Exergy optimization of HVAC systems with air recirculation

Thesis submitted for the degree of
Master in Energy

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Abstract

In the next pages, an energetic and exergetic analysis of one specific Heat, Ventilation and Air Conditioning (HVAC) system, using air recirculation, will be detailed. Different operating conditions will be presented and the ones of least exergy destruction will be highlighted and analyzed.

Both the first and second law of thermodynamics are used to understand and describe these phenomena through a specific performance criterion called **Duty Specific Exergy Consumption**[1], noted **DSExC** in the rest.

The main results presented in this document have been published in a peer-reviewed French journal dedicated to HVAC systems, see (Chacon Chauca, Quintanilla Munoz, & Vaudrey, 2019).

[1] *Performance criterion hitherto called in French « Consommation Opératoire Unitaire d'Exergie » or COUEx, see in (Rivero, de Oliveira Jr., & Le Goff, 1990).*

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Author,

Justo CHACON

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Nomenclature

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Cp_a : Specific heat of air

Cp_v : Specific heat of vapor

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

r_a : Gas constant for the dry air

r_v : Gas constant for the water vapor

T_0 : Ambient temperature

Greek letters

α *: Recirculation rate*

η_{AHU} *: Energy efficiency of the AHU*

θ *: Temperature in degrees Celsius*

φ *: Relative humidity*

ω *: Specific humidity*

Acronyms

AHU : Air Handling Units

ASHRAE: American Society of Heating, Refrigerating and Air-Conditioning Engineers

HRV : Heat Recovery Ventilation

HVAC : Heat, Ventilation and Air Conditioning

A. Context and goals

1. Worldwide importance

The efficient use of our energy sources is one of the major worldwide issues nowadays. The main problem with the word *efficient* is its actual meaning: people too often do not understand it entirely. Efficiency does not have an actual physical meaning; it rather has a mathematical representation. It has to be calculated, and that makes it more difficult to understand. In other words, sometimes we cannot be aware of the inefficient use of the energy and the huge amount of sources/money/time we are wasting/ spending.

Environmental and economic purposes push various investigations to focus on the performances of energy systems and equipment. In the context of the incoming energy transition, Heat, Ventilation and Air Conditioning (HVAC) systems will certainly take an increasing and worldwide importance. The final goal of our study is to help to reduce the energy consumption in being "efficient" with our HVAC systems.

Thermodynamic gives engineers the needed tools to analyze such kind of systems. The energy analysis is based on the first law of thermodynamics and the performance of the system. Exergy is based on the "quality" of that energy in one system. Both can be useful to achieve such an analysis of efficiency.

2. Future and prospects

Thanks to the cuts in energy consumptions they can imply, HVAC systems will certainly play an increasing role in future buildings. One of the strategies available to decrease the energy consumption of such a system is the recirculation of air. A limited fraction of the return air, coming from the inside of the building, is diverted from the exhaust stream and mixed with the incoming outside air one, as represented in Figure 1. This gaseous mix, whom properties – namely temperature and humidity – are closer to the ones required inside the volume to ventilate, is then treated and injected into the latter with a lower energy consumption.

As well as the air recirculation system, another strategy available to increase the "wise use" of energy is to use heat recovery ventilation (HRV). With such an equipment, the exhaust air stream — so the return air stream whom the recirculated air has been removed — is used as a hot flow of a cross flow or counter-flow heat exchanger whom cold flow is the outside air aspirated. This heat exchange increases or decreases the outside air flow temperature — whether the whole system is operating in winter or summer for instance — and makes its temperature closer to the building ones. It helps drastically to decrease the energy consumption of the system.

3. HVAC Systems

Air handling systems[2] or Air Handling Units (AHU), commonly known as HVAC systems, are gradually replacing more traditional heating systems. They are used as equipment for different kind of building: industrial, tertiary and/or detached houses. They allow, in any season, to maintain at a set-point not only the temperature, but also the humidity inside the building or the room. The goal of such a system is: the comfort of the occupants or the maintenance in optimal conditions of various goods, which may require well-defined temperature and humidity levels.

4. Goals of the project

In figure 1, we can see a representation of the kind of system HVAC we are interested in. As we explained above, an air recirculation system is used to improve the energy efficiency of the building we are representing.

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[2] *Or CTA, for* Centrales de Traitement de l'Air *in French.*

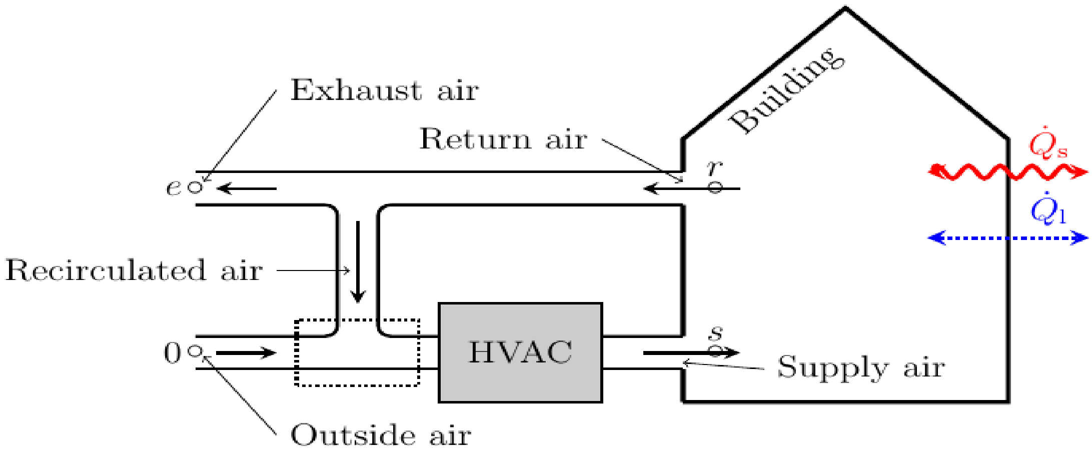

Figure 1: Simplified representation of a building ventilated by an HVAC system equipped with an air recirculation system. A fraction of the return air stream is diverted from the exhaust circuit and mixed with the fresh air aspirated from the outside before its treatment through the HVAC system.

The main purpose of this work is to analyze the influence of the outside air fraction — so the fraction of the aspirated outside air which is injected into the building at the supply point (see figure 1) — on the amount of exergy destroyed inside the whole AHU system, so the HVAC itself and the recirculated air system. The main goal of this analysis is to find, if it exists, the best recirculated air fraction, which produces the lowest total destruction of exergy. The lower the destruction of exergy, the better the performances of the system we have, see (Bejan, 2006) or (Dincer & Rosen, 2013).

In the model here considered, there are two main sources of exergy destruction:

1. - The **air recirculation system**, involving a mixing process of two different air streams at two different levels of temperature and humidity.

2. - The **HVAC system** itself, hat houses various exergy destruction processes, through heat exchange, head losses or humidification processes for instance.

In the incoming chapters, several kinds of balance will be established (mass balance, energy balance and exergy balance) to, finally, estimate the total exergy destruction of our model. Afterward, our experimental apparatus (air handling system) will be presented, as well as all their measurement processes. And, finally, with all the calculation and information obtained in the previous chapters, we are going to display our results and conclusions.

5. Previous exergy analyses of HVAC Systems

Considering the known history of the concept of exergy and of its wide use in engineering (Sciubba & Wall, 2007), its application to HVAC systems is quite recent, with a first dedicated paper in 1979 (Wepfer, Gaggioli, & Obert, 1979). This paper has established an experimental exergy balance; they demonstrated that the exergy flow can move in the inverse direction of the mass flow if the energy has to be supplied or balanced out. They described such a behavior with the following phrase "Exergy is potential to cause change".

In 2002, (Chengqin, Nianping, & Guangfa, 2002) made a review of principles of exergy analysis in HVAC and proposed a different selection of dead-states[3]. This review highlighted the importance of a proper classification of systems and energy flows; as well as the selection of dead-state for simplifying the exergy analysis. They concluded that "Evaporative cooling is accomplished by sacrificing the chemical exergy of the primary and/or secondary air in order to get the thermal exergy increase of the primary air."

[3] *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.*

In 2010, (Sakulpipatsin, Itard, van der Kooi, Boelman, & Luscuere, 2010) presented a method for exergy analysis of HVAC systems and buildings which "intended to enable building designers to compare between the impact of improvements in the building envelope and in HVAC systems". Later, (Tolga Balta, Dincer, & Hepbasil, 2010) compared four options of heating applications (heat pump, condensing boiler, conventional boiler and solar collector) for a building under both the energy and exergy approaches, and they concluded that the solar collector has the highest total exergy efficiency. Finally, the same year, (Marletta, 2010) made an exergy analysis in three plant schemes of air conditioning: all-air, dual-duct and fan-coil systems; this analysis revealed the poor performance of the most common air-conditioning system plant schemes and proposed the following direction: "Chemical dehumidification combined with solar energy into the solar cooling technology seems to be very interesting".

In 2013, (Goncalves, Rodrigues Gaspar, & Gameiro da Silva, 2013) compared different heating systems for buildings under different outdoor conditions; their work has revealed that "the exergy is an useful method for comparison of thermal-based options, with similar primary energy performances, located at different outdoor environmental conditions".

B. Exergy analysis of HVAC systems

Try to have one accurate definition of energy is a quite difficult task. We can try to define it though as:

An algebraic combination of different measurable physical parameters, which can take different forms depending on the concerned phenomena, which can be transformed, transported and stored, and whom value is conserved in any circumstance.

