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# Numerical Study of the Thermal and Exergetic Efficiency of a Brayton Power Cycle with Regeneration

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### Abstract

As the years go by, the growing interest of nations in energy generation becomes more evident. For this reason, this paper presents a study in which an exergetic analysis and the study of the variation of thermal efficiency for a Brayton cycle with regeneration was carried out, varying parameters such as the compression ratio. A spreadsheet was used to study a gas power cycle, an ideal Brayton regeneration cycle with a turbine, a compressor, a regenerator and a combustion chamber which delivers the heat input to the system, finding that if you want to improve the thermal efficiency of a gas power engine using an ideal brayton regeneration cycle you should consider or take into account what has been proposed and demonstrated in this research as a way to achieve this objective, Also thanks to the exergetic analysis that was carried out on the cycle in question, this research can be considered to take action on the components that most destroy useful energy potential.

**Keywords**: Numerical study, Thermal efficiency, Exergetic efficiency, Brayton Power Cycle, Regeneration

## 1. Introduction

The generation of energy is of vital importance for the development of nations. For this reason, several ways have been investigated for the generation of this energy, among which are the energy generated by wind power [1], wave energy [2], generation of energy by means of solar sources [3] and energy generated by means of power cycles [4], [5]. These power cycles are widely used in organizations that have their own energy generation and energy in the form of heat remaining from the generation process is used in other applications in the organization, and this is another great benefit of the self-generation of energy.
The Rankine cycle in which steam is used as the motive fluid for obtaining energy has shown advances such as the use of solar concentrators to generate the steam used in the[6] cycle. In addition, organic fluids have been used for this cycle to seek economic optimization [7]. On the other hand, the Brayton cycle, unlike the Rankine, the driving fluid used is combustion gases. For this the studies have sought to increase the efficiency of the cycle, for this purpose a model of the closed reversible binary Brayton cycle was constructed and the results indicate that the maximum output power is reached when the optimum pressure ratios of the coverage and background cycles are chosen, and the power output increases first and then decreases with the increase in the pressure ratio of the coverage cycle. The thermal efficiency of the cycle increases gradually with the increase in the pressure ratio of the coverage cycle [8]. In the same way, the optimization of the exergetic performance has been sought, for this purpose a new configuration of combined intercooling regenerative Brayton and inverse Brayton cycles with regeneration before the inverse cycle was proposed. By using exergy analysis, the performance of the new configuration is investigated and optimized. After analyzing the effects of the main design parameters on combined cycle performance, it is found that cooling and regenerative progress are beneficial for increasing exergy efficiency when it varies in a certain range. It is also found that when the total pressure is relatively high, the compressor workload is significantly reduced by the intermediate cooling process; when the total pressure is small, the regenerative process can save more fuel [9]. Similarly, a parametric study is conducted to investigate the effects of some decision variables on the effectiveness of the first and second law and the total unit cost of the product of the Brayton supercritical carbon dioxide recompression / organic flash cycle and the Brayton supercritical carbon dioxide recompression cycle [10].
Similarly, a parametric study is conducted to investigate the effects of some decision variables on the effectiveness of the first and second law and the total unit cost of the product of the Brayton supercritical carbon dioxide recompression / organic flash cycle and the Brayton supercritical carbon dioxide recompression cycle.

## 2. Methodology

### 2.1. Description of the process

A spreadsheet was used to study a gas power cycle, an ideal Brayton regeneration cycle with a turbine, a compressor, a regenerator and a combustion chamber which delivers the input heat to the system; the program has the ability to perform case studies from independent and dependent variables, the cycle can be seen in Figure 1.

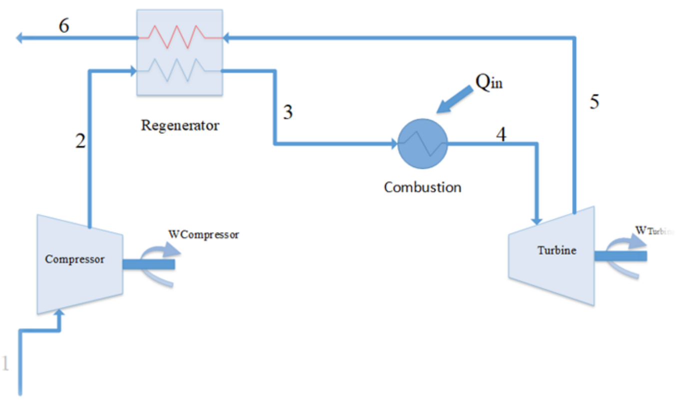


Figure 1. Process Diagram

The Brayton cycle with regeneration that operates with air was chosen as initial data the inlet pressure and temperature to the compressor (P= 100 kPa, T=300K) respectively, and the compressor has a pressure ratio from 2 to 20, the turbine inlet temperature is maintained at T= 1400K, at the compressor inlet there is a volumetric flow AV= 5 m3/s, the regenerator has an adiabatic efficiency of 80%, the turbine and compressor varying from 70%, 80%, 90% and 100% this can be seen organized in Table 1.

