25th ABCM International Congress of Mechanical Engineering
October 20-25, 2019, Uberlândia, MG, Brazil

COB-2019-0020
ENERGETIC AND ECONOMIC ANALYSIS OF THE STEAM COMPRESSION REFRIGERATION SYSTEM WITH INTERCOOLER AND WITH COMPRESSOR SCALE SYSTEM

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Abstract. *The present investigation is developed in the refrigeration industry, whose activity is the production and commercialization of the ice used mainly in the fishing industry. The development consists in evaluating and comparing the steam compression refrigeration system with intercooler and the steam compression refrigeration system with compressor scale system. These evaluations are intended to determine the COP (coefficient of performance) in both systems through an energy analysis, as well as to reduce the freezing time of the ice without affecting its quality. Finally, a thermoeconomic evaluation of these systems will be carried out to allow the company to analyze the convenience in study, by reducing their production costs using their productive processes efficiently, providing quality products, and being competitive in the market. It is evident that the COP will be increased by 8.28%, which allows a reduction in ice freezing time by 22.25% or 3 hours and 39 minutes. In addition to obtaining savings in electricity consumption equivalent to $ 8 563.00 dollars in the next 12 months.*

Keywords: *coefficient of performance, compressor scale system, intercooler, refrigeration industry, steam.*

1. INTRODUCTION

In industrial refrigeration, applications are larger in size and capacity than commercial ones, some typical applications are in ice plants, food preservation and handling, in heat production, refrigerated warehouses, among others (Sánchez *et al.*, 2010).

The refrigeration industries are big consumers of energy, they must establish control strategies that lead them to a framework of competitiveness and efficient production. The production of block ice is one of them. However, in these plants, process control is not normally established other than a fixed-cycle timer that determines the end of the production period.

In the different productive sectors, the variation of the demand in function to the temporality causes that the production varies, reason why in some cases it is necessary to increase the capacity of production, whereas in others simply, to diminish the production (Heard *et al.*, 2016).

The ice production industry is a growing industry, making it one of the most important in recent years as it serves for food preservation; One of the fields with the greatest use of ice in our area is the fishing industry. Ice plays a key role in preserving the quality of export fish, as well as in its internal transport and commercialization.

The refrigeration by compression is a method of refrigeration consists to force mechanically the circulation of a refrigerant in a closed circuit creating zones of high and low pressure with the intention that the fluid absorbs heat in the evaporator and yields it in the condenser (Salhi *et al.*, 2018; Xu *et al.*, 2018).

The purpose of refrigeration is to maintain the temperature of a given environment at temperatures below the value of the ambient temperature, with a constant and sustained value.

In the ideal refrigeration cycle, in the energy balances of the equipment, any heat loss or gain in the pipes is disregarded, considering that the only heat exchanges that take place in the system occur in the evaporator and in the condenser. Although it is known that the refrigerant fluid has a temperature different from that of the pipes and that this would necessarily produce a heat exchange (Castillo *et al.*, 2014; Danlei *et al.*, 2017; Liu *et al.*, 2018).

Cooling systems with intercooler, as it is not possible to control exactly the liquid state of the fluid leaving the evaporator, the system is usually designed so that the fluid comes out as slightly superheated steam instead of saturated steam of the ideal cycle. This ensures that the compressor always circulates the fluid in the vapor phase. This oversizing leads to an increase in the specific volume of the steam compared to the saturated steam at the same pressure.

Unfortunately, this supposes an increase of the power of entrance of the compressor, since the work in steady state, is proportional to the specific volume (Ifaei *et al.*, 2016; Rostamzadeh *et al.*, 2018).

Cascade cooling systems, there are industrial applications where low temperatures are required, ranging from -25 to -75 °C. Unfortunately, a simple vapor compression cycle cannot be used to reach these moderately low temperatures.

To overcome this difficulty without stopping the compression of steam, a cascade system can be used (Cimsit *et al.*, 2012; Heard *et al.*, 2016; Patel *et al.*, 2017).