These different forms of energy can come from various natural sources around the world. If we want to manipulate energy, we have to establish and use energy balances.

Any exergy analysis must be primarily based on mass and energy balances of the same considered system. In the coming pages, expressions/formulas used to elaborate mass, energy and exergy balances of our HVAC system are going to be presented. In addition, the results of the calculations will be showed below.

1. System description

The image in figure 2 represents our simplified system of our HVAC: system equipped building with an air recirculation process. It also shows us the different reference points used in calculations. Each part of this system must be studied separately. Afterward, all this information will be used to estimate the energy generated/used in the HVAC.

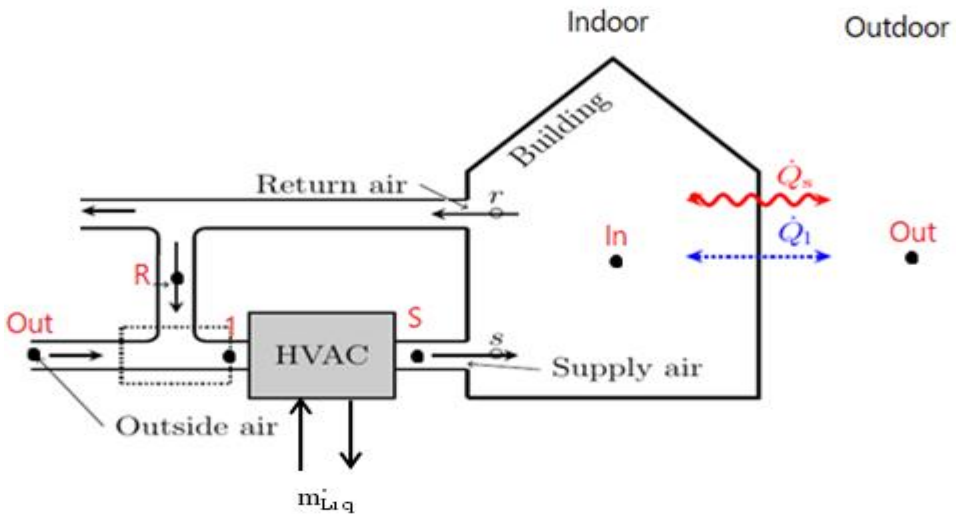

Figure 2: Model considered in the rest of this work, with a possible humidification and dehumidification air process through the HVAC system.

2. Humidity

We can define humidity as "the amount of water vapor present in the air". The amount of water vapor that is needed to achieve saturation increases as the temperature increases. That means as the temperature of a parcel of water becomes lower it will eventually reach the point of saturation without adding or losing any mass of water.

We are going to use two main measurement of humidity: the specific humidity, henceforth noted ω, and the relative humidity noted φ.

2.1. Specific humidity

Specific humidity, also called mixing ratio, is defined as the ratio of the mass of water vapor to the mass of dry air vapor is mixed with (ASHRAE, 2001):

$$\omega = 0.622 \times \frac{p_v}{p_{atm} - p_v} = 0.622 \times \frac{p_{sat} * \varphi}{p_{atm} - p_{sat} * \varphi} \quad (1)$$

p is the pressure, with the subscripts "atm", "v" and "sat" that mean atmospheric, water vapor and saturation, respectively. The calculation of the saturation pressure can be done thanks to the Buck equation (Buck, 1981):

$$p_{\text{sat}}(\theta) = A \cdot \exp\left(\left(B - \frac{\theta}{C}\right) \cdot \left(\frac{\theta}{D + \theta}\right)\right) \quad (2)$$

With the pressure p in Pa, the relative temperature θ in °C and the following constants:

$A = 611{,}21$	B =18,678	C =234,5 °C	D =257,14 °C

2.2. Relative humidity

The relative humidity (φ) of and air-water mixture is defined as the ratio of the actual partial pressure of water vapor in the mixture to the saturation vapor pressure of water. It is expressed as a percentage; a higher percentage means that the air-water mixture is closer to its saturation state.

$$\varphi = \frac{p_v}{p_{sat}} \quad (3)$$

3. Mass balance

A mass balance, also called a material balance, is an application of the law of conservation of mass to the analysis of physical systems. We are going to analyze the air and water mass balances of the considered HVAC system

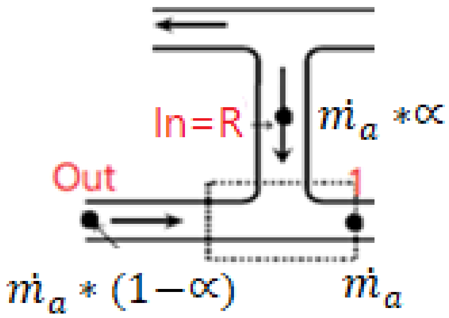

Figure 3: Model of the air recirculation system. References points "Out", "R" and "1".

The figure 3 shows us the three different reference points: Point "Out", for the outside air; point "R", for returned air (in this case, the returned air is the same air inside the building, it could be called "R" or "In"); and point "1", for the air incoming air into the HVAC system. We introduce the recirculated air fraction, represented with the term ∝; or in other words, the outside air fraction, above aforementioned, as $X = (1-\propto)$.

The water vapor mass balance in the air recirculation system can be expressed as:

$$w_1 = \propto * \, w_{int} + (1-\propto) * w_{out} \quad (4)$$

Mass balance of water in the HVCA system itself, the same balance can be written as in equation (4):

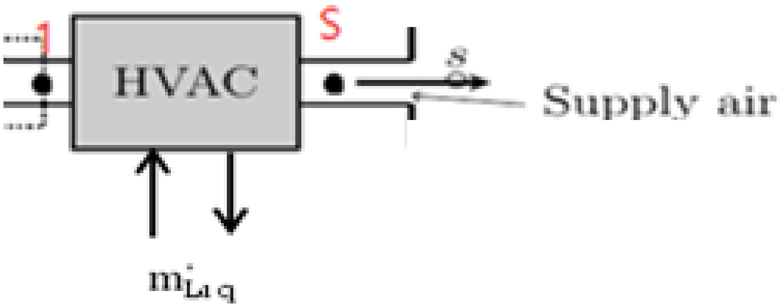

Figure 4: Model of the HVAC system. References points “1” and “S”.

The figure 4 shows us the two references points in the HVAC system and the flow of water (called liquid) in its inside. The point “S” represented the supplied air going into the building. Using this simple model, we could determinate the equation 5 below:

$$\dot{m}_a * (w_1 - w_s) = \dot{m}_{L1q} \tag{5}$$

Mass balance of water in the building:

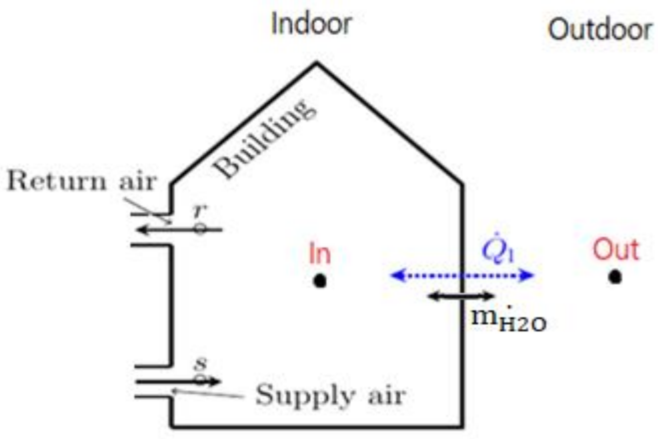

Figure 5: Model of the equipped building and his exchange of water with the exterior.

The figure 5 shows us the representation of the building and his exchange of water with the exterior. This flow of water can be calculated with the equation 6 below:

$$\dot{m}_a * (w_{int} - w_s) = \dot{m}_{H20} \tag{6}$$

4. Energy Balance

Several problems are found if we talk about acclimatization of building and/or thermal barriers. Thermal barrier coatings serve to insulate building components from large and prolonged heat load; they could sustain an appreciable temperature difference between the load-bearing alloys and the coating surface.

4.1. Sensible Load

Sensible heat is the energy related to changes of temperature of a substance or object. **Sensible load** is the thermal power exchanged by the building due to the difference of temperature between its inside and outside (because of its limited thermal barrier capacity); this load has to be balanced out by the HVAC system (supplied air in Point S).

$$\dot{Q}_S = \dot{m}_a * Cp_a * (T_{int} - T_S) \tag{7}$$

4.2. Latent Load

In nature, the phase change of a substance (from liquid to vapor for example) requires the addition or removal of heat. Latent heat is the energy required to change the phase of a body with no temperature change. **Latent load** is the thermal power exchanged by the building due to the exchange of water with its surroundings (See figure 5).

$$\dot{Q}_L = \dot{m_{H2O}} * L_V \tag{8}$$

- The enthalpy of vaporization of water is $L_v = 2501$ kJ/kg.

4.3. Enthalpy of moist air

Specific enthalpy of moist air is the sum of the two specific enthalpies: the **dry air** one and the **water vapor** one. It can be given by the following equation, (ASHRAE, 2001):

$$h = h_a + \omega \cdot h_g = Cp_a \cdot \theta + \omega \cdot (L_v + Cp_v \cdot \theta) \tag{9}$$

As we have already explained, this equation represents two different enthalpies, the dry air one, related with its sensible heat only (so involving temperature variations); and the water vapor one, related with both its sensible and latent heat (so involving temperature variations as well as humidity). The standard physical properties of dry air and water vapor, as presented in equation (9), are (ASHRAE, 2001):

- The specific heat at constant pressure of dry air is $Cp_a = 1.006$ kJ/(kg.K).
- The specific heat at constant pressure of water vapor is $Cp_v = 1.827$ kJ/(kg.K).

