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MEASUREMENT UNCERTAINTIES CALCULATION IN THE COOLING CAPACITY DETERMINATION OF SPLIT AIR CONDITIONERS BY THE AIR-ENTHALPY METHODOLOGY

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Abstract. *The split air conditioner is widely used around the world to control internal temperature of houses and buildings. It is important to understand how to quantify the cooling capacity of this kind of equipment and evaluate carefully the measurement uncertainties involved in the process. This paper describes meticulously the Air-Enthalpy methodology and, in particular, the equations to calculate the uncertainty of measurement when this methodology is utilized. To evaluate the effects that the independent variables errors have on the dependent variables the Error Propagation Theory is applied. For this experiment the relative measurement uncertainty for the cooling capacity calculated was 1.6%.*

Keywords: *split air conditioner, cooling capacity, measurement uncertainty.*

1. INTRODUCTION

Ambient temperature control is an important subject that has been studied exhaustively over the years by scientist all over the world. It is very important to measure adequately the amount of energy that is necessary to keep our houses and buildings in a comfortable internal temperature zone. One of the most common equipment used today to do that is the split air conditioner.

The split air conditioner employs the electric powered refrigeration method of mechanical vapor-compression illustrated in Fig. 1 (ASHRAE Fundamentals, 2009) to reach the desired environmental condition in the indoor room. This method uses a refrigerant fluid that circuits around the system transporting heat from the different zones.

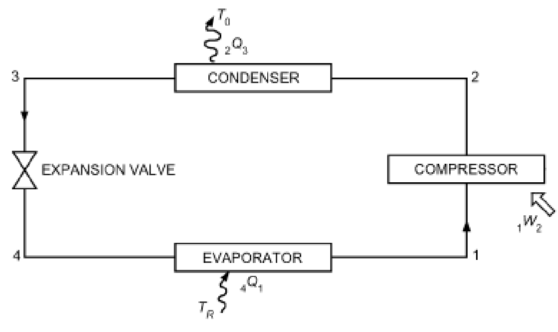

Figure 1 - Mechanical Vapor-compression cycle.

The amount of cooling provided by an air conditioner unit mainly depends on the outdoor and indoor temperature and of the conditioned environment heat and humidity generation.

In order to evaluate the air conditioning system performance during the product development, manufacturers use calorimeters to perform standardized tests. According to ISO 5151:1994, there are three types of calorimeters utilized: psychrometric, calibrated and balanced calorimeter.

In this project, the performance tests were executed on a psychrometric calorimeter that is described in the next section. The mathematical equations to define the cooling capacity by the Air-Enthalpy method are presented, and finally yet most importantly, the measurement uncertainty is evaluated.

2. EXPERIMENTAL APPARATUS

2.1 Split Air Conditioner

The split air conditioner used in this project is a 9000 Btu/h model, 220V/60Hz, operating with R410a as refrigerant fluid. The schematic of the basic components is illustrated in Fig. 2 (Pacheco, 2013). The compressor is a rotating type, the heat exchangers (evaporator and condenser) are of the fin-tube type and the expansion device is a capillary tube. There is a fan assembly powered by an electric motor close to each one of the heat exchangers to generate forced air convection and increase the heat exchange rate. An example of its application is illustrated in Fig. 3 (education, 2016).

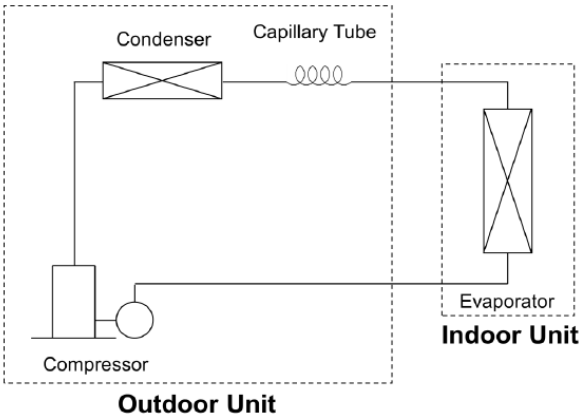

Figure 2 - Schematic of the basic components of a split air conditioner unit.

