CASE REPORT

Comments on a heat recovery unit in refrigeration industry

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ABSTRACT

Heat recovery is one of the measures proposed for the appropriate use of ammonia in tropical countries. This article analyzes a heat recovery system installed in an industrial refrigeration plant. Based on comparative readings of operating parameters of the installation, determined the effectiveness of the heat exchange, the increase in the efficiency of the refrigeration system, as well as the fuel saved by heating water in the industry. The results obtained reported that the thermal design based on heat exchange in annular spaces allows a significant saving of resources and a high rate of thermal utilization.

Keywords: Heat Recovery; Refrigeration Plant; Energy Saving

1. Introduction

Saving energy carriers is one of the main sources of resources. It is a necessity and a responsibility to take measures to reduce costs in order to make efficient use of the installed capacities.

Since the last decade heat recovery in the area of industrial refrigeration has become a scientific part of the object of work of many companies in the world. This vision is due to the possibility of increasing energy efficiency through this way, considerably favoring the process of condensation of the refrigerant and affecting a considerable reduction in water heating costs.

There are reports of certain investigations of the use of heat recovery in the area of industrial refrigeration. Abass and Olajire[1] conducted a review of a group of case studies in the brewing industry, where different techniques are applied to promote energy savings and reduced environmental impact. One of the cases presented was the application of a heat recovery unit in the cooling system of the Canadian Maritime brewery for water heating. The savings reported for this concept were in the order of 45,000/year. Other applications using heat recovery units[2] indicate an efficiency increase of more than 20% in some industrial applications.

Srimuang and Amatachaya[3] made a review of the application of heat recuperators, focused on energy savings and heat transfer effectiveness of the different equipment that have been designed and tested for various applications. Due to the nature of these processes, there is in many cases instability in the heat generating sources. This element was introduced in the analysis of the efficiency of heat recovery equipment, by monitoring various elements of different nature[4].

The development of evaluation methods of recovered heat involving the exergy transferred for each component of the system, served as a basis for recent studies, which propose control strategies to ensure a stable response in industrial heat recovery systems, to achieve a satisfactory performance of the heat recovery process[5,7].

In Cuba, the measures proposed for the adequate use of ammonia in tropical countries include among their recommendations the use of heat recovery systems[6]. In the proposal, heat recovery is recommended with the possibility of the dual purpose of recovering heat through an exchanger for the extraction of sensible heat, and at the same time recovering part of the oil that leaves the compressor in the form of vapor. Since 2000, heat recovery equipment has been designed, built and installed in several industries in eastern Cuba. These equipments present benefits that support the proposal in other refrigeration industries.

However, despite having demonstrated effective advantages in operating time, the potential of these heat exchange systems, their thermal effectiveness under different working conditions and to what extent a change of operation could imply a variation in the levels of fuel savings recorded are not known.

From a set of comparative readings of operating parameters of the industrial refrigeration installation and applying the method of evaluating the number of thermal units transferred per unit area as a function of effectiveness (NTU-effectiveness)[8], the necessary elements for the development of this research are obtained.

2. Methods used and experimental conditions

The refrigeration cycle installed in an ice cream factory is intended to meet the refrigeration needs of the ice cream production and preservation process. **Figure 1** shows the location of the heat recovery unit in the two-stage compression refrigeration system.

