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BEHAVIOR ASSESSEMENT OF VAPOUR COMPRESSION REFRIGERATION SYSTEM DUE TO PRESENCE OF MALFUNCTIONS IN THE COMPRESSOR USING THE THERMOECONOMIC METHODOLOGY

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Abstract: *The chilling and freezing processes are extensively used in the food industry, either during production or storage of manufactured food. And in the last years, the change in the nutrition habits of people around the world caused an increased demand for chilled or frozen foods ready for consumption. It is well known that refrigeration systems are huge energy consumers, either in the food industry or in the air conditioning systems for comfort. Many attempts to reduce the energy consumption of these systems are under study, due their importance in global primary energy consumption. Mathematical models of these systems are frequently used to analyze its operation under various operational conditions and to improve their efficiency. Considering this scenario, the aim of this work is to apply a thermoeconomic methodology to analyze the behavior of vapour compression refrigeration systems, and to assess the influence of compressor malfunctions on the overall system performance. The analysis will be focused on the evaluation of the degradation of the system due to compressor malfunctions, and also on the influence of these degradations on the efficiency, cooling capacity and power consumption of the system. Through this paper 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 especially of the compressor.*

Keywords: Thermoeconomic analysis; refrigeration system; energy consumption; computer modeling.

1. INTRODUCTION

The vapor compression refrigeration systems have been used in numerous applications such as household refrigerators and air-conditioned for residential and public buildings. According to Ding (2007) air conditioning systems represent a amounting up to 40 [%] of the total electricity consumption in the summer in China. The refrigeration systems represent a large part of the energy consumption in commercial and residential sector, and according to Tagliafico et al. (2012) refrigerator freezers are responsible to 8 [%] of the residential electricity consumption in USA.

The monitoring process 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. Computational simulation has been widely used for performance prediction and optimum design of refrigeration systems (Rasmussen and Shenoy, 2012). According to Ahamed et al. (2011) to obtain an appreciable reduction 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 (Santos et al. 2009).

The classical thermoeconomic diagnosis, according to Verda (2004), consists in comparing two conditions of the system, the condition to be tested and the condition set as reference. Thermoeconomic diagnostic techniques are often used in power generation systems, for example, the thermoeconomics concepts of diagnosis and prognosis are discussed in the Lazzaretto et al. (2006). In Torres et al. (2002), a plant with a gas turbine is analyzed through thermoeconomics concepts, focusing on the impacts generated on fuel consumption due to the presence of irreversibility or induced malfunction in the components of the cycle. In Valero et al. (2002), a plant with a gas turbine is also analyzed using a thermoeconomic diagnosis approach and considering its operation under various conditions. Valero et al. (2004) observes that the main aim of thermoeconomic diagnosis is to find the causes of irreversibilities on a given system and assess their impact on energy consumption.

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. Considering the above described scenario, the aim of this work is present a method to conduct a preliminary diagnosis and prognosis of refrigeration systems using the thermoeconomic theory, which will be used to evaluate the degradation due to compressor malfunctions, and also the influence of these degradations in the efficiency and power consumption of the system.

2. DEVELOPMENT

2.1 System Behavior

The computer simulations and mathematical models of refrigeration systems have been developed over the last years to analyse its operation under various operational conditions, according to Blanco et al. (2012). The models and simulations techniques can help in the improvement of refrigeration systems, increasing their efficiency.

The operational behaviour of a refrigeration system as a whole depends on its individual components comportment, which are: compressor, evaporator, condenser and expansion device. Each of these components must be in equilibrium with the other ones. In order to predict the behaviour of refrigeration systems, mathematical models and computer simulations becomes necessary (Richardson et al. 2002).

In this work, the thermoeconomic diagnosis will be conducted through the development of a case study. Hence, 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]. For the cooling load calculation and dimensioning of the system the following temperatures were considered: 32.0 [°C] for the ambient (external) conditions and -2.0 [°C] for the product storage conditions within the refrigerated space. The refrigerant used was the HCFC-22. Fig. 01 shows the idealized system, with its basic components and reference conditions.

