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Energy and Exergetic Analysis of a Regenerative Rankine Cycle with Feed water Heaters

Valencia Ochoa, G.[1]*, Garcia Sierra, Y.[2], Diaz Manotas, D.[2]

[1]Mechanical Engineering Universidad del Atlántico/ Barranquilla, Colombia
[2]Universidad del Atlántico/ Barranquilla, Colombia

Abstract : This paper presents the energy and exergy analysis of a 500 MW steam power plant operating under a regenerative Rankine cycle with one open and three closed heater. Some case studies were developed with the help of Aspen HYSYS® 7.2, in order to study the behavior of the energy efficiency and exergy cycle under changes of the high pressure (32MPa- 40MPa) and high temperature (600°C - 800°C), medium pressure (2MPa-8Mpa), Medium temperature (300°C-600 ° C) and low pressure (5kPa-100kPa), resulting in a nominal operating conditions a thermal efficiency of 46.2%, a second law efficiency of 79% and a 79.05% of irreversibility in the boiler and superheater regarding the exergy supplied.
Keywords : exergy, energy, irreversibilities; Regenerative Rankine cycle, efficiency.

Introduction

The generation of electrical energy from thermal plants has performed an important role in the development of society, allowing a significant increase in productivity, which is reflected in better living conditions of mankind. Today, the energy produced from fossil fuels in thermal power plants is about 80%, while 20% of electricity is obtained from different sources such as hydric, nuclear, wind, solar, among others [1].

There are different energy generation alternatives, all of them are currently under study and improvement, in order to reduce the environmental impact at the lowest possible cost, which is quantified with the efficiency of these processes. One of the most widely used alternatives is the generation of energy through thermal machines which consists of one device and two sources (hot and cold) for its operation. These machines operate under a thermodynamic power cycle whose main objective is to transform the transmitted heat into mechanical work, however this is not entirely possible since in this process heat is always dissipated and not everything can be transformed into work [2].

Different combinations and options have been proposed to thermodynamically optimize these processes. It is from there that one of the most commonly used power cycles called the Rankine cycle emerges [2], that allows to generate power with little environmental impact, short construction and implementation times, compared to conventional thermoelectric power plants [3]. A careful study of this cycle reveals that heat is transferred from a working fluid to a relatively low temperature, this reduces the average temperature at which heat is added and consequently the cycle efficiency [4]. To remedy this deficiency, it is sought to increase the temperature of the liquid leaving the pump, feed water, before it enters to the boiler. A practical regeneration process in the plants not only improves the efficiency of the cycle, but also provides a convenient means of eliminating the air being filtered to the condenser [5], preventing corrosion in the boiler. It also helps to control the volumetric flow of the steam in the turbine final stage, associated with the large specific volumes at low pressures. Additionally, with the extraction or purging of the steam from the turbine in the steam power stations at various points a regeneration process is achieved [6]. This steam, which could produce more work if it were further expanded in the turbine, is instead used to heat the feed water.

The different types of energy also present different qualities, this is because of the possibility of producing work or transforming one type of energy into another [7]. For example, the quality of the heat depends on its temperature; at higher temperature, a heat source can transfer its energy with more possibilities than at lower temperature. In general, it is accepted as a measure of the quality of energy its ability to produce work. The problem with this definition is to choose the appropriate reference level. It must be taken into account that for a thermal machine to perform work, it must take heat from a source at high temperature and give some of that heat to a sink at low temperature. As it is usual for thermal machines to work with the environment surrounding them as a cold focus, the reference level is usually taken at room temperature. Therefore, when calculating the exergy [8] it is necessary to specify the environment in which the thermal machine works. The ability of an energy medium to perform work expresses its potential to transform into other types of energy, and therefore the exergy can be applied to the study of technological processes as well as for power plants, thermodynamic cycles, machines, etc. Unlike energy, there is no a conservation law for exergy. Any irreversible phenomenon causes a loss of exergy, which leads to a reduction in the potential of the useful effects of energy, or on the contrary to an increase in the energy consumption provided by the hot focus (to achieve an equal generation of work).

