THERMODYNAMIC ANALYSIS OF A REFRIGERATION CYCLE USING REGENERATIVE HEAT EXCHANGER – SUCTION/LIQUID LINE

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Summary: *This paper presents results from thermodynamic comparison of a conventional compression cycle and a steam cycle that uses a heat exchanger countercurrent (liquid line/suction line) in an air conditioning system split. The main objective is to study the relationship between the COP and the mass variation of refrigerant to the effectiveness of the heat exchanger. The papers presented in the literature discuss the matter in a theoretical way, are summarized in tables of rare loss statements without specification of methods. The methodology of work is based on testing of an air conditioner operating conventionally and also with the heat exchanger for the determination of values and parameters of interest. The tests were performed in a thermal chamber with temperature controlled and equipped with a data acquisition system for reading and storage results. The refrigerant was R22. Besides making possible an assessment of the feasibility of cost-benefit thermodynamics, it is suggested a different method for installing the equipment type split.*

Keywords: *Heat exchanger, air conditioning, thermodynamic efficiency, split, liquid line/suction line.*

1. INTRODUCTION

Especially the last two decades environmental issues have gained great space in the discussions and surveys of the global society. In this context, industrial processes and domestic refrigeration using type refrigerants: CFC, HFC's and HCFC's suffer gradual restrictions and regulations, because they have high potential for action on the ozone layer and /or greenhouse gases. Due to increasing global temperatures, the use of air conditioning has become inevitable for human comfort, and is essential in certain regions. It is estimated that three quarters of emissions of HFCs and HCFCs are linked to any leakage of air conditioning systems, which features a vicious circle of rising greenhouse gas (HINRICHS & KLEINBACH, 2003). Therefore, all progress, in order to reduce the quantities of these gases present in several types of HVAC systems, is beneficial to both the environment and to operating costs.

Beyond the attention that the refrigerants have another source of discussion and research, is the growing share of electricity produced for the relevant processes related to climate. According to survey statistics, the energy consumption by sector in Brazil is distributed as a Fig.1. In the residential sector, the largest electricity consumption is attributed to refrigeration and air conditioning, representing 33.0% of consumption in this sector. A reduction of only 1% of the consumption of residential cooling equipment, would result in a savings of about 30GWh/year. In the commercial sector, 20% of energy consumption are due to air conditioning (BEN, 2009).

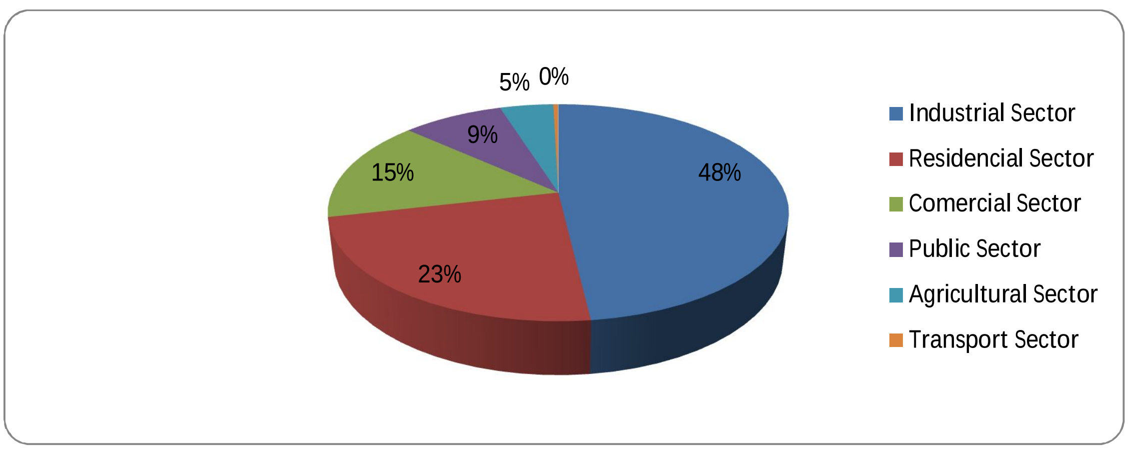

Figure 1 - Distribution of energy consumption for electricity sector.

