behaviour found in the mixing process. Behaviour founded in simple heating process is show in Figure 18.

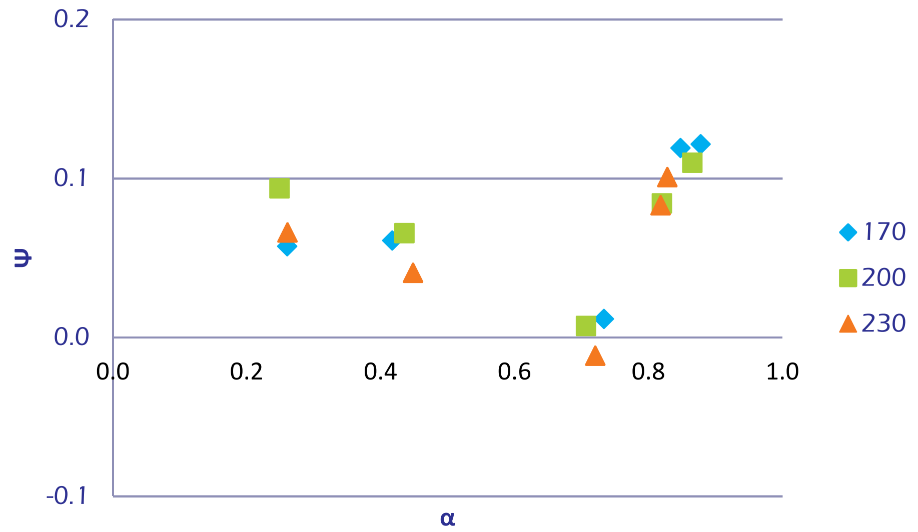

Figure 18 : Exergy efficiencies for cooling with dehumidification at different fan speed

6. Humidification process

Simple heating process may cause problems of low relative humidity. A way to solve this problem is the humidification process. It is possible to introduce a steam or spray of water. First solution will result in humidification with additional heating, second solution result in a cooling of the air stream.

HVAC system analysed use to a flow of steam in order to humidify the air stream.

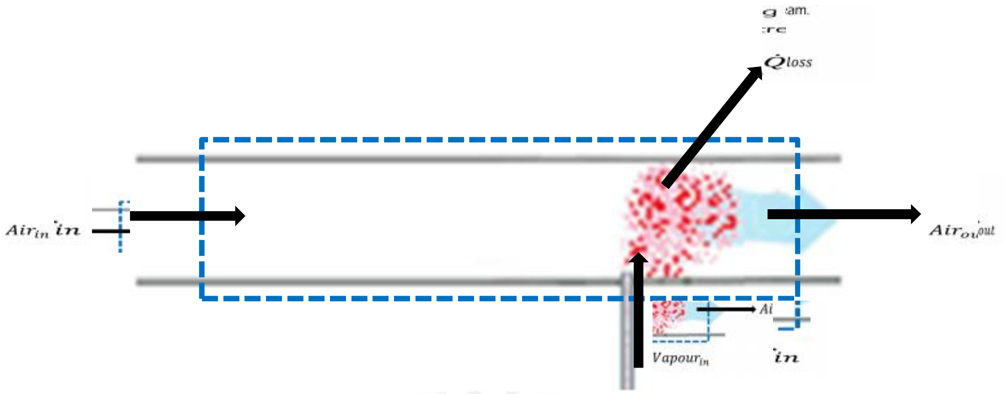

Figure 19 : Complete system of humidification

The energy balance for humidification process reduces to the following :

$$\dot{Q}_{loss} = \dot{m}_C h_C - \dot{m}_D h_D + \dot{m}_W h_W \tag{32}$$

The energy efficiency is expressed as following:

$$\eta = \frac{\dot{m}_D h_D - \dot{m}_C h_C}{\dot{m}_W h_W} \tag{33}$$

Energy efficiencies are higher than 100%. To explain these results it is important to analyse the ambient temperature. Air stream inlet and outlet temperature are lower than ambient temperature, this difference between systems temperature and ambient temperature may enable a heat transfer from the surrounding to the HVAC system, as a result energy efficiencies greater than 100% are obtained.

