

# OPTIMIZATION OF CASCADE REFRIGERATION SYSTEM TO ACHIEVE LOWER TEMPERATURES

## OTIMIZAÇÃO DE SISTEMA DE REFRIGERAÇÃO CASCATA PARA ATINGIR BAIXÍSSIMAS TEMPERATURAS

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**Abstract:** Achieve ultra-low temperatures is usually possible due to the use of cascade configurations refrigeration systems. Commonly, the observed results do not match the expected theoretical predictions. Therefore, in order to increase thermal efficiency and reach lower temperatures, adjustments and modifications in mechanical components of refrigeration system, in refrigerant fluids charges, and in general parts arrangement can be done. The system presented herein features a cascade configuration that uses R-404a and R-508b, and was previously able to reach  $-70^{\circ}\text{C}$ . This study investigates the impact of optimization changes made in this existing cascade system, regarding the use of a more powerful compressor, in order to achieve  $-86^{\circ}\text{C}$  inside the freezer. A series of tests was conducted for each one of the changes, and this paper presents the complete methodology used, as well as the results found. As we will be able to notice, the results did not match the expected temperature, but contributed to the evolution of the experiment.

**Keywords:** cascade systems, refrigeration, ultra-low temperature.

**Resumo:** Atingir temperaturas extremamente baixas geralmente é possível devido ao uso de configurações em cascata em sistemas de refrigeração. Geralmente, os resultados observados não correspondem às previsões teóricas esperadas. Portanto, para aumentar a eficiência térmica e atingir temperaturas mais baixas, é possível fazer ajustes e modificações nos componentes mecânicos do sistema de refrigeração, na carga dos fluidos refrigerantes e no arranjo geral das peças. O sistema aqui apresentado apresenta uma configuração em cascata que utiliza R-404a e R-508b, e anteriormente era capaz de atingir  $-70^{\circ}\text{C}$ . Este estudo investiga o impacto das mudanças de otimização feitas neste sistema em cascata existente, em relação ao uso de um compressor mais potente, a fim de atingir  $-86^{\circ}\text{C}$  dentro do freezer. Uma série de testes completos foi realizada para cada uma das alterações, e este artigo apresenta a metodologia completa utilizada, bem como os resultados encontrados. Como poderemos observar, os resultados não corresponderam à temperatura esperada, mas contribuíram para a evolução do experimento.

**Palavras-chave:** Sistemas cascata, refrigeração, baixíssimas temperaturas

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# 1 INTRODUCTION

The depletion of the ozone layer caused by the use of synthetic fluids over the past decades, especially the CFC (notably used in the refrigeration and climatization industries), led to a global concern about their substitution. In Brazil, due to International Environmental Conventions and Decisions, above all the Montreal Protocol, that took place over the last 30 years, most of commercial refrigeration cycles do not use CFCs anymore. The refrigerant fluids applied are usually HFCs, that are harmless to the ozone layer.

Recent research and development activity is focusing on fluorinated propene (propylene) isomers as potential refrigerants possessing low global warming potentials (GWPs). Propene isomers contain an unsaturated (carbon-carbon double) bond and can also be referred to as olefins or alkenes. In addition to these isomers carrying the designation R, the fluorinated isomers also can be identified with the prefixes HFO (hydrofluoro-olefin), HFA (hydrofluoroalkene), or HFC (hydrofluorocarbon). For example, R-1234yf could also be referred to as HFO-1234yf, HFA-1234yf, or HFC-1234yf (BROWN, 2009).

The refrigerant selection process for refrigeration systems should consider environmental aspects, efficiency and cost factors. Ideally, the refrigerant fluid used should be non-flammable, environmental friendly, and lead the system to high exergetic efficiency (NASRUDDIN et al., 2016). A suitable refrigerant couple is able to provide a large temperature lift while improving system performance (SUN et al., 2016).

