THERMODYNAMIC DIAGNOSIS TECHNIQUES TO ASSESS THE BEHAVIOR OF VAPOUR COMPRESSION REFRIGERATION SYSTEMS

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Abstract. *Nowadays, chilling and freezing processes are extensively used in the food industry, either during production or storage of manufactured food. Thus, it is of extreme importance to have a close control over the equipment used in these facilities, in order to make them able to operate at maximum efficiency under all conditions. Considering that the refrigeration and air conditioning systems are huge consumers of electricity, it is clear that this practice allows for a significant reduction in energy consumption at a global level. In the last years, mathematical models of refrigeration and air conditioning systems have been used to analyze their operation under various working conditions and also to predict their performance. Considering this context, the aim of this work is to develop a method for preliminary diagnosis of vapour compression refrigeration systems, using the thermoeconomic methodology. The developed method will be used to evaluate the degradation of the system components, and also the influence of these degradations on the efficiency and power consumption of the system. Through this analysis, it is expected to obtain a prognosis of the system behavior, which can be used as an auxiliary tool for maintenance scheduling, as well as to perform a complete analysis of the whole system and of each one of its component.*
Keywords*: thermoeconomic diagnosis; refrigeration system; energy consumption; computer modeling.*

1. INTRODUCTION

For appreciable reductions in energy consumption of refrigeration systems it is necessary to conduct a detailed system analysis under the perspective of the first and second laws of thermodynamics. Through the principles of thermodynamics and economics, the thermoeconomy evaluates a given system using concepts of products and inputs in the form of exergetic flows, i.e., considering the second law of thermodynamics.

The monitoring of refrigeration and air conditioning systems is driven by researches focused on energy savings, and also aiming to achieve less expensive repairs, periodic maintenance and minor stoppages. According to Navarro-Esbrı´ et al. (2006), the literature related to methods of fault detection in vapour compression refrigeration systems is limited and only a few papers can be found.

The FDD (Fault Detection Diagnosis) system analyzes operating data, establishing trends for damages or operating standards previously established, recognizing the presence of a fault. D'Accadia and Rossi (1998) simulated failures in components of a refrigeration system. The papers described above, uses of concepts of the thermoeconomic theory to detect the anomalies added in the system in analysis. Considering the above described scenario, the aim of this work is to develop a method to able to conduct the preliminary diagnosis of refrigeration systems using the thermoeconomic theory, which will be used to evaluate the degradation of the system components, and also the influence of these degradations in the efficiency and power consumption of the system.

2. DEVELOPMENT

2.1 System Behavior

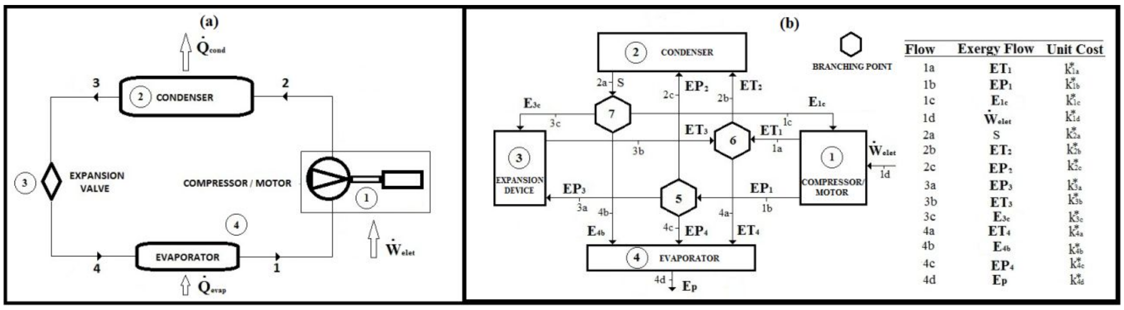

Figure 1: Simplified layout of the refrigeration system (a) and Productive structure of the refrigeration system (b).

The refrigeration system considered was a plant for cooling and freezing of 16 [ton/day] of lamb meat, with a cooling capacity of 27.5 TR. Fig. (1.a) shows the idealized system, with its basic components.

