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Article

# Employment of Computational Analysis to Reduces Electric Energy Consumption of an Ammonia Refrigeration System at a Peak Time

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**Abstract:**

Aspects involving the maximization of the coefficient of performance (COP) of industrial units is a topic of high importance, due to actually exists onerous tariffs of electric energy and any reduction in consumption, results in great improvements not just for the company, but also to the energetic matrix of the country, since large refrigeration units have a high electrical energy consumption. Thereby, this study has the objective of improving the COP in a refrigeration system that uses $NH_3$ as a refrigerant fluid, through the use of mathematical modeling and computational simulation of the system in analyze. Furthermore, the electrical motors of the compressors are driven with variable speed drives (VSDs). Thus, were evaluated different conditions of regime temperature operation and values of frequency of the VSDs, with aim of obtaining the smallest electrical energy consumption, in order to meet the specific conditions of running of the installation. It was identified that the refrigeration system has some water contamination that also impacts the consumption of electrical energy. This study demonstrated that without realizing none structural modification the refrigeration system, it was possible to increase 24% the COP of the refrigeration system during the off-peak time while at peak time the percentage increase was of 63.5%. Comparing January and September months, the electrical energy save was 458438 kWh, being 80668 kWh at peak time. Moreover, comparing August and September months, the electrical energy save was of 214057 kWh, being 67983 kWh of this total consumed in peak time.

**Keywords:** Energy Efficiency; Computational simulation; Variable speed drives; Industrial refrigeration; COP.

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## 1. Introduction

The onerous tariffs related to electric energy charges in Brazil have motivated studies to implement better practices in many fields of industries with the objective of reducing operating costs and the environmental impacts caused by electricity generation. The principal aggravating elements of the energetic consumption are generally related to operating conditions, maintenance and the correct selection of the equipment. Achieving the best of refrigeration installation is a difficult task, whose practices normally used are based on trial and error or associated with the professional experience of technicians and engineers.

In this sense, the more rational use of electricity promoted by the implementation of parameters obtained through the computational simulation of each refrigeration system can be a big ally. Thus, with this methodology is possible to achieve greater coefficients of performance (COP) and predict the behavior of the industrial refrigeration system. Normally, the interventions are made in the pressures of operation, compressors and heat exchangers (evaporators and condensers.

The compressor is the component of the refrigeration system that has the biggest demand for electricity, to the point of significantly affect the operating cost of the installation. Therefore, is very important the correct definition of compression ratio, maintenance plans, proper selection of each component at the time of designing the system, in order to attend the pre-requisites operating conditions and ensure the best energy efficiency (Stoecker and Jabardo, 2002).

VSD is the most effective controller and energy saver for mechanical machines in industries. Modern VSDs are affordable, reliable, flexible, and offer significant electrical energy savings through greatly reduced electric bills. VSD increases efficiency by allowing motors to be operated at the ideal speed for every load condition. In many applications VSDs reduce motor electricity consumption by 30–60% (Saidur et al., 2012).

Plessis et al. (2013) and Peng and Du (2015) presented researches with different techniques to control the capacity of heat exchangers and compressor and the use of variable speed drive (VSDs) demonstrated better flexibility in operation and a greater reduction in electricity consumption compared to standard installation or others control techniques. Yu and Chang (2006) studied the use of fans with variable speed drive (VSD) control in the condenser, coupled with condensation temperature control and used together in an air-cooled screw chiller, allowed an operation with more efficiency and an annual reduction of electric energy consumption normalizes by the total area of the building floor from 56,2 kWh/m² to 44 kWh/m².

Widell and Eikevik (2010) realized an experimental analysis of compressor operation in a large refrigeration system with 5 screw compressor and ammonia as fluid refrigerant. Moreover, the authors developed an optimal model for compression operation with the objective of increase energy efficiency. The process of optimized the conditions of operations was performed with and without variable speed drives. The results demonstrated that the biggest saving in energy consumption has been achieved when not all the tunnels were load. With the optimizing of the operation is assumed that can be saved € 30000-50000 per year in the system analysed.

Acunha Jr et al. (2019) analysed strategies to improve the energy efficiency of industrial refrigeration system of a warehouse through the use of mathematical modelling and computational simulation using the software ESS. The electric motors of the evaporators and condensers are drive by VSDs. The authors evaluated different conditions of regime temperature operation and values of frequency of the VSDs. It was identified that the refrigeration system has presence of water contamination that also impact the consumption of electrical energy. This study demonstrated that without realize none structural modification the refrigeration system, is possible reduce in 9.3% the consumption of electric energy.

This study has the objective of identifying actions to promote the increase of efficiency energy of an industrial refrigeration system, through the use of mathematical modelling and computational simulations using the software EES (Engineering Equation Solver) as a tool to perform the calculations. Moreover, the electric motors of the screw compressors are driven by VSDs, in order to obtain the lowest electricity consumption in a specific condition of operation of the industrial system.

