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Thermodynamic Analysis of Variable Compression Ratio Diesel Engine fuelled with Waste Fried Oil Methyl Ester-diesel blend

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ABSTRACT

The present work is set to explore the effect of compression ratio (CR) and injection timing (IT) on energy and exergy potential of a waste fried oil methyl ester (WFOME) – diesel blend run diesel engine. Experiments are carried out in a single cylinder, direct injection, water cooled variable compression ratio diesel engine at a constant peed of 1500 rpm under a full load of 3.5kW brake power. The study involves three different CRs of 16, 17 and 18; and three different ITs of 20, 23 and 25 ^{0}BTDC. Here, the CR of 18 and IT of 23^{0}BTDC are the standard ones. The energy analysis performed for the experimental data includes shaft power, energy input through fuel, output by cooling water and exhaust, uncounted loss per unit time. The exergy analysis is carried out for availability input, shaft, cooling water and exhaust availability, availability destruction and entropy generation. It shows that higher values of CR increase the shaft availability and cooling water availability, however, they decrease the exhaust flow availability. The retardation and advancement of IT give different results. The exergy analysis also shows that with the increase of CR, increase the shaft availability and exergy efficiencywhile it reduces the exergy destruction, butthe injection retardation and advancement lowers the shaft availability and exergy efficiency. The entropy generation is also reduced for the similar CR and increases for similar IT modifications.

***Keywords*— Injection timing, Performance, waste fried oil methyl ester(WFOME), Thermodynamic Analysis.**

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I. INTRODUCTION

Now a-days not only conventional fuels power IC engine, but also various renewable alternative fuels viz. biofuels, biogas, natural gas, hydrogen etc. are also made use of fuel. Biofuels, especially the methyl and ethyl esters of vegetable oils, popularly known as 'biodiesels', have an important contribution. This is because of the need to reduce the use of fossil fuels in diesel engines without modifying them [1,2]. Biodiesels are oxygenated fuels and can be used in diesel engines to improve combustion efficiency. The internal combustion (IC) engines are the building blocks of modern civilization. This is because of their capability to convert chemical energy of fuel into heat and mechanical energy. In order to study effective utilization of energy, terminologically 'thermodynamic analysis' of biodiesel run IC engines is necessary. However, biodiesels have a comparatively lesser calorific value than diesel, which causes lower power and efficiency [3]. This can be improved by raising the CR.

The energy and exergy analysis for IC engines, especially for diesel engines, have been discussed merely in last two decades [4,5]. The second law analysis performed

on various engine parts (Cummins make, USA) as well as whole diesel plants is reported by Flynn et al. [6]. Van Gerpen and Shapiro [7] performed a detailed analysis for a closed cycle, bringing into focus the belligerent term of chemical availability along with the thermo-mechanical one. Further, Rakopoulos and Andritsakis [8] studied the irreversibility's in direct and indirect injection diesel engines combustion. According to Giakoumis [9], the availability destruction in a low heat rejection (LHR) engine is small, which does not allow the mechanical work to increase. Rather, it increases the potential for extra work recovery owing to the higher availability content of the exhaust gas. Some of other thermodynamic analysis performed by Rakopoulos and his co-workers [10–13] are available in literature. Their study uncovered a method for calculating both combustion irreversibility and working medium availability for a diesel engine [14]. Ajav et al. [15] have showed that the thermal balance of the engine operating on 15% and 20% ethanol–diesel blends is significantly different than 5% and 10% ethanol–diesel blends. It is observed that the thermodynamic analysis on the biodiesel-diesel blend is not studied extensively. The present study provide thermodynamic analysis of Waste fried oil methyl ester (WFOME)–diesel blend (B40: 40% biodiesel, 60 % mineral diesel).

II. Objectives

The theoretical investigation on the effective distribution of energy at various components of IC engine has been done by coupling the first and the second laws of thermodynamics together. This clears the picture of thermodynamic energy distribution of engine manoeuvre. The details study of literature unfurls the fact that the effect of engine design and operating parameters viz., CR and IT variation on energy and exergy distribution of a diesel engine running with waste fried oil methyl ester (WFOME)-diesel blend is not clear. However, in order to establish WFOME as an alternative to diesel fuel, it is necessary to uncover the effect of engine design and operating parameters on thermo mechanical energy–exergy distribution. In this context, experiments are performed in a WFOME-diesel blend (B40) run diesel engine at full load condition for a set of CR and IT. The CRs of 16, 17 and 18 and ITs of 20, 23 and 25 BTDC are considered for the experimentation. The results obtained from the tests are then analysed to explore the energy and exergy potential of fuel input, shaft work, cooling water and exhaust gas potential and exergy destruction. Exergy efficiency (ηII), and entropy generation are also analysed and discussed.