Tellier first proposed the concept of cascade refrigeration in 1867, but low temperatures were probably not in demand at that time (BANSAL and JAIN, 2007). Nowadays, supplying very-low and ultra-low temperatures, required in some scientific research fields, industrial processes and perishable goods storage is, sometimes, essential (NASRUDDIN et al., 2016). Garimella et al. (2011) studied a cascade absorption/compression cycle for naval ship application. Ghorbani et al. (2016) considered cascade refrigeration systems for

integrated cryogenic natural gas process, liquefied natural gas and nitrogen rejection unit.

Single stage refrigeration cycles face a difficulty in reaching such low temperatures and yet remain economically viable. These systems also have limitations regarding large temperature differences between the climatized space and the outside environment. There is a high compression ratio needed in these applications, which increases the discharge temperature of the lubricant and refrigerant fluid, decreasing the volumetric efficiency of the compressor and the system's COP (coefficient of performance) (NASRUDDIN et al., 2016).

The use of cascade refrigeration systems is an effective alternative to systems that require large temperature differences and are widely used in applications that need temperatures below  $-60^{\circ}\text{C}$  (BHATTACHARYYA et al., 2005). Such systems also offer an additional degree of optimization capacity, because of the additional iterations between changes in all the existing stages. In cascade systems, two or more single-stage units are coupled to each other through cascade heat exchanger, working as the evaporator of the High Temperature Circuit (HTC), and, at the same time, as a condensing medium for the Low Temperature Circuit (LTC) (NASRUDDIN et al., 2016).

The results expected from a theoretical system are not always reached in practice, due to wrong considerations, mistakes made during the design and financial and commercial difficulties, for example. For that reason, some practical changes and optimizations have to be done during the conception and assembly of refrigeration systems.

Yamaguchi (2016) studied the improvement of energy efficiency of a cascade system with  $\text{CO}_2$  as refrigerant. Mohammadi and Ameri (2014) used different cooling strategies and compared energy and exergy of a cascade air conditioning system. Asgari et al. (2017) dealt with advanced exergy and exergoeconomic analysis and multi objective optimization of internal auto-cascade refrigeration cycle. Llopis et al. (2016) analyzed and quantified the effects caused by the use of an internal heat exchanger at the LTC in a cascade

refrigeration plant. Yilmaz et al. (2014) performed a thermodynamic analysis of a two stage sub-critical cascade refrigeration system using CO<sub>2</sub> and R404a in order to maximize the Coefficient of Performance (COP). Mumanachit et al. (2012) compared the energy performance and saving of a direct two-stage ammonia system to an ammonia-carbon dioxide cascade system for low temperature (below 40 C) applications.

Instead of R-404a and R-508b fluids, used in this work, Silva et al. (2012) used R-404a in the HTC in a supermarket application. Bansal and Jain (2007) mentioned the use of R-508b in cascade systems to ultra-low systems.

At Institute Federal of Bahia (IFBA), a freezer that features cascade systems was designed and built, using R-404a as HTC fluid, and R-508b as LTC fluid. Initially, it was able to achieve temperatures below -70°C, even though, it has been conceived to reach -86°C. Therefore, in this experiment, optimizations and adjustments that aim to reach -86°C inside the freezer was made and the results are now presented. The strategy consisted in the exchange of the LTC compressor of the freezer for a more powerful one. Then subcooling and superheating adjustments was made in the system.

## 2 THEORETICAL FRAMEWORK

Regarding refrigeration systems that involve high temperature differences between the heat source and the heat sink, the use of single stage systems does not bring performance gain, because the existence of high-pressure ratios leads to a low volumetric efficiency of the compressors, causing a low COP (AMINYAVARI et al., 2014).