## 2. Materials and Methods

### *2.1. Mathematical Modeling*

#### 2.1.1. Evaporator

The efficiency of the external surface $\eta_e$ of a finned surface is evaluated in function to the surface areas finned $A_f$, the total external surface area $A_e$ and the efficiency of the fins $\eta_a$ as shown in Eq. (1) (Mcquiston et al., 1994)

$$\eta_e = 1 - \frac{A_f}{A_e}(1 - \eta_a) \tag{1}$$

The evaluation of the efficiency of the fins depends on how they are placed on the tubes of the heat exchanger. With the analogy of electrical assembly, consisting of a series of resistors, for smooth heat exchanger the Eq. (2) express the total thermal resistance $R_{total}$ in function of the tube thermal resistance $R_{tube}$, of contact $R_{cont}$, of incrustations $R_{incrust}$, of the oil present inside the tube $R_{oil}$, of the ice on the surface $R_{ice}$, of the surface external efficiency $\eta_e$, of the inside and external convective coefficients, respectively, $h_i$ (W m$^{-2}$ k$^{-1}$) and $h_e$ (W m$^{-2}$ k$^{-1}$), of the external area $A_e$ (m²) and internal $A_i$ (m²) of heat exchange.

$$R_{total} = \frac{1}{h_i A_i} + R_{tube} + R_{cont} + R_{incrust} + R_{oil} + R_{ice} + \frac{1}{\eta_e h_e A_e} \tag{2}$$

The overall external heat exchanger coefficient is demonstrated by Eq. (3).

$$\frac{1}{U_e A_e} = \frac{1}{h_i A_i} + R_{tube} + R_{cont} + R_{incrust} + R_{oil} + R_{ice} + \frac{1}{\eta_e h_e A_e} \tag{3}$$

In this study, for the evaporation region, it was considered a psychrometric process with air cooling and dehumidification and the coil in this area of the evaporator was frozen.

Considering the effect of resistance due to the contact resistance of the fin, the presence of oil, the incrustation, the presence of ice and efficiency of the fin as an equivalent resistance ($h_{eq}$ - corrected with support of experimental observations) it is possible to obtain the equation for overall heat exchange coefficient $U$ (W m$^{-2}$ k$^{-1}$) as shown in Eq. (4), trough the simplification of Eq. (3).

$$U = \frac{1}{\frac{d_e}{d_i}\left(\frac{1}{h_i}\right) + \frac{d_e}{d_i}\left(\frac{L}{K_t}\right) + \frac{1}{h_{eq}}} \tag{4}$$

Where $K_T$ is the thermal conductivity of the tube (W m$^{-1}$ k$^{-1}$), $d_i$ and $d_e$ are, respectively, the internal and external diameter of the tube (m), L is the thickness tube (m) and $h_{eq}$ can be obtained through experimental measurements and thermal balances, since the capacity $\dot{q}$ of the system can be determined resolving the Eq. (5).

$$UA = \int U dA_h = \frac{d\dot{q}}{LMTD} \tag{5}$$

*LMTD* (°C) is the logarithmic mean temperature difference between air and ammonia flows.

2.1.2. Compressor

The heat transfer rate in the evaporator must be compatible with that performed by the compressor and their capacity assessed using mathematical modelling, according to the Eq. (6) (Acunha et. Al, 2011). The coefficients *C1* to *C8* are determinate by thermal balances and experimental adjustment.

$$\dot{q} = C1 + C2\left(T_e\right) + C3\left(T_c\right) + C4\left(T_e{}^2\right) + C5\left(T_e T_c\right) + C6\left(T_6{}^2\right) + C7\left(T_e{}^3\right) + C8\left(T_e{}^2 T_c\right) + C8\left(T_e T_c{}^2\right) + C10\left(T_c{}^3\right) \tag{6}$$

Where $T_e$ (°C) and $T_c$ (°C) are the evaporation and condensation temperature, respectively (°C).

2.1.2. Condenser

The determination of the average internal heat exchange coefficient was calculated using the correlations presented by Chato (1962) *apud* Bejan (1996) for internal condensation in horizontal tube dominated by natural convection, for low steam Reynolds number (Re < 3,5x10$^4$), $h_i$ is given by Eq. (7).

$$h_i = 0.55\left[\frac{\rho_L\left(\rho_L - \rho_V\right) g h'_{fg} k_L^3}{\mu_L d\left(T_{sat} - T_p\right)}\right]^{1/4} \tag{7}$$

Where $h'_{fg}$ is the corrected latent heat of vaporization (J/ kg) that has the objective of include the effect of sensible heat due to temperature reduction of the condensed liquid below the saturation temperature in regions near to the wall. The $h'_{fg}$ is given by Eq. (8).

$$h'_{fg} = h_{fg} + \frac{3}{8} C_{p,L}\left(T_{sat} - T_p\right) \tag{8}$$

In this equation $C_{p,L}$ is the liquid specific heat (J kg$^{-1}$ k$^{-1}$).