The exergetic and energetic analysis together, can give a complete representation of the characteristics of the system. Such comprehensive analysis will be the most convenient approach to performance evaluation and determination of cycle improvement potentials [9]. In the literature are found works related to energetic and exergetic analysis in thermal power plants, among which is Reddy and Mohamed [10] who analyzed in a combined power cycle, the effect of the temperature of entry in a gas turbine and the pressure relationship on exergetic efficiency. Rosen [11] presented an energy and exergy analysis based on the comparison of coal and nuclear steam power plants. Datta et al. [12] presented an energy and exergy analysis of an integrated gas turbine cycle with a biomass gasifier for the generation of distributed energy. Woudstra et al. [13] determined the cogeneration process, steam generation levels to reduce heat losses in the heat recovery steam generator (HRSG) and the loss of exergy due to flue gas exhaust unity. Srinivas et al. [14] analyzed the combined cycle power plant with methane as fuel using the first and second law of thermodynamics.

The purpose of this study presented below is to carry out the energy and exergy analyzes for a regenerative Rankine cycle with simulated overheating in Aspen HYSYS® 7.2, developing a detail thermodynamic models based on the equations of mass balance, energy and exergy, that allowed to develop different cases study, which reveal the parametric behavior of the cycle.

Methodology

Thermal Power Plant data Operation

A Thermal Generation Plant generates 500MW of electrical power, where the ambient temperature and pressure conditions are T_0 = 22 ° C and P_0 = 1 atm, the water enters the first stage of the turbine at 32 MPa and 600 ° C, and is expanded to 8 MPa. A portion of the flow is diverted to a closed heater and the remainder of the flow is reheated to 560 ° C before being sent to the second stage of the turbine. When the second stage of the turbine finalized, the pressure reaches 1 MPa, at which time a new extraction is carried out towards another closed heater, the excess current is sent to a third stage of the turbine where it expands up to 0.15 MPa, there makes a new extraction that is sent to an open heater, while the remaining water expands through the fourth stage of the turbine until it reaches 6 KPa. From each of the closed heaters comes out saturated liquid, which is passed through expansion valves and then sent to the open heater. At the outlet of this condenser there is saturated liquid, which is pumped into the boiler, and each stage of the turbine has an efficiency of 85% and the pumps are isentropic. The system schematic for the described cycle is simulated in Aspen HYSYS® 7.2 as shown in Fig. 1.

Energy and Exergy Analysis

The analysis of energy and exergy of the thermal cycle was developed from the mass balances, energy and exergy balances applied to each of the components of the cycle, which are shown in detail below, considering the thermodynamic states shown as follow in the process diagram on Figure 1.

