PONTIFICAL CATHOLIC UNIVERSITY OF PARANA

POLITECHNIC SCHOOL

MECHANICAL ENGINEERING PROGRAM

RICARDO BAASCH PACHECO

EXPERIMENTAL EVALUATION AND EMPIRICAL PERFORMANCE CORRELATIONS OF A DUAL-SPLIT AIR CONDITIONING SYSTEM

CURITIBA

2013

RICARDO BAASCH PACHECO

EXPERIMENTAL EVALUATION AND EMPIRICAL PERFORMANCE CORRELATIONS OF A DUAL-SPLIT AIR CONDITIONING SYSTEM

Dissertation presented to the graduate program in Mechanical Engineering at the Pontifical Catholic University of Parana as a requirement to obtaining the Master's degree in Mechanical Engineering.

Advisor: Prof. Dr. Nathan Mendes

CURITIBA

2013

RICARDO BAASCH PACHECO

EXPERIMENTAL EVALUATION AND EMPIRICAL PERFORMANCE CORRELATIONS OF A DUAL-SPLIT AIR CONDITIONING SYSTEM

Dissertation presented to the examiners of the graduate program in Mechanical Engineering from Pontifical Catholic University of Parana.

President:

____

Prof. Dr. Nathan Mendes (advisor)
PUC/PR

____

Prof. Dr. Luís Mauro Moura
PUC/PR

____

Prof. Dr. Viviana Mariani
PUC/PR

____

Prof. Dr. Arlindo Tribess
USP

Curitiba, 2013

ABSTRACT

A dual-split air conditioning system is an air conditioner with two indoor units sharing the same outdoor unit. This paper presents a proposal for the performance test standard and test procedure of dual-split air conditioners to determine its cooling capacity and energy efficiency ratio. The air enthalpy method was used. Additionally, the air properties of indoor and outdoor test rooms were varied so that the capacity and energy efficiency of the unit was determined on a wide range of climate conditions. An uncertainty analysis was performed to validate the procedure. A regression of these test results in a biquadratic equation were performed to determine the off-design performance of the dual-split. The equation was evaluated in two different forms: one using the procedure suggested by ASHRAE (2009), and one using the enthalpy instead of the wet-bulb temperature of the indoor air. The result of the regressions were accurate in both forms of the equations, so this can be implemented into energy simulation software, such as Domus and EnergyPlus to improve their HVAC database.

Keywords dual-split air conditioner, performance test, uncertainty analysis, performance correlation.

This dissertation is dedicated to my family; specially my wife, Waleska, and my daughter, Letícia, for the great support and understanding.

ACKNOWLEDGMENTS

I wish to thank Nathan Mendes for the competent orientation of this project; my friends from Electrolux do Brasil S.A. Gustavo Cherem Corte Pereira, Carlos Augusto Richter do Nascimento, Fabrizio Bolelli and Jefferson Katsumi Bansho for the support and advice during the experimental tests; my lecturers and colleagues at PUC/PR; Electrolux of Brasil S.A. for the finacial support and for the equipment used on this project and my family.

TABLE OF CONTENTS

Abstract i

List of Figures vi

List of Tables vii

Symbols and Subscriptions viii

1. Introduction 1
   1.1. Objectives 2
   1.2. Literature Review 3
2. Air Conditioning Systems 6
   2.1. Window type Air Conditioner 7
   2.2. Single Split Air Conditioner 8
   2.3. Dual-Split Air Conditioner 9
3. Performance Testing Methods 11
   3.1. Calorimeter Method 13
   3.2. Air-Enthalpy Method 14
   3.3. Multi-Split Test Method 16
4. Performance Calculation 18
   4.1. Total Cooling Capacity 18
      4.1.1. Volumetric Air Flow Rate 19
      4.1.2. Enthalpy of Moist Air 22
   4.2. Sensible Cooling Capacity 22
   4.3. Energy Efficiency Ratio 23
5. Uncertainties of Measurements 24

5.1. Total Cooling Capacity 26

5.2. Sensible Cooling Capacity 32

5.3. Energy Efficiency Ratio 33

6. Tests Results and Analysis 34

7. Performance Characteristics Curve 42

7.1. Total Cooling Capacity 43

7.2. Sensible Cooling Capacity 45

7.3. Energy Efficiency Ratio 47

7.4. Calculation Analysis 48

8. Conclusion 50

8.1. Recommendations for Future Work 52

References 54

LIST OF FIGURES

Figure 2.1 – Refrigeration Circuit of a Vapor Compression Air Conditioner 7

Figure 2.2 – Refrigeration Circuit of a Window Type Air Conditioner 8

Figure 2.3 – Refrigeration Circuit of a Single-Split Air Conditioner.................................. 9

Figure 2.4 – Refrigeration Circuit of a Dual-Split Air Conditioner 10

Figure 3.1 – Calibrated Calorimeter Schematics 12

Figure 3.2 – Balanced Calorimeter Schematics 12

Figure 3.3 – Air-Enthalpy Method Calorimeter 14

Figure 3.4 – Nozzle Chamber of the Air-Enthalpy Method Calorimeter......................... 15

LIST OF TABLES

Table 3.1 – Performance Test Conditions 17

Table 5.1 – Uncertainties of Measurement Devices 25

Table 6.1 – Test Results 1 35

Table 6.2 – Test Results 2 36

Table 6.3 – Test Results 3 37

Table 6.4 – Test Results 4 37

Table 6.5 – Test Results 5 39

Table 6.6 – Test Results 6 40

Table 6.7 – Test Results 7 41

Table 7.1 – Constants of Total Cooling Capacity of Unit A.................... 43

Table 7.2 – Results of Total Cooling Capacity of Unit A.................... 43

Table 7.3 – Constants of Total Cooling Capacity of Unit B.................... 44

Table 7.4 – Results of Total Cooling Capacity of Unit B.................... 44

Table 7.5 – Constants of Sensible Cooling Capacity of Unit A.................... 45

Table 7.6 – Results of Sensible Cooling Capacity of Unit A.................... 45

Table 7.7 – Constants of Sensible Cooling Capacity of Unit B.................... 46

Table 7.8 – Results of Sensible Cooling Capacity of Unit B.................... 46

Table 7.9 – Constants of EER of Unit A 47

Table 7.10 – Results of EER of Unit A 47

Table 7.11 – Constants of EER of Unit B 48

Table 7.12 – Results of EER of Unit B 48

SYMBOLS AND SUBSCRIPTIONS

Symbol	Description	Unit
$a_0 \dots a_5$	Coefficients of equations (7.1) and (7.2)	-
A	Nozzle discharge area	m^2
$ABNT$	Brazilian Association for Technical Standards (from *Associação Brasileira de Normas Técnicas*)	-
$ANSI$	American National Standards Institute	-
$ASHRAE$	American Society of Heating, Refrigerating and Air-Conditioning Engineers	-
$C_1 \dots C_6$	Coefficients of equation (4.15)	-
C_d	Nozzle discharge coefficient	-
c_p	Specific heat	kJ/(kg.K)
D	Diameter of the nozzle	m
$EER\ (COP)$	Energy Efficiency Ratio (Coefficient of Performance)	-
h	Specific enthalpy	kJ/kg
ISO	International Organization for Standardization	
$\dot{m}_w$	Water flow rate condensed by the equipment	kg/s
p	Atmospheric pressure	kPa
p_d	Static pressure drop across the nozzle	kPa
p_{da}	Partial pressure of dry air in moist air	kPa
p_w	Partial pressure of water vapor in moist air	kPa
$\dot{P}_T$	Power input to the indoor-side compartment	W
NBR	Brazilian Standard (from *Norma Brasileira*)	
$\dot{Q}_c$	Total cooling capacity of the air conditioner	W
$\dot{Q}_k$	Heat leakage into the indoor compartment from the outdoor compartment	W
$\dot{Q}_l$	Heat leakage into the indoor compartment from the other walls of the test room	W
$\dot{Q}_s$	Sensible cooling capacity of the air conditioner	W
R	Gas constant	kJ/(kg.K)
Re	Reynolds number	-

RTD	Resistance Temperature Detectors	-
T	Temperature	K (ºC)
u	Uncertainty of measurement	-
$\dot{V}_a$	Volumetric air flow rate through the product	m^3/s
$\dot{W}$	Work performed by the equipment	W
W	Humidity ratio of moist air	kg_a/kg_w
α	Alpha ratio of the nozzle chamber	-
β	Beta ratio of the nozzle chamber	-
γ	Nozzle expansion factor	-
μ	Viscosity	Pa.s
ρ	Density	kg/m^3

Subscript	**Description**
a	Moist air
A	Unit A
B	Unit B
da	Dry air
db	Dry-bulb
l	Liquid water
o	Outdoor chamber
s	Saturation
v	Water Vapor
w	Water
wb	Wet-bulb
0	At 0ºC
1	Air entering the product / of indoor chamber
2	Air leaving the product / of nozzle inlet

1. INTRODUCTION

Being a country of hot and tropical weather on most of its territory, there is, in Brazil, a strong need to control the temperature of indoor areas to enhance their thermal comfort. To perform such task, the most commonly used equipment is the air conditioners. For residential and light commercial applications, the most commonly used air conditioners are the window and the split types.

The window type air conditioners are built in a single body that contains all the components of the product. Because of its constructive characteristics, normally, these air conditioners have relatively low energy efficiency and are considerably noisy due to compressor operation with inefficient noise insulation. Nevertheless, they are very popular for their cost and ease of installation. The split air conditioners are composed of an indoor unit that contains the evaporator and an outdoor unit that contains the compressor, the condenser and the expansion device (although the expansion device can be found on the indoor unit of some products). The main differential of these products is the low noise level of operation, since the compressor and the expansion device are located in the outdoor unit, and far from the room where the equipment is installed. The popularity of these equipments is rising rapidly in Brazil due to the lowering prices of this category.

Within the splits category, there is a specific line of products that is commercially called multi-splits, which are composed of two, three or four indoor units that share a single outdoor unit. These products are commercially called dual-splits, tri-splits or quad-splits, respectively. Normally, despite the fact that the indoor units share the same outdoor unit, there is one refrigeration cycle for each of them, so these units can be considered independent from each other. As there is several indoor units for only one outdoor unit, there is a great demand for those products, especially in apartment buildings and commercial centers where the space for installing the outdoor unit is very limited.

Both window and split type air conditioners are heat pumps. In other words, they operate by mechanical vapor-compression of a refrigerant fluid. The main advantage of a heat pump, when compared to other methods of air conditioning, is the high COP (coefficient of performance) – also known as EER (energy efficiency ratio). In normal operating conditions, according to international standards, they are capable of removing more than three times more energy from the air than the energy consumed by them during the refrigeration cycle.

The capacity and energy efficiency ratio of window type and single split type air conditioners are measured and calculated following the procedures described in several standards available worldwide. Some examples are ISO 5151 (1994), ANSI/AHAM RAC-1 (1992) and NBR 5882 (1983). None of these standards describes such procedure for multi-split air conditioners, thus it is necessary to create a procedure to standardize the test.

In addition, these standards can determine the capacity and EER of the equipment in one specific psychrometric condition of the air. Often, it is desired to determine the capacity and EER of the equipment in a wide range of psychrometric conditions, especially when working with building thermal and energy simulation software.

1.1. OBJECTIVES

The main objective of this dissertation is the development of an experimental procedure to determine the cooling capacity and energy efficiency ratio of a dual-split air conditioner. As there is no standard test for this product, this procedure is developed using the air-enthalpy method described on standards used for window type and split type air conditioners, such as ISO 5151 (1994).

In addition, the calculation method is described to demonstrate the main differences in calculation to obtain the cooling capacity and EER for a single split unit and a dual-split unit. With the calculation presented, the uncertainty analysis of the calculation is also presented and compared to that of a single split unit.

One dual-split air conditioner was tested according to the procedure developed and the test results are used as example of the method described to prove its accuracy.

For last, the off-design operation characteristics of the equipment are calculated. These characteristics are calculated by two different methods; both of them consisting in a regression of the test results into a bi-quadratic function that describes the cooling capacity and EER of the product. The first method is the one recommended by ASHRAE (2009), which calculates the equipment performance in terms of the outdoor dry-bulb temperature and the indoor wet-bulb temperatures. The second method is based on the ASHRAE method, but replaces the variable of indoor wet-bulb temperature by the indoor enthalpy of the air. The accuracy of both methods is compared. The functions determined can be implemented into building energy simulation programs.

1.2. LITERATURE REVIEW

Multi-split air conditioners are growing in popularity due to their flexibility for installation on places with few available spaces for the outdoor unit. Despite this rapid growth in demand, there is no standard test procedure to determine the capacity and energy efficiency ratio of such equipment.

The most commonly used test standard for non-ducted air-conditioners is ISO 5151 (1994) that describes the standard conditions and test methods to be applied for rating a non-ducted air conditioner and it is accepted worldwide. However, this standard *"is limited to systems utilizing a single refrigeration circuit and having one evaporator and one condenser"* (ISO 5151), so it does not describe a test procedure for multi-split air conditioners.

In the United States, the ruling standard for room air conditioners is ANSI/AHAM RAC-1 (1992). Although it describes performance tests for room air conditioners in general, all the test procedures and conditions are based on ASHRAE

16 (1983). In ASHRAE 16, there is no reference for multi-splits air conditioners and the installation procedures are limited to systems with one evaporator.

In Brazil, there are two national standards regarding the test and rating conditions of domestic air conditioners. While NBR 5858 describes the conditions for determining the performance and stablishes the minimum requirements of quality and capacity of domestic air conditioners, NBR 5882 describes the test procedures to be used when determining the characteristics of the air conditioners according to NBR 5858. The test procedures described on NBR 5882 are very similar to the ones described on ISO 5151, and there is no information about the application of this standard on multi-split air conditoners.

Little literature can be found on performance testing of residential multi-split air conditioners. Most of the studies performed on this subject are related to the application of inverter technology to multi-splits, or the performance optimization of these systems, using the concept of variable refrigerant flow (VRF) through the indoor units, such as Park *et al.* (2001), Hu and Yang (2004), Zhang *et al.* (2011) and Tu *et al.* (2011). The VRF concept is determined by the application of a digital scroll compressor and electronic expansion valves.