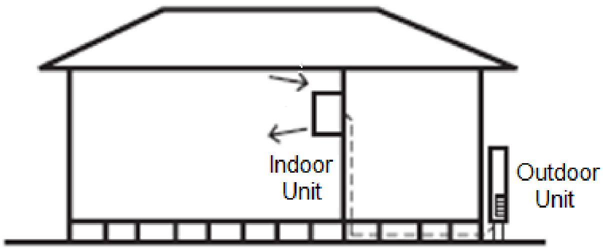

Figure 3 - Split air conditioner application.

2.2 Psychrometric Calorimeter

The psychrometric calorimeter is illustrated in Fig. 4 (Pacheco, 2013) and consists of two rooms that can control inner temperature and relative humidity, creating any desired ambient conditions for air conditioner performance evaluation, both in the indoor-room and in the outdoor-room. There is a standardized Nozzle Chamber inside the indoor-room to measure the airflow rate. It consists of an exhaust fan, a standardized nozzle, a flow straightener and two manometers. This apparatus can also be called Wind Tunnel and is illustrated below, in Fig. 5 (Pacheco, 2013).

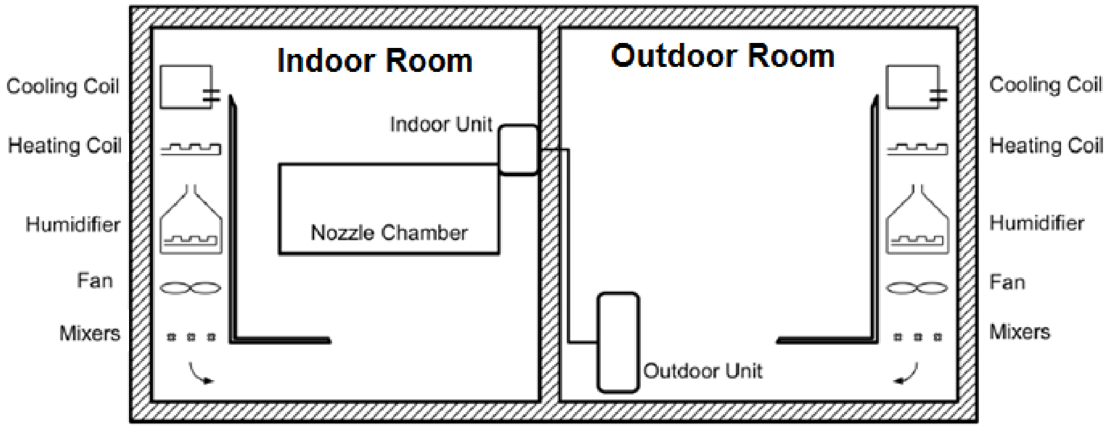

Figure 4 - Illustration of a split air conditioner unit inside a psychrometric calorimeter.

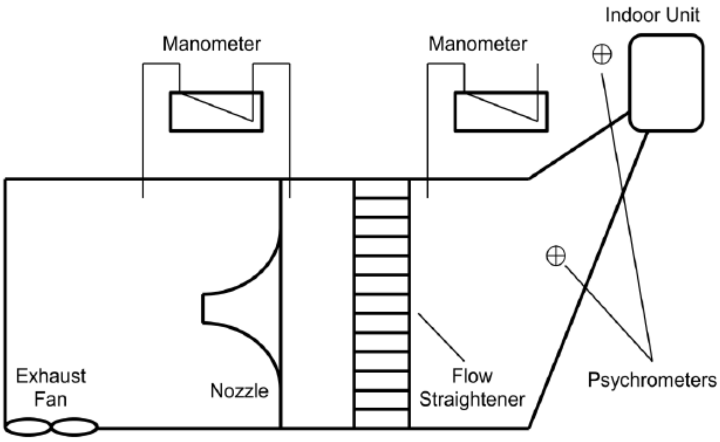

Figure 5 - Nozzle chamber used inside the psychrometric calorimeter.

2.3 Air-Enthaply Methodology

When working with a psychrometric calorimeter, the more convenient way to determine the cooling capacity and energy efficiency of an air conditioner is to use the Air-Enthalpy method, described next.

In the region close to the equipment air suction, a psychrometer measures the dry-bulb and wet-bulb temperatures, defining the psychrometric air state at the evaporator inlet. With these two properties, the specific inlet air enthalpy (h_i) can be calculated. The supply air enters into a nozzle chamber, where another psychrometer measures the dry-bulb and wet-bulb temperatures, defining the psychrometric air state at the evaporator outlet, and now the specific outlet air enthalpy (h_o) can be calculated. A manometer measures the air pressure at this point.

