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THE DIRECT METHOD FROM FINITE SPEED THERMODYNAMICS USED FOR ISENTROPIC EFFICIENCY EVALUATION OF REVERSED QUASI-CARNOT IRREVERSIBLE CYCLES

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The paper presents the analysis of the influence of irreversibilities in a reversed quasi-Carnot vapor cycle. The present computation scheme is based on recent developments of the Direct Method of Finite Speed Thermodynamics (FST). The Direct Method consists in analyzing any irreversible cycle, step by step, by writing the corresponding equation of the First Law of Thermodynamics for finite speed processes and integrating it on the whole cycle, for each process. The First Law expression for finite speed processes includes three of the main sources of internal irreversibilities, namely: finite speed interaction between the piston and the gas/vapor, friction due to the finite piston speed within the cylinder, throttling processes in the valves.

The study aims to take into account the essential differences between the behavior of perfect gases and vapors, and to analyze the changes necessary to develop a methodology for fully analytical calculating the isentropic efficiency. The principal objective of this approach is to avoid the use of property tables, by replacing them with a fully analytical calculation scheme for the irreversible cycle, in a manner similar to the development available in Classical Reversible Thermodynamics.

The analytical results were applied to a particular set of operating parameters for which the optimum piston operating speed corresponds to the minimum refrigerating power and maximum operating efficiency. Corroborating the variation of the performances (COP and power) and the evolution of the isentropic efficiency, the results show the designer what the losses are during the operation of a machine and allow it to "see" where it is necessary to intervene in order to increase the performance of refrigeration machines and heat pumps.

Keywords: Finite Speed Thermodynamics, Direct Method, irreversibilities, isentropic efficiency.

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1. Introduction

The Finite Speed Thermodynamics (FST) has been shown to be able to provide analytical evaluation of internal irreversibilities in several machines (Stirling, Otto, Diesel, Brayton, Carnot) [1-6] -and electrochemical devices [7], as a function of the speed of the piston that is also responsible of external irreversibilities, namely the finite heat transfer rate from source to cycle fluid and then to sink. The computation scheme developed in FST using the Direct Method is based on the First Law of Thermodynamics for Processes with Finite Speed that contains the main internal irreversibility causes of thermal machines expressed as a function of the piston speed that is integrated on each cycle process providing analytical expression for performance (power and efficiency). It can be used to optimize theoretical cycles of actual thermal machines and what's the most important, it was validated for 12 performing Stirling Engines (in 16 operational regimes) [1, 5, 8] and 4 Solar Stirling Motors [1-3].

By using the First Law of Thermodynamics for Processes with Finite Speed and the Direct Method, first optimization analysis of this cycle with external and internal irreversibilities is developed. Heat losses between the two heat reservoirs temperature level through the machine are considered. External irreversibilities are due to the finite rate heat transfer at the source and sink and are modeled by an irreversible coefficient added to the classical expression of heat transfer on isothermal process. Internal irreversibilities are included in the mathematical expression of the First Law of Thermodynamics for Processes with Finite Speed as non-dimensional pressure losses due to the non-uniformity of the fluid pressure in the cylinder and friction.

The aim of this approach is to analyze the influence of irreversibilities in a reversed irreversible quasi-Carnot vapor cycle (the cycle of Refrigeration Machine with Mechanical Compression of vapor), starting from previous paper [9] by using the Direct Method. A similar study was previously conducted for an irreversible Rankine cycle with finite speed [8], which analytically and graphically illustrates the optimized performance deviations from the results obtained by Curzon - Ahlborn [10] in Finite Time Thermodynamics.