4.4. Dew point Temperature

The dew point is the temperature to which air must be cooled to become saturated with water vapor. There are several ways to estimate the dew point temperature; in this analysis, we are going to use the following equation (ASHRAE, 2001):

$$T_{DP} = C14 + C15 * \delta + C16 * \delta^2 + C17 * \delta^3 + C18 * {p_w}^{0.1984} \tag{10}$$

With the following constants:

C14=6.54	C15=14.526	C16=0.7389	C17=0.09486	C18=0.4569

And with $p_w = \varphi * \frac{\text{Psat}}{1000}$ and $\delta = \ln(p_w)$

4.5. HVAC Power consumption

Finally, with all of the expressions above and the equation 11(See figure 3), the HVAC power consumption can be calculated thanks to the following equation:

$$h_1 = \propto * h_{int} + (1-\propto) * h_{out} \tag{11}$$

$$\dot{Q_{CTA}} = \dot{m}_a * (h_1 - h_S) \tag{12}$$

5. Exergy Balance

Exergy can be defined as the maximum amount of *useful energy* we can expect to gather in the best situation possible from a given amount of matter, (Bejan, 2006). As well as useful energy, there is also one lost/destroyed energy, this is known as Anergy, So, energy is divided in Exergy and Anergy, useful energy + destroyed energy.

In ideals processes, exergy and anergy remain the same, but we know that real processes are not ideals, so anergy increase along the process and, as a consequence, exergy decrease.

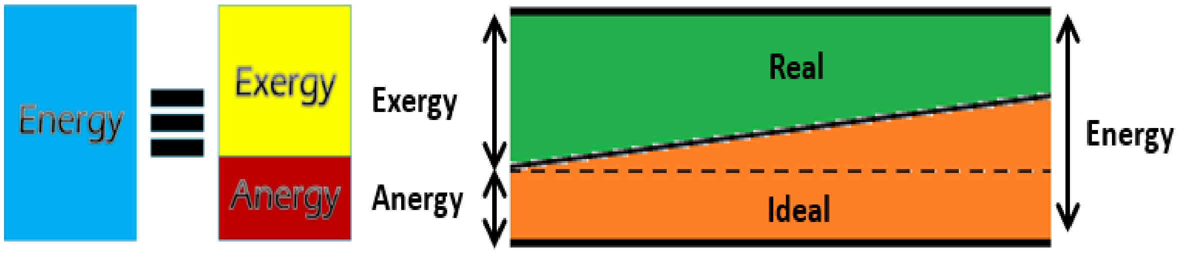

Figure 6: Representation of energy as the sum of exergy and anergy.

In the next pages, expressions/formulas used to elaborate one energy analysis/balance our HVAC system are going to be presented.

According to the concept of exergy itself (Bejan, 2006), we have to consider in our balance that any amount of work can be transformed into useful energy (100% Exergy), but not all the heat could be transformed into useful energy. In this last case, The Carnot Factor (Θ) must be used to calculate the maximum amount of useful energy. Exergy of moist air can be decomposed in three different components, (Wepfer, Gaggioli, & Obert, 1979):

a. Thermal exergy
b. Mechanical exergy
c. Chemical exergy

5.1. Thermal Exergy of moist air

The part of the thermal exergy of most air follows the equation below:

$$ex_{Th} = (Cp_a + \omega \cdot Cp_v) \cdot T_0 \cdot \left(\frac{T}{T_0} - 1 - \ln\frac{T}{T_0}\right) \qquad (13.1)$$

Otherwise, we can use the following modified expression based on the log mean Carnot Factor noted $\breve{\Theta}_\infty$.

$$ex_{Th} = (Cp_a + \omega\ Cp_v) \cdot (T - T_0) \cdot \breve{\Theta}_\infty \qquad (13.2)$$

And it is defined as:

$$\breve{\Theta}_\infty = 1 - \frac{T_0}{\breve{T}_\infty} \qquad (14)$$

With the Logarithmic mean temperature (LMD), noted $\breve{T}_\infty$ and defined by:

$$\breve{T}_\infty = \frac{T - T_0}{\ln\,(T/T_0)} \qquad (15)$$

5.2. Mechanical Exergy of moist air

The part of the mechanical exergy of most air follows the equation below:

$$ex_{Me} = (r_a + r_v \cdot \omega) \cdot T_0 \cdot \ln\frac{p}{p_0} \qquad (16)$$

5.3. Chemical Exergy of moist air

The part of the chemical exergy of most air follows the equation below:

$$ex_{Ch} = \mathrm{T_0} \cdot \left[(\mathrm{r_a} + \mathrm{r_v} \cdot \omega) \cdot \ln\left(\frac{\frac{\mathrm{r_a}}{\mathrm{r_v}} + \omega_0}{\frac{\mathrm{r_a}}{\mathrm{r_v}} + \omega}\right) + \mathrm{r_v} \cdot \omega \cdot \ln\left(\frac{\omega}{\omega_0}\right) \right] \tag{17}$$

Finally, the expression to calculate the specific total exergy considering air is (Wepfer, Gaggioli, & Obert, 1979) (Dincer & Rosen, 2013):

$$\begin{aligned} ex_{TOTAL} = {} & (\mathrm{Cp_a} + \omega\, \mathrm{Cp_v}) \cdot (\mathrm{T} - \mathrm{T_0}) \cdot \breve{\Theta}_\infty + (\mathrm{r_a} + \mathrm{r_v} \cdot \omega) \cdot \mathrm{T_0} \cdot \ln\left(\frac{\mathrm{p}}{\mathrm{p_0}}\right) + \\ & \mathrm{T_0} \cdot \left[(\mathrm{r_a} + \mathrm{r_v} \cdot \omega) \ln\left(\frac{\frac{\mathrm{r_a}}{\mathrm{r_v}} + \omega_0}{\frac{\mathrm{r_a}}{\mathrm{r_v}} + \omega}\right) + \mathrm{r_v} \cdot \omega \cdot \ln\left(\frac{\omega}{\omega_0}\right) \right] \end{aligned} \tag{18}$$

The gas constant for the dry air is $\mathrm{r_a} = 0.287$ kJ/kg.K, used by standard convention.

The gas constant for the water vapor is $\mathrm{r_v} = 0.462$ kJ/kg.K, used by standard convention.

5.4. Destruction of Exergy

After having taken into account both kinds of exergy, we can calculate the destruction of exergy in each part of the HVAC system.

Applying one exergy balance in the recirculation system (See figure 3), we obtain the following equation 19.1:

$$\dot{\mathrm{m}}_\mathrm{a} * \propto * \, ex_{int} + \dot{\mathrm{m}}_\mathrm{a} * (1 - \propto) * ex_{out} - \dot{\mathrm{m}}_\mathrm{a} * ex_1 - \dot{Ex}_{D-RC} = 0 \tag{19.1}$$

Which can be expression by flow of air; we obtain the following expression of the destruction of exergy in the recirculation system:

$$\frac{\dot{Ex}_{D-RC}}{\dot{\mathrm{m}}_\mathrm{a}} = \propto * \, ex_{int} + (1 - \propto) * ex_{out} - ex_1 \tag{19.2}$$

In the other hands, applying one exergy balance in the HVAC system (See figure 4), we obtain the equation 20.1 below:

$$\dot{\mathrm{m}}_\mathrm{a} * ex_1 - \dot{\mathrm{m}}_\mathrm{a} * ex_s - Q_{Cold} * \Theta_{Cold} - \dot{\mathrm{m}}_{\mathrm{liq}} * ex_{H20} - \dot{Ex}_{D-HVAC} = 0 \tag{20.1}$$

Which can be expression by flow of air; after combination with equation (5), we obtain the following simplified expression of the destruction of exergy in the HVAC system:

$$\frac{\dot{Ex}_{D-HVAC}}{\dot{\mathrm{m}}_\mathrm{a}} = ex_1 - ex_s - (h_1 - h_s) * \Theta_{Cold} - (w_1 - w_s) * ex_{H20} \tag{20.2}$$

The following equations must be used to calculate the simplify expression:

The HVAC power consumption is the heat given by the flow of water (See figure 4).

$$Q_{Cold} = (h_1 - h_s) * \dot{\mathrm{m}}_\mathrm{a} \tag{21}$$

The Carnot Factor of this flow of water (Θ_{Cold}):

$$\Theta_{Cold} = 1 - \frac{T_0}{T_{cold}} \tag{22}$$

We introduce the pinch point which is defined as the point where temperature difference between the two composite curves is minimum.

