# Energy and Exergy Analysis of Brayton-Diesel Cycle

**Sanjay, Mukul Agarwal, Rajay**

*Abstract*-- **In this work the energy and exergy analysis of a hybrid gas turbine cycle has been presented. The thermodynamic characteristic of Brayton-diesel cycle is considered in order to establish its importance to future power generation markets. Mathematical modeling of Brayton-diesel cycle has been done at component level. Based on mathematical modeling, a computer code has been developed and the configuration has been subjected to thermodynamic analysis. Results show that, at any turbine inlet temperature (TIT) the plant specific work initially increases with increase of pressure ratio ($r_{p,c}$), and but at very high values of $r_{p,c}$, it starts decreasing. For a fixed value of $r_{p,c}$ (more than 10) with the increase in TIT, plant efficiency and specific work both increase. The cycle is best suited for applications where power requirement ranges between 700-900 kJ/kg. The exergy analysis shows that maximum exergy loss of around 27% occurs in during combustion in the plant.**

*Index Terms*— **Diesel cycle, exergy, exergy loss, gas turbine cycle, hybrid.**

## I. INTRODUCTION

With the increasing population of the world, the energy consumption will increase rapidly. While as the fossil fuels are depleting and one can imagine the future energy crisis unless some alternate cheap energy resources are developed. The increasing energy demand and depletion of fossil fuel resources inevitably necessitate for the optimum utilization of exhaustible fossil fuel and non-renewable energy resources. In this effort, a hybrid gas turbine based power cycle has been conceived for achieving maximum utilisation of thermal energy associated with the gas turbine exhaust.

In regard to the simple-cycle gas turbine technology, the major driver to enhance the engine performance has been the increase in process conditions (temperature and pressure) through advancements in materials and cooling methods. On-going development and near term introduction of advanced gas turbines has improved the efficiency of the simple-cycle operation more than 40%. Thermodynamic cycle developments, such as recuperation, mixed air steam turbines (MAST) are among the possible ways to improve the performance of gas turbine based power plants at feasible costs. Significant work in the field of advanced gas turbine based cycles has been done by Abdallah, H. and Harvey, S [1], Alabdoadaim, M et.al [2], Arrieta, F. R. P [3], Yadav R [4], Bianchi, M [5] , Chiesa. et al.[6] , Gabbrielli [7], Jonsson [8], Kuchonthara [9], Heppenstall[10], Horlock [11], Waldyr [12], Lukas [13], Poullikkas, A [14], etc. After reviewing recent technical papers by leading researchers Brayton-diesel cycle has been identified for detailed study.

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## II. NOMENCLATURE

| | | |
|---|---|---|
| $c_p$ | = | specific heat…………..(kJ-kg$^{-1}$K$^{-1}$) |
| gt | = | gas turbine |
| h | = | specific enthalpy………(kJ kg$^{-1}$) |
| $\Delta H_r$ | = | lower heating value……(kJ.kg$^{-1}$K$^{-1}$) |
| $\dot{m}$ | = | mass flow rate……….. (kg s$^{-1}$) |
| Q | = | heat added/ removed during process |
| $r_p$ | = | cycle pressure ratio |
| p | = | pressure………………..(bar) |
| T | = | temperature……………(K) |
| TIT | = | turbine inlet temperature (K) = combustor exit temperature |
| W | = | specific work………(kJ kg$^{-1}$) |

***Greek symbols***

| | | |
|---|---|---|
| ε | = | effectiveness ………..(%) |
| η | = | efficiency………………(%) |
| γ | = | ratio of specific heat at constant pressure and constant volume |
| $\upsilon$ | = | specific volume ………(m$^3$kg$^{-1}$) |

***Subscripts***

| | | |
|---|---|---|
| a | = | air , ambient |
| b | = | blade |
| c | = | compressor, coolant, |
| C | = | compression (Diesel Cycle) |
| comb | = | combustor |
| dc | = | diesel cycle |