Eini et al. (2016) showed that when there is a large temperature difference between hot and cold environment, the evaporator pressure decreases, leading to air leakage into the system, and to an increment in suction volume and condenser pressure, which requires robust design of pipes and fitting. Air is a non-condensable gas and impact on the correct function of

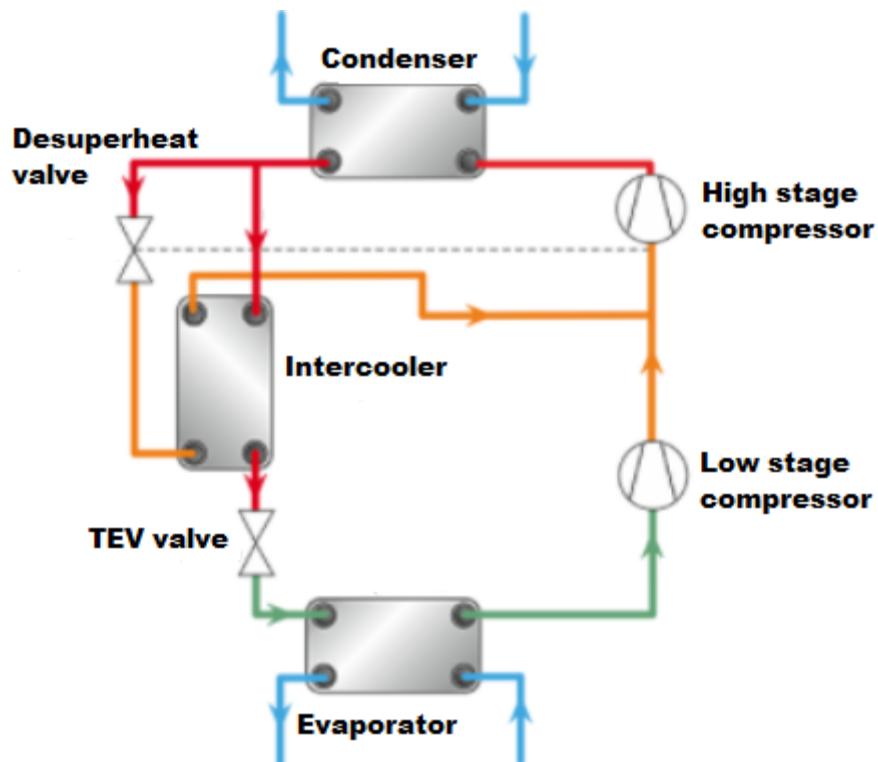


the two-stage system decays more rapidly, comparing to cascade systems at lower suction pressures, due to the fast decline in the booster compressor's mass flow coupled with its decreasing volumetric efficiency (SWEPE, 2015).

## 2.2 Intercooling Systems

The performance of a single stage vapor compression cycle deteriorates at high heat rejection and low evaporation temperatures. This fact is usually overcome by employing multi staging. The use of intercooling with an external fluid between the compression stages, reduce the specific volumes and the discharge temperatures, which, as a consequence, reduces the work input to the compressor. For these applications, the selection of the inter-stage pressure is a key parameter for enhancing the performance of the system (PUROHIT et al., 2015). A schematic layout of an intercooler system is shown in Figure 2.

Figure 2 - Schematic layout of an intercooler system



Source: SWEPE (2015).

Refrigeration coefficient of theoretical cycle of multistage compression system with intercooling is always higher than single-stage compression. By cooling the refrigerant vapor after the first compressor, the discharge gas leaving the high-stage compressor can be held at an acceptable temperature level. The intermediate cooling also increases the compressor efficiency, which reduces the compressor power consumption (KARELIN et al., 2017).

## **2.2 Cascade Systems**

Bansal and Jain (2007) defined a two-circuit cascade system as two independent refrigerating units coupled through a heat exchanger, where the refrigerant vapors of the LTC are condensed, rejecting heat to the refrigerant in the HTC. In order to balance both cycles, the heat rejected by LTC must be absorbed by the HTC. They said that each circuit has a different refrigerant suitable for its temperature. The lower temperature units progressively using lower boiling point refrigerants. The lower boiling point refrigerant will have a higher saturation pressure at low temperatures that keeps the ingress of air under control and requires a smaller compressor for the same refrigerating effect due to higher density of suction vapors. Generally, two-circuit and rarely three-circuit cascade systems are used.