The simulation of the refrigeration system is achieved through the solution of the set of non-linear equations that governs the system operations. The necessary information (design parameters) used for modelling each component of the system were obtained from their respective manufacturer catalogue data. The parameters are listed below:

• Volume displaced by the compressor;

• Volumetric efficiency of the compressor as a function of pressure ratio;
• Capacity per unit difference of temperature for the condenser and evaporator as a function air flow of fans;
• Maximum mass flow rate provided by the expansion device (valve).

The nonlinear equations system was solved with the software EES (Engineering equation solver), and using the bisection method for the vaporization and condensation temperatures, superheat and subcooling. The solution of the system can be considered at steady state.

2.2 First and Second Laws of Thermodynamic Analysis

The energy and exergy analysis requires a mathematical formulation based on principles of thermodynamics and mass conservation, and also the establishment of boundary conditions. The following considerations were taken into account in this work:

- For each condition analysed the refrigeration system operates at steady state.
- Pressure losses in the condenser and evaporator are neglected.
- “Heat losses” except in compressor (compression process is not isentropic) are negligible.
- The kinetic and potential components of energy and exergy are also neglected.

The subscripts "e" and "s" represent respectively inputs and outputs of the control volume that is represented by the subscript "vc". The numerical subscripts are related to the state points shown of the Fig. (1.a).

The physical exergy of the flow is given by (kinetic and potential energy are not considered):

$$e_f = \dot{m}_f \cdot [h - h_o - T_o(s - s_o)] \tag{1}$$

where:

e_f : physical exergy;

$\dot{m}_f$: mass flow rate;

h : enthalpy;

s : entropy.

For some flows, the splitting of their physical exergy into its thermal (ET) and mechanical (EP) parts is necessary to better represent the flow into analysis (Morosuk and Tsatsaronis, 2008), as shown in Eq (2) and (3), where "h_m" and "s_m" are defined pressure P (actual pressure point) and T_0 (reference temperature).

$$ET = \dot{m}_f \cdot [(h - h_m) - T_0(s - s_m)] \tag{2}$$

$$EP = \dot{m}_f \cdot [(h_m - h_0) - T_0(s_m - s_0)] \tag{3}$$

The exergy of heat flow (EQ) according to Ahamed et al. (2011) is given by:

$$EQ = \sum_j \left(1 - \frac{T_0}{T_j}\right) \cdot \dot{Q} \tag{4}$$

where:

$\dot{Q}$: heat flow;

T_j : temperature in the boundary (surface heat exchange).

The negentropy (S) according to Santos et al. (2009) is defined as:

$$S = T_0 \cdot \Delta s \tag{5}$$

The temperature and pressure adopted for the dead state $T_0 = 20\ [°C]$ e $P_0 = 100\ [kPa]$ (reference condition), are respectively. The determination of the properties “h_0” and “s_0” is based on these conditions.

The application of mass conservation principle and the first law of thermodynamic in the control volume formed by the evaporator, results in the cooling capacity ($\dot{Q}_{evap}$):

$$\dot{Q}_{evap} = \dot{m}_f \cdot (h_1 - h_4) \tag{6}$$

The evaporator is modeled according to the procedure presented by Richardson et al. (2002), which is a satisfactory solution to represent this component in engineering applications, once its overall conductance is not usually provided by their manufacturers.

$$\dot{Q}_{evap} = C_{evap} \cdot \left(T_{AEE} - T_{evap}\right) \tag{7}$$

where:

C_{evap} : capacity per unit of temperature;

T_{AEE} : inlet air temperature in the evaporator;

T_{evap} : vaporization temperature.

