ENDOGENOUS AND EXOGENOUS EXERGY DESTRUCTION-A RATIONAL APPROACH TO EVALUATE THE THERMODYNAMIC PERFORMANCE OF A 285 MW GAS TURBINE POWER PLANT

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الملخص

تم في هذا العمل تقسيم تحطيم الإكسيرجي الحادث في كل عنصر من عناصر وحدة توربينات غازية في محطة كهرباء السرير في ليبيا، الى تحطيم داخلي وآخر خارجي، وذلك بهدف الحصول على تقييم ديناميكي حراري عقلاني لاداءها. تتكون المحطة من ثلاث وحدات توربينية غازية سيمنز، نوع SGT5-PAC 4000F، كل وحدة بسعة 285 ميجاوات، تحتوي على العناصر المتعارف عليها الثلاثة (ضاغط الهواء وغرفة الاحتراق والتوربينات). لا يقتصر سبب التحطيم الكلي للإكسيرجي على القصور في أداء العنصر المراد تقييم أداه فقط، بل يحدث أيضا بسبب تأثير القصور الحادث في العناصر الأخرى على ذلك العنصر. يحدث التحطيم الداخلي للإكسيرجي عندما تعمل العناصر الأخرى للمنظومة بشكل مثالي دون أي تحطيم للإكسيرجي بينما يعمل العنصر المعني بحالته الطبيعية. يمكن حساب التحطيم الخارجي في كل عنصر من عناصر المنظومة بطرح التحطيم الداخلي من التحطيم الكلي للإكسيرجي. لذلك يجب الأخذ في الاعتبار عند اتخاذ القرارات بتحسين الأداء الديناميكي الحراري للمنظومة إلى مبدأ تقسيم التحطيم الكلي للإكسيرجي إلى تحطيم داخلي وآخر خارجي. يتم ايضا تقسيم تحطيم الإكسيرجي إلى تحطيم يمكن تجنبه وآخر لا يمكن تجنبه، ومن تم يمكن تقسيم كل منها إلى تحطيم داخلي وآخر خارجي. حيث يمكن تجنب جزء فقط من تحطيم الإكسيرجي، بينما لا يمكن تجنب الباقي لأمور اقتصادية وأخرى تكنولوجية. أظهرت النتائج أنه بالنسبة للضاغط، فإن 82٪ من تحطيم الإكسيرجي داخلي و 18٪ تحطيم خارجي، و للتوربينة الغازية، 96.6٪ من تحطيم الإكسيرجي داخلي و 3.4٪ تحطيم خارجي، و لغرفة الاحتراق، 69.60٪ من تحطيم الإكسيرجي داخلي و 30.40٪ تحطيم خارجي. أي أن 75.82 ميجاوات من تحطيم الإكسيرجي في غرفة الاحتراق خارجي المنشأ، وبالتالي يجب تحسين أداء المكونين الآخرين (الضاغط والتوربينات) أو استبدالهما لرفع الأداء الديناميكي الحراري لغرفة الاحتراق. النتائج الأخيرة لا يمكن تمييزها دون تقسيم التحطيم الكلي للإكسيرجي إلى جزء داخلي وآخر خارجي.

ABSTRACT

In this work, the total exergy destruction in a component of a system is split into endogenous and exogenous parts to assess the thermodynamic performance of a gas turbine unit, which is located in the SARIR power station, in Libya. The power station consists of three units, Siemens gas turbines, type SGT5-PAC 4000F, each unit with 285 MW rated capacity with the three ordinary components (air compressor, combustion chamber and turbine) is considered.

The total exergy destruction is not only due the deficiency of that component, but also occurs by the deficiencies of the remaining components. The endogenous exergy destruction takes place when the other components of the system work perfectly without any exergy destruction, and the considered component works with its normal condition. Splitting the total exergy destruction into endogenous and exogenous parts must be considered when decision is made to enhance the thermodynamic performance of a

system. The exergy destruction can be split into an avoidable and unavoidable exergy destruction, each of them can be split into an endogenous and exogenous exergy destruction. Only part of the exergy destruction can be avoided, the remaining cannot be avoided due to economic issues and technological limit.

The results show that for the compressor, 82% of the exergy destruction is endogenous and 18% is exogenous exergy destruction, for the gas turbine, 96.6% of the exergy destruction is endogenous and 3.4% is exogenous exergy destruction and for the combustion chamber, 69.60% of the exergy destruction is endogenous and 30.40% is exogenous exergy destruction. That is, 75.82 MW of the exergy destruction in the combustion chamber is exogenous, hence the performance of the other two components (compressor and turbine) must be improved or replaced to elevate the thermodynamic performance of the combustion chamber. The later finding cannot be recognized without splitting the total exergy destruction into endogenous and exogenous exergy destruction.

KEYWORDS: Exergy; Exergy Destruction; Endogenous Exergy Destruction; Exogenous.

INTRODUCTION

The classical exergy analysis detects the amount and the site of the exergy destruction, however, it fails to indicate the contribution of the inefficiencies of the other elements of the system on the exergy destruction within the element being considered [1]. The destruction of exergy presents the true inefficiencies of an element, hence, an exergy analysis can indicate the extents of enhancing the thermodynamic performance of a system [2]. Exergy analysis pinpoints the causes of deficiencies by determining the irreversibility within each system element, however, care must be taken when evaluating the thermodynamic performance of an element, since part of the element's exergy destruction (irreversibility) occurs due to the inefficiencies of the other elements of the system. A rational approach is considered by splitting the exergy destruction into an endogenous and exogenous exergy destruction.

The general concept of endogenous and exogenous exergy destruction was presented by Tsatsaronis, et al.[3]. A graphical approach was introduced to investigate the performance of a simple gas turbine cycle and a simple refrigeration cycle by splitting the total exergy destruction into endogenous and exogenous parts. The endogenous exergy destruction in an element is independent of the variation of the exergy destruction in the remaining elements. The exogenous exergy destruction in an element depends on the variation of the exergy destruction in the remaining elements. The total exergy destruction in an element is the summation of the endogenous and exogenous exergy destruction.

The concept of endogenous/ exogenous exergy destruction is applied for different applications of thermodynamic systems, for instance, for an air preheater system [4], within a building [5], for regasification of the liquefied natural gas [6], for a ground-source heat pump dryer used in food drying [7], for a simple gas turbine system and co-generation power plant [8] and for milk processing factory [9].

A discussion of four different methods established by the authors for estimating the endogenous part of exergy destruction and a method established on the structural theory was presented [10]. The performance and cost estimation of Kalina cycle combined with

Parabolic-Trough Solar Collectors applying advanced exergy and exergoeconomic based approaches to detect the enhancement potential and the interface between system elements, was introduced by splitting the exergy destruction into endogenous and exogenous and into avoidable and unavoidable parts [11]. The exergy destruction rate and the total operating cost within the components were divided into endogenous/exogenous and unavoidable/avoidable parts.

An analysis based on conventional and advanced exergy analysis was presented for organic Rankine cycle connected to internal combustion engine [12]. The exergy destruction was divided into avoidable/ unavoidable and endogenous and exogenous parts. The result of this analysis indicated that there was a high potential of performance improvement for the cycle being analyzed. An advanced thermodynamic analysis based on avoidable/unavoidable and endogenous/exogenous exergy destruction was illustrated for a heat pump utilized a low grade energy [13]. The results showed that 50% of the total exergy destruction in the elements of the heat pump could be avoided, and about 30 to 40% of this avoidable exergy destruction was an exogenous exergy destruction.