The calculation of the cold temperature can be determinate using the pinch of the HVAC system. Normally, the pinch used in the HVAC system is 15°C. However, this amount is theoretically and it has to be calculated depending on the machine analyzed.

$$T_{cold} = T_{DP} - Pinch \tag{23}$$

The equation 24 is one simple expression of the exergy of water which is used to try to simplify this calculation (Bejan, 2006):

$$ex_{H20} = -r_v * T_0 * \ln{(\varphi_0)} \tag{24}$$

6. Performance criteria

The main performance criterion used in the rest is the Duty Specific Exergy Consumption or DSExC, initially defined as, (Rivero, de Oliveira Jr., & Le Goff, 1990):

$$DSExC = \frac{\textit{Rate of exergy destroyed}}{\textit{Electricity Consumption}}$$

This criterion, already used in the specific context of psychrometric applications (Réguillet, et al., 2013), shows us the relation between the total exergy destroyed and the total useful transferred load. Applied to our Air-Handling Unit (AHU), this criterion can be expressed as:

$$DSExC = \frac{\dot{E}x_D}{\dot{E}} = \frac{\dot{E}x_{D-RC} + \dot{E}x_{D-HVAC}}{\sum \dot{Q} + \sum \dot{W}} = \frac{\dot{E}x_{D-total}}{\dot{m}_a \cdot (h_1 - h_s)} \cdot \frac{\dot{m}_a \cdot (h_1 - h_s)}{\sum \dot{Q} + \sum \dot{W}} \tag{25}$$

$\dot{E}x_{D-total} = \dot{E}x_{D-RC} + \dot{E}x_{D-HVAC}$ is the rate of exergy destroyed in the whole AHU and $\dot{E} = \sum \dot{Q} + \sum \dot{W}$ is the sum of the rates of energy consumed by the same AHU, while the second part of last equation right side, noted η_{AHU}, is the energy efficiency of the same AHU, given by:

$$\eta_{\mathrm{AHU}} = \frac{\dot{m}_a \cdot (h_1 - h_s)}{\sum \dot{Q} + \sum \dot{W}} \tag{25.b}$$

Equation (25) can then be expressed as the product of two parameters, the first one being related to moist air itself, and the second one to the energy performances of the whole Air-Handling Unit (25.b).

C. Experimental Apparatus

1. HVAC system

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 whole system is presented in Figure 7.

The use of measuring instruments makes possible the statements of energy and mass balances across each *psychrometric*[4] *process*.

Figure 7: Air treatment station A660 Base Unit with A661B Recirculation Duct Upgrade (P.A.Hilton)

The used HVAC system is detailed schematically in Figure 8. The fresh air is sucked in, at the bottom left, at the point marked F. Afterward, it is sucked into the air treatment system by a fan located near A and it goes through various operations:

a. Heating process between B and C.
b. Cooling process between C and D.
c. Humidification process between C and D.

It is finally rejected to supply the room to be ventilated at point E.

A fraction of the air flow treated, called recirculated air flow and about which we already discussed in the previous chapter, is mixed with the incoming fresh air. The fraction of the flow treated is managed from the valve at point 42.

In Figure 8, the most important points to emphasize are the following:

1.-Input air flow	6.-Re-heating resistors	10.- Control valve of fan's speed	41.-Output non-recirculated air flow
2.-Measurements of dry and wet air	7.-Diaphragm	18.-Input water flow	42.-Control valve of recirculated air flow.
4.-Pre-Heating resistors	8.-Supplied air or treated air	24.-Compressor	
5.-Evaporator	9.-Fan	40.-Principal pipe of the HVAC system	

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

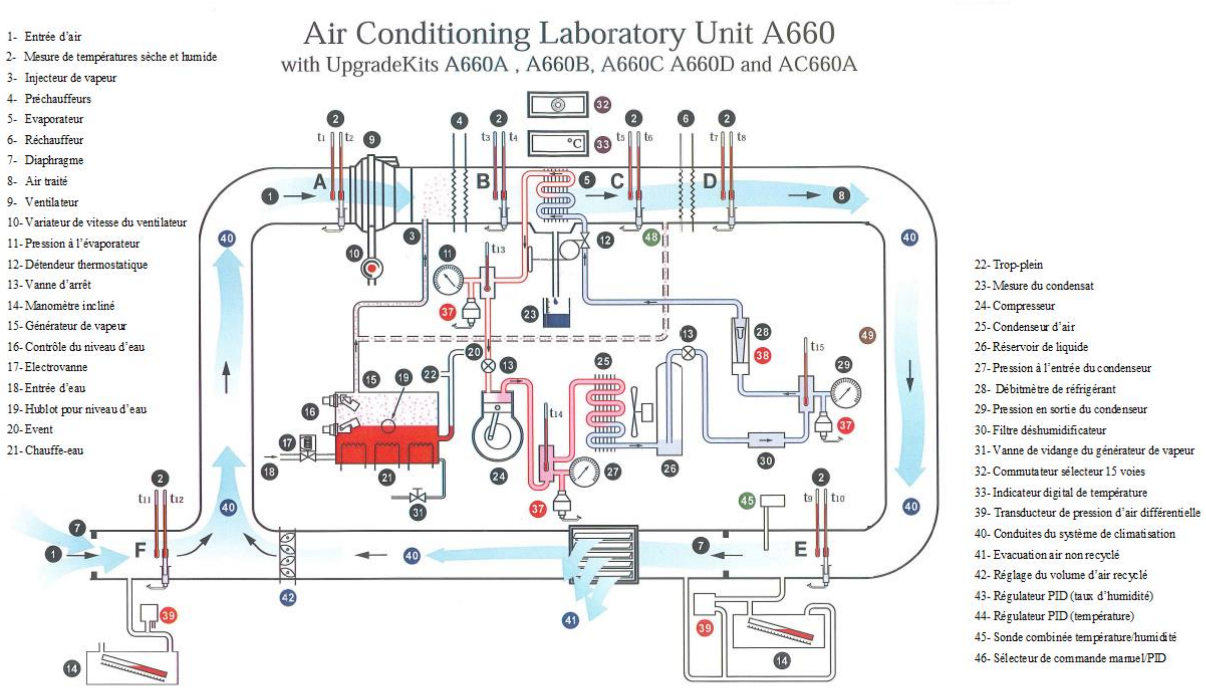

Figure 8: Diagram of the HVAC system of the department of Energy.

2. Measurements processes

2.1. Air mass flow rate

The two air flows required to establish a complete system balance — so the aspirated and the treated one, the latter being the sum of the former and of the recirculated flow— are measured using diaphragms, resulting pressure drop being measured by piezometrics tubes placed at points E and F, as presented in figure 8. The relation between the measured pressure drop and the searched mass flow rate, obtained from previous experiments, is:

$$\dot{m}_a = \rho \cdot S_0 \cdot V_0 = \frac{C' \cdot \sqrt{\frac{\Delta z}{v}}}{1 + \omega} \qquad (26)$$

The factor C' (overall flow coefficient) is respectively equal to:

- 0.0517 For the input point 1
- 0.0529 For the point 7.

Δz is the presssure drop actually measured, and expressed in mm of water, and v is the moist air specific volume, expressed in m^3/kg, and whom value can be obtained thanks to specific softwares or from the following formula:

$$v = \frac{r_v \times (0.622 + \omega) \times T}{(1 + \omega) \times p_{atm}} \qquad (27)$$

T is the air temperature and ω its specific humidity (see Chapter 2.2), whom measurement will be presented below.

2.2. Temperature and Humidity

The humidity of air is measured at different points of the circuit, using sensors of the "wet bulb" type. Such sensors actually measure two different temperatures:

1. The first one is called **dry bulb temperature**, noted T, and is just the usual temperature of air.
2. The second one is the **wet bulb temperature**, noted T_{wet}, which is the temperature read by a thermometer covered in water-soaked cloth over which air is passed.

The dry and the wet temperatures are measured through usual thermocouple temperature sensors (of K type) and recorded by the computer, as for the air mass flow rates previously introduced. These temperatures are noted t1, t2 ... t12 in Figure 8.

Firstly, the measured temperatures are converted into water partial pressure of vapor ($\mathrm{p_v}$) using the following experimental correlation, which demonstration can be found for example in (QUINTANILLA MUÑOZ, 2016):

$$\begin{aligned} p_v \approx p_{sat}(T_{wet}) - \frac{c_{p,a}}{\alpha \cdot L_v} \cdot p \cdot (T - T_{wet}) \\ = p_{sat}(T_{wet}) - 0{,}000666 \times p \times (T - T_{wet}) \end{aligned} \quad (28)$$

p is here the "total pressure" of air. In this correlation, we calculate the saturated pressure using equation (2). Afterward, the partial vapor pressures are converted into specific humidity, using either, the diagram of moist air, a specific software, or equation (1). Finally, as well as specific humidity, the obtained partial pressure of vapor is subsequently converted into relative humidity using equation (3).

2.3. Electricity consumption

The total electricity consumption of the HVAC system is the sum of its fan's power; its compressor's power (both as work consumption); and its Pre and Re-heating resistors consumption (both as heat consumption).

2.3.1. Fan Power Consumption

The air flow rate sucked in by the HVAC system is controlled by the electrical voltage applied to the fan (corresponding to point 10 in Figure 8). Once the fan voltage is set, we have to wait about 10 minutes before taking measurements. In addition, the fan power consumption (W_F) can be obtained thanks to a function of the supply voltage, supplied by the concerned system manufacturer. This function is drawn in Figure 9.