Through technology split (separate) for air conditioners, the risk of leaks from the installation process significantly increased compared to the compact units or "window". This occurs because the refrigerant charge is performed at installation of the system, which does not occur in compact units, where the refrigerant charge is added during a controlled manufacturing process. Therefore, any reduction in refrigerant charge of air conditioning systems, promote consequently, the proportional reduction in emissions of harmful gases to the greenhouse effect.

The air conditioner split of residential and commercial application since 1998 have become more popular in Brazil. The Fig. 2 shows schematically the main components of an air conditioning: compressor, evaporator unit, condenser unit and expansion device.

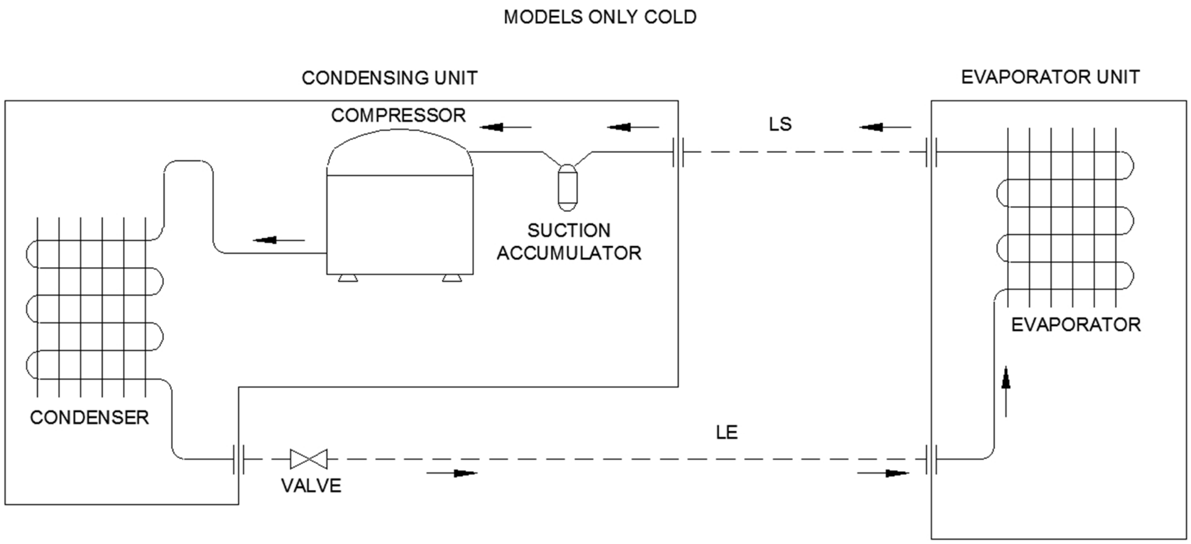

Figure 2 - Components of an air conditioner**.**

Attentive consumer requirements technology split brought the separation of evaporator and condenser unit to:

a) Reduce the noise from the compression process within the air-conditioned environments;
b) Eliminate the openings in the walls of the buildings;
c) Expand the capacity of the apparatus;
d) Become more aesthetic.

This requires the union of two units through copper tubing, which carries the refrigerant in different energy levels, let's look at Fig. 3:

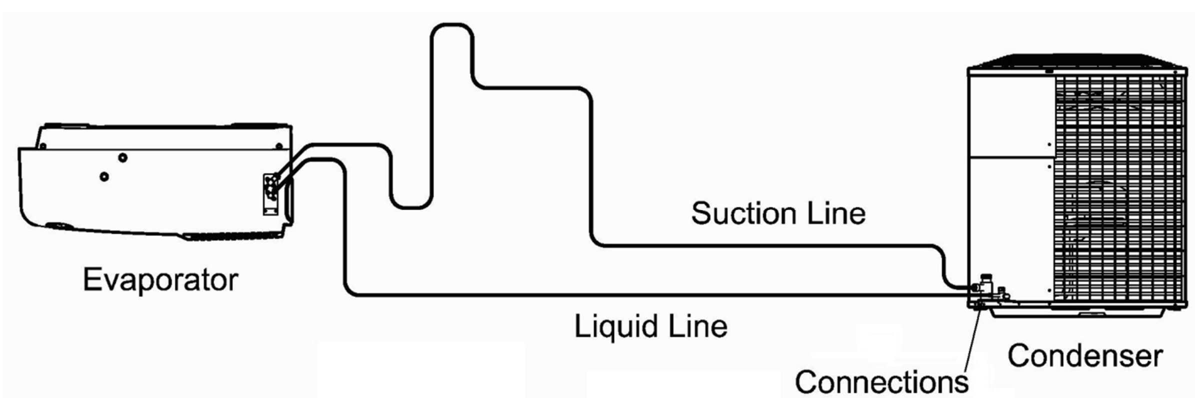

Figure 3 -Type air conditioning system split.