A conventional and advanced thermodynamic analysis was implemented for a combined cycle power plant [14]. The analysis revealed that with the exception of the gas turbine and the high pressure turbine, most of the exergy destruction was unavoidable and constrained by the internal technological limitation which was endogenous. It was concluded that with the innovative analysis, the new advance strategies were exposed that could not otherwise be established. Aiming to identify the causes of the internal deficiencies and the potential of improvement of a power generation system utilizing the cold energy of LNG, the exergy destruction was split into its endogenous and exogenous parts [15]. The results of the advanced thermodynamic analysis, indicated that the main cause of the component exergy destruction was endogenous.

Gas turbine power plants are the dominate technology in generating electricity in many countries, hence a rational approach is needed to estimate their thermodynamic performance. Total exergy destruction in a component in a thermodynamic system does not give the real knowledge of the deficiency of this component, as only part of this destruction could be avoided during the operation, the other part is unavoidable due to the design and economic issues. Furthermore, part of the avoided exergy destruction could be exogenous due to the inefficiency of the other components which is linked to the component being analyzed. The power sector in Libya depends largely on the gas turbine power plants for generating electricity for the public, so it is very important to reassess their performance based on a rational approach. In this work, the concept of the endogenous and exogenous exergy destruction is applied to a gas turbine power plant, which is located in the SARIR power station, in Libya. The power station consists of three units, Siemens gas turbines, type SGT5-PAC 4000F, each unit with 285 MW rated capacity [16]. The analysis is extended to obtain the avoidable-endogenous, avoidable-exogenous, unavoidable-endogenous and unavoidable-exogenous exergy destruction.

THE CONVENTIONAL THERMODYNAMIC MODEL

For the analysis, it is assumed steady-state, steady flow processes, with negligible changes in kinetic and potential energies. The specific heat for air, fuel and gases are assumed to be temperature independent, also the pressure drop in the combustion chamber is neglected. Referring to Figure (1), the net output work and the air mass flow rate can be calculated as follow:

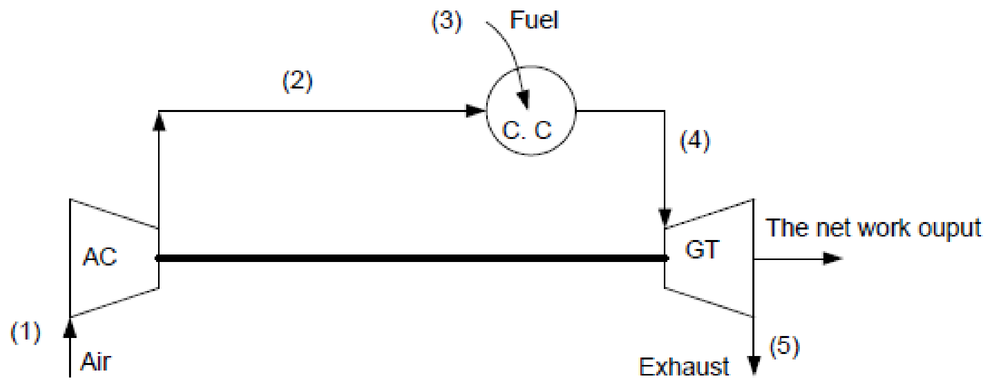

Figure 1: The simple gas turbine cycle

$$\dot{W}_{net} = \dot{W}_{GT} - \dot{W}_{AC} \quad \text{(1-a)}$$

$$\dot{W}_{net} = \dot{m}_g w_{GT} - \dot{m}_a w_{AC} \quad \text{(1-b)}$$

$$\dot{W}_{net} = (\dot{m}_a + \dot{m}_f) w_{GT} - \dot{m}_a w_{AC} \quad \text{(1-c)}$$

$$\dot{W}_{net} = \dot{m}_a (w_{GT} - w_{AC}) + \dot{m}_f w_{GT} \quad \text{(1-d)}$$

And hence the air mass flow rate is given by:

$$\dot{m}_a = \frac{\dot{W}_{net} - \dot{m}_f w_{GT}}{(w_{GT} - w_{AC})} \quad (2)$$

The compressor model

The compressor outlet temperature is given by:

$$T_2 = T_1 \left(1 + \frac{\pi^{\frac{\gamma-1}{\gamma}} - 1}{\eta_{AC}} \right) \quad (3)$$

And hence, the specific work of the compressor is calculated by:

$$w_{AC} = cp_a (T_2 - T_1) = \frac{cp_a T_1}{\eta_{AC}} \left(\pi^{\frac{\gamma-1}{\gamma}} - 1 \right) \quad (4)$$

The concept of fuel and product is applied, where the fuel represents the input power (input exergy) to operate the compressor, which is:

$$\dot{\Psi}_{F,AC} = \dot{m}_a w_{AC} \quad (5)$$

And the product is the exergy gained by the air stream, which is:

$$\dot{\Psi}_{P,AC} = (\dot{\Psi}_2 - \dot{\Psi}_1) = \dot{m}_a (\psi_2 - \psi_1) \quad \text{(6-a)}$$

$$\dot{\Psi}_{P,AC} = \dot{m}_a [(h_2 - h_1) - T_0 (s_2 - s_1)] \quad \text{(6-b)}$$

$$\dot{\Psi}_{P,AC} = \dot{m}_a [w_{AC} - T_0 (s_2 - s_1)] \quad \text{(6-c)}$$

The exergy destruction is given as:

$$\dot{\Psi}_{D,AC} = \dot{\Psi}_{F,AC} - \dot{\Psi}_{P,AC} \quad \text{(7-a)}$$

$$\dot{\Psi}_{D,AC} = \dot{\Psi}_{F,AC}(1 - \varepsilon_{AC}) \tag{7-b}$$

Or:

$$\dot{\Psi}_{D,AC} = \dot{m}_a w_{AC}(1 - \varepsilon_{AC}) \tag{7-c}$$

ε_{AC} is the exergatic efficiency. And hence, the exergy destruction in the air compressor is given by:

$$\dot{\Psi}_{D,AC} = \frac{\dot{W}_{net} - \dot{m}_f w_{GT}}{(w_{GT} - w_{AC})} w_{AC}(1 - \varepsilon_{AC}) \tag{7-d}$$

Equation (7-d) shows the effect of the specific work output of the gas turbine and the fuel mass flow rate on the exergy destruction in the compressor. That means, part of the exergy destruction in the compressor is exogenous. Improving the performance of the gas turbine and the combustion chamber would reduce the exergy destruction in the compressor. When the combustion chamber and the gas turbine operate perfectly without exergy destruction, then the exergy destruction in the compressor is due to the deficiency of the compressor itself and called the endogenous exergy destruction. The difference between the total exergy destruction and endogenous exergy destruction in a component is the exogenous exergy destruction. The exergatic efficiency (effectiveness), ε_{AC} is defined as:

$$\varepsilon_{AC} = \frac{\dot{\Psi}_{P,AC}}{\dot{\Psi}_{F,AC}} \tag{8-a}$$

$$\varepsilon_{AC} = \frac{\dot{m}_a[w_{AC} - T_0(s_2 - s_1)]}{\dot{W}_{AC}} = \frac{[w_{AC} - T_0(s_2 - s_1)]}{w_{AC}} \tag{8-b}$$

For an isentropic compression, $s_2 = s_1$ and hence $\varepsilon_{AC} = 100\%$

The combustion chamber model:

The energy balance on the combustion chamber is given by:

$$\dot{m}_f \eta_{CC}\left[LHV + cp_f\left(T_f - T_1\right)\right] = \left(\dot{m}_a + \dot{m}_f\right)cp_g(T_4 - T_1) - \dot{m}_a cp_a(T_2 - T_1) \tag{9}$$

Then:

$$T_4 = \frac{\dot{m}_f \eta_{CC}\left[LHV + cp_f\left(T_f - T_1\right)\right] + \dot{m}_a cp_a(T_2 - T_1)}{(\dot{m}_a + \dot{m}_f)cp_g} + T_1 \tag{10-a}$$