Park *et al.* (2001) conducted an analysis on the performance of a dual-split inverter air conditioner equipped with dedicated electronic expansion valves for each indoor unit. All the performance results were obtained through modeling simulation of the system and compared to the results of an existing model of air-to air heat pumps. Thus, there was no experimental procedure to be followed.

Hu and Yang (2004) developed and tested the performance of a multi-split air conditioner with a digital scroll compressor and five indoor units using electronic expansion valves to control the refrigerant flow for each of the indoor units. The testing method used is the air-enthalpy method and the set temperatures and relative humidities on indoor and outdoor sides of the testing rooms are according to ASHRAE standards. However, the test procedure used to integrate the performance results of each indoor unit is not detailed and the overall system capacity and energy efficiency ratio is shown without explanations on this procedure.

Zhang *et al.* (2011) also studied the performance of a multi-split air conditioning system, with a digital variable compressor, which is turned on and off by PWM (pulse width modulation) valve, and electronic expansion valves for each of the four indoor units. The testing method used is also the air-enthalpy method, but the tests are conducted on non-insulated rooms that are not equipped with temperature and humidity control devices. On this test, each indoor unit was installed on a separate room, with different dimensions from one another. In this way, it is concluded that the test method is not precise enough for the purpose of this work.

Tu *et al.* (2011) studied a VRF system with high cooling capacity. Although the air-enthalpy method was chosen for the performance tests, there are no further details on the testing procedure.

2. AIR CONDITIONING SYSTEMS

The refrigeration effect can be achieved by several methods. Mechanical vapor-compression and absorption are the most commonly used of them.

The absorption refrigeration is described below by ASHRAE (2009):

> An absorption cycle is a heat-activated thermal cycle. It exchanges only thermal energy with its surroundings; no appreciable mechanical energy is exchanged. Furthermore, no appreciable conversion of heat to work or work to heat occurs in the cycle.
>
> Absorption cycles are used in applications where one or more of the exchanges of heat with the surroundings is the useful product (e.g., refrigeration, air conditioning, and heat pumping). The two great advantages of this type of cycle in comparison to other cycles with similar product are:

The absorption cycle is used in refrigeration applications when a heat source is available in the proximity of the equipment to be installed. Its main advantages are:

• It is not necessary to have large mechanical equipment, as most of the energy input on the cycle is thermal;

• It is possible to use any source of heat, including waste heat from other equipment.

The mechanical vapor compression cycle is described below by Kuehn *et al.* (1998):

> Cooling is accomplished by evaporation of a liquid refrigerant under reduced pressure and temperature. The saturation temperature of the vapor is then elevated by mechanical compression, allowing the vapor to be condensed by heat rejection to ordinary cooling water or atmospheric air. The relatively high-pressure liquid is then expanded to the heat-exchanger, where evaporation occurs. The expansion process is usually accomplished by throttling through a valve.

The refrigeration cycle of a single-stage mechanical vapor-compression equipment is shown in Figure 2.1:

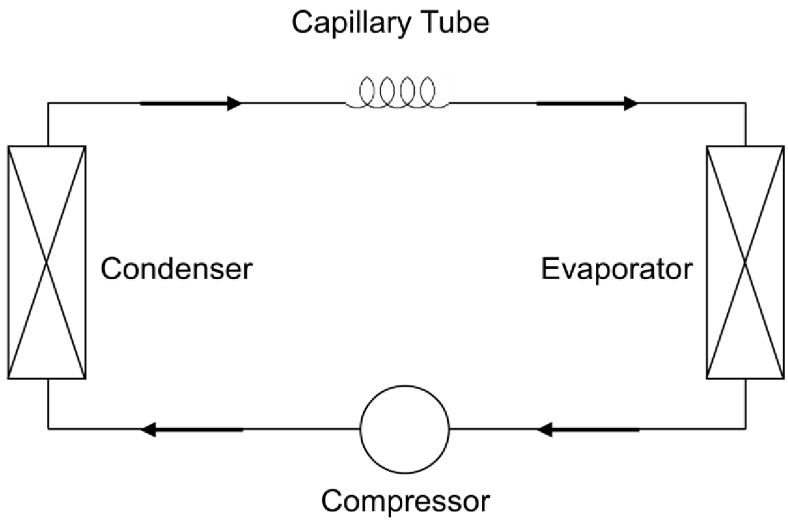

Figure 2.1 – Refrigeration Circuit of a Vapor Compression Air Conditioner

As the application of an absorption air-conditioner is limited by the initial cost and the need of thermal energy availability, most of the residential air conditioners, such as window type and split type, operate with mechanical vapor compression.

2.1. WINDOW TYPE AIR CONDITIONER

Nowadays, the most common air conditioner type in Brazil is the window type. This system is comprised of one single body that contains all the elements of the refrigeration circuit. Its popularity in Brazil is given by the low equipment cost and the ease of installation, as there is no need of special equipment to install it.

As the entire unit is built in a single body, the heat exchangers size is limited, causing the equipment to have a low energy efficiency level when compared to a split air conditioner. Also because of the single body construction – the compressor is located very close to the room to be acclimatized – and the poor noise insulation inside the product, the window type air conditioner often present high noise levels.

Figure 2.2 shows the structure of a window type air conditioner.

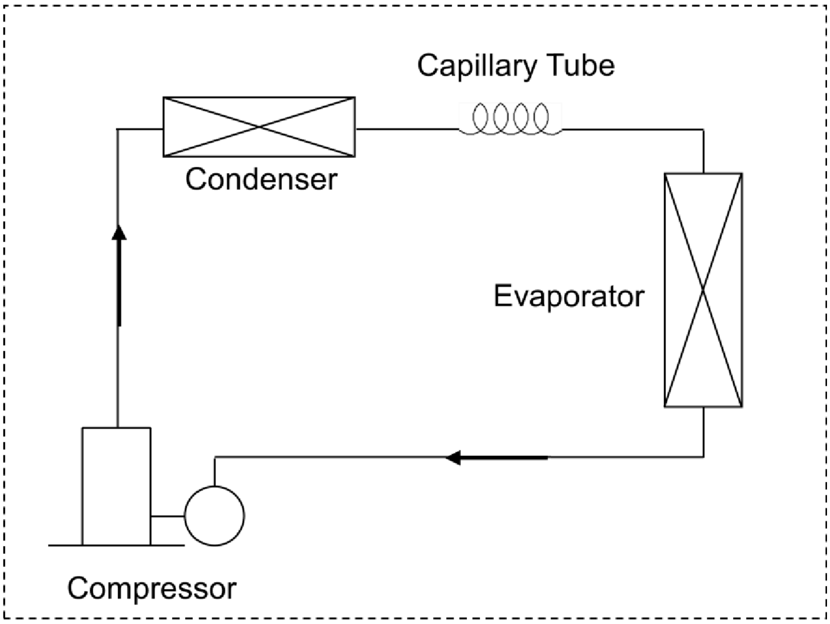

Figure 2.2 – Refrigeration Circuit of a Window Type Air Conditioner

2.2. SINGLE SPLIT AIR CONDITIONER

Differently from the window-type air conditioners, a household split air conditioner is composed of two or more separate units. One of the units contains the condenser, the compressor and the throttling device, which normally is a capillary tube. In order to dissipate the heat from the condenser, this unit is installed outside the environment to be refrigerated, often on an outdoor environment. For this reason it is called the Outdoor Unit (although it can also be referred to as the condenser unit).

The other unit is installed inside the environment to be refrigerated, thus it is called Indoor Unit (although it can also be referred to as the evaporator unit). The indoor unit contains the evaporator of the system. The refrigeration circuit of a single-split type air conditioner can be seen in Figure 2.3.

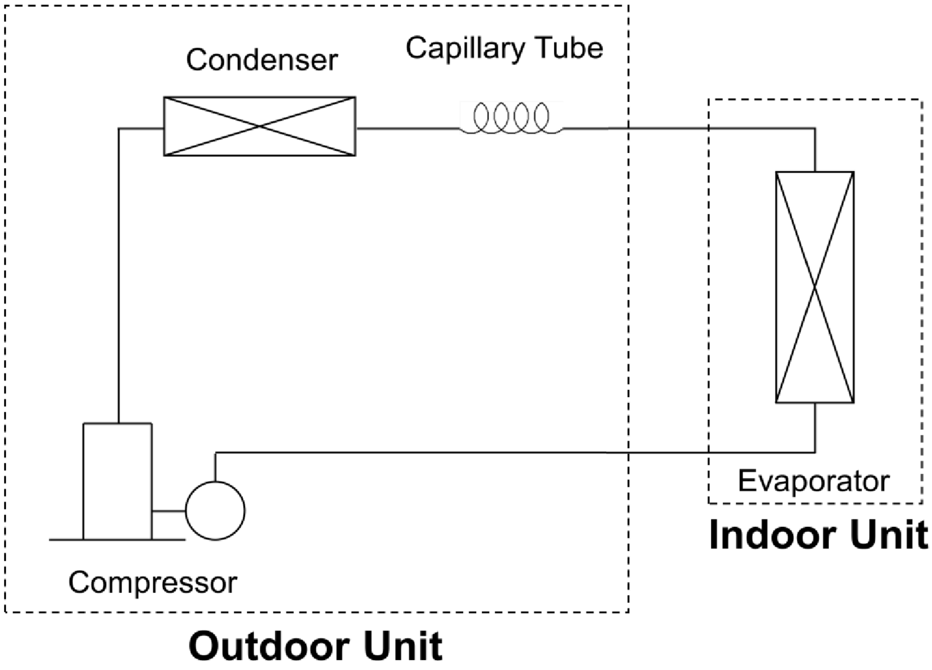

Figure 2.3 – Refrigeration Circuit of a Single-Split Air Conditioner

2.3. DUAL-SPLIT AIR CONDITIONER

A dual-split air conditioner is a commercial name for a unit comprised of two split type air conditioning systems that share one outdoor unit. This outdoor unit differs from those of single-split air conditioners for having two compressors, thus two independent refrigerant circuits.

Each one of the units behaves as single-split type air conditioner. As the condenser, which is air-cooled, has two refrigerant circuits that are isolated from each other, the operation of both units is independent. Their performance, however, is influenced by one another, but this is not the subject of this study.

Figure 2.4 shows the refrigeration circuit a dual-split air conditioner.

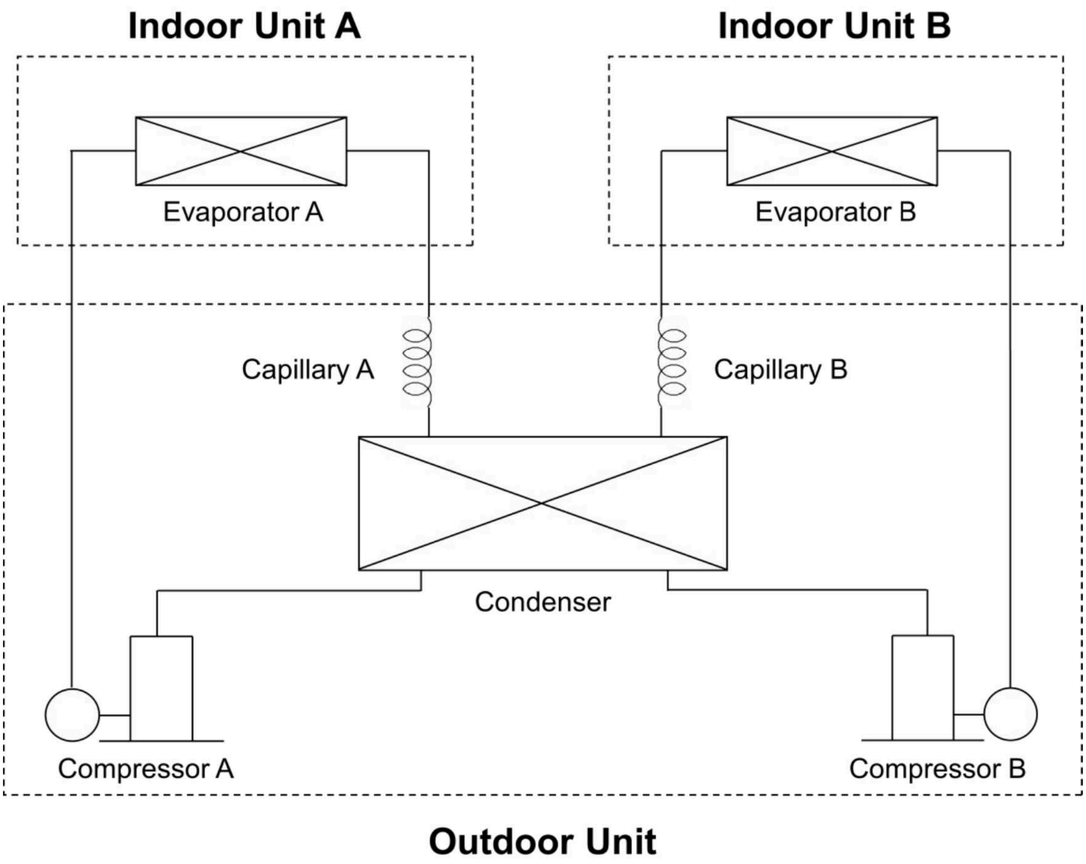

Figure 2.4 – Refrigeration Circuit of a Dual-Split Air Conditioner

The equipment used for this experiment is a residential dual-split air conditioner, 220V~/60Hz, operating with R-22 refrigerant fluid with identical refrigeration systems for each indoor unit. Each system has a compressor of the rotary type, a throttling device – which is a capillary tube – and an evaporator of the fin-tube type. Despite the equipment has only one condenser, also of the fin-tube type, the condenser has two circuits for the refrigerant fluid, one for each system.

Next to each heat exchanger, there is a fan assembly powered by an electric motor to cause forced convection and increase the heat exchange.

3. PERFORMANCE TESTING METHODS

There are several standards that establish procedures and test conditions for measuring and calculating the performance of domestic air conditioners, such as ISO5151 (1994), ANSI/AHAM RAC-1 (1992) and NBR5882 (1983). However, the scope of these standards is limited to “systems utilizing a single refrigeration circuit and having one evaporator and one condenser” (ISO5151). There is no standard available that describe the testing procedure for multi-split air conditioner, thus it is necessary to develop a procedure to standardize the test.