One of the advantages of a cascade system when it comes to achieving very low temperatures is that the two stages do not necessarily contain the same refrigerant fluid. A refrigerant with a higher vapor pressure can be used in the LTC, and a refrigerant with a lower vapor pressure can be used in the HTC.

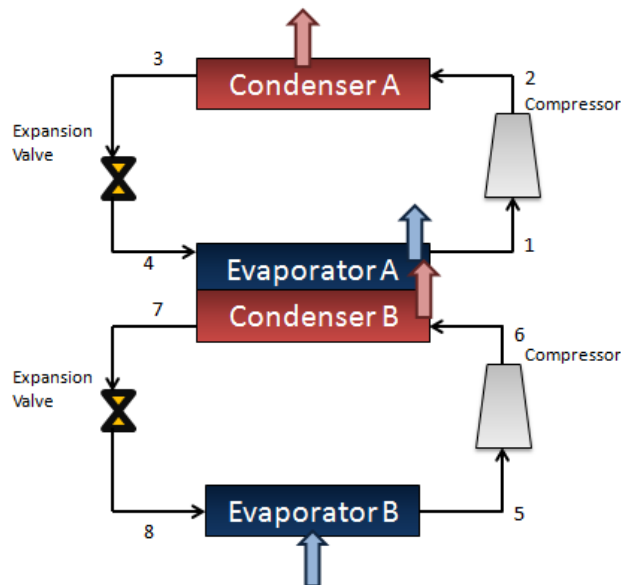
The cascade system can solve several problems coming from the high-pressure ratio in low temperature refrigeration systems (PARK et al., 2013).

In general, if the desired temperature can be easily achieved in a single-stage machine, it will be more efficient than a cascade system due to irreversibility and losses associated with a large number of components (BANSAL and JAIN, 2007).

Multi-stage refrigeration cycles can also achieve very low temperatures efficiently, but there are some major concerns in comparison to cascade cycle. In multi-stage refrigeration, the same refrigerant must work at the highest and the lowest pressure levels. The selection of refrigerant to avoid excessively large pressures in the ambient condenser and evaporation pressures below one atmosphere in the cold evaporator, can be difficult. Vacuum should always be avoided, because it increases the risk of air and moisture leaking into the system, leading to reduced system performance and increased wear on components (SWEF, 2015).

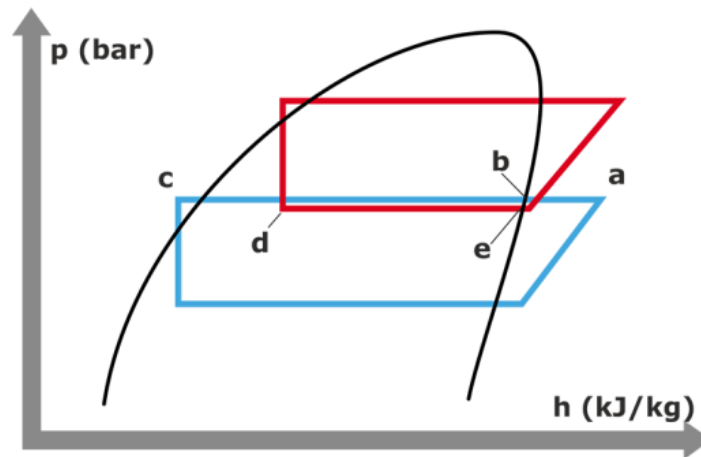
Refrigerant selection and oil distribution for a cascade system can be dealt separately for each circuit. It is important to notice that the cascade heat exchanger will be exposed to temperature and pressure fluctuations. Figure 3 and Figure 4 show, respectively, the lay out and a P/h diagram of a cascade system.

