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Thermodynamic analysis of a twin cylinder CI engine operated on diesel fuel

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Abstract: At the present scenario there is a need of thermodynamic analysis of compression ignition (CI) engine to evaluate its energy utilization efficiency, which will locate and quantify the amount of quality energy lost from the engine at different heads. In this paper, the analysis of energy, exergy, maximum combustion temperature and exhaust gas temperature of a CI engine using diesel is performed. The results indicate that brake thermal efficiency, exergy efficiency, mean gas temperature increases with increase in load and relatively brake specific fuel consumption decreases. At the same time destruction of exergy also increases with engine load, this indicates a potential of waste heat recovery from the CI engine to improve its overall efficiency.
Keywords: Exergy, Energy, Exergy Efficiency, Exergy Losses, Exergy Destruction

I. INTRODUCTION

Energy is a vital part of the economic growth of a country. It enhances economic prosperity, personal comfort and quality of life. India being a developing country requires very large amount of energy to maintain its rate of progress. At present, India is world's eleventh largest energy producer and stands as the sixth largest energy user in global scenario [1]. A large portion of power requirements are obtained using diesel engines. The net brake thermal efficiency of compression ignition engines is difficult to be higher than 45% [2]. A large amount of fuel energy is rejected from the engine to the surroundings in form of exhaust heat, heat lost to the cooling water, heat lost through the engine walls and energy lost with the unburned fuel emissions. Most of the researchers have studied the performance of internal combustion (IC) engines based on brake thermal efficiency (BTE), brake specific fuel consumption (BSFC), brake specific energy consumption (BSEC), indicated power (IP), brake power (BP) and emissions from the engine. These analyses have overlooked the quality of energy lost to the surroundings in form of waste heat [3, 4]. Exergy analysis overcomes the restrictions of the first law of thermodynamics and is based on both the first and second laws of thermodynamics [5]. The exergy of a system is defined as the amount of mechanical work that can be maximally extracted from the system, till the system reaches thermodynamic equilibrium with its surrounding. Exergy analysis measures the deviation of actual performance of the system from ideal performance. It indicates that, the maximum potential for a system to perform work is a function of its internal energy and the ambient conditions [6]. Exergy analysis applied to a system determines the locations, types and magnitude of losses associated with a system which can lead in design of an improved system to reduce inefficiency. This increased energy efficiency benefits in reduction in consumption of energy source and emission. The importance of exergy analysis leads in better analysis of resources to utilise in an efficient way. In this paper, analysis of energy, exergy efficiency and destruction of exergy of a twin cylinder four stroke CI engine was performed.

II. THERMODYNAMIC ANALYSIS OF THE ENGINE

The thermodynamic analysis (energy and exergy) is getting more attention day by day to evaluate and manage the performance of thermal energy system especially IC engines. The IC engine energy balance is also called thermal balance, which deals with IC engine energy distribution based on system integration [7]. Alasfour (1997) carried out an exergy analysis of a petrol engine to consider the impact of butanol-gasoline blends on engine efficiency [8]. Debnath et al. carried out the energy and exergy analysis of a single cylinder diesel engine and concluded that the fuel energy recover can be improved from 26% to 30% and the shaft availability is increased by increase in compression ratio [9].

A. *Energy and Exergy Balance System and Equations*

One of the commonly used methods to construct energy balance system including different energies involved in IC engine operation. Then the magnitude of all these energies can further analyzed for each sub systems. By optimizing the energy distribution of all the systems, losses can be minimized. The total fuel chemical energy can be distributed as useful work, heat transfer loss, frictional losses, exhaust losses, heat energy dissipated to cooling water and chemical energy loss from unburned emissions [10]. The control volume analysis of IC engine energy balance is shown in Fig 1. Out of different types of energy involved in the control volume analysis the incoming energy to the control volume comprises fuel chemical energy and intake

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enthalpy. Similarly outflow of energy from control volume consists of effective output power, exhaust gas and cooling water energy, partially burned fuel energy and heat lost from the engine surface by convection and radiation. The second law analysis is carried out by considering the engine as a heat engine which operates between two temperature reservoirs, the combustion temperature (T_H) and the ambient temperature (T_L) as shown in Fig 2.