Table 1. Operational condition of the thermal cycle.

| **Press. 1** | **Tem. 1** | **Rp** | **Tem. 4** | $\eta_{regenerator}$ | $\eta_{com}$ / $\eta_{turb}$ |
|---|---|---|---|---|---|
| **100 kpa** | 300K | 2-20 | 1400K | 80% | 70-100% |

### 2.2. Fundamental equations

A set of equations were used to determine the thermal efficiency of the cycle and the destruction of exergy in each component of the process, each of which was studied in depth in the literature.

First of all, it is necessary to calculate the mass flow that can be determined by equation (1) because at the compressor inlet there is a volumetric flow rate of 5 $m^3/s$ and the flow rate can be calculated as follows

$$\dot{m} = \frac{(AV)*P}{R*T}\,. \tag{1}$$

The thermal efficiency of the cycle is calculated as shown in equation 2

$$\eta = \frac{\dot{W}n}{\dot{Q}in} \quad , \tag{2}$$

And you have to, in order to calculate $\dot{W}n$ and $\dot{Q}e$ as thermal efficiency variables related in equation (3)

$$\dot{W}n = \dot{W}t - \dot{W}c, \tag{3}$$

This in turn depends on the work carried out by the compressor and the turbine, which were calculated as shown in equations (4) and (5) respectively.

$$\dot{W}c = \dot{m} * (h2 - h1) \tag{4}$$

$$\dot{W}t = \dot{m} * (h5 - h4) \tag{5}$$

In order to determine the input heat ($\dot{Q}e$) equation (6) was used

$$\dot{Qint} = \dot{m} * (h4 - h3)\ . \tag{6}$$

Once the values corresponding to the work produced and consumed by the components of the cycle are obtained, the adiabatic efficiencies of the devices are calculated using equation (7).

$$\eta_c = \frac{\dot{W}iso}{\dot{W}real}; \tag{7}$$

where $\dot{W}iso = \dot{m} * (h2\,iso - h1)$, where $h_{2iso}$ s the value of enthalpy corresponding to the thermodynamic state 2 assuming that the entropy value of state 1 is equal to that of state 2, equation (8) is used here. where $\dot{W}iso = \dot{m} * (h5\,iso - h4)$,

$$\eta_t = \frac{\dot{W}real}{\dot{W}iso}; \tag{8}$$

In addition, an exergetic study was carried out to study the destruction of exergy by component, using equation (9) corresponding to a general exergy balance for systems with stationary flow.

$$\sum\left(1-\frac{T_0}{T_k}\right)\dot{Q}_k-\dot{W}+\sum_{in}\dot{m}\psi-\sum_{out}\dot{m}\psi-\dot{X}_{destroyed}=0\,, \tag{9}$$

This equation was used for each component studied to determine the exergy destroyed by component when varying the compression ratio at which the compressor worked.

## 3. Results and Discussion

The aim was to study the behaviour of the thermal efficiency of the cycle against the pressure ratio of the compressor, varying from 2 to 20 the pressure ratio. In addition, the adiabatic efficiency of the compressor was varied, generating different curves corresponding to the isentropic efficiency of the compressor and the turbine, ranging from 70% to 100%, as shown in Figure 2a, where it can be observed the behaviour of the thermal efficiency of the cycle at different compression ratios with different curves corresponding to adiabatic efficiencies of compressor and turbine. From these result it can be concluded that the compression ratio where the thermal efficiency is maximum in each of the adiabatic efficiency curves is in the order of $P_2/P_1=4$, and after $P_2/P_1=6$ the thermal efficiency of the cycle begins to decrease due to the heat lost on the turbine and the irreversibility presented. On the other hand, in percentage terms it can be identified the growth on the thermal efficiency as function of the compression ratio from 2 to 4 and the decrease from 6 to 20 of 20.12% and 31.12% respectively. Finally, as was increased the adiabatic efficiency of the compressor and turbine, the thermal efficiency of the cycle increases maintaining the same behaviour as when the compression ratio goes from 2 to 4 is maximum and then decreases.

