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Financial, energy, performance and emission analysis of a repurpose used cooking oil (RUCO) diesel fuel blends

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Abstract In this study, exergy, energy, performance and emission analysis were investigated for the repurpose used cooking oil (RUCO), *Jatropha curcas* (JC), *Pongamia Pinnata* (PP) and petroleum diesel fuel (PDF) fueled compression ignition engine under various engine loads. In this study, 20% of each biodiesel was tested in single cylinder, four stroke, diesel engine, given that open literature shows the potential use of biodiesel of up to 20% in a diesel engine without modification. The diesel engine was used to investigate their performance, combustion and emission characteristics of diesel-repurpose used cooking oil, *Jatropha curcas*, and *Pongamia Pinnata* fuel samples at different compression ratios and load condition. The results showed that thermal efficiency is higher with the PDF compared to DRUCO20, DJC20, DPP20 biodiesel blends. The exhaust gas temperature decreased and specific fuel consumption of the engine were increased by adding RUCO, *Jatropha curcas*, *Pongamia Pinnata* to petroleum diesel fuel. Engine ecological analysis showed that blended fuel reduces the average hydrocarbons (HC), carbon monoxide (CO) and NO_x than petroleum diesel fuel. While DRUCO20 showed better performance and reduction in ecological analysis but higher ecological of CO₂ is comparable with DCJ20 and DPP20.

27 **Keywords** Diesel engine. Repurpose used cooking oil. *Jatropha curcas*. *Pongamia Pinnata*.

28 Performance. Ecological analysis. Energy Analysis

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Introduction

Rapid depletion of fossil fuels due to enormous growth of transport sector and release of hazardous emissions from engine tail pipe have gained the attention of many fuel-based industries globally towards novel and renewable based fuels for IC engine applications. Also, the need for reduction of carbon foot print by global emission standards enforced the researchers to search for environmentally friendly fuels without compromising the engine performance. Among the IC engines, diesel engines are considered to be the most favourable one, due to its widespread range of utilizations in various industries such as buildings, transport, agricultural fields and power production (Lloyd and Cackette 2001). The reason for this can be accredited to its high thermal efficiency and greater power production. This resulted in rapid increase of consumption of diesel fuel. But the diesel fuel is one among the non-replenishable and moreover it discharges harmful emissions viz., atmospheric particle pollution, carbon dioxide (CO₂), oxides of nitrogen (NO_x) and sulphur (SO_x) (Agarwal et al. 2006).

In order to overcome these drawbacks many alternate fuels were examined by various researchers viz., derivatives of alcohol, dimethyl ether and biodiesel fuels. Among these alternatives, it has been found that biodiesel fuels are promising one as they release less greenhouse gases, possess high cetane number, replenishable and biodegradable. Further, the biodiesels can be used in the CI engines without any further changes in the prevailing diesel systems (Fugia et al. 2008). Though biodiesel has many advantages, it possesses some significant problems such as low heat content, lower spray properties, inadequate storage stability and cold flow characteristics. These problems can be overcome with the appropriate selection of raw material for the generation of biodiesel which is done by transesterification method. It is vital to study biodiesel as it is capable of diminishing soot discharge from engine tail pipe, because of existence of fuel-bond O₂ and nonexistence of aromatics and

79 sulphur. Moreover, SWOT analysis on biodiesel prospects proposed that it can be appropriate
80 alternate for fossil diesel fuel, as they have better ignition value and 30-70% lesser
81 greenhouse gas discharges (Semwal et al. 2011, Bankovic et al. 2012). Nearly 300 crops are
82 identified which can be capable of producing alternate fuel for engines. Some of the edible
83 oils in use are obtained from Palm, Coconut, Rapeseed and Soybeans (Jayed et al. 2009,
84 Sharma and Singh 2009, Thompson et al. 2010). Due to global food security issues, use of
85 non-edible oils replaced edible ones. Most of the nonedible oils come from Jatropha, Karanja,
86 Neem, Castor, Kusum and Linseed (Atabani et al. 2013, Canoira et al. 2010, Chongkhong et
87 al. 2007).

88 Researchers later focussed on utilization of algae for the generation of biodiesels.
89 Biodiesel from algae was successfully developed in test centres after removal of lipids from
90 the microalgal biomass using organic solvents (Ranjith et al. 2015). The performance of
91 diesel engine using algae biodiesels are in the same range of conventional fuelled diesel
92 engines (Satputaley et al. 2018). Compression ignition engines charged with microalgal
93 biodiesel and conventional diesel blend showed 27.98% reduction in NO_x emissions and
94 30% improved combustion rate (Natarajan et al. 2017). It is reported that algae-based
95 biofuels will show the dominance in fuel industry in the forthcoming decade (Voet al. 2018).
96 These algae can be even cultivated in urban wastewater. Nearly one hundred thousand
97 microalgal species are recognized around the world as stated in algaebase.org and still
98 searching for new potentials of novel types. However, the main disadvantage of all these
99 biodiesel fuels is their unit price, which is nearly twice the unit price of diesels (Bellou et al.
100 2014). Hence current fuel market is still in favour of conventional fossil fuels. Traditionally
101 optimization of various processes involved in fuel and chemical industries were based only
102 on energy analysis, as this is simple and cost-effective technique. But this method is not
103 adequate to point out whether the energy is utilised efficiently or not. Also, this technique is

104 not suitable for comparing two systems in order to choose better one (Jehad et al. 2018). It is
105 the exergy analysis which provides the thermodynamic particulars of various operations.
106 Exergy analysis gives meaningful comparison between the processes. It also aids in designing
107 new systems and optimising the existing designs. The main attribute of exergy analysis is that
108 it pinpoints the thermodynamic details of processes viz., types, causes and location of losses
109 more rationally than former energy analysis (Dincer and Rosen 2011).

110 Hence, from past two decades researchers focussed on conducting exergy analysis of CI
111 engines charged with different biodiesels in order to identify the thermodynamic losses
112 involved in various processes. The performance of a diesel engine was optimised with the aid
113 of exergy and emission analyses (Azoumahet al. 2009). From exergy analysis, it was found
114 that combustion process was the major reason for the irreversibility in a turbocharged CI
115 engine charged with palm oil biodiesel (Pedro et al. 2009). The II law efficiency of a CI
116 engine fuelled with high oleic methyl-ester was enhanced with the drop of dead state
117 temperature (Hakan et al. 2009). On the basis of exergy analysis results, olive-pomace oil
118 blended with biodiesel was chosen to be an alternative for the conventional diesel fuel in
119 compression ignition engines (Lópezet al. 2014). But exergy studies on CI engines fuelled
120 with biodiesel were less in number compared to that of other research areas (Luis2013).
121 Hence it is important to perform exergy analysis along with the search for replacements for
122 fossil fuels in CI engines. Dr. M. M. Kutty, the Secretary of Petroleum aurpraktik gas
123 mantralaya (the Ministry of Petroleum and Natural Gas) of India, as a part of Swachhata
124 Pakhwada (cleanliness fortnight), encouraged researchers to do research in the field of
125 biodiesel on 2nd October 2019, at the event of Rashtriya Swachhta Diwas (National
126 Cleanliness Day). Even the Chief Minister of Gujarat Mr. Vijay Rupani launched a mobile
127 application named RUCO (repurpose used cooking oil) with joint initiative of biodiesel union

in India (BDAI) for gathering waste cooking oil from the restaurants and food handling units and transform it into bio-fuel (FSSAI 2018).

Hence in the current work, RUCO biodiesel is compared with pure diesel and other biodiesel blends generated from *Jatropha* (*Jatropha Curcas*) and *Karanja* (*Pongamia Pinnata*) oil by conducting a thorough investigation of performance, combustion, emission and exergy analysis in a single cylinder, 4-stroke CI engine at varying compression ratios (16.5, 17.5 and 18.5). All the biodiesel blends were produced at 20% concentration by volume to the diesel. As far as author's attention, lesser study has carried out such a comprehensive investigation on these biodiesel blends and hence the results of our study will be of greater interest to the researchers in this field.

Experimental Protocol

Test Fuels and their Properties

The standard fuels used in this investigational work are repurpose used cooking oil (RUCO), *Jatropha curcas* (JC), *Pongamia Pinnata* (PP) biodiesel and petroleum diesel fuel (PDF) fueled. The fuels are collected from local market respectively. The RUCO alternative fuel is produced by transesterification of cooking oil and the complete process is conducted in the IC engine laboratory at Maulana Azad National Institute of Technology, Bhopal. Earlier study done by the scholars stated that RUCO substitute fuel has up to 20% with diesel base fuel shown better engine characteristics. Hence, in the present investigation RUCO, JC, PP blend proportion is fixed at 20% by basis of volume with PDF has been used. In following blends, the repurpose used cooking oil, JC and PP alternative proportion is raised by 20%, thus making blends with 80% PDF. These blends are termed as RUCO20 (Containing 80% PDF and 20% repurpose used cooking oil alternative), JC20 (Containing 80% PDF and 20% *Jatropha curcas* alternative), and PP20 (Containing 80% PDF and 20% *Pongamia Pinnata*

alternative). The characteristics of the PDF and blends samples are indicated in Table 1. The PDF, RUCO20, DJC20 and DPP20 blends are shown in Fig.1.

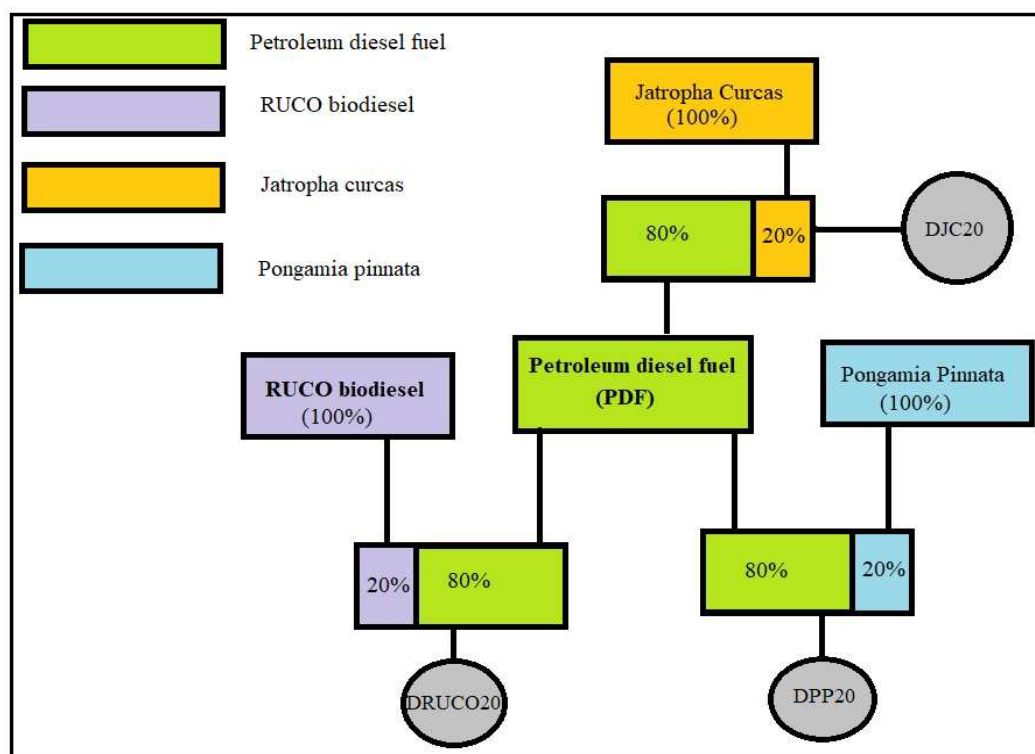


Fig.1 Tested diesel-biodiesel-ethanol Blends

Table 1 Test fuel properties

Fuel	PDF	RUCO	JC	PP	ASTM standards
Lower heating value (MJ/kg)	42.6	36.6	39.64	38.91	ASTM D240
Viscosity at 40°C (mm ² /s)	3.58	4.76	5.11	5.71	ASTM D445
Density (kg/m ³)	830	897	875	880	ASTM D1298
Cetane number	48-52	53.0	52.3	52.0	ASTM D976
Flash point (°C)	75	90	162	163	ASTM D92
Fire point (°C)	79	93	-	-	ASTM D92

Cost Analysis

Although biodiesel has many advantages, the main concern pertains to the usage of biodiesel is its primary cost, which relies on economics of petroleum product, feedstock and transport. The total production price for RUCO, JC and PP substitutes rely on the yield of seed, oil

content, and quality. It is obvious from the examination that the price of 1 litre biodiesel is more than that of the conventional diesel by 3 to 5 %. Estimated cost analysis of RUCO, JC and PP alternative as represented in Table 2, Table 3 & Table 4.