The external heat exchange coefficient between water and tube was the object of study of some researchers that found correlations based in the relation $(\Gamma/d_e)$, where $\Gamma$ represents the water flow per unit of tube length (Kg m$^{-1}$ s$^{-1}$) (Parker and Treybel, 1961).

$$h_e = 704\left(1.29 + 0.022T_w\right)\left(\frac{\Gamma}{d_e}\right)^{1/3} \tag{9}$$

Where $T_w$ is the water temperature (°C).

After the determination of the internal and external heat exchange coefficients is possible to calculate de overall heat exchange coefficient of the evaporative condenser (EC). Eq. (10), allows to determinate the EC (evaporative condenser) capacity.

$$\dot{q} = UA\Delta T \tag{10}$$

Where $A$ is the heat exchange area (m²) and $\Delta T$ is the difference of temperature between $NH_3$ and spray water (°C).

The heat transfer rate can also be determined through experimental data, measuring air flow rate and inlet and outlet air flow conditions by the Eq. (11).

$$\dot{q} = \dot{m}_{air}\left(i_{air,out} - i_{air,inlet}\right) - \dot{m}_{wr} i_{wr} \tag{11}$$

Where $\dot{m}_{air}$ is the air mass flow rate trough the EC (kg m$^{-1}$), $i_{air,out}$ and $i_{air,inlet}$ are, respectively, the outlet and inlet enthalpy of the air flow (kJ kg$^{-1}$), $\dot{m}_{wr}$ is the replacement water mass flow rate (kg s$^{-1}$) and $i_{wr}$ is the enthalpy of the EC water in the sump (kJ kg$^{-1}$).

### *2.2. Characteristics of the industrial intallation*

The study was realized in a poultry food processing company where data related to the operation was collected. The ammonia refrigeration system operates to keep the operation of antechambers, chillers and ice machines to provide cold water, freezing tunnels, acclimatized rooms to process food and cold storage rooms. The installed power by equipment kind, can be verified in the Table 1.

**Table 1.** Installed power by equipment.

| Equipment | Installed power (kW) |
|---|---|
| Air cooler | 166.8 |
| ECs | 99.2 |
| $NH_3$ pumps | 11.04 |
| Screw compressors | 1030.4 |
| Total Installed Power | 1307.44 |

### *2.3. Methodology*

Based on the data measured it was possible to develop the overall mathematical modelling of the system with the objective of provide the computational simulation of the main electricity consumption equipment operating together. Probably the situation verified is different of those predicted in the initial project, due the actual necessities of the company.

Relevant quantities that were used to execute the analysis of the refrigeration systems are: time of operation, refrigeration thermal load, chamber and antechamber temperature, operation pressures, power and electricity consumption.

During the time of collection of data, it was verified an operation with full refrigeration thermal load, where all of the processing locals were in working. Also, it was identified that the time of operation of the refrigeration system was approximately 24 hours, resulting in a low variation of the refrigerated rooms. This analysis is based on 3 months of operation, being 2 months in a usual mode operation and 1 month after implementing energy consumption reduction strategies. The evaluation in the company demonstrated a tendency constant of temperature to each process as shown in Table 02.

**Table 2.** Process temperatures and refrigeration loads.

| Process | Saturation Temperature (°C) | Refrigeration Load (kw) |
|---|---|---|
| Freezing and storage | -35 | 2816 |
| Ice, cold water and intermediary cooling | -10 | 4400 |
| Cold water and acclimated rooms | 0 | 1973 |
| Evaporative Condenser | 35 | 8191 |

During the processing food, the chickens are immersed in a moisture of ice and cold water to chilling until 5°C, approximately. After that, it is goes cutting rooms, freezing tunnels and storage chambers.

Mathematical modelling and computational simulations were done to found optimized operation behaviours for to reduce electrical consumption at peak time. The computational program developed to simulate various operating scenarios to find the best engine room arrangement without compromising the final quality of food products. Figure 1 shows the main window of that simulation program. Computational program allows cut in and cut out screw compressors, evaporative condensers, ice machines, change the product rate in the tunnels as well as change his temperatures, operation pressures of the screw compressors, percentage of the compressors capacity, wet bulb environmental temperature and other variables.