The pressure at the nozzle chamber entrance is maintained equal to the local atmospheric pressure by the exhaust fan so that the nozzle chamber does not interfere on the airflow of the tested air conditioner. After flowing through the flow straightener, the air passes through the standardized nozzle. The second manometer measures the pressure drop caused by the nozzle and the mass flow rate of air ($\dot{m}_{air}$) can be determined. With all these parameters defined, it is possible to determine the total cooling capacity and energy efficiency rate of the product in test.

3. CALCULATION SEQUENCE

When the air-enthalpy method is utilized, the total cooling capacity (Q_C) of the air conditioning equipment under test is derived from the Air-Side energy balance at the evaporator, given by Eq. (1).

$$Q_C = \dot{m}_{air}.(h_i - h_o) \tag{1}$$

The mass flow rate of moist air ($\dot{m}_{air}$) is defined by Eq. (2), depends of the density of the air (ρ_o) leaving the heat exchanger and entering the nozzle chamber and of the volumetric moist airflow rate ($\dot{V}_{air}$).

$$\dot{m}_{air} = \rho_o . \dot{V}_{air} \tag{2}$$

The volumetric airflow rate is determined according to the nozzle chamber methodology described on ANSI/AMCA 210-99:1999 given by Eq. (3).

$$\dot{V}_{air} = \gamma . (C_d . A) . \sqrt{\frac{2.\Delta P}{\rho_o}} \tag{3}$$

Where:

- γ is the air expansion factor.
- C_d is the nozzle discharge coefficient.
- A is the nozzle discharge area.
- ΔP is the static pressure drop across the nozzle.

The nozzle discharge area is calculated by the equation of the area of a circle equation. Equation 4 and 5 define the air expansion factor. The β factor can be considered zero and P_a is the atmospheric pressure measured by the barometer.

$$\gamma = 1 - (0.548 + 0.71.\beta^4).(1 - \alpha) \tag{4}$$

$$\alpha = \frac{P_a - \Delta P}{P_a} \tag{5}$$

To calculate the nozzle discharge coefficient, as shown in Eq. (6), it is necessary to obtain the Reynolds number (R_e). The Reynolds number depends of the mass flow rate of air, as shown in Eq. (7), which depends of the nozzle discharge coefficient. The interdependency between these two parameters requires an iterative calculation.

$$C_d = 0.9986 - \frac{7.006}{\sqrt{R_e}} + \frac{134.6}{R_e} \tag{6}$$

$$R_e = \frac{4.\dot{m}_{air}}{\pi.d.\mu_o} \tag{7}$$

The viscosity of the air (μ_o) is obtained from the linear regression of values from air properties table described by Eq. (8), and depends only of the dry-bulb temperature of the air (T_{dbo}).

$$\mu_o = (17.209 + 0.046.T_{dbo}).10^{-6} \tag{8}$$

The density of the moist air entering the nozzle chamber is calculated using Eq. (9) where R_{da} is the gas constant for dry air, and it is equal to *287.042 J/kgK.*

$$\rho_o = \frac{P_a - 0.3781.p_w}{R_{da}.T_{dbo}} \tag{9}$$

The partial pressure of water vapor present in moist air (p_w) is calculated by the humidity ratio of moist air (W) equation, given below.

$$p_w = \frac{W.P_a}{0.622 + W} \tag{10}$$

From the mass and energy balances of an adiabatic saturation device (Kuehn et al., 1998), it is possible to define the humidity ratio of moist air as shown in Eq. (11).

$$W = \frac{(h_{v,wb} - h_{l,wb}).W_s}{h_{v,db} - h_{l,wb}} - \frac{1.006.(T_{dbo} - T_{wbo})}{h_{v,db} - h_{l,wb}} \tag{11}$$

Where:

- $h_{v,wb}$ is the specific enthalpy of water vapor at wet bulb temperature.
- $h_{v,db}$ is the specific enthalpy of water vapor at dry bulb temperature.
- $h_{l,wb}$ is the specific enthalpy of liquid water at wet bulb temperature.