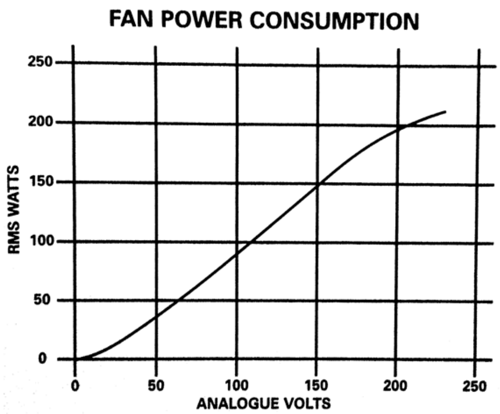

Figure 9: Experimental correlation of the fan consumer electrical power vs. supply voltage

2.3.2. Compressor Power Consumption

The compressor power consumption ($\dot{W}_C$) can be calculated with the following equation (29), coming from the compressor manufacturer, Tecumseh[5]:

$$\begin{aligned} \dot{W}_C = C1 + C2 * T_e + C4 * {T_e}^2 + C7 * {T_e}^3 + \left(C3 + C5 * T_e + C8 * {T_e}^2\right) * T_c \\ + (C6 + C9 * T_e) * {T_c}^2 + C10 * {T_c}^3 \end{aligned} \quad (29)$$

[5] *Datasheet available at : http://www.tecumseh.com/en/europe/Products/Reciprocating-Compressors/AJE4492YGZ?pdf=performance*

With the following constants:

C1= 4.128*10^{2}	C2= 3.000	C3= 9.960	C4= 4.675*10^{-3}	C5=1.603*10^{-2}
C6= -7.876*10^{-4}	C7= 1.354*10^{-3}	C8= -1.867*10^{-3}	C9= 5.002*10^{-3}	C10= -4.390*10^{-4}

T_e and T_c are respectively the evaporation and condensation temperatures of the refrigerant fluid (here the R134a), in °C.

2.3.3. Heating Resistors Consumption

Each of the heating electrical resistors (Pre-heater and Re-heater) dissipates a heat rate of $\dot{Q}_{heater} = 1{,}2\ kW$. The boiler used to humidity air uses two electrical resistances of 3 kW and one of 1 kW.

Finally, the total electricity consumption can be computed thanks to:

$$\dot{E} = \sum \dot{Q} + \sum \dot{W} = \dot{Q}_{H-R} + \dot{W}_F + \dot{W}_C \tag{30}$$

3. Equivalence System

We have presented the model of our HVAC (See figure 2) with the following reference points: in; out; 1; and "s". However, in the diagram of the experimental apparatus (see figure 8) we have different reference points, A-F. In the figure 10, one equivalence system (with A-F reference points) will be presented to make easier the calculations.

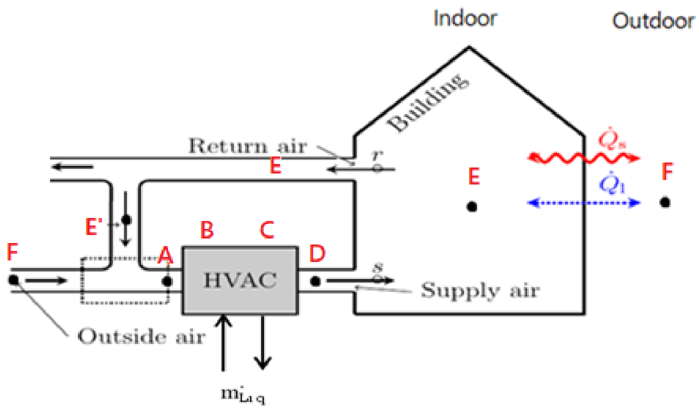

Figure 10: Equivalence HVAC system with reference points A-F.

D. Results

With the help of our HVAC system (see Chapter 3.1), we have done several experimental tests with different values of recirculated air fraction (∝), two main cases processes have been studied:

a. Air cooling process (ignoring the $H_2 0$ exergy destruction)
b. Air cooling and dehumidification process (counting the $H_2 0$ exergy destruction)

The balance and analysis will be done in the following 2 point of view:

1. The total AHU system, between points F-E′ (see figure 8 and 10 above); the air recirculation system (points F -E′-A) and the HVAC system (points A-D)
2. The single cooling system, between points B-C (see figure 8 and 10 above); the portion of our HVAC system which makes the cooling process (points B-C).

1. The total AHU system

The total AHU system (see chapter 1.3) has two main source of exergy destruction:

- The Air Recirculation System.
- The HVAC system itself.

In the incoming pages, the effect of the variation of the recirculated air fraction over the exergy destruction and in the Duty Specific Exergy Consumption will be displayed.

1.1. Air cooling process test

We manipulate the recirculated air fraction (∝) and we calculate the destroyed exergy (by outdoor air flow) in the recirculated system, the air handling units and the whole HVAC system, the results are shown in the table 1 and figure 11.

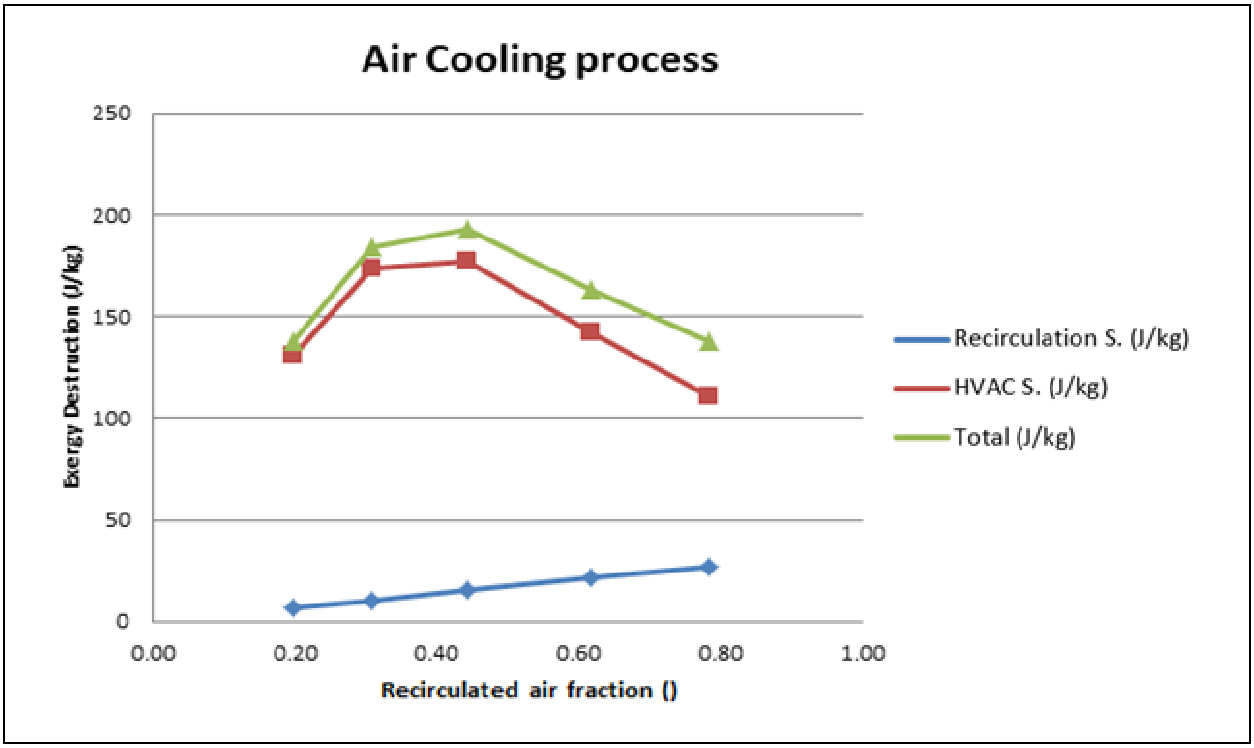

Figure 11: Exergy destruction vs recirculated air fraction for the air cooling process.

Table 1: Data values of the air-cooling process test

∝ (%)	$\dot{m}_a$ (kg/s)	$\dot{m}_{E'}$ (kg/s)	Recirculation S. (J/kg)	HVAC S. (J/kg)	Total (J/kg)	DSExC (%)
19.71%	0.18	0.035	6.51	131.24	137.75	1.11%
30.75%	0.18	0.055	10.25	174.03	184.29	1.51%
44.33%	0.18	0.081	15.83	176.84	192.67	1.60%
61.65%	0.19	0.115	21.67	141.82	163.49	1.40%
78.49%	0.19	0.150	26.99	110.45	137.44	1.20%

Additionally, figure 12 shows us the relation between the recirculated air fraction and the Duty Specific Exergy Consumption calculated for the whole AHU system.