At the main screen, also is possible, to visualize demanded power by process in that industry, being classified like:

- -35°C → At the main screen, also is possible, to visualize demanded power by process in that industry, being classified like:
- -10°C → demanded power for all of the compressors and pumps working with evaporation temperature of -10°C;
- 0°C → demanded power for all of the compressors and pumps working with evaporation temperature of 0°C;
- Evap → demanded power for all of the evaporators working;
- Cond → demanded power for all of the evaporative condensers working;
- Total → total demanded power for all electrical motors working.

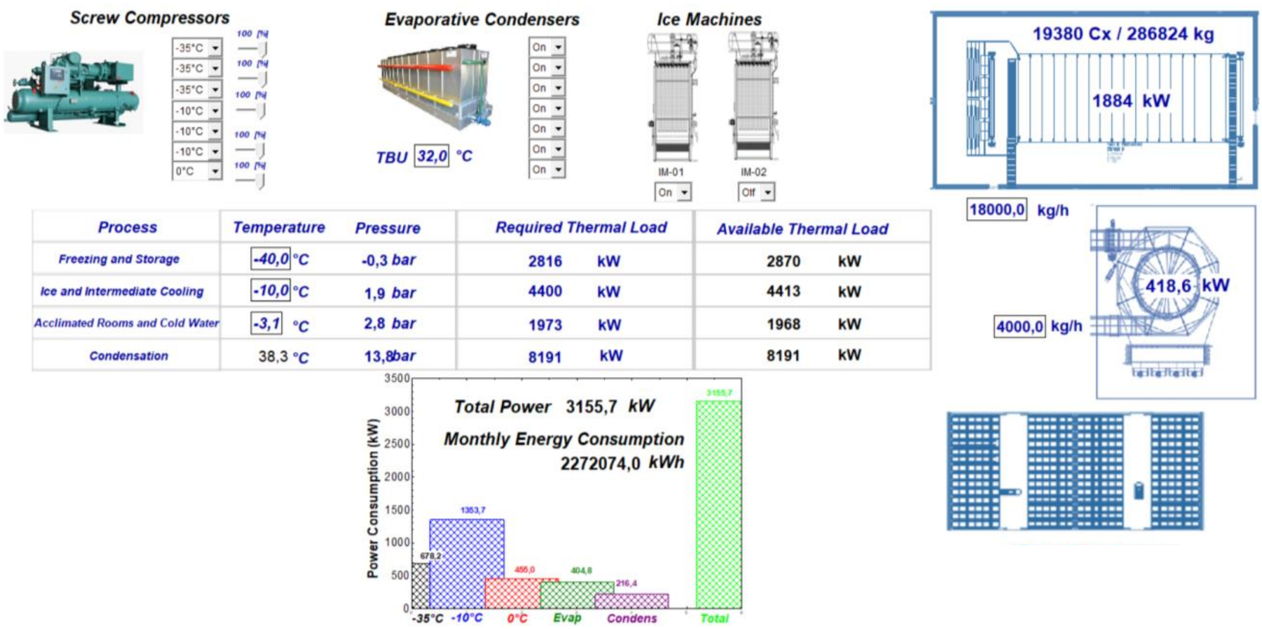


**Figure 1.** Main window of simulation program

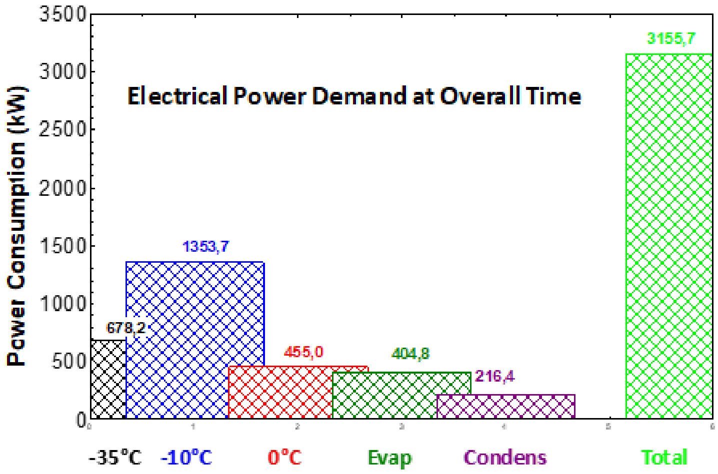


**Figure 2.** Original demanded power by sector

## 3. Results and Discussions

### *3.1. Water Contamination in Ammonia System*

In refrigeration systems, frequently, is found a considerable quantity of water together with the fluid refrigerant. The water contamination can cause undesirable effects such as a decrease in the coefficient of performance (COP), reduction in the refrigeration thermal load capacity, galvanic corrosion in valves and pipes and the increase of oil consumption by the compressors. In order to maintain good conditions of operation, the percentage of water present in the refrigeration system should be inferior to 0.5%. Moreover, for even better results the ammonia purity recommend is 99.95%.