The compression work $\dot{W}_{12}$ is given by:

$$\dot{W}_{12} = \dot{m}_f \cdot \left(h_2 - h_1\right) \tag{8}$$

According to Venturini et al. (1999) both volumetric and isentropic efficiency can be calculated as a function of the compressor pressure ratio (RP). The compressor actual volumetric efficiency (η_{VR}) is given by Eq. (9), whose coefficients are determined through a regression process using data from the component manufacturer. The isentropic efficiency (η_{isent}) is given by Eq. (10).

$$\eta_{VR} = a \cdot RP^2 + b \cdot RP + c \tag{9}$$

$$\eta_{isent} = d \cdot RP^2 + e \cdot RP + f \tag{10}$$

$$RP = \frac{P_{cond}}{P_{evap}} \tag{11}$$

where:

P_{cond} : condensation pressure;

P_{evap} : vaporization pressure.

With the volume displaced by the compressor $\dot{V}_{des}$, also obtained from the catalogue data of the component, and with the refrigerant specific volume in the compressor suction v_s , the mass flow rate can be calculated (Venturini et al. 1999):

$$m_f = \frac{\dot{V}_{desl} \cdot \eta_{VR}}{v_s} \tag{12}$$

Using the mechanical efficiency ($\eta_{mec} = 0,90$) and electrical efficiency ($\eta_{elet} = 0,90$), it is possible to determine the actual power consumed by the compressor (Richardson et al., 2002):

$$\dot{W}_{elet} = \frac{\dot{W}_{12}}{\eta_{mec} \cdot \eta_{elet} \cdot \eta_{isent}} \tag{13}$$

For the condenser, applying the principles of mass conservation and the first law of thermodynamics, it is possible to determine its rate of heat rejection ($\dot{Q}_{cond}$), thus:

$$\dot{Q}_{cond} = \dot{m}_f \cdot \left(h_2 - h_3\right) \tag{14}$$

Using the same method adopted to model the evaporator (Richardson et al., 2002), it is also possible to determine the rate of heat rejection in the condenser as a function of its main operational temperatures, as shown in Eq. (15).

$$\dot{Q}_{cond} = C_{cond} \cdot \left(T_{cond} - T_{AEC}\right) \tag{15}$$

where:

C_{cond} : capacity per unit of temperature difference;

T_{AEC} : inlet air temperature in the condenser (in general, it is equal to the ambient temperature);

T_{cond} : condensation temperature.

The process in the expansion device can be considered isenthalpic, what can also be verified using the principles of mass conservation and the first law of thermodynamics, resulting in:

$$h_3 = h_4 \tag{16}$$

The expansion device used in the system considered in this work was a thermostatic expansion valve. From data usually furnished by its manufacturer one can determine the valve coefficient (Ka) as a function of the vaporization temperature, as shown in Eq. (17). Using Eq. (18) it is possible to determine the maximum mass flow rate provided by the valve ($\dot{m}_{f\max}$). This coefficient and the maximum mass flow rate, together with Eq. (19), can be used to simulate the valve operation in any other condition (Yassuda, 1983 and Koury et al., 2001).

$$Ka = g + h \cdot T_{evap} - i \cdot T_{evap}{}^2 \tag{17}$$

$$Ka = \frac{\dot{m}_{f\max}}{\sqrt{2 \cdot \rho_3 \cdot (P_{cond} - P_{evap})}} \tag{18}$$

$$DTSA_{OPS} = \left(\frac{\dot{m}_{f\max}}{\dot{m}_f} \right) DTSA_{OS} + DTSA_{SS} \tag{19}$$

where:

ρ_3 : density of the refrigerant at the valve inlet;

$DTSA_{OPS}$: superheat operational;

$DTSA_{OS}$: superheat dynamic;

$DTSA_{SS}$: superheat static.

And finally the Eq. (20) presents the coefficient of performance of the refrigeration system (COP):

$$COP = \frac{\dot{Q}_{evap}}{\dot{W}_{comp}} \tag{20}$$

2.3 Thermoeconomic Analysis

The components of the refrigeration system are described by their specific exergy consumptions (k), or the amount of resources (inputs) needed to produce a unit of product. The unit exergy consumption can be understood as the input exergy required to generate one exergy unit of product. Naturally, the more irreversible the process is, the higher the value of the unit exergy consumption. The flows within the productive structure are characterized by their exergy (*E*), their exergy cost (E^*), also known as the exergy resources needed for producing this flow, and their unit exergy cost (k*). (Valero et al., 2004).