- W_s is the humidity ratio of moist air at saturation.
- T_{wbo} is the wet bulb temperature of the air.

Considering that the specific heat of liquid water ($c_{p,lw}$) is equal to *4.1868 kJ/kgK,* the specific heat of saturated water vapor ($c_{p,w}$) is equal to *1.86 kJ/kgK* and the specific enthalpy of water vapor at 0°C ($h_{v,0}$) is equal to *2500.89 kJ/kg,* Eq. (12) defines enthalpy of liquid water at wet bulb temperature and Eq. (13) defines enthalpy of water vapor at wet bulb temperature ($h_{v,wb}$).

$$h_{l,wb} = c_{p,lw}.T_{wbo} \tag{12}$$

$$h_{v,wb} = h_{v,0} + c_{p,w}.T_{wbo} \tag{13}$$

The humidity ratio of moist air at saturation (W_s) is obtained through Eq. (14), where the partial pressure of water vapor at saturation (p_{ws}) is calculated by Eq. (15), a regression of properties table of water data, proposed by Hyland and Wexler (1983), which is within 300 ppm of accuracy for the temperature range of 0°C to 200°C, to calculate p_{ws}.

$$W_s = \frac{0.622.p_{ws}}{P_a - p_{ws}} \tag{14}$$

$$\ln(p_{ws}) = C_1.T_{wbo}{}^{-1} + C_2 + C_3.T_{wbo} + C_4.T_{wbo}{}^2 + C_5.T_{wbo}{}^3 + C_6.\ln(T_{wbo}) \tag{15}$$

Where:
- $C_1 = -5.8002206\ E + 03.$
- $C_2 = +1.3914993\ E + 00.$
- $C_3 = -4.8640239\ E - 02.$
- $C_4 = +4.1764768\ E - 05.$
- $C_5 = -1.4452093\ E - 08.$
- $C_6 = +6.5459673\ E + 00.$

Moist air is considered a binary mixture of perfect gases (dry air and water vapor). The amount of water vapor varies from zero (dry air) to a maximum that depends on temperature and pressure of the air. The enthalpy of a mixture of perfect gases is equal the sum of the individual partial enthalpies of each component. Therefore, the specific enthalpy of moist air (h) can be defined by Eq. (16).

$$h = h_{da} + W.h_{v,db} \tag{16}$$

Equation 17 defines specific enthalpy of dry air (h_{da}) considering that specific heat of dry air ($c_{p,da}$) is constant and equals to *1.006 kJ/kgK*. Equation 18 defines specific enthalpy of water vapor at dry bulb temperature ($h_{v,db}$) considering that the specific heat of saturated water vapor ($c_{p,w}$) is equal to *1.86 kJ/kgK* and the specific enthalpy of water vapor at 0°C ($h_{v,0}$) is equal to *2500.89 kJ/kg.*

$$h_{da} = c_{p,da}.T_{dbo} \tag{17}$$

$$h_{v,db} = h_{v,0} + c_{p,w}.T_{dbo} \tag{18}$$

At last, with all this parameters defined and calculated, it becomes possible to obtain the total cooling capacity of the split air-conditioner using equation 1.

4. MEASUREMENT UNCERTAINTY

To calculate the uncertainties of measurements, the error propagation theory developed in J. Taylor (1988) was used. The same methodology used by Pacheco (2013) and Pereira and Mendes (2011) .This methodology is useful to evaluate the effects that the independent variables error have on the dependent variables and helps to evaluate how accurate the results presented in this project really are.

All measured quantities were obtained by using the sensors presented in Tab. 1. The calorimeter used to perform the tests is able to maintain the dry-bulb and wet-bulb temperatures in a range of ±0.1 ºC from the set temperature. Thus, this variation was added with the uncertainty of measurement of the resistance temperature detectors (RTD) when any of these parameters are being used in the analysis.

Table 1. Sensors specifications with uncertainties of measurements.