The analysis and the optimization of the Carnot cycle were also given in previous paper [11]. In [4] calculation schemes were developed based on the Direct Method of analyzing the irreversibilities produced in an irreversible Carnot cycle, which works with perfect gas as a fluid and flows through four different parts of the installation. All these components are connected by pipes and valves, maintaining the ratio of constant expansion during the isothermal process, which is carried out at a high temperature, T_H. Concerned with proposing methods to optimize the performance coefficients of refrigeration machines and real heat pumps, L. Grosu [12] was also paying special attention to the analysis of pressure losses in the distribution system (suction and discharge valves) of the piston compressor with the help of Finite Time Thermodynamics. In this paper [13] the optimal value of these pressure losses corresponding to the minimum power consumed by the compressor of the refrigerating machine is determined, knowing that the generation of internal entropy in the compressor is largely due to the pressure losses in the valves. These characteristics differ fundamentally from everything previously done in the field, and it is characteristic of using the Direct Method from FST.

2. Materials and Methods

The objective of this approach is to analyze the influence of the piston speed on the isentropic efficiency, which is known from the literature to depend only on the vaporization and condensation temperatures of the refrigerant. In this context, the present approach proposes to obtain a complete analogue calculation scheme of the isentropic efficiency, in a manner similar to the studies developed within the Classical Reversible Thermodynamics, without using the tables of properties of refrigerants, starting from the comparison of the reversible cycle 1-2_r-3-4_r-1 with the irreversible cycle with finite speeds 1-2_{ir}-3-4_{ir}-1 (Fig. 1).

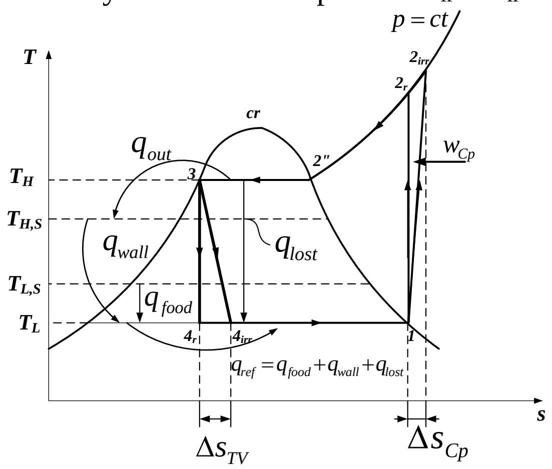

Fig. 1. The reversed quasi-Carnot irreversible cycle in T-s coordinates [9]

The study proposes a completely analytical treatment of the quasi-Carnot cycle, which operates according to the cycle described by the refrigeration machine, and in which the ratio of the extreme pressures (at the inlet and outlet of the compressor) replaces the volumetric ratio used in previous studies [3]. Furthermore, the study aims to take account of the account the essential differences between the behavior of perfect gases and vapors, and to analyze the changes necessary to develop a methodology for calculating fully analytical the irreversibilities (entropy generation [14]) and performances evaluation (efficiency and power [9]) and isentropic efficiency of such a cycle. Each process of the quasi-Carnot irreversible cycle, described in Fig.1, it is produced in different components: compressor ($1\text{-}2_{ir}$), condenser ($2_{ir}\text{-}3$), lamination valve ($3\text{-}4_{ir}$), all of which are connected by pipes (Fig.1).

Equation of the First Law of Thermodynamics for irreversible processes can be integrated (analytically) in order to obtain the process equation and also the expression of the irreversible work and heat exchange in those processes [9]. For the irreversible cycle with finite speed from (Fig. 1), this equation is integrated only for the adiabatic irreversible process $1\text{-}2_{ir}$ from the compressor, by assuming that the working fluid is a perfect gas and imposing the condition of the adiabatic process [9]:

$$mc_v^{'}dT = -p_{m,i}\left(1 \pm \frac{aw_p}{\sqrt{3RT}} \pm \frac{f\,\Delta p_f}{p_{m,i}} \pm \frac{\Delta p_{th}}{p_{m,i}}\right)dV = -p_{m,i}B\,dV \quad (1)$$

The (+) sign is used for compression, and the (-) sign for expansion.

Thus, the equations of the irreversible adiabatic compression in the compressor are obtained (completely new equations compared to previous approaches in the other branches of Irreversible Thermodynamics). These equations will contain the origin of the internal irreversibilities, namely: the finite speed of the piston and the friction between piston and cylinder.

$$-in\ T-V\ coordinates:\quad T_1V_1^{B(k'-1)} = T_{2ir}V_{2ir}^{B(k'-1)} \quad (2)$$

$$-in\ p-V\ coordinates:\quad p_1V_1^{B(k'-1)+1} = p_{2ir}V_{2ir}^{B(k'-1)+1} \quad (3)$$

$$-in\ T-p\ coordinates:\quad \frac{T_1}{T_{2ir}} = \left(\frac{p_1}{p_{2ir}}\right)^{\frac{B(k'-1)}{1+B(k'-1)}} \quad (4)$$

These equations are different from the differential equations of the adiabatic process of Classical Thermodynamics, because of the two terms B and k'. The term B takes into account the internal irreversibilities generated by finite speed, and the term k' emphasizes from the beginning the difference between the behavior of superheated vapors and the perfect gas in the compressor.

$$B = 1 \pm \frac{aw_p}{\sqrt{1.5RT_1\left(1+\lambda_p^{\frac{k'-1}{k'}}\right)}} \pm \frac{2f\cdot\left(A+B^{'}w_p\right)}{p_1(1+\lambda_p)} \quad (5)$$

where: $\lambda_p = p_2/p_1$, and k' is an adiabatic exponent of the superheated vapor [12] (a completely new concept introduced by the authors to ensure the complete analytical treatment without using data from tables with successive iterations), which takes into account the essential difference between the behavior of the perfect gas and that of the R134a vapors. The analytical formulas for calculating the corrected adiabatic exponent, k ', were obtained [9].

Given the fact that the influence of the vapor behavior on the performances and the isentropic efficiency is analyzed, the interest for investigating the variation of the adiabatic exponent according to the saturated vapor temperature and the pressure at the exit of the compressor is justified.

From equation (4), T_{2ir} can be calculated. With the correlation between h and T respectively, h2ir and s2ir can be obtained immediately on the 2”-2_{ir} isobaric process, necessary to calculate the mechanical work in the compressor (h_{2ir}-h_1) (Table 1) [9]. In these equations p2ir is the pressure at the exit of the compressor (p_{2ir}=p_2 -imposed). All the properties of the vapor from the equations in the Table 1 [14] have been expresses analytically as function of the speed of the piston and of other parameters of the cycle and compressor, in Table 2 [14].

Table 1

Analytical expressions for the calculation of COP and power [14]

The cause of the irreversibility	***COP***	***Power***
Finite speed of the piston in the compressor	$(COP)_I = \dfrac{h_1 - h_{4r}}{\left[(h_{2ir})_w - h_1\right] - (h_3 - h_{4r})}$	$P_{irI} = \dot{m}\left\{\begin{matrix}\left[(h_{2ir})_w - h_1\right] - \\ -(h_3 - h_{4r})\end{matrix}\right\}$
Finite speed of the piston and the friction in the compressor	$(\mathrm{COP})_{II} = \dfrac{h_1 - h_{4r}}{\left[(h_{2ir})_{w,f} - h_1\right] - (h_3 - h_{4r})}$	$P_{irII} = \dot{m}\left\{\begin{matrix}\left[(h_{2ir})_{w,f} - h_1\right] - \\ -(h_3 - h_{4r})\end{matrix}\right\}$
Finite speed of the piston, the friction in the compressor and the throttling in the throttling valve	$(COP)_{III} = \dfrac{h_1 - h_{4ir}}{(h_{2ir})_{w,f} - h_1}$	$P_{irIII} = \dot{m}\left[(h_{2ir})_{w,f} - h_1\right]$
Finite speed of the piston, the friction in the compressor, the throttling in the throttling valve and the throttling in the compressor	$(COP)_{IV} = \dfrac{h_1 - h_{4ir}}{(w_{cpr})_{w,f} + w_{th_{Cp}}}$ with: $(w_{cpr})_{w,f} = (h_{2ir})_{w,f} - h_1$ $w_{th_{Cp}} = \Delta p_{thR} \cdot v_{2ir} + \Delta p_{thA} \cdot v_1$	$P_{irIV} = \dot{m}\left[\left((h_{2ir})_{w,f} - h_1\right) + w_{th_{Cp}}\right]$