The first step in the thermoeconomics analysis is the development of the productive structure of the plant being evaluated. The productive structure is indispensable to define a thermoeconomic model of the plant. Once developed, it will allow establishing the productive purpose of the process units, or rather the definitions of the inputs and products as well as the distribution of the resources throughout the plant. (D'Accadia and Rossi, 1998).

With regard to the condenser, in this work, the reduction of entropy of the refrigerant is considered as its product. The expansion valve has the role of change the thermal component of the refrigerant’s exergy by means of a pressure fall, in order to make the desired refrigeration effect possible in the evaporator. These considerations lead to the productive structure shown in Fig. 1.b. From this figure, it can be inferred that a productive structure is a graphical representation of the resources distribution over the whole thermal system. In this case, the system is considered to be composed of four productive units and three fictitious devices.

In this productive structure, "F" represents inputs (resources) and "P" products. The Fuel-Product definition is represented, for each component of the system, in Table 1.

The expansion valve uses the EP_3 flow as input (Fig. 1.b), which is originated in the fictitious device "5", and E_{3c}, originated in the fictitious device "5", to form the thermal component of the exergy. The fictitious device "5" uses the EP_1 flow as input, which is originated in the compressor. The thermal component of the exergy flows ET_1 and ET_3 are considered as product of the compressor and of the expansion valve, respectively. Both are the “fuels” for the fictitious device "6", which provides the input flow ET_2 to the condenser and input flow ET_4 to the evaporator. The fictitious device "7" is responsible for allocating the product of the condenser (reduction of entropy). The condenser is responsible for the reduction of entropy generated by other components of the system. So, its product is allocated according to the entropy generated by each device, as Eq (s). (21), (22) and (23).

Table 1: Fuel-Product definition for productive structure of the refrigeration system.

No.	Process Unit	Input (F)	Product (P)	Losses (L)
1	Compressor/Motor	$W_{elet}+E_{1c}$	ET_1+EP_1	0
2	Condenser	ET_2+EP_2	S	EQ_{cond}
3	Expansion Valve	EP_3+E_{3c}	ET_3	0
4	Evaporator	$ET_4+E_{4b}+EP_4$	E_p	0
5	Branching point	EP_1	$EP_2+EP_3+EP_4$	0
6	Branching point	ET_1+ET_3	ET_2+ET_4	0
7	Branching point	S	$E_{1c}+E_{3c}+E_{4b}$	0
-	Refrigeration system	W_{elet}	E_p	0

For compressor/motor: $E_{1c} = \dot{m} \cdot T_0 (s_2 - s_1)$. (21)

For expansion valve: $E_{3c} = \dot{m} \cdot T_0 (s_4 - s_3)$. (22)

For evaporator: $E_{4b} = \dot{m} \cdot T_0 (s_1 - s_4)$. (23)

The condenser heat flow $\dot{Q}_{cond}$ does not have a useful destination for this system, thus it is not considered a product. The exergy flows EP_2 and EP_4, originated in the fictitious device "5", are inputs to the condenser and evaporator, respectively. Finally, E_p is the main product of the system, and $\dot{W}_{elet}$ is external input of the system.

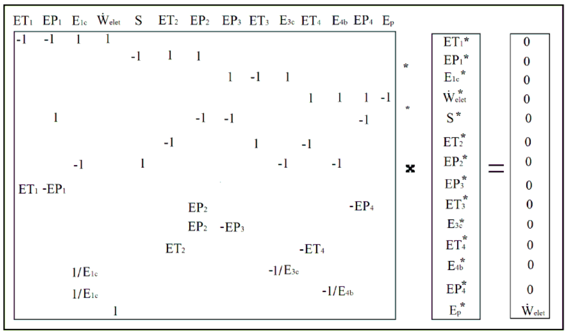

Figure 2: System of equations for calculating unit exergy costs.