The performance tests of an air conditioner are performed in test rooms called calorimeters. The calorimeter is a test room with two insulated compartments that simulate the conditions of the air on the indoor and outdoor environments by controlling its psychrometric properties, such as dry-bulb temperature and relative humidity. These parameters are controlled through a cooling coil and a heating coil as well as a humidifier device on each side of the calorimeter. There are two types of calorimeter that can perform these tests: the Calibrated Calorimeter and the Balanced Calorimeter.

The overall structure of both types of calorimeter is the same. The calibrated calorimeter presents inside each compartment a cooling coil which is responsible for cooling and dehumidification; a heating coil; a humidification device and a fan assembly to allow the temperature and humidity to be homogeneous throughout the compartment. A schematic representation of the calibrated calorimeter is shown in Figure 3.1.

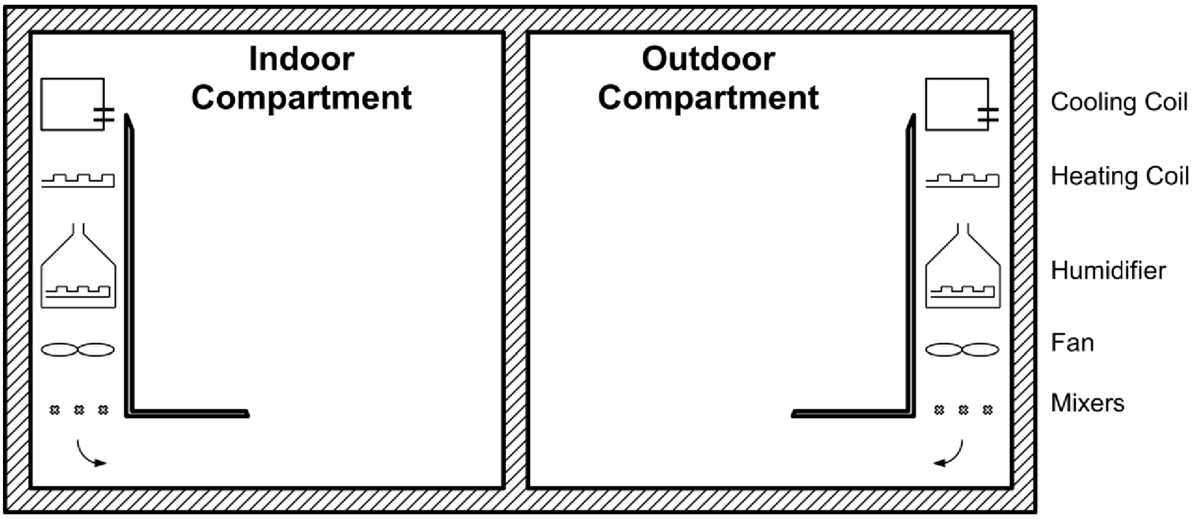

Figure 3.1 – Calibrated Calorimeter Schematics

The structure of a balanced calorimeter is similar to that of a calibrated calorimeter. The difference between these two test rooms is that a balanced calorimeter has a controlled-temperature air space around each compartment. Figure 3.2 shows the schematic representation of a balanced calorimeter.

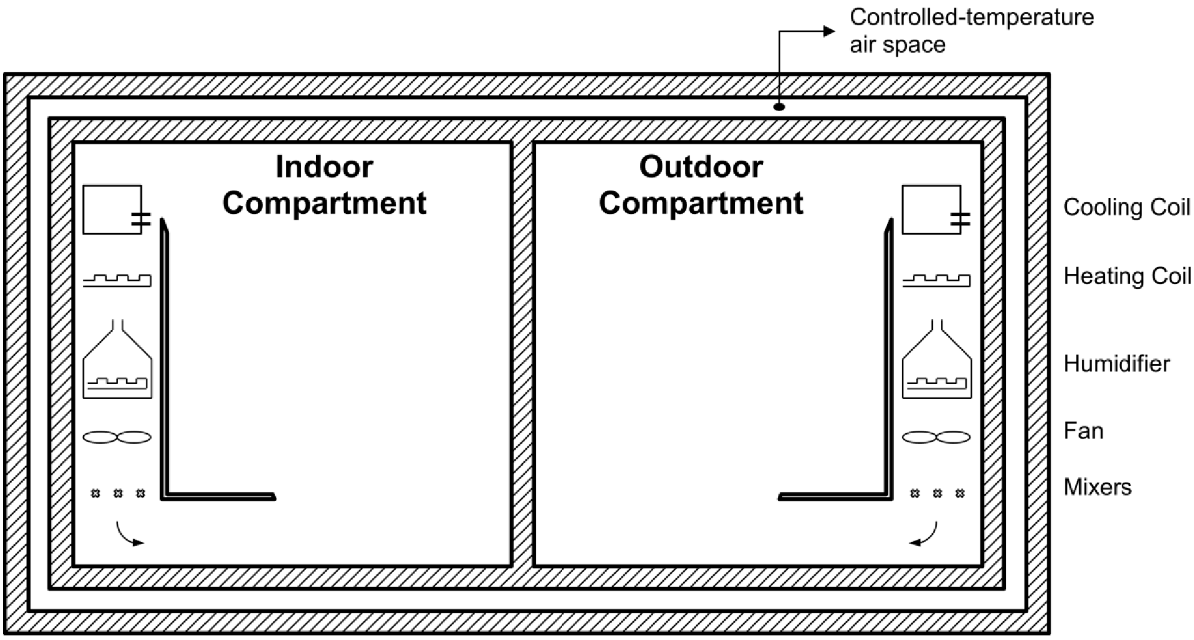

Figure 3.2 – Balanced Calorimeter Schematics

According to ISO5151 (1994), there are two methods to determine the performance of non-ducted air conditioners (which covers window and split air conditioners): the Calorimeter method and the Air-Enthalpy method.

3.1. CALORIMETER METHOD

The calorimeter method can determine the capacity of a product by "balancing the cooling and dehumidifying effects with measured heat and water inputs" (ISO5151). This energy balance is determined by the equation below:

$$\dot{Q}_c = \sum \dot{P}_T + (h_{w1} - h_{w2}).\dot{m}_w + \dot{Q}_k + \dot{Q}_l \tag{3.1}$$

Where:

- $\dot{Q}_c$ is the total cooling capacity of the equipment, in Watts;
- $\sum \dot{P}_T$ is the sum of all power inputs to the indoor-side compartment, in Watts;
- h_{w1} is the specific enthalpy of water or steam introduced to the indoor compartment, in kJ/kg;
- h_{w2} is the specific enthalpy of condensed water that leaves the indoor compartment, in kJ/kg;
- $\dot{m}_w$ is the rate that water is condensed by the equipment during the test, in kg/s;
- $\dot{Q}_k$ is the heat leakage into the indoor compartment from the outdoor compartment, in Watts;
- $\dot{Q}_l$ is the heat leakage into the indoor compartment from the other walls of the test room, in Watts.

This method can be used in any of the calorimeter types described above. The use of the balanced calorimeter is more convenient though, as the heat leakage through the walls ($\dot{Q}_l$) can be neglected.

3.2. AIR-ENTHALPY METHOD

Differently from the calorimeter method, this method uses the psychrometric properties of moist air to determine the cooling capacity of the equipment being tested.

The calorimeter used in this method is similar to a calibrated calorimeter described above. There are two compartments that simulate the temperature and humidity of the air on indoor and outdoor sides. Both sides are equipped with cooling and heating coils and a humidifier device that can maintain the air properties in a controlled level. In addition, the indoor compartment has an air-flow measuring apparatus, where the air properties at evaporator outlet and volumetric air flow rate are measured. Figure 3.3 shows a calorimeter of the air-enthalpy test method.

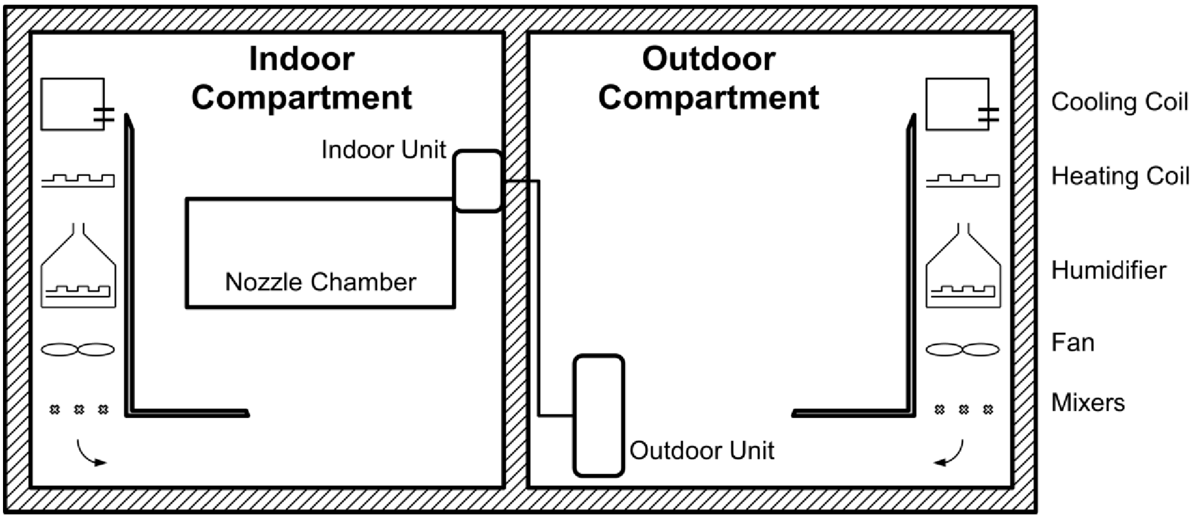

Figure 3.3 – Air-Enthalpy Method Calorimeter

The air-flow measuring apparatus used in this test is a nozzle chamber is equipped with two manometers, one flow straightener, one nozzle, and an exhaust fan. Figure 3.4 shows the nozzle chamber schematics.

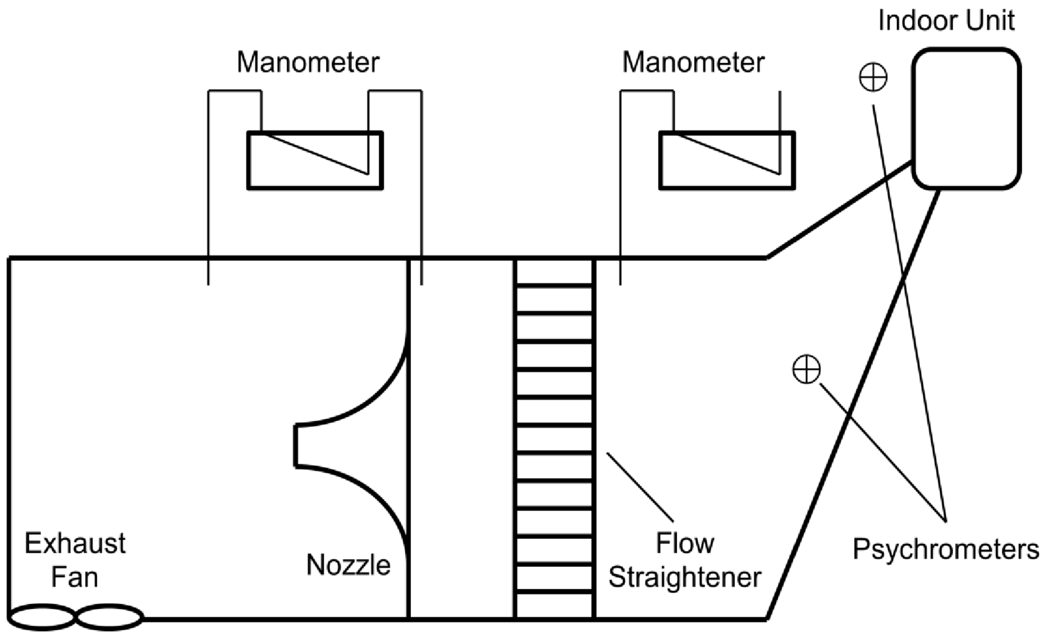

Figure 3.4 – Nozzle Chamber of the Air-Enthalpy Method Calorimeter

The method consists in the following procedure:

In the indoor compartment of the calorimeter, the dry-bulb and wet-bulb temperatures of the ambient air are measured by a psychrometer installed at the inlet of the indoor unit. With these two properties, the specific enthalpy of the air can be determined. Afterwards, the discharge air is blown into a wind tunnel, where another psychrometer measures the dry-bulb and wet-bulb temperatures of the air, hence its specific enthalpy. A manometer measures the air pressure at this point. The air pressure is maintained equal to the local atmospheric pressure by the exhaust fan installed at the end of the chamber. After flowing through a flow straightener, the air passes through the standardized nozzle. The second manometer measures the pressure drop caused by the nozzle. With the calculation shown on chapter 4, the air flow and the product performance parameters are determined.

3.3. MULTI-SPLIT TEST METHOD

The performance tests were performed with both refrigeration cycles operating. As the calorimeter method uses the measurement of the indoor compartment heating coil and the other equipment of the calorimeter to determine the cooling performance of product and the operation of one unit would interfere on the results of the other unit, it would not be possible to determine the cooling capacity and other performance parameters of each refrigeration cycle. On the air enthalpy method, as the air-flow measuring apparatus is connected directly on the discharge of the indoor unit and all the measurements are performed inside this apparatus, the operation of one unit does not interfere on the other unit. For this reason, to test the dual-split unit, the air-enthalpy method was selected.

Ideally, it would be recommended to use two air-flow measuring apparatus, one for each indoor unit, and perform the data acquisition simultaneously for both systems. This would allow a greater precision of air psychrometric properties on the test and avoid the need of precisely setting these conditions of test for each unit. As there was only one nozzle chamber available for testing, the tests performed for this study were performed separately for each indoor unit.

The test condition selected to determine the cooling capacity of the product is the one suggested by ANSI/AHAM (1992):

- Indoor T_{db} | T_{wb}: 26.7ºC | 19.4ºC
- Outdoor T_{db} | T_{wb}: 35.0ºC | 23.9ºC

In addition, six other tests were performed on each refrigeration system, in order to obtain the performance characteristics curves of the product, as described in chapter 7. Table 3.1 shows the test conditions considered for testing.