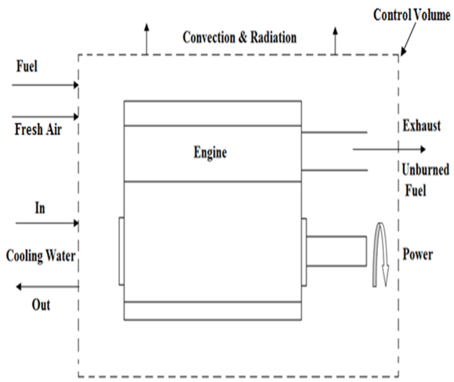

Fig 1. Internal Combustion engine energy.

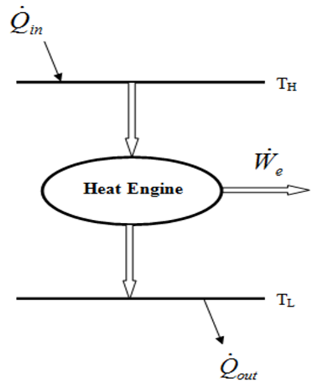

Fig 2. Heat engine balance system.

The following assumptions were considered during the analysis of IC engine energy balance,

A. The diesel engine works in a steady state condition.
B. Ideal gas principles are applied for fresh air intake and exhaust gases.
C. The kinetic and potential energy changes are neglected.
D. The temperature and pressure of dead (environmental) state are taken as the actual ambient conditions.
E. The state of water in the exhaust is generally vapour in IC engines, hence the lower calorific value (LCV) of the fuel is considered.

The equation for energy balance represents the relationship between different types of energy involved such as the chemical energy supplied to the engine $\dot{Q}_{in}$ is calculated from the mass flow rate of fuel and lower calorific value of fuel.

$$\dot{Q}_{in} = \dot{m}LCV \qquad (1)$$

Various approximations for the chemical exergy of fossil, liquid and gaseous fuels are presented by the researchers. Based on the work of Szargut and Styrylska an approximation of ratio of the fuel chemical exergy to the low calorific value $\left(\frac{\psi_{fch}}{LCV}\right)$ at atmospheric pressure of 1.01325 bar and temperature 300 K for diesel fuel ($C_{14.4}H_{24.9}$) is 1.0699, where ψ_{fch} is the input fuel chemical exergy [11].

One part of the heat supplied $\dot{Q}_{in}$ is converted into useful work $\dot{W}_e$ (engine loads) and the other part $\dot{Q}_{out}$ is rejected to the environment. By applying the first law of thermodynamics to this heat engine system, we obtain:

$$\dot{Q}_{in} = \dot{W}_e + \dot{Q}_{out} \qquad (2)$$

$\dot{W}_e = \frac{2\pi\tau N}{60000}$ = Rate of energy converted to useful work i.e engine load (i.e measured using eddy current dynamometer) in kW where $\tau = Torque = W.r$

W = Load applied by dynamometer in Newton.

r = Dynamometer arm length (m)

N= speed of the engine in rpm

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$\dot{Q}_{out}$= Rate of energy rejected to the environment, kW

Equation (2) determines the quantity of the power concerned in the system; however the quality of power can be analyzed from the second law of thermodynamics:

$$\frac{\dot{Q}_{in}}{T_H} - \frac{\dot{Q}_{out}}{T_L} + \dot{S}_{gen} = 0 \qquad (3)$$

Where, $\dot{S}_{gen}$= The rate of entropy generation relative to environment.

Equation (2) and (3) follows the exergy (E_x) function that gives information for both the quantity of the power available and their quality. The exergy considered here indicates the maximum recoverable energy after combustion. For the sake of simplicity chemical exergy is not taken into account. The exergy consumption accounting equation for engine is given by:

$$E_X = \dot{Q}_{in}\left(1 - \frac{T_L}{T_H}\right) = \dot{W}_e + \delta E_X \qquad (4)$$

Where,

E_x= Exergy consumption rate, kW

δE_x= The rate of destruction in exergy, kW.

The destruction of exergy is calculated using equation (4) by putting the value of input energy, cold side and hot side temperature and rate of energy converted to electric power. The Gouy-Stodola version of destruction of exergy is given by:

$$\delta E_X = T_L \dot{S}_{gen} \qquad (5)$$

The brake thermal efficiency (η_e) of the current system which is the ratio of rate of energy converted to electric power to the input fuel energy rate and is given by:

$$\eta_e = \frac{\dot{W}_e}{\dot{Q}_{in}} \qquad (6)$$

The exergy efficiency (η_{ex}) is the ratio of rate of energy converted to electric power to the exergy consumption rate and is given by:

$$\eta_{ex} = \frac{\dot{W}_e}{E_X} \qquad (7)$$

From equation (4), the determination of the combustion chamber temperature is necessary to evaluate the exergy consumption and thus the exergy efficiency.

$\dot{Q}_{out}$ is given by:

$$\dot{Q}_{out} = \dot{Q}_1 + \dot{Q}_2 + \dot{Q}_3 \qquad (8)$$

Where $\dot{Q}_1$ is the rate of heat energy taken away by cooling water in kW, $\dot{Q}_2$ is the rate of heat taken away by exhaust gas in kW and $\dot{Q}_3$ is the unaccounted loss of heat energy in kW.

$$\dot{Q}_1 = \dot{m}_w C_w \left(T_{2w} - T_{1w}\right) \qquad (9)$$

Where $\dot{m}_w$ is the mass flow rate of cooling water in Kg/s, C_w is the specific heat capacity of cooling water, T_{2w} is the cooling water temperature at outlet in K, T_{1w} is the cooling water temperature at inlet in K.

The rate of heat energy taken away by the exhaust gas can be written as the difference between exhaust gas enthalpy and intake gas enthalpy.

$$\dot{Q}_2 = \dot{H}_2 - \dot{H}_1 \quad (10)$$

Where $\dot{H}_1$ and $\dot{H}_2$ are rate of inlet gas enthalpy and rate of exhaust gas enthalpy respectively in kW.

The calculation formula of rate of exhaust gas enthalpy rate is:

$$\dot{H}_2 = \left(\dot{m}_f + \dot{m}_a\right) C_{pex} T_{ex} \quad (11)$$

Where,

$\dot{m}_f$ = mass flow rate of fuel, kg/s

$\dot{m}_a$ = mass flow rate of intake gas, kg/s

C_{p_ex} = Specific heat capacity of the exhaust gas,

T_{ex} = Exhaust gas temperature, K

The calculation formula of rate of intake gas enthalpy rate is:

$$\dot{H}_1 = \left(\dot{m}_f + \dot{m}_a\right) C_{pin} T_L \quad (12)$$

Where,

C_{p_in} = Specific heat capacity of the intake gas

T_L = The cold side at ambient temperature in K

The brake specific fuel consumption (BSFC) kg/kWh is given by:

$$BSFC = \frac{\dot{m}_f}{\dot{W}_e} \quad (13)$$

The percentage of each kind of energy in terms of total energy is given by:

$$\eta_f = \eta_e + \eta_w + \eta_{exh} + \eta_u = 100\% \quad (14)$$

η_f = Total percentage of fuel chemical energy

η_w is the percentage of cooling water energy in total fuel energy and it is calculated by:

$$\eta_w = \frac{\dot{Q}_1}{\dot{Q}_{in}} \quad (15)$$

The percentage of exhaust energy in total fuel energy (η_{exh}) is given by:

$$\eta_{exh} = \frac{\dot{Q}_2}{\dot{Q}_{in}} \quad (16)$$

The percentage of unaccounted loss of energy in total fuel energy (η_u) is calculated by:

$$\eta_u = \frac{\dot{Q}_3}{\dot{Q}_{in}} \quad (17)$$

According to the second law of thermodynamics the evaluation indicator of energy is not only the quantity but also the quality. Waste heat energy is a kind of low-grade energy compared to effective mechanical energy since it requires various kinds of thermodynamic cycles to be converted into effective work. In accordance with Carnot principle, the maximum efficiency of thermodynamic cycles cannot be higher than Carnot cycle efficiency. Thus, only part of waste heat energy can be recovered. Therefore exergy analysis is more useful to evaluate the total heat energy characteristics. Exergy is also called available energy and is used to evaluate the energy quality.

B. Determination of Maximum Combustion Temperature

The maximum combustion temperature T_H is calculated from the measured exhaust temperature. The engine real cycle is approximated to be an ideal diesel cycle. The P-V and T-S diagram for ideal diesel cycle is shown in Fig 3 & Fig 4 respectively.

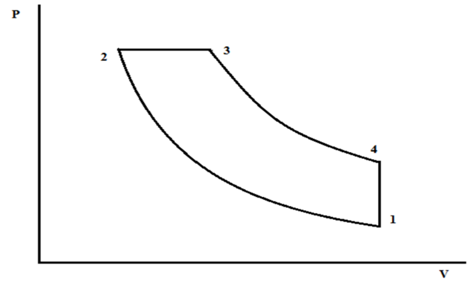

Fig 3. P-V plot of ideal diesel cycle.

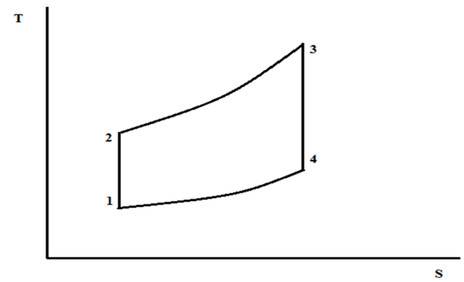

Fig 4. T-S plot of ideal diesel cycle.

T_3 represents the combustion temperature T_H and T_4 represents the exhaust gas temperature. Due to the isentropic expansion of the gas (considered perfect gas) between states 3 and 4, the relationship between T_3 and T_4 is established as:

$$T_3 = T_4 \left(\frac{r_c}{r_{of}} \right)^{\gamma - 1} \qquad (18)$$

γ is the specific heat ratio (C_p/C_v) and its value is 1.4, since the gas considered is perfect. r_c is the engine compression ratio, which is the ratio of the maximum volume (V_1) to the minimum volume (V_2) formed in the engine cylinder.

$$r_c = \frac{V_1}{V_2} \qquad (19)$$

r_{of} is the cutoff ratio of the engine given by:

$$r_{of} = \frac{V_3}{V_2} \qquad (20)$$

where V_3 and V_2 are the engine cylinder volumes after and before combustion respectively.
The cutoff ratio varies with respect to brake thermal efficiency(η_e) as follows:

$$\eta_e = 1 + \frac{1 - r_{of}^{\ \gamma}}{r_c^{\gamma - 1} \gamma \left(r_{of} - 1 \right)} \qquad (21)$$

As the values of T_4 and η_e are determined experimentally and the values of r_c and γ are known, r_{of} and T_3 (T_H, combustion chamber temperature) can be calculated using equations 18 and 21.

III. EXPERIMENTATION

The aim of the study is to perform a detailed analysis of a compression ignition engine regarding energy utilization, available energy or exergy and waste heat recovery potential using diesel.

A. Experimental Setup

The schematic diagram of the engine setup is shown in Fig 5. The engine used in the experimental investigation was a twin cylinder, 4-stroke, 14 hp, 1500 rpm compression ignition engine. The engine was coupled with a three phase, 415 volt AC alternator with loading circuit was developed by designing a suitable load cell for measurement of applied load on the engine. ‘K’ type thermocouples were used to measure gas temperature at the engine exhaust, engine cooling water outlet and ambient temperature. The fuel flow was measured by the use of 50 ml burette and stopwatch with level sensors. Instruments fitted to the test bed are properly calibrated to minimize the possible errors during experimentation. A computerized data acquisition system was used to collect, store and analyse the data during the experiment by using various sensors.

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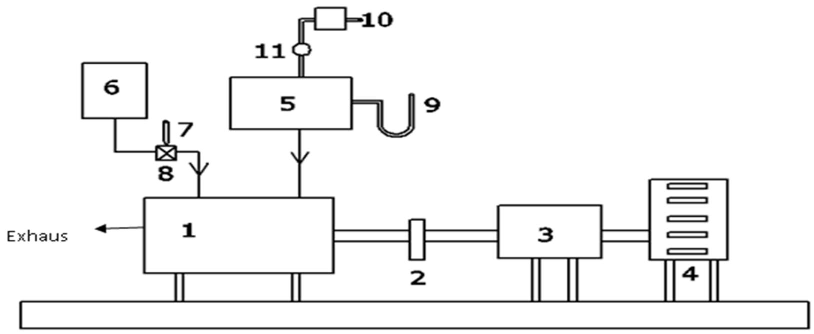

Fig 5. Schematic diagram of the engine setup

1. Engine
2. Coupling
3. Alternator
4. Electrical Loading Unit
5. Air box
6. Fuel Tank
7. Burette
8. Valve
9. Manometer
10. Air Filter
11. Orifice Meter

B. Experimental Procedure

The engine was started using standard diesel runs for 15 minutes at the rated speed of 1500 rpm. When the engine was warmed up and reached the steady state, readings are taken thrice at each load and the mean value was considered for calculation.

IV. RESULTS AND DISCUSSION

During theoretical analysis, the key aspects of various kinds of energy, its quantity, quality and various kinds of losses are illustrated, which shows how energy is consumed during energy conversion and heat transfer processes. The exergy analysis is made to locate the area and maximum potential of energy to perform useful work.

A. Brake Specific Fuel Consumption

Variation of brake specific fuel consumption with engine load is illustrated in Fig 6. The BSFC decreases with increase in load. The main reason for this could be that at higher engine loads the percentage increase in fuel consumption for the engine is less than the increase in brake power. On the other hand with increase in load combustion chamber pressure and temperature increases, this leads to better atomization and combustion of fuel, resulting decrease in BSFC.

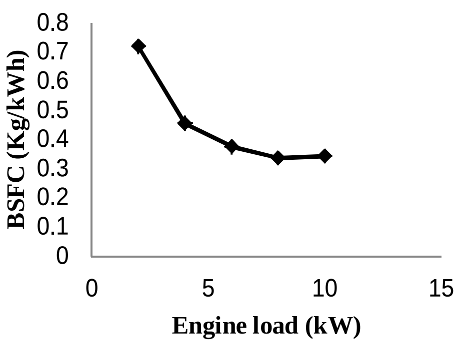

Fig 6. Variation of bsfc with engine load.

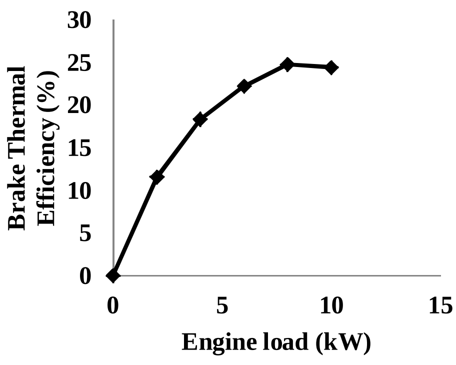

Fig 7. Variation of brake thermal efficiency with engine load.

B. Brake Thermal Efficiency

Fig 7 shows the variation of brake thermal efficiency of the engine with respect to engine load. Brake thermal efficiency increases with increase in engine load. The main reason for this could be relatively lower heat loss and higher power output at higher engine loads. At higher load the increase in average gas temperature have the effect of reducing the fuel ignition delay and because of better mixing and spray characteristics of fuel at higher temperature, leads to increase in brake thermal

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efficiency.

C. Exergy Efficiency

Variation of exergy efficiency with engine load is illustrated in Fig 8. The exergy efficiency increases with increase in load. This can be referred to increase in in-cylinder temperature leading to better atomization and combustion of fuel, resulting in increase in useful work and comparatively lower fuel consumption at higher loads.

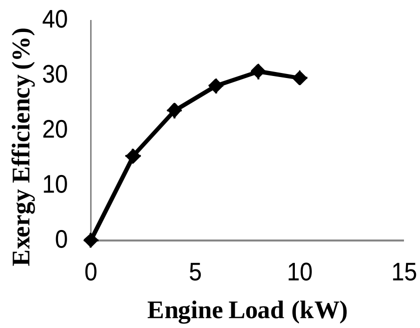

Fig 8. Variation of exergy efficiency with engine load.

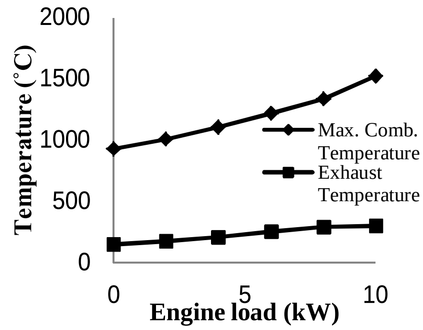

Fig 9. Variation of temperature with engine load.