Once the state of the system has been determined, it is possible to obtain the costs for all the flows that appear interrelated in productive structure shown in Fig. 1.b. The system of equations for calculating unit exergy costs is presented as a matrix in Fig 2. The equations presented on lines 1-7 (Fig. 2) are obtained through the cost balances (inputs x outputs) for each component of the productive structure.

The next equations (lines 8-13), according to Valero et al. (1986) apud D'Accadia and Rossi (1998), imposes that all the flows leaving a given unit has the same unit exergy cost. The last equation allows one to assign the cost of the resource consumed by the system (electricity, $\dot{W}_{elet}$), whose unit exergy cost is equal to one (unity).

In a general way, the thermoeconomic diagnosis consists in identifying the degradation of the components performance of a thermal system, assessing and quantifying their effects in terms of additional "fuel" (external input) consumption. The early detection and diagnosis of process faults, while the system is still operating, can prevent the progression of malfunctions and reduce economic losses.

The diagnosis method based in the following parameters: Variation in the Specific Exergy Consumption (Eq. (24)), Malfunction or irreversibility increase due to inefficiency (Eq (25)) and Fuel Impact (Eq. (26)).

$$\Delta k_j = k_j - k_{j_{ref}} \tag{24}$$

$$MF = \Delta k \cdot P^0 \tag{25}$$

$$\Delta F_T = \sum_{i=1}^{n} \left(\sum_{j=0}^{n} k_p{}^* \Delta k_{ji} \right) \cdot P_i^0 \tag{26}$$

where:

k_j : Unit exergy consumption for component *j*.

$k_{j_{ref}}$: Unit exergy consumption in reference state for component *j*.

P^0 : Product in the reference state.

k_p^* : Unit exergy cost for inlet of the plant.

P_i^0 : Product in the reference state for component *i*.

Δk_{ji} : Variation of the exergetic unit consumption for the product that enters on component *i* originated of the component *j*.

3. RESULTS AND DISCUSSION

In this analysis, the ambient temperature of 32 °C and the refrigerated space temperature of -2 °C will be assumed as reference states, i.e., the design conditions for the refrigeration plant being analyzed. The thermoeconomic diagnosis method presented will be analyzed through the introduction of hypothetical malfunctions in components of the refrigeration system.

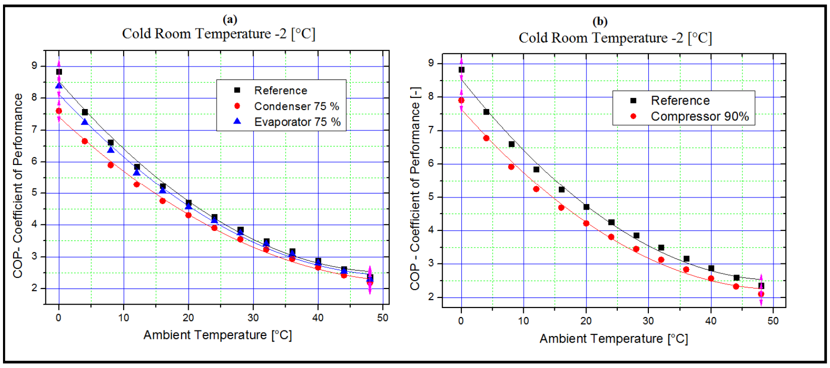

Figure 3: Impact in the COP for capacity reduction of the condenser, evaporator (a) and compressor (b).

The first malfunctions introduced in the system refer to a 25% of the capacity reduction in the condenser and evaporator. The reductions in the capacity of the heat exchangers were introduced in their capacity per unit of temperature difference, i.e., their respective C coefficient. Table 2 shows the exergetic flows for the system operating on its design condition, and also these flows when considering the existence of the malfunctions in the condenser and in the evaporator (decrease of 25 in their design capacity).

Figure 3.a shows the COP variation of the refrigeration system as a function of the ambient temperature, for a constant temperature in the refrigerated space. The COP represents the overall performance of the system in terms of the first law of thermodynamics. It is possible to observe that the reduction in the capacity of the condenser has a greater impact on the coefficient of performance of the system (COP) than the reduction in the evaporator capacity.