III. Experimental setup and Procedure

The experiments are conducted in a single cylinder, four stroke, direct injection variable compression ratio (VCR) diesel engine (Kirloskar make, India) fuelled with WFOME-Diesel blend. The engine is connected to a hydraulic cooling type eddy current dynamometer for loading. A tilting cylinder block arrangement is used to vary the CR without stoppingthe engine and altering the combustion chamber geometry. The brief engine specification are: bore 87.5 mm, stroke 110 mm, CR range 12–18, capacity 661 cc (at standard CR 17.5), IT range 0-25BTDC. The engine produces 3.5 kW of rated power with diesel at full load (12 kg) at a rated speed of 1500 rpm. A total of six thermocouples (four PT100 type and two K type) are installed at various locations of the setup for measurement of water and exhaust gas temperature. The setup has a stand-alone panel box consisting of air box, fuel tank, manometer, fuel measuring burette. The fuel measurement is performed by differential pressure transducer (Yokogawa make, Model No: EJA110A-DMS5A-92NN). It is connected through a fuel line and the signal of flow rate is transferred to the National Instrument made data acquisition device (DAD). This DAD is connected to the computer with USB port and measurement of fuel flow is stored in computer in kg/h. Rotameters are used for cooling water flow measurement through the jackets of engine block, cylinder head and calorimeter. All the analog signals recorded from different locations of the test rig are supplied to the 'Enginesoft' software for performance analysis. The schematic diagram of the VCR diesel engine setup is shown in the Fig. 1.

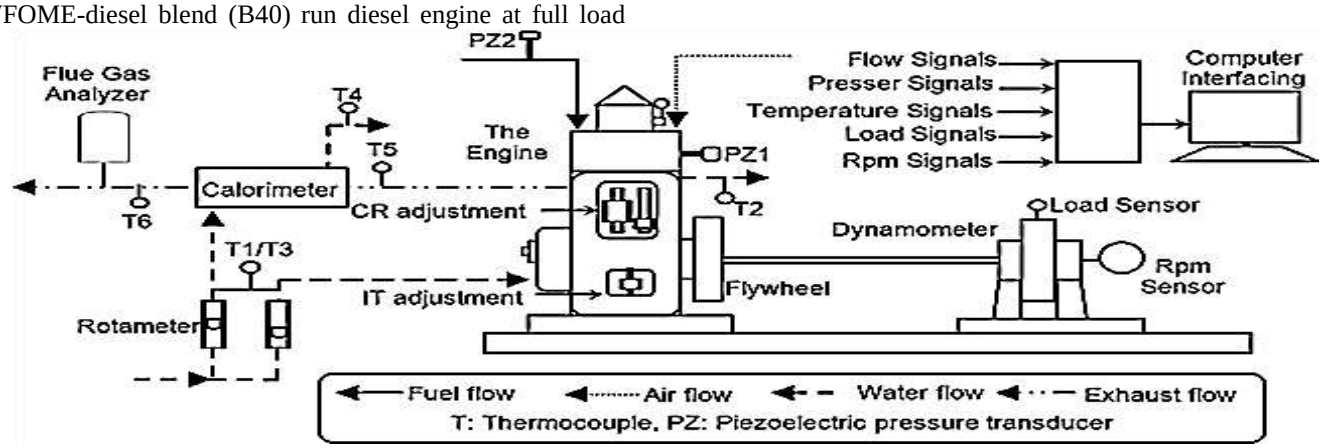

Fig.1 The experimental setup

The VCR engine is first run using diesel at standard diesel specification; CR of 18 and IT of 23BTDC. As the load is increased, the engine speed reduces. In order to maintain a constant BP, the engine consumes more fuel resulting a higher heat release, and hence, a higher temperature inside cylinder. This increases temperatures at the outlet of the cooling water and exhaust gas. When the full load condition (12 kg) is achieved, the engine is allowed to run for few minutes and the temperatures at the outlet of cooling water and exhaust gas are monitored closely at the

computer display until it reaches a steady state condition. This indicates that the combustion inside the cylinder becomes steady and the engine is ready for data acquisition. The readings of temperatures, air and fuel flow rate, speed, cylinder and fuel pressure variation are automatically recorded by the DAD. Thereafter, the engine is brought back to no load condition slowly and allowed to run for few minutes. Later, WFOME-diesel blend is tested in the VCR engine at various CRs and ITs. The VCR engine allows online modification of CR variation. The CRs under investigation are 16, 17 and 18

along with the standard CR of 18. Initially, the engine is set at CR of 18 and IT of 23BTDC for WFOME-diesel test. Once the data are recorded at this particular setting, the CR is changed by rotating the CR adjustor knob. Once all the CRs at 23BTDC are tested, the engine is then allowed to run at other ITs by rotating the injection point adjustment nut. Thereafter, for each IT, the engine is tested at all the CRs. The appropriate IT is confirmed from the fuel pressure data. The same process as discussed above is repeated to study other IT before complete shutdown. The key properties of fuels used in this study are listed in the Table I.