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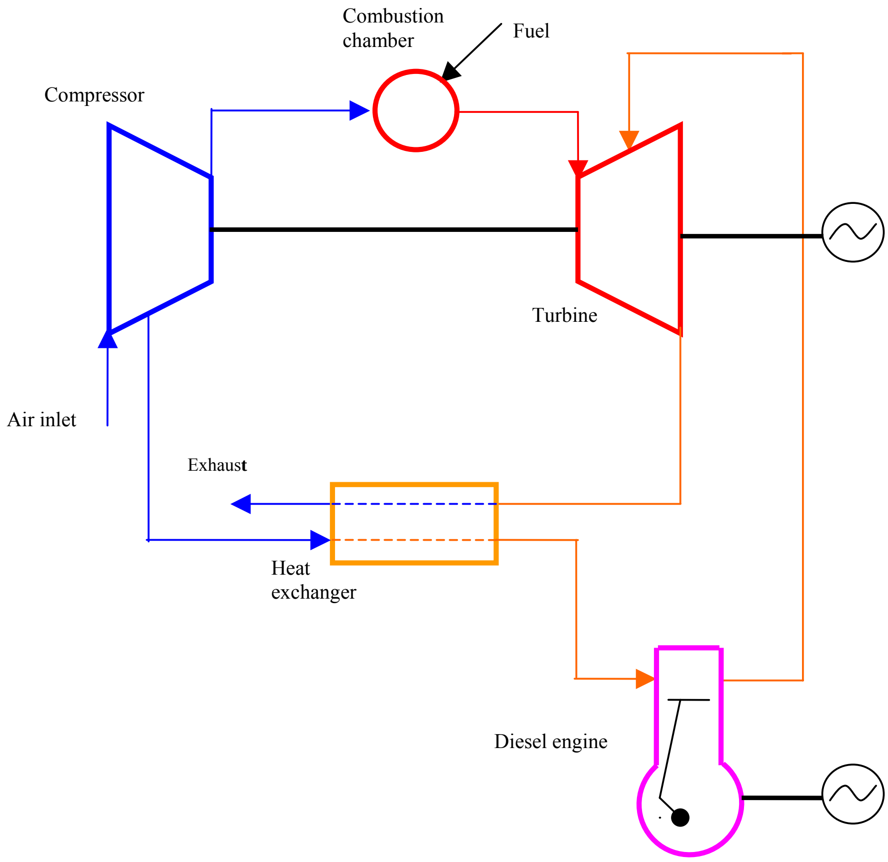


Fig. 1. Schematic of a Brayton-Diesel cycle

| | | |
|---|---|---|
| e | = | exit |
| E | = | expansion (Diesel Cycle) |
| f | = | fuel |
| g | = | gas |
| gen | = | alternator |
| gt | = | gas turbine |
| he | = | heat exchanger |
| i | = | inlet, stage of compressor |
| in | = | inlet |
| j | = | coolant bleed points |
| net | = | difference between two values |
| p | = | pressure |
| plant | = | brayton-diesel cycle plant |
| z | = | cooled row stage |

***Acronym***

| | | |
|---|---|---|
| C | = | Compressor |
| CC | = | Combustion chamber |
| GT | = | Gas turbine |

## III. BRAYTON-DIESEL- CYCLE CONFIGURATION

The configuration identified for parametric study is built up of various types of components. Preheating of the inlet air of a diesel engine can sufficiently improve its performance. The gas turbine exhaust can be applied in order to increase the temperature of the air, which is extracted from the compressor and fed into the diesel engine. Subsequently, the engine outlet flow expands through the low-pressure stage of the gas turbine as illustrated in Fig. 1. Main components of this cycle are gas turbine, gas-to-gas heat exchanger and diesel engine. The turbine exhaust is used to heat up the air bled from the compressor, which goes into diesel engine for further power generation. Modeling of all these components/elements that constitute the cycle configuration has been done. Modeling of these components are based on mass and energy balance across the control volume boundary of each of these components. Modeling the various elements of a cycle and derived governing equations is detailed in the following section.

### A. *Modeling of Components*

Modeling of various elements like working fluid/gas, compressor, combustor, and cooled gas turbine has been done in the earlier work [15,16,17]. Following are a few of the important equations of the model.