The capacity reduction in the condenser and evaporator leads to a reduction in the exergy flow E_p, which is related with the cooling capacity of the system. The Table 3 shows the Specific Consumption of Exergy for system when operating at reference state and for its operation with the presence of malfunctions in the condenser and evaporator.

Table 2: Exergetic flows for system operating at design condition and with malfunctions in the condenser and evaporator.

Flow	ET_1	EP_1	E_{1c}	W_{elet}	S	ET_2	EP_2	ET_3	EP_3	E_{3c}	ET_4	E_{4b}	EP_4	E_p
Reference State [kW]	9.810	13.060	4.243	33.470	112.200	9.761	0.015	9.607	12.730	3.123	9.656	104.8	0.310	9.724
Condenser (75%) [kW]	10.930	12.860	4.503	34.920	108.800	10.750	0.016	9.039	12.530	3.495	9.218	100.8	0.306	9.284
Evaporator (75%) [kW]	9.182	13.020	4.181	32.570	106.900	9.185	0.014	9.649	12.710	3.064	9.645	99.670	0.294	9.711

Table 3: Specific Consumption of Exergy for components.

Process Unit	k_{comp}	k_{cond}	k_{evap}	k_{disp}
Reference State [-]	1.6492	0.0871	11.8057	1.6497
Condenser (75%) [-]	1.6576	0.0989	11.8868	1.7719
Evaporator (75%) [-]	1.6553	0.0860	11.2871	1.6352

Table 4 shows the Variation in the Specific Exergy Consumption of the system operating with the presence of malfunctions in the condenser and evaporator. In this table it is observed that the presence of malfunctions in the condenser and evaporator leads to an increase in the Specific Exergy Consumption in the compressor, since they also increase the generation of the irreversibility in compressor. In Table 4 it can also be observed an increase in the Specific Exergy Consumption of all remaining devices, especially in the expansion valve due to the increased condensing pressure of the system.

The Table 5 shows the Malfunction in the equipments due to the anomalies in the condenser and evaporator. Thus it is showed the result of irreversibility on compressor due to the presence of anomalies in the condenser and evaporator.

The Table 6 shows the Fuel Impact in the compressor due to malfunctions in the condenser and evaporator. One can observe that there is an increase of energy consumption in the compressor due to both malfunctions, and especially due to the malfunction in the condenser.

Table 4: Variation in the Specific Consumption of Exergy.

Process Unit	Δk_{comp}	Δk_{cond}	Δk_{evap}	Δk_{disp}
Condenser (75%) [-]	0.0084	0.0118	0.0817	0.1208
Evaporator (75%) [-]	0.0061	-0.0011	-0.5180	-0.0159

Table 5: Malfunction for Components.

Process Unit	MF compressor	MF condenser	MF evaporator	MF expansion valve
Reference State [kW]	0	0	0	0
Condenser (75%) [kW]	0.1930	1.3229	0.7941	1.1603
Evaporator (75%) [kW]	0.1397	-0.1224	-5.0369	-0.1528

Table 6: Fuel Impact for Components.

Process Unit	ΔF_T compressor
Reference State [kW]	0
Condenser (75%) [kW]	0.2373
Evaporator (75%) [kW]	0.1706

In this analysis it was also considered as another malfunction a reduction in the capacity of the compressor (10% reduction). The reduction in the capacity of the compressor was introduced through its isentropic efficiency, simulating, for example, a reduction in the heat rejection from its cylinders during compression process. The Table 7 shows the exergetic flows of the system when working at its design condition and with the presence of malfunctions in the compressor. Fig. 3.a shows the variation of the COP of the refrigeration system as a function of the ambient temperature, for a constant temperature in the refrigerated space.

The capacity reduction in the compressor leads to a reduction in the exergetic flow Ep, which is related with cooling capacity and leads to a increased consume of electrical power of the system. The Table 8 shows the Specific Exergy Consumption of the system when operating at reference state and for its operation with the presence of malfunction in the compressor.