The refrigerant used is Ammonia, due to the high refrigeration requirement of this in its latent heat change with respect to other refrigerants (Cajo *et al.*, 2009; Seckin, 2018).

To avoid the presence of sludge or scale in the cooling system due to the presence of foreign bodies and drag the base salt, it is preferable to use industrially obtained refrigeration solutions, which thanks to their chemical composition act as anti-corrosive, are odorless, toxic and ensure a longer service time (Castro *et al.*, 2008).

On the other hand, the purpose in an exergy analysis is usually to determine the exergy of each of the elements that make up a system and thus achieve their efficiencies. The destruction of exergy of a component can be determined by an exergy balance of the component. It is important to remember that changes in the system caused by kinetic and potential energy are neglected (Cengel *et al.*, 2008; Xing *et al.*, 2015; Zhang *et al.*, 2018).

The irreversibilities, although they cannot be avoided, must be reduced to a minimum value, because they bring as a consequence the need to supply an additional power, to achieve the desired cooling since as long as the irreversibilities increase the same happens with the power supplied to the compressor (Liu *et al.*, 2017).

The exergy analysis allows to optimize the design of the different equipment of an installation. And if the equipment is already installed, the exergetic cost of the equipment could be determined, which is fundamental in the decision making process; With small modifications, significant exergy and economic savings are achieved (Bellos *et al.*, 2017; Lounissi *et al.*, 2017).

During the process of production of block ice of the PRC. S.A.C. refrigeration plant, it is observed that, as in any productive system, the cycle time necessary for the freezing process is a problem that occurs daily during the high season, due to the increase in its demand, in other words, there is more extraction of fish. Also, there are occasional clients, which, not being perennial clients of the company, arrive to make their purchase at the least expected moment and without having placed their order with anticipation. Therefore, the company to deliver their orders on time is in need of reducing the freezing time of the ice causing 2 things; the first, that the block does not complete its cycle and that its quality is affected by obtaining brittle ice, and the second, that the plant reaches 100% utilization of its installed capacity.

The refrigeration plant PRC. S.A.C., currently uses traditional technology for the cooling system of ice blocks, in other words, it does not have equipment that helps to improve the process and therefore obtaining a finished product of quality is almost impossible.

Therefore, due to the multitude of applications and their relevant importance, the refrigeration plant, in its steam compression cold production facilities has a high percentage of energy consumption as well as a high economic and environmental impact. On the one hand, the indirect greenhouse effect associated with the origin of the energy used, and on the other, the direct effect associated with refrigerant leaks because refrigerants with a high global warming potential (PCM) are used.

The objective of this research is to evaluate the best production system for the refrigeration plant. Among these systems are: steam compression refrigeration system with intercooler and steam compression refrigeration system with compressor scale system with and without intercooler.

The evaluations will be carried out through an energy analysis to obtain the coefficient of performance of the systems evaluated. Later, with the system that presents the best COP (coefficient of performance), the new freezing time of the ice will be determined and it will be compared with the current time (20 hours) used by the refrigeration plant to obtain block ice.

Finally, the economic evaluation tool will be used, using the electricity consumption in kW-h per month, the active electric energy cost in dollars per kW-h and the average hours of operation of the equipment that makes up the plant during a day of operation.

The purpose is to obtain savings in production, delivering quality products in the shortest time and that the refrigeration plant is competitive in the market.

2. METHODOLOGY

In order to carry out a successful investigation, the descriptive technique, the use of index cards, the use of bibliographic and electronic material will be used. Energy diagnostic information will be obtained both by the author and by qualified technical personnel of the company. Microsoft and Coolpack softwares to perform scenario models on the data obtained, which will allow us to perform the energy analysis.

The required information is made up of the following: measurement of the current operating parameters such as the water temperature in the molds, final brine temperature, condenser temperature, evaporator temperature, high pressure and low pressure, daily hours of work, amount of refrigerant (NH_3) used, power consumed by the compressor, equipment information and data sheet of similar equipment.

2.1. Energetic analysis of the steam compression refrigeration system

The following energetic analysis is used for the correct development of the research.