As in any process of refrigeration, air conditioning is based on a thermodynamic cycle of a particular refrigerant called: compression cycle steam or cooling cycle. The cycle has two pressure levels well defined: the condensation pressure (high pressure) and evaporation pressure (low pressure). This gradient provides the change of physical state of the refrigerant, allowing it to reject or absorb heat. For this reason, the suction and liquid have different temperatures, which in the conventional method of installation are prevented from interacting energetically. This occurs due to individual isolation insulating pipes with special LDPE (low density polyethylene foam) to prevent heat exchange between the tubes and/or external environment.

Anyway, are known the benefits of using a heat exchanger line/suction line (SLHX) for the use of energy present in pipelines. The three main benefits are referenced below, ASHRAE (1998):

a) Elevation of COP (coefficient of performance) due to a greater sub-cooling the refrigerant which in turn promotes an increase in the refrigeration effect;
b) Reduction bubble at the beginning of the process of expansion that could impede the flow of the refrigerant;
c) Evaporate completely residual fluid present in the compressor suction.

However the literature alert for some refrigerants fluids the coefficient of performance may be reduced by the fact of coolant in the aspiration of compressor have greater specific volume. This way, the effects are clear, so that the heat exchanger does not provide advantages of thermodynamic point of view. The Fig. 4 represents schematically the heat exchanger.

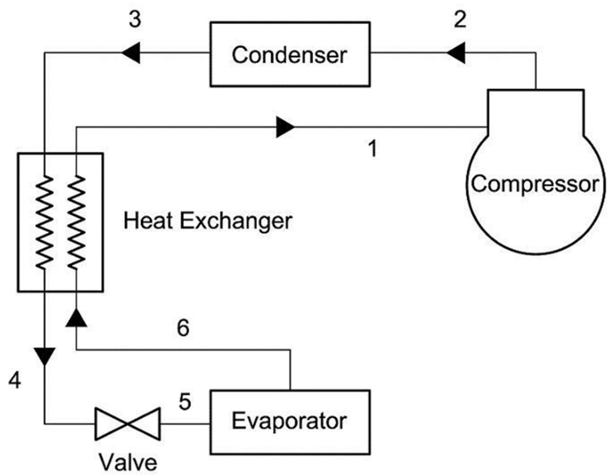

Figure 4. Schematic representation of the heat exchanger.

Anyway, the makers of air conditioners split indicate in their manuals that the addition of refrigerant systems to run until the attainment of a certain superheating of the steam. Superheating indicated by most manufacturers is 5°C to 7°C, confirming that the priority is to guarantee the life of the compressor. Due to the cost of the compressor in relation to other components, it avoids any form of compression wet problems and therefore dilution of lubricant. Below, the Fig. 5 illustrates the measurement points of temperature and pressure for the addition of refrigerant based on the evaporator's superheating.

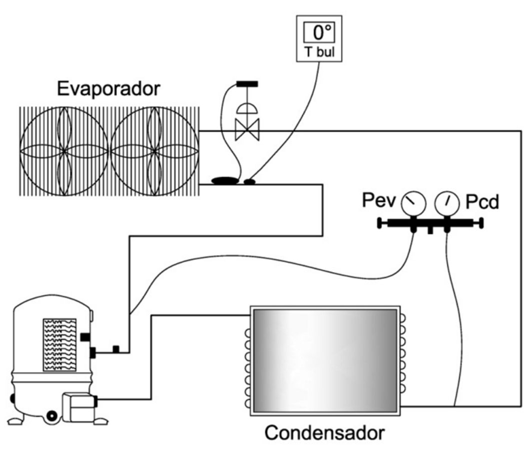

Figure 5. Measurement of the evaporator's superheating.

The articles surveyed are based on analysis of systems with fixed refrigerant charge for the comparison of systems using SLHX and those who do not use it. However, it is important to know the impact of SLHX over charge of refrigerant added to the system, since the load may vary. The purpose of this study is testing an air conditioner for the comparison of results between the thermodynamic conventional method of installation and the method that employs the SLHX, adopting fluid charge variable.

