Numerical Approach for Energy Analysis of Gt Power Plant.

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Abstract

Gas turbine power plants are of major interest because of their power capacity and lower investment. However, engineers and researchers still struggle in order to maximize their energy-economic efficiency and reduce their greenhouse gas emissions. These targets were formerly simulated using numerical integration, and uncommonly are tackled experimentally. In this work, a numerical analytical model has been established emphasizing the effect of pressure ratio on simple and regenerative gas turbine cycle performance which is successively used in the energy and exergy analysis of the power system. The elaborated analytical model permits a simple and quick calculation of maximum power output and efficiency based on pressure ratio and turbine inlet temperature using a direct and exact equation which can generate a rapid calculation for specific parameters. These equations will be helpful for designers and industrialists. The established analytical calculation is validated against some numerical examples and available data, and the agreement is observed to be fairly acceptable for such a universal and quite simple analytical model.

Keywords:

Energy efficiency, Net power output, Pressure ratio, Energy, Exergy.

1. Introduction

Gas turbines are progressively used as a principal source of power and heat for several industries. Precisely for this reason, over the past few years, a recorded industry effort and research investigation rate on gas turbine cycle have increased meaningfully to enhance their performance and reduce energy consumption with CO2 emission [1, 2].

In order to analyze the effect of rotating speed and nozzle stagger angle for a gas turbine, Hao et al. [3] established a thermodynamic analysis for three different gas turbines with diverse load scale. It was concluded that for higher performance of gas turbine the power turbine rotating speed should be lowest and it was founded an optimal value of the nozzle stagger angle for higher thermal efficiency. Antonio et al. [4] used Visual Basic code to establish a computer simulation for a real case of a combined-cycle gas turbine (CCGT), It is perceived that including a regenerator and a source of thermal energy running with solar energy can improve the efficiency of 2.25% to 3.29% and a lessening of gas consumption around 6.25% and 9.45%, It should be pointed out that on the other hand the net power of the proposed innovative cycle had been reduced with a mean value of 7.5%. A commercial simulator is used in [5] to predict the functioning at part-load conditions of a power plant with a combined cycle gas turbine (CCGT). It is observed that an increase in ambient temperature leads to lessen the power plant performance. Therefore author offered a laborious simulation based on a multivariable optimization method involving two major parameters fuel flow control (FFC) and inlet guide vane control (IGVC) to get a functioning approach that leads to enhanced efficiency. Zuming et al. [6] extend their study in [5] by using dynamic process simulation software instead of GateCycle to simulate the off-design operation of a triple-pressure reheat CCGT plant. The results show a good agreement between the two software. Whereas, the Aspen HYSYS offers an easy incorporation with several energy systems accompanied by a real-time simulation. Abigail et al. [7] assesses the result of ambient temperature and relative humidity on a natural gas combined cycle power plant according to Mexico environment, the results revealed that for augmenting ambient temperature from 15 °C to 45 °C both thermal efficiency and power decrease respectively from 50.8% to 48% and from 700MW to 530MW which in turn affect significantly the price of electricity from 52.58 $/MWh to 64 $/MWh. Ashley and Sarim [8] conducted an exhaustive study on the effect of the ambient temperature on the turbine power output, an empirical relationship between the ambient conditions in site Dubai, UAE and turbine power capacity was established. It is observed that thermal efficiency and power output decrease respectively with 0.1% and 1.47 MW. It is well known that a Heat recovery (HR) system incorporated in any thermal system will reduce greenhouse gas emissions and enhance the performance of the overall energy system, recently Bo et al. [9] established an innovative modeling outline incorporating both atmospheric dispersion modeling system (AERMOD) and computational fluid dynamics (CFD) model. The effect of HR on NOx was explored in a real case for a simple gas turbine cycle. The results show that maximum ground-level concentrations reached for a small stack exit temperature. Recently, some research results shown that applying steam injection on simple gas turbine cycle leads to augments both thermal efficiency and net work output however reduces the amount of CO2, therefore STIG process is considered environmentally friendly. Hasan and Yasin [10] studied parametrically STIG process to perform an adequate optimization for pertinent and specification parameters, it is found that all relevant parameters environmental, fuel price investment, and performance are strictly related to each other. The study in [3] was additionally examined experimentally in [11] to incorporate a hybrid cooling model, the calculated thermodynamics parameters and the experiment data are compared and displayed a good agreement. The main results prove that the angle nozzle meaningfully alters the performance of the gas turbine. Hasan and Yasin [12] develop an assessment method to incorporate together performance and environmental aspects for gas turbine cycles with chemical equilibrium. A multi-criterion optimization is implemented for variable pressure ratio and steam injection, their results revealed that for 5% of steam injection at fixed turbine inlet temperature and pressure ratio equal to 10 the thermal efficiency augments 4.3,% and at pressure ratio equal to 40 the increase of thermal efficiency will be 8.1%. Barinyima et al. [13] investigated numerically the performance of turboshaft engine cycles based on existing simple cycles; it was found that the thermal efficiency and specific fuel consumption are higher compared to traditional simple cycle engines. Sankaran and Christopher [14-15] perform an innovative method to maximize the gas turbine engine efficiency by employing an approach of architectures for heat- and work-regenerative engines by minimizing the irreversibility of the system based on optimization of the arrangements process and they used attractor-trajectory optimization procedure. Mohanad [16] did a detailed parametric study on the basic thermodynamics of gas turbine power plants incorporating reheating, cooling, and regeneration processes. It is perceived that the effectiveness of regenerator and intercooler augments the thermal efficiency as well as turbine and compressor efficiencies. Furthermore, lessening ambient temperature leads to reduced specific fuel consumption.

One of the most promising techniques used to enhance gas turbine and combined cycle power plants is inlet air cooling, it has been the subject of inspections in the last period. A detailed and comprehensive review of several inlet air cooling in gas turbine cycles besides their advantage and disadvantages have been presented by Ibrahim and Varnham [17]. An evaluation of the effect of two various approaches of inlet air cooling on the performance of a gas turbine combined cycle plant has been examined by Alok and Sanjay [18]. It has been perceived that an enhancement of efficiency and work output has been realized respectively around 4.88% and 14.77% when using the vapor compression cooling method. While using vapor absorption cooling this improvement augments efficiency and will be 17.2% whereas the decrease in work output will be 9.47%. Saleh et al. [19] reported that the optimum value of the turbine inlet temperature of the cooled air is 8 °C and the cooling capacity is 36 kW/m3 s−1, and for higher performance, it is recommended to use a sub-cooling multistage compressor system coupled with the wet condenser. A novel steam injection on gas turbine cycles has been introduced by Mahmoud and Hany [20] it is represented by a steam addition in the section after the combustion chamber exit and before the gas turbine inlet. The results revealed that for the proposed modified gas turbine cycle the efficiency and power output augment considerably which leads to reduce the specific cost. In order to assess the overall capability of inlet air cooling for gas–steam combined cycle power plant Cheng et al. [21] developed an analytical technique to analyze the efficiency ratio and relative payback period for such a system, The proposed cooling system based on absorption chiller and saturated evaporative cooler. The main results revealed that at ambient temperature between 15–20 °C fogging get more power efficiency compared to chilling. A comprehensive calculation of isentropic efficiency for the most important components in a gas turbine is presented in [22]. The isentropic efficiency of the compressor and the turbine are the main objectives of the analysis. Based on the numerical approach the simulation and modeling are liable to the pressure ratio and the turbine inlet temperature. The results show that the thermal efficiency is maximum for a pressure ratio of about 70. Ashkan et al. [23] studied numerically gas turbine and Stirling engine combined cycle and a separate gas turbine cycle, The modeling based on thermodynamic and economic investigation has for objective of minimizing electricity cost and energy losses and maximizing the overall performance. It is found that the proposed hybrid system (Stirling engine and combined cycle of gas turbine) leads to an important enhancement of efficiency of around 16.1% more than the traditional simple gas turbine cycle. Another thermodynamic performance exploration for a gas turbine cycle incorporated with a Stirling engine has been studied by Mahmood et al. [24], the most pertinent parameters selected in this study are length, diameter, and porosity of the regenerator with turbine inlet temperature and pressure ratio, it is observed that the efficiency of the overall system increase from 23.6 to 38.8%. Rui et al. [25] developed a computer code to analyze the effect of steam addition in a single annular combustor on the performance of a gas turbine, it is remarked that the new dimension of various mechanisms in the combustor must be considered greater than the case without steam injection. It is proved that an injection of 20% steam in the combustor needs a 10 % increase in the height of the combustor. Xiaoguo et al. [26] presented a novel technique to achieve higher performance of gas turbines designed for Building Cooling, Heating, and Power based on optimization of specification parameters. Results show that in the case of gas turbine efficiency among the two specific values, the gas turbine capacity can be determined by Following the Thermal Load (FTL). On the other hand, the gas turbine capacity can be dimensioned according to Following the Electricity Load (FEL).