To calculate the refrigeration load or cooling capacity, it is necessary to determine the necessary loads of all the components immersed in the brine pool.

It is determined as the sum of the cooling water from ambient temperature to 0 °C, freezing and subcooling of ice to -3 °C. The equations to be used are the following: (Colorado *et al.*,2015).

2.1.1. Refrigerating load for cooling water

It is given by the variation of sensible heat, and that must be extracted to the water to pass from the initial water temperature to 0 °C:

$$R_{cw} = \dot{m}_w * Cp_w * (T_{iw} - T_{fw}) \tag{1}$$

Where, R_{cw} is the refrigerated load to cool the water from the ambient temperature to the freezing point, m_w is the mass flow of water, Cp_w it is the constant pressure specific heat of the water and T_{iw}, T_{fw} represent the initial water temperatures up to the final freezing temperature at 0 °C.

2.1.2. Refrigerated load for freezing water

It is the variation of latent heat or phase change of water from the liquid state to the solid state at atmospheric pressure.

$$R_{fw} = \dot{m}_w * h_{fg} \tag{2}$$

Where, R_{fw} is the refrigeration load for freezing and h_{fg} is the latent heat of the fusion water at 0 °C.

2.1.3. Refrigerated load for subcooling water

It is the sensible heat variation that must be extracted to the ice for subcooling from the freezing temperature to the subcooling temperature of 3 °C.

$$R_{sw} = \dot{m}_w * Cp_i * (T_{fw} - T_i) \tag{3}$$

R_{sw} is the refrigeration load for the subcooling of the water, Cp_i it is the constant pressure specific heat of the ice and T_i is the final temperature of subcooled ice (-3 °C).

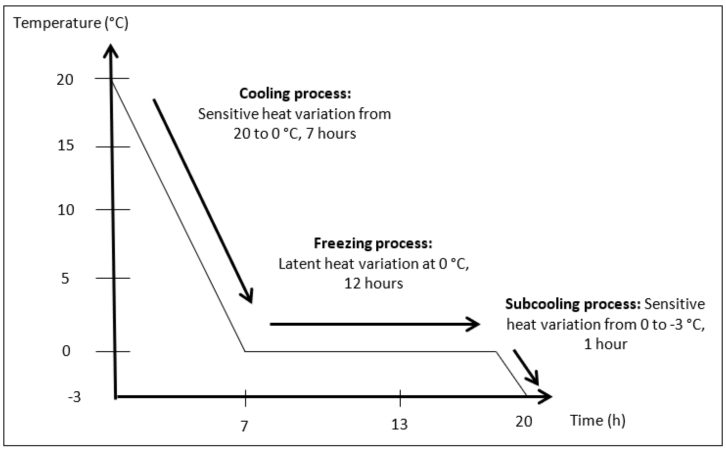


Figure 1. Temperature profile for block ice generation, starting from the cooling process, after freezing and finally subcooling of the ice, having a period of 20 hours per day. (The cooling process has a period of 7 hours; the freezing process has a period of 12 hours and the subcooling process has a period of 1 hour).

2.2. Indicators of the steam compression refrigeration system (Hegazy *et al.*, 2016; Rangel, 2003; Zhu *et al.*, 2012)

2.2.1. Refrigerant effect

$$RE = \dot{m}_r * (h_1 - h_4) \quad (4)$$

Where, m_r is the mass flow of the refrigerant, h_1 is the enthalpy of the refrigerant at the evaporator outlet and h_4 is the enthalpy of the refrigerant at the inlet of the evaporator and its value is equal to the enthalpy of saturated liquid.

2.2.2. Compressor power

$$CP = \dot{m}_r * (h_2 - h_1) \quad (5)$$

Where, h_2 is the enthalpy of the refrigerant at the outlet of the evaporator inlet compressor and its value is equal to the enthalpy of the refrigerant in the state of superheated steam.