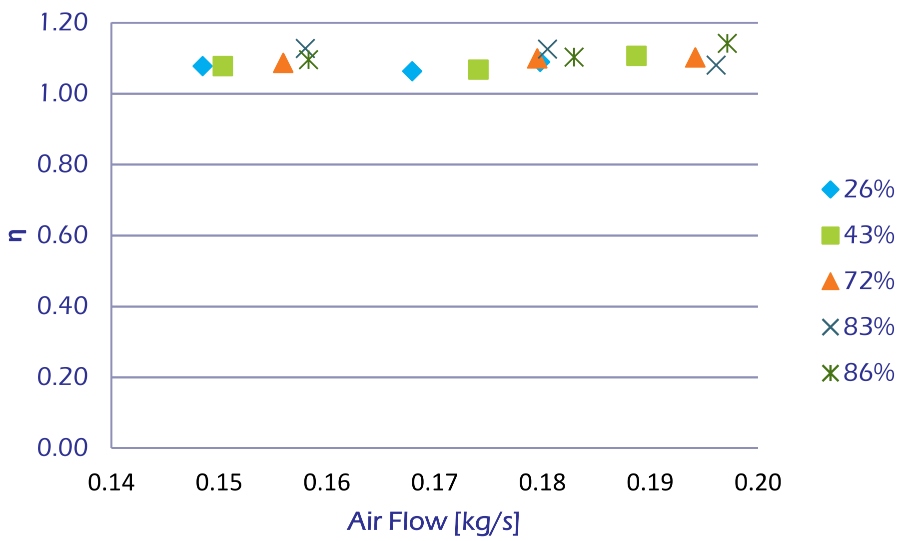

Figure 20 : Energy efficiencies for humidification at different recirculation

The exergy balance for humidification process reduces to the following:

$$\dot{E}x_d = \dot{m}_C ex_C - \dot{m}_D ex_D + \dot{m}_W ex_W + \dot{Q}_{loss}\left(1 - \frac{T_0}{T}\right) \tag{34}$$

The exergy efficiency is expressed as following:

$$\psi = \frac{\dot{m}_D ex_D - \dot{m}_C ex_C}{\dot{m}_W h_W} \tag{35}$$

The humidification process in the HVAC system uses a steam. Water flow rates interact with the air stream. Kinetic and potential term are considered negligible. The exergy of a steam (water flow rate) can be assessed as:

$$ex_W = c_{pW}(T - T_0) - T_0 c_{pW} \ln\frac{T}{T_0} - R_W T_0 \ln \emptyset_0 \tag{36}$$

Exergy efficiencies values obtained are very low (less than 6%).

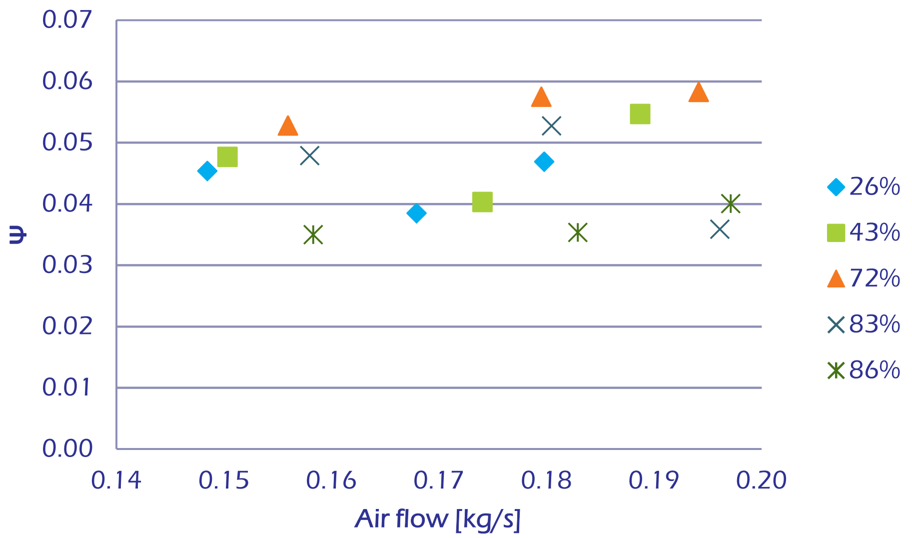

Figure 21 : Exergy efficiencies for humidification at different recirculation rates

If exergy efficiency results are presented as function of air recirculation rates for each fan tension (fan speed-air flow), it is interesting to note the same behaviour for all different air flows. This behaviour is similar to mixing stream process behaviour. Values obtained are show in Figure 22.