Independent Variable [x_i]	*Measurement Equipment*	*Manufacturer*	*Uncertainty of Measurement [$U(x_i)$]*
Atmospheric pressure (P_a)	Barometer [650-750 mmHg]	Princo	± 4.2 mmHg
Dry-bulb temperature (T_{db}) Wet-bulb temperature (T_{wb})	Resistance Temperature Detectors [0-60 °C]	Consistec	± 0.08 °C
Electrical Power (E_C)	Power Transducer [0-2500 W]	Yokogawa	± 4.0 W
Mass Flow Rate of Air ($\dot{m}_{air}$)	Liquid Column Manometer [0-50 mmH_2O]	Dwyer	± 0.2 mmH_2O
	Nozzle diameter [87.5mm]	Machined	± 0.1 mm

According to J. Taylor (1988), when a parameter to be measured (G) is a dependent variable of an analytical expression and the independent variables (x_1, x_2, …x_n) are obtained by different measuring systems, the measurement total uncertainty of the dependent variable due to the x_i uncertainties is given by equation below.

$$\frac{U_G}{G} = \pm\sqrt{\left|\frac{U_G(x_1)}{G}\right|^2 + \left|\frac{U_G(x_2)}{G}\right|^2 + \left|\frac{U_G(x_3)}{G}\right|^2 + \cdots + \left|\frac{U_G(x_n)}{G}\right|^2} \tag{19}$$

Each addition term of Eq. (19) represents the relative uncertainty measurement related to each parameter x_i, that was calculated using Eq. (20).

$$\frac{U_G(x_i)}{G} = \frac{\partial G}{\partial x_i}.\frac{U(x_i)}{G} \tag{20}$$

When the parameter to be defined is obtained by linear regression of a table of properties, the uncertainty of this regression is given by the standard deviation (S) of the linear regression, given by Eq. (21).

$$S = U_x = \pm\sqrt{\frac{\sum_{i=1}^{n}(x_i - x_t)^2}{n-1}} \tag{21}$$

Where:

- x_i is the i^{th} value of parameter x.
- x_t is the value of parameter x obtained in the table of properties.
- n is the amount of different values of x used for the linear regression.

The cooling capacity calculation process incorporates many parameters and equations, involving moist air properties, as described before, so, in order to summarize, the calculation step by step of the partial derivatives will not be presented is this project. Therefore, applying Eq. (19) and Eq. (20) in Eq. (1), the uncertainty of measurement of the cooling capacity (U_{Q_C}) is given by:

$$\frac{U_{Q_C}}{Q_C} = \pm\sqrt{\left|\frac{U_{\dot{V}_{air}}}{\dot{V}_{air}}\right|^2 + \left|\frac{U_{\rho_o}}{\rho_o}\right|^2 + \left|\frac{U_{h_i}}{h_i - h_o}\right|^2 + \left|\frac{U_{h_o}}{h_o - h_i}\right|^2} \tag{22}$$

Applying Eq. (19) and Eq. (20) in Eq. (3) the uncertainty of measurement of the volumetric airflow ($U_{\dot{V}_{air}}$) is given by:

$$\frac{U_{\dot{V}_{air}}}{\dot{V}_{air}} = \pm\sqrt{\left|\frac{U_\gamma}{\gamma}\right|^2 + \left|\frac{U_{C_d}}{C_d}\right|^2 + \left|\frac{U_A}{A}\right|^2 + + \left|-\frac{U_{\rho_o}}{2.\rho_o}\right|^2 + \left|\frac{U_{\Delta P}}{2.\Delta P}\right|^2} \tag{23}$$

The nozzle discharge coefficient (C_d) uncertainty is 1.2%, according to ANSI/AMCA 210-99:1999 requirements.

Applying Eq. (19) and Eq. (20) to the outlet area of the nozzle equation, the uncertainty of measurement (U_A) of the area is given by:

$$\frac{U_A}{A} = \pm\left|\frac{2.U_d}{d}\right| \tag{24}$$

Applying Eq. (19) and Eq. (20) in Eq. (4) the uncertainty of measurement of the air expansion factor (U_γ) is given by:

$$\frac{U_\gamma}{\gamma} = \pm\left|\frac{0.548.U_\alpha}{1-0.548.(1-\alpha)}\right| \tag{25}$$

Applying Eq. (19) and Eq. (20) in Eq. (5) the uncertainty of measurement of the alpha ratio (U_α) is given by:

$$\frac{U_\alpha}{\alpha} = \pm\sqrt{\left|\frac{\Delta P.U_{P_a}}{P_a.(P_a-\Delta P)}\right|^2 + \left|-\frac{U_{\Delta P}}{P_a-\Delta P}\right|^2} \tag{26}$$