In the second case study results, the change in the compressor inlet temperature affected the thermal efficiency of the process, and the cycle operating at different ambient temperature conditions ranging from 25 to 40°C where studied, changing the compression ratio from of 2 to 20 as shown in Figure 2b, where the behaviour of the thermal efficiency of the cycle due to the change in the compression ratio with different curves corresponding to compressor inlet temperatures from 25 to 40°C, assuming that the same thermodynamic cycle is operating in different environmental conditions. In addition, the same behaviour was presented as the previous results with adiabatic efficiencies, highlighting that it has steeper slopes, such as the one was observed from a compression ratio of 2 to 6, with a growth of 23.94%, but already from 6 to 20 a decrease of 17.86%. In addition, the lower inlet temperature to the compressor of 25°C produce the maximum thermal efficiency of 0.4632%, and the optimum compression ratio of 6. On the other hand, the lower thermal efficiency of the cycle of 0.4461% was presented at 40°C at the same compression ratio, results that confirms the statement, the lower inlet temperature to the compressor, produce a higher thermal efficiency of the cycle.

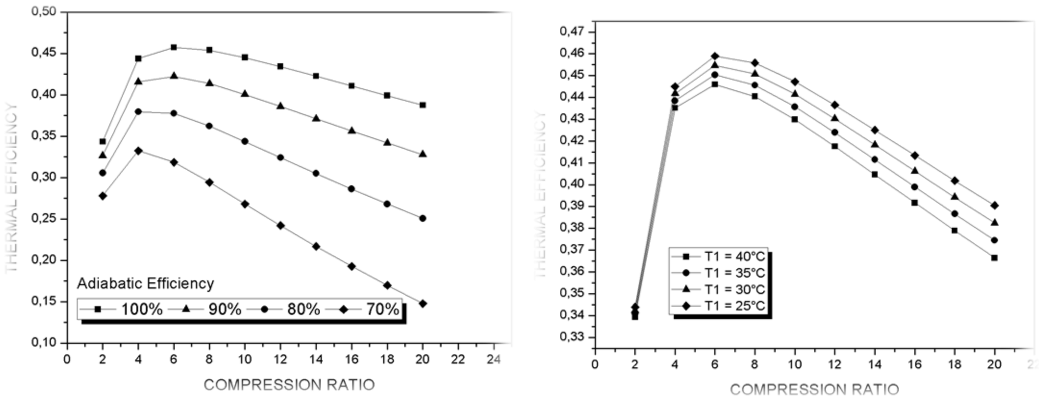


Figure 2. Results of Thermal efficiency versus compression ratio changing, a) the adiabatic efficiency, b) the inlet compressor temperature

Also, the exergetic analysis of the compressor was studied against the compression ratio at different adiabatic efficiencies, being the adiabatic efficiency an independent variable varied from 70% to 100% with a step of 10, as shown in Figure 3a, where the exergy destruction of the compressor versus the compression ratio, from 2 to 20, and the adiabatic efficiency is presented, resulting that when the compression ratio increases, the exergy destroyed by the compressor also increase, only in the adiabatic efficiency curve of 70% the exergy destroyed has an increase from 2 to 20 of 69.32%, and when the adiabatic efficiency of the compressor is increasing, the exergy destroyed of the component is less, as expected, also less inclined is the curve of the exergy destroyed versus the compression ratio, due to increasing the compressor efficiency, the compressor destroys less exergy and can operate with a higher compression ratios. In the same way, for the destroyed exergy of the turbine, it was also carried out a parametric study changing the compression ratio from 2 to 20, and the adiabatic efficiency of the same from 70% to 100%, thus providing 4 curves to allow a better study of the effect of this efficiency in the destruction of exergy due to this component as shown in Figure 3b.