Finite speed of the piston, the friction in the compressor, the throttling in the throttling valve, the throttling in the compressor, but also the heat losses between sources	$(COP)_V = \frac{q_{ref} - q_{lost}}{\lvert w_{cpr} \rvert + w_{th_{Cp}}} = \frac{\dot{m}(h_1 - h_{4ir}) - \dot{Q}_{lost}}{\dot{m}\left[\left((h_{2ir})_{w,f} - h_1\right) + w_{th_{Cp}}\right]}$ where: $\dot{m} = \rho_1 \frac{\pi D^2}{4} w_p$, $\rho_1 = \frac{1}{v_1}$ $\dot{Q}_{lost} = KA(T_H - T_L)$ A – the average area between the vaporizer and the condenser $A = 0.5 \cdot (A_{ev} + A_{Cd})$	$P_{irV} = \dot{m}\left[\left((h_{2irr})_{w,f} - h_1\right) + w_{th_{Cp}}\right] = P_{irIV}$

In the irreversible process 3-4$_{ir}$, the enthalpy is constant: $h_3 = h_{4ir}$, and because of this there is no need for an equation similar to the one for the compression process, in this case. Taking into account, however, that the intention is to carry out a totally analytical calculation scheme, the properties *h* and *s* on the limit curves, depending on the pressure and the temperature are expressed in Table 2 [14].

The irreversibilities generated in the throttling valves have been evaluated separately, based on the illustration from Figure 2 and taken into account when computing the COP of the entire cycle and the consumed power [9].

Table 2

Expressions of the refrigerant (R134a) properties in the main states of the cycle [14]

STATE 1	STATE 2r	STATE 2ir	STATE 3	STATE 4a	STATE 4r	STATE 4ir
$T_1 = 74.582 (p_1)^{0.1035}$	$T_{2r} = T_1 \left(\frac{p_2}{p_1}\right)^{\frac{k'-1}{k'}}$	$T_{2ir} = T_1 \left(\frac{p_2}{p_1}\right)^{\frac{B(k'-1)}{B(k'-1)+1}}$	$T_3 = -2 \cdot 10^{-11} (p_3)^2 + 8 \cdot 10^{-5} (p_3) + 253.87$	$T_{4a} = T_1$	$T_{4r} = T_1 = 74.582 (p_1)^{0.1035}$	$T_{4ir} = T_1 = 74.582 (p_1)^{0.1035}$
$p_1 - imposed$	$p_{2r} - imposed$	$p_{2ir} = p_{2r}$	$p_3 = p_{2irr} = p_{2r}$	$p_{4a} = p_1$	$p_{4r} = p_1$	$p_{4ir} = p_1$
$s_1 = -15.51 \ln(p_1) + 1128.8 = 0.0045 (T_1)^2 - 3.0762 (T_1) + 1436.2$	$s_{2r} = -0.0032 (T_{2r})^2 + 5.3361 (T_{2r}) - 428.19$	$s_{2ir} = -0.0032 (T_{2ir})^2 + 5.3361 (T_{2ir}) - 428.19$	$s_3 = -10^{-10} (p_3)^2 + 0.0004 (p_3) + 113.3 = 4.725 (T_3) - 1084.8$	$s_{4a} = -10^{-10} (p_1)^2 + 0.0004 (p_1) + 113.3 = 4.725 (T_1) - 1084.8$	$s_{4r} = s_3 = -10^{-10} (p_3)^2 + 0.0004 (p_3) + 113.3 = 4.725 (T_3) - 1084.8$	$s_{4ir} = s_{4a} + (s_1 - s_4) \left(\frac{h_3 - h_{4a}}{h_1 - h_{4a}}\right)$

$h_1 = -1.3367 (T_1)^2 + 1305.8(T_1) - 6407.8 = 15846\ln(p_1) + 51610$	$h_{2r} = 1085 (T_{2r}) - 68451$	$h_{2ir} = 1085 (T_{2ir}) - 68451$	$h_3 = -3\cdot10^{-8}(p_3)^2 + 0.1086(p_3) + 25166 = 2.1886(T_3)^2 + 124.47(T_3) - 145250$	$h_{4a} = -3\cdot10^{-8}(p_1)^2 + 0.1086(p_1) + 25166 = 1450.6(T_1) - 1084.8$	$h_{4r} = h_{4a} + x_4\ (h_1 - h_a)$	$h_{4ir} = h_3 = -3\cdot10^{-8}(p_3)^2 + 0.1086(p_3) + 25166 = 1450.6(T_3) - 1084.8$
$v_1 = 8\cdot10^{21}(T_1)^{-9.46} = 15715 (p_1)^{0.981}$	$v_{2r} = v_1 \left(\frac{T_1}{T_{2r}}\right)^{\frac{1}{k'-1}}$	$v_{2ir} = v_1\ \theta^{B(k'-1)}$	$v_3 = -2\cdot10^{-17}(p_3)^2 + 2\cdot10^{-10}(p_3) + 0.0007 = 2\cdot10^{-8}(T_3)^2 - 9\cdot10^{-6}(T_3) + 0.0018$	$v_{4a} = -2\cdot10^{-17}(p_1)^2 + 2e^{-10}p_1 + 0.0007$	$v_{4r} = v_3\left(\frac{p_3}{p_4}\right)^{k'}$	$v_{4ir} = v_3 \cdot \left(\frac{T_3}{T_1}\right)^{\frac{1}{B(k'-1)}}$
$x_1 = 1$			$x_3 = 0$	$x_{4a} = 0$	$x_{4r} = \frac{s_{4r} - s_{4a}}{s_1 - s_{4a}}$	$x_{4ir} = \frac{h_3 - h_{4a}}{h_1 - h_{4a}}$
$B = 1 \pm \frac{aw_p}{\sqrt{1.5RT_1\left(1+\lambda_p^{\frac{k'-1}{k'}}\right)}} \pm \frac{2f\left(A' + B'w_p\right)}{p_1\ (1+\lambda_p)}$				$k' = 10^{-6}(T_{2r})^2 - 0.0007(T_{2r}) + 1.2303 = 5\cdot10^{-6}(p_{2r})^2 + 0.0002(p_{2r}) + 1.1298$		

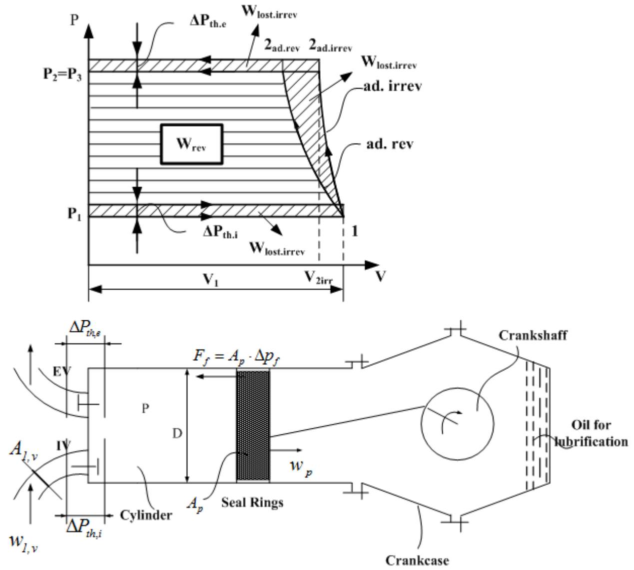

Fig. 2. The p-V diagram of the ideal reversible and irreversible compressor [9]

In the literature [3], it is appreciated that through the piston compressor valves, the flow of refrigerant vapor is laminar when the valve is opened and closed (when the flow varies linearly with the pressure gradient) and turbulent when the section provided by the valve opening is around its maximum. In the present study proposed, the losses through lamination are expressed in a manner similar to the case of internal combustion engines proposed by Heywood [15], and adapted so that it can be included in the expression of the First Law for the finite speed processes that take place in the piston machines [7,17].