Applying Eq. (19) and Eq. (20) in Eq. (9) the uncertainty of measurement of the density of moist air (U_{ρ_o}) is given by:

$$\frac{U_{\rho_o}}{\rho_o} = \pm\sqrt{\left|\frac{U_{P_a}}{P_a-0.3781.p_w}\right|^2 + \left|\frac{0.3781.U_{p_w}}{P_a-0.3781.p_w}\right|^2 + + \left|\frac{(0.3781.p_w-P_a).U_{T_{dbo}}}{T_{dbo}.(P_a-0.3781.p_w)}\right|^2} \tag{27}$$

Applying Eq. (19) and Eq. (20) in Eq. (10) the uncertainty of measurement of the partial pressure of water vapor (U_{p_w}) is given by:

$$\frac{U_{p_W}}{p_W} = \pm\sqrt{\left|\frac{U_{P_a}}{P_a}\right|^2 + \left|\frac{0.622.U_W}{W.(W+0.622)}\right|^2} \tag{28}$$

Applying Eq. (19) and Eq. (20) in Eq. (11) the uncertainty of measurement of the humidity ratio of moist air (U_W) is given by:

$$\frac{U_W}{W} = \pm\sqrt{\left|\frac{U_W(h_{v,wb})}{W}\right|^2 + \left|\frac{U_W(h_{v,db})}{W}\right|^2 + \left|\frac{U_W(h_{l,wb})}{W}\right|^2 + \left|\frac{U_W(W_s)}{W}\right|^2 + \left|\frac{U_W(T_{dbo})}{W}\right|^2 + \left|\frac{U_W(T_{wbo})}{W}\right|^2} \tag{29}$$

Where each term can be written as:

$$\frac{U_W(h_{v,wb})}{W} = \frac{\partial W}{\partial h_{v,wb}}.\frac{U(h_{v,wb})}{W} = \frac{W_s.U_{h_{v,wb}}}{W.(h_{v,db}-h_{l,wb})} \tag{29a}$$

$$\frac{U_W(h_{v,db})}{W} = \frac{\partial W}{\partial h_{v,db}}.\frac{U(h_{v,db})}{W} = -\frac{[(h_{v,wb}-h_{l,wb}).W_s-1.006.(T_{dbo}-T_{wbo})].U_{h_{v,db}}}{W.(h_{v,db}-h_{l,wb})^2} \tag{29b}$$

$$\frac{U_W(h_{l,wb})}{W} = \frac{\partial W}{\partial h_{l,wb}}.\frac{U(h_{l,wb})}{W} = -\frac{[(h_{v,wb}-h_{v,db}).W_s-1.006.(T_{dbo}-T_{wbo})].U_{h_{l,wb}}}{W.(h_{v,db}-h_{l,wb})^2} \tag{29c}$$

$$\frac{U_W(W_s)}{W} = \frac{\partial W}{\partial W_s}.\frac{U(W_s)}{W} = \frac{(h_{v,wb}-h_{v,db}).U_{W_s}}{W.(h_{v,db}-h_{l,wb})} \tag{29d}$$

$$\frac{U_W(T_{dbo})}{W} = \frac{\partial W}{\partial T_{dbo}}.\frac{U(T_{dbo})}{W} = -\frac{1.006.U_{T_{dbo}}}{W.(h_{v,db}-h_{l,wb})} \tag{29e}$$

$$\frac{U_W(T_{wbo})}{W} = \frac{\partial W}{\partial T_{wbo}}.\frac{U(T_{wbo})}{W} = \frac{1.006.U_{T_{wbo}}}{W.(h_{v,db}-h_{l,wb})} \tag{29f}$$

Applying Eq. (19) and Eq. (20) in Eq. (12) the uncertainty of measurement of the enthalpy of liquid water at wet bulb temperature ($U_{h_{l,wb}}$) is given by:

$$\frac{U_{h_{l,wb}}}{h_{l,wb}} = \pm\sqrt{\left|\frac{4.1868.U_{T_{wbo}}}{h_{l,wb}}\right|^2 + \left|\frac{U_{h_{l,wb}}(reg)}{h_{l,wb}}\right|^2} \quad (30)$$