Table 2 Making cost of 1 litre of PP alternative

Particular	Quantity	Rate (Rs.)	Amount
Karanja seeds	3 kg	20	60
Transportation cost	4 kg	4	16
Chemical charges	-----	-----	15.0
Electricity	2.00 units	8	16.0
Manpower (1 day), helpers (2)	2	6	12.0
Total			119.0
By-product Income			
Seed Cake	1.7 kg	15	25.5
Glycerin	-----	-----	20.5
Total			46.0
<u>Net production cost of biodiesel for 1 litre</u>			
Total biodiesel manufacturing cost			119.0
By-product Income			46.0
Total			73.00/-litre.

Table 3 Production cost of 1 litre of JC alternative

Particular	Quantity	Rate (Rs.)	Amount
Jatropha seeds	5 kg	6	30
Transportation cost	5 kg	2	10
Chemical charges	-----	-----	10
Electricity	1.34 units	8.0	10.7
Manpower (1 day), helpers (1)	1	6	6
Total			66.7/-litre.

Table 4 Making cost of 1 litre of RUCO alternative

Particular	Quantity	Rate (Rs.)	Amount
Waste Cooking oil	1 litre	45	45
Transportation cost		5	5
Chemical charges	-----	-----	10
Electricity	1.30 units	8.0	10.4
Manpower (1 day), helpers (1)	1	6	6
Total			76.4/-litre.

To read the engine load value, a Kistler equipment, which magnetically measures the load, is used. With gathering the relevant data from the tests, the SFC, and BTE are calculated using Equations 1-3.

$$Pe = \frac{2\pi \cdot \omega \cdot T}{1000} \quad (1)$$

$$SFC = \frac{\dot{m}_f \cdot 10^6}{Pe} \quad (2)$$

$$BTE = \frac{3600}{BSFC \cdot LHV} \cdot 100 \quad (3)$$

All symbols in the previous equations are explained in the nomenclature part of the manuscript. A k-type thermocouple is employed to read the EGT from the tailpipe. Engine speed, crank angle, and top dead centre are recorded with the aid of Kubler encoder. Besides, the in-cylinder pressure values are measured with air cooled piezoelectric sensor. After this sensor reads the in-cylinder pressure value, it synchronously sends it to combustion analysis software. In the study, software analysis to draw the combustion data is used, and each curve given in the combustion graphs represents the mean of more than 1000 cycles for this study.

After recording the in-cylinder pressure values, the heat discharge rate curves are calculated from the Eq. (4) in harmony with the first law of thermodynamics.

$$\frac{\partial Q_n}{\partial \phi} = \frac{k}{k-1} P \frac{\partial V}{\partial \phi} + \frac{1}{1-k} V \frac{\partial P}{\partial \phi} \quad (4)$$

The symbols in equation (1) is explained in the nomenclature part of the paper. Here, the authors accept that there is no gas leakage from the valves or segments during the experiments, and the gas in the cylinder is accepted as ideal. The investigational setup is as indicated in Fig.2. The investigational setup specifications are given in Table 5.

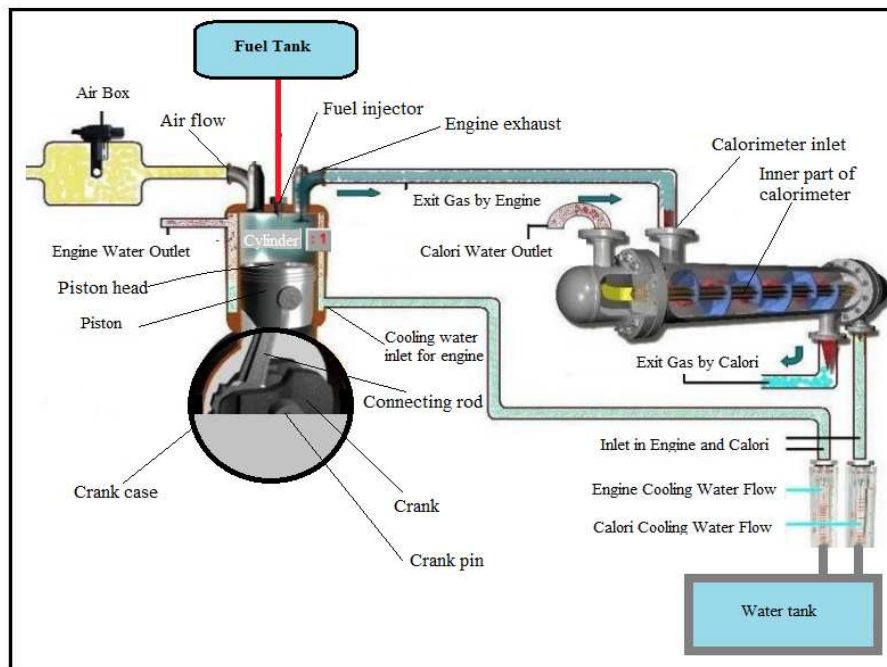


Fig. 2 Experimental rig

Table 5 Specifications of Test Engine

Parameters	Value
Engine cylinder/stroke	1/4
Higher than fuel injection pressure	220 bar
Rated speed	1500 rpm
Bore/stroke	80/110 mm

Injection timing	23.5° b TDC
Compression ratio	15.5-18.5
Method of cooling	Water

An uncertainty analysis is of great importance for the experimental studies because its result gives an idea about the accuracy and repeatability of the presented results to the readers. Therefore, a total uncertainty value of the results is calculated using the following equation according to the data given in Table 6. The explanation of the symbols is given in the nomenclature part of the paper.

$$U_R = \left[\left(\frac{\partial R}{\partial x_1} u_1 \right)^2 + \left(\frac{\partial R}{\partial x_2} u_2 \right)^2 + \dots + \left(\frac{\partial R}{\partial x_n} u_n \right)^2 \right]^{1/2} \quad (5)$$

Table 6 Uncertainties

Apparatus	Uncertainty
Temperature transducer	±0.15%
Speed indicator	±1.0%
Load gauge	±0.2%
Pressure transducer	±0.5%
Crank Angle encoder	±0.2%
FGA	
CO ₂	± 1.0%
NO _x	± 0.5%
Smoke meter	± 1.0%
Burette fitted measurement	±0.5%
Exhaust gas calorimeter	-
Heat value measurement	±1.0%

208 Exergy analysis considerations

209 Exergy investigation has been carried out from I & II law of thermodynamics. In this
210 investigation total energy can be utilized in various ways. The work from engine consumes
211 the whole power of fuel, which takes away the heat by the tail pipe emissions. Heat transfer
212 losses by cooling medium and unaccounted losses are also considered. Energy losses in the
213 engine is calculated by using subsequent equations (Eq. 6 to 10). The total energy efficacy of
214 the system enhanced by improving the energy distribution. The following assumptions are
215 made:

- 216 • Steady state engine run throughout the examination.
- 217 • Total engine is considered as control volume.
- 218 • The ideal gas is formed in the combustion chamber.
- 219 • Neglect the influence of changes in potential and kinetic energy.

220 Fuel provided to the engine is as given in Eq, (6)

$$221 \quad Q_{supplied} = m_f \times LCV, kW \quad (6)$$

222 Where m_f is the rate of fuel provided in kg/Sec, LCV indicates lower heating value

223 Brake power is stated by Eq. (7).

$$224 \quad Q_{brake\ power} = 2 \times \pi \times N \times T \quad (7)$$

225 Where, N is the crankshaft rpm and T is the torque induced (kN-m)

226 Cooling water heat loss is given by Eq. (8)

$$227 \quad Q_{cooling\ water} = C_{pw} \times m_{we} \times (T_2 - T_1) \quad (8)$$

228 Where m_{we} is the water mass flow rate in the engine in kg/Sec, C_{pw} specific heat of water
229 in kJ/Kg. K, T_1 is the water temperature at inlet, whereas and T_2 is the exit water temperature
230 from engine ($^{\circ}$ C).

231 Heat taken away by the tailpipe gases is expressed by Eq. (9)

$$232 \quad Q_{exhaust\ gas} = C_{pe} \times m_{eg} \times (T_5 - T_a) \quad (9)$$

Where m_{eg} is the rate of discharge gases from engine tail pipe in kg/Sec, C_{pe} is the specific heat of the discharge gas (kJ/Kg. K), T_5 is the temperature of discharge gas at calorimeter (K) T_a is the atmosphere temperature (K). Not accounted energy losses can be examined by Eq. (9) by considering subsequent facts: The engine power developed employed to run pump, water circulation, pump lubrication.

$$Q_{unaccounted} = Q_s - (Q_{bp} + Q_{cw} + Q_{eg}) \quad (10)$$

Where $Q_{unaccounted}$ is the unaccounted energy losses (kJ) while Q_s , Q_{bp} , Q_{cw} , and Q_{eg} are heat supplied, brake power, cooling water and heat taken away by tailpipe gas respectively.

Energy analysis is indicated in Fig.3 (a, b and c).