**Figure 3.** - Schematic layout of a cascade system



**Source:** Ribeiro et al. (2017)

**Figure 4.** - A log P/h diagram of a cascade system showing the LTC in blue and the HTC in red.



Source: SWEF (2015)

In the process of a cascade system design, one of the most important issues is to find an optimal intermediate temperature, which is the evaporating temperature of a high temperature cycle or the condensing temperature of a low temperature cycle, to obtain the best efficiency of the system (PARK et al., 2013).

The intermediate temperature between the two cascade circuits is a design parameter that plays an important role to determine the coefficient of performance (COP) of the overall system. For reversible cycles, assuming no temperature difference between the two fluids in the cascade condenser, the optimum cascade temperature is the geometric mean of the condensing and evaporating temperatures of the cascade system, i.e., to have the same temperature ratio in each of the circuits. However, it has been shown that the optimum temperature depends on the type of refrigerants used in the two circuits, though the actual values may not differ from ideal values by more than 10%. The larger the temperature difference, the lower the COP of the system. A comparison with R-404A shows energy savings only at low outdoor temperatures up to about 14°C (BANSAL and JAIN, 2007).

Further, the assumption of a single temperature for both the fluids is not practically feasible, as it would demand infinite area for the cascade condenser. The optimum temperature difference between the two fluids should depend not only on the heat transfer characteristics of the refrigerants in the two circuits but

also on the economics of the design (operating versus capital cost). The larger the temperature difference, the lower the COP of the system (BANSAL and JAIN, 2007).

### 3 MATERIALS AND METHODS

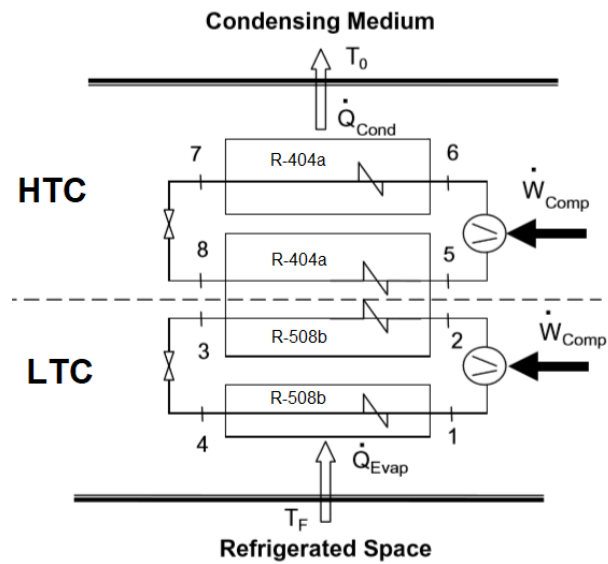
#### 3.1 System Description

The existing cascade system, discussed in this work, is part of a freezer located at IFBA – Campus Salvador, inside the Refrigeration Workshop, in Brazil. This freezer was manufactured by some of IFBA's professors, students and technicians in order to achieve -86°C internally.

Before this study, the system featured a 1,25 HP alternative hermetic compressor in the HTC, a 1 HP alternative hermetic compressor in the LTC, a shell and coil heat exchanger, and a 0,036" capillary expansion device. The walls are composed of one external galvanized steel sheet, one layer of vacuum panel, PU (expanded polyurethane) insulation and one internal galvanized steel sheet. The coil, made of a 3/8" copper tube with 38m length, seats between the PU insulation and the internal galvanized steel sheet.

The cascade system is made of two single-stage systems coupled by an intermediate heat exchanger (cascade heat exchanger), where the refrigerant vapors of the LTC are condensed, and the heat is rejected to the refrigerant in the HTC. A schematic diagram of the cascade system, focus of this work, is shown in Figure 5.