Table 3.1 – Performance Test Conditions

Test Nº	Indoor Compartment		Outdoor Compartment	
	T_{db}	T_{wb}	T_{db}	T_{wb}
1	26.7 ºC	19.4 ºC	35.0 ºC	23.9 ºC
2	21.0 ºC	16.2 ºC	45.0 ºC	32.0 ºC
3	21.0 ºC	16.2 ºC	31.0 ºC	21.0 ºC
4	21.0 ºC	16.2 ºC	18.0 ºC	12.0 ºC
5	35.2 ºC	32.0 ºC	18.0 ºC	12.0 ºC
6	35.2 ºC	32.0 ºC	31.0 ºC	21.0 ºC
7	35.2 ºC	32.0 ºC	45.0 ºC	32.0 ºC

4. PERFORMANCE CALCULATION

The calculation of the performance of the air conditioner can be resumed by the calculation of three parameters:

- Total Cooling Capacity;
- Sensible Cooling Capacity; and
- Energy Efficiency Ratio.

4.1. TOTAL COOLING CAPACITY

When an air conditioner is tested with the air-enthalpy method, its cooling capacity is obtained with the volumetric air flow rate of air through the equipment under tested, the density of moist air and the difference between the specific enthalpy of moist air entering and leaving the equipment.

$$\dot{Q}_{c,AB} = \dot{V}_a \, \rho_a \, (h_{a_1} - h_{a_2}) \tag{4.1}$$

Where:

- $\dot{Q}_{c,AB}$ is the total cooling capacity of unit A or B, in Watts;
- $\dot{V}_a$ is the volumetric air flow rate through the product, in m^3/s;
- ρ_a is the density of moist air, in kg/m^3;
- h_{a1} is the specific enthalpy of moist air entering the product, in kJ/kg;
- h_{a2} is the specific enthalpy of moist air leaving the product, kJ/kg;

Equation (4.1) describes the total cooling capacity of each indoor unit of the dual-split air conditioner. The total cooling capacity of the equipment is the sum of the cooling capacity of each indoor unit.

$$\dot{Q}_c = \dot{Q}_{c,A} + \dot{Q}_{c,B} \tag{4.2}$$

4.1.1. Volumetric Air Flow Rate

The volumetric air flow rate of each indoor unit was determined according to the method described on ANSI/AMCA 210-99 (1999). According to this standard, the volumetric air flow rate measured at a nozzles chamber is given by:

$$\dot{V}_a = \gamma \sum (C_d\ A) \sqrt{\frac{2\ p_d}{\rho_2}} \tag{4.3}$$

In the test performed, the nozzles chamber had only one nozzle, thus equation (4.3) becomes:

$$\dot{V}_a = \gamma\ (C_d\ A) \sqrt{\frac{2\ p_d}{\rho_2}} \tag{4.4}$$

Where:

- γ is the nozzle expansion factor, dimensionless;
- p_d is the static pressure drop across the nozzle, in kPa;
- ρ_2 is the density of the air at the nozzle inlet, in kg/m^3;
- C_d is the nozzle discharge coefficient, dimensionless;
- A is the nozzle discharge area, in m^2;

The nozzle expansion factor is obtained through:

$$\gamma = 1 - (0.548 + 0.71\ \beta^4)(1 - \alpha) \tag{4.5}$$

Where the alpha ratio is given by equation (4.6) and the beta ratio, for the nozzle chamber approach, is taken as zero.

$$\alpha = \frac{p - p_d}{p} \tag{4.6}$$

The nozzle discharge coefficient depends on Reynolds number and the Reynolds number depends on the volumetric air flow rate, which depends on the nozzle discharge coefficient, as shown on equations (4.7) and (4.8). Because of this interrelation, these parameters are obtained by iterative calculation:

$$C_d = 0.9986 - \frac{7.006}{\sqrt{Re}} + \frac{134.6}{Re} \tag{4.7}$$

$$Re = \frac{4\,\dot{V}_a \rho_2}{\pi\, D\, \mu_a} \tag{4.8}$$

Where:

- Re is the Reynolds number, dimensionless;
- D is the diameter of the nozzle, in m;
- μ_a is the viscosity of the air, in Pa.s.

The density of moist air is calculated with equation (4.9).

$$\rho_2 = \frac{p_w}{R_w\, T_{db}} + \frac{p_{da}}{R_{da}\, T_{db}} = \frac{p - 0.3781\, p_w}{R_a\, T_{db}} \tag{4.9}$$

Where p_w is the partial pressure of water vapor in moist air, in kPa; and R_a is the gas constant for air, equals 287.042 J/kg.K (ASHRAE, 2009).

The partial pressure of water vapor in moist air is calculated with the humidity ratio (W) of moist air according to equation (4.10)

$$p_w = \frac{W\, p}{0.622 + W} \tag{4.10}$$

The humidity ratio of moist air is given by mass and energy balances of an adiabatic saturation device (Kuehn *et al.*,1998). The result of these balances is shown on equation (4.11):

$$W = \frac{\left(h_{v,wb} - h_{l,wb}\right) W_s - 1.006\,(T_{db} - T_{wb})}{h_v - h_{l,wb}} \tag{4.11}$$

Where:

- $h_{v,wb}$ is the specific enthalpy of water vapor at wet-bulb temperature, in kJ/kg;
- h_v is the specific enthalpy of water vapor at dry-bulb temperature, in kJ/kg;
- $h_{l,wb}$ is the specific enthalpy of liquid water at wet-bulb temperature, in kJ/kg; and
- W_s is the humidity ratio of moist air at saturation, in kg_w/kg_a.

Considering that the specific heat of liquid water equals 1 cal/(g.K) or 4.1868 kJ/(kg.K); the specific heat of saturated water vapor equals 1.86 kJ/(kg.K) and the specific enthalpy of water vapor at 0ºC is 2500.89 kJ/kg (ASHRAE 2009), the enthalpy of liquid water and water vapor at wet-bulb temperature are:

$$h_{l,wb} = c_{p_{lw}} T_{wb} = 4.1868\, T_{wb} \tag{4.12}$$

$$h_{v,wb} = h_{v,0} + c_{p_w} T_{wb} = 2500.89 + 1.86\, T_{wb} \tag{4.13}$$

The humidity ratio of moist air at saturation is obtained through equation (4.14):

$$W_s = \frac{0.622\, p_{ws}}{p - p_{ws}} \tag{4.14}$$

The parameter p_{ws} is obtained by regression of values of properties table of water. Hyland and Wexler (1983, *apud* ASHRAE 2009) proposed an equation that calculates it. This equation, which is within 300 ppm of accuracy, is shown below:

$$\ln(p_{ws}) = C_1 {T_{wb}}^{-1} + C_2 + C_3 T_{wb} + C_4 {T_{wb}}^2 + C_5 {T_{wb}}^3 + C_6 \ln(T_{wb}) \tag{4.15}$$

Where:

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

The viscosity of air is obtained from the linear regression of values from properties table of air (Bird et al. 2002). Equation (4.16) shows the result of this regression for atmospheric pressure, in Pa.s:

$$\mu_a = (17.20905 + 0.04569\, T_{db})\, 10^{-6} \tag{4.16}$$

Finally, the nozzle discharge area is obtained with the equation of the area of a circle.

4.1.2. Enthalpy of Moist Air

As moist air is considered a mixture of perfect gases (dry air and water vapor), its enthalpy equals the sum of the enthalpy of its components. Hence, the enthalpy of moist air is given by equation (4.17).

$$h_a = h_{da} + W\, h_v \tag{4.17}$$

Where h_{da} is the specific enthalpy for dry air in kJ/kg$_{da}$, and h_v is the specific enthalpy for saturated water vapor in kJ/kg$_w$.

The specific heat of dry air is assumed to be constant and equals 1.006 kJ/(kg.K) (ASHRAE 2009). Similarly to the previous section, the specific heat of saturated water vapor is 1.86 kJ/(kg.K) and the specific enthalpy of water vapor at 0ºC is 2500.89 J/kg (ASHRAE 2009). Thus, the specific enthalpy of dry air and water vapor at dry-bulb temperature can be expressed as:

$$h_{da} = c_{p_a}\, T_{db} \;=\; 1.006\, T_{db} \tag{4.18}$$

$$h_v = h_{v,0} + c_{p_w}\, T_{db} = \; 2500.89 + 1.86\, T_{db} \tag{4.19}$$

It is now possible to calculate the cooling capacity of moist air with equation (4.1).

4.2. SENSIBLE COOLING CAPACITY

The total cooling capacity of the air conditioner refers to the sum of the two types of heat considered in the process. Latent heat is the heat associated with the change of phase of a substance at constant temperature. Sensible heat is the heat added to or removed from a substance resulting in temperature rise or fall, respectively.

The sensible cooling capacity of an air conditioner is the cooling capacity referring only to the sensible heat, or temperature variation. For each unit, it is calculated by equation (4.20).

$$\dot{Q}_{s,AB} = \dot{V}_a\, \rho_a\, c_{p_a}\, (T_{db_1} - T_{db_2}) \tag{4.20}$$

The sensible cooling capacity of the equipment is the sum of the sensible cooling capacity of each indoor unit.

$$\dot{Q}_s = \dot{Q}_{s,A} + \dot{Q}_{s,B} \tag{4.21}$$

4.3. ENERGY EFFICIENCY RATIO

The energy efficiency ratio (EER), also known as the coefficient of performance (COP) expresses the “energy effectiveness of a refrigeration cycle” (Kuehn et al. 1998). By definition, it calculates the rate of energy that is removed by an air conditioner over the work performed by the equipment.

$$EER = \frac{\dot{Q}_c}{\dot{W}} \tag{4.22}$$

Where $\dot{W}$ is the work of the equipment under test.

As the multi-split unit has one fan motor for both refrigeration cycles it would be inaccurate to calculate the energy efficiency ratio of each unit. Thus, only the total EER was calculated.

5. UNCERTAINTIES OF MEASUREMENTS

There are several factors that can influence accuracy of the tests performed in a calorimeter. According to Cherem-Pereira (2003), there are three factors that are the most important:

- the variation of dry-bulb and wet-bulb temperatures during the test;
- the variation of power supply during the test; and
- the uncertainty of measurement of the equipments used to measure the properties of the air.

The calorimeter used to perform the tests is able to maintain the dry-bulb and wet-bulb temperatures in a range of ±0.1ºC from the set temperature. Thus, this variation is summed with the uncertainty of measurement of the resistance temperature detectors (RTD) when any of these parameters are being used in the analysis.

The calorimeter is also able to keep the voltage inputted to the product within less than 0.5% variation from the set point, so the uncertainty due to this characteristic can be neglected from calculations.

The uncertainty of the devices used to measure atmospheric pressure; dry-bulb and wet-bulb temperatures and volumetric flow rate of the air, as well as the power input on the product are important and were considered to determine the uncertainty of measurement of cooling capacity and EER. The uncertainties of measurement of these devices are informed by the manufacturer or obtained from the calibration report. Table 5.1 shows the uncertainties of measurement of the equipment used on the tests performed.

Table 5.1 – Uncertainties of Measurement Devices

Independent Variable [x_i]	**Measurement Equipment**	**Measuring Range**	**Manufacturer**	**Uncertainty of Measurement [$u(x_i)$]**
Atmospheric pressure	Barometer	650 – 750 [mm Hg]	Princo	± 4.2 mmHg
Dry-bulb and Wet-bulb Temperatures	Resistance Temperature Detectors (RTD)	0 – 60 [ºC]	Consistec	± 0.08 ºC
Power input	Power Transducer	0 – 2500 [W]	Yokogawa	± 4.0 W
Volumetric Air-Flow Rate	Liquid Column Manometer	0 – 50 [mm H_2O]	Dwyer	± 0.2 mmH_2O
	Nozzle	87,5 [mm]	Machined	± 0.1 mm

When a given parameter (A) is defined by indirect measurement, meaning that it is obtained by the measurement of its independent variables (x_1, x_2, …x_n), its uncertainty of measurement must be obtained by the combination of the uncertainties of these independent variables. In this case, the method suggested by Taylor (1988) – expressed by equation (5.1) – was used:

$$\frac{u_A}{A} = \pm\sqrt{\left|\frac{u_A(x_1)}{A}\right|^2 + \left|\frac{u_A(x_2)}{A}\right|^2 + \cdots + \left|\frac{u_A(x_n)}{A}\right|^2} \tag{5.1}$$

Where each term is obtained for each x_i by:

$$\frac{u_A(x_i)}{A} = \frac{\partial A}{\partial x_i}\,\frac{u(x_i)}{A} \tag{5.2}$$

Where:

- $\frac{\partial A}{\partial x_i}$ is the sensitivity coefficient of parameter A related to x_i;

- $u(x_i)$ is the uncertainty of measurement of x_i.

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 of the linear regression:

$$S = u_x = \pm\sqrt{\frac{\sum_{i=1}^{n}(x_i - x_t)^2}{n-1}} \tag{5.3}$$

Where S is the standard deviation of the linear regression, x_i is the ith value of parameter x, x_t is the value of parameter x obtained from the table of properties and n is the amount of values of x used for the regression.