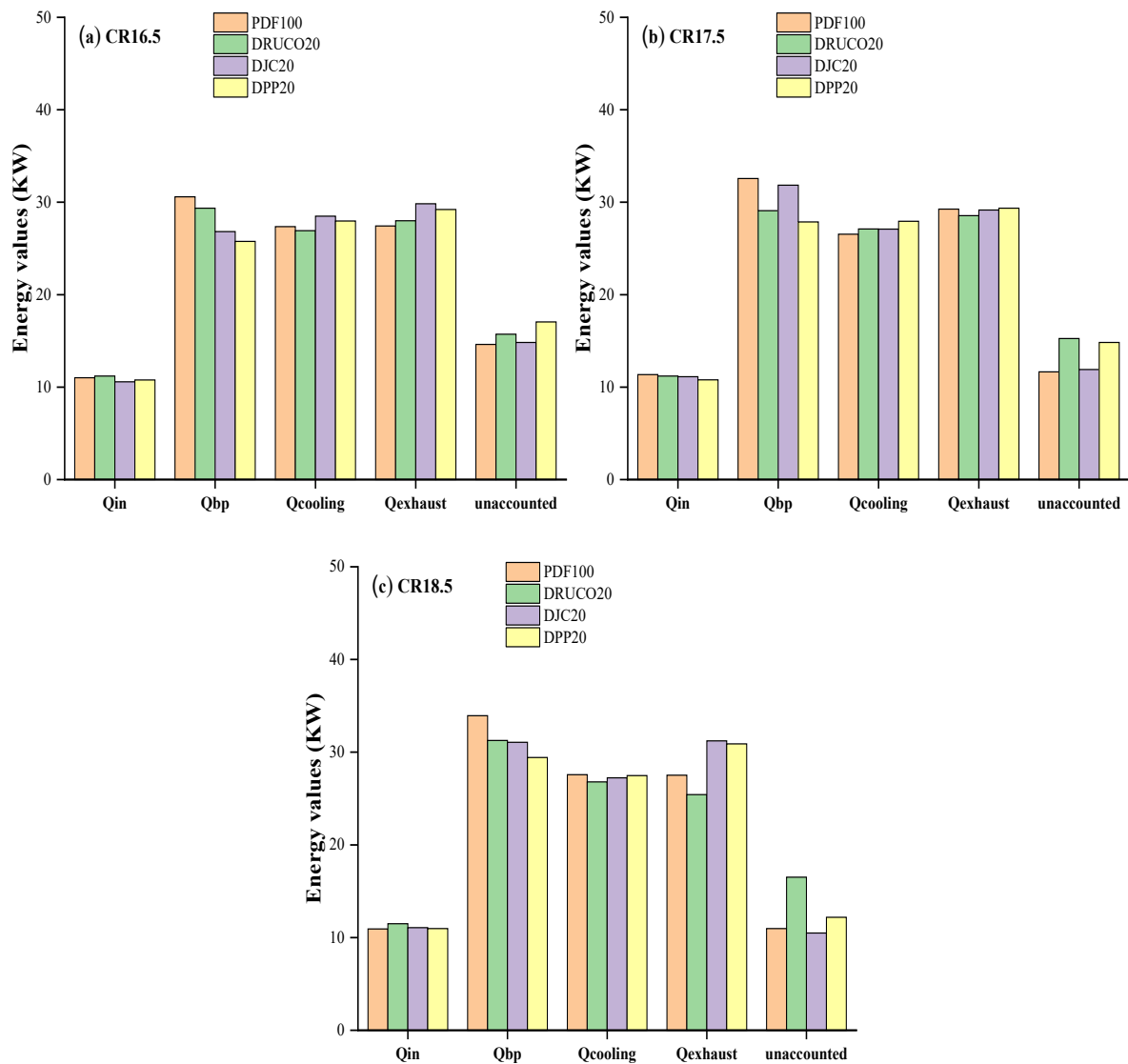


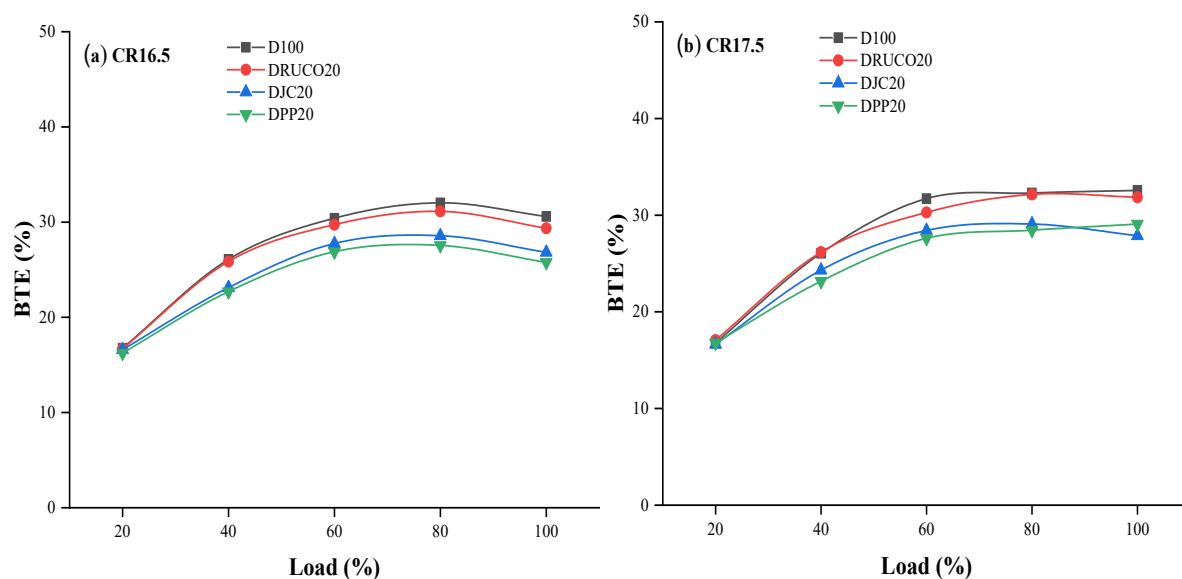
Fig.3 Energy values analysis for a) CR16.5, b) CR17.5 and c) CR18.5

Results and discussions

Performance breakdown

Brake thermal efficiency

BTE is the engine's brake power which depends on total heat energy from the fuel, which is the equivalent to product of lower heating value and fuel flow rate (Pali et al. 2015). Fig. 4 indicates the variations in BTE values in relation with load for pure diesel and biodiesel blends at 16.5, 17.5, and 18.5 CRs. It can be clearly seen that increasing compression ratios from 16.5 to 18.5 would not significantly rise the BTE. Also, above 60% of the engine load, the BTE for all the tested fuels are relatively the same. In general, the BTE enhanced with the rise of engine loads for all the fuels. At high loads, the gaseous fuel air mixture is rich due to high pilot quantity, thus the rise of combustion rates and BTE. It is important to note that D100 shows the highest value of BTE compared to DRUCO20, DJC20, DPP20 as depicted in Fig. 4. Although biodiesel blends contain more oxygen content that could result in higher BTE as a result of enhanced combustion efficiency as reported (Gharehghani et al. 2019), the lower calorific value of DRUCO20, DJC20 and DPP20 would result in relatively higher fuel consumption, thus deteriorating their brake thermal efficiencies. However, among the biodiesel blends, DRUCO20 has the highest BTE with its value being close to pure diesel.



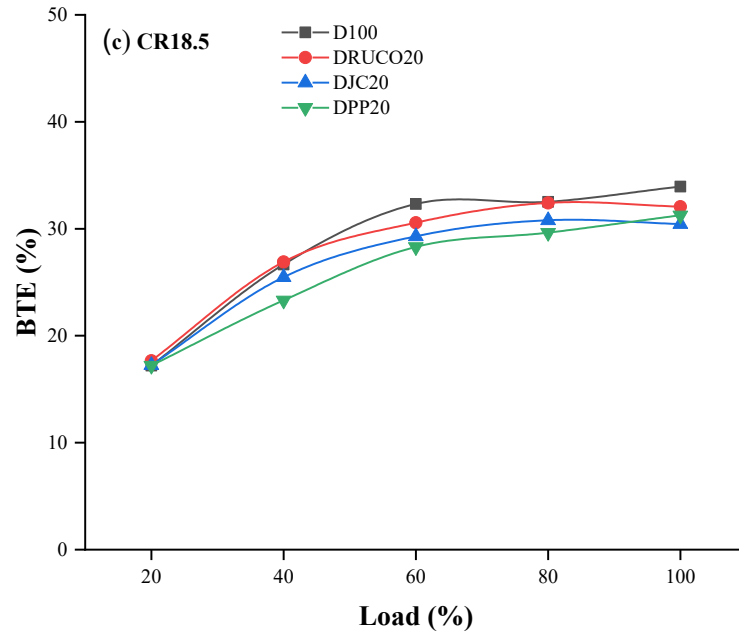


Fig.4 BTE against engine load for various CRs

Specific fuel consumption

The SFC of an engine denotes the efficiency to convert the energy in chemical form of a fuel into the work. A low fuel consumption indicates that the fuel energy is utilised efficiently and at a lower combustion temperature owing to a higher air-fuel ratio. Consequently, fuel with improved SFC having a lower combustion temperature will reduce the amount of heat transferred to the coolant. This would be beneficial to downsize the radiator so that smaller cooling fan could be used in an engine. Fig. 5 indicates the changes in SFC values in relation with load for pure diesel and biodiesel blends at three compression ratios 16.5, 17.5, and 18.5. Overall, the SFC diminished with the rise in the engine loads. This is possibly attributed to high in-cylinder temperatures at 100% engine loads, thus increasing engine efficiencies and lessening fuel consumption (Veza et al. 2019). At full load for all the CRs, the pure diesel gave the lowest SFC among the tested fuels. This is because the pure diesel has the highest lower heating value (42.5 MJ/kg) compared to the biodiesel blends as can be seen in Table 1 (Tan et al. 2017). As a result, less fuel is enough to achieve same engine power as of using pure diesel.

Enhancing the CR does not seem to improve the value of SFC significantly as shown in Fig. 5. However, it is noteworthy to observe that at the highest compression ratio (18.5), DPP20 gives the highest SFC compared to pure diesel and other biodiesel amalgamations tall engine loads. Note that the SFC and BTE have inverse relationship (Veza et al. 2021), and this inverse tendency is obvious from Figs. 4 and 5. In Fig 4, it is shown that DPP20 gives the lowest value of BTE, while in Fig. 5, it is depicted that the DPP20 gives the highest of SFC, confirming the inverse trend between SFC and BTE. Therefore, lower the SFC, greater the BTE. Also, as DRUCO20 gives the second highest BTE as discussed previously (Fig. 4), its BSFC confirms the inverse BTE-SFC trend by giving the second lowest BSFC among all the tested fuels as depicted in Fig. 5

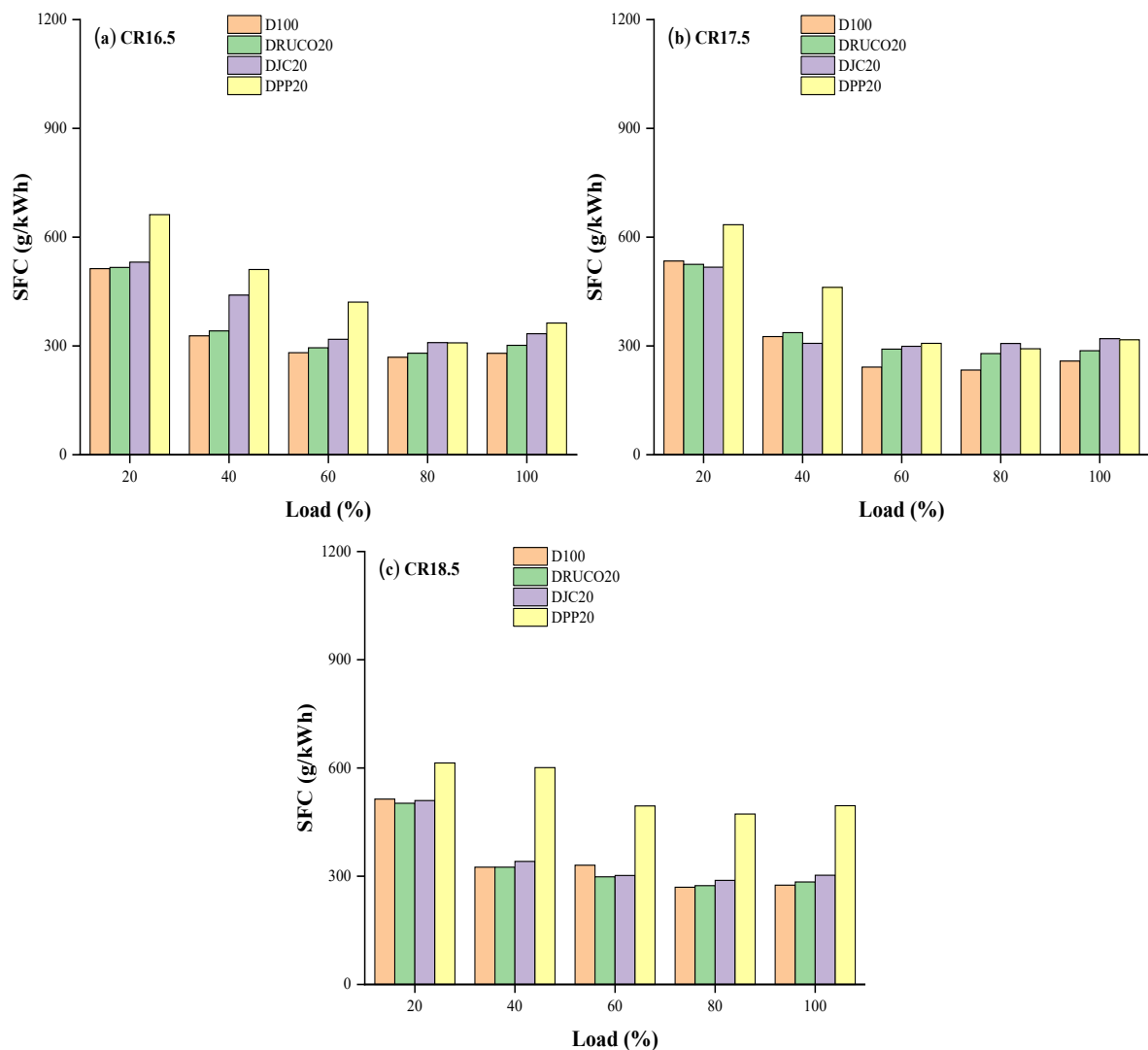
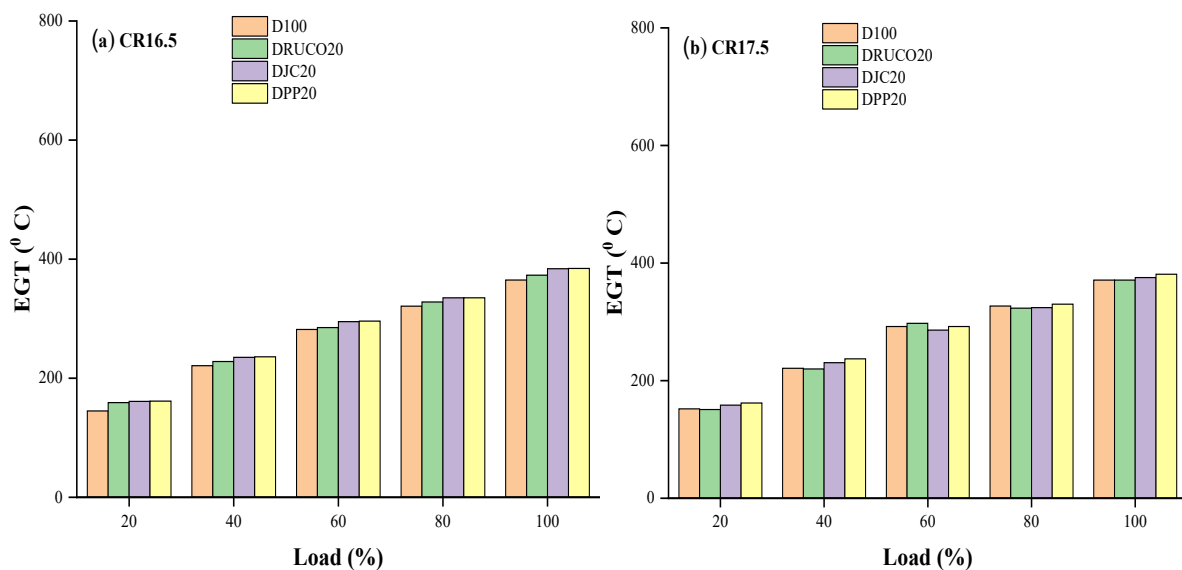