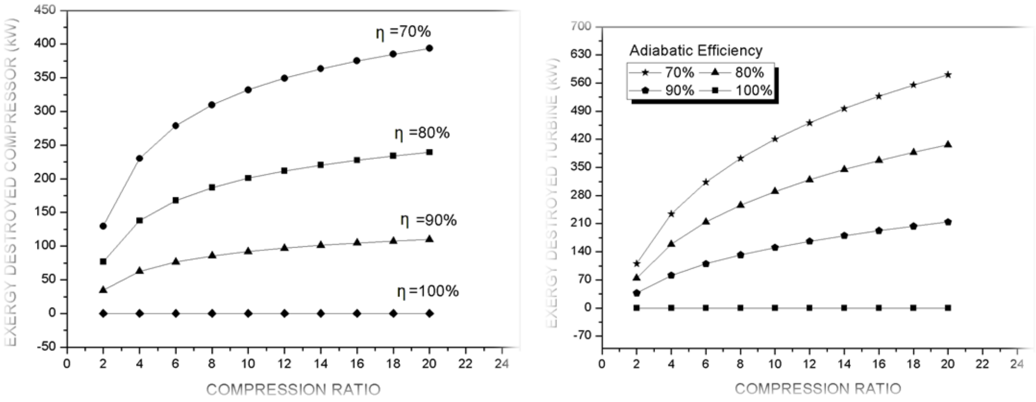


Figure 3. Exergy destroyed, a) compressor, b) Turbine

The results obtained presents the same behaviour of the destroyed exergy versus the compression ratio of the compressor but, to different degrees, as the compression ratio increases, the destroyed exergy of the turbine also increases, and becomes greater if the adiabatic efficiency of the turbine decreases; also, when the adiabatic efficiency of the turbine increases, the resistance of the high compression ratio increases, as shown in Figure 4a. Finally, the analysis of the destroyed exergy of the regenerator was plotted versus the compression ratio from 2 to 20, and also versus the adiabatic efficiency of the compressor and turbine as shown in Figure 4b. In this case a decreasing curve was observed from the destroyed exergy versus the compression ratio at different adiabatic efficiencies (which in this case does not have a significant contribution to the exergy destruction), i.e. as the compressor compression ratio increases the regenerator destroys less exergy. From a compression ratio from 2 to 10, the decrease in exergy destroyed due to the regenerator to a constant turbine and compressor adiabatic efficiency of 96% was observed to a greater extent, then continues to decrease, but to a lesser extent, and as the adiabatic efficiency of the turbine and compressor increases, this curve runs upward. In addition, Figure 4a shows this detail more clearly; that is, when the efficiency of the compressor and turbine is at its maximum, the regenerator destroys more exergy as the compression ratio increases.

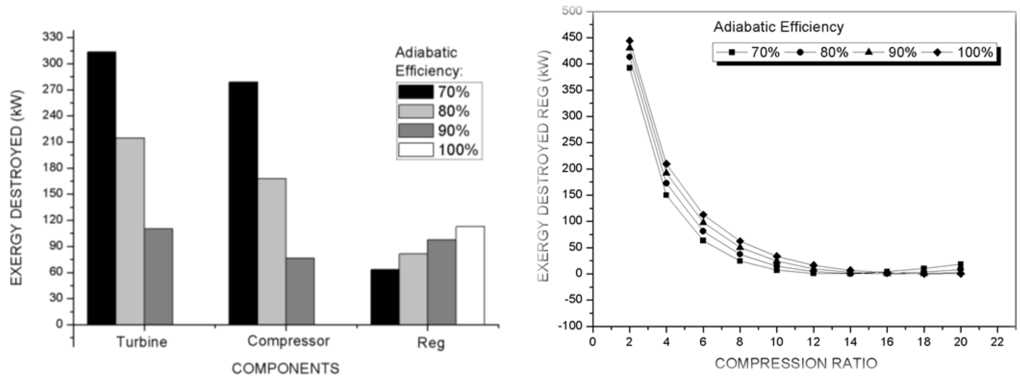


Figure 4. Destruction of exergy a) by components, b) regenerator

## 4. Conclusions

In the study that was carried out at an ideal Brayton cycle with regeneration, the behaviour of the thermal efficiency of the cycle was observed when varying parameters such as the temperature and the adiabatic efficiency of the components at different compression ratios of the compressor, in order to find ways to improve the thermal efficiency of the cycle. The results obtained when varying these parameters and their influence on the thermal efficiency of the cycle and on the destroyed exergy of the components of the ideal Brayton cycle with regeneration were also demonstrated and discussed; it is clear that if you want to improve the thermal efficiency of a gas-powered engine that uses an ideal Brayton cycle with regeneration, you must consider or take into account what has been proposed and demonstrated in this research as a way to achieve this objective. Also, thanks to the exergetic analysis carried out on the cycle in question, you can consider this research to take measures on the components that most destroy useful energy potential (Exergy) in order to have an increasingly more efficient gas-power cycle. As a final point, it can be stated that the objectives of the research on the study of the behaviour of the components of this cycle were met when varying parameters such as the temperature and the adiabatic efficiency of the components at different compressor compression ratios.

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