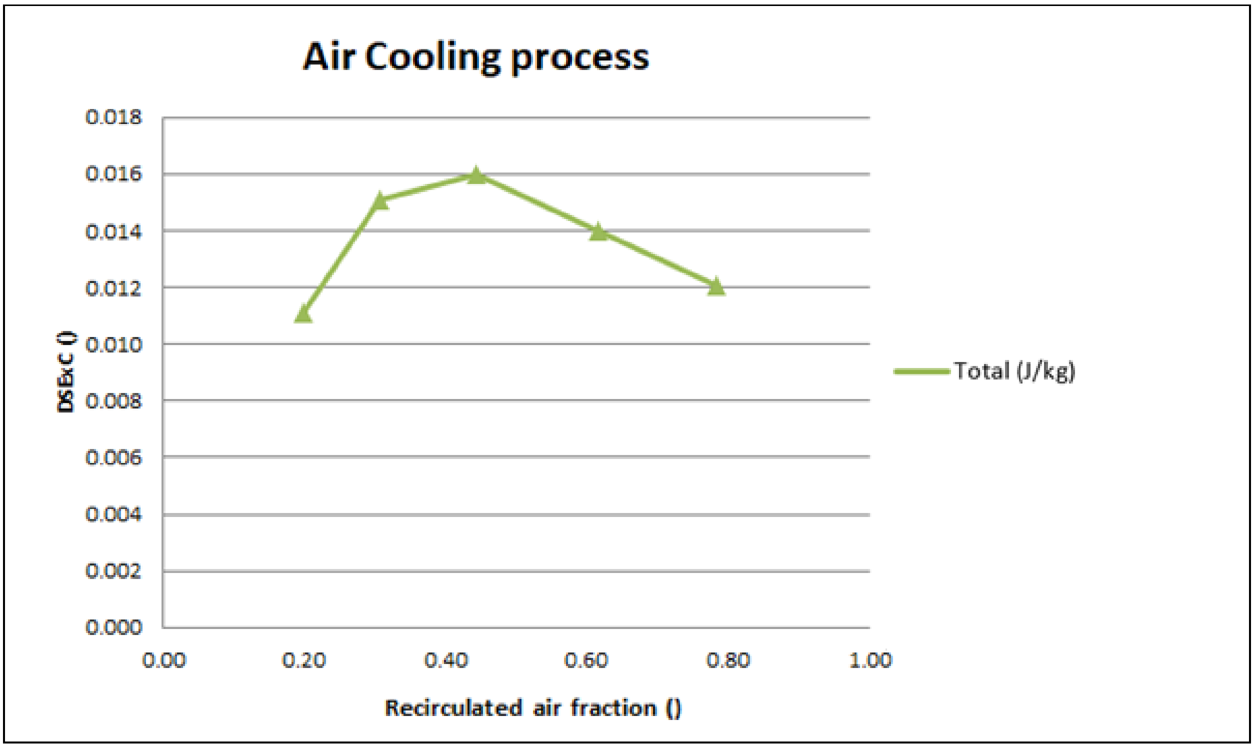

Figure 12: DSExC vs recirculated air fraction for the air cooling process.

1.2. Air cooling and dehumidification process test

One dehumidification process is added in this test. As well as in first test, we manipulate the recirculated air fraction (∝) and we calculate the destroyed exergy in the recirculated system, the air handling units and the whole AHU system, the results are shown in the table 2 and figure 13.

Table 2: Data values of the air cooling/dehumidification process test

∝ (%)	$\dot{m}_a$ (kg/s)	$\dot{m}_{E'}$ (kg/s)	Recirculation S. (J/kg)	HVAC S. (J/kg)	Total (J/kg)	DSExC (%)
22.64%	0.17	0.040	26.27	636.10	662.37	11.33%
35.34%	0.18	0.064	35.66	552.01	587.66	11.04%
57.57%	0.19	0.108	60.04	460.69	520.72	10.36%
70.23%	0.19	0.135	73.50	398.37	471.87	9.85%

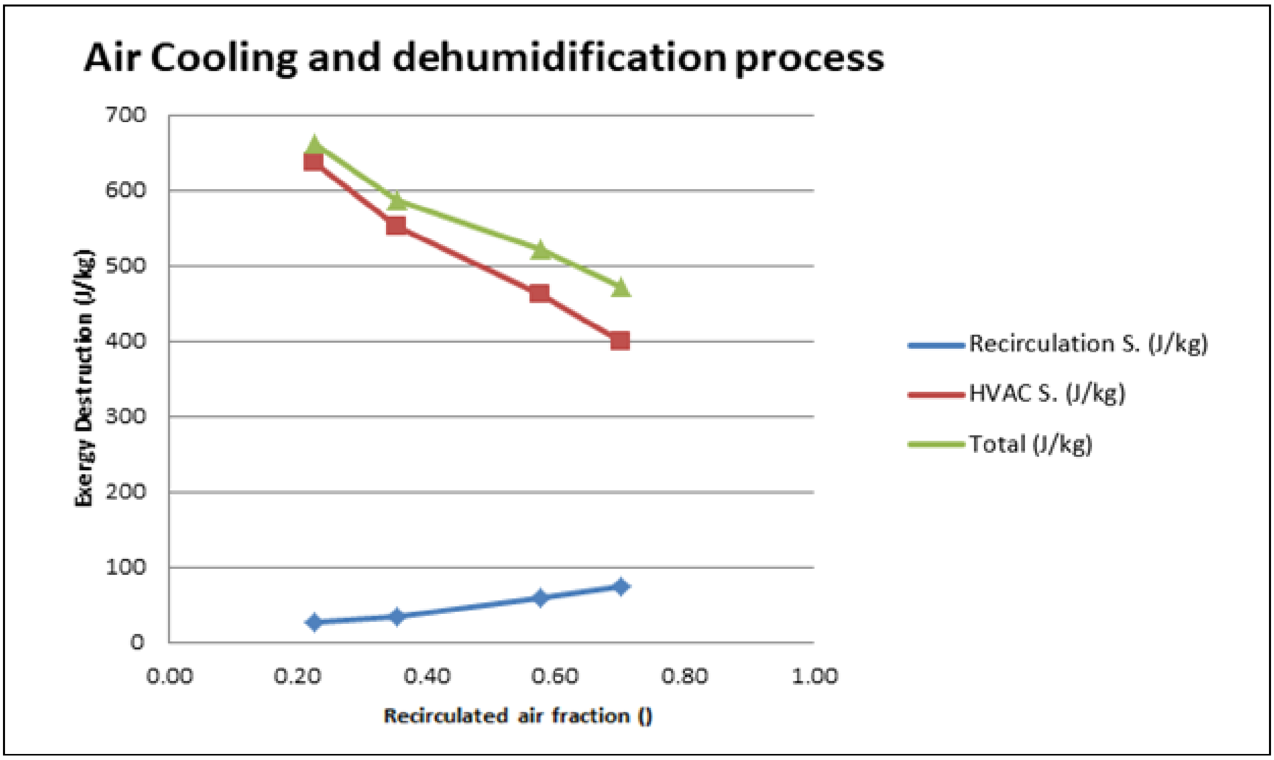

Figure 13: Exergy destruction vs recirculated air fraction for the air cooling/dehumidification process.

Afterward, figure 14 shows us the relation between the recirculated air fraction and the Duty Specific Exergy Consumption calculated for the whole AHU system in this test.

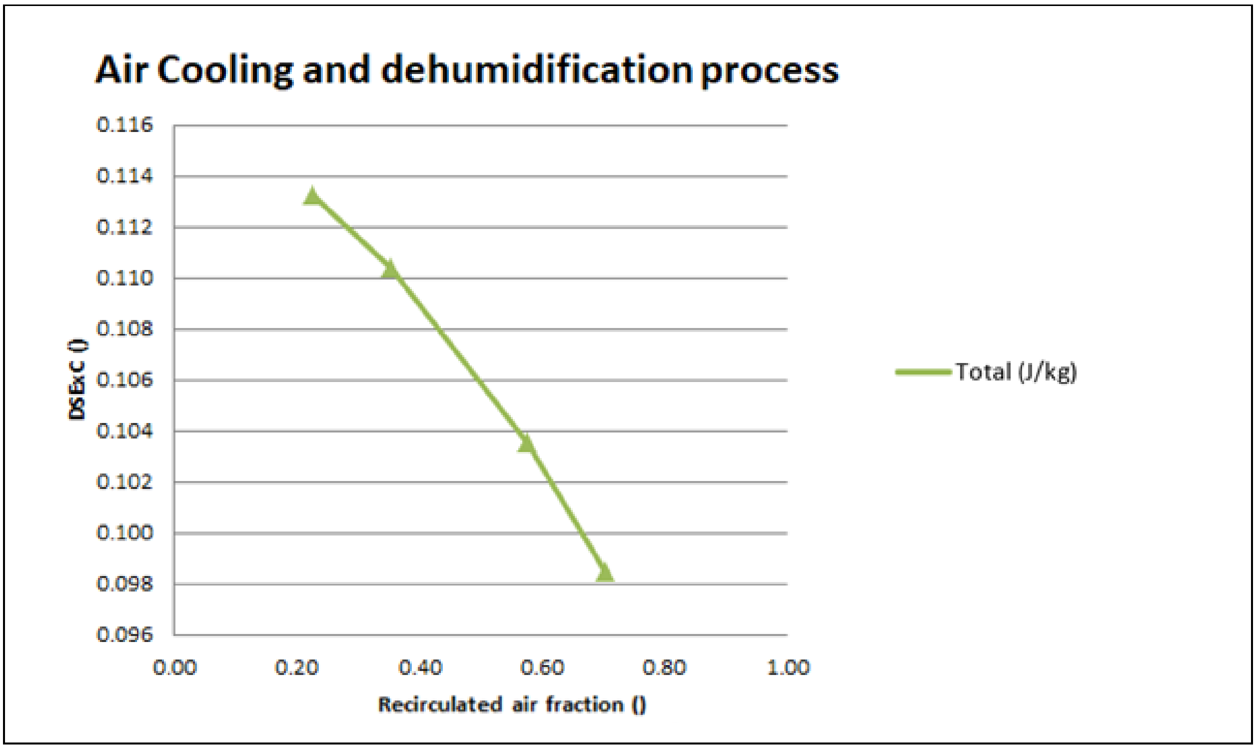

Figure 14: DSExC vs recirculated air fraction for the air cooling/dehumidification process.