Fig.5 SFC against engine load for various CRs

Exhaust gas temperature

EGT indicates the quantity of heat energy transported by the exhaust gasses. Therefore, for an efficient engine, low EGT is desired. However, to reach the standards of stringent emission policy, the development of after treatment techniques such as selective catalytic reduction (SCR), diesel oxidation catalyst (DOC) and diesel particulate filter (DPF) requires a least EGT of around 200°C for catalyst light-off since the conversion efficacy of such after treatment techniques rely greatly on the temperature of tailpipe gas (Guan et al. 2018, Nanthagopal et al. 2019). Fig. 6 depicts the changes in the ECT values in relation with engine load for D100, DRUCO20, DJC20 and DPP20 at 16.5, 17.5 and 18.5 CRs. As the engine loads increases, the tailpipe gas temperature rises for all the tested fuels. Nearly the same trends were also observed for the 3 CRs. Furthermore, except for the 20% engine load, a minimum EGT of approximately 200°C for the most operating conditions were achieved. The burning of biodiesel blends was seen to enhance the EGT than that of pure diesel but the variances among the tested fuels were insignificant for each CR. With the rise in CR, EGT reduced slightly. As the CR inclined, the temperature at the start of compression stroke would lessen owing to the reduction in burnt gases mass. Therefore, the EGT will drop at high CRs and as a result lower temperature is found at the start of the compression stroke.



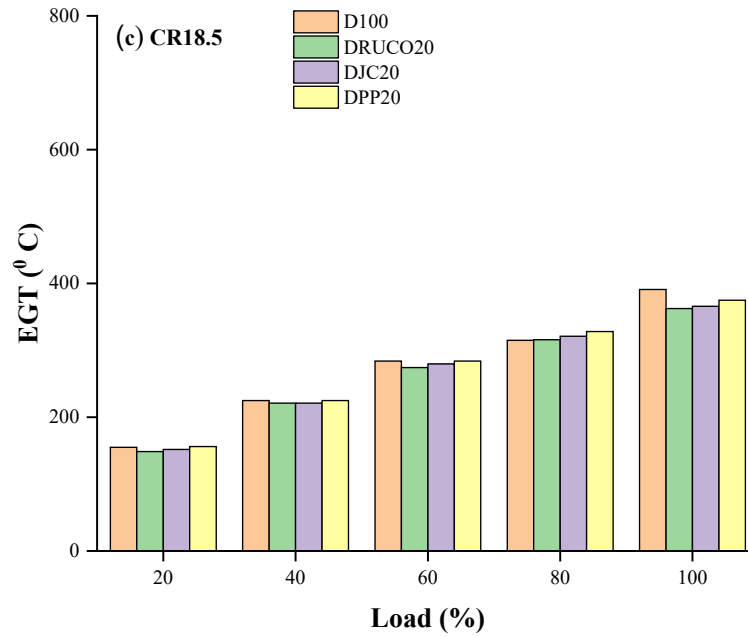


Fig.6 EGT analysis for all test conditions

Ecological analysis

Ecological analysis of CO

CO emissions from diesel combustion are greatly rely on the A/F ratio. Insufficient O₂ is the major reason for the CO formation. Fig. 7 shows that the CO emissions were relatively the same for the all-test conditions. At the maximum load, highest CO emissions are seen for all the fuels. The insignificant changes in CO emissions among DRUCO20, DJC20 and DPP20 were down to their comparable levels of O₂ content. Moreover, it is also noticed from Fig. 7 that biodiesel blends produced relatively smaller CO emissions than that of pure diesel. The highest reductions in CO emissions were found to be at 60% engine load for DRUCO20, whereas the highest value of CO exhaust was given by pure diesel at maximum loads for all 3 CRs.

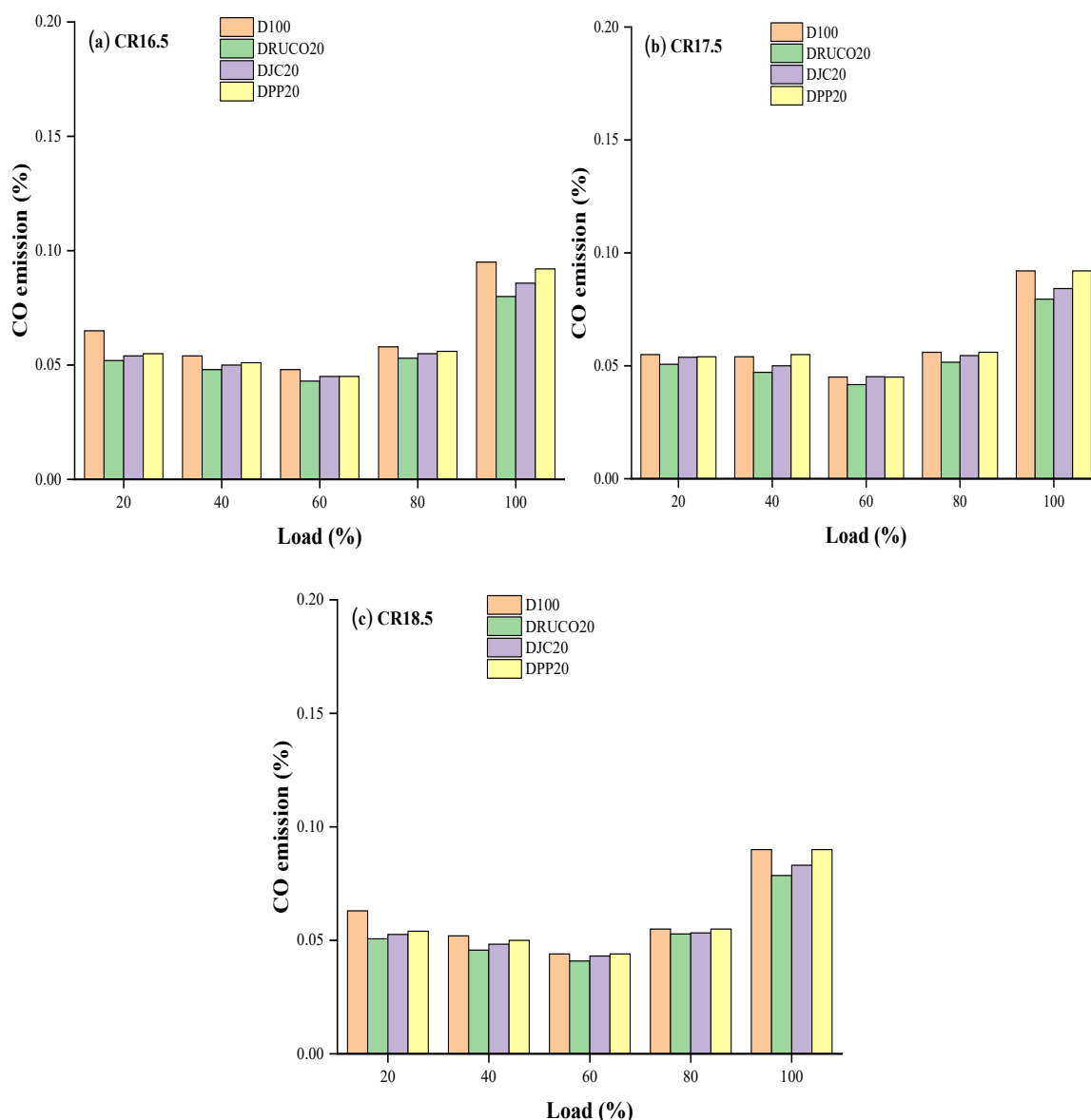


Fig.7 CO ecological analysis against engine load for 3 CRs

Ecological of HC

It is generally agreed that the HC emissions are caused by either rich or lean mixture regions that results in unburned hydrocarbon. Overall, HC emissions give analogous trends for all the CRs as depicted through Fig. 8. For all the 3 CRs, the greatest reductions in HC emissions were accomplished by DJC20 at 50% engine load, while the maximum HC emissions were given by D100 under maximum load condition. Like CO emissions, the HC emissions for all biodiesel blends are lesser than the base fuel (pure diesel) at all CRs. The presence of O₂ and absenteeism of aromatics in biodiesel are the causes for lesser HC emissions with the aid of

DRUCO20, DJC20 and DPP20. The intrinsic molecular oxygen in biodiesel blends facilitates more complete combustion resulting in lower HC emissions.