Or:

$$T_4 = \frac{\eta_{CC}\left[LHV + cp_f\left(T_f - T_1\right)\right] + \lambda cp_a(T_2 - T_1)}{(1 + \lambda)cp_g} + T_1 \tag{10-b}$$

The air/ fuel ratio is given by:

$$\lambda = \frac{\dot{m}_a}{\dot{m}_f} \tag{11}$$

The fuel mass flow rate is given by:

$$\dot{m}_f = \frac{\dot{m}_a\left[cp_g(T_4 - T_1) - cp_a(T_2 - T_1)\right]}{\eta_{CC}\left[LHV + cp_f\left(T_f - T_1\right)\right] - cp_g(T_4 - T_1)} \tag{12}$$

The energy efficiency is expressed as:

$$\eta_{CC} = \frac{(1+\lambda)cp_g(T_4-T_1)-\lambda cp_a(T_2-T_1)}{\left[LHV+cp_f(T_f-T_1)\right]} \tag{13}$$

The exergy destruction is given by:

$$\dot{\Psi}_{D,CC} = \dot{\Psi}_F - \dot{\Psi}_P \tag{14-a}$$

Which is expressed by:

$$\dot{\Psi}_{D,CC} = \dot{m}_f\psi_f - \left[\left(\dot{m}_f + \dot{m}_a\right)\psi_4 - \dot{m}_a\psi_2\right] \tag{14-b}$$

$$\dot{\Psi}_{D,CC} = \dot{m}_f\left\{\psi_f - [(1+\lambda)\psi_4 - \lambda\psi_2]\right\} \tag{14-c}$$

$$\dot{\Psi}_{D,CC} = \frac{\dot{m}_a}{\lambda}\left\{\psi_f - [(1+\lambda)\psi_4 - \lambda\psi_2]\right\} \tag{14-d}$$

The substitution with the air mass flow rate (as given by Equation (2)) results in:

$$\dot{\Psi}_{D,CC} = \frac{\dot{W}_{net}-\dot{m}_f w_{GT}}{\lambda(w_{GT}-w_{AC})}\left\{\psi_f - [(1+\lambda)\psi_4 - \lambda\psi_2]\right\} \tag{14-e}$$

$$\dot{\Psi}_{D,CC} = \frac{\dot{W}_{net}-\dot{m}_f w_{GT}}{\lambda(w_{GT}-w_{AC})}\psi_f(1-\varepsilon_{CC}) \tag{14-f}$$

Equation (14-f) shows the effect of the turbine and compressor specific works on the exergy destruction in the combustion chamber. Increasing the turbine specific work and reducing the compressor specific work will cause a drop in the exergy destruction in the combustion chamber. The exergy destruction in the combustion chamber could be reduced by increasing its exergatic efficiency. When the isentropic works for the turbine and the compressor are applied, and by fixing the exergatic efficiency of the combustion chamber at its normal value, then the exergy destruction in the combustion chamber presents the endogenous exergy destruction.

The exergatic efficiency is given by:

$$\varepsilon_{CC} = \frac{(1+\lambda)\psi_4-\lambda\psi_2}{\psi_f} \tag{15}$$

The gas turbine model

The temperature of the exhaust gases at the turbine outlet is given by:

$$T_5 = T_4\left[1-\eta_{GT}\left(1-\frac{1}{\pi^{\frac{\gamma_g-1}{\gamma_g}}}\right)\right] \tag{16}$$

And hence, the specific work is calculated by:

$$w_{GT} = cp_g(T_4-T_5) = \eta_{GT}cp_gT_4\left(1-\frac{1}{\pi^{\frac{\gamma_g-1}{\gamma_g}}}\right) \tag{17}$$

The exergy destruction is obtained by:

$$\dot{\Psi}_{D,GT} = \left(\dot{\Psi}_4 - \dot{\Psi}_5\right) - \dot{W}_{GT} \tag{18-a}$$

$$\dot{\Psi}_{D,GT} = \left(\dot{m}_a + \dot{m}_f\right)[(\psi_4-\psi_5) - w_{GT}] \tag{18-b}$$

$$\dot{\Psi}_{D,GT} = \left(\dot{m}_a + \dot{m}_f\right)w_{GT}\left[\frac{1}{\varepsilon_{GT}} - 1\right] \tag{18-c}$$

By substituting with the air mass flow rate (Equation (2)), the exergy destruction becomes:

$$\dot{\Psi}_{D,GT} = \frac{\dot{W}_{net} - \dot{m}_f w_{AC}}{w_{GT} - w_{AC}} w_{GT} \left(\frac{1}{\varepsilon_{GT}} - 1\right) \tag{18-d}$$

Equation (18-d) shows the effect of the specific work of the compressor and the fuel mass flow rate on the exergy destruction in the turbine. The endogenous exergy destruction in the gas turbine is obtained if the exergy destruction in the compressor and combustion chamber are set to zero.

The exergatic efficiency is given by:

$$\varepsilon_{GT} = \frac{w_{GT}}{\psi_4 - \psi_5} = \frac{w_{GT}}{w_{GT} - T_0(s_4 - s_5)} \tag{19}$$

For the isentropic expansion, $s_5 = s_4$ and hence $\varepsilon_{GT} = 100\%$

A heat exchanger model

If a system contains a heat exchanger, see Figure (2), then the exergatic efficiency could be expressed as:

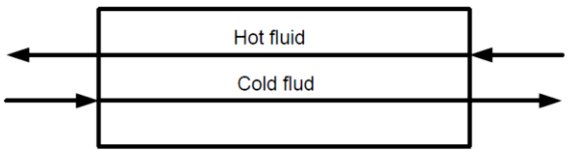

Figure 2: Heat exchanger

$$\varepsilon_{H/X} = \frac{(\Delta\Psi)_{cold}}{(\Delta\Psi)_{hot}} = \frac{(\Delta H - T_0 \Delta S)_{cold}}{(\Delta H - T_0 \Delta S)_{hot}} \tag{20}$$

Theoretically, the exergatic efficiency is 100% for an adiabatic and reversible heat transfer process (isentropic), that requires an infinitesimal temperature difference between the two streams. Practically the heat transfer is not isentropic and the exergatic efficiency cannot be 100% as a finite temperature difference must be kept between the two streams.

THE GENERAL APPROACH OF THE ENDOGENOUS/EXOGENOUS EXERGY DESTRUCTION [3]

The advanced thermodynamic model is based on the endogenous/ exogenous exergy destruction, such that for a component "k":

$$\dot{\Psi}_{D,k} = \dot{\Psi}_{D,k}^{EN} + \dot{\Psi}_{D,k}^{EX} \tag{21}$$

In general, the exergy destruction in the k^{-th} component is expressed as:

$$\dot{\Psi}_{D,k} = \dot{\Psi}_{F,k} - \dot{\Psi}_{P,k} \tag{22}$$

And for the whole system:

$$\dot{\Psi}_{D,tot} = \sum \dot{\Psi}_{D,k} = \dot{\Psi}_{F,tot} - \dot{\Psi}_{P,tot} - \dot{\Psi}_{L,tot} \tag{23}$$

The effectiveness (exergatic efficiency) for the component "k" is given by:

$$\varepsilon_k = \frac{\dot{\Psi}_{P,k}}{\dot{\Psi}_{F,k}} = \frac{\dot{\Psi}_{F,k} - \dot{\Psi}_{D,k}}{\dot{\Psi}_{F,k}} = 1 - \frac{\dot{\Psi}_{D,k}}{\dot{\Psi}_{F,k}} \tag{24}$$

The concept of the endogenous/exogenous is initially illustrated by using Figure (3), which is composed of three components A, B and C. The three components are linked together such that: $\dot{\Psi}_{F,A} = \dot{\Psi}_{F,tot}$, $\dot{\Psi}_{P,A} = \dot{\Psi}_{F,B}$, $\dot{\Psi}_{P,B} = \dot{\Psi}_{F,C}$ and $\dot{\Psi}_{P,C} = \dot{\Psi}_{P,tot}$.