**Figure 5** - Schematic diagram of the R-404a/R-508b cascade refrigeration system.



Source: Dopazo et al. (2009)

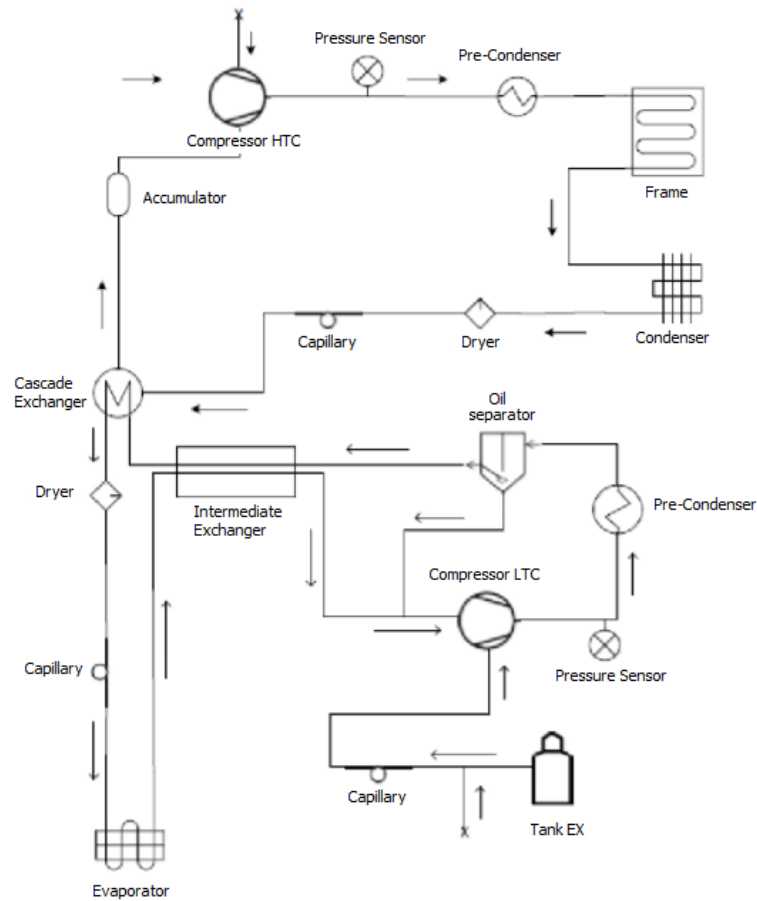
The HTC uses R-404a and the LTC uses R-508b as refrigerant fluids respectively. In Figures 6 to 8, a general overview of the freezer and the cascade system is shown.

Figure 6 - Freezer front, showing main digital display, and internal chamber.



Source: The Autors (2020)

Figure 7 - Cascade system flow components



**Source:** The Autors (2020)

**Figure 8** – Condensing unit



**Source:** The Autors (2020)

As designed, the system was not able to reach the desired internal temperature during the tests, the lowest temperature achieved was  $-65^{\circ}\text{C}$ . Therefore, the optimization was proposed.

The original LTC compressor was replaced by a more powerful one (1.25 HP instead of the former 1 HP). Thus, the heat transfer between the R-404a and the R-508b and the fluid flow inside the coil will be increased, making it possible to remove more heat from internal part of the freezer.

### 3.2 Experiment Details

During the experiment, it was crucially to take pressure and temperature measurements in key spots, because, in order to achieve the internal temperature goal, the amount of R-404a and R-508b put, respectively, into the HTC and into the LCT had to be empirically adjusted.

The freezer features temperature and pressure sensors. The measurements made by them are digitally displayed. Inside the internal chamber, the temperature measurements are performed in three points. there was a list of the external temperature measurement points:

- HTC Compressor suction;
- LTC Compressor suction;
- HTC Capillary entrance;
- LTC Capillary entrance;
- Coil outlet.

Regarding the pressure, two analogic manifolds were used to measure both Suction and Discharge pressures of the HTC and LTC compressors. Besides that, redundant measurements were taken by two pressure sensors existing in the freezer.

- HTC Compressor suction, Manifold 1 (blue);
- HTC Compressor discharge, Manifold 1 (red) and internal pressure sensor 1;
- LTC Compressor suction, Manifold 2 (blue);
- LTC Compressor discharge, Manifold 2 (red) and internal pressure sensor 2.