5.1. TOTAL COOLING CAPACITY

Equation (4.2) expresses the total cooling capacity of the equipment considering both refrigerant cycles of the dual-split. Applying equations (5.1) and (5.2) to it, the uncertainty of measurement of the total cooling capacity is obtained:

$$\frac{u_{\dot{Q}_c}}{\dot{Q}_c} = \pm\sqrt{\left|\frac{u_{\dot{Q}_{c,A}}}{\dot{Q}_{c,A}}\right|^2 + \left|\frac{u_{\dot{Q}_{c,B}}}{\dot{Q}_{c,B}}\right|^2} \tag{5.4}$$

From equation (4.1), and knowing that the uncertainty of measurement is given by equation (5.1), the uncertainty of measurement of the total cooling capacity of each refrigeration cycle is given by:

$$\frac{u_{\dot{Q}_{c,AB}}}{\dot{Q}_{c,AB}} = \pm\sqrt{\left|\frac{u_{\dot{Q}_{c,AB}}(\dot{V}_a)}{\dot{Q}_{c,AB}}\right|^2 + \left|\frac{u_{\dot{Q}_{c,AB}}(\rho_2)}{\dot{Q}_{c,AB}}\right|^2 + \left|\frac{u_{\dot{Q}_{c,AB}}(h_{a_1})}{\dot{Q}_{c,AB}}\right|^2 + \left|\frac{u_{\dot{Q}_{c,AB}}(h_{a_2})}{\dot{Q}_{c,AB}}\right|^2} \tag{5.5}$$

With equation (5.2), each of the terms on equation (5.5) will become:

$$\frac{u_{\dot{Q}_{c,AB}}(\dot{V}_a)}{\dot{Q}_{c,AB}} = \frac{\partial \dot{Q}_{c,AB}}{\partial \dot{V}_a}\frac{u(\dot{V}_a)}{\dot{Q}_{c,AB}} = \frac{u_{\dot{V}_a}}{\dot{V}_a} \tag{5.6}$$

$$\frac{u_{\dot{Q}_{c,AB}}(\rho_2)}{\dot{Q}_{c,AB}} = \frac{\partial \dot{Q}_{c,AB}}{\partial \rho_2}\frac{u(\rho_2)}{\dot{Q}_{c,AB}} = \frac{u_{\rho_2}}{\rho_2} \tag{5.7}$$

$$\frac{u_{\dot{Q}_{c,AB}}(h_{a1})}{\dot{Q}_{c,AB}} = \frac{\partial \dot{Q}_{c,AB}}{\partial h_{a_1}} \frac{u(h_{a_1})}{\dot{Q}_{c,AB}} = \frac{u_{h_{a1}}}{h_{a_1} - h_{a_2}} \tag{5.8}$$

$$\frac{u_{\dot{Q}_{c,AB}}(h_{a2})}{\dot{Q}_{c,AB}} = \frac{\partial \dot{Q}_{c,AB}}{\partial h_{a_2}} \frac{u(h_{a_2})}{\dot{Q}_{c,AB}} = \frac{u_{h_{a2}}}{h_{a_2} - h_{a_1}} \tag{5.9}$$

Thus, equation(5.5) becomes:

$$\frac{u_{\dot{Q}_{c,AB}}}{\dot{Q}_{c,AB}} = \pm\sqrt{\left|\frac{u_{\dot{V}_a}}{\dot{V}_a}\right|^2 + \left|\frac{u_{\rho_2}}{\rho_2}\right|^2 + \left|\frac{u_{h_{a1}}}{h_{a_1} - h_{a_2}}\right|^2 + \left|\frac{u_{h_{a2}}}{h_{a_2} - h_{a_1}}\right|^2} \tag{5.10}$$

From equation (4.3), which defines the volumetric air flow rate, the uncertainty of this parameter can be defined by:

$$\frac{u_{\dot{V}_a}}{\dot{V}_a} = \pm\sqrt{\left|\frac{u_{\dot{V}_a}(\gamma)}{\dot{V}_a}\right|^2 + \left|\frac{u_{\dot{V}_a}(p_d)}{\dot{V}_a}\right|^2 + \left|\frac{u_{\dot{V}_a}(\rho_2)}{\dot{V}_a}\right|^2 + \left|\frac{u_{\dot{V}_a}(C_d)}{\dot{V}_a}\right|^2 + \left|\frac{u_{\dot{V}_a}(A)}{\dot{V}_a}\right|^2} \tag{5.11}$$

Applying equation (5.2) to the terms of equation (5.11):

$$\frac{u_{\dot{V}_a}(\gamma)}{\dot{V}_a} = \frac{\partial \dot{V}_a}{\partial \gamma} \frac{u(\gamma)}{\dot{V}_a} = \frac{u_\gamma}{\gamma} \tag{5.12}$$

$$\frac{u_{\dot{V}_a}(p_d)}{\dot{V}_a} = \frac{\partial \dot{V}_a}{\partial p_d} \frac{u(p_d)}{\dot{V}_a} = \frac{u_{p_d}}{2\,p_d} \tag{5.13}$$

$$\frac{u_{\dot{V}_a}(\rho_2)}{\dot{V}_a} = \frac{\partial \dot{V}_a}{\partial \rho_2} \frac{u(\rho_2)}{\dot{V}_a} = -\frac{u_{\rho_2}}{2\,\rho_2} \tag{5.14}$$

$$\frac{u_{\dot{V}_a}(C_d)}{\dot{V}_a} = \frac{\partial \dot{V}_a}{\partial C_d} \frac{u(C_d)}{\dot{V}_a} = \frac{u_{C_d}}{C_d} \tag{5.15}$$

$$\frac{u_{\dot{V}_a}(A)}{\dot{V}_a} = \frac{\partial \dot{V}_a}{\partial p_d} \frac{u(A)}{\dot{V}_a} = \frac{u_A}{A} \tag{5.16}$$

Then, equation (5.11) becomes:

$$\frac{u_{\dot{V}_a}}{\dot{V}_a} = \pm\sqrt{\left|\frac{u_\gamma}{\gamma}\right|^2 + \left|\frac{u_{p_d}}{2\,p_d}\right|^2 + \left|-\frac{u_{\rho_2}}{2\,\rho_2}\right|^2 + \left|\frac{u_{Cd}}{C_d}\right|^2 + \left|\frac{u_A}{A}\right|^2} \tag{5.17}$$

The nozzle expansion factor is function of the alpha ratio only, as the beta ratio is taken as zero. Considering equation (5.2) the uncertainty of measurement due to this parameter is:

$$\frac{u_\gamma}{\gamma} = \pm\left|\frac{u_\gamma(\alpha)}{\gamma}\right| = \pm\left|\frac{\partial\gamma}{\partial\alpha}\,\frac{u(\alpha)}{\gamma}\right| = \pm\left|\frac{0.548\,u_\alpha}{1-0.548\,(1-\alpha)}\right| \tag{5.18}$$

Additionally, alpha ratio is a function of the atmospheric pressure and the pressure drop on the nozzle. So, its uncertainty of measurement is:

$$\frac{u_\alpha}{\alpha} = \pm\sqrt{\left|\frac{u_\alpha(p)}{\alpha}\right|^2 + \left|\frac{u_\alpha(p_d)}{\alpha}\right|^2} \tag{5.19}$$

These terms can be described as below:

$$\frac{u_\alpha(p)}{\alpha} = \frac{\partial\alpha}{\partial p}\,\frac{u(p)}{\alpha} = \frac{p_d\,u_p}{p\,(p-p_d)} \tag{5.20}$$

$$\frac{u_\alpha(p_d)}{\alpha} = \frac{\partial\alpha}{\partial p_d}\,\frac{u(p_d)}{\alpha} = -\frac{u_{p_d}}{p-p_d} \tag{5.21}$$

And equation (5.19) can be written as below:

$$\frac{u_\alpha}{\alpha} = \pm\sqrt{\left|\frac{p_d\,u_p}{p\,(p-p_d)}\right|^2 + \left|-\frac{u_{p_d}}{p-p_d}\right|^2} \tag{5.22}$$

The density of air is defined on equation (4.9). Its uncertainty of measurement is given by the combined uncertainty of measurement of its independent variables, as shown on equation (5.23).

$$\frac{u_{\rho_2}}{\rho_2} = \pm\sqrt{\left|\frac{u_{\rho_2}(p)}{\rho_2}\right|^2 + \left|\frac{u_{\rho_2}(p_w)}{\rho_2}\right|^2 + \left|\frac{u_{\rho_2}(T_{db})}{\rho_2}\right|^2} \tag{5.23}$$

Where:

$$\frac{u_{\rho_2}(p)}{\rho_2} = \frac{\partial\rho_2}{\partial p}\,\frac{u(p)}{\rho_2} = \frac{u_p}{p-0.3781p_w} \tag{5.24}$$

$$\frac{u_{\rho_2}(p_w)}{\rho_2} = \frac{\partial\rho_2}{\partial p_w}\,\frac{u(p_w)}{\rho_2} = -\frac{0.3781\,u_{p_w}}{p-0.3781\,p_w} \tag{5.25}$$

$$\frac{u_{\rho_2}(T_{db})}{\rho_2} = \frac{\partial\rho_2}{\partial T_{db}}\,\frac{u(T_{db})}{\rho_2} = \frac{(0.3781\,p_w - p)\,u_{T_{db}}}{T_{db}\,(p-0.3781\,p_w)} \tag{5.26}$$

Equation (5.23) can be re-written as below:

$$\frac{u_{\rho_2}}{\rho_2} = \pm\sqrt{\left|\frac{u_p}{p - 0.3781 p_w}\right|^2 + \left|\frac{0.3781\, u_{p_w}}{p - 0.3781\, p_w}\right|^2 + \left|\frac{(0.3781\, p_w - \; p)\, u_{T_{db}}}{T_{db}\,(p - 0.3781\, p_w)}\right|^2} \tag{5.27}$$

The partial pressure of water vapor in moist air was calculated with equation (4.10). The uncertainty of this parameter will be given by:

$$\frac{u_{p_w}}{p_w} = \pm\sqrt{\left|\frac{u_{p_w}(W)}{p_w}\right|^2 + \left|\frac{u_{p_w}(p)}{p_w}\right|^2} \tag{5.28}$$

Where:

$$\frac{u_{p_w}(W)}{p_w} = \frac{\partial p_w}{\partial W}\,\frac{u(W)}{p_w} = \frac{0.622\, u_W}{W\,(W + 0.622)} \tag{5.29}$$

$$\frac{u_{p_w}(p)}{p_w} = \frac{\partial p_w}{\partial p}\,\frac{u(p)}{p_w} = \frac{u_p}{p} \tag{5.30}$$

So:

$$\frac{u_{p_w}}{p_w} = \pm\sqrt{\left|\frac{0.622\, u_W}{W\,(W + 0.622)}\right|^2 + \left|\frac{u_p}{p}\right|^2} \tag{5.31}$$

The uncertainty of the humidity ratio, given by equation (4.11), is shown below:

$$\frac{u_W}{W} = \pm\sqrt{\left|\frac{u_W(h_{v,wb})}{W}\right|^2 + \left|\frac{u_W(h_{l,wb})}{W}\right|^2 + \left|\frac{u_W(W_s)}{W}\right|^2 + \left|\frac{u_W(T_{db})}{W}\right|^2 + \left|\frac{u_W(T_{wb})}{W}\right|^2 + \left|\frac{u_W(h_v)}{W}\right|^2} \tag{5.32}$$

Each of the terms on equation (5.32) is calculated below:

$$\frac{u_W(h_{v,wb})}{W} = \frac{\partial W}{\partial h_{v,wb}}\,\frac{u(h_{v,wb})}{W} = \frac{W_s\, u_{h_{v,wb}}}{W\,(h_v - \; h_{l,wb})} \tag{5.33}$$

$$\frac{u_W(h_{l,wb})}{W} = \frac{\partial W}{\partial h_{l,wb}}\,\frac{u(h_{l,wb})}{W} = \frac{\big((h_{v,wb} - h_v)W_s - 1.006(T_{db} - T_{wb})\big)\, u_{h_{l,wb}}}{W\,(h_v - h_{l,wb})^2} \tag{5.34}$$

$$\frac{u_W(W_s)}{W} = \frac{\partial W}{\partial W_s}\,\frac{u(W_s)}{W} = \frac{(h_{v,wb} - h_v)\, u_{W_s}}{W\,(h_v - h_{l,wb})} \tag{5.35}$$

$$\frac{u_W(T_{db})}{W} = \frac{\partial W}{\partial T_{db}}\,\frac{u(T_{db})}{W} = -\frac{1.006\, u_{T_{db}}}{W\,(h_v - h_{l,wb})} \tag{5.36}$$

$$\frac{u_W(T_{wb})}{W} = \frac{\partial W}{\partial T_{wb}} \frac{u(T_{wb})}{W} = \frac{1.006\, u_{T_{wb}}}{W\, (h_v - h_{l,wb})} \tag{5.37}$$

$$\frac{u_W(h_v)}{W} = \frac{\partial W}{\partial h_v} \frac{u(h_v)}{W} = -\frac{\left((h_{v,wb} - h_{l,wb})W_s - 1.006(T_{db} - T_{wb})\right) u_{h_v}}{W\, (h_v - h_{l,wb})^2} \tag{5.38}$$

The uncertainty of W_s in equation (5.35) is calculated through:

$$\frac{u_{W_s}}{W_s} = \pm\sqrt{\left|\frac{u_{W_s}(p_{ws})}{W_s}\right|^2 + \left|\frac{u_{W_s}(p)}{W_s}\right|^2} \tag{5.39}$$

Where:

$$\frac{u_{W_s}(p_{ws})}{W_s} = \frac{\partial W_s}{\partial p_{ws}} \frac{u(p_{ws})}{W_s} = \frac{p\, u_{p_{ws}}}{p_{ws}\, (p - p_{ws})} \tag{5.40}$$

$$\frac{u_{W_s}(p)}{W_s} = \frac{\partial W_s}{\partial p} \frac{u(p)}{W_s} = -\frac{u_p}{p - p_{ws}} \tag{5.41}$$

Finally, equation (5.39) can be re-written as below.

$$\frac{u_{W_s}}{W_s} = \pm\sqrt{\left|\frac{p\, u_{p_{ws}}}{p_{ws}\, (p - p_{ws})}\right|^2 + \left|-\frac{u_p}{p - p_{ws}}\right|^2} \tag{5.42}$$

The partial pressure of water vapor at saturation is obtained by a regression of values from properties table of water. Although the uncertainty of the regression is considered negligible, the uncertainty due to T_{wb} was considered:

$$\frac{u_{p_{ws}}}{p_{ws}} = \pm\left|\frac{u_{p_{ws}}(T_{wb})}{p_{ws}}\right| = \pm\left|-\frac{C_1}{{T_{wb}}^2} + C_3 + 2\, C_4\, T_{wb} + 3\, C_5\, {T_{wb}}^2 + \frac{C_6}{T_{wb}}\right| u_{T_{wb}} \tag{5.43}$$

The uncertainties of the enthalpy terms on equations (5.33), (5.34) and (5.38) are calculated in terms of the uncertainty of the temperature measurement and the uncertainty of the linear regression of each equation. Hence:

$$\frac{u_{h_{v,wb}}}{h_{v,wb}} = \pm\sqrt{\left|\frac{u_{h_{v,wb}}(T)}{h_{v,wb}}\right|^2 + \left|\frac{u_{h_{v,wb}}(reg)}{h_{v,wb}}\right|^2} \tag{5.44}$$

$$\frac{u_{h_{l,wb}}}{h_{l,wb}} = \pm\sqrt{\left|\frac{u_{h_{l,wb}}(T)}{h_{l,wb}}\right|^2 + \left|\frac{u_{h_{l,wb}}(reg)}{h_{l,wb}}\right|^2} \tag{5.45}$$

$$\frac{u_{h_v}}{h_v} = \pm\sqrt{\left|\frac{u_{h_v}(T)}{h_v}\right|^2 + \left|\frac{u_{h_v}(reg)}{h_v}\right|^2} \tag{5.46}$$

If the first term of each equation is detailed:

$$\frac{u_{h_{v,wb}}}{h_{v,wb}} = \pm\sqrt{\left|\frac{1.86\ u_T}{h_{v,wb}}\right|^2 + \left|\frac{u_{h_{v,wb}}(reg)}{h_{v,wb}}\right|^2} \tag{5.47}$$

$$\frac{u_{h_{l,wb}}}{h_{l,wb}} = \pm\sqrt{\left|\frac{4.1868\ u_T}{h_{l,wb}}\right|^2 + \left|\frac{u_{h_{l,wb}}(reg)}{h_{l,wb}}\right|^2} \tag{5.48}$$

$$\frac{u_{h_v}}{h_v} = \pm\sqrt{\left|\frac{1.86\ u_T}{h_v}\right|^2 + \left|\frac{u_{h_v}(reg)}{h_v}\right|^2} \tag{5.49}$$

The uncertainty of the coefficient of discharge of the nozzle is 1.2%, as required by ANSI/AMCA (1999).

The outlet area of the nozzle is given by the area of a circle of diameter D. Hence, using the same method that was used so far, its uncertainty of measurement is:

$$\frac{u_A}{A} = \pm\left|\frac{u_A(D)}{A}\right| = \pm\left|\frac{\partial A}{\partial D}\ \frac{u(D)}{A}\right| = \pm\left|\frac{2u_D}{D}\right| \tag{5.50}$$

For the last two terms of equation (4.1), the uncertainty of measurement is given by the uncertainty of measurement of the specific enthalpy of moist air, which is dependent of the specific enthalpy of dry air, the humidity ratio of moist air and the specific enthalpy of water vapor.

$$\frac{u_{h_a}}{h_a} = \pm\sqrt{\left|\frac{u_{h_a}(h_{da})}{h_a}\right|^2 + \left|\frac{u_{h_a}(W)}{h_a}\right|^2 + \left|\frac{u_{h_a}(h_v)}{h_a}\right|^2} \tag{5.51}$$

Expanding the terms on equation above:

$$\frac{u_{h_a}(h_{da})}{h_a} = \frac{\partial h_a}{\partial h_{da}}\ \frac{u(h_{da})}{h_a} = \frac{u_{h_{da}}}{h_a} \tag{5.52}$$

$$\frac{u_{h_a}(W)}{h_a} = \frac{\partial h_a}{\partial W}\ \frac{u(W)}{h_a} = \frac{h_v\ u_W}{h_a} \tag{5.53}$$

$$\frac{u_{h_a}(h_v)}{h_a} = \frac{\partial h_a}{\partial h_v} \frac{u(h_v)}{h_a} = \frac{W\, u_{h_v}}{h_a} \tag{5.54}$$

So:

$$\frac{u_{h_a}}{h_a} = \pm \sqrt{\left|\frac{u_{h_{da}}}{h_a}\right|^2 + \left|\frac{h_v\, u_W}{h_a}\right|^2 + \left|\frac{W\, u_{h_v}}{h_a}\right|^2} \tag{5.55}$$

Equations (4.18) and (4.19), which define the specific enthalpy of dry air and saturated water vapor are a linear regression of properties tables of air and water, respectively. Although the uncertainty of the values on the tables will not be considered, the uncertainties of the linear regressions are important and were calculated through equation (5.3). In addition, the uncertainties of these parameters due to T_{db} are calculated with equations (5.56) and (5.57), respectively.

$$\frac{u_{h_{da}}(T_{db})}{h_{da}} = \frac{\partial h_{da}}{\partial T_{db}} \frac{u(T_{db})}{h_{da}} = \frac{u_{T_{db}}}{T_{db}} \tag{5.56}$$

$$\frac{u_{h_v}(T_{db})}{h_v} = \frac{\partial h_v}{\partial T_{db}} \frac{u(T_{db})}{h_v} = \frac{1.86\, u_{T_{db}}}{h_v} \tag{5.57}$$

Finally:

$$\frac{u_{h_{da}}}{h_{da}} = \pm \sqrt{\left|\frac{u_{T_{db}}}{T_{db}}\right|^2 + \left|\frac{u_{h_{da}}(reg)}{h_{da}}\right|^2} \tag{5.58}$$

$$\frac{u_{h_v}}{h_v} = \pm \sqrt{\left|\frac{1.86\, u_{T_{db}}}{h_v}\right|^2 + \left|\frac{u_{h_v}(reg)}{h_v}\right|^2} \tag{5.59}$$

5.2. SENSIBLE COOLING CAPACITY

As the sensible cooling capacity of the product is given by equation (4.21), its uncertainty of measurement will be given by:

$$\frac{u_{\dot{Q}_s}}{\dot{Q}_s} = \pm \sqrt{\left|\frac{u_{\dot{Q}_{s,A}}}{\dot{Q}_{s,A}}\right|^2 + \left|\frac{u_{\dot{Q}_{s,B}}}{\dot{Q}_{s,B}}\right|^2} \tag{5.60}$$

The sensible cooling capacity of each unit is calculated through equation (5.61):

$$\frac{u_{\dot{Q}_{s,AB}}}{\dot{Q}_{s,AB}} = \pm\sqrt{\left|\frac{u_{\dot{Q}_{s,AB}}(\dot{V}_a)}{\dot{Q}_{s,AB}}\right|^2 + \left|\frac{u_{\dot{Q}_{s,AB}}(\rho_2)}{\dot{Q}_{s,AB}}\right|^2 + \left|\frac{u_{\dot{Q}_{s,AB}}(c_{p_a})}{\dot{Q}_{s,AB}}\right|^2 + \left|\frac{u_{\dot{Q}_{s,AB}}(T_{db_1})}{\dot{Q}_{s,AB}}\right|^2 + \left|\frac{u_{\dot{Q}_{s,AB}}(T_{db_2})}{\dot{Q}_{s,AB}}\right|^2} \tag{5.61}$$

Considering that the uncertainty of the specific heat of moist air is negligible and using the same method that was used for the total cooling capacity, each term of the equation above can be expanded as below:

$$\frac{u_{\dot{Q}_{s,AB}}(\dot{V}_a)}{\dot{Q}_{s,AB}} = \frac{\partial \dot{Q}_{s,AB}}{\partial \dot{V}_a}\frac{u(\dot{V}_a)}{\dot{Q}_{s,AB}} = \frac{u_{\dot{V}_a}}{\dot{V}_a} \tag{5.62}$$

$$\frac{u_{\dot{Q}_{s,AB}}(\rho_2)}{\dot{Q}_{s,AB}} = \frac{\partial \dot{Q}_{s,AB}}{\partial \rho_2}\frac{u(\rho_2)}{\dot{Q}_{s,AB}} = \frac{u_{\rho_2}}{\rho_2} \tag{5.63}$$

$$\frac{u_{\dot{Q}_{s,AB}}(T_{db_1})}{\dot{Q}_{s,AB}} = \frac{\partial \dot{Q}_{s,AB}}{\partial T_{db_1}}\frac{u(T_{db_1})}{\dot{Q}_{s,AB}} = \frac{u_{T_{db_1}}}{T_{db_1} - T_{db_2}} \tag{5.64}$$

$$\frac{u_{\dot{Q}_{s,AB}}(T_{db_2})}{\dot{Q}_{s,AB}} = \frac{\partial \dot{Q}_{s,AB}}{\partial T_{db_2}}\frac{u(T_{db_2})}{\dot{Q}_{s,AB}} = \frac{u_{T_{db_2}}}{T_{db_2} - T_{db_1}} \tag{5.65}$$

Hence, equation (5.61) is written as:

$$\frac{u_{\dot{Q}_{s,AB}}}{\dot{Q}_{s,AB}} = \pm\sqrt{\left|\frac{u_{\dot{V}_a}}{\dot{V}_a}\right|^2 + \left|\frac{u_{\rho_2}}{\rho_2}\right|^2 + \left|\frac{u_{T_{db_1}}}{T_{db_1} - T_{db_2}}\right|^2 + \left|\frac{u_{T_{db_2}}}{T_{db_2} - T_{db_1}}\right|^2} \tag{5.66}$$

The calculation of each term was demonstrated on section 5.1.

5.3. ENERGY EFFICIENCY RATIO

The energy efficiency ratio is given in terms of the total cooling capacity and the total power consumption of the product, as shown on equation (4.22). The uncertainty of measurement of EER is given by:

$$\frac{u_{EER}}{EER} = \pm\sqrt{\left|\frac{u_{\dot{Q}c}}{\dot{Q}_c}\right|^2 + \left|\frac{u_{\dot{W}}}{\dot{W}}\right|^2} \tag{5.67}$$

6. TESTS RESULTS AND ANALYSIS

As there was only one nozzle chamber available for the test, the test conditions on unit A and unit B were slightly different. On test 1, while the measured dry-bulb temperature on the indoor chamber during the test of unit A was 26.8ºC, this parameter during the test of unit B was 26.6ºC. Although the measured values are different, both values are within the tolerance requested for the test. Thus, the calculation of total cooling capacity was performed neglecting this difference. The same analysis is performed for wet-bulb temperature of the indoor chamber and the dry-bulb temperature of outdoor chamber.

The humidifier of the outdoor chamber was defective during the tests and it was not possible to precisely control the relative humidity, hence the wet-bulb temperature, of the outdoor chamber. For this reason, this parameter presented great variations from the test of unit A to the test of unit B.However, as the humidity of outdoor chamber does not have significant impact on the product performance, this difference was neglected.

When testing the unit A, the enthalpy of air on the indoor chamber is 60.00 kJ/kg. After flowing through the evaporator, the dry-bulb temperature of the air drops to 13.84ºC and its enthalpy drops to 38.28 kJ/kg. As the calculated air flow rate is 0.1289 m^3/h, the total cooling capacity of unit A is 3006 W, or 10256 BTU/h.

During the test of unit B, the enthalpy of air was 59.16 kJ/kg before flowing through the evaporator and 39.80 kJ/kg after flowing through the evaporator. Although the indoor unit B should be exactly the same of the indoor unit A, the fan motor and axial flow fan of unit B were more effective than those of unit A, and the airflow of unit B was 0.1391m^3/h – greater than that of unit A. The total cooling capacity of unit B is 2974 W, or 10147 BTU/h.

Combining the capacity of both units, the total cooling of the product at the test conditions suggested by ANSI/AHAM RAC-1 is 5980 W, or 20403 BTU/h. The calculated uncertainty of measurement is 173 W, or 2.90%. The sensible cooling capacity is 3739W and its uncertainty of measurement is 152 W, or 4.07%.

As the measured power consumption was 2299 W, the energy efficiency ratio is 2.60 and the uncertainty of measurement is 0.08 or 2.90%.

The test results from test 1 are shown on Table 6.1.

Table 6.1 – Test Results 1

Test 1				
Parameter			Unit A	Unit B
Indoor Side	T_{db_1} | T_{wb_1}	ºC	**26.79 | 19.42**	**26.60 | 19.24**
	Φ	%	52.87%	52.65%
	h_{a_1}	kJ/kg	60.00	59.16
Outdoor Side	T_{db} | T_{wb}	ºC	**34.99 | 26.44**	**35.16 | 24.01**
Evaporator Outlet	T_{db_2} | T_{wb_2}	ºC	13.84 | 12.74	14.37 | 13.13
	Φ	%	90.00%	88.81%
	h_{a_2}	kJ/kg	38.82	39.80
Air Flow Rate	$\dot{V}_a$	m^3/s	0.1289	0.1391
Total Cooling Capacity	$\dot{Q}_c$	W	3006	2974
			5980 ± 173	
Sensible Cooling Capacity	$\dot{Q}_s$	W	1849	1890
			3739 ± 152	
Power Consumption	$\dot{W}$	W	2299	
EER	EER	-	**2.60 ± 0.08**	

For test 2, the indoor dry-bulb temperature of air was 21ºC – the lowest level of all tests performed. The outdoor dry-bulb temperature was 45ºC – the highest level of all tests performed. These test conditions present the lowest temperature difference on both heat exchangers, so this test resulted on the lowest total cooling capacity and lowest EER of the product.

The total cooling capacity of unit A was 2227 W, or 7599 BTU/h and unit B was 2159, or 7366 BTU/h. The total cooling capacity of the product was 4386 W, or 14965 BTU/h. The test results of test 2 are presented in Table 6.2.

Table 6.2 – Test Results 2

Test 2				
Parameter			Unit A	Unit B
Indoor Side	$T_{db_1} \mid T_{wb_1}$	ºC	**21.07 | 16.11**	**21.02 | 16.10**
	Φ	%	62.71%	62.77%
	h_{a_1}	kJ/kg	48.81	48.51
Outdoor Side	$T_{db} \mid T_{wb}$	ºC	**44.98 | 32.43**	**45.04 | 31.91**
Evaporator Outlet	$T_{db_2} \mid T_{wb_2}$	ºC	11.19 | 10.38	11.91 | 11.02
	Φ	%	91.98%	91.43%
	h_{a_2}	kJ/kg	32.58	34.07
Air Flow Rate	$\dot{V}_a$	m^3/s	0.1235	0.1339
Total Cooling Capacity	$\dot{Q}_c$	W	2227	2159
			4386 ± 142	
Sensible Cooling Capacity	$\dot{Q}_s$	W	1363	1370
			2733 ± 136	
Power Consumption	$\dot{W}$	W	2546	
EER	EER	-	**1.72 ± 0.06**	

Test 3 was performed with the same indoor conditions of test 2, but outdoor temperature was reduced to 31ºC. Due to this variation, which increased the temperature difference between air and refrigerant fluid on outdoor unit, the equipment enhanced its cooling performance. The total cooling capacity was 5475 W, or 18680 BTU/h and the EER was 2.57.