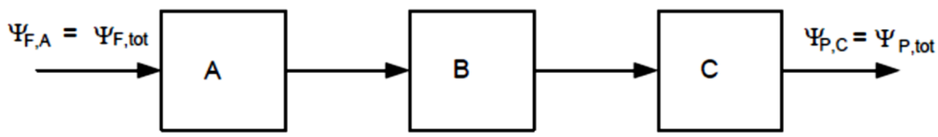

Figure 3: Illustration case for endogenous/exogenous exergy destruction

The exergy destruction in component "C" is determined by:

$$\dot{\Psi}_{D,C} = \dot{\Psi}_{F,C} - \dot{\Psi}_{P,C} = \dot{\Psi}_{F,C} - \dot{\Psi}_{P,tot} \tag{25-a}$$

$$\dot{\Psi}_{D,C} = \dot{\Psi}_{P,tot}\left(\frac{\dot{\Psi}_{F,C}}{\dot{\Psi}_{P,C}} - 1\right) = \dot{\Psi}_{P,tot}\left(\frac{1}{\varepsilon_C} - 1\right) \tag{25-b}$$

ε_C is the exergatic efficiency of the component "C". As can be seen, by fixing the total exergy of the product of the whole system ($\dot{\Psi}_{P,tot}$), the exergy destruction in this particular component (C) is a function only on its exergatic efficiency (ε_C) and has nothing to do with inefficiencies of the other components, that means, $\dot{\Psi}_{D,C}^{EX} = 0.0$, and hence $\dot{\Psi}_{D,C} = \dot{\Psi}_{D,C}^{EN}$.

The exergy destruction in component "B" is given by:

$$\dot{\Psi}_{D,B} = \dot{\Psi}_{F,B} - \dot{\Psi}_{P,B} \tag{26-a}$$

$$\dot{\Psi}_{D,B} = \dot{\Psi}_{P,B}\left(\frac{\dot{\Psi}_{F,B}}{\dot{\Psi}_{P,B}} - 1\right) = \dot{\Psi}_{P,B}\left(\frac{1}{\varepsilon_B} - 1\right) = \dot{\Psi}_{F,C}\left(\frac{1}{\varepsilon_B} - 1\right) \tag{26-b}$$

$$\dot{\Psi}_{D,B} = \frac{\dot{\Psi}_{P,C}}{\varepsilon_C}\left(\frac{1}{\varepsilon_B} - 1\right)\frac{\dot{\Psi}_{P,tot}}{\varepsilon_C}\left(\frac{1}{\varepsilon_B} - 1\right) \tag{26-c}$$

Equation (26-c) indicates, by fixing the total exergy of the product of the whole system, the exergy destruction of this component (B) is:

$$\dot{\Psi}_{D,B} = f(\varepsilon_B, \varepsilon_C) \tag{26-e}$$

That means, the component "C" contributes to exergy destruction in the component "B", and the exergy destruction in the component "B" varies with the change in ε_C. In other words, part of the exergy destruction in the component "B" is endogenous due to its internal deficiency and the other part is exogenous due to the deficiency in the component "C". The values of ε_B and ε_C determine the endogenous and exogenous exergy destruction in the component "B", respectively.

By setting ε_C=1, the endogenous exergy destruction in the component "B" is given by:

$$\dot{\Psi}_{D,B}^{EN} = \dot{\Psi}_{P,tot}\left(\frac{1}{\varepsilon_B} - 1\right) \tag{27}$$

Since:

$$\dot{\Psi}_{D,B}^{EX} = \dot{\Psi}_{D,B} - \dot{\Psi}_{D,B}^{EN} \tag{28-a}$$

Then, the exogenous exergy destruction in the component "B" is given by:

$$\dot{\Psi}_{D,B}^{EX} = \dot{\Psi}_{P,tot}\left(\frac{1}{\varepsilon_B} - 1\right)\left(\frac{1}{\varepsilon_C} - 1\right) \tag{28-b}$$

For the component "A", the exergy destruction is determined by:

$$\dot{\Psi}_{D,A} = \dot{\Psi}_{F,A} - \dot{\Psi}_{P,A} \tag{29-a}$$

$$\dot{\Psi}_{D,A} = \dot{\Psi}_{P,A}\left(\frac{\dot{\Psi}_{F,A}}{\dot{\Psi}_{P,A}} - 1\right) = \dot{\Psi}_{P,A}\left(\frac{1}{\varepsilon_A} - 1\right) = \dot{\Psi}_{F,B}\left(\frac{1}{\varepsilon_A} - 1\right) \tag{29-b}$$

$$\dot{\Psi}_{D,A} = \frac{\dot{\Psi}_{P,B}}{\varepsilon_B}\left(\frac{1}{\varepsilon_A} - 1\right) = \frac{\dot{\Psi}_{F,C}}{\varepsilon_B}\left(\frac{1}{\varepsilon_A} - 1\right) = \frac{\dot{\Psi}_{P,tot}}{\varepsilon_B \varepsilon_C}\left(\frac{1}{\varepsilon_A} - 1\right) \quad (29\text{-c})$$

Equation (29-c) indicates, by fixing the total exergy of the product of the whole system, the exergy destruction in the component (A) is:

$$\dot{\Psi}_{D,A} = f(\varepsilon_A, \varepsilon_B, \varepsilon_C) \quad (29\text{-d})$$

The foregoing equations reveals that, the components “C” and “B” contribute to the exergy destruction in the component “A”, and the exergy destruction in the component “A” varies with the change in ε_C and ε_B. In other words, part of the exergy destruction in the component “A” is endogenous due to its internal deficiency and the rest is exogenous due to the deficiencies in the components “B” and “C”. The value of ε_A, determines the endogenous part of the exergy destruction in the component “A”, and ε_B and ε_C determine the exogenous exergy destruction in the component “A”.

By setting ε_B =1 and ε_C =1, the endogenous exergy destruction in the component “A” is given by:

$$\dot{\Psi}_{D,A}^{EN} = \dot{\Psi}_{P,tot}\left(\frac{1}{\varepsilon_A} - 1\right) \quad (30$$

Since:

$$\dot{\Psi}_{D,A}^{EX} = \dot{\Psi}_{D,A} - \dot{\Psi}_{D,A}^{EN} \quad (31\text{-a})$$

Then the exogenous exergy destruction in the component “A” is given by:

$$\dot{\Psi}_{D,A}^{EX} = \dot{\Psi}_{P,tot}\left(\frac{1}{\varepsilon_B \varepsilon_C} - 1\right)\left(\frac{1}{\varepsilon_A} - 1\right) \quad (31\text{-b})$$

The foregoing analysis indicates that, care must be considered when evaluating the thermodynamic performance of a component in a thermodynamic system since part of its exergy destruction could be due to the deficiencies of the other components of the system being analyzed.

THE BASIS OF THE GRAPHICAL APPROACH FOR THE ENDOGENOUS/EXOGENOUS EXERGY DESTRUCTION [3]

As discussed before, the endogenous exergy destruction of a component “k” is obtained by keeping its real exergatic efficiency “ε_k” , and setting the thermal efficiencies of the other components to 100% (isentropic processes). Furthermore, it was shown that, the exergatic efficiency cannot attain a value of 100% for the combustion and heat transfer processes for instance, such processes are existing in the gas turbine power stations. A graphical approach is developed to overcome this limitation.