The refrigeration system under analysis is basically composed of a compressor bank, a water-cooled evaporative condenser, a liquid receiver tank, an intermediate separator cooling tank, a low pressure evaporator system consisting of the coolers, the paddle machine and the freezing equipment, as well as an intermediate pressure evaporation system where the flavoring cooler, the storage tanks and the ice bank are located.
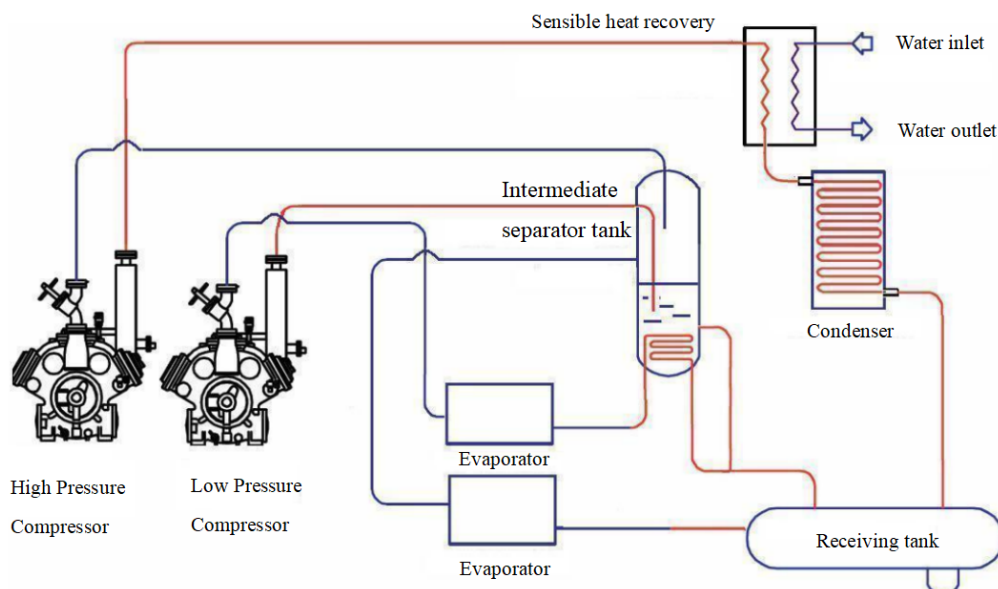
Figure 1. Location of the heat recovery unit in the refrigeration system.

Recovering heat from the compressor in the cooling of the compressor jackets, considering also consider the refrigerant vapor leaving the compressor in the discharge, is for the authors

criterion, the most important opportunity for heat recovery in industrial refrigeration systems. The heat recovered in the industrial system will depend not only on the compressor technology, but also on

the operational parameters of the system.

Tube and shell heat recuperators are well known and widely used in thermo-energy industries due to their high efficiency in heat transmission, significantly increasing the heat transfer from the energy carrier to the cooling medium. In Cuba, a heat exchanger was patented, which contemplates among its uses that of serving as a heat recuperator[9], in refrigeration installations. This recuperator has the disadvantage of requiring granulated material with particular characteristics, which makes its construction more expensive and complex. This granular material has limited time, because it needs to be in contact with the refrigerant and oil, deteriorating its properties, so it needs to be replaced periodically. In addition, because it will have to circulate through it all the coolant, it will cause an additional pressure drop.

The heat recovery unit that is evaluated in this research, is designed from the elements set out by Professor Fernando Brossard at the International Conference of the IIR "Natural Refrigerant" in Denmark in 1996[6], which recommends the use of heat recovery units that allow the advantages of an economical design, with simple operation. Installed at the discharge of the high stage compressor and before the condenser, this heat exchanger operates as a heat recovery unit, heating water and allowing the condensation of part of the lubricant vapors that are discharged together with ammonia, to be periodically purged through its lower part.

This paper evaluates the proposed heat recovery unit, a more compact heat exchanger with lower construction and operating costs. This evaluation takes into account elements of the operation of the equipment as a heat exchanger, assessing the impact of heat recovery on the cooling system, as well as elements of fuel savings in the steam generation of the industrial facility. This evaluation will provide quantitative evidence on the application of this technique in industrial facilities.

The installed heat recovery unit has novel characteristics in its construction and operation. Both flows circulate in countercurrent annular spaces, with tangential axial inlet to increase the residence time of the fluids and increase the turbulence of their hydrodynamics (**Figure 2**). There are no references in the open scientific literature of the use in Cuba of this technology in two-stage industrial refrigeration systems using ammonia as a refrigerant.
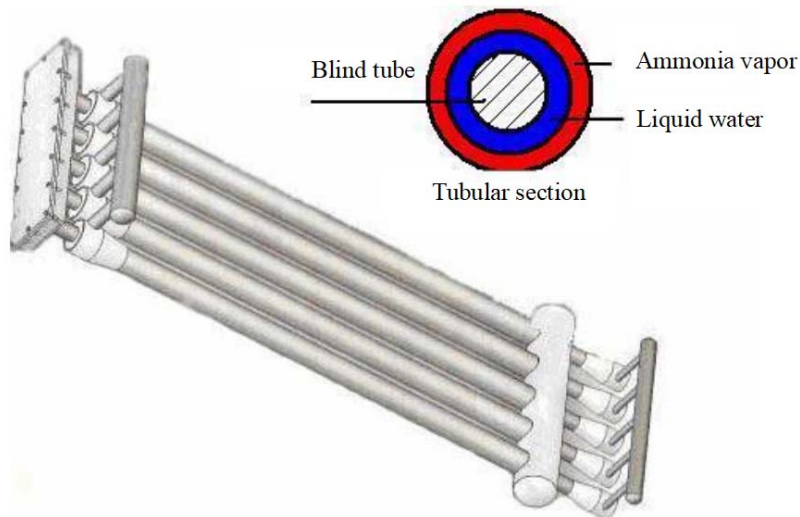
Figure 2. Schematic of the installed heat recovery equipment and tubular section of one of the heat exchanger tubes**.**