2. The single Cooling System

In this part of the chapter, the exergy balance will be done only in the single cooling system of our experimental apparatus (between B-C, see figure 8 and 10). However, this analysis change all equation explained before. Those equations will be updated:

Firstly, the system doesn't count the air recirculation system.

Then, the exergy destruction equation 20.2 will be changed for the following equation 31, where we don't have exergy of water:

$$\frac{\dot{E}x_D}{\dot{\mathrm{m}}_\mathrm{a}} = ex_B - ex_C - (h_B - h_C) * \Theta_{Cold} \tag{31}$$

Finally, the calculation of the DSExC (equation 25) will be calculated counting only the compressor power, we can see it in the equation 32; also, the energy efficiency of the cooling system (η_{CS}) will be updated in the equation 32.b.

$$DSExC = \frac{\dot{E}x_D}{\dot{E}} = \frac{\dot{E}x_D}{\sum \dot{Q} + \sum \dot{W}} = \frac{\dot{E}x_D}{\dot{m}_a \cdot (h_B - h_C)} \cdot \frac{\dot{m}_a \cdot (h_B - h_C)}{\dot{W}_C} \tag{32}$$

$$\eta_{\mathrm{CS}} = \frac{\dot{m}_a \cdot (h_B - h_C)}{\dot{W}_C} \tag{32.b}$$

As we have explained, the only source of exergy destruction is our cooling system, where we will analyse the effect of the variation of the recirculated air fraction over the exergy destruction and in the Duty Specific Exergy Consumption.

2.1. Air cooling process test

We manipulate the recirculated air fraction (∝) and we calculate the destroyed exergy in the air-cooling system, the results are shown in the table 3 and figure 15.

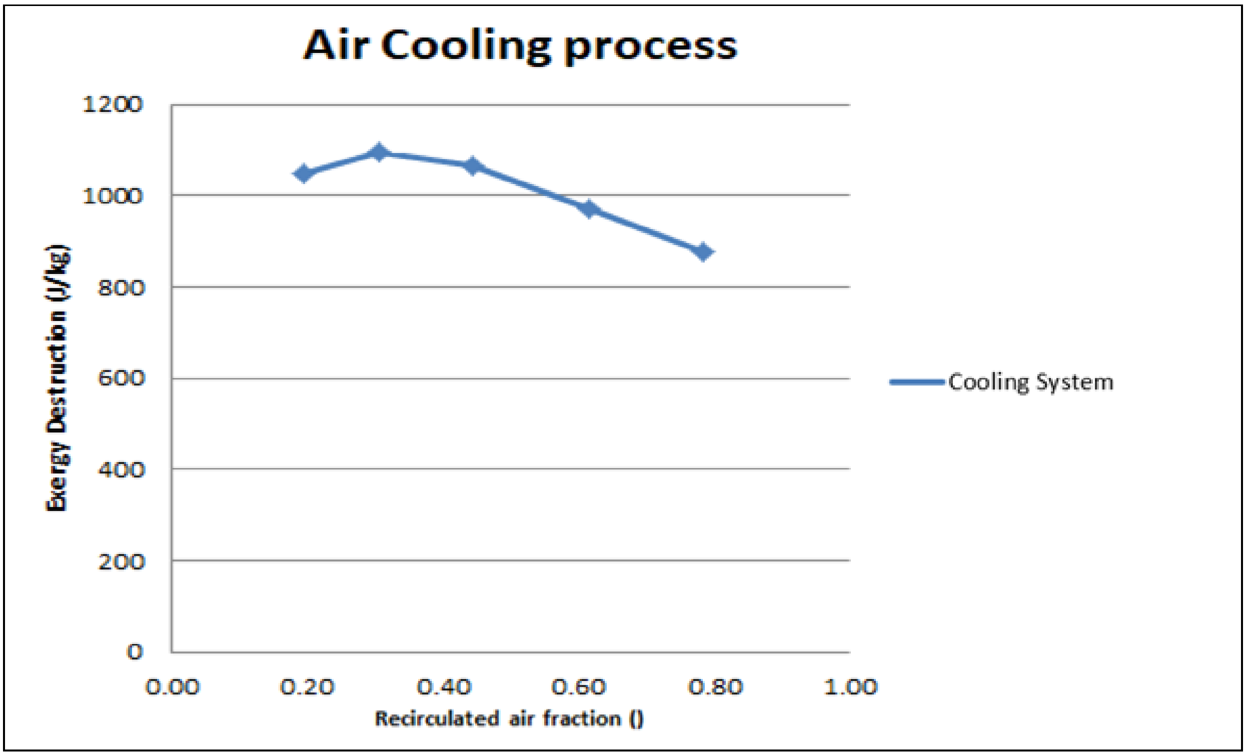

Figure 15: Exergy destruction vs recirculated air fraction for the air cooling process in the single cooling system.

Table 3: Data values of the air-cooling process test in single cooling system

∝ (%)	$\dot{m}_a$ (kg/s)	Cooling S (J/kg)	DSExC (%)
19.71%	0.18	1048.94	24.44%
30.75%	0.18	1093.50	25.68%
44.33%	0.18	1065.03	25.39%
61.65%	0.19	971.81	23.99%
78.49%	0.19	875.79	22.22%

Additionally, figure 16 shows us the relation between the recirculated air fraction and the Duty Specific Exergy Consumption calculated for the cooling system.

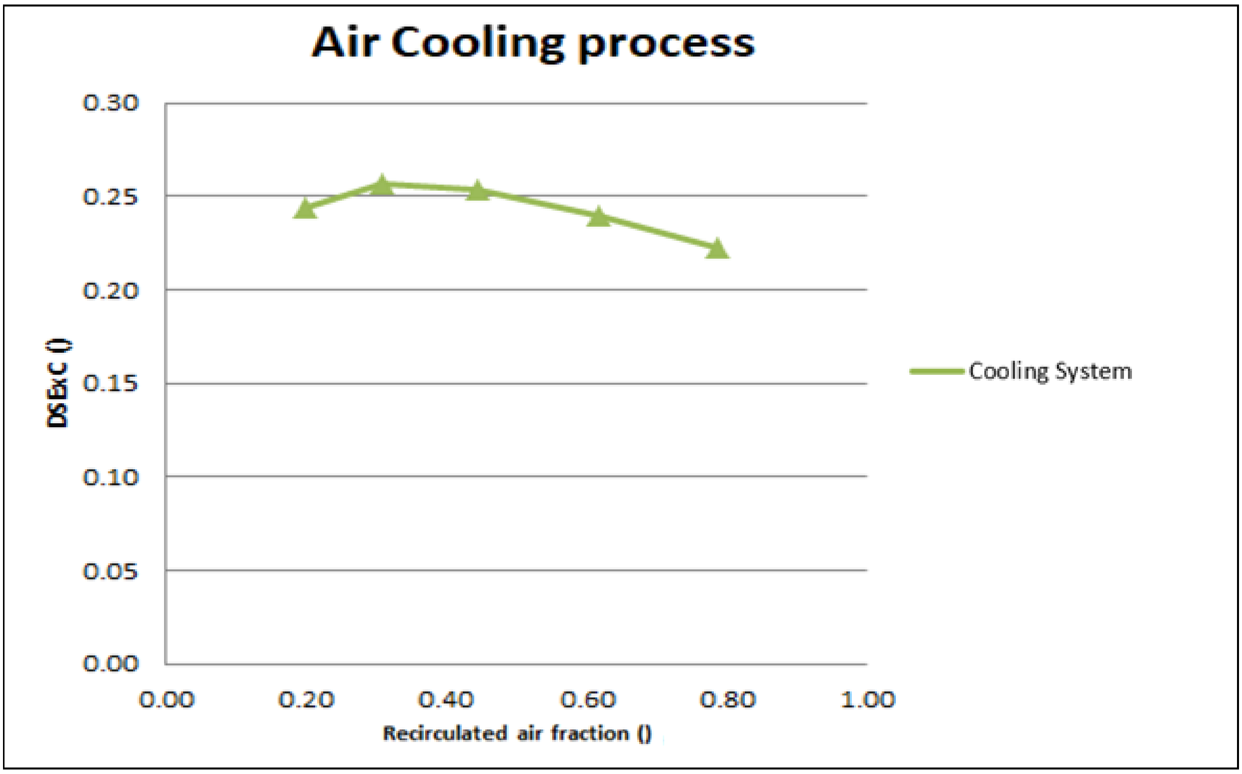

Figure 16: DSExC vs recirculated air fraction for the air cooling process process in the single cooling system.

2.2. Air cooling and dehumidification process test

One dehumidification process is added in this test. As well as in first test, we manipulate the recirculated air fraction (∝) and we calculate the destroyed exergy in the air-cooling system, the results are shown in the table 4 and figure 17.