For ideal system without exergy destruction and with a fixed supply of product “$\dot{\Psi}_{P,tot}$” the exergy balance is given as:

$$\dot{\Psi}_{F,tot}^{ID} - \dot{\Psi}_{L,tot}^{ID} - \dot{\Psi}_{P,tot} = 0 \quad (32)$$

If there is only one component “k” characterized by some kind of irreversibility (exergy destruction) then, additional exergy must be supplied ($\Delta\dot{\Psi}_{F,tot}^{k}$) and the exergy losses increases by $\Delta\dot{\Psi}_{L,tot}^{k}$. The last equation is then modified as follow:

$$\left(\dot{\Psi}_{F,tot}^{ID} + \Delta\dot{\Psi}_{F,tot}^{k}\right) - \left(\dot{\Psi}_{L,tot}^{ID} + \Delta\dot{\Psi}_{L,tot}^{k}\right) - \dot{\Psi}_{P,tot} = \dot{\Psi}_{D,k} \quad (33)$$

Since the other components work without exergy destruction, then the exergy destruction in the component "k" is the endogenous exergy destruction, hence the last equation could be written as:

$$\left(\dot{\Psi}_{F,tot}^{ID} + \Delta\dot{\Psi}_{F,tot}^{k}\right) - \left(\dot{\Psi}_{L,tot}^{ID} + \Delta\dot{\Psi}_{L,tot}^{k}\right) - \dot{\Psi}_{P,tot} = \dot{\Psi}_{D,k}^{EN} \quad (34)$$

When there are irreversibilities in the all components, then the exergy balance equation is given by:

$$\left(\dot{\Psi}_{F,tot}^{ID} + \Delta\dot{\Psi}_{F,tot}^{RS}\right) - \left(\dot{\Psi}_{L,tot}^{ID} + \Delta\dot{\Psi}_{L,tot}^{RS}\right) - \dot{\Psi}_{P,tot} = \dot{\Psi}_{D,others} + \dot{\Psi}_{D,k} \quad (35)$$

As $\dot{\Psi}_{D,others}$ tends to zero in equation (35), the left hand side of the same equation approaches the left hand side of equation (33), and hence the right hand side of both equation approaches each other, i. e. $\dot{\Psi}_{D,k}$ approaches $\dot{\Psi}_{D,k}^{EN}$. Then by plotting the expression $\left(\dot{\Psi}_{F,tot}^{ID} + \Delta\dot{\Psi}_{F,tot}^{RS}\right) - \left(\dot{\Psi}_{L,tot}^{ID} + \Delta\dot{\Psi}_{L,tot}^{RS}\right) - \dot{\Psi}_{P,tot}$ against $\dot{\Psi}_{D,others}$, the intercept will be the endogenous exergy destruction ($\dot{\Psi}_{D,k}^{EN}$) of the component "k".

EXERGY BALANCE APPROACH [10]

For ideal process, that is process without exergy destruction, we may write:

$$\dot{\Psi}_F = \dot{\Psi}_P \quad (36)$$

Equation (36) is valid for isentropic processes, such as in an a gas turbine where $\varepsilon_{GT} = 1.0$ and in a compressor, where $\varepsilon_{AC} = 1.0$. Back to Figure (1). For ideal processes, Equation (36) may be applied to the combustion chamber as follow:

$$\dot{\Psi}_f = \dot{\Psi}_4 - \dot{\Psi}_2 \quad (37)$$

Which can be written as:

$$\dot{\Psi}_f = \dot{m}_g \psi_4 - \dot{\Psi}_2 \quad (38)$$

Then:

$$\dot{m}_g = \frac{\dot{\Psi}_f + \dot{\Psi}_2}{\psi_4} \quad (39)$$

All parameters in the right hand side of Equation (39) are fixed as in the real processes, and hence a fictitious mass flow rate of gasses ($\dot{m}_g$)is calculated to satisfy Equation (37).

AVOIDABLE AND UNAVOIDABLE EXERGY DESTRUCTION

The exergy destruction can be split into avoidable and unavoidable destructions [17] as follow:

$$\dot{\Psi}_{D,k} = \dot{\Psi}_{D,k}^{AV} + \dot{\Psi}_{D,k}^{UN} \quad (40)$$

$$\dot{\Psi}_{D,k,A}^{UN} = \dot{\Psi}_{P,k,A} \left(\frac{\dot{\Psi}_D}{\dot{\Psi}_p}\right)_k^{UN} \quad (41)$$

The subscript "A" denotes the operating point of the equipment [17].

MODELING THE GAS TURBINE POWER PLANT

For the analysis, the gas turbine power unit which is located in Sarir power station, in Libya. Sarir power station is selected. It comprises of three units Siemens gas turbines, type SGT5-PAC 4000F and each unit with 285 MW rated capacity Generally the GT power units consist of three major components namely the air compressor (AC), combustion chamber (CC) and turbine as shown in Figure (1), the design data is given in Table (1) [16].

Table 1: Design parameters

Item	Parameter	Value	Unit
Compressor	Inlet temperature	288.15	[K]
	Inlet pressure	1.01325	[bar]
	Pressure ratio	17.43	[-]
	Air-mass flow rate	672	[kg/s]
	Air-c_{Pa}	1.00351	[kJ/kg.K]
	Air-specific heat ratio	1.40	[-]
	Thermal efficiency	87.468 %	[-]
Combustion chamber	Fuel inlet temperature	288.15	[K]
	Fuel inlet pressure	12	[bar]
	Fuel mass flow rate	14.994	[kg/s]
	Efficiency	95 %	[-]
	Lower heating value	52000	[kJ/kg]
	Fuel chemical exergy	52025	[kJ/kg]
	Fuel-c_{Pf}	2.25	[kJ/kg.K]
Gas turbine	Inlet temperature	1584	[K]
	Pressure ratio	17.43	[-]
	Thermal efficiency	88.8 %	[-]
	Gas-specific heat ratio	1.33	[-]
	Gas c_{pg}	1.148	[kJ/kg.K]
Reference state	Temperature (T_0)	288.15	[K]
	Pressure (P_0)	1.01325	[bar]

RESULTS AND DESCUSSION

Thermodynamic parameters, operating powers, and exergy destruction are calculated and tabulated in Tables (2-4).

Table 2: Thermodynamic parameters at each state

STATE	T(K)	P(kPa)	$\dot{m}\ (\frac{kg}{s})$	$\psi\ (\frac{kJ}{kg})$	$\dot{\Psi}\ (MW)$
1	288.15	101.3	672	0	0
2	705.6952	1765.659	672	396.1434	266.2083
3 (Fuel)	288.15	1200	14.99405	52025	780.0653
4	1583.83	1765.659	686.994	1159.916	796.8556
5	866.0886	101.3	686.994	299.4303	205.7068

Table 3: Compressor, Turbine and Net power

$\dot{W}_C$	281.1567 (MW)
$\dot{W}_T$	566.0602 (MW)
$\dot{W}_{Net}$	284.9036 (MW)

As can be seen, the exergy destruction in the combustion chamber contributes to about 86 % of the total exergy destruction.

Table 4: Exergy destruction and the effectiveness of each component

Component	$\dot{\Psi}_F$	$\dot{\Psi}_P$	$\dot{\Psi}_D$	$\dot{\Psi}_D$ %	ε
AC	281.1567	266.2083	14.94832	5.164299	0.946833
CC	780.0653	530.6473	249.418	86.16817	0.68026
GT	591.1488	566.0602	25.08859	8.667528	0.95756
Total exergy destruction			289.46		

The graphical approach is applied to calculate the endogenous exergy destruction in the compressor. Here, the exergatic efficiency of the compressor is kept at its nominal value of 94.6833 % (Table 4). The thermal efficiency of the gas turbine is set to 100 %, hence the exergy destruction in the turbine is eliminated. Equation (16) is used to calculate T_{5s} and the gas turbine isentropic work is calculated by Equation (17). Since, T_2 and T_4 are both fixed to their design value (Table 2), the exergy destruction in the combustion chamber cannot be reduced directly. One way to reduce the exergy destruction in the combustion chamber is by heating the air before entering the combustion chamber adiabatically and reversibly to a new elevated inlet temperature T_2^* by using adiabatic-reversible heater, as shown in Figure (4).

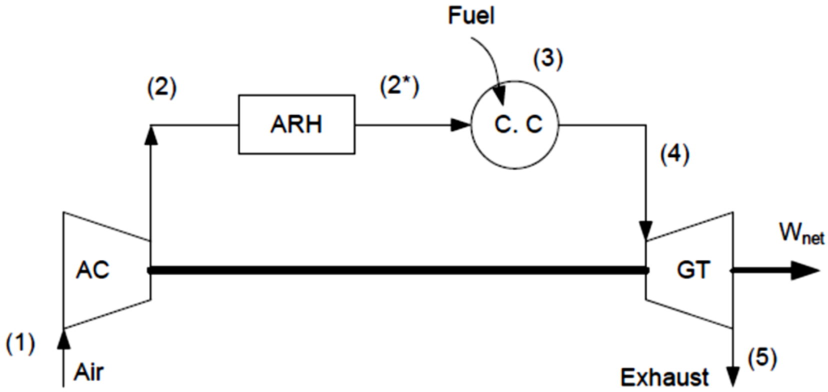

Figure 4: The modified gas turbine cycle

The input temperature is increased gradually to 668, 683 and 698 K, as shown in Figure (5). Then, less fuel is needed to heat the working stream from state 2* to state 4, hence the exergatic efficiency of the combustion chamber is increased and its exergy destruction is reduced.

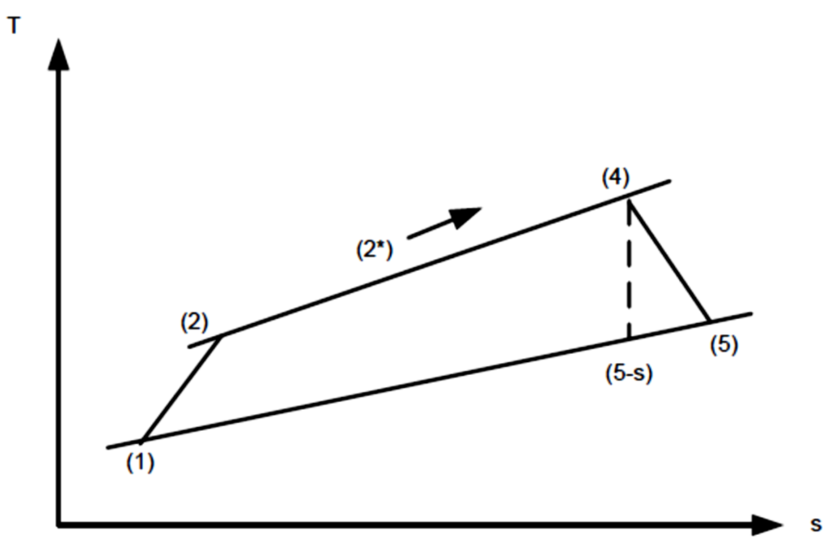

Figure 5: Variation of the combustion chamber inlet temperature and the specific work of the gas turbine

Equation (12) is used to calculate the new mass flow rate of the fuel, Equation (15) is used to calculate the new exergatic efficiency of the combustion chamber and Equation (14—f) is used to calculate the reduced exergy destruction of the combustion chamber.

Equation (7-d) is used to calculate the exergy destruction in the compressor. Here, the net power output is set equal to the design value of 285 MW, and the specific work of the turbine is replaced by the isentropic specific work. The expression $\dot{\Psi}_{F,tot} - \dot{\Psi}_{L,tot} - \dot{\Psi}_{P,tot}$ is plotted against $\dot{\Psi}_{D,others}$, the intercept is the endogenous exergy destruction ($\dot{\Psi}_{D,C}^{EN}$) in the compressor as shown in Figure (6).

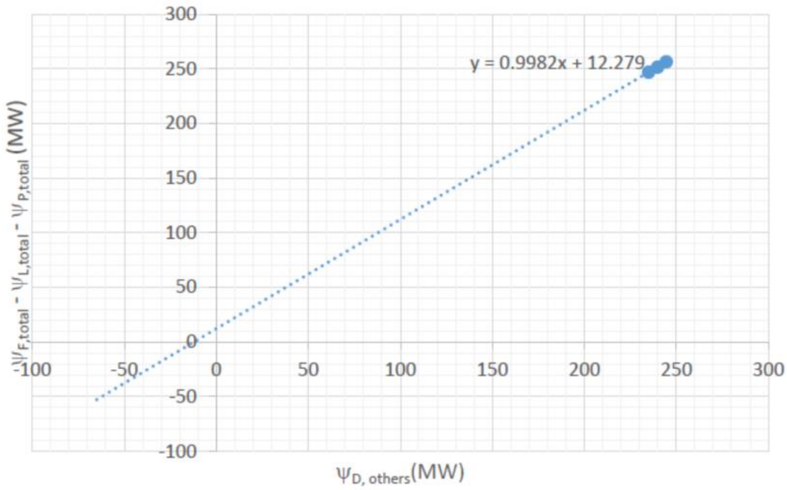

Figure 6: Endogenous exergy destruction in the compressor

To obtain the endogenous exergy destruction in the gas turbine, the exergatic efficiency of the gas turbine is fixed at its original value of 95.756%, see Table (4) while the thermal efficiency of the compressor is elevated to 100%, then the exergy destruction in the compressor is eliminated as shown in Figure (7). Equation (3) is used to calculate T_{2s} and equation (4) to calculate the isentropic work. To reduce the exergy destruction in the combustion chamber and hence elevate its exergatic efficiency, the adiabatic reversible heater is used, and the inlet temperature of the air entering the combustion chamber is gradually increased to 668, 683 and 698 K, as shown in Figure (7). Then, less fuel is needed to heat the working stream from state 2* to state 4, hence the exergatic efficiency of the combustion chamber is raised and its exergy destruction is reduced. The new fuel mass flow rate, the combustion chamber exergatic efficiency and the exergy destruction are calculated as previously discussed.

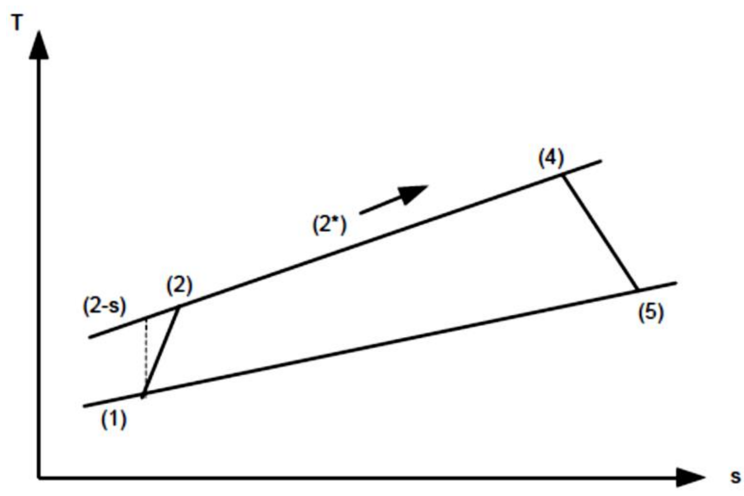

Figure 7: Variation of the combustion chamber inlet temperature and the specific work of the compressor

The expression $\dot{\Psi}_{F,tot} - \dot{\Psi}_{L,tot} - \dot{\Psi}_{P,tot}$ is plotted against $\dot{\Psi}_{D,others}$, the intercept is the endogenous exergy destruction ($\dot{\Psi}_{D,GT}^{EN}$) in the gas turbine, as shown in Figure (8).