The compressor is driven by an electric motor, so the power supplied by the electric motor is equal to:

$$W_{em} = \eta_{comp} * \eta_{em} * CP \quad (6)$$

η_{comp} is the efficiency or mechanical performance of the compressor and is a function of the type of compressor used, so we have for reciprocating compressors is equal to 0.8 and for screw compressors is 0.85, η_{em} is the efficiency or performance of the electric motor that drives the compressor and W_{em} is the Power supplied by the electric motor.

2.2.3. Thermal rejection power

It is the thermal power associated with the energy that the refrigerant fluid must release from the refrigeration cycle into the environment, in order to enter the thermodynamic conditions necessary to produce the cooling effect again.

$$TRP = \dot{m}_r * (h_2 - h_3) \quad (7)$$

2.2.4. Coefficient of performance:

It is an indicator that denotes the efficiency of a refrigeration cycle and is evaluated by comparing the refrigerant effect between the power absorbed by the compressor.

For the systems of refrigeration by compression of steam is a value superior to the unit, being in maximum values of 3.5 to 5 for industrial plants with generation of ice in block and cold rooms.

$$COP = \frac{refrigerant\ effect}{compressor\ power} \quad (8)$$

With the methodology described, the equations and laws that are important, give us an overview of what is intended to investigate, in this sense, we can give sustenance to this research by demonstrating that it already has an established foundation, and will be developed later.

3. RESULTS AND DISCUSSIONS

In this section, the thermal load of the ice plant in blocks is determined and from this we continue with the procedure, for this we use the data of the Tab. 1.

Table 1. Income data for determining the thermal load of the ice industry

Physical parameters(1)	Symbology	Specification	Units
Amount of water in the molds	m_w	125	Ton/day
Ambient temperature	T_a	20	°C
Initial water temperature	T_{iw}	20	°C
Final temperature of the ice	T_{fw}	-3	°C
Specific heat of water	Cp_w	4.18	kJ/kg °C
Heat melting water	Δh_{fg}	334	kJ/kg
Specific heat of ice	Cp_i	2.1	kJ/kg °C
Specific heat of the mold	Cp_m	0.5	kJ/kg °C
Brine final temperature	T_{fs}	-5	°C

(1)Management reports of the maintenance area of the refrigeration plant PRC. S.A.C.

The most important parameter is the amount of water used for the production of ice per day that is 125 ton per day.

3.1. The determination of the thermal load or refrigerated load

By performing the various procedures and apply the Eq. (1) to (9), we obtain the following thermal load or refrigeration load, shown in Tab. 2.

Table 2. Determination of the refrigeration load of the refrigeration plant PRC. S.A.C.

N°	Refrigerated load	Power (kW)	(%)
1	Refrigerated load for block ice production	735.94	90.48
2	Refrigerated load for the cooling of the molds	6.98	0.86
3	Refrigerating load of brine water agitators	29.80	3.66
4	Additional losses	40.67	5
	Total	813.39	100

The highest percentage of power consumed by the refrigeration cycle is used in the refrigerated load for block ice production with a percentage of 90.48 % and a power of 735.94 kW.

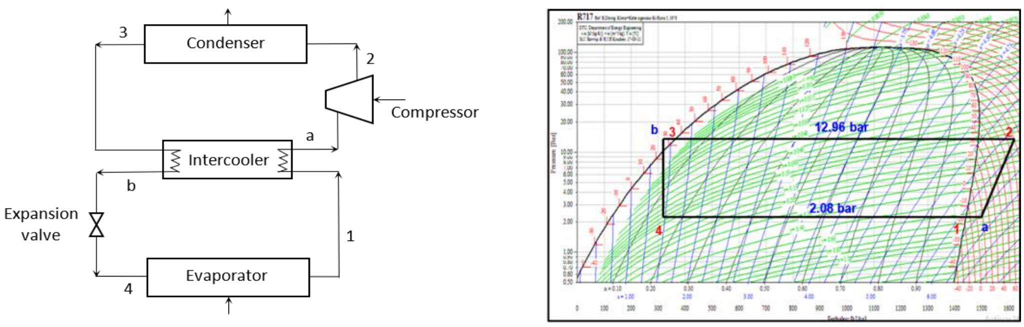
The thermal load determined in Tab. 2 will be used to calculate the Indicators of the steam compression refrigeration system.

3.2. Energy balance in the steam compression refrigeration system with intercooler

Using the thermodynamic table, the properties of ammonia are determined at each point of the basic cycle, parameters shown in the Tab. 3.

Table 3. Refrigerant values in the steam compression refrigeration system with intercooler, Fig. 2.

Parameters		Symbology	Quantity	Units
High pressure		P_a	12.96	Bar
Low pressure		P_b	2.075	Bar
Inlet temperature in the condenser		T_2	113.24	°C
Final condenser temperature with cooling		T_{cond}	27	°C
Temperature in the evaporator		T_{evap}	-17	°C
Pressure drop at the evaporator outlet		ΔP	1	Bar
Values obtained from the thermodynamic table - R717 (Moran *et al.*, 2004)				
Point	Calculation criteria		Quantity	Units
1a	Saturated steam at P_{low} with $\Delta T = 11°C$		1447.1	KJ/kg
2R	P_{high} and at the temperature T_2 at the entrance of the cond.		1687.96	KJ/kg
3	Subcooled liquid at P_{high} with $\Delta T = 1°C$		312.72	KJ/kg
4	Value equal to the point 3		312.72	KJ/kg



A1 A2

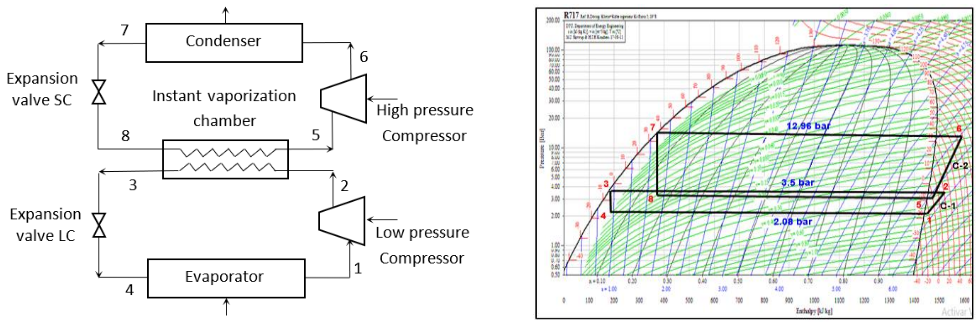
Figure 2. A1. Flow diagram and A2. Diagram P vs h of the steam compression refrigeration system with intercooler (subcooling and reheating). Cycle equipment: evaporator, compressor, condenser, expansion valve and intercooler - Coolpack software.

3.3. Energy balance in the steam compression refrigeration system with compressor scale system without intercooler

Using the thermodynamic table, the properties of Ammonia are determined at each point of the basic cycle, parameters shown in the Tab. 4.

Table 4. Refrigerant values in the steam compression refrigeration cycle with compressor scale system without intercooler, Fig. 3.

Parameters	Symbology	Quantity	Units
High pressure	P_a	12.96	Bar
Average pressure	P_m	3.5	Bar
Low pressure	P_b	2.075	Bar
Inlet temperature in the condenser	T_2	113.24	°C
Condenser final temperature	T_{cond}	27	°C
Temperature in the evaporator	T_{evap}	-17	°C
Pressure drop at the evaporator outlet	ΔP	0.1	Bar
Values obtained from the thermodynamic table - R717 (Moran *et al.*, 2004)			
Point	Calculation criteria	Quantity	Units
1	Saturated steam at P_{low}	1420.71	KJ/kg
2R	P_{medium} and isentropic to the point 1	1487.09	KJ/kg
3	Saturated liquid at P_{medium}, (T=-5,36 ºC)	155.20	KJ/kg
4	Value equal to that of point 3, isentropic process	155.20	KJ/kg
5	Saturated steam at P_{medium}	1436.01	KJ/kg
6	P_{high} and isentropic to the point 5	1615.07	KJ/kg
7	Saturated liquid at P_{high}, (T=33,49 ºC)	339.48	KJ/kg
8	Value equal to that of point 7, isentropic process	339.48	KJ/kg



B1 B2

Figure 3. B1. Flow diagram and B2. Diagram P vs h of the steam compression refrigeration system with compressor scale system without intercooler; (SC) superior cycle: high pressure compressor, condenser and expansion valve SC. (LC) lower cycle: evaporator, low pressure compressor, expansion valve LC - Coolpack software.