Table 4: Data values of the air cooling/dehumidification process test in single cooling system

∝ (%)	$\dot{m}_a$ (kg/s)	Total (J/kg)	DSExC (%)
22.64%	0.17	914.88	20.08%
35.34%	0.18	751.81	18.45%
57.57%	0.19	629.54	16.45%
70.23%	0.19	545.41	15.09%

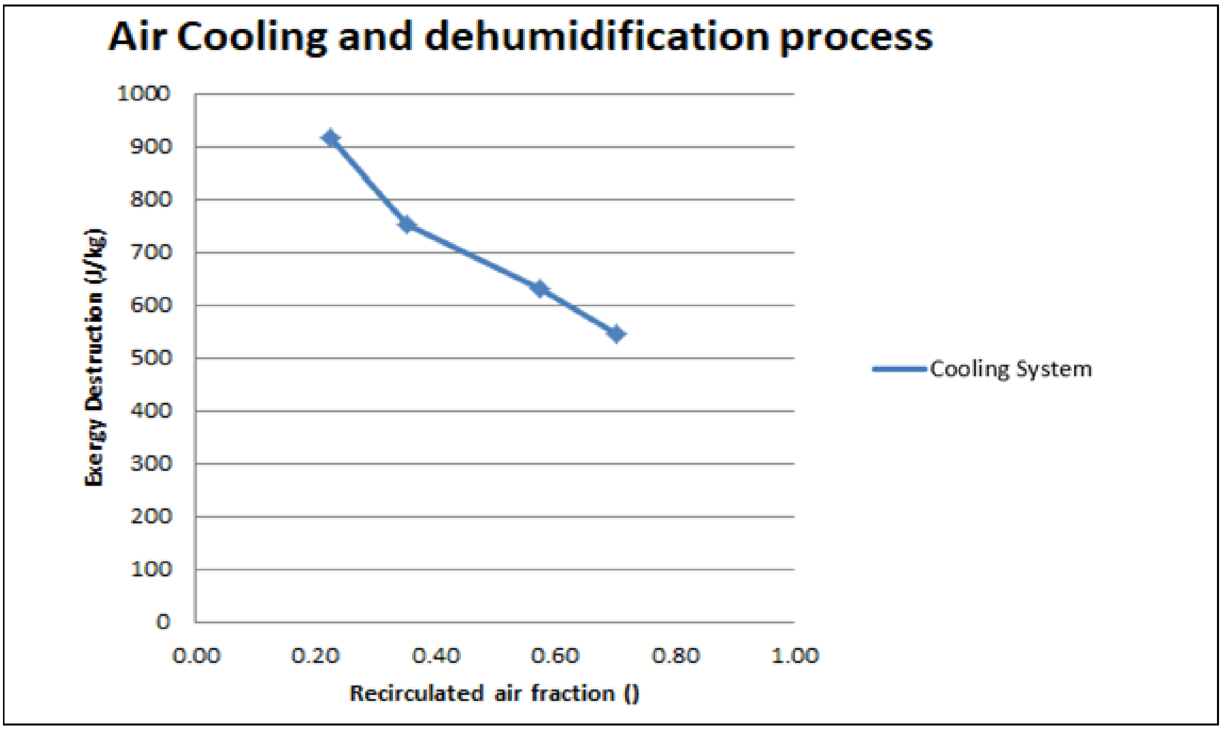

Figure 17: Exergy destruction vs recirculated air fraction for the air cooling/dehumidification process in the single cooling system.

Afterward, figure 18 shows us the relation between the recirculated air fraction and the Duty Specific Exergy Consumption calculated for the simple cooling system in this test.

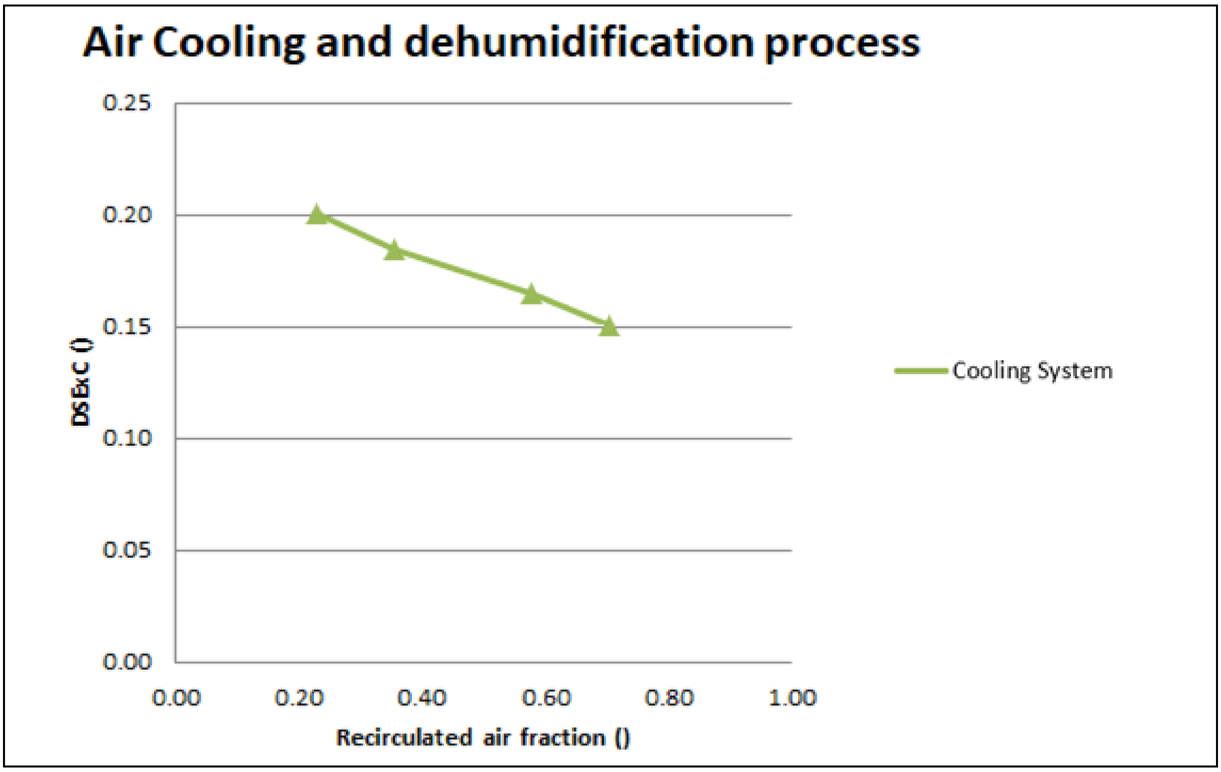

Figure 18: DSExC vs recirculated air fraction for the air cooling/dehumidification process in the single cooling system.

E. Conclusions

-In the balance of the whole AHU system, the condensed water has an important influence. The absence of this condensed water (absence of exergy of the water) makes the single air-cooling process exergy destruction lower than the air cooling and dehumidification's one. As we can see, in figure 11 and figure 13, the difference between their exergy destructions rises into 500 J/kg. Although, depending of the outdoor conditions, this amount could change. In addition, two different kind of graphic trend has been obtained in this balance of the whole AHU system:

1) In the single air-cooling process, a concave trend with the worst-case scenario around 50% of recirculated air fraction and the best case's one around 80% (figure 11).

2) In the air cooling and dehumidification process, an only decreasing trend with the best-case scenario around 80% (figure 13).

This phenomenon happens because in both analyses the same initial reference point has been used; in theory, this analysis should be correct, but the huge difference between their exergy destruction makes us think that one different or adjusted initial reference point should be used for the calculation in presence of condensed water. This adjustment could be done in any incoming exergy analysis research.

-For the total exergy destruction; normally, the highest recirculated air fraction applied, the lowest exergy destruction obtained. This trend works for the air cooling and dehumidification test, but not for the air-cooling test. In the last case, there is one maximum point of exergy destruction around 50% of recirculated air fraction which is the case the least favorable (reasons explained in the first conclusion). In other words, for this case, the best-case scenario for the exergy destruction should be 0% or 100% of recirculated air fraction applied. Unfortunately, both scenarios cannot be physically possible; the recirculated air fraction 0 % gives us one system without recirculation system, which is the main objective of the AHU; and the recirculated air fraction 100% gives us one system without input air flow, a minimum amount of outdoor air must be supplied to begin and continue the circuit air flow. In this last case, with this experimental apparatus, the maximum recirculated air fraction could be around 80% and 85%. Depending of the machine, this value could rise until 90%.

Although, the maximum value of $\propto$ (recirculated air fraction) also depends on the quality of the air inside the building/room/space, the comfort of the occupant has to be in mind while recirculating more return air or supplying more fresh air to the system. Its importance will be presented in 2 examples:

1) In a family house or a student classroom, the quality of the return air is similar to the quality of the input air (fresh air), these places don't generate any kind of particles which could change the quality of the return air. In this kind of cases, the recirculation air fraction could be the largest possible without affecting the comfort of the occupants.

2) In a gym or a physical exercise place, the occupants generate themselves different kind of fluids, for example sweat, which affect the quality of the return air. In this kind of cases, more fresh air has to be supplied into the room, or in other words, the recirculation air fraction would be strongly affected(decreased) to maintain a correct comfort for the occupants.

-In the balance of the cooling system (partial balance) between C-D (see figure 8), the exergy destruction is higher than the balance of the whole system (total balance). This could sound contradictory, but in some part of the AHU system, the specific exergy is negative. The reason of this phenomenon is the conditions of the temperature outside and inside the building. The tests have been done in summer times, which means, the temperature outside the building is higher than the temperature inside it, or in other words, the Carnot Factor is negative and the specific exergy from the heat is also negative.

-The main conclusion discussed in the paper published in the French journal dedicated to HVAC system is the following one: "The way we define the reference state of the process of humidification or dehumidification has to be checked to avoid artificial and questionable values of exergy destruction", see (Chacon Chauca, Quintanilla Munoz, & Vaudrey, 2019).

-A solution to avoid this kind of artificial and questionable values of exergy destruction has been already published, see (Chengqin, Nianping, & Guangfa, 2002), and it is the following one: "The reference state of water must be based on the saturated ambient humidity and not on the actual one".

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