The fundamental dimensions of the designed equipment together with the measurements of its operation are of vital importance for its evaluation. The outer and inner diameter of the annulus through which the water flows measures 0.057 m and 0.050 m respectively. The outer diameter of the blind tube corresponding to the design was 0.042 m, leaving the inner diameter of the annulus through which the ammonia circulates at 0.068 m. The heat exchange length was 2.7 m.

Water temperature measurements at the inlet (Te_{H2O}) and outlet (Ts_{H2O}) and ammonia temperature

at the inlet (Te_{NH3}) and outlet (Ts_{NH3}) were carried out using an Electro-Term Model SRH77A digital thermometer with a range of -40 °C to 200 °C, with an accuracy of 0.01 °C. The water flow (m_{H2O}) was measured by the gauging method, using a container graduated in milliliters and a stopwatch. To measure the pressure in the system, we used the pressure gauges installed in the industry. Using a Fluke 337A Model 321 hook ammeter and a Fluke 430 Series II network analyzer, we were able to determine the voltage (V) and amperage per phase of the motors installed in the compressors (I_p) and the power factor ($\cos\varphi$).

The NTU-effectiveness method is used to evaluate the equipment. This method was first developed in detail by Kays and London in 1955[10], and it is based on determining the amount of heat transferred, with respect to the maximum possible amount of heat to be transferred during the given process.

In this evaluation, the stable operation of the refrigeration system was considered, performing the data collection in the periods of stable operation and from identical cooling load, starting from the variation of the exchanger water flow. This variation of the flow causes a readjustment of the parameters of the refrigeration system, so it was necessary to observe a reasonable time interval before each measurement to achieve stability in the system again.

Determining the ammonia flux, from the heat balance, can correspond to the application of the NTU-effectiveness method[11].

$$C\max = m_{H20} * Cp_{H20}$$
(1)

$$C\min = m_{NH3} * Cp_{NH3}$$
(2)

$$Cr = \frac{C\min}{C\max}$$
(3)

Where Cmin and Cmax are the heat capacities of each fluid.

$$Q_{max} = C\min^* \left(Te_{NH3} - TeH_{H2O} \right)$$
(4)

$$\varepsilon = \frac{Q}{Q_{max}}$$
(5)

Knowing the flow configuration, the effectiveness (ε) and the heat capacity ratio (Cr), it is possible to determine the number of thermal units transferred per unit area (NTU).

$$NTU = \frac{1}{1-Cr} \ln \left(\frac{\varepsilon - 1}{Cr^* \varepsilon - 1} \right)$$
(6)

Substituting (1) and (7) into (8) we can find:

$$\varepsilon = \frac{UALMTD}{C\min \left(Te_{N_3} - T_{H_2O} \right)}$$
(7)

Noting that the NTU coefficient can be calculated as:

$$NTU = \frac{UA}{C\min}$$
(8)

Another concept that is introduced in the evaluation and that has certain practical value, despite being little used, is that of the efficiency of an exchanger.

$$\psi = \frac{\varepsilon}{NTU}$$
(9)

This parameter represents the extent to which this heat transfer is carried out at the maximum temperature, so it can lead us directly to evaluate the performance of a unit as a function of the temperature of the fluids present in the heat exchanger.

In order to carry out the measurements, a stable operation of the cooling system was considered, taking data in a comparative way in the water flow interval in which the exchanger is operating. The measurements shown in **Table 1** and **Table 2** are a comparative sample of those taken to evaluate the impact of this modification. They correspond to the average values for the same cooling load in the three months before the heat exchanger was put into operation and after it was put into operation.

