Study on Combustion Parameters of Pongamia Biodiesel and Di-Tert Butyl Peroxide in Dual Fuel Mode Diesel Engine using Hydrogen as Gaseous Fuel

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Abstract: Because of its environmental benefits, biodiesel made from pongamia oil has been viewed as a possible alternative for diesel engines. Reducing exhaust pollutants and dependency on traditional petroleum fuels are two of the biggest issues facing compression ignition (CI) engines these days. Even so, the present engine pollution reduction technologies are almost at the legal limitations, and those restrictions are anticipated to get stricter soon. At low load, the maximum value of net heat release (NHR) and the peak of mean gas temperature (MGT) in cylinder were observed to be 13.4 j/deg and 541.03 ^oC by the addition of 10% BKO with substitution of 7% hydrogen as a secondary fuel. However, at high load, the maximum value of net heat release (NHR) and the peak of mean gas temperature (MGT) in cylinder were observed to be 12.83 j/deg and 600.29 ^oC by the addition of 5% DTBP with substitution of 25% hydrogen as a secondary fuel.

***Keywords**: Diesel, