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Effect of hydrogen-diesel quantity variation on brake thermal efficiency of a dual fuelled diesel engine

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Abstract

The twenty first century could well see the rise of hydrogen as a gaseous fuel, due to it being both environment friendly and having a huge energy potential. In this paper, experiments are performed in a compression ignition diesel engine with dual fuel mode. Diesel and hydrogen are used as pilot liquid and primary gaseous fuel, respectively. The objective of this study is to find out the specific composition of diesel and hydrogen for maximum brake thermal efficiency at five different loading conditions (20%, 40%, 60%, 80% and 100% of full load) individually on the basis of maximum diesel substitution rate. At the same time, the effects on brake specific fuel consumption, brake specific energy consumption, volumetric efficiency and exhaust gas temperature are also observed at various liquid gaseous fuel compositions for all the five loadings. Furthermore, second law analysis is carried out to optimize the dual fuel engine run. It is seen that a diesel engine can be run efficiently in hydrogen-diesel dual fuel mode if the diesel to hydrogen ratio is kept at 40:60.

Keywords: Diesel Engine, Diesel Replacement Ratio, Hydrogen, Dual Fuel, Efficiency, Second Law

1. Introduction

The use of conventional fossil fuels has reached a perceived crisis point. A number of reasons are responsible for this, such as finite reserves of what are non-renewable energy sources and the damage fossil fuels cause to the environment [1]. Therefore, researchers around the world are exploring every option to find suitable alternatives to replace fossil fuels, whether partially or fully [2]. Some of the alternative fuels that have been used to replace petroleum-based fuels include vegetable oils, alcohols, liquefied petroleum gas (LPG), liquefied natural gas (LNG), compressed natural gas (CNG), bio gas, producer gas, hydrogen etc. In this context, hydrogen (H_2), a non-carboniferous and non-toxic gaseous fuel, has attracted great interest and has huge potential. H_2 is only one of many possible alternative fuels that can be derived from various natural resources. Others include: coal, oil shale and uranium or renewable resources based on solar energy. H_2 can be commercially formed from electrolysis of water and by coal gasification; thermo-chemical decomposition of water and solar photo-electrolysis, although these are still in the developmental stage at present [3]. The energy required to ignite H_2 is very low and thus its usage in spark ignition

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(SI) engines is not suitable. Again, in compression ignition (CI) engines, H_2 will not auto ignite due to its high auto ignition temperature (858 K). Therefore the 'dual fuel' mode appears the best way to utilize H_2in internal combustion (IC) engines [4]. The dual fuel environment can be created by initially using a small amount of diesel (as pilot fuel) to launch the combustion and then supplying H_2 (as primary fuel) to deliver the rest amount of energy to run the cycle. Regarding power output, hydrogen enhances the mixture's energy density at lean conditions during a dual fuel run by increasing the hydrogen-to-carbon ratio, and thereby improves torque at the wide open throttle condition [5]. H_2 can be supplied in the engine by carburation, manifold or port injection or by cylinder injection [6, 7]. However, the injection of H_2 in the intake manifold or port requires a minor modification in the engine and offers a better power output than carburetion [8–10]. The experimental works of Yi et al. [11] established that intake port injection delivers higher efficiency than in-cylinder injection at different equivalence ratios too.

Varde and Frame figured out that the brake thermal efficiency (η_{bth}) of H_2diesel dual fuel mode is primarily dependent upon the amount of H_2 added. The larger the amount of H_2, the higher the value of η_{bth} is [3, 12]. It has been seen in H_2 diesel dual fuel mode that 90% enriched H_2 gives higher efficiency than 30% at 70% load, but cannot complete the load range beyond that due to knocking problems [3]. However, η_{bth} was found to drop when the amount of H_2 is less than or equal to 5%. In their analysis, an extremely lean air H_2 mixture restricts the flame to propagate faster, which lowers H_2 combustion efficiency [12]. However, experimental works done later, with H_2 diesel dual fuel mode, do not prove this drop in η_{bth} with H_2 addition as mentioned above [13]. According to Shudo et al. hydrogen combustion causes higher cooling loss to the combustion chamber wall than hydrocarbon combustion, because of its higher burning velocity and shorter quenching distance [14]. A study performed by Wang and Zhang indicates that the introduction of hydrogen into the diesel engine causes the energy release rate to increase at the early stages of combustion, which increases the indicated efficiency [15]. This is also the reason for the lowered exhaust temperature. According to them, for fixed H_2 supply at 50%, 75% and 100% load, H_2 replaces 13.4%, 10.1% and 8.4% energy respectively with high diffusive speed and high energy release rate.

The practice of normal and heavy exhaust gas recirculation (EGR) in H_2 diesel dual fuel mode is found to lower power production and fuel consumption [16]. Increases in compression ratio (CR) for H_2 fuelled diesel engine improves power, efficiency, peak pressure, peak heat release rate and emission of oxides of carbon, but increases NO_x emission [17]. A study of injection timing variation shown that advancing injection timing although provides favorable emission reduction, but makes engine operation more inefficient and unstable [18]. Sahoo et al. performed an experimental study on syngas diesel dual fuel mode for H_2:CO ratio of 100:0 at 20%, 40%, 60%, 80% and 100% of full load at maximum possible supply of hydrogen until knocking [19]. The study reveals that at 80% load, the engine offers a maximum 19.75% brake thermal efficiency at a maximum 72.3% diesel replacement ratio. A few researchers [4, 20] have studied the variation of H_2-diesel quantity for constant diesel supply at each load to improve the brake power (BP). The increase in the supply of H_2 in inlet manifold causes a reduction in the air flow to the engine. As a result, the volumetric efficiency (η_{vol}) and consequently the η_{bth} of the engine reduces. Therefore, there is scope to study and understand engine performance by varying both H_2 and diesel supply while maintaining constant BP at each load condition.

In light of this fact, the objective of the present study is to determine the best composition of H_2-diesel for maximum η_{bth} by varying the quantity of fuel (pilot and primary) and maintaining constant speed and BP at each of the five load conditions correspondingly. Some of the important physical and thermodynamic properties of diesel and H_2 are shown in Table 1. The load conditions selected are 20%, 40%, 60%, 80% and 100% of full load. As reported by Sahoo et al. [19], the maximum

Table 1: Properties of H_2 and diesel [19]

Properties	Diesel	Hydrogen
Chemical composition	$C_{12}H_{26}$	H_2
Density $\left(\frac{kg}{m^3}\right)$	850	0.085
Calorific value $\left(\frac{MJ}{kg}\right)$	42	119.81
Cetane number	45–55	–
Auto-ignition temperature (K)	553	858
Stoichiometric air fuel ratio	14.92	34.3
Energy density $\left(\frac{MJ}{Nm^3}\right)$	2.82	2.87

diesel replacement ratios during a dual fuel run are considered as 26%, 42%, 58%, 72% and 44% at the aforementioned loads respectively. Other performance parameters studied alongside are brake specific fuel consumption (BSFC), brake specific energy consumption (BSEC), η_{vol} and exhaust gas temperature (EGT). In order to endorse the experimental results and analysis, the Second Law analysis is performed to provide histograms of calculated availability of fuel, cooling water, exhaust gas, availability destruction and exergy efficiency. In this way, the present experimental and analytical studies will establish the optimized quantity of H_2-diesel composition for best efficiency at constant power at each load.

2. Experimental setup

The experiments are carried out in a Kirloskar TV1 CI diesel engine installed at the Centre for Energy of the Indian Institute of Technology (IIT), Guwahati, India. Figure 1 shows a schematic diagram of the engine test bed. The original engine specifications are shown in Table 2. The engine loading is performed by an eddy current type dynamometer. The liquid fuel is supplied to the engine from the fuel tank through a fuel pump and injector. The fuel injection system of the engine consists of an injection nozzle with three holes of 0.3 mm diameter with a 120° spray angle. A U-tube type manometer is used to quantity the head difference of air flow to the engine, while allowing the intake air to pass through an orifice meter. The engine block and cylinder head are surrounded by a cooling jacket through which water flows to cool the engine. To measure the specific heat of exhaust gas, a calorimeter of counter flow pipe-in pipe heat exchanger is also provided. Temperature measurement is performed by K-type thermocouples, which are fitted at relevant positions [21].

In order to convert the diesel engine test bed into dual fuel mode, some additional equipment is installed in the setup. These include: hydrogen gas cylinder with regulator, coriolis mass flow meter, flame trap with fine tuning regulator, non return valve (NRV) and gas carburetor. The coriolis mass flow meter measures the mass flow rate of hydrogen; while the flame trap and the NRV are used to prevent fire hazards due to accidental engine backfire. In the dual fuel mode $H_2$is supplied to the engine by the induction method. In this method, H_2 mixes with the intake air in the inlet manifold outside the cylinder. A gas carburetor [16] is fixed in the intake manifold of the engine to provide the H_2 supply. The liquid fuel supply is controlled through a fuel cut off valve for various diesel fuel replacement ratios by a lever-arm arrangement, as shown in Fig. 2.

3. Experimental procedure

Table 3 illustrates the designed experimental matrix of the H_2-diesel test at different loads. Initially, the engine is allowed to run on diesel at no load condition for a few minutes to attain a steady state. The cooling water supplies for the engine and calorimeter are set to 270 and

Table 2: Diesel engine specification [21]

Parameter	Specification
Engine type	Kirloskar TV1
General details	Single cylinder, four stroke diesel, water cooled, compression ignition
Bore and stroke	87.5×110 mm
Compression ratio	17.5:1
Rated output	5.2 kW (7 BHP) 1500 rpm
Air box	With orifice meter and manometer
Dynamometer	Eddy current loading unit, 0–16 kg
Fuel injection opening	205 bar 23° BTDC static
Calorimeter type	Pipe in pipe arrangement
Rotameter	For water flow measurement

Table 3: The experimental matrix

Load	Diesel replacement ratio	Engine operation
20%	10, 20, 26	Speed:
40%	10, 20, 30, 40, 42	1500±50 RPM
60%	10,20, 30, 40, 50, 58	Injection timing:
80%	10, 20, 30, 40, 50, 60 ,70, 72	23° BTDC
100%	10,20, 30, 40, 44	

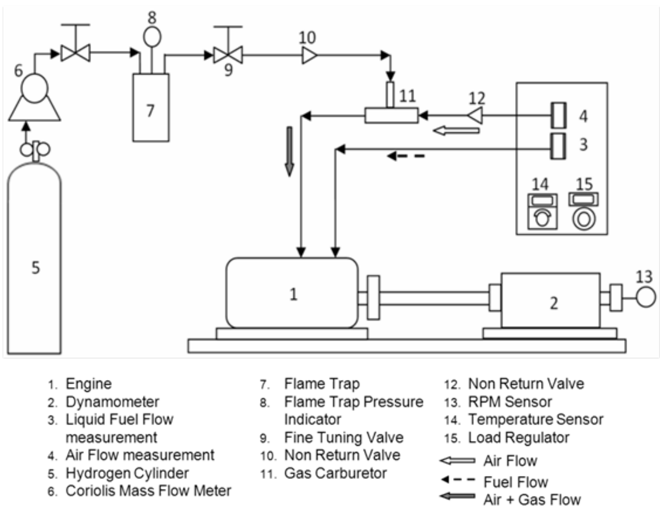

Figure 1: Schematic diagram of the setup

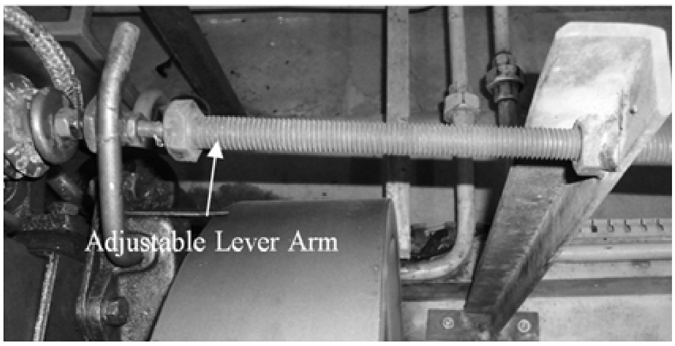

Figure 2: Adjustable lever arm arrangement

80 liters per hour, as per the engine provider instructions. Thereafter, the load is gradually increased to 3.2 kg (20% load) and the engine is allowed to run until it reaches a steady state. Then, the inlet and outlet temperatures of engine cooling water, calorimeter cooling water and exhaust gas are measured. Water head difference, diesel flow rate and engine speed are also recorded. The adjustable lever arm is then rotated to press the fuel cut off valve, which will reduce the fuel supply and speed.

The lever arm is then fixed at a point where diesel supply is reduced by 10%. At this point H_2 (99.99% purity) is allowed to flow from the high pressure cylinder to the flame trap, through the coriolis mass flow meter. At the outlet of the flame trap, one fine tuning regulator is connected to control H_2 flow accurately and is delivered to the intake manifold through the NRV and gas carburetor. The added supply of chemical energy in the form of H_2 in the cylinder is converted into mechanical energy after combustion. This increases the speed and BP of the engine. The quantity of H_2 is adjusted precisely to return the engine speed and BP to its previous value, recorded during the pure diesel run. The pressure of the H_2 outlet is not allowed to exceed 1.2 bar. After the engine reaches a steady state, the values of temperatures, water manometer head and mass flow of H_2 from coriolis flow meter are recorded. The H_2 supply is then stopped and the adjustable lever depressed further to reduce the diesel fuel supply by 20%. At this point, H_2 supply is initiated and the procedure is repeated.

Once the data of all the fuel replacement ratios are recorded, the engine is restored to its diesel mode. The load is increased by the eddy current dynamometer, and the measurement procedure for all the diesel replacement ratios are repeated at that load. The maximum fuel replacement ratios (shown in Table 3) for five loading conditions (20%, 40%, 60%, 80% and 100% of full load) are taken from the work reported by Sahoo et al. [19]. Finally, the H_2 supply is stopped completely, and the engine is allowed to run at "no load condition" prior to complete shutdown.

4. Analysis procedure

After collecting the data sets at each diesel replacement ratio and for each load, the dependent parameters are calculated according to the following equations [22, 23].

The diesel replacement ratio (Z) is given by

$$Z = \frac{\dot{m}_d - \dot{m}_{pd}}{\dot{m}_d} \cdot 100\% \tag{1}$$

The brake power can be written as

$$BP = \frac{2 \cdot 3.142 \cdot N \cdot W \cdot r}{60000} \tag{2}$$

The brake thermal efficiency for diesel mode is measured as

$$(\eta_{bth})_{diesel} = \frac{BP}{\dot{m}_d \cdot LHV_d} \cdot 100\% \tag{3}$$

The brake thermal efficiency for dual fuel mode is given by

$$(\eta_{bth})_{dual} = \frac{BP}{\dot{m}_{pd} \cdot LHV_{pd} + \dot{m}_h \cdot LHV_h} \cdot 100\% \tag{4}$$

The brake specific fuel consumption for dual fuel mode is computed from

$$BSFC = \left(\frac{\dot{m}_{pd} + \dot{m}_h}{BP}\right) \cdot 3600 \tag{5}$$

The brake specific energy consumption for dual fuel mode is given by

$$BSEC = \frac{\dot{m}_{pd} \cdot LHV_{pd} + \dot{m}_h \cdot LHV_h}{BP} \tag{6}$$

The volumetric efficiency can be computed from

$$\eta_{vol} = \frac{\dot{m}_a\left(\frac{kg}{s}\right) \cdot 3600}{\left(\frac{3.142}{4}\right) \cdot D^2 \cdot L \cdot \frac{N}{n} \cdot 60 \cdot K \cdot \rho_a} \cdot 100\% \tag{7}$$

5. Thermodynamic analysis

The results of the hydrogen-diesel dual-fuel experiment are analyzed using the Law of Thermodynamics. It provides significant information regarding the appropriate distribution of energy supplied by fuel in different parts of the engine [24]. Also, the energy that is utilized or destroyed is quantified through availability analysis. This analysis, finally, gives the exact amount of hydrogen and diesel composition which should be maintained to extract the maximum amount of energy from the fuel energy supplied. Hence, the "First Law (Energy)" along with the "Second Law (Exergy)" study of the engine is described in the following section with correct equations.

5.1. Energy analysis

According to the First Law of thermodynamics, the energy supplied in a system is conserved in its different processes and components [25]. In a CI engine, the fuel energy supplied (Q_i) is transferred in its different processes, viz. Shaft power (P_s), Energy in cooling water (Q_c), Energy in exhaust gas (Q_e) and Uncounted energy losses (Q_u)

in the form of friction, radiation, heat transfer to the surroundings, operating auxiliary equipments, etc. These different forms of energies are calculated according to the following analytical expressions [26].

The fuel energy supplied, i.e., the energy input can be calculated as follows:

$$(Q_i)_{diesel} = \frac{\dot{m}_d}{3600} \cdot LHV_d \tag{8}$$

$$(Q_i)_{dual} = \frac{\dot{m}_{pd}}{3600} \cdot LHV_{pd} + \frac{\dot{m}_h}{3600} \cdot LHV_h \tag{9}$$

The energy transferred into the shaft can be measured as

$$P_s = Brake\ power\ of\ the\ engine \tag{10}$$

The energy transferred into cooling water can be computed as

$$Q_C = \left(\frac{\dot{m}_{pd}}{3600}\right) \cdot C_{pw} \cdot (T_{wo} - T_{wi}) \tag{11}$$

The energy flow through exhaust gas can be estimated as

$$Q_e = \left(\frac{\dot{m}_e}{3600}\right) \cdot C_{pe} \cdot (T_{ei} - T_{eo}) \tag{12}$$

For a more precise thermodynamic analysis, the specific heat of exhaust gas is calculated from the energy balance of the exhaust gas calorimeter. Finally, from the energy balance, the uncounted energy losses can be estimated as

$$Q_u = Q_i - (P_s + Q_c + Q_e) \tag{13}$$

5.2. Exergy analysis

The availability can be described as the ability of the supplied energy to perform a useful amount of work [27]. In the CI engine the chemical availability of fuel (A_i) supplied is converted into different types of exergy, viz., Shaft availability (A_s), Cooling water availability (A_c), Exhaust gas availability (A_e) and Destructed availability (A_d) in the form of friction, radiation, heat transfer to the surroundings, operating auxiliary equipments, etc. These forms of energies are calculated according to the following analytical expressions as described in the literature [28–30].

The chemical availability of the fuel supplied is given by

$$(A_i)_{diesel} = 1.0338 \cdot \frac{\dot{m}_d}{3600} \cdot LHV_d \tag{14}$$

$$(A_i)_{dual} = 1.0338 \cdot \frac{\dot{m}_{pd}}{3600} \cdot LHV_{pd} + 0.985 \cdot \frac{\dot{m}_h}{3600} \cdot LHV_h \tag{15}$$

The availability transferred through the shaft is recorded as

$$A_s = Brake\ power\ of\ the\ engine \tag{16}$$

The cooling water availability can be measured as

$$A_C = Q_C - \left(\frac{\dot{m}_w}{3600}\right) \cdot C_{pw} \cdot \ln\left(\frac{T_{wo}}{T_{wi}}\right) \tag{17}$$

Exhaust gas availability can be calculated as

$$A_e = Q_e + \left(\frac{\dot{m}_w}{3600}\right) \cdot T_o \cdot \left\{C_{pw} \cdot \ln\left(\frac{T_o}{T_{ei}}\right) - R_e \cdot \ln\left(\frac{P_o}{P_e}\right)\right\} \tag{18}$$

The exhaust gas constant (R_e) is estimated from the energy balance of the exhaust gas calorimeter and the products of complete combustion of the diesel fuel.

The uncounted availability destruction is determined from the availability balance as

$$A_d = A_i - (A_s + Ac_e + A_e) \tag{19}$$

Therefore, the exergy efficiency (η_{II}) can be estimated as

$$\eta_{II} = 1 - \frac{Destructed\,availability}{Fuel\,availability} = 1 - \frac{A_d}{A_i} \tag{20}$$

6. Results and discussion

The results and discussion part of this H_2-diesel dual fuel experiment work is divided into two sections; viz., performance analysis and Second Law analysis. The performance analysis discusses η_{bth}

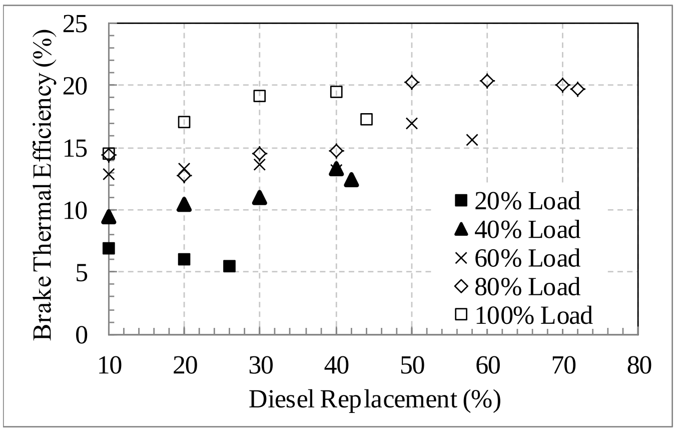

Figure 3: Variation of brake thermal efficiency with diesel replacement

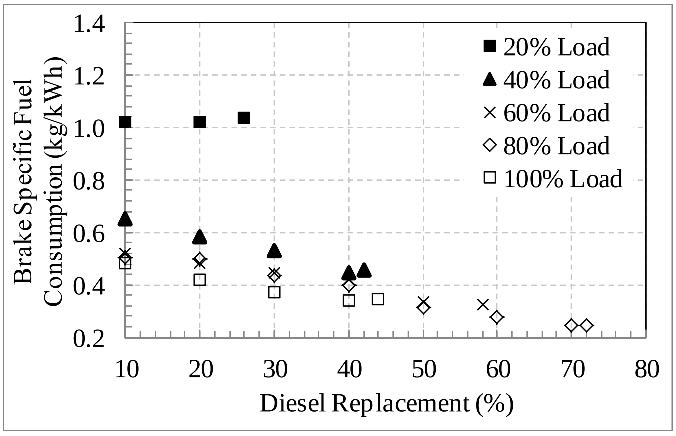

Figure 4: Variation of brake specific fuel consumption with diesel replacement

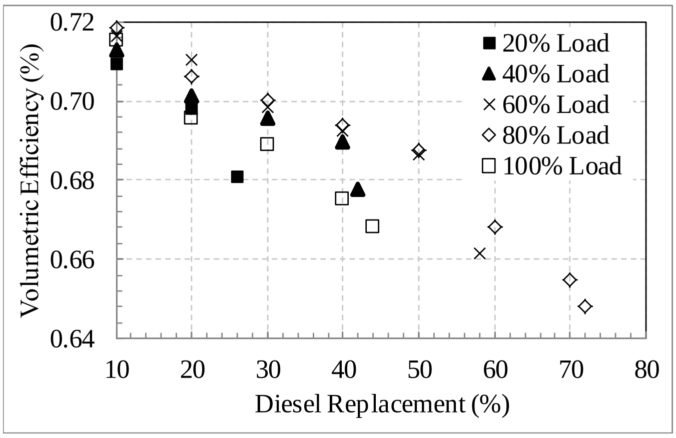

Figure 5: Variation of volumetric efficiency with diesel replacement

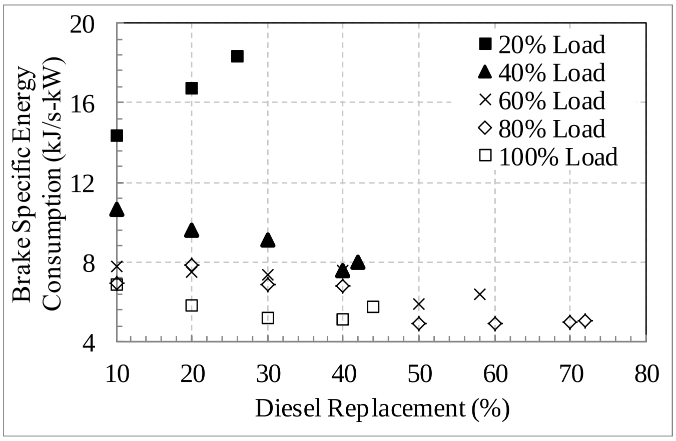

Figure 6: Variation of brake specific energy consumption with diesel replacement

, BSFC, BSEC, η_{vol}, EGT and a comparison of maximum brake thermal efficiencies for diesel and dual fuel modes. Later on, the Second Law analysis shows the availabilities of fuel, cooling water and exhaust gas, destroyed availability and exergy efficiency.

6.1. Performance analysis

The effect of variation of H_2-diesel quantity on η_{bth} for the five loading conditions is shown in Fig 3. Except for the 20% load, all other loading conditions show that an increase in H_2 quantity increases η_{bth}, but only up to a certain limit. This indicates that in the lower load region, H_2 cannot burn properly with diesel and results in poor combustion efficiency. However, this condition improves with the increase in load. The maximum value of η_{bth} obtained is around 20% at 80% load condition for both 50% and 60% diesel replacement ratio. Along with the increase in the η_{bth} there is also a reduction in BSFC encountered with the increase in load and H_2 substitution rate (except for the 20% load) which is exemplified in Fig. 4. This is because with the increase in H_2, the quantity of energy supply rate into the cylinder increases. Therefore, the total amount of fuel needed for the same BP is alleviated as far as energy supply is concerned. However, after a certain point of H_2 replacement, the engine may not run more efficiently, resulting in a reduction in η_{bth}. This is because of the large reduction in volumetric efficiency caused by a reduction of air (or more precisely oxygen) accessibility inside the cylinder. This can be clearly understood from Fig. 5. The reduction in BSEC with the increase

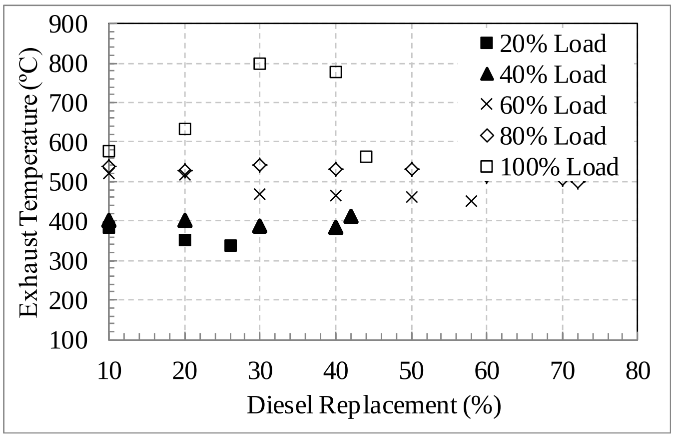

Figure 7: Variation of exhaust temperature with diesel replacement

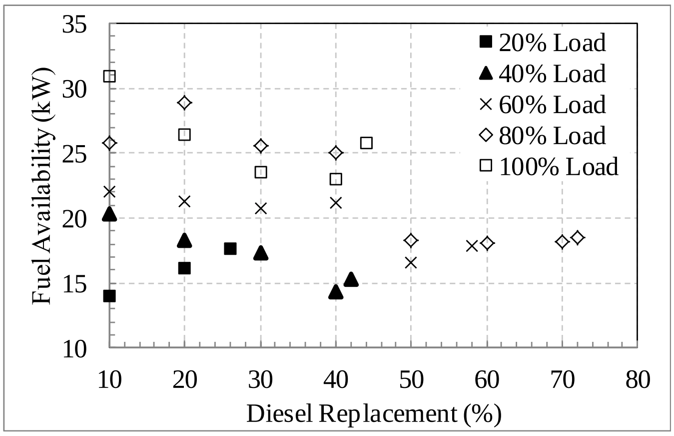

Figure 9: Variation of fuel availability with diesel replacement

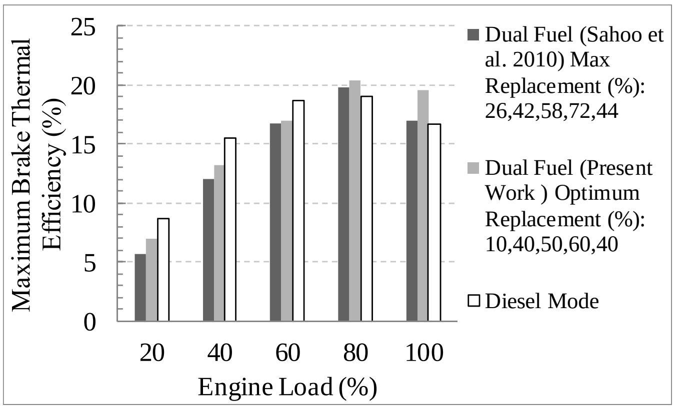

Figure 8: Comparison of maximum brake thermal efficiency between diesel and dual fuel modes

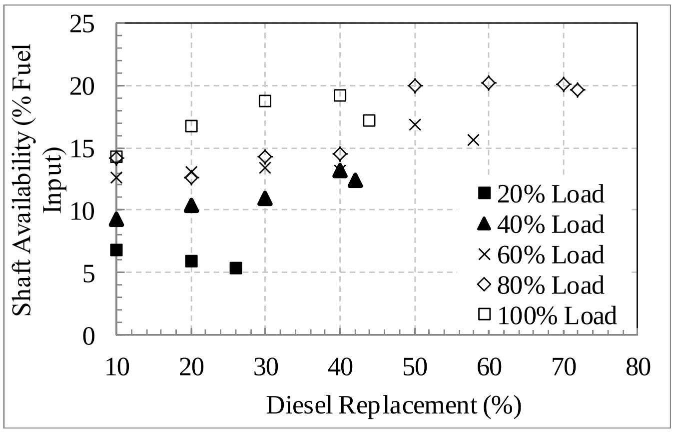

Figure 10: Variation of shaft availability with diesel replacement

in load and percentage of diesel replacement is shown in Fig. 6. Since H_2 carries more energy per unit mass than diesel (Table 1), the quantity required for H_2 to replace diesel for same power at each load is lower. However, at 20% load condition, traditional poor combustion efficiency meant slightly more H_2 was required to achieve the same power for the three diesel replacement ratios studied.

Figure 7 describes the effect of load and diesel replacement by H_2 on the EGT. In the higher load region H_2 burns more rapidly than it does at lower loads. However, at the highest load, the low efficiency of engine indicates higher fuel consumption. This fact combined with the high burning rate increased the exhaust gas temperature [31]. Figure 8 was added to this paper to compare the maximum brake thermal efficiencies obtained from this experiment and the findings of Sahoo et al. [19] (at 100% H_2 mode) with diesel mode for each loading condition. The maximum η_{bth} found from the present study at 20%, 40%, 60%, 80% and 100% load are 7%, 13%, 17%, 20% and 20% by replacing 10%, 40%, 50%, 60% and 40% diesel while maintaining constant speed and BP. These values are 1.5 to 19% higher than the brake thermal efficiencies found for maximum diesel replacements [19]. By increasing the H_2 substitution, η_{bth} can be increased only up to a certain limit, beyond which any increase in H_2 reduces the quantity of air in the cylinder to burn it. This results in incomplete combustion, thereby negatively impacting engine performance.

6.2. Second Law analysis

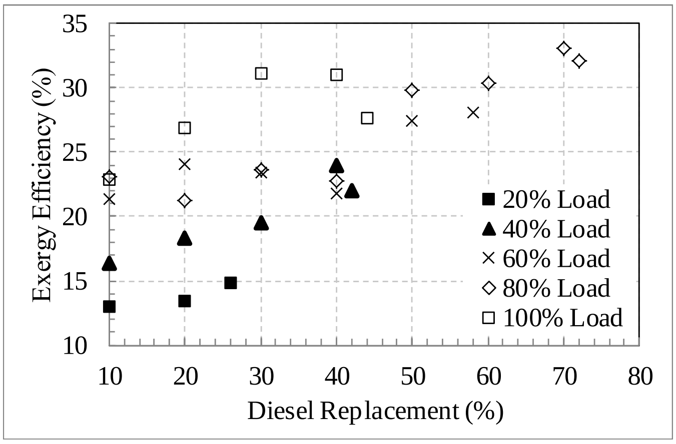

Figure 11: Variation of exergy efficiency with diesel replacement

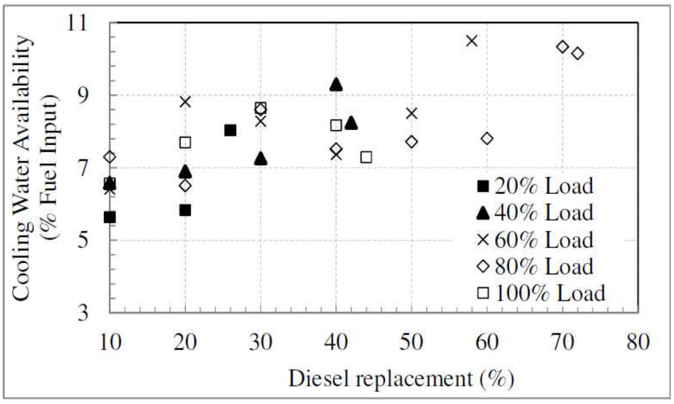

Figure 12: Variation of cooling water availability with diesel replacement

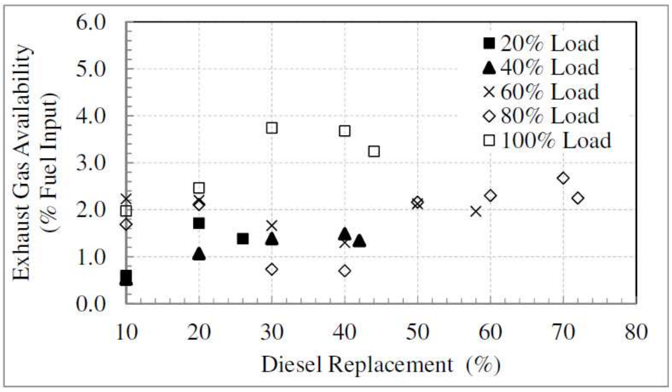

Figure 13: Variation of exhaust gas availability with diesel replacement

The outcome of experimental observation performed in the present work is again processed by Eqs. (14) through (20) to achieve the Second Law analysis, which are presented in Figs 9 through 11 as a function of diesel replacement. With the increase in load the engine needs more fuel to burn and to achieve higher power. However, with the increase in H_2 quantity and the reduction in the pilot diesel supply, fuel availability decreases. This is because with an increase in H_2, the engine receives more fuel (i.e., H_2) with a high energy density, which can make up the energy needed to run the engine at the same BP at that particular load (Fig. 9). Although the BP is kept constant for each loading condition throughout the diesel replacement study, the reduction in fuel availability alongside the increase in H_2 substitution results in an increase in the percentage of shaft availability, as is clear from Fig. 10.

The percentage of cooling water availability with diesel replacement is shown in Fig. 12. Although there is a slight increase in the available work obtained in the cooling water with the increase in load, the increase in H_2 again balances up to 50% diesel replacement. This is because an increase in H_2 in the dual fuel system reduces the chances of energy being wasted in the exhaust cooling water due to its better utilization during the combustion process. However, beyond 50% diesel replacement, the available work in the cooling water increases due to more unharnessable energy flowing out though the cooling water.

The engine exhaust too has some exergy, a potential to cause change, as a consequence of not being in stable equilibrium with the environment. When released to the environment, this exergy represents an opportunity to change the environment. If trapped, this exergy may cause a potentially useful change [32]. The exhaust gas availability (Fig. 13) gives the details of lower available work in the low load range (20% and 40%) due to incomplete combustion and the low exhaust gas temperature (Fig. 7). At highest load, the very high exhaust gas temperature results in a huge amount of available energy. However, at 60% and 80% loads, better combustion and hence higher efficiency causes lower exergy flow through exhaust gas. It is clear from Fig. 14 that the increase in H_2 and load reduces the chances of energy to destroy. Finally, an increase in load and H_2 in-

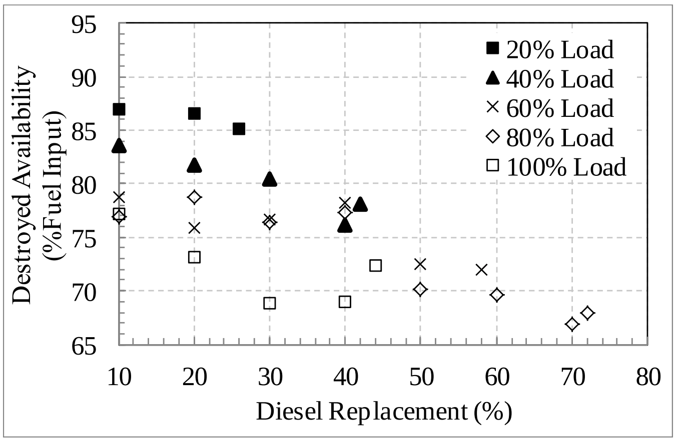

Figure 14: Variation of destroyed availability with diesel replacement

creases exergy efficiency, but only up to a certain limit as seen from Fig. 11. The maximum exergy efficiencies obtained are around 15%, 24%, 28%, 33% and 31% for the loading conditions studied.

7. Uncertainity analysis

Uncertainty analysis for the various parameters of engine performance and availability is performed by using perturbation techniques [33, 34]. The uncertainties calculated for various independent parameters are: engine speed (1.1%), engine load (1.5%), liquid fuel flow rate (2%), gas flow rate (1.3%), water flow rate (1.2%), LHV of liquid and gaseous fuel (1%) and temperature (1.6%). Using these values, the computed engine performance and availability parameters are expected to be accurate within ± 4.3%.

8. Conclusions

In this investigation, an experimental study and Second Law analysis are performed for an H_2-diesel dual fuel in a CI diesel engine. The experiments were performed for various diesel replacements with H_2 in five different loadings to obtain the best performance point and then Second Law analysis is performed to establish the findings. The findings from this study can be summarized as follows.

1. The increase in load and H_2 substitution increases η_{bth} of the dual fuel engine up to a particular point. For 80% load this happens up to 60% diesel substitution. The fall in η_{bth} beyond this range is due to a reduction in η_{vol}.
2. The increase in H_2 supply for a constant BP at each load results in a reduction in BSFC and BSEC. This is because H_2 has a higher energy rate than diesel.
3. The increase in EGT at the highest load is severe due to the high burning rate and lower efficiency at almost similar fuel consumption.
4. A comparison of maximum efficiency shows that the increase in H_2 supply will not raise the engine efficiency endlessly. In order to get most efficient performance from the engine, it has to be operated within 40% to 60% diesel replacement in dual fuel mode during loading conditions.
5. Fuel availability increases with the increase in load to cope with the rise in BP. However, the increase in high energized H_2 supply compensates for this fact, and hence fuel availability reduces. The above fact again increases the shaft availability as a percentage of fuel input; although BP is maintained fixed at each load.
6. In the low to mid range of diesel replacement, the increase in H_2 supply compensates for the increase in cooling water availability with the increase in load. Better combustion and higher efficiency result in a reduction in exhaust exergy flow in this region in the medium to high load range.
7. The H_2-diesel dual fuel system runs more efficiently and delivers better performance when H_2-diesel compositions are kept within 40%, 50%, 60% and 40% for the 40% to 100% load range, keeping the BP constant. However, the dual fuel run is not preferred for the 20% load due to the poor showing of the engine.

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Nomenclature

$\dot{m}$ Mass flow rate $\left(\frac{kg}{s}\right)$

η_{bth} Brake thermal efficiency

η_{II} Exergy efficiency

η_{vol} Volumetric efficiency

ρ Density $\left(\frac{kg}{m^3}\right)$

$_a$ Air

$_c$ Cooling water

$_d$ Diesel

$_e$ Exhaust gas

$_h$ Hydrogen

$_i$ Input

$_o$ Atmospheric condition

$_s$ Shaft

$_u$ Uncounted

$_w$ Water

$_{ei}$ Exhaust gas inlet to calorimeter

$_{eo}$ Exhaust gas outlet from calorimeter

$_{pd}$ Pilot diesel

$_{wi}$ Water inlet to calorimeter

$_{wo}$ Water outlet from calorimeter

A Availability (kW)

BHP Brake horse power

BP, P_s Brake power (kW)

BSEC Brake specific fuel consumption $\left(\frac{kJ}{kW \cdot s}\right)$

BSFC Brake specific fuel consumption $\left(\frac{kg}{kW \cdot hr}\right)$

BTDC Before top dead centre

cp Specific heat $\left(\frac{kJ}{kg \cdot K}\right)$

CA Crank angle (degree)

CI Compression ignition

CO Carbon monoxide

CR Compression ratio

D Engine cylinder diameter (m)

DI Direct injection

EGT Exhaust gas temperature (°C)

IC Internal combustion

K Number of cylinder

L Engine stroke length (m)

LHV Lower heating value $\left(\frac{MJ}{kg}\right)$

N Revolutions per minute (RPM)

n Number of revolutions per cycle (2 for four stroke)

NRV Non return valve

p Pressure (bar)

Q Energy (kW)

R Gas constant $\left(\frac{kJ}{kg \cdot K}\right)$

r Dynamometer arm radius (m)

SI Spark ignition

T Temperature (K)

Z Diesel replacement rate