2. EXPERIMENTAL APPARATUS

The Fig. 6 shows the experimental apparatus used in this work that consists of a bench composed of an air-conditioning unit capacity 30.000 Btu/h, monitored by sensors for humidity and temperature. The discharge and suction of evaporator were mounted with isolated parts of galvanized steel in order to facilitate the measurements. The bench was tested in house thermal hydraulic laboratory at the Federal University of Paraná (UFPR), which has two distinct compartments with controlled temperature, simulating a calorimeter psychometric. A data acquisition system computation was used for the storage of measurements of temperature, relative humidity and air speed in real time. It is a system composed of a computer and two PCI cards; a multimeter, called Ni-4060 and control, Ni-6703, manufactured by National Instruments™. The supervisory system is the LabView 8.0.

Figure 6. Experimental apparatus.

The Fig. 7 shows the location of measurement points in experimental apparatus. The meter had the purpose of measuring temperature *(T)*, relative humidity (∅), pressure *(P)*, electrical current *(I)* and tension *(V)* for the determination of parameters used for evaluating the thermodynamic cycle. For the measurement of power of compressor was used a wattmeter of ESB Meters, manufacturing Mod. Saga 2300. The pressure of the refrigerant was measured through a manifold two-way and three hoses, with analog gauges for R22 directly connected appliance. valves

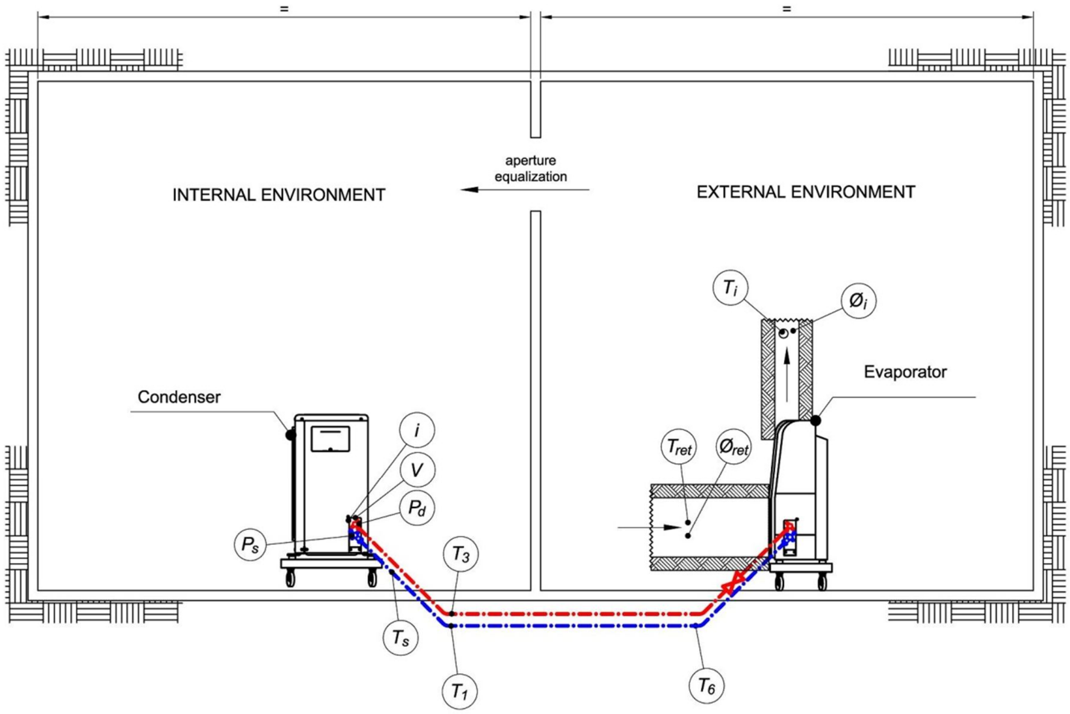

Figure 7. Schematic diagram of placement of Thermistors and relative humidity sensors in the system.