Several researchers and engineers inspected thermodynamically the gas turbine cycle since the mid-twentieth century; Palmer [27] established a code structure for the calculation of the performance of a given thermodynamic cycle by an electronic digital computer. This study was further extended in [28] to include the Algol version, suitable for most of the larger computers in this period. In 1974, Macmillan [29] developed the work of Palmer and Annand [27, 28] using Fortran IV to predict the performance of a steady-state model for any gas turbine pattern. Furthermore, Shapiro and Caddy [30] established a new program, entitled NEPCOMP “Navy Engine Performance Program" which aims to calculate the thermodynamic cycle. In 1975, NASA [31] developed a computer code for exploring the performance of steady and non-steady gas turbine cycles. In 1985, the study in [27-28] was further extended in [32-33] based on the Newton-Raphson iteration method and developed a code for the investigation of any gas turbine engines.

Enrico [34] had done a detailed analytical study based on the general thermodynamic model for blade cooling, the established calculation estimate directly gas temperature over the channel and the flow rate of cooling air. Leszek and Monika [35] elaborated an analytical model based on energy and exergy analysis for gas microturbines. Results revealed that major exergy destruction and loss exist in the regenerator and combustion chamber. Kyoung et al. [36] develop a simplified analytical treatment to simulate and model the wet-compression process for Evaporative gas turbine cycles, the thermodynamic examination of the real-time heat and mass transfer procedures leads to observe that there is an optimum value of water injection ratio at which the compression and evaporation time are equals. Similarly to this study, Zheng et al. [37] prove the enhancement of efficiency for gas turbine cycles in cathe se of wet-compression process, a thermodynamic model was developed, and it is observed that in the case of the dry air compression process, the polytropic index is higher compared to wet compression process. Rahman et al. [38] offered a parametric study of gas turbine performance based on running parameters such as turbine inlet and exhaust temperature, compression ratio, and ambient temperature. Results exposed that ambient temperature and air-to-fuel ratio augment fuel consumption.

The previous detailed technical literature review confirmed that the gas turbine cycle technology is completely interesting and still receiving increasing attention since the mid-twentieth century until nowadays even after the spread of renewable energy technology, due to their potential for higher thermal efficiency, and greenhouse gas emission with low cost and investment. The majority of the investigations are numerically due to the complexity of phenomena that reigns the overall system and are accompanied by heat mass transfer complication, moreover, the experimental investigation is very expensive.

To the best of the author's apprehension, nobody has discussed the effect of pertinent parameters such as compression ratio rp and turbine inlet temperature TIT on the exergy and the performance of simple and regenerative gas turbine cycles. This paper presents a simple and general thermodynamic model for both simple and regenerative gas turbine cycles based on an analytical approach, therefore the results process are devoid of any numerical methodology permitting a simple and quick manipulation using a direct and exact equation can be helpful for design and direct calculation for specific parameters. The model forecasts the power output, and the thermal efficiency and can minimize the exergy loss of exhaust gasses. The analytical calculation is validated against some numerical examples and available data, and the agreement is observed to be fairly acceptable.

2. System configuration and basic assumption

A schematic diagram for the modeled simple and regenerative gas turbine power plant cycle is presented in Figures 1 and 2. The expression of the mathematical model for simple and regenerative gas turbine cycles is assumed in the succeeding assumptions:

- The specific heat of air and gasses is assumed as 1.005 kJ/kg.K and 1.14 kJ/kg.K.
- In both simple GT-Cycle and Regenerative GT-Cycle processes are assumed to be Ideal. This means pressure drop is negligible.
- The major variables are turbine inlet temperature and pressure ratio.

3. Thermodynamic model and methodology for simple gas turbine cycle

3.1. Analytical model

$$W_c = \frac{m_a.c_{pa}T_1}{\eta_c}\left(r_p^k - 1\right) = AT_1\left(r_p^k - 1\right) \qquad (1)$$

Where

$$k = \frac{\gamma - 1}{\gamma} \text{ and } A = \frac{m_a.c_{pa}}{\eta_c}$$

$$W_T = m_a.c_{pg}.T_3.\eta_t.\left(1 - r_p^{-k'}\right).\left\{\frac{LCV - c_{pa}.T_2}{LCV - c_{pg}.T_3}\right\}$$

$$= m_a.c_{pg}.T_3.\eta_t.\left(1 - r_p^{-k'}\right).\left\{\frac{LCV}{LCV - c_{pg}.T_3} - \frac{c_{pa}.T_1}{LCV - c_{pg}.T_3}.\left(1 + \frac{r_p^k - 1}{\eta_c}\right)\right\}$$

$$= m_a.c_{pg}.T_3.\eta_t.\left(1 - r_p^{-k'}\right).\left\{\frac{LCV - c_{pa}.T_1}{LCV - c_{pg}.T_3} - \frac{c_{pa}.T_1}{LCV - c_{pg}.T_3}.\left(\frac{r_p^k - 1}{\eta_c}\right)\right\} \qquad (2)$$

or

$$W_T = B.T_3.\left(1 - r_p^{-k'}\right).\left[C - D\left(r_p^k - 1\right)\right] \qquad (3)$$

Where

$$\left.\begin{aligned} k' &= \frac{\gamma' - 1}{\gamma'} \\ B &= m_a.c_{pg}.\eta_t \\ C &= \frac{LCV - c_{pa}.T_1}{LCV - c_{pg}.T_3} \\ D &= \frac{c_{pa}.T_1}{\eta_c.\left(LCV - c_{pg}.T_3\right)} \end{aligned}\right\}$$

and

$$W_{net} = W_T - W_c \qquad (4)$$

3.1.1. Analytical calculation for minimum network

For minimum W_{net}

$$W_T = W_c \qquad (5)$$

$$B.T_3.\left(1 - r_p^{-k'}\right).\left[C - D\left(r_p^k - 1\right)\right] = AT_1\left(r_p^k - 1\right) \qquad (6)$$

It is noted that for any value of T_1 and T_3 the "D' is much less than "C", therefore neglecting this term gives the approximate solution of r_p for W_{net} minimum

$$\Rightarrow B.C.T_3.\left(1 - r_p^{-k'}\right) \quad AT_1\left(r_p^k - 1\right) \qquad (7)$$

This gives

$$r_p^k + \alpha.r_p^{-k'} = 1 + \alpha \qquad (8)$$

Where

$$\left.\begin{aligned} \alpha &= C.\eta.c_p.T_r \\ c_p &= \left(c_{pg}/c_{pa}\right) \\ \eta &= \eta_c.\eta_T \\ T_r &= \left(T_3/T_1\right) = \left(T_{max.}/T_{min.}\right) \end{aligned}\right\} \qquad (9)$$

It is very clear that for any value of a, the first solution of r_p is one and the second solution depends on the value of a. For second root of this equation it is noted that there is mismatching between the graphical solution and the solution found by above equation, so in order to minimize the error between both results the above equation is modified as given below

$$r_p^k + \alpha . r_p^{-k'} = B + \alpha \quad (10)$$

Where

$$B = m_a . c_{pg} . \eta_t$$

3.1.2. Analytical calculation for maximum network

For maximum W_{net}

$$\frac{\partial W_{net}}{\partial r_p} = 0 \text{ and } \frac{\partial^2 W_{net}}{\partial r_p^2} = -ve$$

$$\frac{\partial W_{net}}{\partial r_p} = \frac{\partial}{\partial r_p}\left[W_T - W_c\right]$$

$$= \frac{\partial}{\partial r_p}\left[B.C.T_3.\left(1 - r_p^{-k'}\right) - AT_1\left(r_p^k - 1\right)\right] \quad (11)$$

A, B, C are independent of r_p therefore from $\frac{\partial W_{net}}{\partial r_p} = 0$

$$r_p^{k+k'} = C\eta . c_p . T . (\frac{k'}{k}) = \alpha . (\frac{k'}{k}) \quad (12)$$

$$r_p = \left(\alpha . \frac{k'}{k}\right)^{\frac{1}{k+k'}} \quad (13)$$

3.1.3. Analytical calculation for maximum thermal efficiency (n_{th})

$$\eta_{th} = \frac{W_{net}}{Q_s} \quad (14)$$

Let

$$\xi = 1 - \eta_{th} = \frac{Q_s - W_{net}}{Q_s} \quad (15)$$

$$W_{net} = m_g . c_{pg} . \left(T_3 - T_4\right) - m_a . c_{pa} . \left(T_2 - T_1\right) \quad (15)$$

$$Q_s = m_g c_{pg} . T_3 - m_a . c_{pa} . T_2 \quad (17)$$

$$Q_s - W_{net} = m_g c_{pg} . T_4 - m_a . c_{pa} . T_1 \quad (18)$$

$$\xi = \frac{m_g c_{pg} . T_4 - m_a . c_{pa} . T_1}{m_g c_{pg} . T_3 - m_a . c_{pa} . T_2} \quad (19)$$

$$\xi = \frac{\lambda . T_4 - T_1}{\lambda . T_3 - T_2} \quad (20)$$

Where $\lambda = C.c_p$

For minimum ξ

$$\frac{\partial \xi}{\partial r_p} = 0 \text{ and } \frac{\partial^2 \xi}{\partial r_p^2} > 0$$

Therefore

$$\frac{\partial \xi}{\partial r_p} = \frac{\lambda . \left(\lambda . T_3 - T\right) . \frac{\partial T_4}{\partial r_p} + \left(\lambda . T_4 - T_1\right) . \frac{\partial T_2}{\partial r_p}}{\left(\lambda . T_3 - T\right)^2} = 0 \quad (21)$$

This will be possible if

$$\Rightarrow \lambda . \left(\lambda . T_3 - T\right) . \frac{\partial T_4}{\partial r_p} + \left(\lambda . T_4 - T_1\right) . \frac{\partial T_2}{\partial r_p} = 0 \quad (22)$$

But $T_4 = T_3 . \left(1 - \eta_T\right) + T_3 . \eta_T . r_p^{-k'}$, therefore

$$\frac{\partial T_4}{\partial r_p} = -k' . T_3 . \eta_T . r_p^{-k'-1} \quad (23)$$

And

$$T_2 = T_1\left(1-\frac{1}{\eta_c}\right)+\frac{T_1.r_p^k}{\eta_c} \tag{24}$$

Therefore

$$\frac{\partial T_2}{\partial r_p} = \frac{T_1.r_p^{k-1}}{\eta_c} \tag{25}$$

$$\lambda.(\lambda.T_3 - T).(k'.T_3.\eta_T.r_p^{-k'-1}) = (\lambda.T_4 - T_1).\left(\frac{T_1.r_p^{k-1}}{\eta_c}\right) \tag{26}$$

$$\Rightarrow r_p^{k+k'} = \lambda\left(\frac{\lambda.T_3 - T}{\lambda.T_4 - T_1}\right).\left(\frac{k'}{k}\right).T_r.\eta \tag{27}$$

On solving further we get

$$r_p^{k''} + \phi.r_p^k = \psi \tag{28}$$

Where

$$\left.\begin{aligned}
k'' &= k + k' \\
C &= \frac{LCV - c_{pa}.T_1}{LCV - c_{pg}.T_3} \\
\lambda &= C.c_p \\
\phi &= \frac{\lambda.T_r.\eta_c + \beta/\eta_c}{\lambda.T_r.(1-\eta_c)-1} \\
\psi &= \frac{\lambda.T_r.\beta + \beta.\left(1/\eta_c - 1\right)}{\lambda.T_r.(1-\eta_c)-1} \\
\beta &= \lambda.\eta.T_r.\left(\frac{k'}{k}\right)
\end{aligned}\right\}$$

4. Model validation for simple gas turbine cycle

4.1. Given parameters

$T_1 = 300K \quad \eta_c = 0.85 \quad c_{pa} = 1.005kJ/kg.K$

$k = 0.285 \quad m_a = 1$

$T_3 = 1200K \quad \eta_t = 0.85 \quad c_{pg} = 1.14kJ/kg.K$

$k' = 0.248 \quad CV = 42000kJ/kg$

4.1.1. Numerical example of calculation for minimum network W_{net}

$$r_p^k + \alpha.r_p^{-k'} = B + \alpha$$

Where

$$\left.\begin{aligned}
\alpha &= C.\eta.c_p.T_r = 1.026\times1.134\times0.7225\times4 = 3.3625 \\
C &= \frac{LCV - c_{pa}.T_1}{LCV - c_{pg}.T_3} = \frac{42000-1.005\times300}{42000-1.14\times1200} = 1.026 \\
c_p &= \left(c_{pg}/c_{pa}\right) = \left(1.14/1.005\right) = 1.134 \\
\eta &= \eta_c.\eta_T = 0.85\times0.85 = 0.7225 \\
T_r &= \left(T_3/T_1\right) = \left(T_{max.}/T_{min.}\right) = \left(1200/300\right) = 4
\end{aligned}\right\}$$

$$r_p^{0.285} + 3.3625\times r_p^{-0.248} = 1\times1.14\times0.85 + 3.3625$$

On solving this equation we get its root as the value of r_p at which W_{net} become zero

$$\left.\begin{aligned} r_p &= 1.07 \\ r_p &= 49.88 \end{aligned}\right\}$$

Both values of r_p are very close to the graphical solution as shown in figure 1.

4.1.2. Numerical example of calculation for maximum network W_{net}

The value of at which is maximum is given by

$$r_p = \left(3.3625.\frac{0.248}{0.285}\right)^{\frac{1}{0.248+0.285}}$$

$$r_p = \left(3.3625\times\frac{0.248}{0.285}\right)^{\frac{1}{0.248+0.285}}$$

This gives r_p=7.49 . Again this value is very close to the graphical solution as shown in figure 3.