Electric currents of compressors were also measured by ammeter pliers. Figures 9 to 11 illustrated the main and the auxiliary display, and show pictures of the experimental preparation.

**Figure 9** - Main Display, showing the locations of the internal temperature measurement spots.



**Source:** The Autors (2020)

**Figure10** - Auxiliary Display, showing real time temperature measurements.



**Source:** The Autors (2020)

**Figure 11** - Details of experimental preparation.



**Source:** The Autors (2020)

The main goal of the installation of the 1.25 HP LTC alternative hermetic compressor was to increase the mass flow of R-508b. This way the fluid would be able to extract more heat from the internal chamber of the freezer, increasing its heat extraction rate and lowering its temperature. Also, the heat exchange in the cascade heat exchanger would be optimized. The coil, where the R-508b runs, is located inside the walls of the freezer, between the PU insulation and the inner galvanized steel sheet.

During the operation startup the gas load in the HTC and in the LTC, were carefully adjusted, because specific volumes of the refrigerant fluids were too high before the compressors actuate and reduce them. To avoid the compressors maximum pressure safety shutdown, the gas load was gradually increased in both HTC and LTC, based on amounts used previous to the tests. Superheat and subcooling parameters above-mentioned were adjusted by adding or removing gas.

Adjustments regarding the superheating and subcooling were made in both HTC and LTC, in order to make sure that there is only superheated vapor in the compressors suction line and only subcooled liquid before the evaporator (coil).

## 4 RESULTS AND DISCUSSION

In the HTC, 1.2 kg of R-404a was added to the system. When R-404A is used, liquid charging is mandatory. R-404A refrigerant cylinders have a dip tube and liquid is charged with the cylinder upright. In the LTC 1.4 kg of R-508b was added alongside with 30g of N-pentane, used to stabilize the LTC compressor oil viscosity in order to ensure that it circulates well throughout the system, regardless of the large temperature variation it is subjected to. In the Table 1, all the measurements results are presented.

**Table 1 - Results.**

<b>Measurement Spot</b>	<b>Temperature (°C)</b>	<b>Pressure (Bar)</b>
HTC Compressor suction	-6.2	0.14
HTC Compressor discharge	-	16.1
LTC Compressor suction	-13	Vacuum
LTC Compressor Discharge	-	11.5
HTC Capillary entrance	-	-
LTC Capillary entrance	-31.7	-
Coil outlet	-60	-
LTC Cascade inlet	-24.4	-
Internal Chamber	-80	-

**Source:** The Autors (2020)

The freezers overall electric current was 6.7 A. The HTC superheat measures during the tests were stable and close to 37.4 °C. The LTC Subcooling measured was unstable, and fluctuating between 1°C and 3.4 °C.

Over the intermediate heat exchangers, in the LTC, the R-508b enters from the compressor unit with -13°C and from the coil outlet at -60°C.

The internal temperature inside the freezer of -80°C was obtained. Despite not reaching the desired temperature of -86°C, there was an advance in the equipment's settings, which will allow reaching the desired temperature with small adjustments.

## **5 FINAL CONSIDERATIONS**

Tests results showed that the main goal of this work was not achieved. The freezer could only reach -80°C inside its internal chamber.

An analysis made in order to identify the root cause to this issue leads to two main issues. First, the capillary was not changed alongside with the compressor. The capillary used in the testes presented in this work was the same used with the former, and less powerful, compressor.

The second cause regards the freezers insulation. During the tests were observed spots where water was condensing on the outer walls of the freezer. Therefore, an excessive heat flow between the internals and the outside of the freezer was created, reducing the systems refrigeration capacity.

The subcooling measured showed some instability during the tests. Therefore, strategies to guarantee a stable value must be applied for further testing.

In future works and experiments, a new LTC capillary will be installed, and thermal insulation will be reinforced, in order to achieve lower temperatures.

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