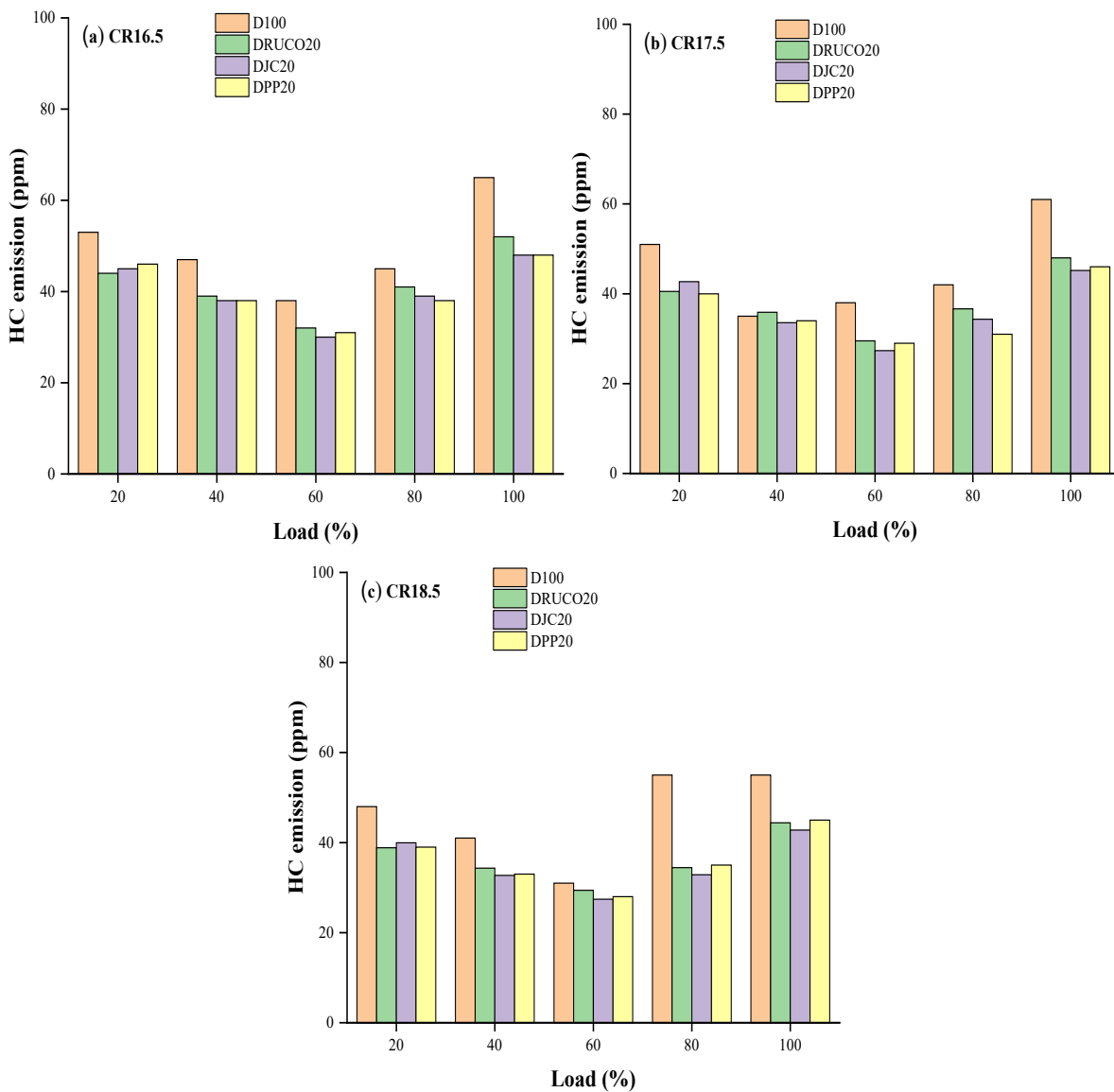
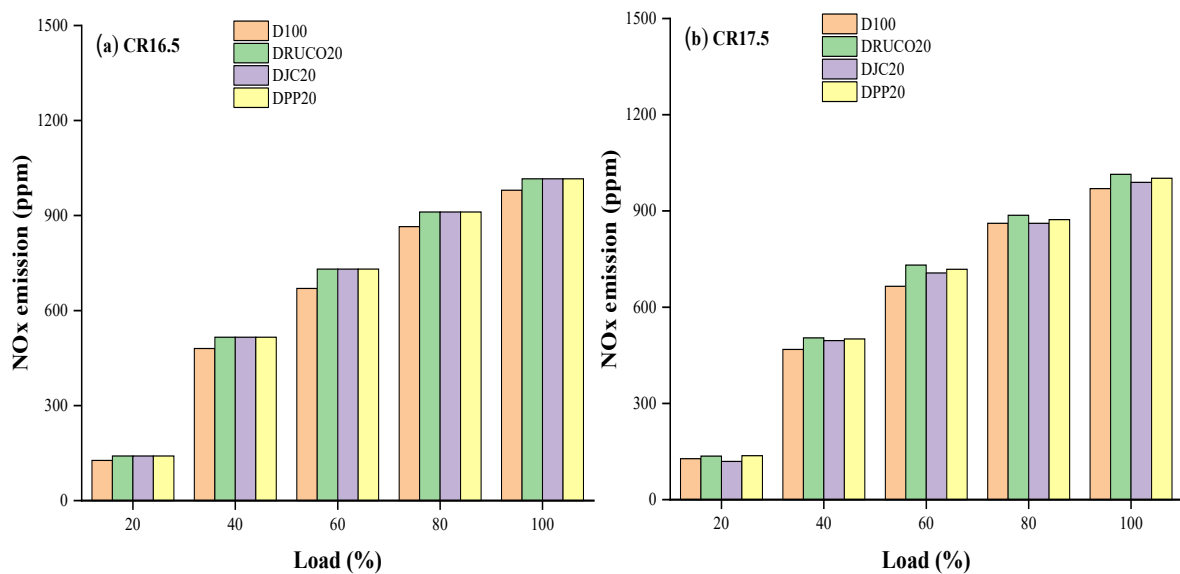


Fig.8 HC ecological analysis against engine load for all CRs

Ecological analysis of NO_x

NO_x emission in a CI engine sowing to its high in-cylinder temperature. It is known that the NO_x formation relies on the combustion temperature where the NO_x formation rates rises exponentially once combustion temperature reaches above 1800 K. It is identified from Fig. 9 that NO_x emissions enhances with rise in engine loads for all the tested fuels. The figure also shows that biodiesel blends emit slightly higher NO_x emissions than pure diesel for all

test conditions. The increased level of NO_x emissions in DRU20, DJC20 and DPP20 is attributed to the additional oxygen content in biodiesel which enhance the combustion in the chamber, thus increasing the in-cylinder temperature. As a consequence, high NO_x emissions for biodiesel combinations are noticed than those of diesel fuel. These findings are in reasonable harmony with those attained (Nabi and Rasul2018, Nanthagopal et al. 2019). Note that at 20% of engine loads at CR16.5, CR17.5 and CR 18.5, the NO_x emissions for all the tested fuels are significantly low because of lesser in-cylinder temperature at lower engine load. As the load enhances, the A/F ratio caused a significant rise in the gas temperature inside the combustion chamber, thus significantly increasing the NO_x formation, which is known considerably sensitive to the rise in the combustion temperature. Therefore, the highest amount of NO_x emissions was generated at the maximum engine load.



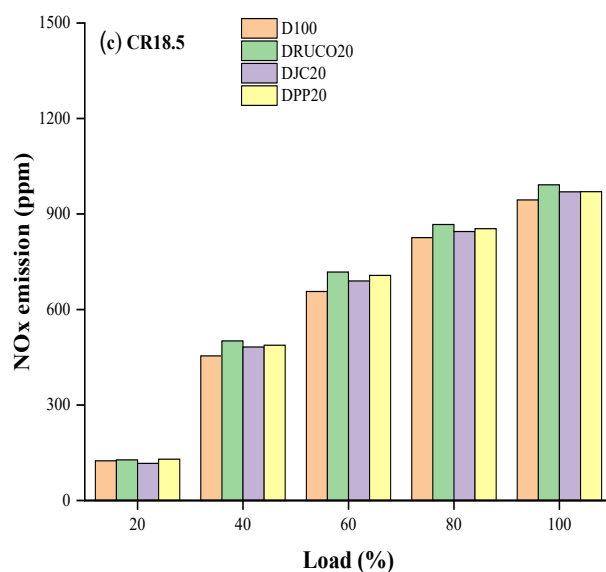


Fig.9 NO_x ecological analysis against engine load at all 3 CRs

Ecological analysis of CO₂

CO₂ emissions are the major sources of global climate change. In the last 200 years, the worldwide concentration of CO₂ has raised by 31.4% (Alami et al. 2020). Increased level of carbon dioxide in the atmosphere from transportation and industrialisation leads to the higher sea level, severe floods and droughts (Mokhtara et al. 2020, ckbarali et al. 2017). Overall, CO₂ emissions raised with mounting engine loads for all the tested fuels as depicted in Fig. 10. The figure also shows that biodiesel blends produced relatively higher CO₂ emissions than pure diesel for all test conditions. At CR16.5 and 17.5, DJC20 gives the highest CO₂ emissions of all the tested fuels except for 20% load at CR16.5 where pure diesel gives the highest values. At 18.5, the highest level was given by DRUCO20 with approximately 3% concentration of CO₂. This is because the molecular oxygen found in biodiesel promotes more complete combustion, thus increasing the conversion of CO into CO₂. It interesting to note that although DRU20, DJC20, DPP20 emits more CO₂ than pure diesel (D100), the biodiesel blends are expected to have net negative CO₂ emission owing to its renewable nature. The CO₂ released from the CI engine is absorbed by the trees of biodiesel feedstocks.

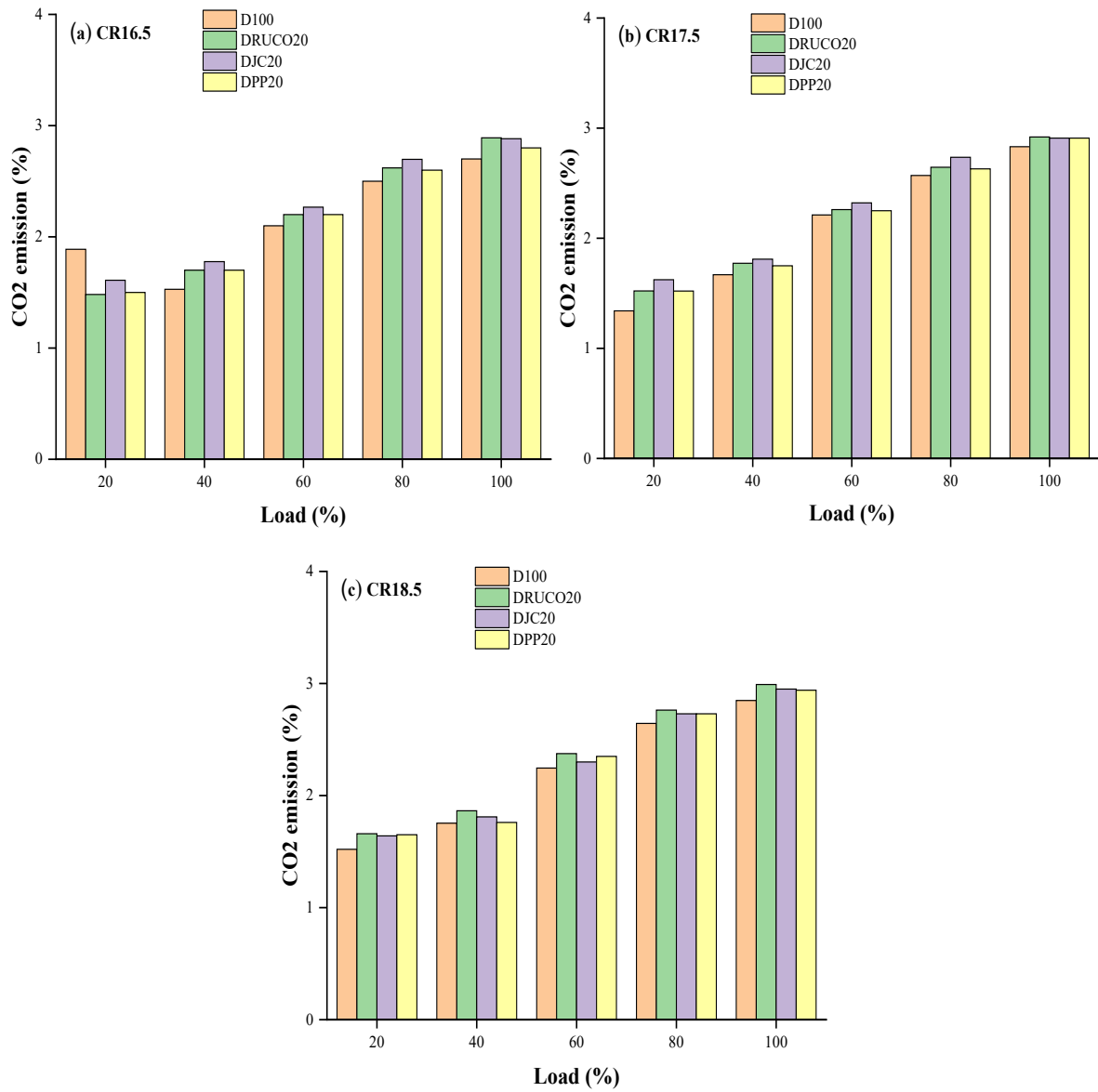


Fig.10 CO₂ ecological analysis against various engine loads for CR16.5, CR17.5 and CR18.5

397 **Conclusion**

398 Several vital issues regarding financial, energy, performance and tail pipe emissions of CI
399 engine charged with various biodiesel blends are examined under various engine loads and
400 CRs (16.5, 17.5 and 18.5) and subsequent inferences are drawn:

401 The BTE of all the investigated fuels did not significantly increased with a rise in CR, but it
402 improved with enhancement in engine loads for all 3 CRs with D100 showing the higher
403 value compared to DRUCO20, DJC20 and DPP20 throughout under all load conditions. Of
404 all the biodiesel blends, DRUCO20 shown the highest BTE with its value being close to
405 D100. In terms of SFC, rising the CR did not substantially improve fuel consumption. At
406 18.5 CR, the DPP20 gave the highest SFC value for all engine loads. The second lowest
407 BSFC among all the tested fuels was given by DRUCO20. The SFC shown an inverse
408 relationship with BTE.