The thermal sensors used in this study were thermistors or semiconductor sensors. They are composed of manganese, nickel and cobalt oxide, which are mixed in equal proportions, getting the form of a small sphere 2.4 mm in diameter. Thermistors are nonlinear, when compared with other temperature sensors that are linear, e.g. sensors drivers of Platinum and tungsten. The relationship between resistance and temperature in a thermistor is well correlated by Eq. (1):

$$R = R_0 \exp\left[\beta\left(\frac{1}{T} - \frac{1}{T_0}\right)\right] \quad (1)$$

where,

R = electrical resistance, thermistor, Ω
R_0 = electrical resistance of the thermistor standard temperature, Ω
β = thermistor constants , K
T = thermistor temperature, K
T_0 = standard thermistor temperature, K

The thermistors were calibrated in a laboratory to determine their intrinsic limits ("Bias limits"). For both, were immersed in a thermal bath temperature constant brand Hakke, CD30 model. A total of 64 temperature measurement was performed at levels 0, 10, 20, ..., 100°C. The largest of these measurements standard deviation was 0.001 ° C and, therefore, the intrinsic limit of every thermistor was established as ±0.001°C, which is consistent with the precision intrinsic limit listed on a manual instrumentation Dally et al. (1993).

Relative humidity sensors are capacitive type, model HU1015NA, manufactured by the company Crown industrial Co., USA. Its intrinsic limit of accuracy is ± 5%, showing high reliability and repeatability. The sensor allows direct playback of relative humidity, through an output voltage, with roughly quadratic behavior, defined by Eq. (2):

$$V_{su} = -0{,}00009\,\phi^2 + 0{,}0323\,\phi + 0{,}581 \quad (2)$$

where,

V_{su} = Voltage, V

ϕ = Relative humidity, %

The Fig. 8a and 8b show respectively the thermistors high precision and relative humidity sensors used in this work. The sensors are connected to the data acquisition system for the measurement of temperature and humidity for the subsequent calculations.

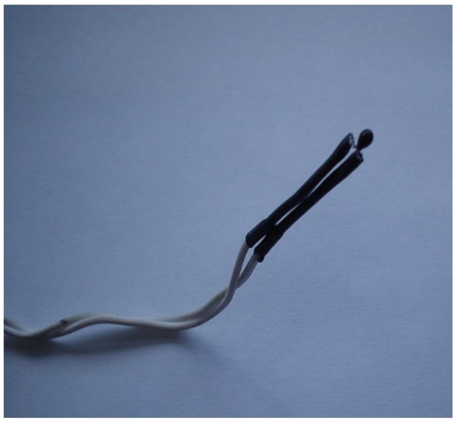

Figure 8a. Thermistor.

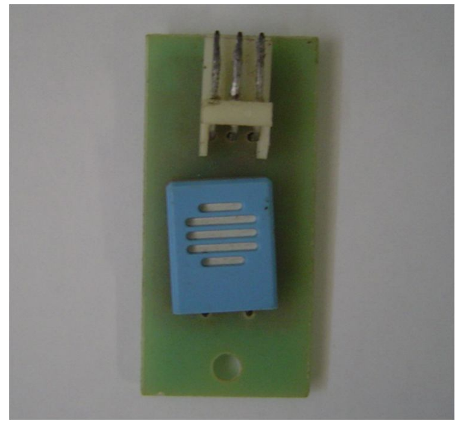

Figure 8b. Relative humidity sensor.

3. EXPERIMENTAL PROCEDURE

According to Fig. 7, each unit (evaporator and condenser) was experienced in separate compartments, with controlled temperatures respectively 26°C and 25°C. The units are joined by two hard copper pipes with track gauges of 3/8" and 3/4" (wall thickness of 1/32") manufacturing of Eluma S.A. The insulating used in pipes is the Polipex Plus ® UV with wall thickness of 10 mm and manufactured in polyethylene of low density polyethylene film coated with additives.

The bench was tested by the conventional method of installation (type I) and also according to the alternate (type II) that uses the heat exchanger. The changer was obtained by Union welded tubing and suction, isolated liquid together according to Fig. 9. To investigate the influence of exchanger were tested three types of sample based on distance between condenser and evaporator:

a) Type I: 1,0m of distance between the units;
b) Type II: 1,5m of distance between the units;
c) Type III: 2,0m of distance between the units;
d) Type IV: 2,5m of distance between the units;
e) Type V: 3,0m of distance between the units.