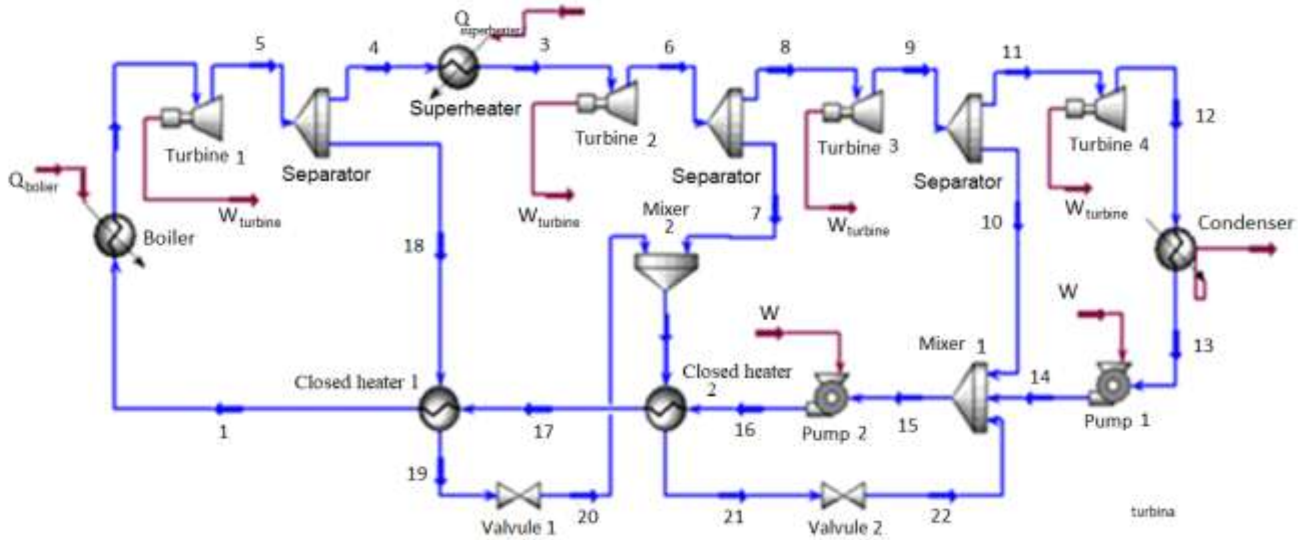

Figure 1.Regenerative Rankine Cycle Schemes with Power Heaters Feed Water in ASPEN Hysys 7.2

Mass and Energy Balances

The components of the cycle were modeled as stationary flow systems, for which energetic and matter considerations can be made as needed, starting from the following equation (1)

$$\dot{Q} - \dot{W} = \sum \dot{m}_e \left(h_e + \frac{V_e^2}{2} + gz_e \right) - \sum \dot{m}_i \left(h_i + \frac{V_i^2}{2} + gz_i \right), (1)$$

wherein general terms the variation of the kinetic and potential energies was not considered, so the expression of the energy balance finally obtained is

$$\dot{Q} - \dot{W} = \dot{m}_e (h_e - h_i), (2)$$

so the transfer of energy by heat or work on the different devices depends on their enthalpies of input and output; while the thermal efficiency of the Rankine cycle is

$$\eta_{ter} = \frac{W_{net}}{Q_{in}}. (3)$$

In the simulation was taking into account the deviations of the real cycle from the idealized, whereby the irreversibilities that take place within the turbine were assumed (since the pump is modeled with a η_s = 100%) assuming an isotropic flow under ideal conditions.

The existing deviations of the real and the isentropic turbines can be taken into account using their isentropic efficiency, defined as shown on equation (4) as follow

$$\boldsymbol{\eta_T} = \frac{\boldsymbol{W_r}}{\boldsymbol{W_s}} = \frac{\boldsymbol{h_i - h_{er}}}{\boldsymbol{h_i - h_{es}}}, (4)$$

where states (r) are the real states of output and input of the pump and turbine, respectively, whereas (s) are the corresponding states for the isentropic case.

The phenomenological equations for the different devices were based on the principle of energy conservation, resulting for the first closed heater the equation (5)

$$y'.h_{18} + h_{17} = y'.h_{19} + h_1, (5)$$

wherey' is the mass fraction passing through the line 18. The equation applied to the second closed heater is

$y''.h_7 + y'.h_{20} + (h_{16} - h_{17}) - (y'' + y')h_{21} = 0$,(6)

wherey'' is the mass fraction passing through line 7, while the balance on the mixer1 is shown as follow

$0 = y'''.h_{10} + (y' + y'').h_{22} + (1 - y' - y'' - y''').(h_{14} - h_{15})$,(7)

where $y'''$is the mass fraction passing through line 10.

The energy balance selecting all expansion stages as control volume is shown on equation (8) as follow

$\frac{\dot{W}_t}{\dot{m}} = (h_2 - h_5) + (1 - y').(h_3 - h_6) + (1 - y' - y'').(h_8 - h_9) + (1 - y' - y'' - y''').(h_{11} - h_{12})$,(8)

for a control volume in the pumps is

$\frac{\dot{W}_b}{\dot{m}} = (h_{16} - h_{15}) + (1 - y' - y'' - y''').(h_{14} - h_{13})$,(9)

andthe input heat supplied to the boiler and superheateris calculated as

$\frac{\dot{Q}}{\dot{m}_1} = (h_2 - h_1) + (1 - y').(h_3 - h_4)$,(10)

whilethe energy balance for the condenser heat exchanger is

$\dot{m}_{cw}(h_{cwsalida} - h_{cwentrada}) = \dot{m}_2\,(1 - y' - y'' - y''').(h_{12} - h_{13})$.(11)

The same procedure can be assumed to determine the calculation of the mass flows, taking into account the division that these carry during the cycle, the mass flow was calculated in point 2, considering that the total work is 500 kW, considering that$\dot{m}$ is the mass flow at the outlet of the boiler, and the powers of the turbine and pump are $\dot{W}_t$ y$\dot{W}_b$respectively, resulting the equation (12) as follow