409 The biodiesel blends were found to enhance the EGT than the pure diesel in spite of the
410 variances among the tested fuels were insignificant for each CR. Except for the 20% engine
411 load, at all test conditions a minimum EGT of 200°C was identified.

412 The CO emissions at different test conditions were compared. At 100% load, the higher CO
413 emissions were recorded for all test conditions. Biodiesel blends emitted relatively lesser
414 level of CO emissions than those of pure diesel. The most significant lessening was achieved
415 by DRUCO20 at 60% engine load, while the highest rise of CO was given by D100 at full
416 loads for all the three CRs.

417 Alike CO emissions, the HC emissions for all biodiesel blends were also lesser than D100.
418 Under all test environments, the peak lower HC emissions was attained by DJC20 at 50%
419 load, while the peak higher HC discharge was given by D100 at 100% load state.

420 The NO_x emissions were noticed to rise with enhancing engine loads for all the test fuels.
421 Compared to pure diesel, biodiesel blends emit slightly higher NO_x discharge for all test

conditions. The higher level of NO_x emissions in DRU20, DJC20, DPP20 are owing to the additional oxygen content in biodiesel which enhance the combustion in the cylinder, thus elevating the in-cylinder temperature and NO_x exhaust.

As regards CO₂, biodiesel blends produced higher amount of CO₂ than D100. At CR16.5 and 17.5, DJC20 gave the highest CO₂ level except for the 20% load condition at CR16.5 where D100 showed the highest values. At 18.5, the highest level was given by DRUCO20 with approximately 3% concentration of CO₂. Overall, the inherent molecular oxygen found in biodiesel facilitates complete combustion, thus improving the CO conversion into CO₂.

447 **Ethical Approval**

448 All the author(s) agrees to the guidelines provided by the Journal. This research has
449 considered all ethical issues associated with the methodology being employed.

450 **Consent to participate**

451 Not applicable

452 **Consent for publication**

453 The Author(s) permits the publisher to publish this research article in this esteemed Journal.

454 The Corresponding Author gives consent from individuals to publish the data associated with
455 this research article.

456 **Availability of data and materials**

457 This research article doesn't contain any data (published) or datasets and materials generated
458 during the study.

459 **Conflict of interest**

460 The authors declare that they have no conflict of interest.

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463 **Authors' contributions**

- **Upendra Rajak:** Methodology; Writing original draft.
- **Abhishek Dasore:** Literature Review.
- **Pankaj Shrivastava:** Editing the Manuscript.
- **Ümit Ağbulut:** Reviewing the Manuscript.
- **Suat Sarıdemir:** Reviewing the Manuscript.
- **Ibham Veza:** Editing the Manuscript.
- **V. Nageswara Reddy:** Editing the Manuscript.
- **Tikendra Nath Verma:** Reviewing & final drafting.

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500 Nomenclature

BTE	Brake thermal efficiency
SFC	Specific fuel consumption
PDF	Petroleum diesel fuel
CI	Compression ignition
EGT	Exhaust gas temperature
HC	hydrocarbons
CO	carbon monoxide
NO _x	Nitrogen oxides
CO ₂	Carbon dioxide
RUCO	repurpose used cooking oil
JC	Jatropha curcas
PP	Pongamia Pinnata
DRUCO20	20% repurpose used cooking oil and 80% Petroleum diesel fuel
DJC20	20% Jatropha curcas and 80% Petroleum diesel fuel
DPP20	20% Pongamia Pinnata and 80% Petroleum diesel fuel
U _R	Total uncertainty
n	Polytropic index
Pe	Brake power (kW)
Ø	Crank angle
ω	Angular velocity, 1/s
U _n	Independent variables
T	Torque, Nm