This way, the effects of the effectiveness of SLHX on the load of soda and thermodynamic performance could also be assessed. The effectiveness, ε, is set in Eq. (3):

$$\varepsilon = \frac{(T_1 - T_6)}{(T_3 - T_6)} \tag{3}$$

Where numeric values entered into subscripts temperatures *(T)* correspond to points represented in Fig. 4. For the acquisition of temperatures of Eq. (3), thermistors were glued to the pipeline through a thermal paste.

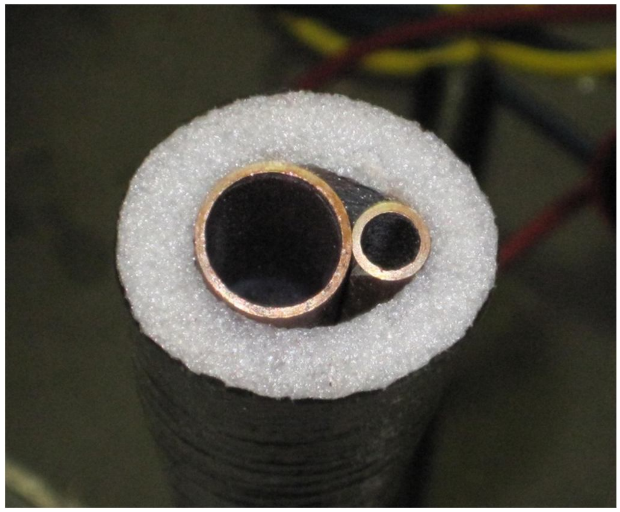

Figure 9. Heat exchanger

The air-conditioning efficiency was measured by the coefficient of performance, *COP*, which is the more applied to evaluate cooling systems. The *COP* is defined by Eq. (4):

$$COP = \frac{\dot{Q}_e}{\dot{P}_{cp}} \tag{4}$$

Where,
Q_e = Cooling capacity, W
$P_{el\ Total}$ = Total electrical power consumed by the compressor, W

To determine the cooling capacity, Q_e, was chosen by psycrometric method, which consists of the knowledge of the return air enthalpy and inflated by evaporation plant. Thus, the cooling capacity was determined by Eq. (5):

$$\dot{Q}_e = \frac{\dot{V}}{v}(h_{ins} - h_{ret}) \tag{5}$$

Where,
Q_e = Cooling capacity, W
V = Volumetric flow rate of air, m^3/s
ν = Specific volume of air, m^3/kg
h_{ins} = Inflate air Enthalpy, J/kg
h_{ret} = Return air Enthalpy, J/kg

The power of compression, $P_{el\ Total}$, was determined from reading the pot with the system on a permanent basis. All psycrometric calculations were based on standard atmospheric pressure of the city of Curitiba, Paraná-Brazil, according to ASHRAE (2001)-90, 88kPa and resolution of Eq. (5) was performed through the data provided by software CYT Soft Psycrometric®.

The volumetric flow rate of evaporation plant was known by measuring the speed of flow-through by a vane anemometer, this was followed by the ASHRAE's recommendation (2001, pp. 14.16.17). For each test, the load was measured by weighing refrigerant cylinder R22 before and after the procedure for adding. Stop criterion of the process of adding refrigerant was the degree of superheating, when the same was achieved the cylinder was closed. The parameter adopted to the extent of superheating was the range of 5°C to 7°C. The procedure of opening and closing cylinder R22 was manual, and based on suction temperature reading, T_s, measure directly on the surface of the suction tubing, subtracted of suction pressure, P_s, converted into evaporation temperature.

4. RESULTS AND DISCUSSION

The experimental measurements considered were purchased with the air-conditioning system and on a permanent basis. The criterion of stability of the temperature of the chamber and temperatures measured system was adopted for the determination of this regime. During a time measuring points identified in Fig. 7 were monitored and the results stored for later analysis. Once the system was superheating specified operator waited until it stabilizes the charge cylinder were definitely closed. In all cases where the degree of superheating remained virtually constant, oscillating between ± 0.3°C on a permanent basis. For samples of type II the degree of superheating had greater variation ± 0.5° C. Only after this step closed the cylinder was weighed on a calibrated digital scale. It should be emphasized that this phase was the most critical from the standpoint of measurement, since it depended heavily on the sensibility of the performer.