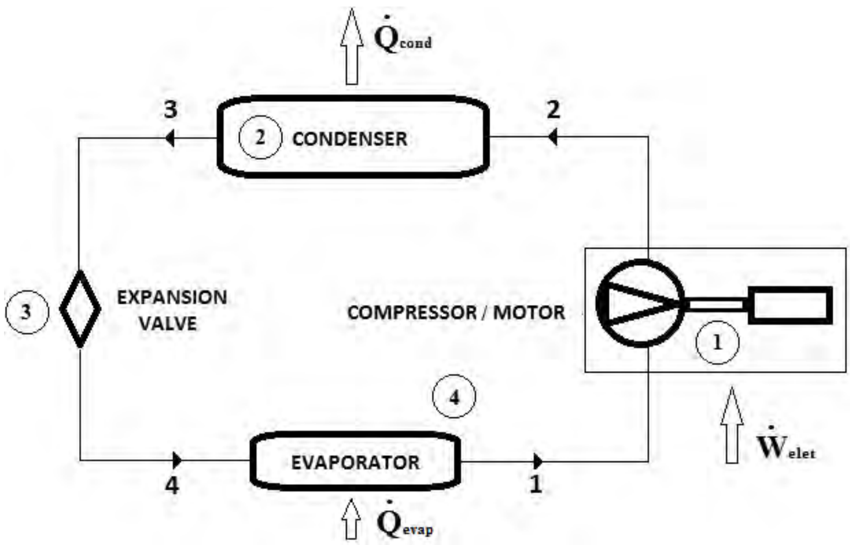

Figure 01: Simplified layout of the refrigeration system.

The simulation of vapor compression refrigeration systems is beneficial to establish system performance over a rigidly controlled set of independent system variables. (Zhou et al. 2010). The simulation of the refrigeration system is achieved through the solution of a set of non-linear equations that governs the system operations. With the model, one can obtain the behaviour of the system for various conditions of the refrigerated space and ambient temperature. 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 of the air flow of fans;
- Maximum mass flow rate provided by the expansion device (valve).

The parameters mentioned above were utilized to characterize the components and to simulate the behavior of system components working together. The nonlinear equations system obtained through the mathematical modeling of the compressor, condenser, evaporator, and expansion device was solved with the software EES (Engineering equation solver), and using the bisection method for the vaporization and condensation temperatures, superheat and subcooling. For each condition considered, the solution of the system formed by the governing equations corresponds to an equilibrium condition of the refrigeration system, i.e., a point where the system can be modeled at quasistatic process. (Van Wylen et al. 2003).

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2.2 First and Second Laws of Thermodynamic Analysis

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

- For each analysed condition, the refrigeration system operates at quasistatic process and may be considered in equilibrium 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).
The physical exergy of the flow is given by Eq. (1) (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 Eq. (3), where "h_m" and "s_m" are calculated at 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. (4).

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

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 in the Eq. (5).

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

The temperature and pressure adopted for the dead state are, respectively, $T_0 = 20\ [°C]$ e $P_0 = 100\ [kPa]$ (reference condition). 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 ($\dot{Q}_{evap}$) in Eq. (6).

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

The evaporator is modeled according to the procedure presented by Richardson et al. (2002), which is a satisfactory

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solution to represent this component in engineering applications, once its overall conductance is not usually provided by their manufacturers (Eq. (7)).

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

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 Eq. (8).

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

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 in the Eq. (12) (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 in the Eq. (13) (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 ($\dot{Q}_{cond}$) in the Eq. (14).

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

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).

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

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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 thermodynamic, resulting in:

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

The expansion device used in the system 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;

$DTSA_{OPS}$: Operational superheating;

$DTSA_{OS}$: Dynamic superheating;

$DTSA_{SS}$: Static superheating.

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 thermoeconomic analysis, according to Erlach et al. (1999), is a branch of Thermodynamics that combines the exergy analysis and economic principles to provide information usually not available through conventional first law (energy) analysis. This methodology is largely used for solving engineering problems (Valero et al. (2002). For its application, it is considered that the system interacts with their environment, consuming some external resources, inputs (F), also designated as fuel, which is transformed into products (P). Eq. (21) shows the basic relation of the fuel-product for the productive units, as shown in Eq. (21).

$$\mathrm{F} - \mathrm{P} = \mathrm{I} \geq 0 \tag{21}$$

where $\mathrm{I} = \mathrm{T_0 S_{ger}}$ is the irreversibility generated in the process (Valero et al., 2002).

