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A COMPARATIVE ANALYSIS OF HYPOTHESES IN THE MODELING OF A HYBRID COMPRESSION REFRIGERATION CYCLE

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Abstract: *Comparative analysis between two thermodynamic models for an evacuated tube solar hybrid refrigeration system; The first analysis models the passage of the refrigerant fluid through the solar thermal exchanger, considering, during the passage, a constant specific volume, this added to the temperature increase suffered by the energy gain, raises the fluid pressure. The second approach models the passage of the refrigerant through the solar exchanger as a compressible flow system, this hypothesis becomes of great relevance to the study, because with it the inlet and outlet velocities of the heat exchanger become a crucial point, since they directly influence the thermodynamic state in the system, because when reaching the zone of velocities greater than that of sound in a flow with added heat, the pressure tends to increase. The results are divided into two parts, the first analysis showed that the thermodynamic proposal adopted for constant specific volume proved to be compatible with a model already studied. The second analysis, assuming the flow as compressible, was not representative for the comparison in question, but it showed promise for comparisons that adopt favorable working conditions.*

Keywords: *Evacuated Tube, Solar Collector, Electricity Saving, Thermal Compression, Refrigerant*

1. INTRODUCTION

There are numerous works already dedicated to the study of refrigeration systems. Solar cooling is a growing system. In the world, the search for more efficient systems that depend less and less on the conventional energy matrix has become a reality, as this reflects both in the reduction of energy-related costs, and in the curbing of the emission of gases that contribute to the greenhouse effect. It is no different with the air conditioning system, more and more ways have been sought to make the system more and more efficient, and sometimes the free energy available from the sun is used, either in the thermal form or in the form of light, to achieve these goals.

This work presents a theoretical analysis of a refrigeration system with hybrid solar/mechanical compression, the energy savings achieved by this system when compared to the conventional vapor compression system has already been studied by some researchers in recent times, and these studies point to encouraging values, as savings ranging from 25% to 40% have been indicated (Kumar and Patel, 2020).

Not many authors have contributed to the study of this system, but some of them are highlighted here. Brahmankar et al. (2018) brought a comparative study between a conventional model and a hybrid model admitting the refrigerant as the ideal gas inside the solar exchanger, reaching results close to 30% of savings, Assadi et al. (2016) similarly considers the gas as ideal and the constant volume in the evacuated tube exchanger, and also models through the ANSYS-FLUENT software the behavior of the heat added to the system by the solar exchanger and points to savings of 25% if the air conditioning is used for 24 hours. An experimental comparison approach is presented by Vakiloroaya et al. (2013), where a work was carried out to monitor the thermodynamic states of the real plant, and the equation was based on the ideal gas condition. Ishak (2014) concluded that the system saves 30% compared to the conventional system, and its payback equals 3 years. Dhiraviam et al. (2017) brought a study of the temperature in the heat exchanger in order to maximize the COP. Yen (2015) brings a deeper study on the solar thermal exchanger through the ANSYS software. Kumar and Patel (2020) brings an analysis under the hypothesis of ideal gas thermodynamically developing the whole equation from that, this article aims to rely on these results in order to compare them to the results obtained here.

The purpose of this work is to bring two thermodynamic modeling proposals for the solar hybrid system under analysis. The first proposal models the inlet and outlet of the solar exchanger with constant specific volume, and the gas as an ideal gas. Inserting heat into the system under such conditions results in a pressure increase when analyzed intermittently. This study brings a comparative analysis with Kumar and Patel (2020), where the same hypotheses are used, with different models. The second proposal models the passage of the refrigerant through the solar exchanger as a compressible flow system, this hypothesis becomes of great relevance for the study, because with it the inlet and outlet velocities of the heat exchanger become a crucial point, since they directly influence the thermodynamic state in the system, this is because when reaching the zone of velocities greater than that of sound in a flow with addition of heat, an increase in the outlet pressure is observed in relation to the inlet; In short, when adding heat in a supersonic compressible flow, the pressure increases, which becomes desirable for the analysis of the system in question, the study will then be around the adjustment of such velocities and states.

This work brings the proposal of modeling the system through the library of thermodynamic states of the EES software, becoming less dependent on the ideal gas hypothesis, in addition to proposing a new thermodynamic modeling for the solar thermal solar exchanger.