By analyzing the above parameters it was possible to determine the electrical power of the compressor motor that was operating:

$$W = \sqrt{3}(I_p)V * \cos\varphi$$
(10)

Where

I_p—Average amperage of electric current of the three phases (A)

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V—Compressor phase voltage (V)   Cosφ—Power factor

Table 1. Sample of the comparative parameters evaluated in the system

Suction pressure (kgf/cm^2)	Intermediate Pressure (kgf/cm^2)	Discharge Pressure (kgf/cm^2)	V (Volt)	I$_p$ (Amp)	Cosφ
Without the use of the heat recovery unit					
-0,11	2,1	12,7	440,3	123,3	0,78
-0.12	3,1	12,5	443.9	124,8	0.78
-0.10	2,8	12.3	439,2	121,5	0.79
With the use of the heat recovery unit					
-0,13	1,8	11,5	440,0	103,4	0,79
-0,18	1,6	12,2	442,1	103,6	0,78
-0,17	2,1	11,2	441,0	102,1	0,78

Table 2. Heat recovery unit operating parameters evaluated in the analysis

m$_{H2O}$ (kg/s)	0,050	0,049	0,045	0,040	0,032	0,029	0,025	0,021	0,019
Te$_{H2O}$ (ºC)	27,3	27,3	27,3	27,3	27,3	27,3	27,3	27,3	27,3
Ts$_{H2O}$ (ºC)	43,1	49,5	51,9	55,2	65,1	72,5	81,3	85,1	92,6
Te$_{NH3}$ (ºC)	123,9	122,5	121,7	121,0	120,3	118,4	119,2	118,1	116,1
Ts$_{NH3}$ (ºC)	75,9	70,3	67,9	61,4	55,4	50,3	45,9	43,7	41,2

To assess the impact of the equipment on the refrigeration system, it was considered that the system also works with a circulating pump, and that the cooling capacity in the analyzed intervals does not vary. Based on these criteria it was possible to calculate:

$$COP = \frac{\Sigma Q_e}{\Sigma W}$$

(11)

Where:

ΣQ_e = Sum of the cooling capacity operating in the installation (kw).

ΣW = Sum of the electrical power of the motors in the installation (kw).

In the industry there is a water heating system that uses fuel oil as fuel. From the results of the operation of the heat recovery unit it is possible to determine comparatively the amount of fuel that can be saved in the system.

Starting from the consideration that:

$$G_{comb} = \frac{G_{agua}(i_v - i_{ao})}{\eta \cdot vci}$$

(12)

Where:

G_{comb}—Amount of fuel used to heat the feed water (kg/h)

G_{agua}—Quantity of water (kg/h)

i_v—Enthalpy of the steam at boiler working pressure (kJ/kg)

i_{ao}—Enthalpy of feed water (kJ/kg)

vci—Low calorific value of fuel (kJ/kg)

η—Boiler efficiency

These calculations took into account the working pressure of the steam generator of the plant 5.50 kgf/cm^2, the boiler efficiency is $\eta = 0.80$ and the low heating power of the fuel is vci = 41,800 kJ/kg.
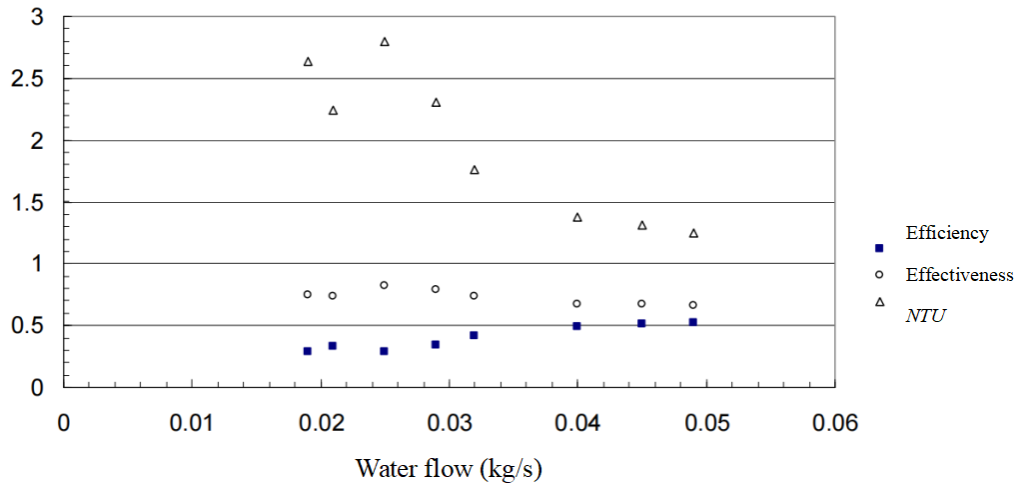
3. Results and discussion

From the equations developed above, it was possible to obtain the variation of different indicators of the heat exchanger performance. For the analysis, data were taken from the system operating in stationary regime, varying the water flow within a certain range that allows obtaining water outlet temperatures between 45 ºC and 90 ºC, independently of the hydrodynamics of the water.