The product exergy is defined according to the component under consideration, while the fuel represents the source that is consumed in generating the product. In the cost balance formulation there is no cost term directly associated with the exergy destruction of each component. The cost associated with the exergy destruction in a component or process is a hidden cost according to Lazzaretto et al. (2006). The exergy (or exergetic) efficiency (η_{exe}) can be written in mathematical terms as Eq. (22).

$$\eta_{exe} = \frac{P}{F} \tag{22}$$

The components are described by their specific exergy consumptions, or the amount of resources needed to produce a unit of product. Therefore, the reciprocal of the exergy efficiency is defined as the unit exergy consumption (k), as shown in Eq. (23).

$$k = \frac{F}{P} \tag{23}$$

The unit exergy consumption (k) 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*), as follow in Eq. (24).

$$k^* = \frac{E^*}{E} \tag{24}$$

The first step in the thermoeconomics analysis is the development of the productive structure of the plant being evaluated. The productive structure is constructed from the physical model and is made up of productive and dissipative units (represented as squares), junction (rhombus) and bifurcation (circles). The productive and the physical components do not necessarily coincide (Arena and Borchiellini, 1999). 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.

The definition of the inputs (resources) and products for the compressor and evaporator is fairly easy, whereas it is not trivial for the dissipating devices, such as the condenser and the expansion valve (D'Accadia and Rossi, 1998).

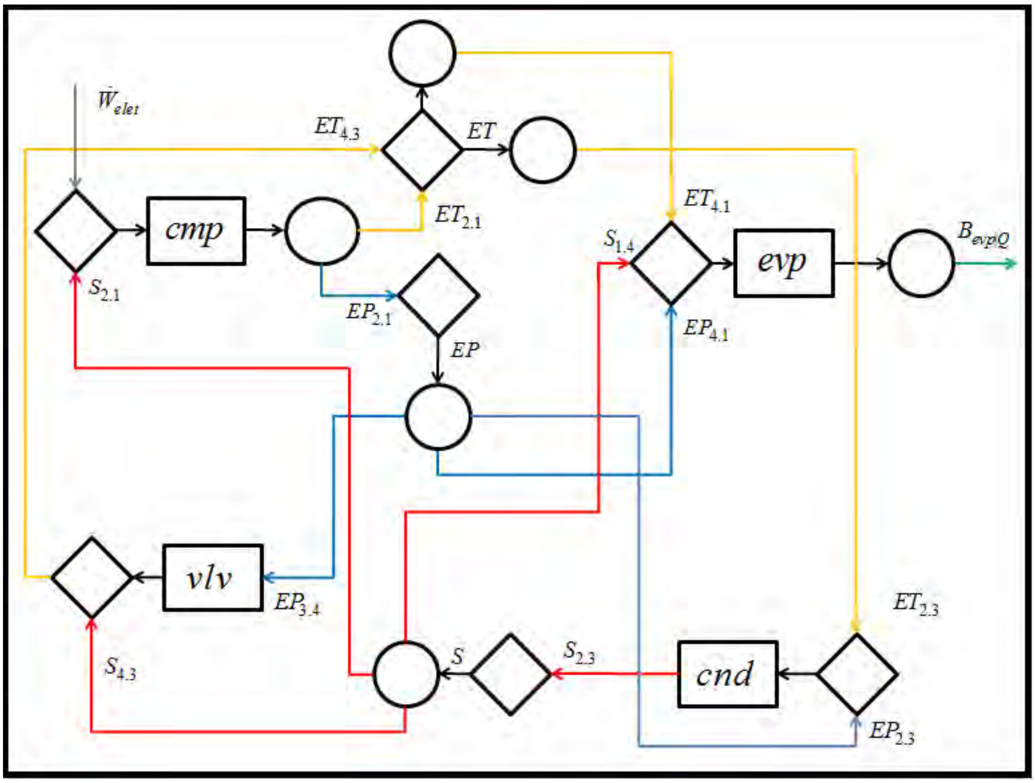

Figure 02: Productive structure of the refrigeration system of Fig. (1).

With regard to the condenser, in this work, the reduction of entropy of the refrigerant is considered as its product. Negentropy was defined as the negative variation of entropy multiplied by the temperature of the environment

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(Frangopoulos, 1987). This application represented a great advance in the discipline, since it allows one to quantify the condenser product, because the condenser is a dissipative component, whose product cannot be expressed in terms of exergy.

The expansion valve has the role of change the thermal component of the refrigerant's exergy by means of a pressure fall, in order produce the desired refrigeration effect in the evaporator. The splitting of physical exergy into its thermal and mechanical parts helps one to represent the product and input for this component (Morosuk and Tsatsaronis 2008). These considerations lead to the productive structure shown in Fig. 02. 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: compressor (cmp), condenser (cnd), expansion valve (vlv) and evaporator (evp), and three fictitious devices Bifurcation/Junction ET , EP and S .

In this productive structure, "F" represents inputs (resources) and "P" products. The Fuel-Product definition is represented, for each component of the system, in Tab. 01.

Table 01: Fuel-Product definition for productive structure of the refrigeration system.

No.	Process Unit	Input (F)	Product (P)
1	Compressor/Motor	$W_{elet}+S_{2.1}$	$ET_{2.1}+EP_{2.1}$
2	Condenser	$ET_{2.3}+EP_{2.3}$	$S_{2.3}$
3	Expansion Valve	$EP_{3.4}+S_{4.3}$	$ET_{4.3}$
4	Evaporator	$ET_{4.1}+EP_{4.1}+S_{1.4}$	$B_{evp\|Q}$
5	Bifurcation/Junction ET	$ET_{2.1}+ET_{4.3}$	$ET_{2.3}+ET_{4.1}$
6	Bifurcation/Junction EP	$EP_{2.1}$	$EP_{2.3}+EP_{3.4}+EP_{4.1}$
7	Bifurcation/Junction S	$S_{2.3}$	$S_{2.1}+ S_{1.4}+ S_{4.3}$
-	Refrigeration system	W_{elet}	$B_{evp\|Q}$

The expansion valve uses the flows $EP_{3.4}$ and $S_{4.3}$ as input (Fig. 02), which are originated in the Bifurcation/Junction EP and S, respectively. The Bifurcation/Junction EP uses the $EP_{2.1}$ flow as input, which is originated in the compressor. The thermal component of the exergy flows $ET_{2.1}$ and $ET_{4.3}$ are considered as product of the compressor and of the expansion valve, respectively. Both are the "fuels" for the Bifurcation/Junction ET, which provides the input flow $ET_{2.3}$ to the condenser and input flow $ET_{4.1}$ to the evaporator. Bifurcation/Junction S 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 follows:

For compressor/motor: $S_{2.1} = \dot{m}\cdot T_0(s_2 - s_1)$. (25)

For expansion valve: $S_{4.3} = \dot{m}\cdot T_0(s_4 - s_3)$. (26)

For evaporator: $S_{1.4} = \dot{m}\cdot T_0(s_1 - s_4)$. (27)

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 $EP_{2.1}$ and $EP_{4.1}$, originated in the Bifurcation/Junction EP, are inputs to the condenser and evaporator, respectively. Finally, $B_{evp|Q}$ is the main product of the system, and $\dot{W}_{elet}$ is external input of the system. Once the state of the system has been determined, it is possible to obtain the unit exergy cost for the overall productive, dissipative and fictitious units that appear interrelated in productive structure shown in Fig. (2). The equations system for calculating unit exergy costs is presented as a matrix in Fig (3).

The equations presented on lines 1-7 are obtained through the cost balances (inputs x outputs) for each component of the productive structure. According to Valero et al. (1986) *apud* D'Accadia and Rossi (1998), it is necessary to impose that all the flows leaving a given unit has the same unit exergy cost. The unit exergy cost of the resource consumed by the system (electricity) is equal to one (unity).

Finally, through the utilization of the thermoeconomic concepts, the diagnosis of the refrigeration system in analysis can be realized. 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 faults, while the system is still operating, can prevent the progression of malfunctions and reducing economic losses.

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$$
\begin{array}{ccccccc}
cmp & cnd & vlv & evp & ET & EP & S
\end{array}
$$

$$
\begin{bmatrix}
(ET_{2.1}+EP_{2.1}) & 0 & 0 & 0 & 0 & 0 & -S_{2.1} \\
0 & S_{2.3} & 0 & 0 & -ET_{2.3} & -EP_{2.3} & 0 \\
0 & 0 & ET_{4.3} & 0 & 0 & -EP_{3.4} & -S_{4.3} \\
0 & 0 & 0 & B_{evp|Q} & 0 & -EP_{4.1} & -S_{1.4} \\
ET_{2.1} & 0 & -ET_{4.3} & 0 & (ET_{2.3}+ET_{4.1}) & 0 & 0 \\
-EP_{2.1} & 0 & 0 & 0 & 0 & (EP_{2.3}+EP_{3.4}+EP_{4.1}) & 0 \\
0 & -S_{2.3} & 0 & 0 & 0 & 0 & (S_{2.1}+S_{1.4}+S_{4.3})
\end{bmatrix}
\times
\begin{bmatrix}
k^*_{cmp} \\ k^*_{cnd} \\ k^*_{vlv} \\ k^*_{evp} \\ k^*_{ET} \\ k^*_{EP} \\ k^*_{S}
\end{bmatrix}
=
\begin{bmatrix}
\dot{W}_{elet} \\ 0 \\ 0 \\ 0 \\ 0 \\ 0 \\ 0
\end{bmatrix}
$$

Figure 3: Equations system for calculating unit exergy costs.

Performance degradation in a vapor compression cycle increases the irreversibility generated, in the component itself, and in components related to the specific production of that component. Therefore an analysis of the use of thermoeconomic diagnosis methods is required according Ommen and Elmegaard (2012). The diagnosis method based on the structural theory and thermoeconomy introduces the following parameters according to Valero et al. (2004): The Variation in the Unit Exergy Consumption (Δk_j) in Eq. (28), Malfunction (MF) or irreversibility increase due to inefficiency in Eq. (29) and Fuel Impact (ΔF_T) in Eq. (30).

$$\Delta k_j = k_j - k_{j_{ref}} \tag{28}$$

$$MF = \Delta k \cdot P^0 \tag{29}$$

$$\Delta F_T = \Delta\, {}^t k_e P^0 + {}^t k^*{}_p \Delta \langle KP \rangle P^0 + {}^t k^*{}_p \Delta P_s \tag{30}$$

where:

k_j : Unit exergy consumption for component j.

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

Δ : Increment;

Δk : Increment in the unit exergy consumption;

${}^t k_e$: Unit exergy consumption vector of the external input (transpose);

P^0 : Product vector in the reference state;

${}^t k^*{}_p$: Unit exergy cost of the product (transpose);

$\langle KP \rangle$: Matrix of unit exergy consumption;

P_s : Final product vector.

3. RESULTS AND DISCUSSION

In this analysis, the ambient temperature of 32 [°C] and the refrigerated space temperature of -2 [°C] will be considered as reference states, i.e., the design conditions for the refrigeration plant being analyzed. The thermoeconomic diagnosis method presented will be applied to refrigeration system, for which a hypothetical malfunctions was introduced in the compressor.

The first malfunction introduced refers to a reduction in the capacity of the compressor. The anomaly was introduced through its isentropic efficiency (10% reduction), simulating, for example, a reduction in the heat rejection

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from its cylinders during compression process. The second malfunction was introduced through its decreased of the suction pressure (10 [%] reduction) in the compressor. This procedure simulates, for example, refrigerant leak in the service valve in the compressor.

The capacity reduction in the compressor leads to a reduction in the exergetic flow $B_{evp|Q}$, which is related with cooling capacity (Tab. 02). The reduction in the isentropic efficiency of the compressor leads to an increased consume of electrical power (W_{elet}) of the system. However, the reduction in the suction pressure of the compressor leads to a reduced consume of electrical power (W_{elet}) of the system, but leads also to a reduction in the exergetic flow $B_{evp|Q}$, which is related with cooling capacity (Tab. 02). These effects combined leads to a reduction in the COP, coefficient of performance of the system (Fig. 04).

Table 02: Exergetic flows for system operating at design condition and with malfunctions in the compressor.

Flow	$ET_{2.1}$	$EP_{2.1}$	W_{elet}	$ET_{2.3}$	$EP_{2.3}$	$ET_{4.3}$	$EP_{3.4}$	$ET_{4.1}$	$EP_{4.1}$	$B_{evp\|Q}$
Reference State [kW]	9.806	13.060	33.472	9.762	0.015	9.612	12.735	9.656	0.310	7.691
Isent. Efficiency (90[%]) [kW]	10.459	13.045	37.314	10.406	0.016	9.570	12.720	9.623	0.310	7.670
Pres. Suction (90[%]) [kW]	8.854	12.838	31.862	8.844	0.014	8.481	11.100	8.487	0.276	6.906

The Fig. 04 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. It is possible to observe that the malfunctions related to the reduction in the isentropic efficiency and refrigerant leak in the service valve reduce of coefficient of performance (COP) of the system.

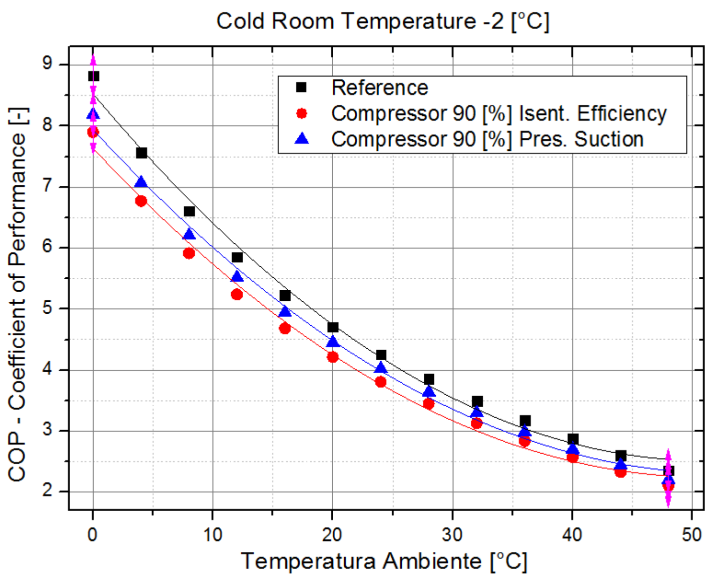

Figure 04: Impact in the COP due to malfunctions in the compressor.

The Tab. 03 shows the input and product for real productive and dissipative units for system operating at design condition and with malfunctions in the compressor. It is possible to observe that the reduction in the isentropic efficiency of the compressor reduces the input and product of the evaporator and increases the input of the own compressor. The malfunction related with refrigerant leak in the suction valves leads to a reduction of the input and product of the evaporator (Tab. 03).

Table 03: Input and product in the real productive and dissipative units for system operating at design condition and with malfunctions in the compressor.

Input/Product	F_{cmp}	P_{cmp}	F_{cnd}	P_{cnd}	F_{vlv}	P_{vlv}	F_{evp}	P_{evp}
Reference State [kW]	37.717	22.866	9.778	112.130	15.857	9.612	114.727	7.691
Isent. Efficiency (90 [%]) [kW]	44.035	23.504	10.422	114.330	15.870	9.570	114.392	7.670
Pres. Suction (90 [%]) [kW]	35.980	21.691	8.857	102.482	13.720	8.481	102.638	6.906

The Tab. 04 shows the Specific Exergy Consumption of the system when operating at reference state and for its operation with the presence of malfunctions in the compressor. The reduction in the isentropic efficiency of the

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compressor leads to an increased Unit Exergy Consumption of the own compressor. In the Tab. 04 one can observes that the refrigerant leak service valve in the compressor leads also to an increased Unit Exergy Consumption of the own compressor.

Table 04: Unit Exergy Consumption in the real productive and dissipative units for system operating at design condition and with malfunctions in the compressor.