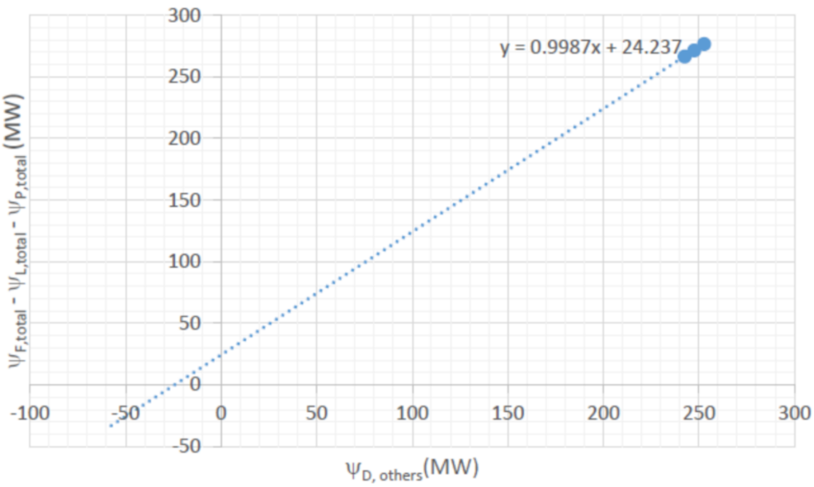

Figure 8: Endogenous exergy destruction in the gas turbine

For the combustion chamber, there is no need to use the adiabatic-reversible heater to obtain the endogenous exergy destruction since the exergatic efficiency should be fixed to its original value of 68.026% as given in Table (4). The thermal efficiency of the compressor is set to 100%, the isentropic work of the compressor and its new outlet temperature is calculated as previously discussed. T_{2s} is calculated as 653.3684 K., where T_2 = 705.6952 K as given in Table (2). By keeping the same amount of the fuel flow rate, the exergatic efficiency of the combustion chamber is reduced to 66.9433% (the original value is 68.026). To retain the original value of the exergatic efficiency of the combustion chamber, Equations (12 and 15) are used to calculate the new mass flow rate of the fuel and the combustion chamber outlet temperature, it is found T_4 = 1669.412 K and $\dot{m}_f =$ 17.14258 kg/s. The exergy destruction in the gas turbine is calculated by varying its thermal efficiency between 90% and 94%.

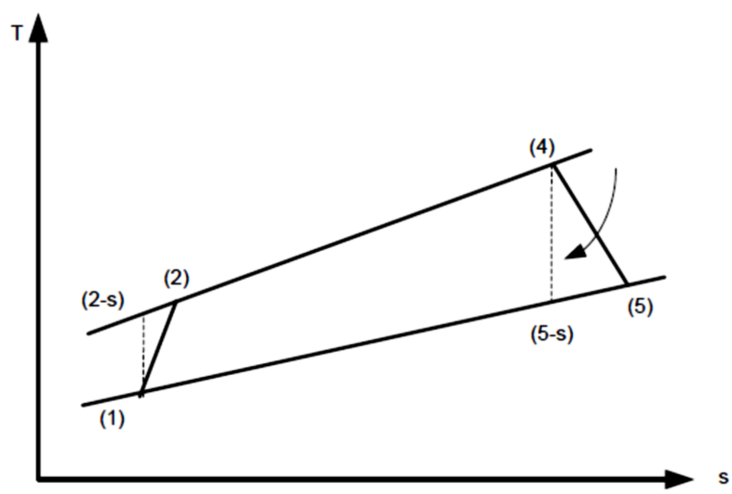

Figure 9: Direction of variations in the compressor and gas turbine efficiencies

The expression $\dot{\Psi}_{F,tot} - \dot{\Psi}_{L,tot} - \dot{\Psi}_{P,tot}$ is plotted against $\dot{\Psi}_{D,others}$, the intercept is the endogenous exergy destruction ($\dot{\Psi}_{D,C.C}^{EN}$) in the combustion chamber as shown in Figure (10).

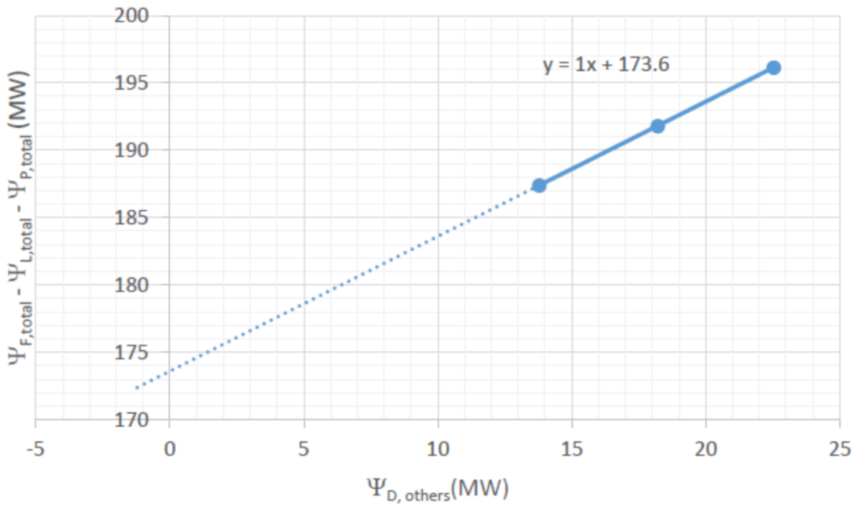

Figure 10: Endogenous exergy destruction in the combustion chamber

Figure (11) summarizes the results of the exergy destruction in MW Figure (12) summarizes the results in percentage. The largest exergy destruction occurs in the combustion chamber, which is 249.42 MW, its exogenous part is 75.82 MW.

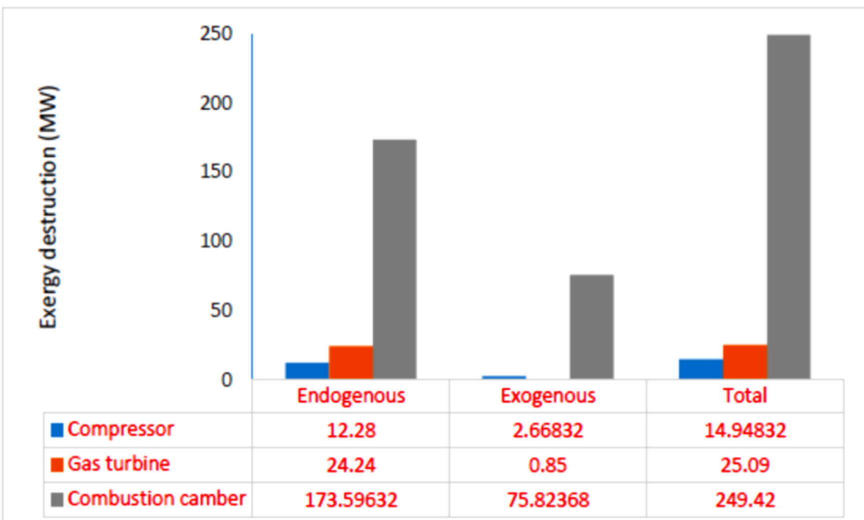

Figure 11: Endogenous, exogenous and total exergy destruction in MW

The exogenous exergy destruction in the combustion chamber contributes to 30.4% of its total exergy destruction, hence, decisions for adjustments should be to raise the performance of the other two components. This finding cannot be obtained without splitting the total exergy destruction into endogenous and exogenous exergy destruction.