2. PHYSICAL STRUCTURE

The object of analysis of this work is a refrigeration system that is based on the vapor compression refrigeration system, with the presence of its 4 main components, namely evaporator, expansion valve, condenser, and compressor, the hybrid system is distinguished by the insertion of a tube solar thermal collector evacued (Fig. 1), between the compressor and the condenser. Such a configuration has been shown to be able to reduce the electrical consumption of the system globally.

When passing through the solar thermal exchanger, the fluid receives heat and raises its pressure, this pressure increase reduces the demand on the compressor, resulting in lower electrical consumption.

As much as this system shows gains in terms of electrical energy, it is not correct to say that the system is more energy efficient, since we have more energy in the form of heat being inserted. However, the system proves to be efficient when analyzed only in relation to electrical energy, since the demand on the mechanical compressor is reduced when supported by such an abundant and free source of energy as the sun.

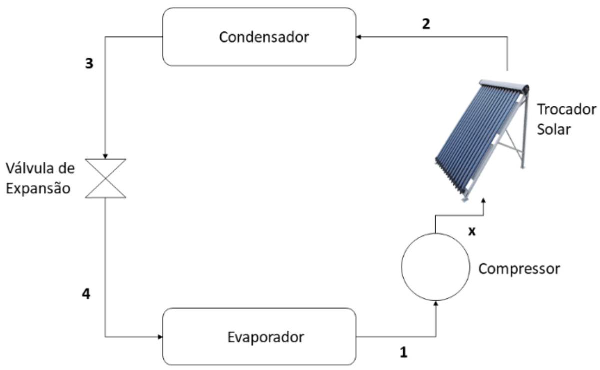

Figura 1 – Physical Structure

2.1 Thermodynamic modeling

The thermodynamic analysis of the process is the purpose of this work. The search is for a model that is minimally representative of the real system, and aims to describe the thermodynamic states before and after the evacuated solar exchanger. The expected behavior of the hybrid system is described in Fig. 2 b), and brings the concept raised in a visual way, making evident the decrease in demand on the compressor between points 1 and 2, and the increase in heat rejection in the condenser between points 3 and 4, while fig. 2 a) shows the conventional system. Tab. 1 indicates the proposed values:

Table 1. Input data

Evaporator Temperature	283 K
Condenser Temperature	312 K
Coolig Load	4,1 kW

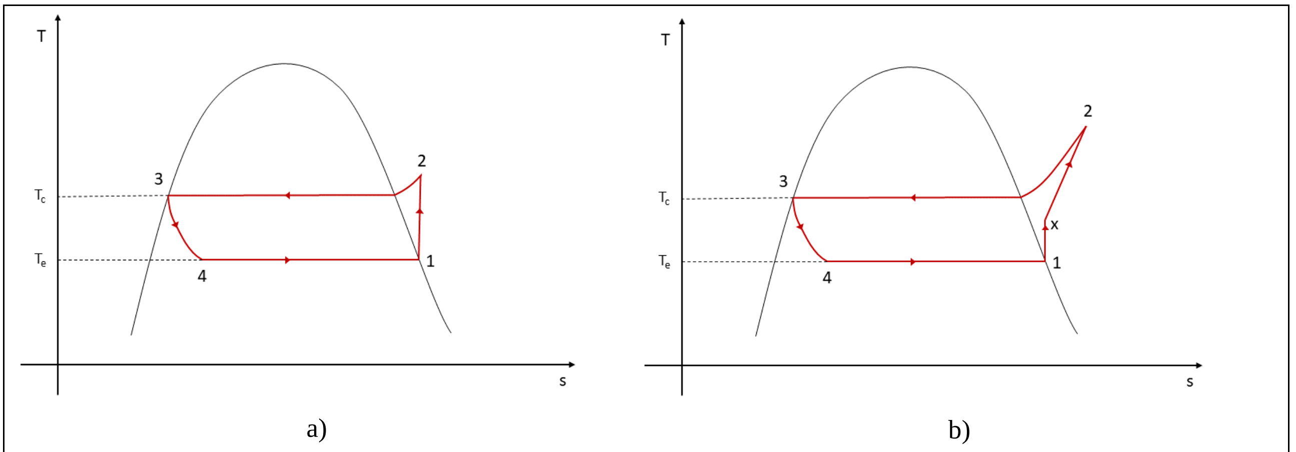

Figure 2. T x s diagram a) Conventional System b) Hybrid System

2.1.1 Mathematical equations

The energy conservation law for a given control volume can be described mathematically by Eq. 1 (Moran et al., 2009).