4.1.3. Numerical example of calculation for maximum thermal efficiency (n_{th})

The value of at which   is maximum is given by

$$r_p^{k''} + \phi . r_p^k = \psi$$

$$\left.\begin{aligned}
k'' &= 0.285 + 0.248 = 0.533 \\
C &= \frac{LCV - c_{pa}.T_1}{LCV - c_{pg}.T_3} = \frac{42000 - 1.005 \times 300}{42000 - 1.14 \times 1200} = 1.026 \\
\lambda &= C.c_p = 1.026 \times \frac{1.14}{1.005} = 1.164 \\
\phi &= \frac{\lambda . T_r . \eta_c + \beta / \eta_c}{\lambda . T_r . (1 - \eta_c) - 1} = \frac{1.164 \times 4 \times 0.85 + 2.927/0.85}{1.164 \times 4 \times (1 - 0.85) - 1} = -24.54 \\
\psi &= \frac{\lambda . T_r . \beta + \beta . \left(1/\eta_c - 1 \right)}{\lambda . T_r . (1 - \eta_c) - 1} = \frac{1.164 \times 4 \times 2.927 + 2.927 \times \left(1/0.85 - 1 \right)}{1.164 \times 4 \times (1 - 0.85) - 1} = -46.89 \\
\beta &= \lambda . \eta . T_r . \left(\frac{k'}{k} \right) = 1.164 \times (0.85 \times 0.85) \times 4 \times \frac{0.248}{0.285} = 2.927
\end{aligned}\right\}$$

After putting these values, the above equation is given below

$$r_p^{0.533} - 24.54 \times r_p^{0.285} = -46.89$$

This gives r_p=12.8 . Again this value is very close to the graphical solution as shown in figure 3.

Variations of bot thermal efficiency n_{th} and network W_{net} with compressor pressure ratio r_p for were depicted in Figure 3. It is perceived that the network and thermal efficiency for simple gas turbine cycle augment expressively for small range of pressure ratio until attainment a maximum value formerly starts lessening meaningfully for higher range of pressure ratio. It can be interpreted on this the existence of optimum values of pressure ratio for which the network and thermal efficiency are maximum, same results proved numerically by previous researches [9-16]. It is worthwhile to conclude that figure 3 validate the elaborated analytical model since that the optimum value for pressure ratio for both thermal efficiency and network are too close to graphical solution.

5. Thermodynamic model and methodology for regenerative gas turbine cycle

5.1. Analytical model

5.1.1. Pressure ratio optimization (Heat exchanger inefficient case)

The effectiveness (ε) of HE is defined as

$$\varepsilon = \frac{T_3 - T_2}{T_5 - T_2} \tag{29}$$

Therefore

$$\Rightarrow T_3 = T_2 + \varepsilon.\left(T_5 - T_2\right) \tag{30}$$

When heat exchanger (HE) become ineffective $T_3=T_2$ and this is possible when $T_5=T_2$

$$T_1\left\{1+\frac{r_p^k-1}{\eta_c}\right\} = T_4\left\{1-\eta_T\left(1-r_p^{-k'}\right)\right\} \tag{31}$$

$$\Rightarrow r_p^k + A_1.r_p^{-k'} = B_1 \tag{32}$$

Where

$$\left.\begin{aligned} A_1 &= -\eta.T_r \\ B_1 &= \left(1-\eta_c\right)+\eta_c\left(1-\eta_T\right).T_r \end{aligned}\right\}$$

5.1.2. Pressure ratio optimization (Heat exchanger with higher performance case)

$$T_3 = T_2 + \varepsilon.\left(T_5 - T_2\right) \tag{33}$$

But

$$T_2 = T_1\left(1-\frac{1}{\eta_c}\right)+\frac{T_1.r_p^k}{\eta_c} \tag{34}$$

And

$$T_5 = T_4.\left(1-\eta_T\right)+T_4.\eta_T.r_p^{-k'} \tag{35}$$

For maximum T_3 ,T_2 should be minimum and T_5 should be maximum, the minimum value of T_2 is T_1 and maximum value of T_5 is T_4 , this is possible at $r_p=1$, therefore the maximum possible value of T_3 is

$$\left(T_3\right)_{r_p=1} = T_1 + \varepsilon.\left(T_4 - T_1\right) = T_{\min.} + \varepsilon.\left(T_{\max.} - T_{\min.}\right) \tag{36}$$

5.1.3. Exergy analysis for pressure ratio optimization

The exergy loss of exhaust gasses will be minimum, when T_6 become minimum, the energy balance of H.E. is given below

$$m_a c_{pa}\left(T_3 - T_2\right) = \varepsilon.m_g c_{pg}\left(T_5 - T_6\right) \tag{37}$$