Applying Eq. (19) and Eq. (20) in Eq. (13) the uncertainty of measurement of the enthalpy of water vapor at wet bulb temperature ($U_{h_{v,wb}}$) is given by:

$$\frac{U_{h_{v,wb}}}{h_{v,wb}} = \pm\sqrt{\left|\frac{1.86.U_{T_{wbo}}}{h_{v,wb}}\right|^2 + \left|\frac{U_{h_{v,wb}}(reg)}{h_{v,wb}}\right|^2} \quad (31)$$

Applying Eq. (19) and Eq. (20) in Eq. (14) the uncertainty of measurement of the humidity ratio of moist air at saturation (U_{W_s}) is given by:

$$\frac{U_{W_s}}{W_s} = \pm\sqrt{\left|-\frac{U_{P_a}}{P_a - p_{ws}}\right|^2 + \left|\frac{P_a.U_{p_{ws}}}{p_{ws}.(P_a - p_{ws})}\right|^2} \quad (32)$$

Applying Eq. (19) and Eq. (20) in Eq. (15) the uncertainty of measurement of the partial pressure of water vapor at saturation ($U_{p_{ws}}$) is given by:

$$\frac{U_{p_{ws}}}{p_{ws}} = \pm\sqrt{\left|-\frac{C_1 U_{T_{wbo}}}{T_{wbo}{}^2} + C_3 U_{T_{wbo}} + 2C_4 U_{T_{wbo}} + 3C_5 T_{wbo}{}^2 U_{T_{wbo}} + \frac{C_6 U_{T_{wbo}}}{T_{wbo}}\right|^2 + \left|\frac{U_{p_{ws}}(reg)}{p_{ws}}\right|^2} \quad (33)$$

Applying Eq. (19) and Eq. (20) in Eq. (16) the uncertainty of measurement of the specific enthalpy of moist air (U_h) is given by:

$$\frac{U_h}{h} = \pm\sqrt{\left|\frac{U_{h_{da}}}{h}\right|^2 + \left|\frac{h_{v,db}.U_W}{h}\right|^2 + \left|\frac{W.U_{h_{v,db}}}{h}\right|^2} \quad (34)$$

Applying Eq. (19) and Eq. (20) in Eq. (17) the uncertainty of measurement of the specific enthalpy of dry air ($U_{h_{da}}$) is given by:

$$\frac{U_{h_{da}}}{h_{da}} = \sqrt{\left|\frac{U_{T_{dbo}}}{T_{dbo}}\right|^2 + \left|\frac{U_{h_{da}}(reg)}{h_{da}}\right|^2} \quad (35)$$

Applying Eq. (19) and Eq. (20) in Eq. (18) the uncertainty of measurement of the specific enthalpy of water vapor at dry-bulb temperature ($U_{h_{v,db}}$) is given by:

$$\frac{U_{h_{v,db}}}{h_{v,db}} = \pm\sqrt{\left|\frac{1.86.U_{T_{dbo}}}{h_{v,db}}\right|^2 + \left|\frac{U_{h_{v,db}}(reg)}{h_{v,db}}\right|^2} \quad (36)$$

All uncertainties related to linear regression of properties table,$U(reg)$, were calculated applying Eq. (21) for each case separately.

5. RESULTS

Following all the equations described in section 3 and 4, with the inputs shown in Tab. 2 it was calculated the parameters and their uncertainties of measurements presented in Tab. 3. The final relative uncertainty for the total cooling capacity of the split air conditioner tested in this work was 1.6 %. The ISO standard requests a value lower than 3% for the uncertainty of measurement of the cooling capacity, so this methodology complies.

Table 2. Data input for the experimental test realized in the split air conditioner.

	xi	U(xi)
T_{dbi} [°C]	26.79	± 0.08
T_{wbi} [°C]	19.42	± 0.08
T_{dbo} [°C]	13.84	± 0.08
T_{wbo} [°C]	12.74	± 0.08
d [mm]	87.5	± 0.1
ΔP [mmH2O]	27.0	± 0.2
Pa [mmHg]	693.9	± 4.2

Table 3. Values calculated using all the equations described in this paper.

	xi	U(xi)
$\dot{V}_{air}$ [m³/s]	0.128	± 0.002
ρ_o [kg/m³]	1.117	± 0.009
h_i [kJ/kg]	58.74	± 0.12
h_o [kJ/kg]	38.18	± 0.12
Q_C [W]	2939	± 47

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