Table 6.3 – Test Results 3

<table>
<tr><th colspan="5">Test 3</th></tr>
<tr><th colspan="3">Parameter</th><th>Unit A</th><th>Unit B</th></tr>
<tr><td rowspan="3">Indoor Side</td><td>$T_{db_1} \mid T_{wb_1}$</td><td>ºC</td><td>20.95 | 16.01</td><td>20.88 | 16.00</td></tr>
<tr><td>Φ</td><td>%</td><td>62.67%</td><td>62.90%</td></tr>
<tr><td>h_{a_1}</td><td>kJ/kg</td><td>48.50</td><td>48.11</td></tr>
<tr><td>Outdoor Side</td><td>$T_{db} \mid T_{wb}$</td><td>ºC</td><td>31.39 | 22.23</td><td>31.11 | 20.67</td></tr>
<tr><td rowspan="3">Evaporator Outlet</td><td>$T_{db_2} \mid T_{wb_2}$</td><td>ºC</td><td>9.35 | 8.74</td><td>10.49 | 9.40</td></tr>
<tr><td>Φ</td><td>%</td><td>93.62%</td><td>88.55%</td></tr>
<tr><td>h_{a_2}</td><td>kJ/kg</td><td>28.55</td><td>29.93</td></tr>
<tr><td>Air Flow Rate</td><td>$\dot{V}_a$</td><td>m^3/s</td><td>0.1231</td><td>0.1334</td></tr>
<tr><td rowspan="2">Total Cooling Capacity</td><td rowspan="2">$\dot{Q}_c$</td><td rowspan="2">W</td><td>2746</td><td>2729</td></tr>
<tr><td colspan="2">5475 ± 195</td></tr>
<tr><td rowspan="2">Sensible Cooling Capacity</td><td rowspan="2">$\dot{Q}_s$</td><td rowspan="2">W</td><td>1607</td><td>1568</td></tr>
<tr><td colspan="2">3175 ± 154</td></tr>
<tr><td>Power Consumption</td><td>$\dot{W}$</td><td>W</td><td colspan="2">2132</td></tr>
<tr><td>EER</td><td>EER</td><td>-</td><td colspan="2">2.57 ± 0.09</td></tr>
</table>

Test 4 was performed with the same indoor conditions of tests 2 and 3, but outdoor temperature was reduced to 18ºC. These conditions improved even more the cooling performance of the equipment. The total cooling capacity tested was 6565 W, or 22400 BTU/h and the EER was 3.85.

Table 6.4 – Test Results 4

Test 4				
Parameter			Unit A	Unit B
Indoor Side	$T_{db_1} \mid T_{wb_1}$	ºC	**20.89 | 16.72**	**21.03 | 16.15**
	Φ	%	68.29%	63.02%
	h_{a_1}	kJ/kg	50.77	48.59
Outdoor Side	$T_{db} \mid T_{wb}$	ºC	**17.44 | 12.05**	**18.38 | 12.35**
Evaporator Outlet	$T_{db_2} \mid T_{wb_2}$	ºC	8.58 | 8.13	9.92 | 8.83
	Φ	%	95.21%	88.27%
	h_{a_2}	kJ/kg	27.09	28.55
Air Flow Rate	$\dot{V}_a$	m^3/s	0.1289	0.1388
Total Cooling Capacity	$\dot{Q}_c$	W	3428	3137
			6565 ± 245	
Sensible Cooling Capacity	$\dot{Q}_s$	W	1791	1749
			3540 ± 171	
Power Consumption	$\dot{W}$	W	1704	
EER	EER	-	**3.85 ± 0.14**	

The humidifier control of the indoor chamber was defective during the tests 5, 6 and 7 of unit B, thus it was not possible to maintain this parameter to the set level. In addition, the temperature control as well as the humidity control of outdoor chamber also presented problems. For this reason the test results of units A and B presented a greater divergence between themselves.

Among all the tests performed, the test conditions of test 5 presented the highest temperature differences between air and refrigerant fluid on both heat exchangers. The indoor chamber was set to 35.2ºC and 32.0ºC and the outdoor chamber was set to 18ºC and 12ºC on dry-bulb and wet-bulb temperatures.

While the enthalpy difference of unit A was 39.27 kJ/kg, on unit B this difference was 30.81 kJ/kg. The difference between the values found for the two units was slightly compensated by the greater airflow of unit B, which reduced the difference between the total cooling capacity of both units. The total cooling capacity of unit A was 5296 W, or 18070 BTU/h and the total cooling capacity of unit B was 4458 W, or 15211 BTU/h. Consequently, unit A rejected 18.8% more heat than unit B. The EER of the combined units were 5.27, meaning that the product was able to reject more than 5 times more heat than its power consumption. The summarized data of test 5 is presented on Table 6.5.

Table 6.5 – Test Results 5

<table>
<tr><th colspan="5">Test 5</th></tr>
<tr><th colspan="3">Parameter</th><th>Unit A</th><th>Unit B</th></tr>
<tr><td rowspan="3">Indoor Side</td><td>$T_{db_1} \mid T_{wb_1}$</td><td>ºC</td><td>35.23 | 31.75</td><td>35.14 | 33.06</td></tr>
<tr><td>Φ</td><td>%</td><td>81.10%</td><td>89.46%</td></tr>
<tr><td>h_{a_1}</td><td>kJ/kg</td><td>120.40</td><td>128.39</td></tr>
<tr><td>Outdoor Side</td><td>$T_{db} \mid T_{wb}$</td><td>ºC</td><td>17.58 | 15.03</td><td>19.03 | 13.77</td></tr>
<tr><td rowspan="3">Evaporator Outlet</td><td>$T_{db_2} \mid T_{wb_2}$</td><td>ºC</td><td>25.28 | 24.53</td><td>28.40 | 27.95</td></tr>
<tr><td>Φ</td><td>%</td><td>96.37%</td><td>99.14%</td></tr>
<tr><td>h_{a_2}</td><td>kJ/kg</td><td>81.13</td><td>97.58</td></tr>
<tr><td>Air Flow Rate</td><td>$\dot{V}_a$</td><td>m^3/s</td><td>0.1280</td><td>0.1381</td></tr>
<tr><td rowspan="2">Total Cooling Capacity</td><td rowspan="2">$\dot{Q}_c$</td><td rowspan="2">W</td><td>5296</td><td>4458</td></tr>
<tr><td colspan="2">9754 ± 229</td></tr>
</table>

Sensible Cooling Capacity	$\dot{Q}_s$	W	1350	982
			2332 ± 120	
Power Consumption	$\dot{W}$	W	1850	
EER	EER	-	**5.27 ± 0.12**	

Table 6.6 and Table 6.7 show the test results of tests 6 and 7 respectively.

Table 6.6 – Test Results 6

Test 6				
Parameter			Unit A	Unit B
Indoor Side	$T_{db_1} \mid T_{wb_1}$	ºC	**35.28 | 31.78**	**35.20 | 33.12**
	Φ	%	80.97%	89.53%
	h_{a_1}	kJ/kg	120.64	128.92
Outdoor Side	$T_{db} \mid T_{wb}$	ºC	**30.93 | 25.42**	**30.66 | 22.92**
Evaporator Outlet	$T_{db_2} \mid T_{wb_2}$	ºC	25.39 | 24.62	27.99 | 27.28
	Φ	%	96.24%	97.18%
	h_{a_2}	kJ/kg	81.60	94.15
Air Flow Rate	$\dot{V}_a$	m^3/s	0.1281	0.1244
Total Cooling Capacity	$\dot{Q}_c$	W	5262	4538
			9799 ± 231	
Sensible Cooling Capacity	$\dot{Q}_s$	W	1341	947
			2287 ± 114	
Power Consumption	$\dot{W}$	W	2352	
EER	EER	-	**4.17 ± 0.10**	

Table 6.7 – Test Results 7

<table>
<tr><th colspan="5">Test 7</th></tr>
<tr><th colspan="3">Parameter</th><th>Unit A</th><th>Unit B</th></tr>
<tr><td rowspan="3">Indoor Side</td><td>T_{db_1} | T_{wb_1}</td><td>ºC</td><td>35.37 | 31.79</td><td>35.24 | 33.06</td></tr>
<tr><td>Φ</td><td>%</td><td>80.51%</td><td>88.93%</td></tr>
<tr><td>h_{a_1}</td><td>kJ/kg</td><td>120.76</td><td>128.64</td></tr>
<tr><td>Outdoor Side</td><td>T_{db} | T_{wb}</td><td>ºC</td><td>44.20 | 35.39</td><td>44.78 | 35.03</td></tr>
<tr><td rowspan="3">Evaporator Outlet</td><td>T_{db_2} | T_{wb_2}</td><td>ºC</td><td>26.05 | 25.24</td><td>28.35 | 27.62</td></tr>
<tr><td>Φ</td><td>%</td><td>95.98%</td><td>97.08%</td></tr>
<tr><td>h_{a_2}</td><td>kJ/kg</td><td>84.52</td><td>95.98</td></tr>
<tr><td>Air Flow Rate</td><td>$\dot{V}_a$</td><td>m^3/s</td><td>0.1231</td><td>0.1246</td></tr>
<tr><td rowspan="2">Total Cooling Capacity</td><td rowspan="2">$\dot{Q}_c$</td><td rowspan="2">W</td><td>4680</td><td>4256</td></tr>
<tr><td colspan="2">8936 ± 211</td></tr>
<tr><td rowspan="2">Sensible Cooling Capacity</td><td rowspan="2">$\dot{Q}_s$</td><td rowspan="2">W</td><td>1210</td><td>904</td></tr>
<tr><td colspan="2">2114 ± 109</td></tr>
<tr><td>Power Consumption</td><td>$\dot{W}$</td><td>W</td><td colspan="2">2999</td></tr>
<tr><td>EER</td><td>EER</td><td>-</td><td colspan="2">2.98 ± 0.07</td></tr>
</table>

7. PERFORMANCE CHARACTERISTICS CURVE

The residential air conditioner systems are designed to have an optimum performance at the conditions required by the performance test standard; usually at 35ºC at the outdoor dry-bulb temperature and 26.7ºC and 19.4ºC at indoor dry-bulb and wet-bulb temperatures. To determine the off-design operation conditions, six tests were performed for each unit at different temperature and humidity conditions on indoor and outdoor testing rooms. With the results of these tests, it was possible to determine a function to describe the performance curve of the unit at any temperature and humidity conditions within a certain range.

Two methods were used to obtain these functions; both of them consisting in a regression of the test data into a bi-quadratic function that describes the cooling capacity and EER of the product. The first method utilized was the one recommended by ASHRAE (2009), which calculates the equipment performance as a function of the outdoor dry-bulb temperature and the indoor wet-bulb temperatures, as shown in Equation (7.1):

$$f = a_0 + a_1.T_{wb} + a_2.T_{wb}^2 + a_3.T_{db,o} + a_4.T_{db,o}^2 + a_5.T_{wb}.T_{db,o} \tag{7.1}$$

The second method is based on ASHRAE method, but replaces the variable of indoor wet-bulb temperature by the indoor enthalpy of the air. This is a new method that is proposed in this study. It will be referred to as the Internal Enthalpy Method. Replacing the indoor wet-bulb temperature by the internal enthalpy, Equation (7.1) becomes:

$$f = a_0 + a_1.h_a + a_2.h_a^2 + a_3.T_{db,o} + a_4.T_{db,o}^2 + a_5.h_a.T_{db,o} \tag{7.2}$$

On both methods, f can express the total cooling capacity, the sensible cooling capacity or the energy efficiency ratio of the equipment. The second method was compared to the one from ASHRAE to determine which one can describe the performance of the equipments more accurately. These equations can be programmed into software of building thermal and energetic simulation, such as DOE and Domus, to enhance their calculation and database. The regressions were performed in a subroutine of Domus software.

7.1. TOTAL COOLING CAPACITY

The result of the regression of the total cooling capacity of unit A is shown in Table 7.1.

Table 7.1 – Constants of Total Cooling Capacity of Unit A

Constants	ASHRAE Method	Internal Enthalpy Method
a_0	3937.4020000	2768.3550000
a_1	-84.0046800	14.5520400
a_2	4.0339920	0.0557586
a_3	-26.2738600	-18.5286600
a_4	-0.5563736	-0.5703255
a_5	1.1687250	0.2493146

It is possible to calculate the off-design total cooling capacity of unit A by replacing the constants in Equations (7.1) and (7.2). The result, as well as the deviation of the regression from the experimental result is shown in Table 7.2.

Table 7.2 – Results of Total Cooling Capacity of Unit A

Test #	Experimental Result	Regression ASHRAE		Regression Internal Enthalpy	
1	3006 W	3022 W	0.50%	3019 W	0.42%
2	2227 W	2171 W	-2.53%	2172 W	-2.49%
3	2746 W	2841 W	3.45%	2841 W	3.46%
4	3428 W	3374 W	-1.57%	3375 W	-1.54%
5	5296 W	5356 W	1.12%	5354 W	1.10%
6	5262 W	5145 W	-2.21%	5147 W	-2.18%

7	4680 W	4737 W	1.21%	4736 W	1.20%
Standard Deviation		181 W		179 W	

The same analysis is performed for unit B, and the results of the regression are shown on Table 7.3.

Table 7.3 – Constants of Total Cooling Capacity of Unit B

Constants	ASHRAE Method	Internal Enthalpy Method
a_0	2224.0720000	1782.8830000
a_1	68.2757400	36.7111700
a_2	-0.3958521	-0.1490214
a_3	-20.4305400	-9.8553230
a_4	-0.6826231	-0.6953079
a_5	1.6935150	0.3590732

Replacing the constants above on equations (7.1) and (7.2) for total cooling capacity, the following results are obtained:

Table 7.4 – Results of Total Cooling Capacity of Unit B

Test #	Experimental Result	Regression ASHRAE		Regression Internal Enthalpy	
1	2974 W	2975 W	0.04%	2974 W	0.01%
2	2159 W	2143 W	-0.74%	2143 W	-0.73%
3	2729 W	2761 W	1.19%	2762 W	1.22%
4	3137 W	3120 W	-0.56%	3120 W	-0.56%
5	4458 W	4478 W	0.44%	4478 W	0.44%
6	4538 W	4502 W	-0.78%	4502 W	-0.78%
7	4256 W	4272 W	0.38%	4272 W	0.39%
Standard Deviation		57 W		58 W	

7.2. SENSIBLE COOLING CAPACITY

The equations for the sensible cooling capacity of the dual-split were obtained by the same methods of the total cooling capacity. The constants of unit A are shown in Table 7.5.