This gives

$$T_6 = T_5 - \frac{\left(T_3 - T_2\right)}{\varepsilon.\lambda} \tag{38}$$

This equation shows that T_6 become minimum, when T_3 become maximum, this means the minimum value of T_6 at $r_p=1$ and is given by

$$\left(T_6\right)_{r_p=1} = T_4 - \frac{\left(\left(T_3\right)_{r_p=1} - T_1\right)}{\varepsilon.\lambda} \tag{39}$$

6. Model validation for regenerative gas turbine cycle

6.1. Given parameters

$T_1 = 300K \quad \eta_c = 0.85 \quad c_{pa} = 1.005kJ/kg.K$

$k = 0.285 \quad m_a = 1$

$T_3 = 1200K \quad \eta_t = 0.85 \quad c_{pg} = 1.14kJ/kg.K$

$k' = 0.248 \quad CV = 42000kJ/kg$

6.1.1. Numerical example of calculation for (Heat exchanger inefficient case)

The condition of r_p at which heat exchanger become ineffective at particular cycle $T_{\max}$ and $T_{\min}$

$$r_p^k + A_1.r_p^{-k'} = B_1$$

Where

$$\left.\begin{aligned} A_1 &= -\eta.T_r = -0.85\times0.85\times4 = -2.89 \\ B_1 &= (1-\eta_c)+\eta_c(1-\eta_T).T_r = (1-0.85)+0.85\times(1-0.85)\times4 = 0.66 \end{aligned}\right\}$$

After putting the values of A_1 and B_1 the above equation reduces to

$$r_p^k - 2.89\times r_p^{-k'} = 0.66$$

This gives r_p=14.5 this value is close to the graphical solution as shown in figure 2.

It is noted that beyond r_p=14.5 the value of T_2 is more than the value of $T_5 > T_2$ this means that the heat is transfer from the compressed air to the exhaust gasses which results in increases in exergy loss of exhaust gasses and mass flow rate of flue in order to attain same peak cycle temperature *(T_{max})* of the cycle. In order to avoid this By-pass value is place in the path of exhaust gasses before the heat exchanger. When this By-Pass Value is open (BPV) allowing the exhaust gasses to pass through the heat exchanger if $T_5 > T_2$ and if $T_5 > T_2$ then by-pass valve is not allowing the exhaust gasses to pass through the heat exchanger and in that case exhaust gasses directly leave to the environment. This arrangement is shown in figure 5. The effect of this by-pass valve on the temperature after the heat exchanger and the thermal efficiency of the cycle is shown in figure 6 and figure 7 respectively.

Figure 6 illustrates the effect of compressor pressure ratio r_p on exit heat exchanger temperature T_3 for different value of turbine inlet temperature TIT=1000K, TIT=1200K and TIT=1400K, for two cases of bypass valve one open and the second closed. It is important to note that in case without bypass valve the exit heat exchanger temperature decreases significantly with compressor pressure ratio r_p, while it increase with turbine inlet temperature. Whereas, in second case with bypass valve the exit heat exchanger temperature increase meaningfully with compressor pressure ratio r_p but remain invariant when changing turbine inlet temperature TIT. It is worthwhile to note that in order to attain higher performance for the regenerative gas turbine cycle it required to use the bypass valve.

Figure 7 exhibits the effect of compressor pressure ratio r_p for TIT=1000K, TIT=1200K and TIT=1400K thermal efficiency of regenerative gas turbine cycle for two cases one with bypass valve and the second without the bypass valve . It is important to note that the thermal efficiency increase considerably for small range of pressure ratio r_p till maximum then decrease significantly until reach zero value. We realize the bypass valve improve substantially the thermal efficiency of regenerative gas turbine cycle.

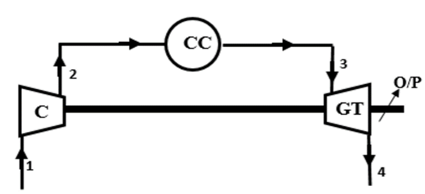

Figure 1: Schematic diagram of the simple gas turbine cycle

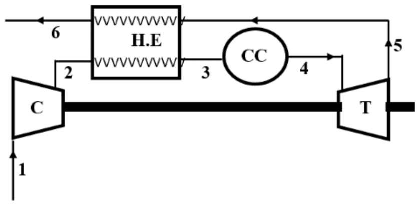

Figure 2: Schematic diagram of the regenerative gas turbine cycle

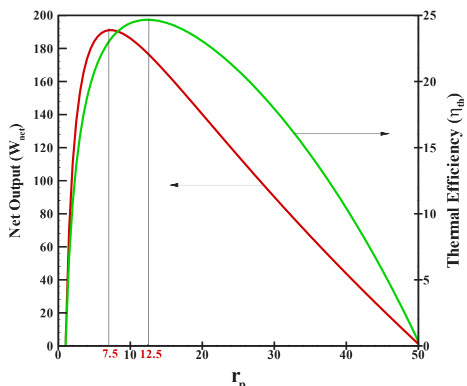

Figure 3: Network and thermal efficiency variation with pressure ratio (Model validation)