TABLE I. PROPERTIES OF FUEL BLENDS

Properties	B20	B40	B60	Diesel	Method
Viscosity at 40 ^{0}C (cSt)	4.94	5.56	6.18	4.320	ASTM D445
Specific gravity	0.838	0.846	0.854	0.830	ASTM D941
Calorific value (kJ/kg)	42200	41,400	40600	43,000	ASTM D240
Flash point (^{0}C)	95	120	145	70	ASTM D93

IV. Analysis Methodology

The sequence of events happening in the engine operation can be identified as fuel and air entrainment, combustion, conversion of chemical energy into mechanical work, heat loss through cooling water, friction, radiation, surroundings and exhaust gas [16]. Further it is assumed that the combustion air and exhaust gases are ideal gas mixtures and their potential and kinetic energy changes are minor [17]. Reference atmospheric conditions are considered as 1 atm and 28 ^{0}C of pressure (Pamb) and temperature (Tamb). The basic equations used for performance calculations are as follows;

- Brake power (BP):

$$BP = (2 \times \pi \times N \times W \times r)/60000, kW \quad (1)$$

Where N is speed of engine in RPM, W is dynamometer load in N and r is the dynamometer arm radius in m.

- Brake thermal efficiency (BTHE):

$$BTHE = \left[\frac{BP}{\dot{m}_f \times LHV_f}\right] \times 100, \% \quad (2)$$

where BP is the brake power of the engine in kW, $\dot{m}_f$ is the mass flow rate of fuel in kg/s and LHV_f is the lower heating value of the fuel in kJ/kg.

A. *Energy Analysis*

In compression ignition (CI) engine, the fuel energy supplied per unit time (Q_{in}) is transferred in its different processes, viz. shaft power (Q_s), energy in cooling water per unit time (Q_w), energy in exhaust gas per unit time (Q_e) and uncounted energy losses per unit time (Q_u) in the form of friction, radiation, heat transfer to the surrounding, operating auxiliary equipment's, etc. These different forms of energies are calculated according to the following analytical expressions [18].

- Fuel energy supplied per unit time (Q_{in}):

$$Q_{in} = \dot{m}_f \times LHV_f, kW \quad (3)$$

- Shaft power (Q_s):

$$Q_s = Brake\ power\ of\ engine,\ kW \quad (4)$$

- Energy in cooling water per unit time (Q_w):

$$Q_w = \left[\dot{m}_{we} \times C_{pw} \times (T_{woe} - T_{wie})\right], \quad kW \quad (5)$$

where$\dot{m}_{we}$ is the mass flow rate of cooling water in kg/s passing through engine jacket, Cpw is the specific heat of water in kJ/kg K and Twie and Twoe are the inlet and outlet temperature of cooling water passing through engine jacket.

- Energy in exhaust gas per unit time (Q_e):

$$Q_e = \left[\left(\dot{m}_f + \dot{m}_a\right) \times C_{pe} \times (T_{eic} - T_{eoc})\right], \ kW \quad (6)$$

Where $\dot{m}_a$ is the mass flow rate of air in kg/s, specific heat of exhaust gas (Cpe) is obtained from the energy balance of the flows passing through the calorimeter, as follows:

$$C_{pe} = \frac{[\dot{m}_{we} \times C_{pw} \times (T_{woe} - T_{wie})]}{[(\dot{m}_f + \dot{m}_a) \times (T_{eic} - T_{eoc})]}, \ kJ/kg\ K \quad (7)$$

where $\dot{m}_{wc}$, Twic and Twoc are the mass flow rate, inlet and outlet temperature of the cooling water passing through the calorimeter and Teic and Teic are the inlet and outlet temperatures of exhaust gas passing through calorimeter.