$\dot{W}_{total} = \dot{m}_2.\left[\frac{\dot{W}_t}{\dot{m}} - \frac{\dot{W}_b}{\dot{m}}\right]$,(12)

where the total power generated by the turbine is the sum of the power of each stage calculated by the equations 13 to 16 as follow

$\boldsymbol{W_{t1}} = \dot{m}_2\,(h_2 - h_{5r})$,(13)
$\boldsymbol{W_{t2}} = (1 - y').\dot{m}_2.(h_3 - h_{6r})$,(14)
$\boldsymbol{W_{t3}} = (1 - y' - y'').\dot{m}_2.(h_8 - h_{9r})$,(15)
$\boldsymbol{W_{t4}} = (1 - y' - y'' - y'').\dot{m}_2.(h_{11} - h_{12r})$.(16)

On the other hand, the total powerrequired by the pumps is calculated bye the equations (17) and (18) as

$\boldsymbol{W_{b1}} = (1 - y' - y'' - y'').\dot{m}_2\,(h_{14} - h_{13})$,(17)
$\boldsymbol{W_{b2}} = \dot{m}_2\left[\frac{\dot{W}_b}{\dot{m}}\right]$.(18)

Thus it can be obtained all the variables and analyzes required of first law to arrive at clear conclusions on the variations in the thermal efficiency of the cycle.

Exergetic analysis and cycle irreversibilities

Exergy is a generic term for a group of concepts that define the maximum possible working potential of a system, a matter stream and/or heat interaction; the state of the (conceptual) environment that is used as state data. In an opened flow system, there are three types of energy transfer across the known control surface, work transfer, heat transfer, and energy associated with mass transfer or flow.The heat transfer exergy (ψ_Q) from the control surface to the temperature T, is determined from the maximum rate of conversion of thermal energy into total work W_{max}, which is given by the equation (19)

$$W_{max} = \psi_Q = Q\left(1 - \frac{T_0}{T}\right), \quad (19)$$

where the constant flow current exergy of matter is the sum of kinetic, potential and physical exergy. The exergy flow for continuous flow process of an open system is given by the equations (20)

$$\sum\left(1 - \frac{T_0}{T_k}\right) Q_k + \sum_{in} \dot{m}\psi_i = \psi_W + \sum_{out} \dot{m}\psi_o + T_0\left[\dot{S}_{gen}\right], \quad (20)$$

where ψ_i and ψ_0 are exergy associated with an output mass and input respectively, ψ_W is the useful work performed on o on the system, $I_{destroyed}$ is the irreversibility of the process, and h^0 is the enthalpy as the sum of enthalpy and T_0 must be calculated in absolute grades.The second law efficiency is defined as

$$n_{II} = \frac{outlet\ Exergy}{imput\ exergy}. \quad (21)$$

The exergy balance for a thermal system is given by the equation (22)

$$\psi_W = \sum_{k=1}^{n}\left(1 - \frac{T_0}{T}\right) Q_k + \sum_{k=1}^{r}[(\dot{m}\psi)_i - (\dot{m}\psi)_o]_k - T_0\,\dot{S}_{gen}. \quad (22)$$

Thus, according to the above, for the study of the second thermodynamic law it is necessary to introduce the calculations corresponding to the exergies present in the cycle, based on the calculations of enthalpies and entropies determined from our input data, it is important to see that also the irreversibilities or destruction of exergy to reach a relationship between this and optimize clear and reliable energy costs to be able to operate the company. The exergy was calculated under the following ambient conditions P_0 = 101.3 kPa and T_0 = 22°C, resulting the exergyfor the boiler as shown on equations (23) as follow

$$\dot{m}_2\,(\psi_2 - \psi_1) = \dot{m}_2\,[(h_2 - h_1) - T_0.(s_2 - s_1)], \quad (23)$$

for the superheater

$$\dot{m}_3\,(\psi_3 - \psi_4) = \dot{m}_3\,[(h_3 - h_4) - T_0.(s_3 - s_4)], \quad (24)$$

while the Irreversibilities for the Condenser is

$$\dot{\mathbf{I}}_{\boldsymbol{cond}} = T_0[\,\dot{m}_{11}.(s_{13} - s_{12}) + \dot{m}_{cw}.(s_{ent} - s_{sal})], \quad (25)$$