The heat losses generated by the leaks between heat sources were taken into account.

In this way, the analytical expression of the coefficient of performance, COP and of the power is obtained, allowing analyzing the successive influence of all the five internal losses.

Thus, the losses through lamination can be calculated using the relation (6), depending on the piston speed:

$$\Delta p_{th} = C \cdot w_p^2 \quad where: C = 0.0045\,[8] \tag{6}$$

Once determined the losses caused by the internal irreversibilities and the losses by lamination, and starting from the relation of defining the isentropic efficiency:

$$\eta_{is} = \frac{|l_{1-2r}|}{|l_{1-2ir}|} = \frac{h_{2r} - h_1}{h_{2ir} - h_1} = \frac{T_L}{T_H} \tag{7}$$

The efficiency of the real compression process was approximated as the ratio of the absolute saturation temperatures corresponding to the discharge pressure and, respectively, the suction pressure [18]:

In order to analytically calculate it, all the causes of the irreversibilities generated during the operation of the machine were considered:

$$\eta_{is} = \frac{h_{2r} - h_1}{h_{2ir} - h_1} = \frac{80921.47\,\dfrac{p_2^{\left(\frac{k'-1}{k'}\right)}}{p_1^{\left(\frac{k'-1}{k'}-0.135\right)}} - 15846\ln(p_1) - 120061}{80921.47\,\dfrac{p_2^{\left(\frac{B(k'-1)}{k'}\right)}}{p_1^{\left(\frac{B(k'-1)}{k'}-0.135\right)}} - 15846\ln(p_1) - 120061} \tag{8}$$

where h_{2ir} it is the term that takes irreversibility into account.

Thus, for the first time in Irreversible Thermodynamics, the analytical expression of the isentropic efficiency is obtained.

3. Results and Discussions

The calculations were performed considering the same dimensions and properties, used in [9]. In the present paper, the same technical characteristics of the studied equipment are used for the calculation of losses through lamination, i.e.: the evaporator pipes length *L=1 m*, pipe diameter *D=0.05m*, $N_{pipes}=8$, $\alpha_e = 7W/m^2K$, $\alpha_i = 5W/m^2K$, $\lambda = 0.044W/mK$, $\delta_{ins} = 0.1m$ *and* $A_{Ev} = A_{Cd} = 0.176\,m^2$. It was found that the adiabatic compression process can be described qualitatively, by using an equation of the adiabatic process corrected by an adiabatic exponent that includes the deviation of the vapor behavior from that of the gas, denoted *k'*. Table 2 highlights its variation.

Using the derived (completely analytical) expressions of COP and power from [9] and isentropic efficiency, the effect of internal irreversibilities introduced progressively in the calculation of cycle performances is illustrated in Table 2 and Figure 3. Thus, the major reduction of the coefficient of performance, COP is registered when the friction losses are considered (COP_{II}), respectively the throttling, in the compressor valves (COP_{IV}) was considered. As expected, the power needed by the compressor, increases with each new irreversibility. The power required for the compressor is not affected by the heat losses between the heat exchangers, but the COP depends on each irreversibility, thus justifying the existence of 4 power curves ($P_V=P_{IV}$) corresponding to the 5 COP curves.