In relation to contamination of the ammonia, it was identified contamination by water, oil and incondensable gases. The contamination index by water and oil in the $NH_3$ is capable of increasing the annual consumption of electricity by approximately 1440000 kWh. Furthermore, the contamination by incondensable gases should increase in annual electricity over-consumption of 865000 kWh.

Due to the condensers operating conditions, the capacity of these heat exchangers significantly compromised. The Figure 3 shows the condensation capacity curve (red), the corrected condensation capacity (green), the real condensation capacity (black) and the required condensation load to the plant. Through this graph is evidenced that the equilibrium between the real condensation capacity and the required capacity should be in the condensation temperature of 28.7°C (10.4 bar). However, due to the problems encountered and the inoperability of two condensers, the operating conditions verified are presented in Figure 4, where the system needed to amplify the correction coefficient of the condensers to compensate the missing capacity, increasing the condensation temperature to approximately 40.2 °C (14.9 bar). For this operating condition, only 1/3 of the condensers working would be needed.

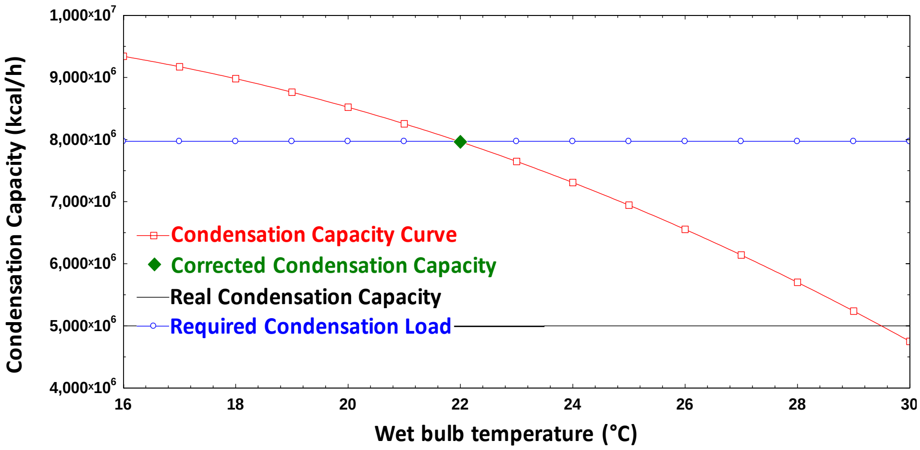


**Figure 3.** Real Thermal capacity verified

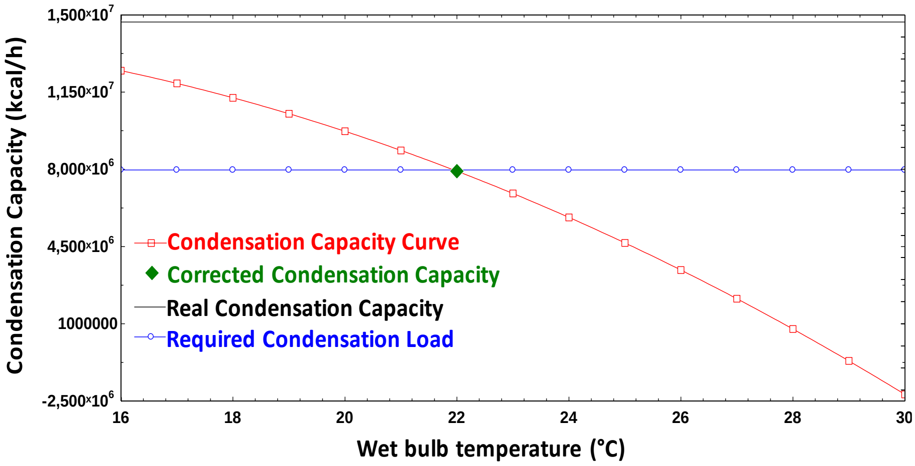


**Figure 4.** Thermal capacity with all condenser running.

*3.2. Changes in operating conditions*

A possibility of improving the energy efficiency in an industrial refrigeration system is related to all of the operating conditions. Firstly, the incondensable gas was purged of the system, due to your impact directly on the condensation head pressure and, consequently in electrical energy consumption. Secondly, the computational simulations are done aim to predict better equipment arrangements to provide higher energy efficiency levels.

Thus, it was realized two kind of simulations: first one using the peak time strategies in order to energy saving costs. Second one, using strategies for off-peak time. Results presented at Figure 5 was predicted by simulation program developed to the industrial application. Those results show that que demanded power was reduced at all of the process in relationship to the current operation behaviors and it can be summarized at Table 03.