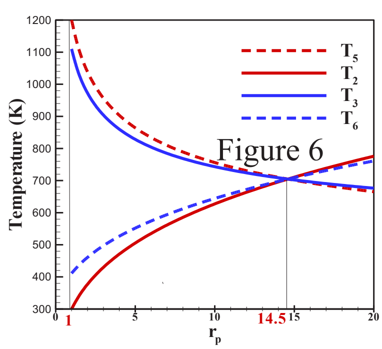

Figure 4: Variation of exit heat exchanger temperature with pressure ratio (Model validation)

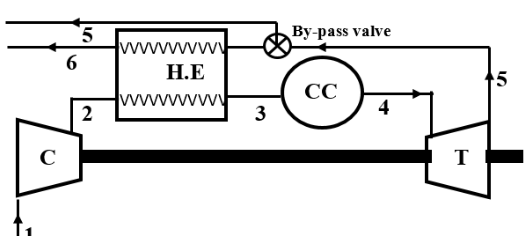

Figure 5: Schematic diagram of the modeled regenerative gas turbine cycle

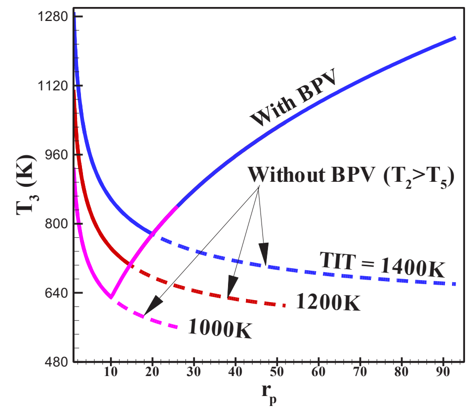

Figure 6: Exit heat exchanger temperature variation with pressure ratio

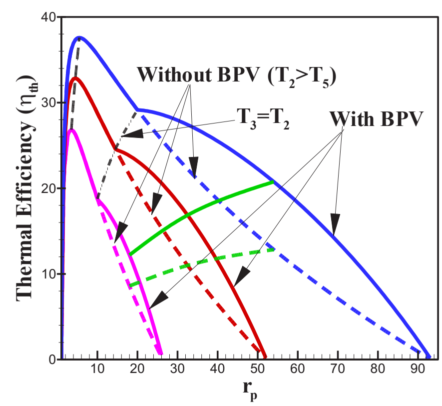

Figure 7: Thermal efficiency variation with pressure ratio

Nomenclature	
W	Power output (W)
h	Specific enthalpy (J/kg)
$\dot{m}$	Mass flow rate (kg/s)
q	Heat Supplied
η	efficiency,
	Effectiveness of HE
P	pressure
v	Specific Volume
HE	Heat Exchanger
HRSG	Heat Recovery Steam Generator
T	Temperature (K)
r_p	Pressure ratio
s	Specific entropy (J/kg K)
C_p	Specific Heat (kJ/kg.K)
LCV	Lower Calorific value of fuel

E	Exergy Loss (W)
TIT	Gas Turbine Inlet Temperature (K)
η	Efficiency
Subscripts	
c	compressor
tr	turbine
cc	Combustion Chamber
p	Pump
a	air
g	Exhaust gas
s	Steam
f	fuel
t	topping
b	bottoming
γ	Ratio of specific heat

7. Conclusion:

This work analyzes the energy and exergy performance of simple and regenerative gas turbine cycle using method. The following are the major conclusion.

- The pressure ratio and turbine inlet temperature significantly effect the network output and energy efficiency of both simple and regenerative gas turbine cycle.
- The numerical results for minimum as well as maximum network out and maximum efficiency are very close to the actual results, therefore the suggested equations could be use to findout the value of pressure ration for maximum efficiency and network output for simple GT-cycle.
- Bypass value valve (BPV) a significant role for the heat exchanger in case of regenerative GT-cycle.
- As the turbine inlet tempearture increases, the exit temperature of gasses from the turbine is higher than the exit temperature of compressed air from the air compressor for higher range of pressure ratio.
- With bypass value valve (BPV), the thermal efficiency is higher than the thermal efficiency without bypass value valve (BPV). This results in decrease in exergy losses by the exhaust gasses.

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