and the Heater Closed I and II are calculated as shown on equations (26) and (27) as follow

$$\dot{\mathbf{I}}_{\boldsymbol{cc1}} = \dot{m}_2\,.T_0.\left(\frac{\dot{\sigma}}{\dot{m}_1}\right) = \dot{m}_2\,.T_0[y'.(s_{19} - s_{18}) + (s_1 - s_{17})], \quad (26)$$

$$\dot{\mathbf{I}}_{\boldsymbol{cc2}} = \dot{m}_2\,.T_0[s_{17} - s_{16} + (y' + y'').s_{21} - y'.s_{20} - y''.s_7]. \quad (27)$$

The irreversibilities in the Mixer, purge valve I and II are presented in equations (28), (29) and (30) respectively as follow

$$\dot{\mathbf{I}}_{\boldsymbol{ofw}} = \dot{m}_2\,.T_0[s_{15} - y''''.s_{10} - (y' + y'').s_{22} - (1 - y' - y'' - y''')s_{14}], \quad (28)$$

$$\dot{\mathbf{I}}_{\boldsymbol{purge1}} = \dot{m}_2\,.T_0[(s_{20} - s_{19}).y'], \quad (29)$$

$$\dot{\mathbf{I}}_{\boldsymbol{purge2}} = \dot{m}_2\,.T_0[(s_{22} - s_{21}).(y' + y'')]. \quad (30)$$

Finally, the irreversibility estimation for first, second, third y fourth stage of the Turbine are conducted respectively by mean of the equations (31) to (34) as follow

$$\dot{\mathbf{I}}_{\boldsymbol{t1}} = \dot{m}_2\,.T_0.(s_{5r} - s_2), \quad (31)$$

$$\dot{\mathbf{I}}_{\boldsymbol{t2}} = \dot{m}_2\,.T_0.(s_{6r} - s_3), \quad (32)$$

$$\dot{\mathbf{I}}_{\boldsymbol{t3}} = \dot{m}_2\,.T_0.(1 - y' - y'').(s_{9r} - s_8), \quad (33)$$

$$\dot{\mathbf{I}}_{\boldsymbol{t4}} = \dot{m}_2\,.T_0.(1 - y' - y'' - y''').(s_{12r} - s_{11}). \quad (34)$$

The exergetic efficiency is defined as the quotient between the total work obtained and the total exergy given to the cycle ($\psi_{boiler} + \psi_{superheater}$), this is:

$$\boldsymbol{\varphi} = \frac{W_{net}}{\psi_{net}}. \quad (35)$$

Results and Discussion

The actual operating data of the components, such as temperature and pressure of the feed water in the turbine stages, portion of the flow entering the closed heater and the boiler, and then being pumped under certain specified conditions of pressure and temperature; Mass flow at the inlet and outlet of the turbines were collected for analysis and thus calculate the enthalpies and exergies at different state points of the cycle and their influence on the behavior of the energetic and exergetic efficiencies of the cycle, which was the fundamental analysis of the present study. Figure 2 presents the powers delivered in each of the stages of the turbines and the power consumed by each pump.

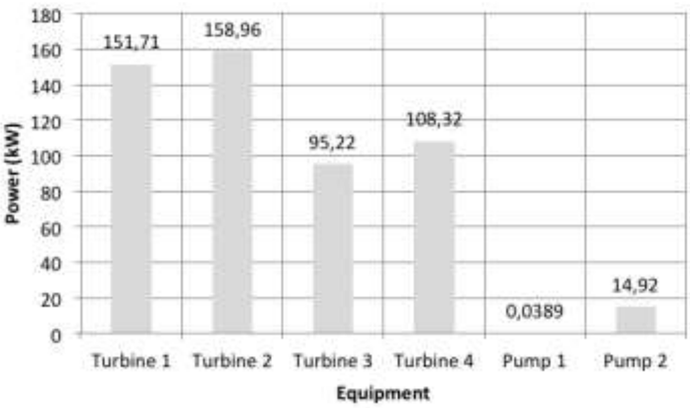

Figure 2.Powers Main Cycle Equipment

Increasing the inlet temperature and pressure during the process of adding heat to the boiler immediately increases the thermal efficiency of the cycle, that is, that an increase in temperature generates greater advantage of expansion in the turbine, since the steam has higher internal energy, as shown in Figure 3.