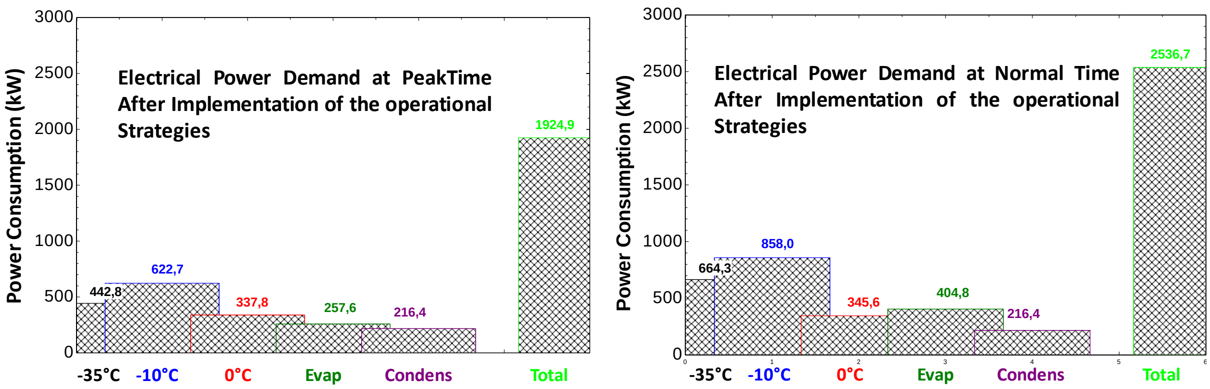


**Figure 4.** Demand power by sector after strategies implementation.

**Table 3.** Power reduction by sector

| Process | Power Reduction (kW) | |
|---|---|---|
| | Peak Time | Off-Peak Time |
| -35°C | 235.4 | 13.9 |
| -10°C | 731 | 495.7 |
| 0°C | 117.2 | 109.4 |
| Evaporators | 147.2 | 0 |
| Evaporative Condenser | 0 | 0 |
| Total | 1230.8 | 619 |

After one month of operation, the results referents to electric energy consumption were compared with the verified consumption in the months between January and September of 2020, through the measurement system existing in the company.

Comparing the results obtained by implementing the strategies between the months of January and September 2020, as well as August and September 2020, it was possible to construct the graphs presented in Figures 5 and 6, respectively. The data presented in these graphics cover the three hours of peak time, excluding holidays and weekends, comparing the same days of weeks in each month. Two unexpected events occurred during September month during the measurements. In the first three days of tests, the installation was contaminated with air, as described before and on the day 11/09 the freezing tunnel presented failure near 5 pm, making it impossible to implement the principal strategies to reduce the electricity consumption and limiting the electric energetic gain that should be possible. Analyzing just the days when the operation was in normality, is possible to verify that between August and September 2020, the lowest energetic gain was of 15.1%, while the biggest was of 55.4%. The average energetic gain achieved was of 31.7%. Executing the same comparative analyze between the August and September of 2020, the lowest energetic gain was of 15.8%, while the biggest was of 48%. The average energetic gain achieved was of 32.5%.