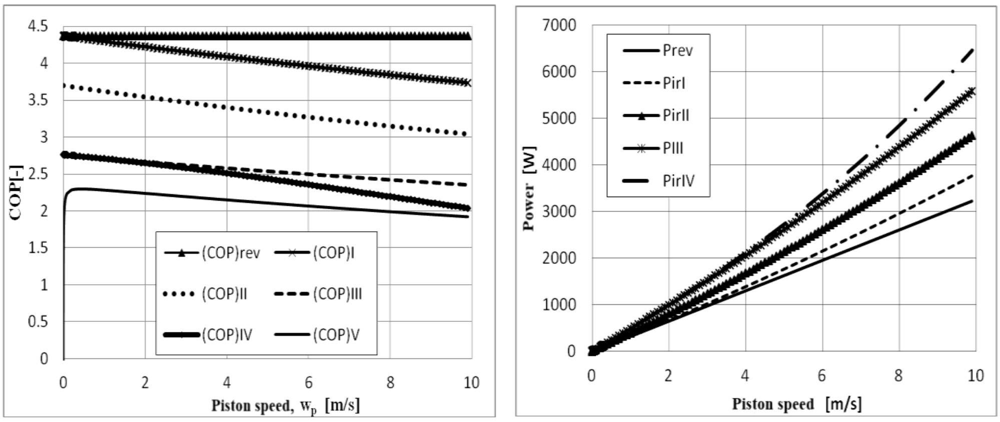

Fig. 3. The progressive influence of irreversibilities on machine performance (COP and power)

Thus P_{IV} corresponds to curves for COP_{IV} and COP_V. As expected, it is found that the Direct Method is essentially a graph-analytical method. In the reversible cycle 1-2r-3-4r, the consumed mechanical work is minimal; COP is maximum, by comparing it with any other cycle that can provide the same cooling power because in any other cycle the internal and external irreversibilities appear.

The horizontal variation in Figure 4 is obtained from the literature [18], based on the only one (empirical) formula existing until now.

It should be mentioned that the irreversibilities that differentiate the 3 curves correspond to those mentioned for calculating the powers P_I, P_{II}, and P_{IV}.

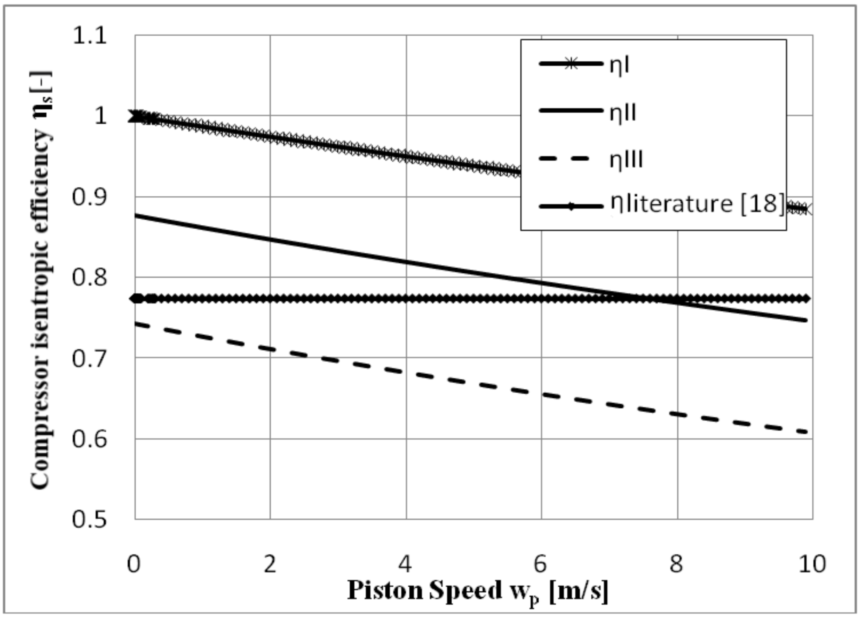

Fig. 4. Compressor isentropic efficiency depending on the piston speed, w_p.

It is found that by introducing a new irreversibility, the isentropic efficiency decreases, with the increase of the piston speed, contrary to those stated in the literature [18], according to which the efficiency is not influenced by the speed of the processes, but only by the condensation and vaporizing temperatures.