$$\dot{Q}-\dot{W}=\left(\frac{\partial E_{VC}}{\partial t}\right)+\dot{m}_{out}\cdot\left(h+\left(\frac{V^2}{2}\right)+gz\right)_{out}-\dot{m}_{in}\cdot\left(h+\left(\frac{V^2}{2}\right)+gz\right)_{in} \tag{1}$$

From the mathematical manipulation of the law of conservation of energy, combined with simplifying hypotheses, equations are obtained in simplified forms that define the entire ideal cycle, Fig. 2. The definition of the mass flow rate ($\dot{m}$) can be found by Eq. 2

$$\dot{m}=\frac{\dot{Q}_{in}}{h_1-h_4} \tag{2}$$

The work on the compressor ($\dot{W}_c$), by Eq.3.

$$\dot{W}_c=\dot{m}\cdot(h_2-h_1) \tag{3}$$

The coefficient of performance (COP_c), in the conventional system is estimated by the Eq.4.

$$COP_c=\frac{(h_1-h_4)}{(h_2-h_1)} \tag{4}$$

For the hybrid system, Fig. 2b), one can observe some equations that differ from those already presented. The coefficient of performance in the hybrid system (COP_h) can be calculated by Eq. 5.

$$COP_h=\frac{(h_1-h_4)}{(h_x-h_1)} \tag{5}$$

Por outro lado, o trabalho no compressor do sistema híbrido ($\dot{W}_h$), by Eq.6.

$$\dot{W}_h = \dot{m} \cdot (h_x - h_1) \tag{6}$$

The work salved ($\dot{W}_s$), by Eq.7.

$$\dot{W}_s = \dot{W}_c - \dot{W}_h \tag{7}$$

And finally, the ratio between the solar heat absorbed ($\dot{Q}_n$), by Eq. 8:

$$\dot{Q}_n = \frac{\dot{Q}_{adc}}{\dot{Q}_{adc_{real}}}$$

(8)

2.1.2 Approach admitting isovolumetric model

For the implementation of the hybrid system in this work, two proposals of thermodynamic analysis will be addressed, for the first one, it is considered between states 2 and 3, input and output respectively of the evacuated tube exchanger, flow where the specific volume is kept constant through from the EES software library. Modeling makes the heat exchanger outlet pressure equal to the pressure inside the condenser in the conventional cycle, the heat exchanger inlet pressure, P2, is also established using the EES software state library added to the equation manipulated from the first law of thermodynamics. To estimate the enthalpy of state "x", we have Eq. 9.

$$h_x = h_2 - \frac{\dot{Q}_{adc}}{\dot{m}} \tag{9}$$

2.1.3 Approach admitting supersonic compressible flow

For this approach, the problem is assumed to be a Rayleigh problem, for which crucial hypotheses are modeled. The heat exchanger process came to be considered a frictionless compressible flow, with the section having a continuous area. An also important hypothesis is that the refrigerant behaves as ideal, for this hypothesis the Compression Factor is evaluated in each refrigerant state, so there is a parameter close to behaving as an ideal gas. Through the equation for the Rayigh Problem, it is observed that using the momentum equation one can define the velocity in state 2 (V_2), through Eq. 10.

$$V_2 = V_1 + \left(\frac{P_2 - P_1}{\rho_1 \cdot V_1} \right) \tag{10}$$

The Mass equation (Eq.11) was used to define the density of state 2 (ρ_2).

$$\rho_2 = \left(\frac{m}{A \cdot V_2} \right) \tag{11}$$

Equation of state (Eq.12) to define T_2:

$$T_2 = \left(\frac{P_2}{\rho_x \cdot R} \right) \tag{12}$$

For (M) being the Mach in the fluid and (k) being the adiabatic expansion coefficient at the stagnation temperature is given by Eq.13

$$T_{02} = T_2 \cdot M_2^2 \cdot \left(1 + \frac{k-1}{2} \right) \tag{13}$$

The enthalpy at 2, (h_2) like all other state 2 parameters, is now defined by the EES Software library.

Where (c_p) is the specific heat, the stagnation temperature at x (T_{ox}) is given by Eq. 14.

$$T_{0x} = T_{02} + \left(\frac{Q \cdot c_P}{m} \right) \tag{14}$$

To set the temperature in x (T_x), was used Eq. 15.

$$T_X = \frac{T_{0x}}{\left(1+\left(\frac{k-1}{2}\right)\right)\cdot M_2^2} \tag{15}$$

The enthalpy at x (h_x) is then defined by the isentropic efficiency (η), the state x isentropic (h') and the enthalpy at 2 (h_2) by the Eq. 16.

$$h_x = \left(\frac{h' - \eta \cdot h_2}{1-\eta}\right) \tag{16}$$

Now in possession of two properties of state x it will also be defined by the EES Software library.

The velocities in the states are defined by the mass equation, once the mass flow is known and the specific mass is defined in each state, the area of the flow section based on the guide article of this work.

The Rayleigh problem defines that for subsonic flows at the entrance of a frictionless heat exchange pipe, the Mach will increase along the pipe to the maximum value of M=1 if heat is added, and its pressure will consequently decrease. However, for flows with inlet velocities above the speed of sound, the Mach will decrease to M=1 if heat is added, and in opposite to the first case, the pressure will consequently increase.

It is reasonable to consider this approach, because for the refrigerant fluids under study, in the states in which they are being evaluated, in theory, the speed of sound propagation is something achievable within the solar exchanger. Rayleigh modeling is used to model Scramjet engines, which gives the model greater reliability, but it is not useful in terms of comparison in this study, since the gas modeled in these engines is air. So far, no approaches have been found that make use of the Rayleigh problem to model this hybrid refrigeration system.

3. RESULTS AND DISCUSSIONS

The two thermodynamic models proposed for this work will be evaluated, the first considering the constant specific volume inside the solar heat exchanger, the second considering the flow as compressible and modeling the system accordingly. Finally, some working fluids will be evaluated for the two analyses. For each analysis, the results and a pre-analysis for such results were displayed.

3.1 Constant volume

The first analysis is a comparison between the model proposed in this work for the hypothesis of constant volume inside the evacuated tube solar thermal heat exchanger, and the model proposed by Kumar and Patel (2020).

To this end, graphs were developed using the same parameters, in order to compare the two works, and bring up the discussions about the difference between the two models, it is worth mentioning here once again that the reference work (Kumar and Patel, 2020) was based on the same assumption of constant volume inside the solar heat exchanger, with the difference that the data and parameters were supported and confronted with a real model of the system, where real tests were carried out. To obtain the graphs, it was necessary to enter four Parameters, all based on the comparison article; The temperature inside the evaporator (Tevap), the outlet temperature of the condenser (Tcond), the temperature at the outlet of the solar heat exchanger (T2) and the cooling in the evaporator (Qevap); The temperature unit was given in Kelvin and the cooling in kW; The initial parameters for this modeling are described in Tab. 1. Subsequently, the three temperatures described were varied and indicated in each graph, while the cooling was fixed at 4.1 kW.

In order to simplify the modeling, unlike the article cited, this modeling considered an ideal system with an isentropic efficiency of 100%.

3.1.1 Temperature variation in the solar exchanger

The first analysis, Fig. 3 a) and b), sets the evaporator and condenser temperatures and varies the temperature in state 2, and evaluates the COP's and Compressor work.

The COP and W in the conventional system are fixed, while the COP in the hybrid system changes.

When compared to the equivalent graphs (Kumar and Patel, 2020), the results show small differences, supposedly due to the fact that different models were adopted, but the trend of the curves are similar, showing the equivalence between the models, this trend indicates the result expected by the system, that by increasing the outlet temperature in the heat exchanger, which is translated by a greater amount of heat entering, we have an increase in the COP of the system, and a greater work saving in the compressor.

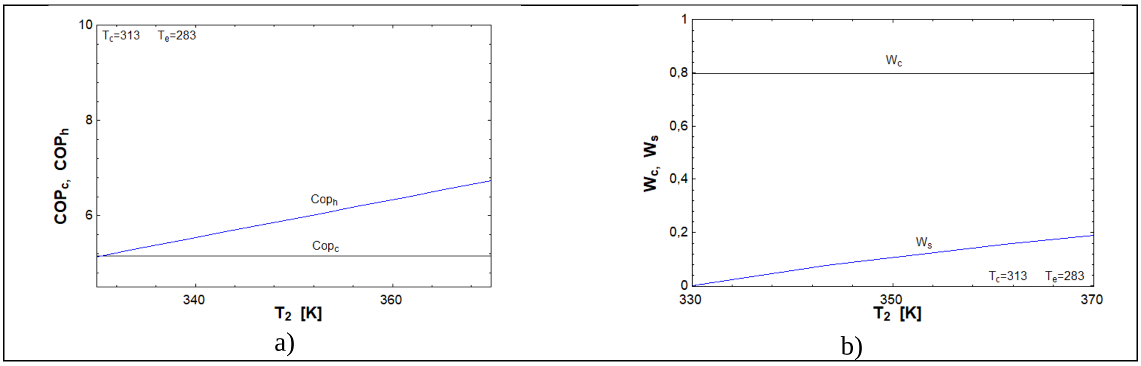

Figure 3. Relationship between solar exchanger outlet temperature and a) COP b) Compressor work

3.1.2 Temperature variation in the evaporator

The second analysis, Fig. 4 a) and b), varies the temperature in the evaporator, and evaluates the effects on COPs and Works:

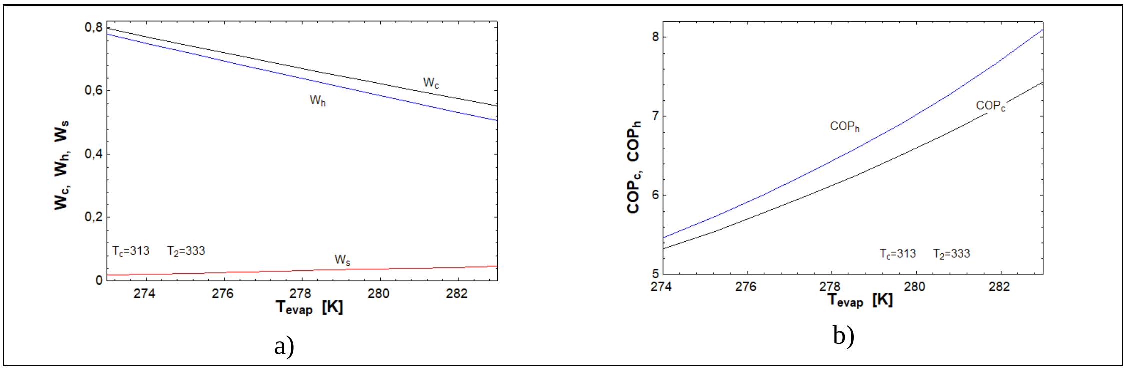

Figure 4. Relationship between Evaporator Temperature and a) Compressor Work b) COP.

As in the first analysis, the results had small divergence of values, but followed the same body and curve trend, increasing the COP and reducing the work with increasing temperature in the evaporator.

3.1.3 Temperature variation in the condenser

In the next analysis, Fig. 5 a) and b), the temperature in the condenser varies, and maintains the same evaluation of the comparisons and :

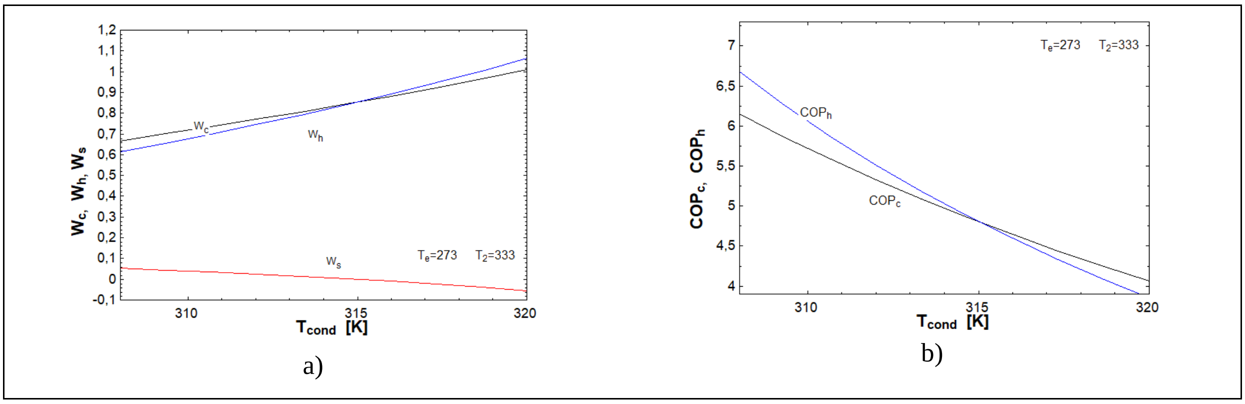

Figure 5. Relationship between Condenser Temperature and a) Compressor Work b) COP

Graphs maintained as expected, pointing to energy savings, and the analyzes remain satisfactory.