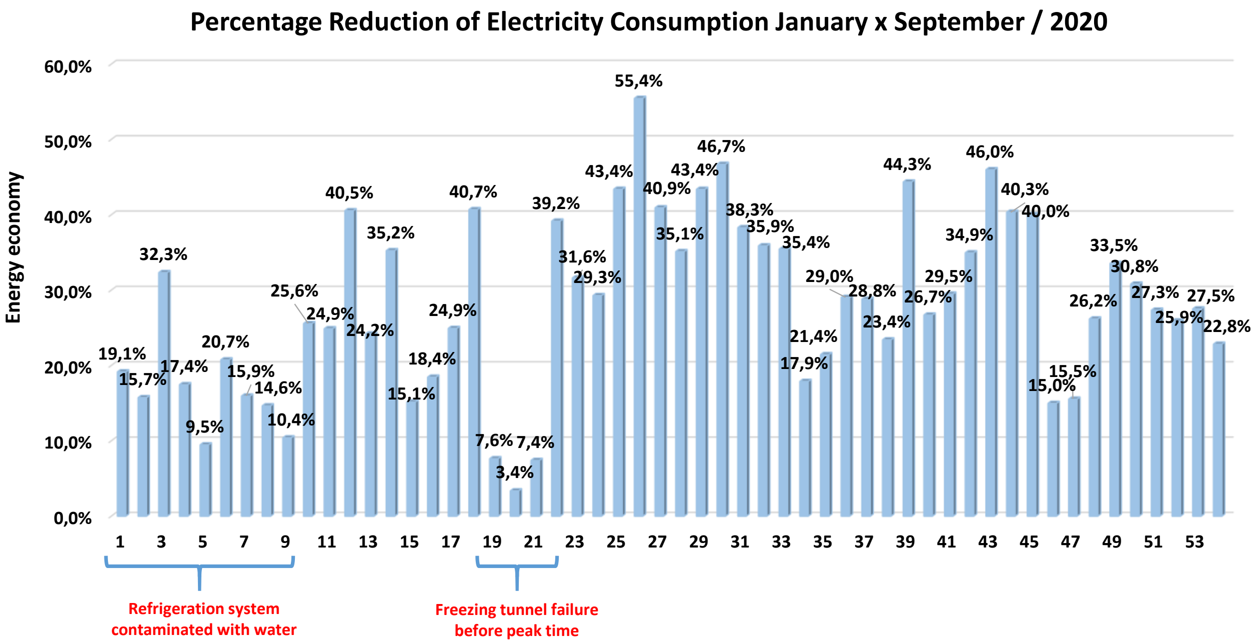


**Figure 5.** Real electrical energy reduction in relationship january/2020.

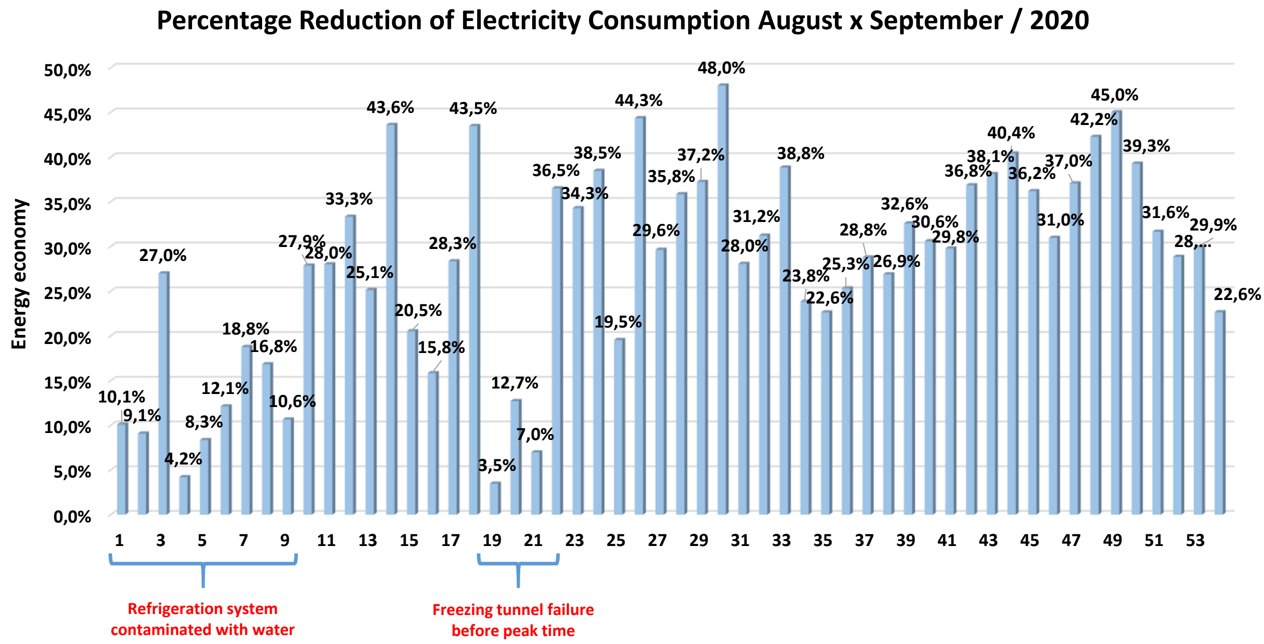


**Figure 6.** Real electrical energy reduction in relationship august/2020.

Taking into account the reduction in consumption of the full month of operation in September, in comparison with the months of January and August 2020, we have the results obtained in Figures 7 and 8, respectively. The total economy of electricity in September was of 458438 kWh, being that 80668 kWh occurred in peak time. Evaluating the economy of electric energy between August and September, the economy of electric energy was of 214057 kWh, being 67983 kWh of this total consumed in peak time. Making a projection of the economy of electricity between August and September for the others months of the year, can be expected a reduction of the consumption in approximately 2568684 kWh. Moreover, it was possible to reduce contracted electricity demand at peak times in 1000 kW, representing a reduction of approximately 17%.