- Uncounted energy losses per unit time (Q_u):

$$Q_u = [Q_{in} - (Q_s + Q_w + Q_e)], \quad kW \quad (8)$$

Exergy analysis

The availability can be described as the ability to perform useful amount of work by the supplied energy [19]. In the CI engine the availability of fuel (Ain) supplied is converted into different types of exergy, viz., shaft availability (As), cooling water availability (Aw), exhaust gas availability (Ae)

and destructed availability (Ad) in the form of friction, radiation, heat transfer to the surrounding, operating auxiliary equipment's, etc. These forms of energies are calculated according to the following analytical expressions as described in the literature [20–21].

- Input availability of fuel (A_{in}):

$$A_{in} = [\dot{m}_f \times LHV_f \times \{1.041 + 0.1728(H/C) + 0.0432(O/C) + 0.2169(S/C) \times (1 - 2.0628(H/C))\}], \quad kW \tag{9}$$

where H, C, O and S are the mass fractions of hydrogen, carbon, oxygen and sulfur contents [22].

- Shaft availability (A_s):

$$A_s = Brake\ power\ of\ engine, \ \ kW \tag{10}$$

- Cooling water availability (A_w):

$$A_w = Q_w - [\dot{m}_{we} \times C_{pw} \times T_{amb} \times \ln(T_{woe}/T_{wie})], \quad kW \tag{11}$$

- Exhaust gas availability (A_e):

$$A_e = Q_e + [(\dot{m}_f + \dot{m}_a) \times T_{amb} \times \{C_{pe} \ln\left(\frac{T_{amb}}{T_{eic}}\right) - R_e \times ln(P_{amb}/P_e)\}], \ \ kW \tag{12}$$

where Re is the specific gas constant of the exhaust gas in kJ/kg K. It is calculated from the thermodynamic relation Re = (RU/molecular weight). RU is the universal gas constant in kJ/kmol K and the molecular weight (kg/kmol) of combustion products is calculated taking into account complete combustion.

- Destructed availability (A_d):

The availabilities As, Aw and Ae are the exergies that can be recovered. Destructed availability

$$A_d = A_{in} - (A_s + A_w + A_e), \quad kW \tag{13}$$

- Exergy efficiency (η_{II}):

$$\eta_{II} = (1 - (A_d/A_{in}) \tag{14}$$

- Entropy generation rate:

The procedure of entropy generation is fairly a novel technique to determine perfectly, the losses in various components in an energy system and to indentify the scopes of enhancement of overall system performance [23]. The entropy generation can be expressed as

$$\dot{S} = [A_d/T_{amb}], \quad kW/K \tag{15}$$

V. Results and Discussion

As the results from RSM shows optimum results for B20 the thermodynamic analysis will be discussed in detail for B20. Results for B40, B60 with reference B0 are tabulated (Table II).

TABLE II. RESULTS OF ENERGY ANALYSIS

CR	IT	Ain (kW)	As (kW)	Aw (kW)	Ae (kW)	Ad (kW)	η_{II}(%)
Diesel							
18	23	13.97	3.43	0.01	0.70	9.82	30.41
B20 (20% WFOME and 80% diesel)							
18	20	15.27	3.45	0.11	0.54	11.17	26.83
18	23	14.47	3.44	0.07	0.63	10.33	28.62
18	25	15.14	3.46	0.06	0.63	10.99	27.37
17	20	15.54	3.47	0.04	0.61	11.42	26.53
17	23	14.74	3.45	0.10	0.61	10.57	28.24
17	25	15.27	3.43	0.06	0.64	11.14	27.08
16	20	16.08	3.46	0.04	0.56	12.02	25.25
16	23	15.41	3.44	0.05	0.53	11.38	26.13
16	25	15.81	3.46	0.04	0.55	11.76	25.59
B20 (20% WFOME and 80% diesel)							
18	20	16.23	3.44	0.04	0.77	11.99	26.13
18	23	15.26	3.45	0.00	0.85	10.96	28.18
18	25	16.09	3.43	0.01	0.77	11.88	26.16
17	20	16.51	3.40	0.06	0.67	12.38	24.99
17	23	15.95	3.47	0.02	0.75	11.72	26.54
17	25	16.37	3.45	0.06	0.66	12.20	25.47
16	20	17.34	3.43	0.05	0.61	13.25	23.58
16	23	16.65	3.46	0.02	0.67	12.50	24.92
16	25	17.48	3.44	0.04	0.63	13.37	23.53
B20 (20% WFOME and 80% diesel)							
18	20	17.63	3.46	0.05	0.71	13.41	23.95
18	23	16.76	3.44	0.10	0.57	12.64	24.56

18	25	17.19	3.45	0.11	0.68	12.96	24.64
17	20	18.07	3.43	0.04	0.70	13.90	23.07
17	23	17.49	3.45	0.05	0.72	13.26	24.18
17	25	17.78	3.46	0.05	0.70	13.57	23.64
16	20	18.80	3.44	0.06	0.57	14.72	21.69
16	23	18.21	3.45	0.06	0.61	14.10	22.59
16	25	18.65	3.45	0.05	0.65	14.50	22.24

A. *Energy analysis*

The distributions of energy per unit time through different process calculated are included in Table II. The standard deviation among the fuel energy input values lies within 0.00–0.06. Hence, the average value of standard deviation falls under a very negligible range (<0.03). The mean fuel energy input per unit time for entire CR and IT combinations studied are 13.38 kW. A portion of this input energy in the form of chemical energy of fuel has been converted into mechanical shaft work (3.44 kW). Some amount of energy is flown through engine cooling water (3.91 kW) and exhaust gas (1.89 kW). The rest amount of energy (4.13 kW) has been lost due to friction, radiation, heat transfer to surroundings. Therefore approximately 26% of input energy is converted into mechanical work and 74% of input energy is lost in various ways from the system. The WFOME has a lower calorific value which is merely 7% lesser than diesel. The lower BTE for the WFOME-diesel blend run engine is due to the increase in fuel energy input. Since the engine runs at a constant speed, it has to produce a constant power at a particular load. In order to achieve this, it has to consume a little more WFOME – diesel blend to cover up its lower energy content. Thus, it is seen that, at CR = 17 and IT = 23, the B60 run engine produces around 6.5% lower BTE than diesel. However, altering the engine design and operating parameters (CR and IT) and BX %, the BTE is found to improve for the B20 run engine showing a decrease of 0.4% as opposed to a decrease of 6.5% as observed earlier. Hence, it is justified that WFOME-diesel blends with a little lower LHV can supply proportionately a substantial amount of energy to run the engine.

The effects of CR and IT on energy per unit time sharing are included in Figs. 8 and 9. The values are obtained by using the analysis Eqs. To study the effect of CR on various energy distributions, the values of three ITs are averaged and included in Fig. 8. The engine is operated at constant speed and constant BP for all the CR and IT combinations. However, the rise in CR causes surge in the temperature during the compression stroke. This is because the high temperature environment at higher CR causes the pressure to elevate to a higher value.

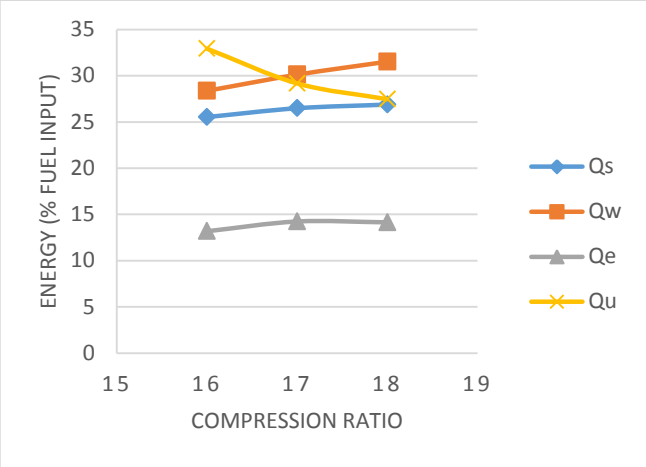

Fig. 2. Variation in energy distribution with compression ratio

Side-by-side, it is also observed that, advancing the IT increases peak pressure and peak heat release rate. Advancement of IT means that fuel is applied at cooler environment to that of retardation. Hence, fuel consumption becomes more for the same BP and retardation of IT gives lesser time for efficient combustion hence more fuel consumption for the same BP. However, as the piston reaches close to TDC, the remaining high fuel quantity attains a favourable environment for combustion. This increases the rate of combustion, rises the peak pressure, and hence, the peak heat release rate. The input fuel energy is reduced with the increase in CR (Table II). This causes an increase in shaft power (Qs) (% of fuel input) which is observed from Fig. 2 as well. The situation is also attributed to the increase in the BTE with the increase in CR for B20 run diesel engine. The values of shaft energies are 25.52%, 26.48% and 26.88% of respective fuel input for CR of 16, 17 and 18. The ITs have similar effect at low and at high CR settings. However, in the intermediate range of CR, the combustion can be described as fully premixed and partially premixed combustion. This is because, at 25 BTDC, the B20 is injected slightly earlier than other ITs and well before the piston reaches the TDC. This allows B20 to have more time to mix with air, which in other words, can be called partially premixed charge. The burning of this charge releases a higher amount of heat upon combustion thereby increasing more cooling water heat loss unlike 23 BTDC. However, the uncounted energies are unaffected at higher CRs.Fuel injection advancement and retardation have a significant effect on the energy distribution (Fig. 3). Advancing (25 BTDC) and retarding (20 BTDC), the IT has decreased the shaft power (% of fuel input) and hence BTE. This is because of the increment of fuel supply during IT advancement or retardation than the standard IT. The mean values of fuel energy supplied per unit time during 20 BTDC and 25 BTDC are 13.41 kW and 13.23 kW which, in turn, are 5.01% and 3.6% higher than the rate of energy

supplied during standard IT, respectively. There is a fluctuation of uncounted heat loss, considering CR and IT variation. This is because of the rise of EGT with IT advancement and increase in average cooling heat loss per unit time. All these facts coupled with lower rate of fuel energy input at 23 BTDC cause increase in uncounted heat loss for IT retardation and advancement.

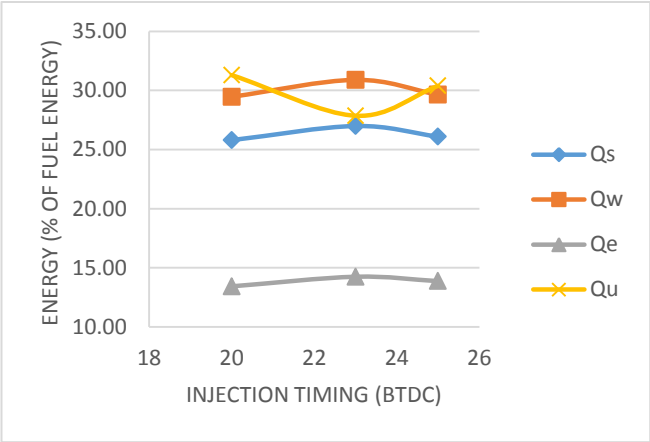

Fig. 3. Variation in energy distribution with Injection timing

B. *Exergy Analysis*

The findings of exergy (the second law) analysis are included in Table III. It represents the availability values and standard deviations of various terms including fuel availability, shaft availability, availability associated with engine cooling water and exhaust gas.The standard deviation among the fuel exergy input values lies between 0.01 and 0.05. Hence, the average value of standard deviation falls below a very negligible range (<0.03). The mean fuel exergy for entire CR and IT combinations studied is 16.43 kW. This can also be termed as total input exergy; since input exergy through air for combustion is neglected. A portion of this input has been converted into mechanical shaft work (3.44 kW). Some amount of exergy is flown through engine cooling water and exhaust gas. The rest amount of exergy has been lost due to friction, radiation, heat transfer to surroundings. However, with respect to cumulative availability, the exergy associated with the cooling water and exhaust gas also come under consideration. Therefore, approximately 30% of input exergy is found which can be called as available energy from the thermodynamic viewpoint. As a result, 70% of input exergy is destroyed from the system.

Figs. 4 and 5 describe the variation of availability associated with shaft, cooling water, exhaust gas and availability destruction with respect to CR and IT separately. The trend of shaft availability for CR variation (Fig. 4) is almost same to that of shaft power as described in Fig. 8. The shaft availabilities for the CRs of 16, 17 and 18 are 21.91%, 22.75% and 23.07% of fuel input, respectively. This reduction of absolute value of fuel availability is responsible for the increase in shaft availability although it is considered

TABLE III. RESULT OF EXERGY ANALYSIS

CR	***IT***	***Ain (kW)***	***As (kW)***	***Aw (kW)***	***Ae (kW)***	***Ad (kW)***	***η_{II}(%)***
Diesel							
18	23	13.97	3.43	0.01	0.70	9.82	30.41
B20 (20% WFOME and 80% diesel)							
18	20	15.27	3.45	0.11	0.54	11.17	26.83
18	23	14.47	3.44	0.07	0.63	10.33	28.62
18	25	15.14	3.46	0.06	0.63	10.99	27.37
17	20	15.54	3.47	0.04	0.61	11.42	26.53
17	23	14.74	3.45	0.10	0.61	10.57	28.24
17	25	15.27	3.43	0.06	0.64	11.14	27.08
16	20	16.08	3.46	0.04	0.56	12.02	25.25
16	23	15.41	3.44	0.05	0.53	11.38	26.13
16	25	15.81	3.46	0.04	0.55	11.76	25.59
B20 (20% WFOME and 80% diesel)							
18	20	16.23	3.44	0.04	0.77	11.99	26.13
18	23	15.26	3.45	0.00	0.85	10.96	28.18
18	25	16.09	3.43	0.01	0.77	11.88	26.16
17	20	16.51	3.40	0.06	0.67	12.38	24.99
17	23	15.95	3.47	0.02	0.75	11.72	26.54
17	25	16.37	3.45	0.06	0.66	12.20	25.47
16	20	17.34	3.43	0.05	0.61	13.25	23.58
16	23	16.65	3.46	0.02	0.67	12.50	24.92
16	25	17.48	3.44	0.04	0.63	13.37	23.53
B20 (20% WFOME and 80% diesel)							
18	20	17.63	3.46	0.05	0.71	13.41	23.95
18	23	16.76	3.44	0.10	0.57	12.64	24.56
18	25	17.19	3.45	0.11	0.68	12.96	24.64
17	20	18.07	3.43	0.04	0.70	13.90	23.07
17	23	17.49	3.45	0.05	0.72	13.26	24.18
17	25	17.78	3.46	0.05	0.70	13.57	23.64

16	20	18.80	3.44	0.06	0.57	14.72	21.69
16	23	18.21	3.45	0.06	0.61	14.10	22.59
16	25	18.65	3.45	0.05	0.65	14.50	22.24

as the shaft work or BP, which is maintained constant throughout. The cooling water availabilities for all the CRs studied are very low. Only a maximum of around 0.5% of fuel availability is found to be associated with the cooling water. This is probably because of the lesser increase of engine cooling water temperature during B20 test. The variation of exergy flow through the exhaust gas also has a little effect on CR variation. On the other hand, the availability destruction trend has shown a lowering trend. The increasing trend of shaft availability coupled with diminishing of fuel availability are probably be the reason of the reduction of availability destruction at the circumstances of almost unchanged cooling water and exhaust gas availabilities for CR variation.

The effect of IT on availability balance shows (Fig. 5) that the shaft availability decreases with IT advance and retardation. The mean shaft availabilities at 20 BTDC and 25 BTDC are 4.42% and 3.33% lower than the same at 23 BTDC. B20 having a higher Carbon number than diesel provides lower ignition delay. However, no significant variation is attained from cooling water availability because; the maximum cooling water availability is found as 0.5% of fuel input. The reason is discussed in the earlier paragraph. There is a slight increase in exhaust gas availability is encountered with the increase in IT. This is probably because of the increase in the EGT. The trend of availability destruction for IT retardation and advancement are found to be higher than

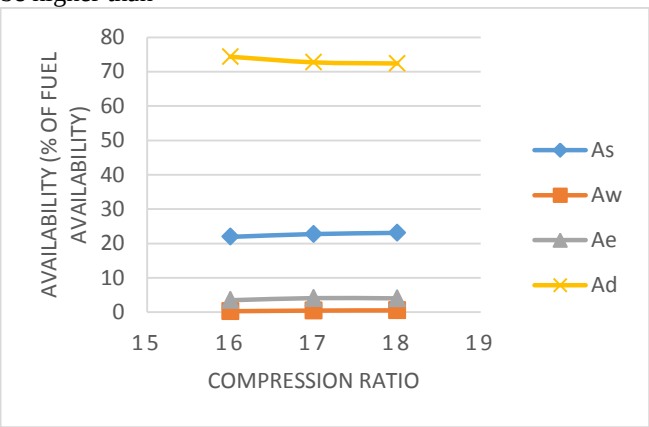

Fig.4. variation in exergy distribution with compression ratio

the standard 23 BTDC of IT. The countable decrease in shaft availability due to IT advancement and retardation comparable to other forms of exergy (cooling water and exhaust flow) is the reason of rise of exergy destruction.

The variations of exergy efficiency with respect to CR and IT are included in Fig. 6. It is clear that B20 provides a better second law (exergy) efficiency with standard IT (23BTDC) at higher CR range. That means a lower compression ratio with standard diesel IT is not much efficient while running B20 in diesel engine as far as the paramount deployment of the rate of available energy of fuel

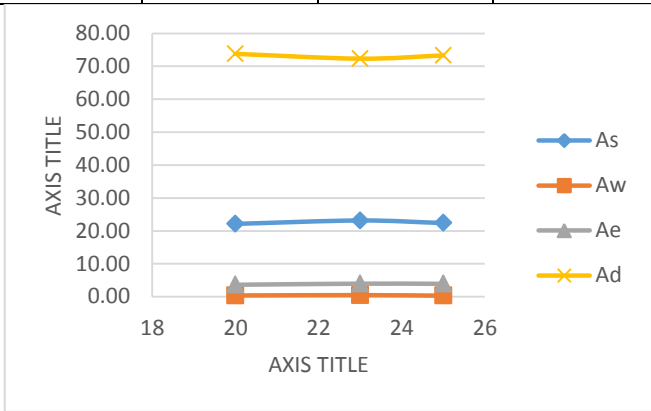

Fig. 5. Variation in exergy distribution with Injection Timing

is concerned. The variation of entropy generation with respect to CR and IT are shown in Fig. 7. The trend of entropy generation is found almost reciprocal of exergy efficiency as expected. The trend suggests that a decrease in the CR increases the entropy generation. Side-by side, 23BTDC is bestowed with lower entropy generation than 20BTDC and 25BTDC of IT.

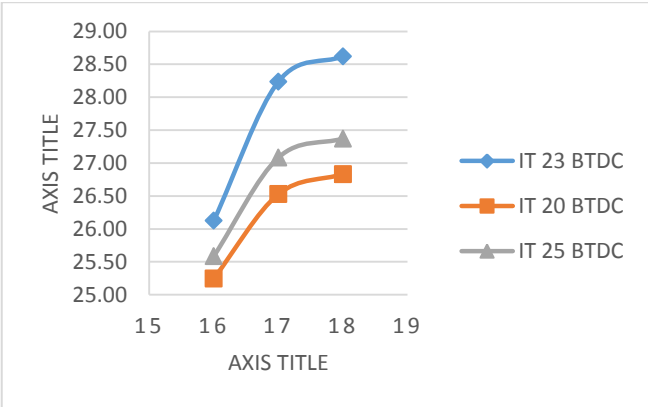

Fig. 6. Effect of injection timing and compression ratio on exergy efficiency.

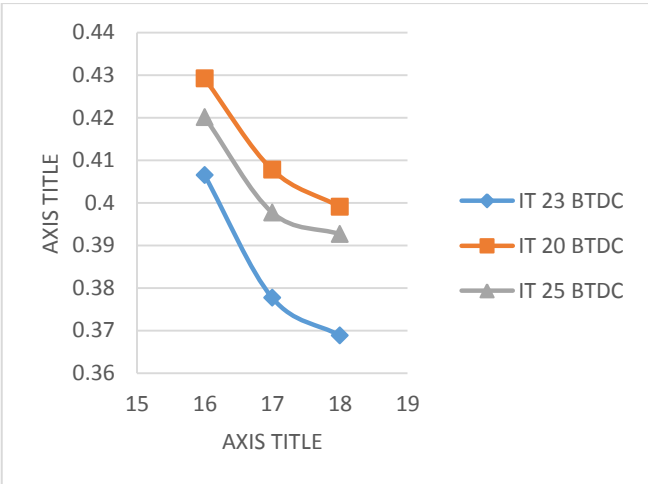

Fig. 7. Effect of injection timing and compression ratio on entropy generation.

VI. CONCLUSION

The thermodynamic analysis (energy and exergy) as a contrivance, is getting attractiveness day by day to judge and control the performance of thermal energy system especially IC engines from a small scale range to robust entity. In the meantime, this type of analysis locates and estimates the energy and exergy distribution, destruction and directs people to find approaches for better available energy management. In this study all the three approaches viz. experimental, analytical (or mathematical) and thermodynamic are applied on a single cylinder, variable compression ratio diesel engine. Combining all results of this study the following conclusion can be drawn.

- The experiments are performed for a set of CR and IT at full load condition. The energy and exergy analyses are inflated for shaft, cooling water, exhaust gas, exergy destruction and entropy formation.
- The analysis demonstrates that WFOME run engine can recover around 26% of the energy supplied by the fuel. Rest of the energy is flown through the cooling water, exhaust gas and other uncounted losses.
- The increase in CR causes a decrease in fuel supply for same BP or shaft power. Therefore, the shaft power per unit fuel supply is increased. The increase in CR also results an increase in cooling water energy flow rate and a reduction in exhaust energy flow rate. However, IT retardation and advancement causes a decrease in shaft energy as a percentage of fuel input. This decreases BTE too for IT retardation and advancement.
- Exergy analysis, on the other hand, has shown that around 30% of the fuel exergy input can be converted to useful means of energy or exergy.
- The shaft availability is increased with the increase of CR and decreases with IT retardation and advancement.
- However, the cooling water and exhaust gas availability variation is found to be very low for both CR and IT variation.
- The exergy destruction is reduced with the increase in CR and increase with IT retardation and advancement.
- Finally, the plot of entropy generation confirms that the increase in CR provide a minor entropy generation and this is found to give a better thermodynamic performance for the WFOME – diesel blend run diesel engine.

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