The present paper is particularly important because only by developing these analytical models for calculating the irreversibilities, the design of new machines is optimized. In this context, it is no longer necessary to construct and experiment in various fields of values of the constructive and functional parameters in order to identify the optimum in operation, designing these machines and achieving them, achievement with substantial savings of material and financial costs.

4. Conclusions

In this paper, a reversed irreversible quasi-Carnot vapor cycle was analyzed. The method of presenting the original results chosen by the authors, in order to be able to illustrate the contribution of each loss comparatively, so that the optimized machine designers can find out and know which losses to focus more on in order to reduce them. The objective of the study was the development of a methodology and its numerical exemplification. The results emphasize optimum speed values generating maximum COP; minimum consumed mechanical power, as well as the effect of irreversibilities on the isentropic efficiency.

Analysing the variation of the power and COP depending on the piston speed, it is clear that the low values of the piston speed characterize an economical operating regime, especially in terms of power consumption. For example, going from *0.5 m/s* to *1 m/s* the power increases twice, respectively from 290 W to 580 W. As previously mentioned, the power required by the compressor does not depend on the expansion process ($3\text{-}4_{ir}$), nor on the heat losses in the heat exchangers, but the COP depends on the irreversibility generated by each process.
Regarding the isentropic efficiency, which is the subject of this study, it is found that it decreases with increasing piston speed from 0.8 to 0.66 for low piston speeds (between 0.5 m/s and 1 m/s).

The study is completely original, being the first time that the Direct Method is applied in the analysis of a refrigerating machine with mechanical compression of vapor, the difference between the behavior of the perfect gas and the vapor being considered in the study of the isentropic efficiency. Regarding that, the original achievement in this paper, namely the influence of the finite speed of the piston w_p on the isentropic efficiency of the compressor with vapor is a worldwide premiere. Here also, a remarkable fact is that the progressive influence of 3 cases of irreversibilities is illustrated by tree curves: finite speed term *aw/c*, friction and throttling with the finite speed in the valves.

Corroborating the variation of the performances (*COP* and *power*) and the evolution of the isentropic efficiency with the piston speed, it appears clearly that small values of the piston speed provide economic operational regime, mainly from the power consumption reason.

Nomenclature

A	area, m^2
a	coefficient ($= \sqrt{3\gamma}$)
c_p, c_v	specific heats, $J\ kg^{-1}\ K^{-1}$
COP	coefficient of performance
D	diameter, m
f	coefficient related to the friction contribution $\in (0, 1)$
FST	Finite Speed Thermodynamics
h	specific enthalpy, $J\ kg^{-1}$
k'	corrected adiabatic exponent

K	the overall heat transfer coefficient, $W\ m^{-2}\ K^{-1}$
L	length, m
m	mass, kg
N	number of pipes, [-]
p	pressure, Pa
Δp	pressure loss, Pa
Q	heat, J
R	gas constant, $J\ kg^{-1}\ K^{-1}$
s	specific entropy, $J\ kg^{-1}K^{-1}$
T	temperature, K
ΔT	temperature variation, K
U	internal energy, J
V	volume, m^3
v	specific volume, $m^3\ kg^{-1}$
ΔV	volume variation, m^3
w	specific work, $J\ kg^{-1}$
w_p	piston speed, $m\ s^{-1}$

Greek symbols

α	convection heat transfer coefficient, $W\ m^{-2}K$
λ	thermal conductivity, $W\ m^{-1}\ K^{-1}$
λ_p	compression ratio in the compressor
k	ratio of the specific heats
ρ	density of vapor, $kg\ m^{-3}$
δ	thickness, m
η	efficiency, [%]

Subscripts

Cd	condenser
Cp	compressor
Ev	vaporizer
f	friction
H	the hot-end of the machine
i	instantaneous
ins	insulation
irr	irreversible
L	the cold-end of the engine
med	average
r	reversible
th	throttling

R E F E R E N C E S

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