Effectiveness values reported for various types of recuperators in other applications by Srimuang and Amatachaya[3], range from 0.16 to 0.76, although the average values are around 0.55. According to literature reports consulted[12-14], this indicator should be maintained above 60 % to be considered good.

From **Figure 3** it can be observed that the effectiveness of the equipment is maintained at values above 60 % in practically the entire operating range. This indicates that the recuperator is operating with good effectiveness. Only at water flow rates above 0.05 kg/s, there is a decrease in this indicator.

The efficiency parameter in a heat exchanger refers to the extent to which the heat exchange takes place at the maximum temperature. With regard to a heat exchanger, the water flow is limited to a

certain level, since a certain thermal level in the heat transfer fluid is desirable in order to be used effectively. Considering the above and considering that the efficiency of a heat exchanger operating in steady state must be above 35% to be considered acceptable, we can affirm, in the evaluated case, that the water flow between 0.03 kg/s [1.9 L/min] and 0.04 kg/s [2.4 L/min], provides the highest efficiency values that do not compromise the temperature level at the outlet of the heat recovery unit, as shown in **Figure 3**.

Figure 3. Behavior of effectiveness (e), efficiency (y) and thermal units transferred per unit area (NTU) as heat recovery unit water flow rate varies**.**

Derived from the heat recovery from the discharge of reciprocating compressors, there is a reduction of the temperature at the discharge and therefore the condensing temperature. This could be seen as a certain disadvantage for heat recovery, as the thermal level of the gases decreases and the amount of heat potentially recoverable also. Authors such as Reind[15] have expressed that for screw compressors this decrease can be up to 4.8 ºC/bar. The refrigeration system of the ice cream factory, object of evaluation in this work, where the refrigeration system operates with reciprocating compressors, the reduction was 7.5 ºC/bar on average.

The incorporation of this exchanger according to the proposed scheme (**Figure 1**) causes the decrease in the condensing temperature of the refrigerant and the pressure at the discharge of the compressors, as shown in **Table 1**. These results support the cause of the appreciable decrease in the power delivered by the compressor motor of the refrigeration system.

The coefficient of operation (COP), is the indicator of thermal efficiency most used to evaluate the performance of a refrigeration system. **Figure 4** shows that under the same conditions analyzed, an increase of 0.4 is achieved as an average of the operating coefficient in the refrigeration system. The increase in this coefficient that is reflected graphically, is the result of a stable evaluation for three months before and three months after the commissioning of this exchanger equipment. Recently Mihail-Dan[16] showed theoretically that it is possible to increase the COP of cooling systems up to three stages of compression.
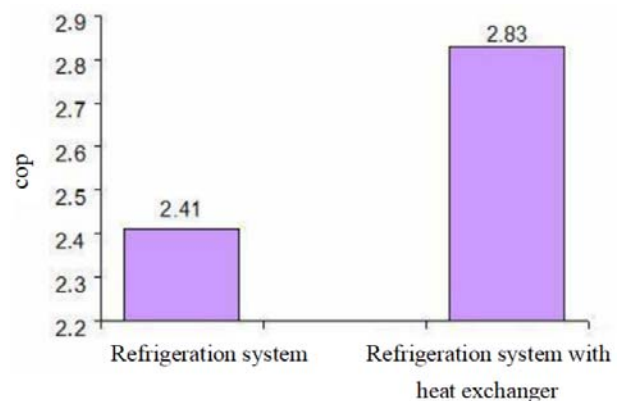
Figure 4. Variation of the coefficient of operation (COP) in the refrigeration system, from the incorporation of the heat recovery unit in the system**.**

The hot water obtained in the heat recovery unit has multiple industrial uses, one of them is to supply the steam generation system. In the industry

where the heat recovery unit is located, a decrease of 0.2 kg/h in the fuel required by the steam generation system to heat the water was observed. This decrease is supported by the daily records in the installation itself, which has corroborated the validity of the results obtained.