Table 9 shows the Variation in the Specific Exergy Consumption of the system operating with the presence of the malfunction in the compressor. In this table it is observed that the presence of malfunction in the compressor leads to an increase in the Specific Exergy Consumption n own compressor, since they also increase the generation of the irreversibility on this component. In table 9 it can also be observed an increase in the Specific Exergy Consumption of all remaining devices.

Table 7: Comparison between reference state and state with anomaly in the compressor.

Flow	ET_1	EP_1	E_{1c}	W_{elet}	S	ET_2	EP_2	ET_3	EP_3	E_{3c}	ET_4	E_{4b}	EP_4	E_p
Reference State [kW]	9.810	13.060	4.243	33.470	112.200	9.761	0.015	9.607	12.730	3.123	9.656	104.8	0.310	9.724
Compressor (90%) [kW]	10.460	13.040	6.716	37.310	114.400	10.400	0.016	9.565	12.720	3.150	9.623	104.500	0.310	9.690

Table 8: Specific Consumption of Exergy for compressor.

Process Unit	k_{comp}	k_{cond}	k_{evap}	k_{disp}
Reference State [-]	1.64921	0.08714	11.80571	1.64970
Compressor (90%) [-]	1.87312	0.09109	11.81194	1.65868

Table 9: Variation in the Specific Consumption of Exergy.

Process Unit	Δk_{comp}	Δk_{cond}	Δk_{evap}	Δk_{disp}
Reference State [-]	0	0	0	0
Compressor (90%) [-]	0.22390	0.00400	0.00680	0.00760

Table 10: Malfunction for compressor.

Process Unit	MF compressor	MF condenser	MF evaporator	MF expansion valve
Reference State [kW]	0	0	0	0
Compressor (90%) [kW]	5.11984	0.44337	0.06634	0.07310

The Table 10 shows Malfunction in the equipments due to anomaly in the compressor. Thus it is showed the result of irreversibility on compressor due to the presence of anomalies in this device.

Table 11: Fuel Impact for compressor.

Process Unit	FI compressor
Reference State [kW]	0
Compressor (90%) [kW]	6.33788

The Table 11 shows the Fuel Impact in the compressor due to its own malfunction. One can observe that there is an increase in the energy consumption of the compressor due to lower mass flow rate.

4. CONCLUSIONS

This study aimed to develop a method for preliminary thermodynamic diagnosis of vapour compression refrigeration systems. So, for a further satisfactory evaluation of the refrigeration system, due to the presence of anomalies in their components, the thermoeconomic analysis (exergetic) should be used, since it helps to understand how the irreversibilities can affect the operation of the system as a whole.

The anomaly’s condenser had more impact in the COP, compared with the anomaly’s evaporator. The anomaly’s condenser had more impact in the pressure ratio resulted t, which results in higher the compression work.

The reduction in the electrical work is observed in the evaporator's anomaly, due to a higher specific volume in the suction of compressor (reduced mass flow). This effect is responsible also for decreased the flow related with cooling capacity. This explained the negative values which are obtained in the Malfunction and also a variation in the Specific Consumption of Exergy for anomalies in the evaporator.

The reduction in the cooling capacity is observed in the condenser's anomaly, due to a higher pressure ratio which reducing the compressor volumetric efficiency (reduced mass flow). This effect is responsible also for decreased the flow related with cooling capacity. This explained the values which are obtained Variation of Exergy Specific Consumption, malfunction and Fuel Impact. This anomaly has a higher impact in the pressure ratio, affecting the isentropic and volumetric efficiency.

The reduction in the cooling capacity is observed in the compressor’s anomaly due to a reduced mass flow rate. This anomaly also increased the pressure ratio that reduces the volumetric and isentropic efficiency. This explained the increasing of the flow related to the electrical work. This anomaly has a higher impact in the isentropic efficient, thus explaining, the higher value compressor’s Fuel Impact.

Therefore, the thermoeconomic diagnostic helps in the monitoring and maintenance of the refrigeration system. This technique predicts the impact, for example, in the system performance (COP), in the input and in the cooling capacity due to elimination of anomalies.

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