D. Maximum Combustion Temperature And Exhaust Gas Temperature

Fig 9 shows the variation of maximum combustion temperature and exhaust gas temperature with engine load. Maximum combustion temperature increases with increase in load; this can be referred to increase in heat release rate due to increase in fuel supply at higher loads. It is observed that the exhaust temperature also increases with increase in engine load. It is due to the fact that at higher loads there is increase in fuel richness and slightly combustion progresses to the afterburning stage, leading to increase in exhaust temperature.

E. Exergy Destruction

The destruction of exergy is caused by irreversibility due to combustion of fuel, frictional loss and by heat loss through the engine exhaust, cylinder wall and cooling water. Fig 10 shows the variation of destruction of exergy for various loads. From the figure it can be observed that the destruction of exergy increases with increase in load. At higher loads there is an increase in fuel richness and ignition delay causing more energy and heat transfer loss.

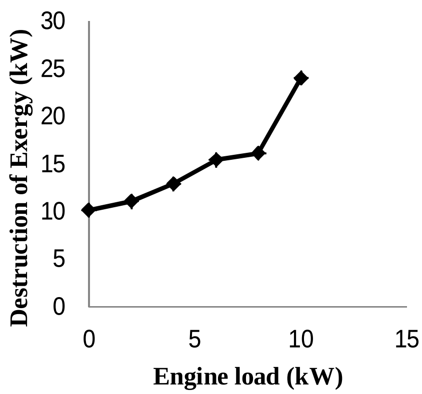

Fig 10. Variation of destruction of exergy with engine load.

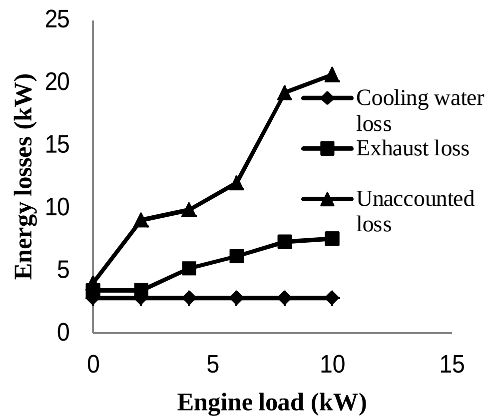

Fig 11. Comparison of energy losses with engine load.

F. Energy Losses

A comparison energy loss with engine load is shown in Fig 11. Heat energy taken by the cooling water remains same at all loads, as flow of cooling water was kept constant. Heat energy lost away through exhaust increases with load indicated by the increase in exhaust gas temperature. Unaccounted loss increases at a higher rate with increase in load compared to other losses.

G. Energy Balance Analysis

Fig 12 shows the percentages of various kinds of energy for the fuels analyzed at full load. The total energy includes useful energy, cooling water loss, exhaust gas loss and unaccounted loss. The unaccounted loss is obtained by subtracting useful energy or brake power, exhaust energy losses and cooling water losses from the input energy. At full load percentage of cooling water loss is found to be 6.8%, percentage of exhaust energy loss is 18.45% and percentage of unaccounted loss is found to be 50.33%.

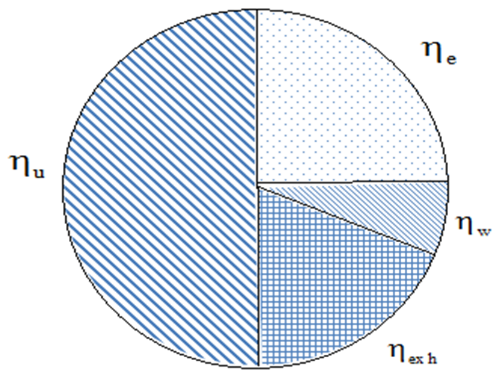

Figure 12. Energy distribution chart at full load.

V. CONCLUSIONS

From the study it was observed that brake thermal efficiency, exergy efficiency increases with increase in load with a relative reduction in brake specific fuel consumption. Increase in mean gas temperature with load indicates the improvement in conversion of fuel energy to heat energy. At the same time increase in destruction of exergy with increase in load indicates the losses in energy due to energy conversion reaction and heat transfer.

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