Figure 5 illustrates a correspondence between the fuel consumption (Gcomb) required for the steam generation process in the industry and the temperature of the water leaving the heat recovery unit. It is possible to appreciate that there is an operating point that allows reaching a temperature of about 70 ºC in the water leaving the recuperator and a fuel consumption of 0.8 kg/h in the steam generation installation. Considering that the quality of this heat recovery is determined by the temperature level of the water, and taking into account that the equipment must be operated with effectiveness levels above 0.6, we can recommend that the water flow to be operated in the system is 0.3 kg/s, since it guarantees adequate parameters from the thermal point of view, guaranteeing an outlet water temperature of around 70 ºC. The amount of energy that is possible to recover from

the compressor discharge will depend on the minimum temperature of the process where it will be used.

The amount of energy that can be recovered from the compressor discharge will depend on the minimum temperature of the process to which it will be destined. In the system under study, heat is recovered to supply the steam generation system, which requires high temperatures. If the primary use of the hot water does not require such high temperature levels (cleaning, industrial processes), the analysis could lead to other results since it is possible to maintain high levels of efficiency using water flows close to 0.05 kg/s.

An analysis of the use of hot water from the heat recovery process is recommended, to indicate the water flow at which to operate the system, performing an associated thermal analysis to ensure efficient operation of the heat exchanger.

Considering a water temperature at the recuperator outlet of 60 °C, the recovered heat rates of 0.15 $kW_{thermal}/kW_{cooling}$, higher than the one reported by Reindl (0.11 $kW_{thermal}/kW_{cooling}$)[14], for heat recovery in screw compressor systems.
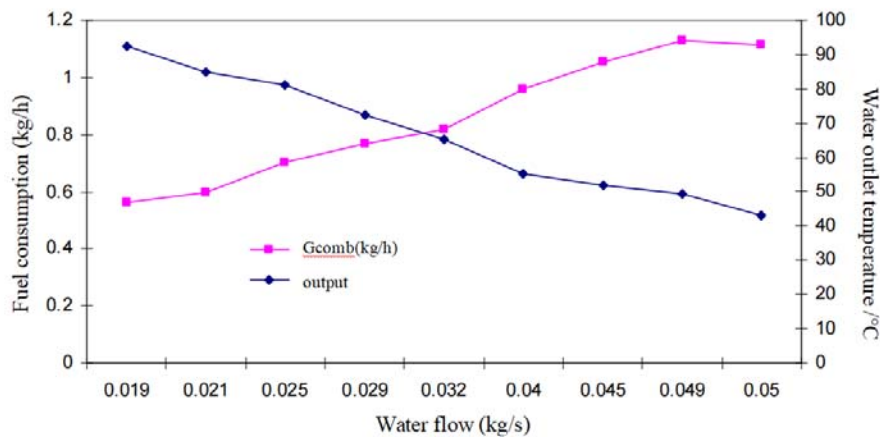
Figure 5. Variation of fuel consumption required to achieve water heating in the steam generator of the industrial facility.

4. Conclusions

The results obtained show that the performance of the installed heat recovery system is operating with adequate effectiveness rates (67%), with an efficiency higher than 35%, for 68.9 ºC water outlet temperature. In the evaluated water flow range, the thermal operating parameters of the heat exchange system are at adequate levels. The incidence of the outlet water temperature on the

quality of the heat transfer process requires a balance between the thermal level at which the water is required and the efficiency of the heat recovery unit, in order to guarantee an adequate operation of the system from the thermal point of view.

The implementation of the heat recovery allows the increase of the efficiency of the double stage compression refrigeration system (COP) by 0.43, with respect to its operation without the heat

recovery system. This was caused by the decrease of the condensing pressure and the lower consumption of the electric motors of the compressors of the refrigeration system. This allows saving of resources and more stable operation of the refrigeration system.

The saving of the fuel used for heating water for steam generation after the heat recovery unit start-up was approximately 0.2 kg/h, taking into account the water flows used in the system. This is a very important element since it makes it possible to reduce the cost associated with the production of the industry and to increase industrial efficiency.

The rates of thermal utilization (0.15 $kW_{thermal}/kW_{refrigeration}$), and of reduction in discharge temperature (7.5 ºC/bar), are comparatively higher than those reported in the literature, which makes it possible to recommend its use in industrial refrigeration installations.

Conflict of interest

The authors declared that they have no conflict of interest.

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