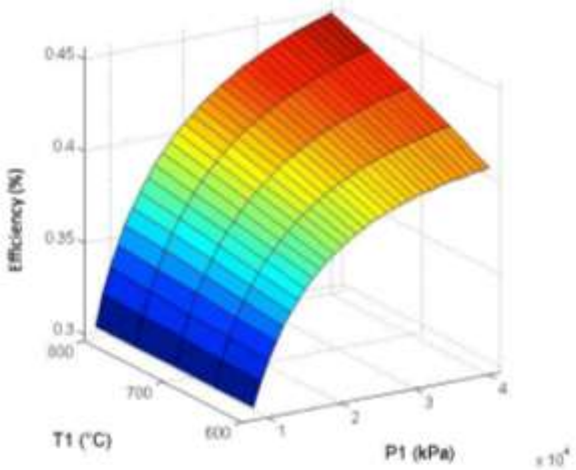

Figure 3.Behavior of the Thermal Efficiency of the Cycle as a function of the Pressure and High Temperature.

In addition, another way of increasing the average temperature during the heat addition process is to increase the operating pressure of the boiler, which automatically raises the temperature at which the boiling occurs. This in turn raises the average temperature at which heat is transferred to the steam and thereby increases the thermal efficiency of the cycle.

The variations in low temperature and pressure (which is the pressure at which the condenser will operate during the cycle) taking into account that the water exists as moist vapor in the condenser at the saturation temperature corresponding to said low pressure. Consequently the reduction of the operating pressure of the condenser automatically reduces the temperature of the vapor, and therefore the temperature at which the heat is transferred. The reason why the behavior of the low pressure tends to be more linear with the thermal efficiency thanwith to the high pressure, as seen in Figure 4, which is due to the fact that it has a direct effect on the cycle by the operation of the condenser, so this pressure is kept constant up to the pump, which implies that at a lower low pressure, less will be the work required by the pump according to the pressure; which automatically generates an increase in the total work required for an improvement in the thermal efficiency of the cycle.

As to the variation of the total work and the heat supplied as a function of the average pressure, it can be concluded that under given pressure conditions, an increase in the work generated reduces the need to supply more heat in the boiler. In other words, the easier it is to heat the steam required to operate the turbine, the less heat must be supplied to the boiler and therefore the thermal efficiency will be improved.

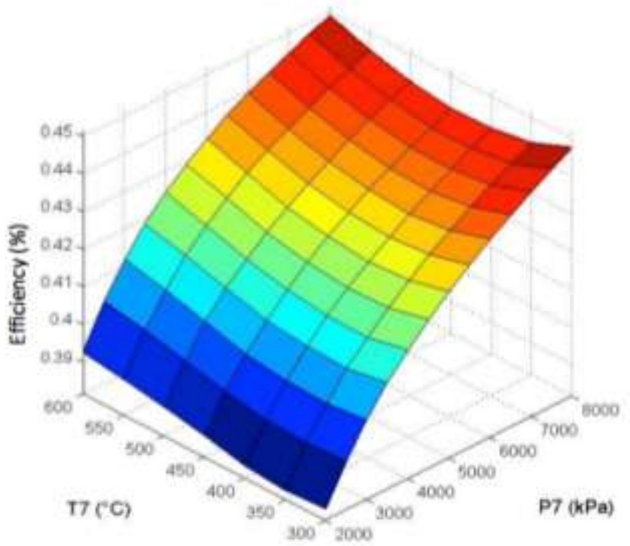

Figure 4.Behavior of the Thermal Efficiency of the Cycle as a function of the Average Pressure and Temperature.

In practice this process is almost unreal because it will always need to supply more heat from the source than the one required to generate mechanical work and a process that demands little heat supplied to the water is almost difficult to complete, since large amounts of work at the expense of little heat to the boiler is not possible for our substance of work because at high pressures will be difficult to achieve boiling water.