3.2 Compressible flow

This modeling brings a little more complexity, both in its equation and in its results. At first, when entering the input parameters adopted in the constant volume modeling, Tab. 1, the code created in the EES software was not able to converge to a result, and it was not possible to create comparative graphs between the two models. Although such a result may lead to the conclusion that the adopted modeling is not representative for the model, as it does not find satisfactory results, it should be noted that according to the comparison article (Kumar and Patel, 2020) the first tests with the real physical model were not satisfactory, and for the model to generate some significant savings, it was necessary to add a component, a solenoid valve, to restrict the output of the fluid from the solar exchanger, thus favoring the isovolumetric modeling, since by restricting this passage the volume inside the solar exchanger remains constant as long as the valve is closed.
This solution adopted is a technological and adaptive solution to the problem, however, in the reviews of the available works, no other situation was found in which it had been adopted. This does not detract from the reliability of the article, it was just the solution adopted by its authors to be able to model the problem, but this leads us to the compressible flow model, which already showed that in the free flow condition, the conditions adopted as parameters of input would not really bring satisfactory results in terms of energy savings.
Modeling by compressible flow is directly linked to both the flow velocity and the thermodynamic state at the entrance of the solar exchanger, according to the literature (Fox et al., 2006), so that there is a pressure increase during a compressible flow with addition of heat, the inlet velocity must be greater than Mach 1 in the inlet conditions, but with the value of the tubular section area and the thermal load (cooling) adopted, the inlet speed in the heat exchanger was not sufficient for Mach 1.

Table 2. Initial data for compressible flow

Evaporator Temperature	283 K
Condenser Temperature	312 K
Tubular section area (Solar Exchanger)	0,00023 m^2
Pressure in X	P_2

With this, an analysis was made of Figure 6, of the Mach analysis at the entrance of the solar exchanger as a function of the thermal load of the evaporator, having as input data, the values described in Tab. 2.

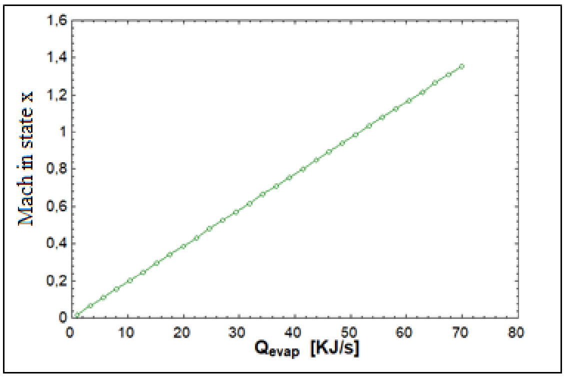

Figure 6. Mach x Thermal Load

To evaluate the area of the tubular section in the solar exchanger, and to have parameters to define an area value from which the real model would start to work, it was analyzed, Fig. 7, Mach behavior as a function of tubular section area for this, the same parameters of Table 1 were used, and the same thermal load (cooling) used in the comparison article (Kumar and Patel, 2020) Qevap = 4.1 kW.
Figure 7 then indicates that Mach is approximately 1, when the area is equal to 0.000016 m^2, showing that sufficient speeds for the model to work will be reached with values of areas smaller than this value, we will not have tools here capable of evaluating whether these values are physically possible as this is not part of the scope of work.
It is possible to bring up several analyses, varying all parameters and thus find a large range where the compressible flow model is able to work, but we will stick only to maintaining the original conditions of the analysis, Tab. 2, and leave the pressure at point x free to vary, working now with thermal load values of 60 kW, and with that we will evaluate the work done by the compressor and the COP of the system, and compare with the conditions between the hybrid system and the conventional system.

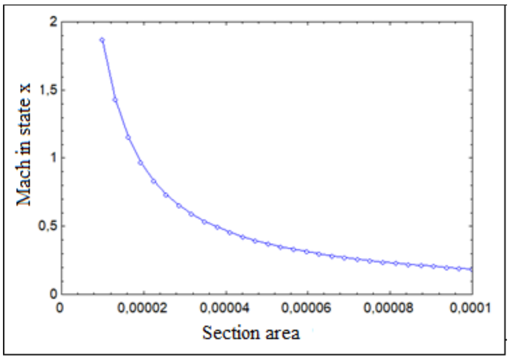

Figure 7. Mach x Section Area

The heat added by the sun in the evacuated tube exchanger is represented here by Qadc, in this modeling it is defined by Eq. 15, and it was the varied parameter, Fig. 8 a) and b), for the evaluation of the parameters of interest, COP and Work in the compressor. The other parameters are defined in Tab. 2, with the exception of the pressure at x.

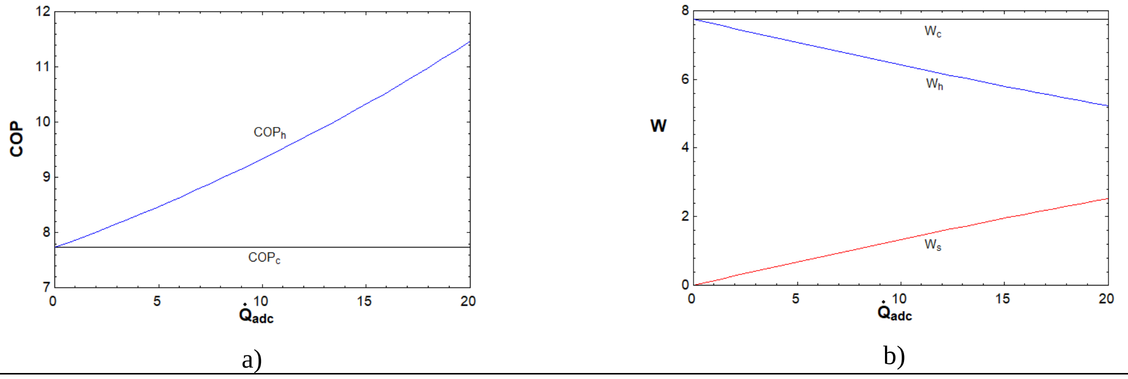

Figure 8. Added Heat in the Solar Exchanger and a) COP b) Work on the Compressor

The analysis shows trends similar to the analyzes made under the hypothesis of constant volume, the COP of the hybrid system, COPh , is shown to increase in relation to the COP of the conventional system, on the other hand, and as expected, the work done by the compressor decreases, Wh, while the work of the conventional system remains constant and the work saved increases. This analysis shows how the system responded as expected when heat was added to the evacuated solar exchanger.

Assuming the ideal gas is essential for modeling, however, an analysis of such a hypothesis must be carried out. For that, the heat inserted in the solar exchanger modeled with the constant Cp, Eq. 15, was compared to the heat calculated by the EES software, which defined it using the enthalpies of states x and 2 as a parameter, through equation 13. These two quantities were related through Eq. 8, (Qn) thus bringing the percentage difference between the two, the analysis is in Fig. 14. For each heat unit inserted into the solar exchanger, the compressibility factor was also evaluated, as it indicates how close a gas is to behaving as ideal.

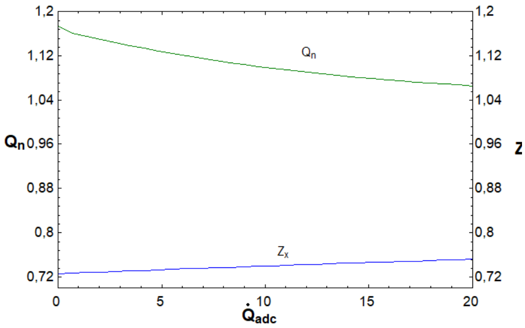

Figure 9. Evaluation of the ideal gas hypothesis

The values of Qn show that the difference between the theoretical heat and the real heat does not reach 20%, and falls when the compressibility factor increases, this is between 0.7 and 0.8 and needs a more detailed analysis to measure how satisfactory it is. Figure 10 shows a simulation under the same conditions, but with a fixed amount of heat absorbed in the solar exchanger equal to 10 kW. The graph brings polygonal lines, as the software only marks the points on the dome and interconnects them with straight lines, but it is clear that the tendency of the lines is to approach the expected result, as proposed in Fig. 2 b).

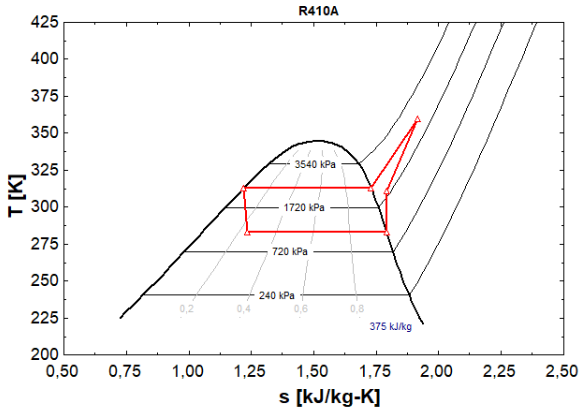

Figure 10. Real TxS diagram

3.3 Comparison of different working fluids

A comparison between the working fluids was also raised, Fig. 11 a) and b), evaluated the behavior of the hybrid COP for the system, respecting their respective modeling.

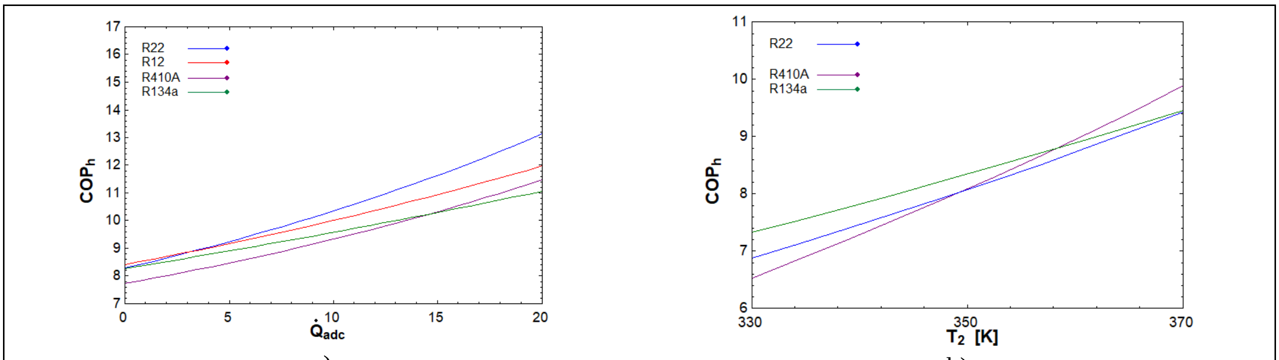

Figure 11.2 Evaluation of Fluids for a) compressible flow b) Constant volume

In the compressible flow modeling, the heat added to the system was varied by the hybrid sole exchanger, and the refrigerant fluid R22 showed a better performance than R410A, for the compressible flow model, Fig. 11 a).
For the constant volume modeling, the outlet temperature of the heat exchanger was varied, and the three fluids evaluated had an equivalent performance, it is worth mentioning that the evaluation conditions were not exactly the same, since the modeling in Fig. 11 a) used a heat load of 60 kW and the one in Fig. 11 b) used 4.1 kW, as in their respective models.

4. CONCLUSION

The hybrid refrigeration system once again showed promise in terms of energy savings, since the analysis in general led to a substantial reduction in electrical energy consumption by the conventional compressor as heat is added to the system through the evacuated tube solar thermal exchanger. The isovolumetric analysis proposed here when compared to the analysis performed by Kumar and Patel (2020) presented equivalent, but not identical, results, since there were no conditions for such results to be the same, as the approach here made use of the properties in the library of the EES software for each thermodynamic state, while the cited work made use of the equation that models the refrigerant as an ideal gas. Even so, the results are considered promising, as the curves followed the same trend, showing the equivalence between the models.

The compressible fluid analysis did not prove to be sufficient when related to the models available here for comparison, but this is justified by the fact that the inlet conditions, such as flow area and thermal load, directly influence this type of modeling, since for if having the expected result the flow velocity is a crucial point of the analysis. Even not being representative for the proposed comparison, this analysis showed promising trends for a specific work interval. In order to validate this model, it is necessary to carry out a work that compares it with a real model or mathematical models that adopt the appropriate initial conditions. The comparative curves of the refrigerant fluids brought another validation parameter to the models, since the curves also followed the same trend, and gave support to a possible choice of fluid for a real model. In general, the results corresponded, within their restrictions, to the model expected for this work.

Nomenclature				**Subscribed**			
P	Pressure	ρ	Especific mass	1	state	real	Real
COP	performance coefficient	A	Area	2	state	o	Stagnation
W	Compressor work	Cp	specific heat	3	state	s	saved
s	Entropy	R	Gas Constant	4	state	evap	Evaporator
m	Pasta			x	state	cond	Condenser
h	enthalpy			c	Convencional		
Q	Heat			h	Hybrid		
V	Speed			adc	Added		

5. ACKNOWLEDGMENTS

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