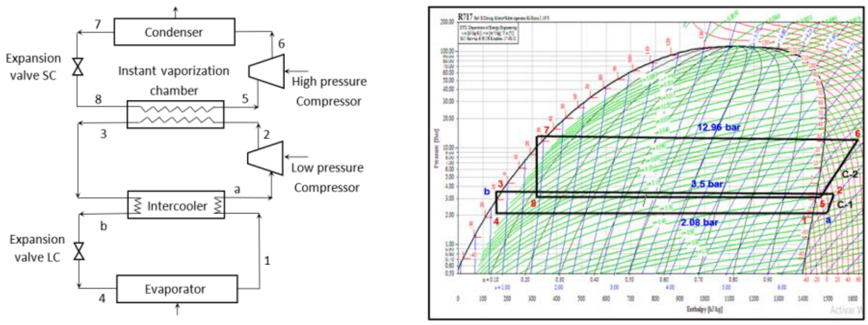
3.4. Energy balance in the steam compression refrigeration system with compressor scale system with intercooler

Using the thermodynamic table, the properties of Ammonia are determined at each point of the basic cycle, parameters shown in the Tab 5**.**

Table 5. Refrigerant values in the steam compression refrigeration cycle with compressor scale system with intercooler, Fig. 4.

Parameters	Symbology	Quantity	Units
High pressure	P_a	12.96	Bar
Average pressure	P_m	3.5	Bar
Low pressure	P_b	2.075	Bar
Inlet temperature in the condenser	T_2	90.15	°C
Condenser final temperature	T_{cond}	27	°C

Temperature in the evaporator	T_{evap}	-17	°C
Pressure drop at the evaporator outlet	ΔP	0.1	Bar
Values obtained from the thermodynamic table - R717 (Moran *et al.*, 2004)			
Point	Calculation criteria	Quantity	Units
1	Saturated steam at P_{low} with $\Delta T = 11°C$	1447.9	KJ/kg
2R	P_{medium} and isentropic to the point 1 with $\Delta P = 0{,}5bar$	1495.01	KJ/kg
3	Subcooled liquid at P_{medium}, (con $\Delta T = -5°C$)	137.04	KJ/kg
4	Value equal to that of point 3, isentropic process	137.04	KJ/kg
5	Saturated steam at P_{media} con $\Delta P = 1bar$	1450.8	KJ/kg
6	P_{high} and isentropic to point 5 with condensation temperature	1630.01	KJ/kg
7	Subcooled liquid at P_{high}, with $\Delta = 1°C$	312.72	KJ/kg
8	Value equal to that of point 7, isentropic process	312.72	KJ/kg



C1 C2

Figure 4. C1. Flow diagram and C2. Diagram P vs h of the steam compression refrigeration system with compressor scale system with intercooler; (SC) superior cycle: high pressure compressor, condenser and expansion valve SC. (LC) lower cycle: evaporator, low pressure compressor, expansion valve LC and intercooler - Coolpack software.

3.5. Results of the application of energy balances to the steam compression refrigeration system

An energy balance was made to the current steam compression refrigeration system with intercooler and with compressor scale system in the refrigeration plant shown in the Tab. 6 and Tab. 7 respectively.

Table 6. Results of the energy balance to the steam compression refrigeration system with intercooler.

Steam compression cooling system	A. With intercooler
Mass flow of the refrigerant	0.734 kg/s
Refrigerant effect	813.39 kW
Compressor power	172.69 kW
Thermal rejection power	986.05 kW
Coefficient of performance	4.71
Ton of refrigeration	231.34 ton

Table 7. Results of the energy balance to the steam compression refrigeration system with compressor scale system without and with intercooler.