Thermal efficiency improves by 4 to 5% with the implementation of a reheating step in the Rankine cycle, which goes hand in hand with the intermediate pressure as seen in Figure 4. However, growth can be seen in the graph at high pressures, this is because at a temperature equal to the input and a higher pressure the quality of the steam is reduced and it generates humidity, affecting the conditions of the low turbine.Likewise, Figure 5 shows that as the temperature increases at the inlet of the first turbine stage, the exergy supplied increases as a function of said temperature in a linear manner, while the supplied exergy decreases respect to the variation of high pressure that shows a slight inclination in the curve, however in certain intervals there is some balance between the exergy supplied and the variation of high pressure and temperature.

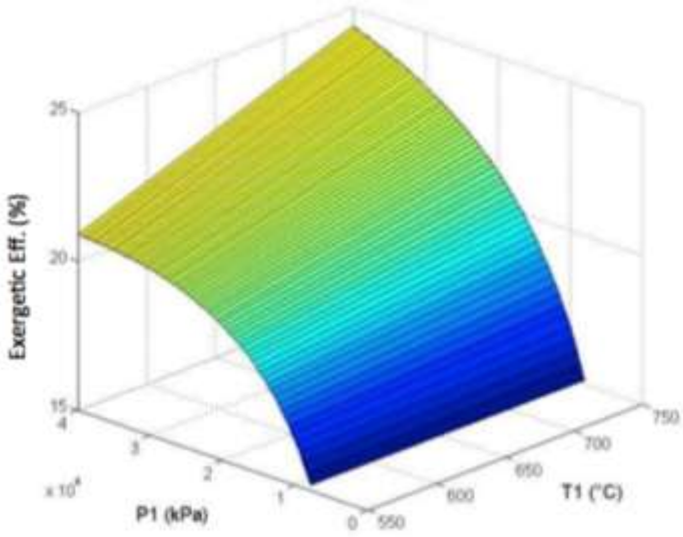

Figure 5.Behavior of the Exergetic Efficiency of the Cycle as a function of the Pressure and Average Temperature.

The exergetic analysis was developed to all components of the cycle, as shown in Figure 6.

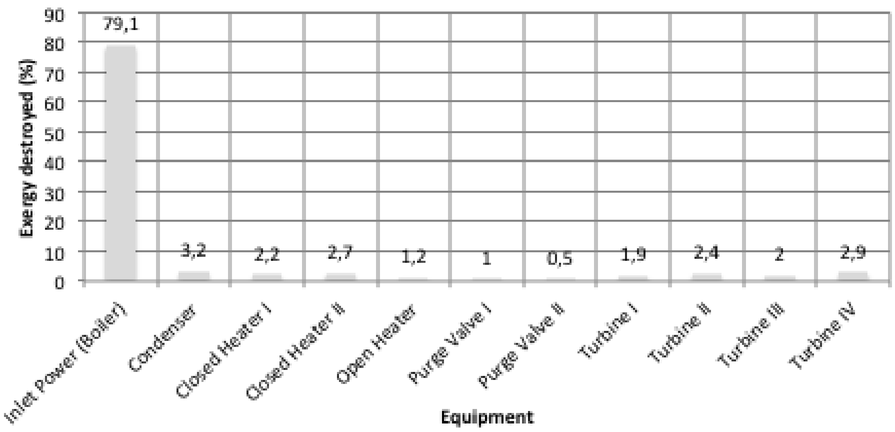

Figure 6. Percentage distribution of the exergy destroyed as a function of the exergy supplied to the cycle by equipment.

By varying the pressure and the average temperature, the supplied exergy that is plotted against both, shows a decrease. However, the behavior of average pressure vs. supply exergy presents a significant slope, while the curve generated between the average temperature vs exergy supply is completely linear.

Conclusion

This research article developed a deep energy and exergetic analysis of a Regenerative Rankine Cycle with open and closed heaters used in thermal power generation plants. It was implemented in Aspen HYSYS® 7.2, where it was possible to carry out some study cases that show the behavior of the thermal and exergetic efficiencies of the cycle in the presence of variation of operational parameters such as high pressure and temperature, providing operational zones of maximum efficiency and better use of energy resources.

Among the most important results it is indicated that approximately fifty percent (50%) of the total thermal energy generated in the combustion chamber is available to do useful work, thus having its effect on the working output of the turbine. This is due to the inherent irreversibility in the process of heat supply in the boiler and superheater.As a recommendation for future work of the present work, it is proposed to model the combustion chamber and perform a thermo economic study of the cycle.

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