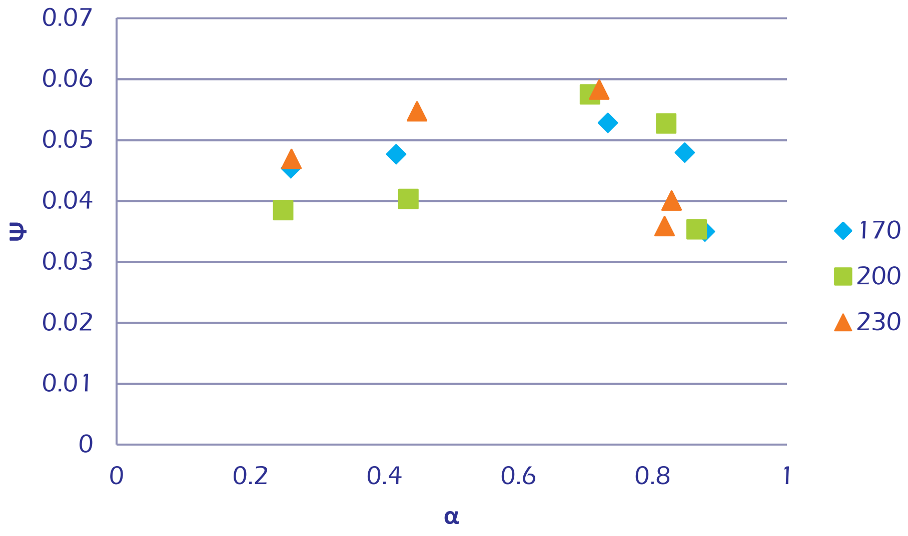

Figure 22 : Exergy efficiencies for humidification at different fan speed

7. Example of operating condition

In this part, an example of measured values and results at a particular operating condition are shown in Table 2 and Table 3 respectively.

Table 2 : Measured values at particular operating condition

	Units	A	B	C	D	E	F
Ambient temperature	[°C]	23.6					
Recirculation rate	[%]	73.3					
Flow	[kg/s]	0.156					0.042
Dry-bulb temperature	[°C]	20.4	30.5	17.1	18.1	18.1	23.4
Wet-bulb temperature	[°C]	16.4	20.2	15.3	17.8	16.6	16.1

Table 3 : Results for particular operating condition

Process (Reference Figure 4)		Energy Efficiency	Exergy Efficiency	Ex_d [kW]	Irreversibility
Mixing air streams	FE → A	99.48%	43.91%	0.003	0.18%
Simple heating	A → B	136.63%	0.80%	1.34	82.41%
Cooling with dehumidification	B → C	104.71%	1.16%	0.225	13.84%
Humidification	C → D	108.63%	5.27%	0.058	3.57%
HVAC System	-	-	-	1.626	100%

D. Conclusions

The exergy analysis of the air conditioning unit reveals the performance of different psychometrics processes. Simple heating and cooling with dehumidification processes have the worst exergy performance with values around 1% and both processes represent around 96% of irreversibilities in the HVAC system.

In detail, simple heating process represent ≈ 82% and cooling with dehumidification process represent ≈14% of irreversibilities on the system.

The second law analysis indicates processes with higher room for improvement in the HVA system: simple heating process and cooling with dehumidification. In order to improve exergy performance of an HVAC system, it should be interesting to investigate these processes and their possible alternatives.

E. References

[1] U. F. d. l'Électricité, «UFE,» 18 June 2014. [Online]. Available: http://ufe-electricite.fr/IMG/pdf/5.pdf [08/31/2016].

[2] R. Chengqin, L. Nianping and T. Guangfa, "Principles of exergy analysis in HVAC and evaluation of evaporative cooling schemes," *Building and Environment,* no. 37, pp. 1045-1055, 2002.

[3] M. Kanoglu, Y. Cengel and I. Dincer, Efficiency Evaluation of Energy Eystems, Springer, 2012.

[4] P. Sakulpipatsin, L. Itard, H. van der Kooi, E. Boelman and P. Luscuere, "An exergy applicatoin for analysis of buildingsand HVAC systems," *Energy and Buildings,* no. 42, pp. 90-99, 2010.

[5] M. Tolga Balta, I. Dincer and A. Hepbasil, "Performance ans sustainability assessment of energy options for building HVAC applications," *Energy and Buildings,* no. 42, pp. 1320-1328, 2010.

[6] L. Marletta, "Air Conditioning Systems from 2nd Law Perspective," *Entropy,* no. 12, pp. 859-877, 2010.

[7] P. Goncalves, A. Rodrigues Gaspar and M. Gameiro da Silva, "Comparative energy and exergy performance of heating options in buildings under different climatic conditions," *Energy and Buildings,* no. 61, pp. 288-297, 2013.

[8] P.A.Hilton. [Online]. Available: http://www.p-a-hilton.co.uk/pdfs/lab-pdf_a660v6_061115113834.pdf [08/31/2016].

[9] W. Wepfer, R. Gaggioli and E. Obert, "Proper evaluation of available energy for HVAC," *ASHRAE Transactions,* vol. 85, no. 1, pp. 214-230, 1979.