Table 7.5 – Constants of Sensible Cooling Capacity of Unit A

Constants	ASHRAE Method	Internal Enthalpy Method
a_0	-2013.3400000	-762.1877000
a_1	361.6790000	73.5849600
a_2	-8.2555360	-0.4791578
a_3	1.7262920	3.2750470
a_4	-0.3737273	-0.3630068
a_5	0.5128761	0.1194995

These constants lead to the following result for sensible cooling capacity of unit A:

Table 7.6 – Results of Sensible Cooling Capacity of Unit A

Test #	Experimental Result	Regression ASHRAE		Regression Internal Enthalpy	
1	1849 W	1849 W	-0.04%	1849 W	-0.02%
2	1363 W	1364 W	0.08%	1363 W	0.01%
3	1607 W	1605 W	-0.15%	1607 W	-0.04%
4	1791 W	1792 W	0.07%	1791 W	0.01%
5	1350 W	1349 W	-0.08%	1350 W	0.00%
6	1341 W	1344 W	0.21%	1341 W	0.05%
7	1210 W	1209 W	-0.13%	1210 W	-0.05%
Standard Deviation		4 W		1 W	

The same analysis is performed for unit B, and the results of the regression are shown in Table 7.7.

Table 7.7 – Constants of Sensible Cooling Capacity of Unit B

Constants	ASHRAE Method	Internal Enthalpy Method
a_0	-2954.0730000	-1200.8270000
a_1	478.1175000	92.8238400
a_2	-10.8836200	-0.5937349
a_3	-18.4864500	-13.5797300
a_4	-0.0933614	-0.1194216
a_5	0.6494650	0.1454763

Replacing the constants above in Equations (7.1) and (7.2) for sensible cooling capacity, the following results are obtained:

Table 7.8 – Results of Sensible Cooling Capacity of Unit B

Test #	Experimental Result	Regression ASHRAE		Regression Internal Enthalpy	
1	1890 W	1890 W	0.00%	1890 W	0.00%
2	1370 W	1371 W	0.02%	1369 W	-0.10%
3	1568 W	1567 W	-0.08%	1571 W	0.17%
4	1749 W	1750 W	0.05%	1748 W	-0.06%
5	982 W	981 W	-0.12%	983 W	0.10%
6	947 W	948 W	0.18%	944 W	-0.28%
7	904 W	903 W	-0.05%	905 W	0.15%
Standard Deviation		3 W		4 W	

7.3. ENERGY EFFICIENCY RATIO

Finally the analysis is performed for the Energy Efficiency Ratio of the product. The constants obtained for unit A by regression of the test results are shown below:

Table 7.9 – Constants of EER of Unit A

Constants	ASHRAE Method	Internal Enthalpy Method
a_0	4.9179230	4.4778570
a_1	-0.0089398	0.0188354
a_2	0.0029119	0.0000570
a_3	-0.0923633	-0.0978938
a_4	0.0004956	0.0004956
a_5	-00010092	-0.0002212

The results of the calculation using the constants above are shown on Table 7.10.

Table 7.10 – Results of EER of Unit A

Test #	Experimental Result	Regression ASHRAE		Regression Internal Enthalpy	
1	2.52	2.53	0.43%	2.53	0.37%
2	1.69	1.65	-2.85%	1.65	-2.84%
3	2.53	2.60	2.85%	2.60	2.89%
4	3.87	3.83	-0.99%	3.83	-0.98%
5	5.49	5.54	0.92%	5.54	0.92%
6	4.29	4.20	-2.10%	4.20	-2.07%
7	3.00	3.04	1.39%	3.04	1.41%
Standard Deviation		0.14		0.14	

For unit B:

Table 7.11 – Constants of EER of Unit B

Constants	ASHRAE Method	Internal Enthalpy Method
a_0	3.4468370	3.7638340
a_1	0.1930073	0.0558763
a_2	-0.0022702	-0.0002185
a_3	-0.1300858	-0.1306405
a_4	0.0008680	0.0008565
a_5	-0.0001897	-0.0000386

Using the same method as above, the result is:

Table 7.12 – Results of EER of Unit B

Test #	Experimental Result	Regression ASHRAE		Regression Internal Enthalpy	
1	2.69	2.69	0.14%	2.69	0.09%
2	1.75	1.73	-1.36%	1.73	-1.36%
3	2.61	2.65	1.70%	2.65	1.76%
4	3.84	3.82	-0.58%	3.82	-0.59%
5	5.04	5.07	0.45%	5.07	0.45%
6	4.03	3.98	-1.23%	3.98	-1.25%
7	2.96	2.98	0.86%	2.98	0.86%
Standard Deviation		0.08		0.08	

7.4. CALCULATION ANALYSIS

The results of the regressions of the three parameters – total cooling capacity; sensible cooling capacity and energy efficiency ratio – showed that both

ASHRAE and Internal Enthalpy methods presented very similar accuracy. On the total cooling capacity, for unit A, the standard deviation was 181 W when calculated with ASHRAE method, while it was 179 W when calculated with Internal Enthalpy method. The similarity is repeated on all regressions performed.

When comparing with the expectation of test 1, on total cooling capacity, the standard deviation represents approximately 6% on unit A and approximately 2% on unit B – considering any of the methods of regression. As the test of each unit was performed separately, the variations of test conditions may have caused unit B to have more consistent test results than unit A.

Analyzing the sensible cooling capacity of unit A, the method proposed by ASHRAE presented a much higher standard deviation, while on unit B, this method was more precise. When compared to the expectation of the values, however, the relative error is less than 1%. That means that both methods presented similar results.

Finally, analyzing the energy efficiency ratio, it can be noted that both methods presented similar results. On unit A, the standard deviation compared to the expectation of test 1 is 5.6% and on unit B it is 2.9%. Once again, the variation of test conditions on both units may have caused the variation of precision of the regression on both units.

8. CONCLUSION

The air conditioners in Brazil are having a great increase in popularity, mainly because of the price decrease within the past years. Among the air conditioners, there is a particular group of products that are becoming more and more popular recently: the multi-splits air conditioners. The main benefits of this equipment are the possibility of installation on places where there is little space for the outdoor unit and the high energy efficiency ratio that they offer. These characteristics make this equipment attractive in hot places with high demographic density, such as the big cities in south-east Asia for example.

Despite the fact that the market of multi-splits is growing rapidly, there is not – worlwide – a clear standard to determine the test procedure and test conditions for this category. The main standards regarding the test procedures of residential air conditioners are:

- ISO5151 (1994) – applicable mainly in Europe and recognized globally;
- ANSI/AHAM RAC-1 (1992) – applicable in the USA; and
- NBR5882 (1983) – applicable in Brazil.

These standards are valid for window type and split type air conditioners. But all of them either declare in their scope that it is not valid for multi-splits or ommit any information about it and does not describe a test procedure for these equipments.

Given the lack of test standard, even though the product represents a good share of the splits market; a new test procedure is proposed. Using the existing standards for single splits, the new method is created using the air-enthalpy testing method. The air-enthalpy method was chosen because it allows the measurement of the performance of each indoor unit separately. If a different method was used – the calibrated or the balanced ones – the operation of one indoor unit would interfere on the test results of the other unit, and it would be impossible to evaluate the performance of the indoor units.

The calculation of the total cooling capacity, the sensible cooling capacity and the energy efficiency ratio was performed according to ISO 5151 recommendation for the air-enthalpy method. The air flow rate was determined following ANSI/ASHRAE 51(1999) - Laboratory Methods of Testing Fans for Aerodynamics Performance Rating. Following these standards, it was possible to determine the nominal performance parameter of the equipment.

As in any experimental analysis, the tests performed are subject of uncertainties of measurement, mainly due to the uncertainty of measurement of the equipment used to measure and acquire data of the properties of air. The uncertainty of the devices used to measure atmospheric pressure, dry-bulb and wet-bulb temperatures, power input and the air flow rate are either informed by the manufacturer or found in calibration reports. The calorimeter is capable of maintaining the dry-bulb and wet-bulb temperatures within a range of 0.1ºC, so this value was summed with the uncertainty of the resistance temperature detectors. The calorimeter maintained the variation of the power supply below 0.5%, so this factor was considered negligible for the uncertainty analysis.

The total cooling capacity of unit A is 3006 W whereas, the capacity of unit B is 2974 W. Combining the capacity of both units, the total cooling capacity of the dual-split is 5980 W, from which the sensible cooling capacity represents 3739 W. The calculation showed that the uncertainty of the total cooling capacity of the dual-split is 2.90%, the uncertainty of the sensible cooling capacity is 4.07% and the uncertainty of the EER is 2.90%. It is concluded that the experiment presented a high level of accuracy.

In addition to the test performed at standard conditions to determine the nominal performance of the dual-split, six performance tests were performed at off-design conditions for each indoor unit. With the results of all the tests, there was enough data to determine a performance characteristics curve of each circuit of the dual-split. As the tests were performed on one unit at a time and the ventilation of each unit presented a significant variation due to construction variation of the fan motors, a different curve was calculated for each of the units.

The curve is calculated through the regression of the test data into a bi-quadratic equation. Two forms of the equation were evaluated. The first one is recommended by ASHRAE, which uses the wet-bulb temperature of air in the indoor chamber and the dry-bulb temperature of air in the outdoor chamber as the parameters of the curve. The second form uses the enthalpy of air in the indoor chamber (instead of the wet-bulb temperature) and the dry-bulb temperature of air in the outdoor chamber. As the wet-bulb temperature and the enthalpy present similar relative variations according to the conditions of the air, both methods presented very similar results and can be used with the same level of accuracy. The regression of both methods was performed in a subroutine implemented on DOMUS.

In addition to the performance characteristics curves of each unit, the tests performed at off-design conditions allows the analysis of performance variation of the equipment due to the variation of the condition of the air in which the equipment installed. While the nominal total cooling capacity is 5980 W, this parameter varied from 4386 W on test 2 to 9754 W on test 5, depending on the air properties during the test. This represents a variation of 122% of the total cooling capacity. Similarly, the energy efficiency ratio varied from 1.72 on test 2 to 5.27 on test 5, while its nominal value is 2.60 determined on test 1. It means that the energy efficiency ratio varied 206%.

8.1. RECOMMENDATIONS FOR FUTURE WORK

The development of a test procedure for residential multi-split air conditioners will allow the testing and labeling of this kind of equipment. It is recommended that more models of multi-splits are tested following this standard to validate the procedure.

The performance characteristics curves calculated can be implemented on building energy simulation programs to enhance their database with a new type of split air conditioners.

It was noted that most of the heat absorbed by the evaporator in the indoor unit is rejected to the atmospheric air by the condenser in the outdoor unit. This rejected heat can be used to heat water to aid to supply hot water to a residence or small commercial building. A dual-split can be improved with an additional water-cooled condenser placed inside a hot water tank of a residence. Besides the benefit of utilizing the rejected heat to heat water, the additional water-cooled condenser would also increase the condensing capacity of the unit and lower the condensation pressure of the system, consequently increasing the cooling capacity and the energy efficiency ratio of the equipment.

REFERENCES

ABNT MB-3341:1990 . **Condicionador de Ar Doméstico – Determinação do Coeficiente de Eficiência Energética.**

ANSI/AHAM RAC-1-1992. **American National Standard Room Air Conditioner.**

ANSI/ASHRAE 16-1983 (RA 2009) **Method of Testing for Rating Room Air Conditioners and Packaged Terminal Air Conditioners.**

ANSI/ASHRAE 51-1999. **Laboratory Methods of Testing Fans for Aerodynamics Performance Rating.**

ASHRAE, American Society of Heating, Refrigeration and Air Conditioning Engineers. **Handbook of HVAC Applications.** Atlanta: ASHRAE. 2007.

ASHRAE, American Society of Heating, Refrigeration and Air Conditioning Engineers. **Handbook of Fundamentals.** Atlanta: ASHRAE. 2009.

BIRD, R. Byron; STEWART, Warren E.; LIGHTFOOT, Edwin N. **Transport Phenomena**. 2nd ed. John Wiley and Sons Inc. 2002.

CHEREM-PEREIRA, Gustavo C. **Modelagem de Condicionadores de Ar Residenciais**. 2003. 193 p. Dissertation (Master's Degree in Mechanical Engineering). Pontifícia Universidade Católica do Paraná. Curitiba. 2003.

CHEREM-PEREIRA, Gustavo C. and MENDES, Nathan. **Empirical Modeling of Room Air Conditioners for Building Energy Analysis**. Energy and Buildings 47. 2012.

HU, Shih-Cheng and YANG, Rong-Hwa. **Development and Testing of a Multi-Type Air Conditioner without Using AC Inverters.** Energy Conversion and Management 46. 2005.

ISO 5151: 1994. **Non-Ducted Air Conditioners and Heat Pumps – Testing and Rating for Performance.**

KUEHN, Thomas H.; RAMSEY, James W.; THRELKELD, James L. **Thermal Environmental Engineering**. 3rd ed. Prentice Hall. 1998.

NBR 5858:1983. **Condicionador de Ar Doméstico.**

NBR 5882:1983. **Condicionador de Ar Doméstico – Determinação das Características.**

PARK, Youn C.; KIM, Young C.; MIN, Man-Ki. **Performance Analysis on a Multi-Type Inverter Air Conditioner.** Energy Conversion and Management 42. 2001.

TU, Qiu; DONG, Kaijun; ZOU, Deqiu; LIN, Yongman. **Experimental Study on Multi-Split Air Conditioner with Digital Scroll Compressor.** Applied Thermal Engineering 31. 2011.

TAYLOR, J. L. **Fundamentals of Measurement Error.** Neff Instrument Corporation. 1988

WANG, Shan K. **Handbook of Air Conditioning and Refrigeration.** 2nd ed. McGraw-Hill. 2000.

ZHANG, Donglian; ZHANG, Xu; LIU, Jun. **Experimental Study of Performance of Digital Variable Multiple Air Conditioning System Under Part Load Conditions.** Energy and Buildings 43. 2011.