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Figures

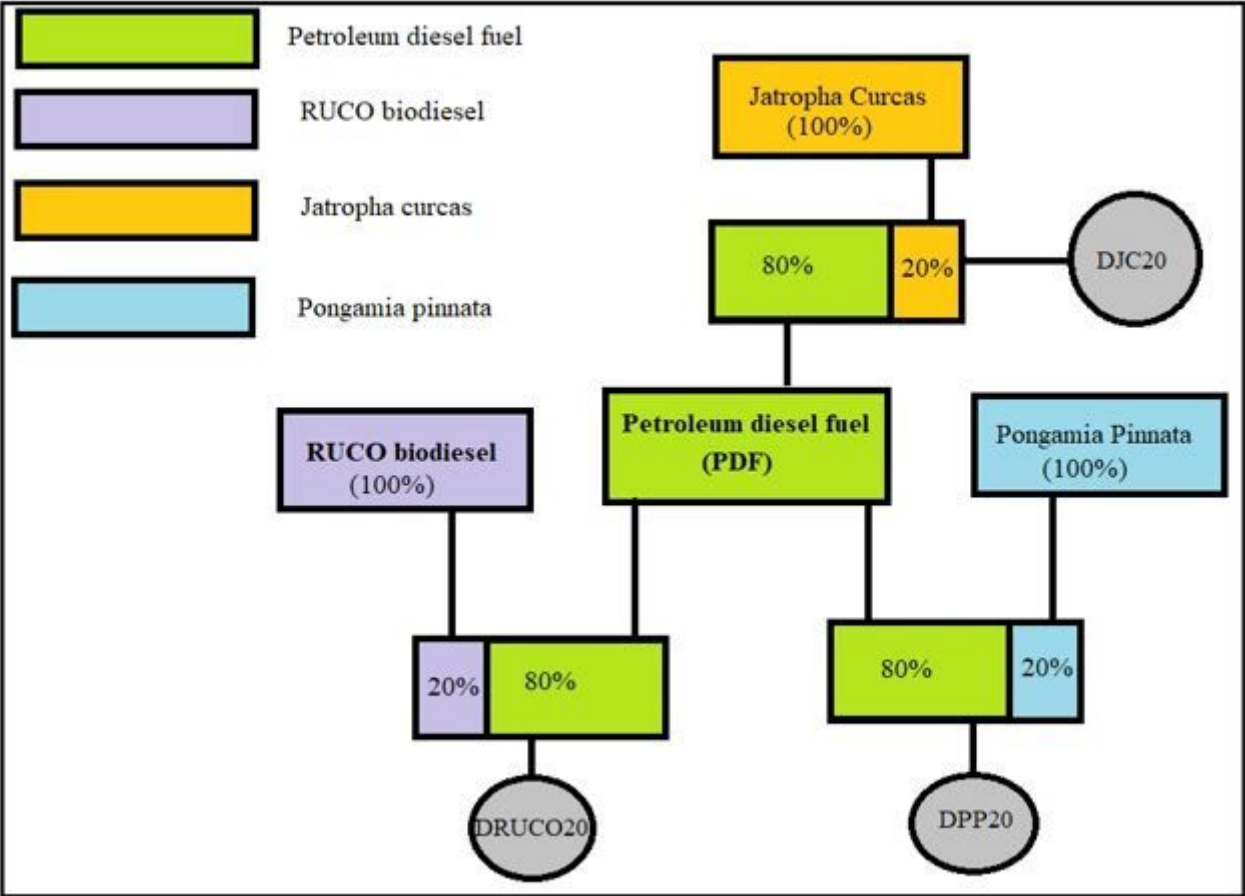


Figure 1

Tested diesel-biodiesel-ethanol Blends

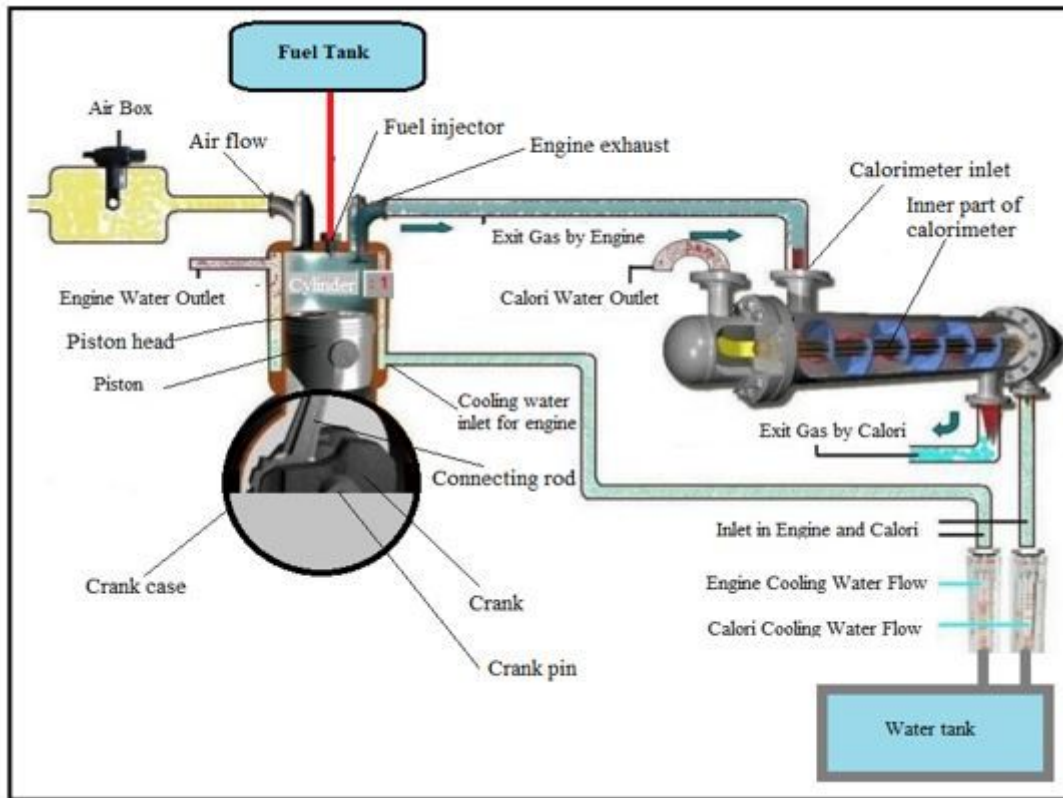


Figure 2

Experimental rig

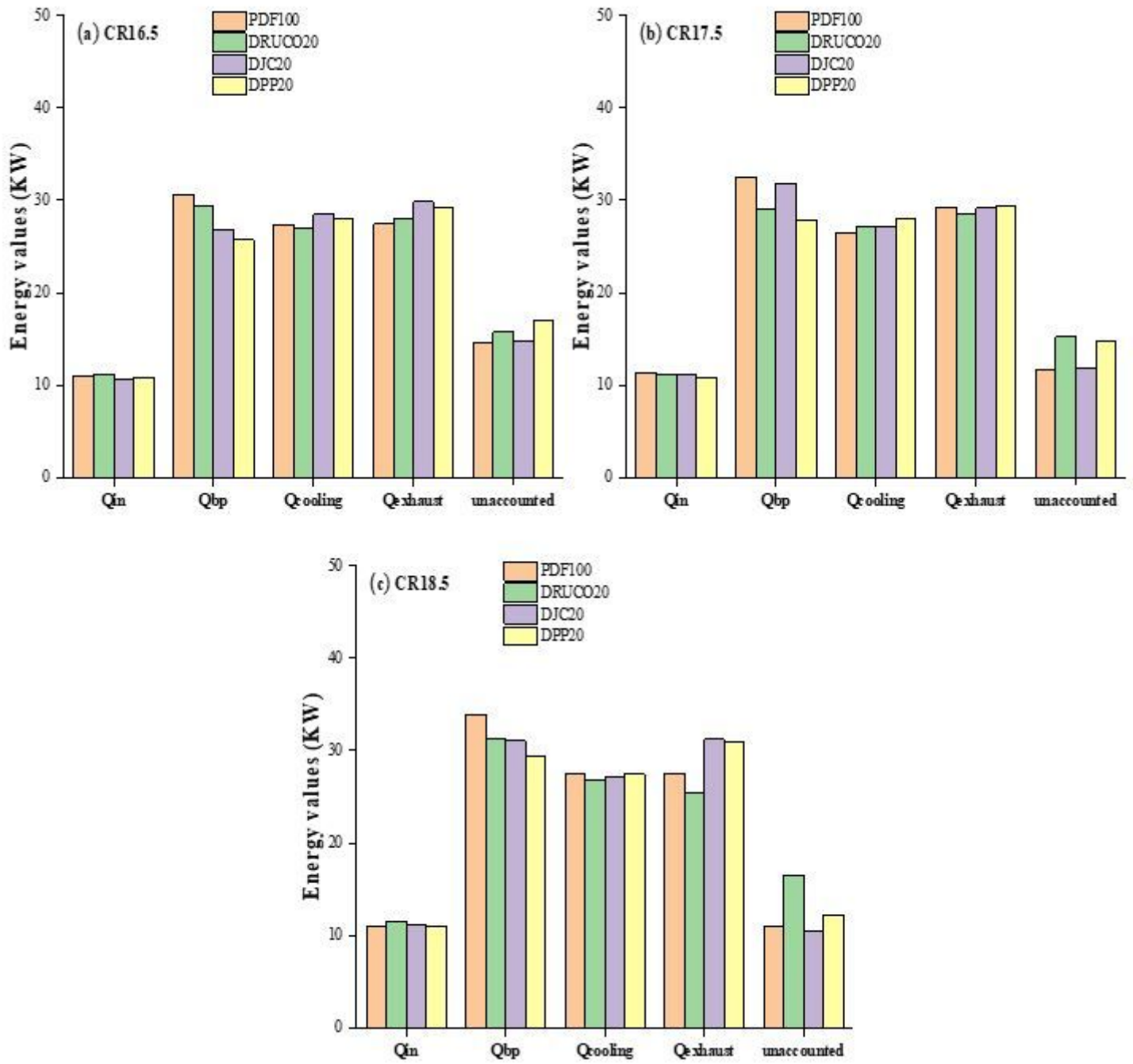


Figure 3

Energy values analysis for a) CR16.5, b) CR17.5 and c) CR18.5

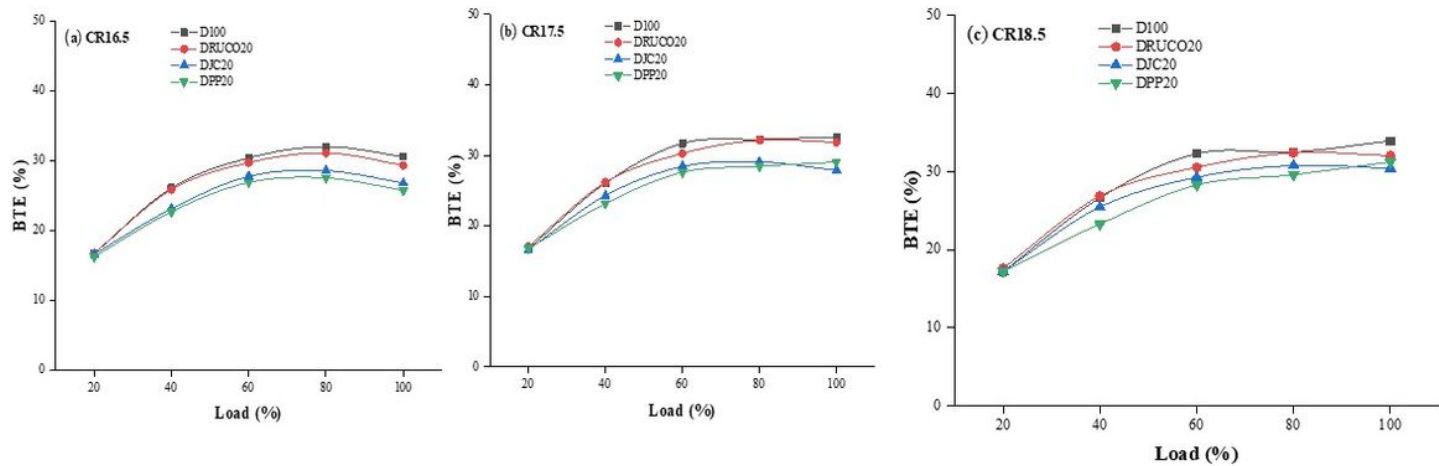


Figure 4

BTE against engine load for various CRs

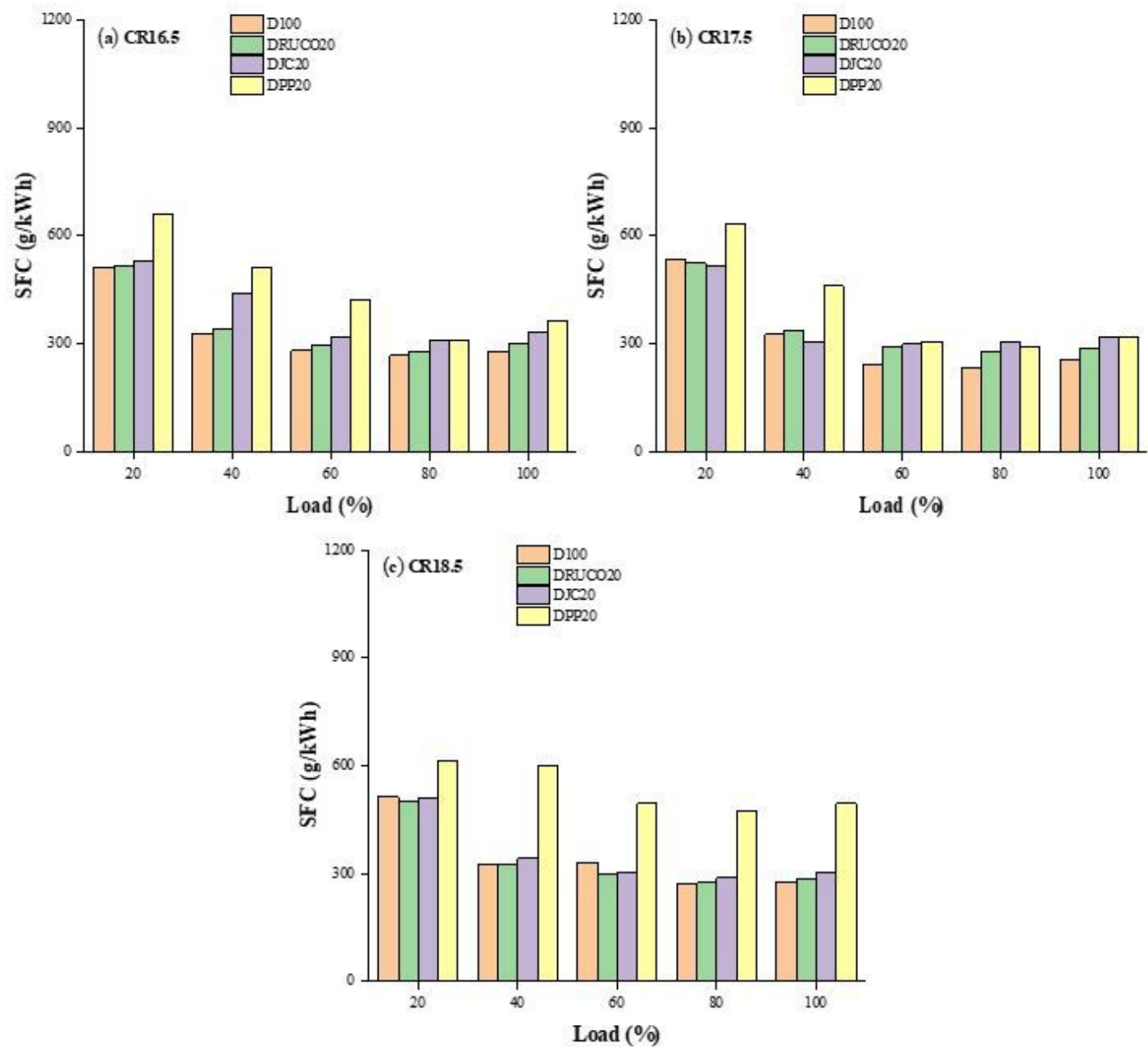


Figure 5

SFC against engine load for various CRs