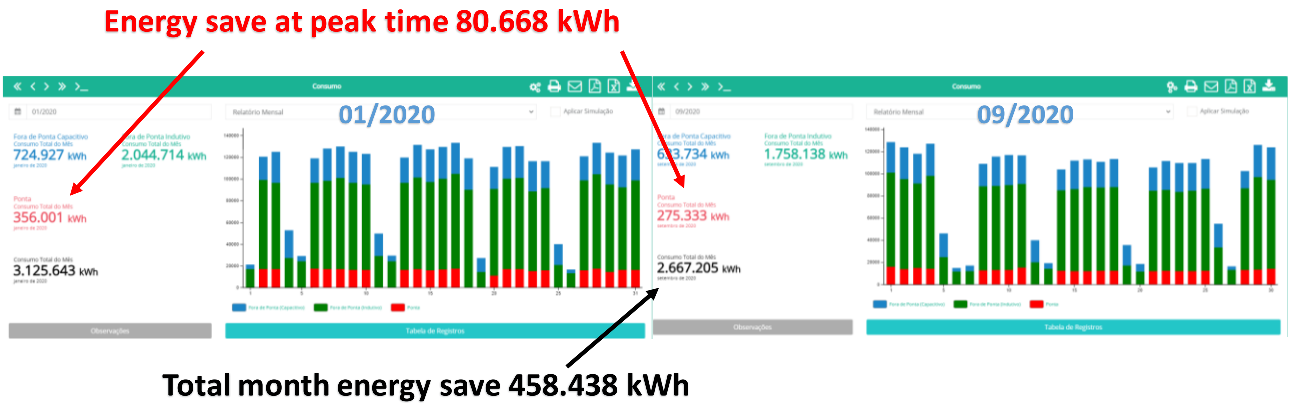


**Figure 7**. Comparison monthly energy consumption between January and September of 2020

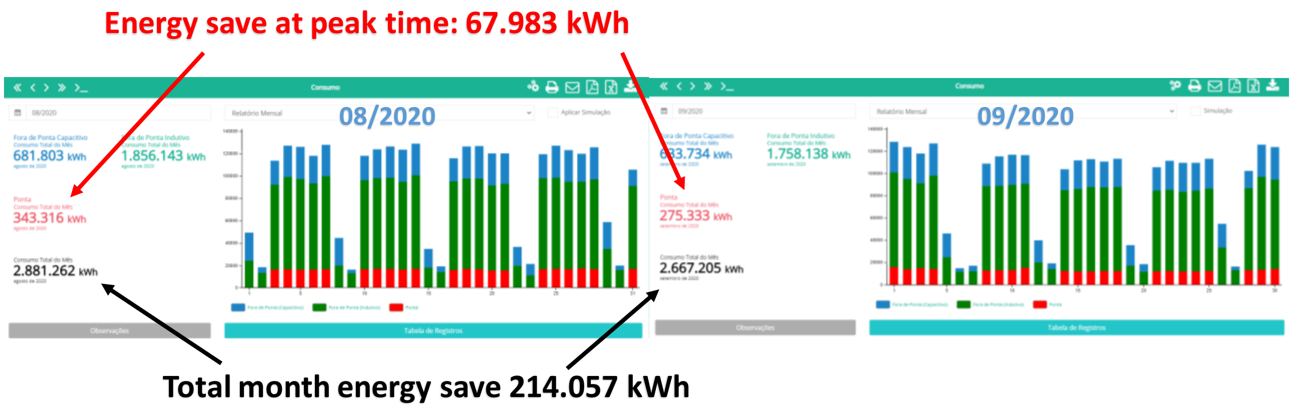


**Figure 8**. Comparison monthly energy consumption between August and September of 2020

The implementation of the actions and strategies described above had a positive impact on the refrigeration system COP. The Table 4 presents a comparison of the COP between original and modified system. In off-peak time there was an increase of 24% while at peak time the percentage increase was of 63.5%.

**Table 4.** Comparative COP

| Original COP | | Increased COP | |
|---|---|---|---|
| Peak | Off-Peak | Peak | Off-Peak |
| 2.92 | 2.92 | 4.77 | 3.62 |

## 4. Conclusions

The present article presented a methodology through the use of mathematical modeling and computational simulation to promote increase the COP of an industrial refrigeration system. Thus, it was possible to simulate different conditions of operation of the refrigeration cycle and also modified the components that were running. The action was implanted in a real installation to verify the gains verified on simulation.

The elimination of water contamination and the modification in the operation conditions resulted a total energy saving in September of 458438 kWh, being that 80668 kWh occurred in peak time compared with January and August 2020. Evaluating the of economy of electric energy between August and September 2020, the electrical energy save was of 214057 kWh, having been 67983 kWh of this total consumed in peak time. Making a projection of the electricity save between August and September for the others months of the year, can be expected a reduction of the consumption in approximately 2568684 kWh. These results were achieved without any structural modification in the refrigeration installation.

Moreover, the refrigeration unit has potential to saving contracted electricity demand at peak times in 1000 kW, representing a reduction of approximately 17%. To this end, a new assessment after the initial changes must be carried out in order to determine new operational standards. Finally, it is recommended that this procedure be performed whenever the operational condition of the refrigeration system needs to be changed, either by imposition of production or by climatic condition.

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