The Fig. 10 shows the relationship of the change in refrigerant charge, *m'/m*, depending on the effectiveness of the heat exchanger, where, *m'*, represents the refrigerant charge required by the system SLHX. The points on the graph represent the comparison of the average of three replicates for each sample. Thus, it is possible to compare methods of installation from the point of view of the variation of refrigerant charge, the main purpose of the research

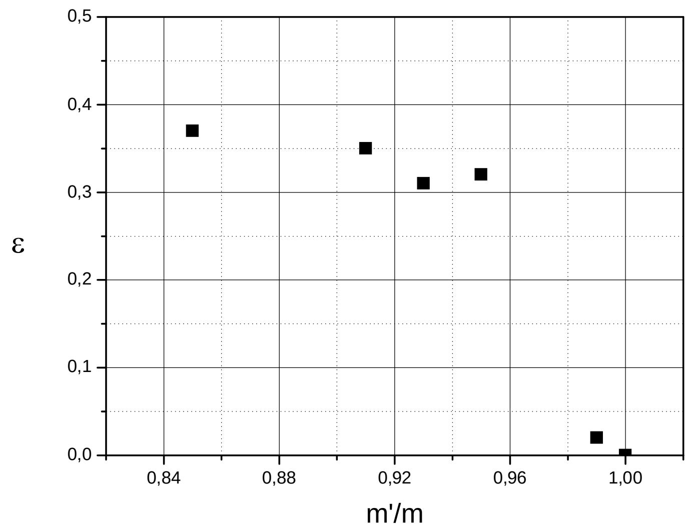

Figure 10. Charge variation of refrigerant for the effectiveness of the changer.

The Figure 11 shows the relationship between the effectiveness and the change in *COP*, and, *COP'*, system performance with SLHX. As the state psychrometric insufflation air and return of evaporation varied very little between each test, and thus the cooling capacity. The points on the graph represent the average of three replicates for each sample.

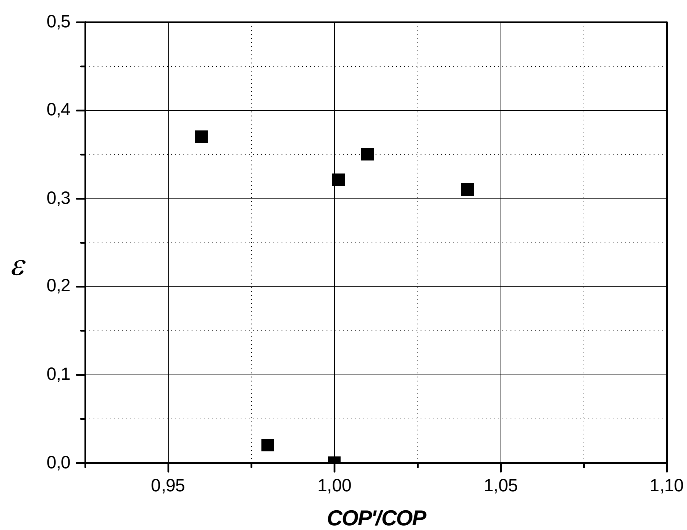

Figure 11. Variation of effectiveness with the *COP*.

5. CONCLUSIONS

As the results, note that for the range of effectiveness experienced, the refrigerant charge was reduced in all cases. The largest reduction reached 15%, this corresponds approximately 275g of refrigerant. Even for low effectiveness decreased 1%. As the variation of the *COP*, there was an increase of up to 5.0% maximum and minimum reduction of 4.0%. Therefore, from the standpoint of the refrigerant charge SLHX promotes a considerable reduction to a nearly constant COP.

For the sample I obtained the exchanger effectiveness and low temperatures T_1 and T_6 were very close. As for the other samples were obtained effectiveness up to 37% and the difference reached 6.0°C. The case was great for the show type III, where the reduction of refrigerant charge was 9.0% and increased *COP* of 1.0% compared to mode I. This gain of *COP* was due mainly to a reduction of power compression, because the cooling capacity varied little. It was found that the power consumed by the compressor is more dependent on mass flow rate than the temperature of suction and discharge.

Finally, it is important to emphasize that it is necessary to widen the search for greater effectiveness of the heat exchanger. Other fluids may also be a source of research, since the R22 legislation has restrictions. Aspects such as the quantification of financial costs resulting from the use of SLHX may also be addressed.

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