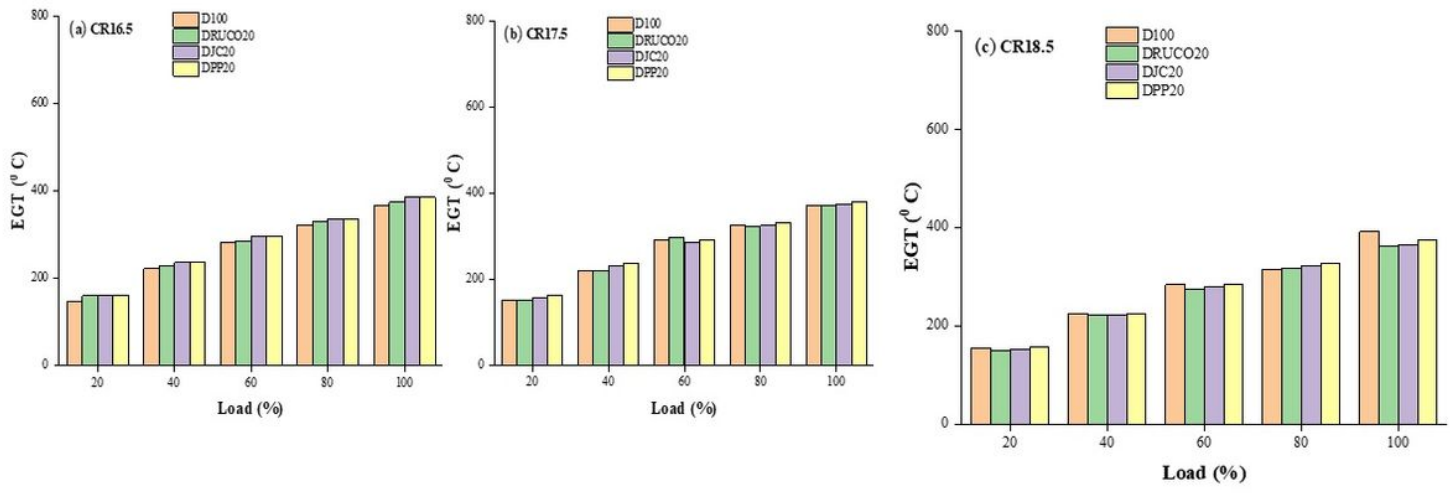


Figure 6

EGT analysis for all test conditions

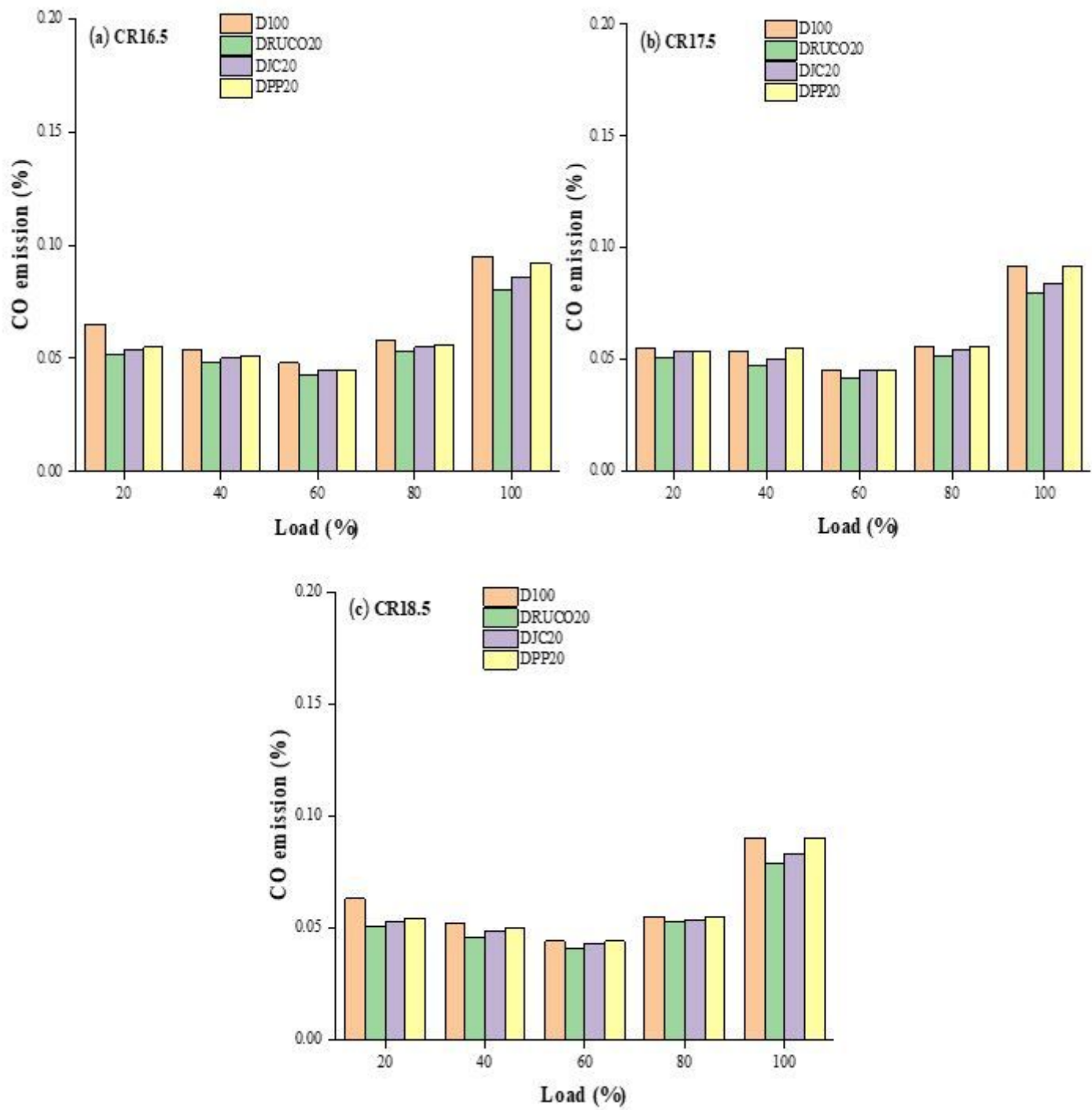


Figure 7

CO ecological analysis against engine load for 3 CRs

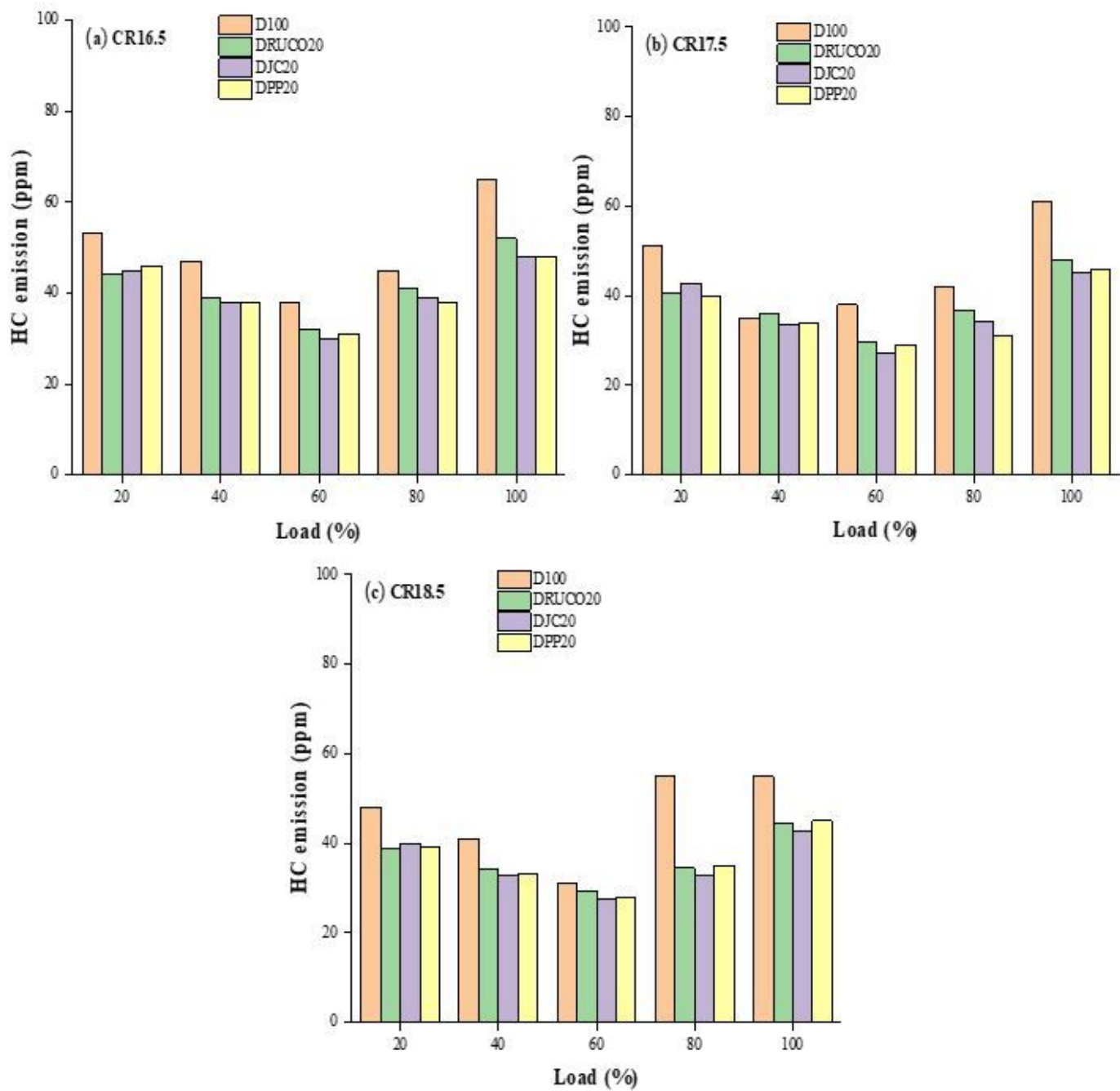


Figure 8

HC ecological analysis against engine load for all CRs

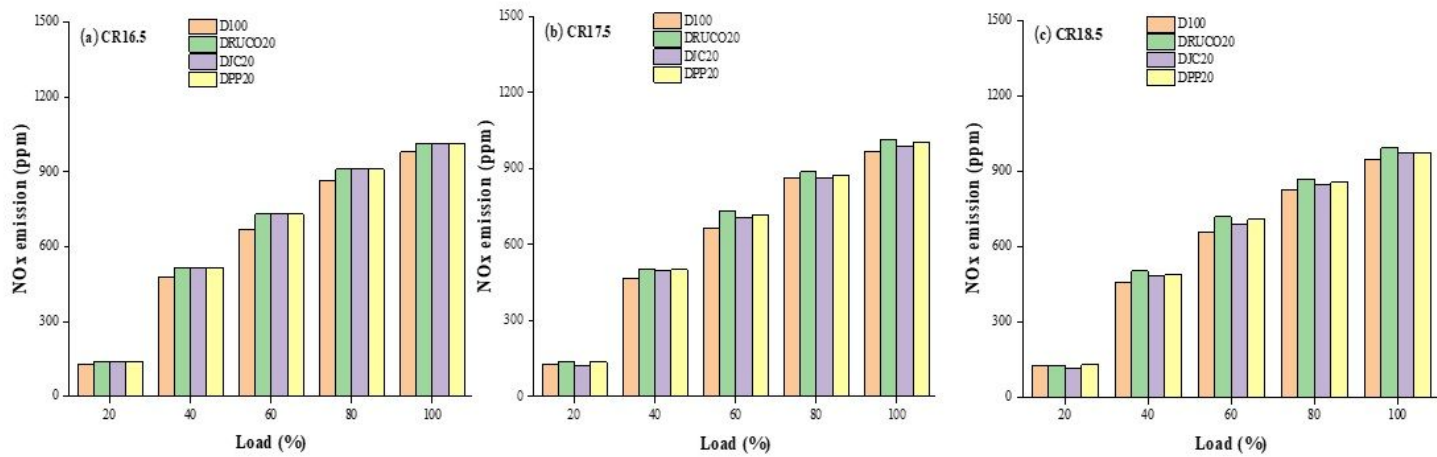


Figure 9

NOx ecological analysis against engine load at all 3 CRs

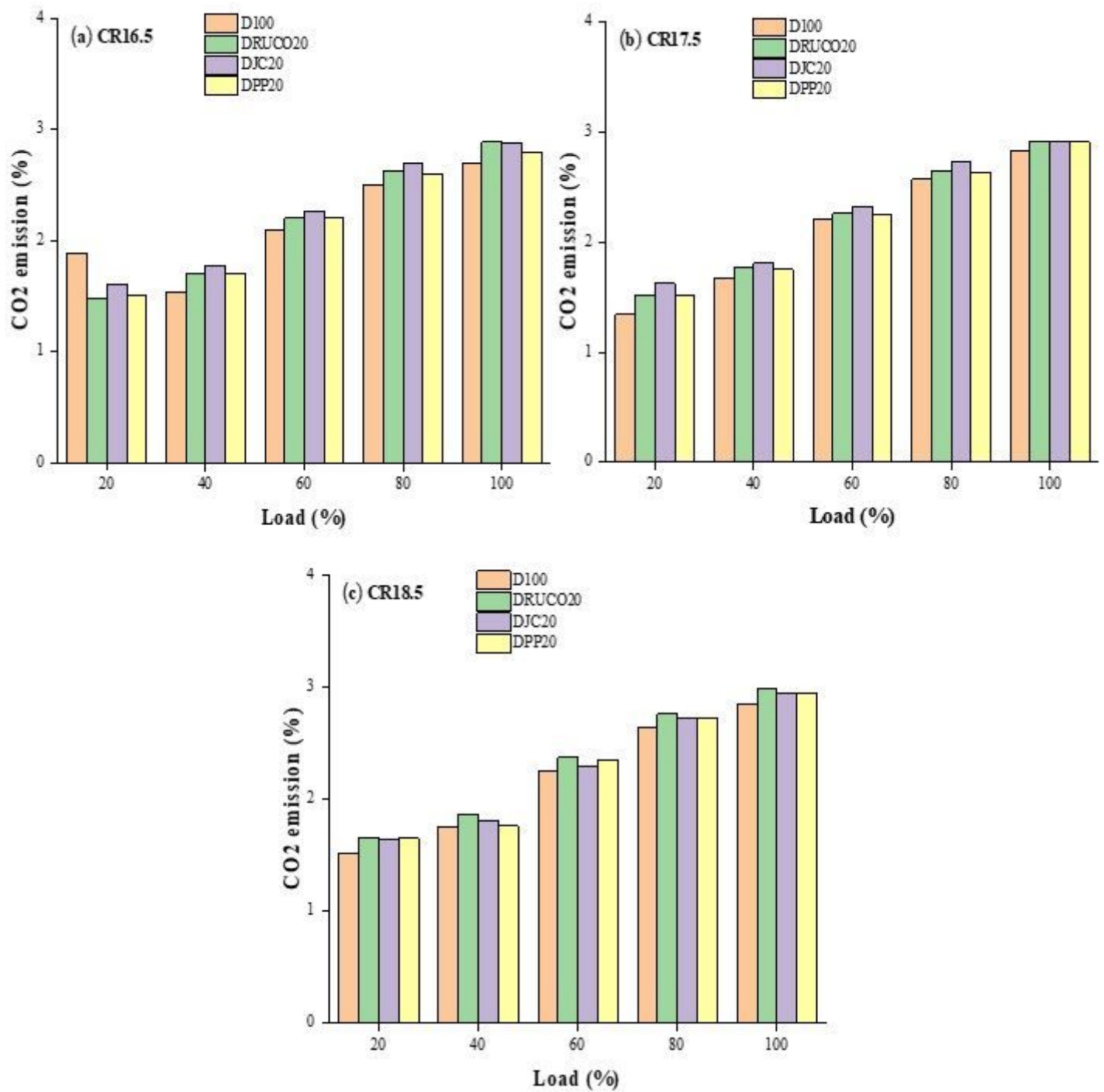


Figure 10

CO₂ ecological analysis against various engine loads for CR16.5, CR17.5 and CR18.5