Enthalpy of air/gas, h = $\int_{T_o}^{T} c_p(T)\,dt$ (1)

Compressor work, $W_c$ =

$$\dot{m}_e h_e + \left(\sum m_{c,j} h_{c,j} - \dot{m}_{c,i} h_{c,i}\right) + \dot{m}_{dc} h_{dc} \quad (2)$$

Energy balance of combustor :

$$\dot{m}_f . \Delta H_r \eta_{comb} = (\dot{m}_{g,e} h_{g,e} - \dot{m}_{a.in} h_{a.in})_{comb} \quad (3)$$

The gas turbine work is the sum of the work done by all rows of bladings having open loop air-cooled blades.

$$W_{gt} = \Sigma \dot{m}_{g.i}(h_{g.a_z} - h_{g.b_z})_{cooled} + \Sigma \dot{m}_{g.i}(h_{g.i} - h_{g.e})_{uncooled} \quad (4)$$

where 'cooled' and 'uncooled' in the equation represent rows of blade requiring cooling and rows of blade not requiring cooling. Heat exchanger is used to transfer heat from the heated stream to the colder stream flowing through it. The energy balance equation of heat exchanger gives:

$$\dot{m}_h \cdot c_{p,h} \cdot \varepsilon_{he}\left[\left(T_{he,h}\right)_i - \left(T_{he,h}\right)_e\right] = \dot{m}_c \cdot c_{p,c} \cdot \left[\left(T_{he,c}\right)_i - \left(T_{he,c}\right)_e\right] \quad (5)$$

where 'h' stand for hot stream and 'c' stands for colder stream.

*B. Diesel Engine Cycle*

Diesel engine works on diesel cycle comprises of two isentropic, one isochoric and one isobaric process. For 1 kg of air in the cylinder, the efficiency analysis of the cycle can be made as given below.
The efficiency may be expressed in terms of any two of the following three ratios

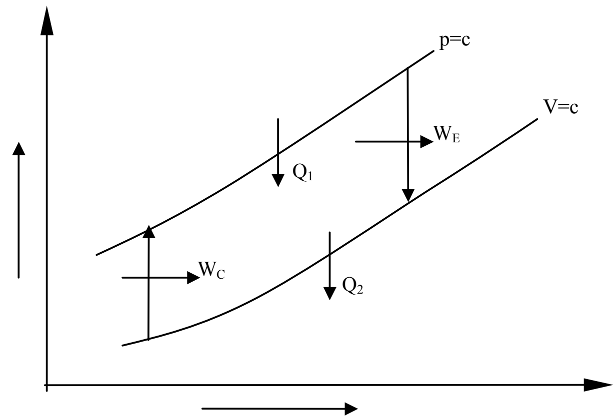


Fig. 2. T-s representation of ide     sel cycle

Compression ratio, $r_k = \dfrac{V_1}{V_2}$ (6)

Expansion ratio, $r_e = \dfrac{V_4}{V_3}$ (7)

Cut –off ratio, $r_c = \dfrac{V_3}{V_2}$ (8)

It is seen that $r_k = r_e . r_c$ (9)

Process 3-4 $$\frac{T_4}{T_3} = \left[\frac{V_3}{V_4}\right]^{\gamma-1} = \frac{1}{r_e^{\gamma-1}} \quad (10)$$

Process 2-3 $$\frac{T_2}{T_3} = \frac{p_2 V_2}{p_3 V_3} = \frac{V_2}{V_3} = \frac{1}{r_c} \quad (11)$$

Process 1-2 $$\frac{T_1}{T_2} = \left[\frac{V_2}{V_1}\right]^{\gamma-1} = \frac{1}{r_k^{\gamma-1}}$$

So

$$T_1 = T_2 . \frac{1}{r_k^{\gamma-1}} = \frac{T_3}{r_c} . \frac{1}{r_k^{\gamma-1}} \quad (12)$$

As we know that
Thermal efficiency

$$\eta = 1 - \frac{Q_2}{Q_1} = 1 - \frac{\dot{m} \cdot c_v (T_4 - T_1)}{\dot{m} \cdot c_P (T_3 - T_2)} = 1 - \frac{T_4 - T_1}{\gamma (T_3 - T_2)} \quad (13)$$

Now substituting the values from equation ( 7) , (8) , (9) , (10) in (13)

$$\eta_{Diesel} = 1 - \frac{1}{\gamma} . \frac{1}{r_k^{\gamma-1}} . \frac{r_c^{\gamma} - 1}{r_c - 1} \quad (14)$$

## IV. RESULTS AND DISCUSSIONS

For predicting the performance of the Brayton-diesel-cycle, a computer code based on the modeling of various cycle components discussed in the previous section has been developed.
For predicting the performance of the Brayton-diesel-cycle, mathematical modeling of various cycle components has been discussed in previous section. Based on this modeling a computer code in C++ language has been developed. Results have been obtained for input data listed in Table I. and the results obtained have been plotted using graphic package ORIGIN 6.0. Based on results of exergy analysis, a Sankey diagram has been drawn for the cycle. The exergy distribution quantifies the losses in various elements of the cycle. The results (Design Monograms and Sankey diagrams) have been discussed.

Table I. Input data for analysis for Brayton-Diesel Cycle

| Component | Parameters |
|---|---|
| Gas property | $c_p$= f(T)<br>Enthalpy h= $\int c_p(T)\,dT$ |
| Ambient condition | $T_a$=288K<br>$P_a$=1.013 bar<br>Relative humidity=60% |
| Inlet section | $\Delta p_{loss}$=1 percent of entry pressure |
| Compressor | Isentropic efficiency ($\eta_c$)=86%<br>Mechanical efficiency ($\eta_m$)=98% |
| Combustor | $\Delta p_{loss}$=2% of entry pressure<br>$\eta_{cc}$=98% |
| Fuel | Natural gas<br>$(LCV)_f$=42000 kJ/kg<br>Fuel inlet pressure = 110% of compressor exit pressure |
| Gas turbine | Isentropic efficiency ($\eta_t$)=86%<br>Mechanical efficiency($\eta_m$)=98%<br>Exhaust pressure=1.08 bar |
| Heat exchanger | $\Delta p_{loss}$=2% of entry pressure<br>Effectiveness($\varepsilon_{rec}$)=92%<br>Temperature gain in Heat exchanger=100K |
| Diesel engine | Compression ratio=5<br>Cut off ratio=2.5 |
| Alternator | Efficiency ($\eta_{alt}$)=98.5% |

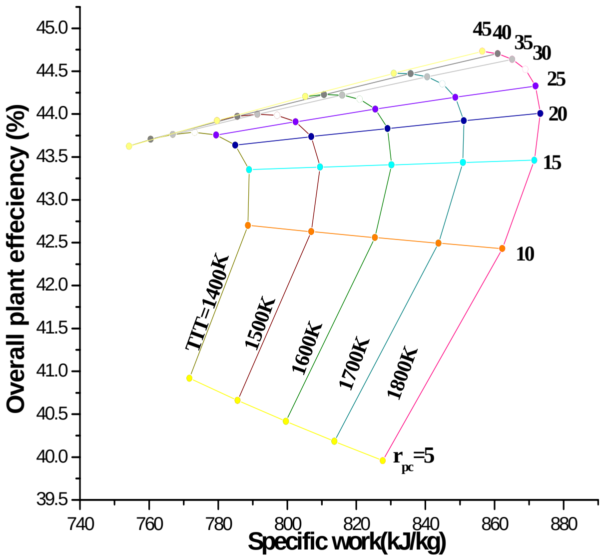


Fig. 3. Influence of TIT and $r_{p,c}$ on plant efficiency and specific work Brayton-Diesel cycle

Fig. 3. shows the variation of specific work and plant efficiency with pressure ratio ($r_{p,c}$) and turbine inlet temperature (TIT).

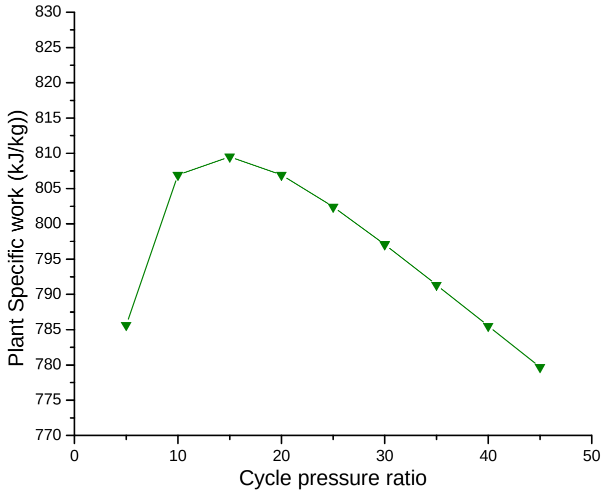


Fig. 4. Effect of $r_{p,c}$ on plant specific work at TIT=1500K

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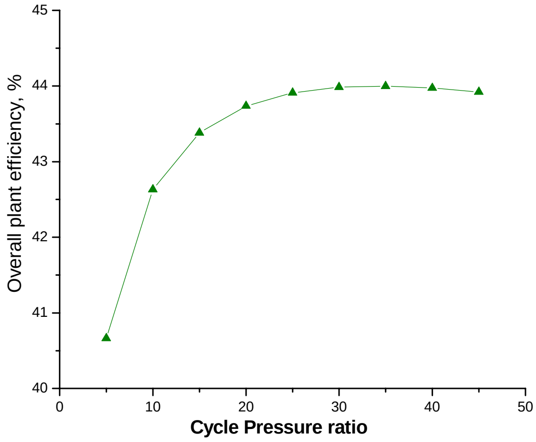


Fig. 5 Effect of $r_{p,c}$ on plant efficiency at TIT=1500K

Results show that at TIT less than 1600K and upto $r_{p,c}$=15, the specific work increases with increase in $r_{p,c}$. At values of TIT higher than 1600K, specific work increases with TIT, till value of $r_{p,c}$ is less than 20, beyond which specific work reduces with increase in TIT. At any TIT value of the plant efficiency increases with increase in $r_{p,c}$ upto a certain value, when after the plant efficiency curve loops back. This design monogram can be used to select the operating parameters for the brayton-diesel cycle plant depending on required plant capacity.
Most of the gas turbines today operate at a TIT of 1500K, hence performance of the configuration at this value of TIT is shown in Fig. 4 and Fig. 5. Fig. 4 shows the effect of pressure ratio on plant specific work for the configuration. The graph shows the range of pressure ratio over which the cycle can operates and also the optimized value of specific work for the selected TIT. Fig. 5 shows the effect of

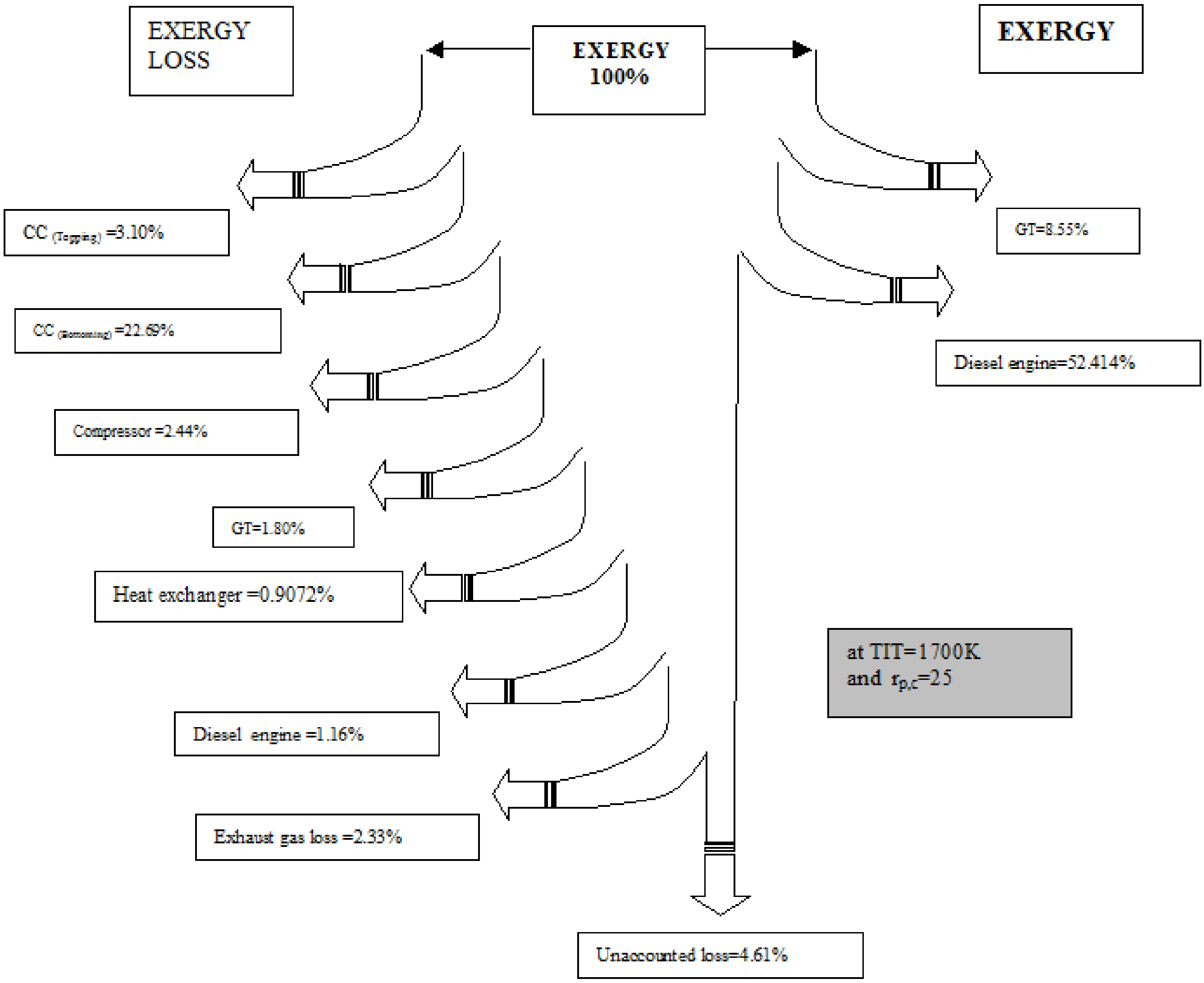


Fig. 6. Exergy flow diagram for Brayton-Diesel cycle

pressure ratio on plant efficiency for the configuration. The graph shows the range of pressure ratio over which the cycle operates and also the optimized value of plant efficiency for the selected TIT.

Fig. 6. shows the exergy flow at $r_{p,c}$ =25 and TIT=1700K among various components of the configuration. Diesel engine has the exergy of about 52.4% while gas turbine exergy is about 8.55% because the bleeding from the compressor has been done at 5 bar, so mass flow of working fluid through the diesel engine is very large in comparison of gas turbine. Exergy loss in combustion chamber of Brayton cycle is about 3.10%, because in turbine working fluid handled is lesser than in Diesel engine. Also the combustion process in Diesel engine exhibits an exergy loss of about 22.70%. This configuration has lower exhaust gas exergy loss (about 2.33%), because the exhaust gas heat has been recovered considerably well in the heat exchanger at the exit of turbine. Unaccounted exergy losses are about 4.61%.

## V. CONCLUSIONS

The brayton-diesel cycle is a gas turbine hybrid cycle and its thermodynamic analysis has been carried out to ascertain its potential as a energy conversion system. From the analysis following conclusions can be drawn:

1) Brayton-Diesel cycle provides the maximum available energy (about 60%)
2) Braton-Diesel cycle gives maximum work output between 840-875 kJ/kg.

Thus the cycle is found to be useful as a energy conversion cycle and performance in the above mentioned range.

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