Process Unit	k_{cmp}	k_{cnd}	k_{vlv}	k_{evp}
Reference State [kW/kW]	1.64950	0.08720	1.64977	14.91639
Isent. Efficiency (90 [%]) [kW/kW]	1.87348	0.09115	1.65839	14.91509
Pres. Suction (90 [%]) [kW/kW]	1.65872	0.08643	1.61773	14.86315

Tab. 05 shows the Variation in the Unit Exergy Consumption of the system operating with the presence of malfunctions in the compressor. In this table it is observed that reduction in the isentropic efficiency of the compressor leads to an increase in the Unit Exergy Consumption in the compressor, since they also increase the generation of the irreversibility in this component. The same can be observing for refrigerant leak in the suction valves in the Variation in the Unit Exergy Consumption in the compressor.

Table 05: Variation in the Unit Exergy Consumption in the real productive and dissipative units for system operating with malfunctions in the compressor.

Process Unit	Δk_{cmp}	Δk_{cnd}	Δk_{vlv}	Δk_{evp}
Isent. Efficiency (90 [%]) [kW/kW]	0.22399	0.00395	0.00862	-0.00131
Pres. Suction (90 [%]) [kW/kW]	0.01234	0.00187	-0.00209	-0.01135

The Tab. 06 shows the Unit Exergy Cost for real productive and dissipative units for system, when operating at design condition and with malfunctions in the compressor. In this table one can observed that the reduction in the isentropic efficiency of the compressor leads to an increase in the Unit Exergy Cost in the own compressor and evaporator. In the Tab. 06 one can observe that the refrigerant leak in the service valve leads to an increased Unit Exergy Cost of the own compressor.

Table 06: Unit Exergy Cost in the real productive and dissipative units for system operating at design condition and with malfunctions in the compressor.

Process Unit	k^*_{cmp}	k^*_{cnd}	k^*_{vlv}	k^*_{evp}
Reference State [kW/kW]	1.49396	0.16236	2.23815	4.61595
Isent. Efficiency (90 [%]) [kW/kW]	1.64168	0.18929	2.55254	5.25954
Pres. Suction (90 [%]) [kW/kW]	1.50218	0.17209	2.50104	4.90504

The Tab. 07 shows the Malfunction in the real productive and dissipative units for system operating at design condition and with malfunctions in the compressor. Thus it is showed the result of irreversibility on real productive and dissipative units for system due to the presence of anomalies in the compressor. In the Tab. 07 it is possible to observe that the reduction in the isentropic efficient and refrigerant leak in the service valve leads to an increased Malfunction of the own compressor, since they also increase the generation of the irreversibility in this component.

Table 07: Malfunction in the real productive and dissipative units for system operating with malfunctions in the compressor.

Process Unit	MF_{cmp}	MF_{cnd}	MF_{vlv}	MF_{evp}
Isent. Efficiency (90 [%]) [kW]	6.43411	0.03866	0.09627	-0.01152
Pres. Suction (90 [%]) [kW]	0.26500	-0.00753	-0.35806	-0.46999

The Tab. 08 shows the Fuel Impact in the compressor due to its own malfunction.

Table 08: Fuel Impact in the compressor for system operating with malfunctions in the compressor.

Process Unit	$\Delta F_{T,cmp}$
Isent. Efficiency (90 [%]) [kW]	2.44092
Pres. Suction (90 [%]) [kW]	0.10051

In the Tab. 8 one can observe that there is an increase of energy consumption in the compressor due to both malfunctions, and especially due to the malfunction related to reduction in the isentropic efficiency.

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 the compressor, 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 compressor refers a reduction in the heat rejection from its cylinders during compression process had more impact in the COP. This anomaly has a higher impact in the pressure ratio, affecting the isentropic and volumetric efficiency, which results in higher the compression work.

The reduction in the electrical work is observed in the anomaly refers a refrigerant leak in the service valve in the compressor, 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 Unit Exergy Consumption for this anomaly.

The reduction in the cooling capacity is observed in the anomaly refers a reduction isentropic efficient, 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. This effects combined leads the reduction in the COP, coefficient of performance of the system.

Through of the variation the input, product and Exergy Consumption of the product units can be understand how anomalies affecting the individual components of the system. The parameter Malfunction helps measured how anomalies add irreversibilities in the components of the system. Finally the Fuel Impact shows the impact in the main input of the system due to anomalies. The effects in the cooling capacity and power consumption together with parameters and concepts thermoeconomic reported in the previous may be used to assessment in the identification of anomalies in the system as a whole.

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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