For the same gas turbine unit the avoidable and unavoidable exergy destruction as given [17], shows that, $\left(\frac{\dot{\Psi}_D}{\dot{\Psi}_P}\right)_k^{UN}$ is 0.05, 0.43 and 0.03 for the compressor, combustion chamber and gas turbine, respectively. The percentage of the endogenous and exogenous exergy destruction is shown in Figure (12). The total exergy destruction is 289.46 MW and the product exergy for each component is given in Table (4). Applying equations (36, 37), the voidable and unavoidable exergy destruction are calculated. Table (5), tabulates the avoidable-endogenous, avoidable-exogenous, unavoidable–endo and unavoidable-exogenous exergy destruction. The same parameters are given in Figure (13) as percentage.

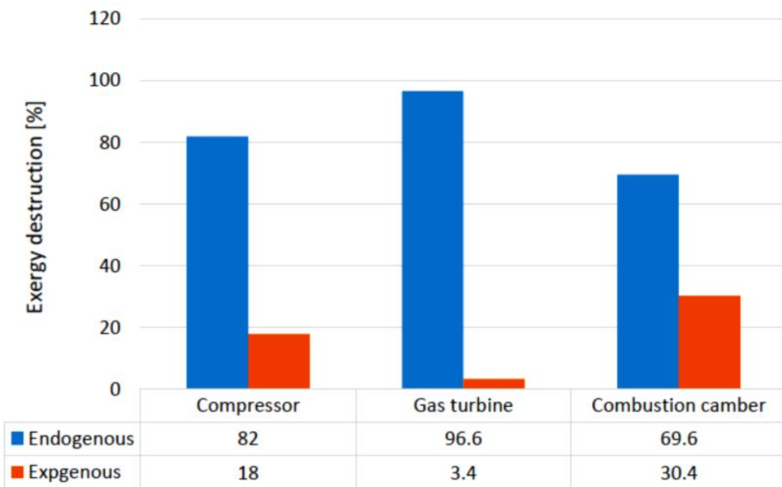

Figure 12: Endogenous/ exogenous exergy destruction in percentage

Table 5: Avoidable-Unavoidable, Endogenous-Exogenous exergy destruction

	$\dot{\Psi}_D^{AV}$ (MW) [17]	$\dot{\Psi}_D^{UN}$ (MW) [17]	$\dot{\Psi}_D^{AV,EN}$ (MW)	$\dot{\Psi}_D^{AV,EXO}$ (MW)	$\dot{\Psi}_D^{UN,EN}$ (MW)	$\dot{\Psi}_D^{UN,EXO}$ (MW)
AC	2.23	12.72	1.83	0.40	10.45	2.27
C.C	18.96	230.46	13.20	5.76	160.40	70.06
GT	6.81	18.28	6.58	0.23	17.66	0.62
Subtotal	**28**	**261.46**	**21.61**	**6.39**	**188.51**	**72.95**
Total	**289.46**			**289.46**		

Table (5) shows the avoidable exergy destruction in the whole system is 28 MW which presents 9.7% of the total exergy destruction (of 289.46 MW), and 22.83% of this destruction is exogenous (6.39 MW of 28 MW).

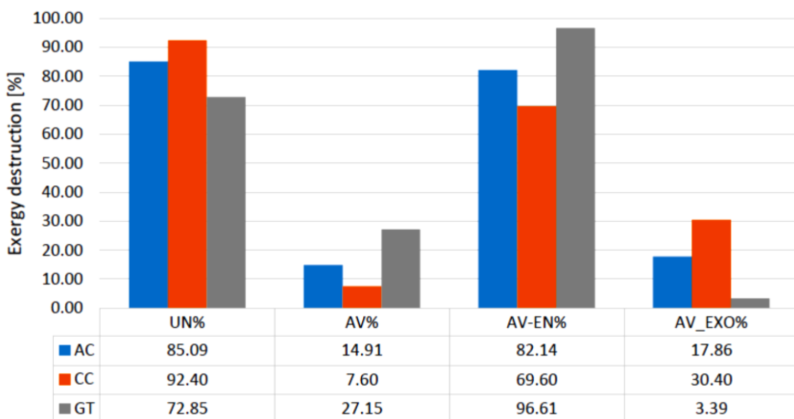

Figure 13: Avoidable-unavoidable, endogenous-exogenous exergy destruction in percentage

The largest exergy destruction occurs in the combustion chamber, which is 249.42 MW, only 18.96 MW is avoidable and 230.46 MW is unavoidable. The avoidable-exogenous exergy destruction in the combustion chamber is 5.76 MW which represents 30.4% of the avoidable exergy destruction in the combustion chamber, this result shows the need to improve the performance of the other two components.

The results of the graphical approach are compared with the results obtained from the exergy balance approach, the results of comparison are shown in Table (6). Almost both approaches give the same results.

Table 6: The comparison between graphical and exergy balance approaches

Component	Graphical approach		Exergy balance approach	
	$\dot{\Psi}_D^{IN}$ (MW)	$\dot{\Psi}_D^{EXO}$ (MW)	$\dot{\Psi}_D^{IN}$ (MW)	$\dot{\Psi}_D^{EXO}$ (MW)
AC	12.28	2.67	12.42	2.53
CC	173.60	75.82	173.60	75.82
GT	24.24	0.85	23.94	1.15

CONCLUSIONS

The exergy destruction in a component of a system is split into endogenous and exogenous exergy destruction. The former occurs internally due to the imperfection in the structure and the operation of the component itself, while the later occurs due to the interaction with the remaining components. The general concept of endogenous and exogenous exergy destruction is applied to 285 MW gas turbine power unit. The graphical approach is applied for the analysis. The concept of the endogenous/ exogenous exergy destruction increases our understanding of the interaction between system components, and makes the optimization of the overall system more convenient. It can help engineers to decide whether an alteration should be made to a particular component or to the other components of the system. Important knowledge for improving the potential of every system component is recognized and the need for components' adjustment, are ordered. This knowledge cannot be recognized by the conventional thermodynamic analyses to obtain the endogenous exergy destruction in a component, the exergatic efficiency of this particular component is held constant at its nominal value, while the thermal efficiency of the other components are increased to reduce their exergy destruction. However, a reduction in the exergy destruction in one component might increase the exergy destruction in another component. For instance, by eliminating the exergy destruction in the compressor (isentropic process), an increase in the exergy destruction in the combustion chamber is noticed. To overcome this limitation, an adiabatic-reversible heat exchanger is inserted such that part of the heat transfer takes place adiabatically and reversibly.

NOMENCLATURE

		Subscripts	
c_P (kJ/kg.K)	constant pressure specific heat	a	air
LHV (kJ/kg)	lower heating value	AC	air-compressor
$\dot{m}$ (kg/s)	mass flow rate	D	destruction
s (kJ/kg.K)	specific entropy	f, F	fuel
T (K)	temperature	g	gas
$\dot{W}$ (MW)	power	GT	gas turbine
w (kJ/kg)	Specific work	k	component
Greek symbol		P	product
ψ (kJ/kg)	specific exergy	**Superscript**	
η	thermal efficiency	EN	endogenous
ε	exergatic efficiency	EX	exogenous
Δ	change	ID	ideal
λ	air/fuel ratio	k	component
π	pressure ratio	RS	real system
$\dot{\Psi}$ (MW)	exergy rate	AV	avoidable
γ	specific heat ratio	UN	unavoidable

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