Steam compression refrigeration system with compressor scale system	B. Without intercooler	C. With intercooler
Refrigerant effect	928.88 kW	939.88 kW
Power of the first compressor	48.72 kW	33.21 kW
Power of the second compressor	159.64 kW	153.04 kW
Mass flow of coolant 1	0.734 kg/s	0.717 kg/s
Mass flow of refrigerant 2	0.892 kg/s	0.856 kg/s
Coefficient of performance	4.46	5.10

From the energy balances performed to the refrigeration systems and the results presented in Tab. 6 and Tab. 7, we can determine that the most important parameter is the coefficient of performance (COP), which will be shown in Fig. 5.

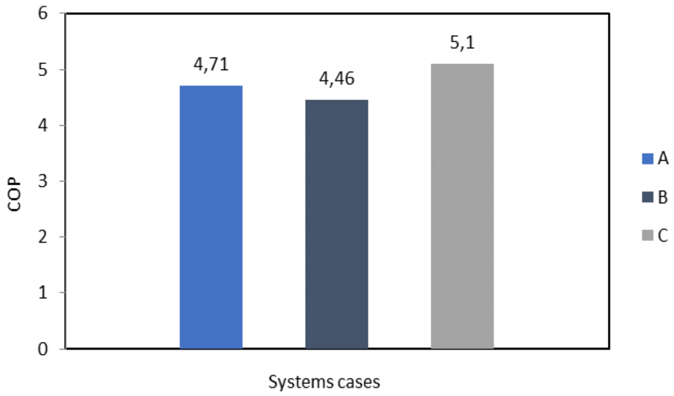


Figure 5. Comparison of the coefficient of performance (COP) of refrigeration systems (A. steam compression refrigeration system with intercooler, B. steam compression refrigeration system with compressor scale system without intercooler and C. steam compression refrigeration system with compressor scale system with intercooler).

The steam compression refrigeration system with compressor scale system with intercooler presents a greater COP with a value of 5.10. Therefore, as this system presents a better result, it will be used to determine the new freezing time of the ice and compare it with the current time that is 20 hours per day used by the refrigeration plant for its production.

3.6. Freezing time of ice in the steam compression refrigeration system with compressor scale system with intercooler

With the result of the cooling effect or thermal load obtained in the previous section we will proceed to perform the reverse process to determine the freezing time of the ice using the steam compression refrigeration system with the compressor scale system.

The reduction of the freezing time is 3 hours and 39 minutes equivalent to 22.25% of the time spent.

3.7. Thermoeconomic evaluation of refrigeration systems

Carrying out the economic analysis, relating the COP obtained in the steam compression refrigeration system with compressor scale system with intercooler and the COP of the steam compression refrigeration system with Intercooler, the electricity consumption in kW-h per month, the active electric power cost in dollars per kW-h and the average hours of operation of the equipment that make up the plant during a day of operation, there is a saving in electricity of $ 23.77 dollars per day, equivalent to $ 8 563.00 dollars per year (see Appendix D of the supplementary material).

4. CONCLUSIONS

Due to the improvements mentioned above, the implementation of this project that is to replace the current system that is the steam compression refrigeration system with Intercooler by the proposed system that is the steam compression refrigeration system with compressor scale system with intercooler must be carried out as soon as possible for the refrigeration plant PRC. S.A.C.

Likewise, more efficient technologies should be implemented in the condenser and evaporator materials that allow the limits of maximum and minimum temperature of the refrigerant to be narrowed, such as the use of evaluative capacitors or evaporators of plates. Pressure and temperature gauges must be placed at strategic points in the refrigeration system, in order to better record the data and obtain better results in the future. And above all, giving greater priority to the control of refrigeration systems for ice production since the energy savings are considerable and production costs will be reduced.

5. ACKNOWLEDGEMENTS

The authors are grateful to FAPESP (Fundação de Amparo à Pesquisa do Estado de São Paulo) for support of this work through Project 2017/08975-1.

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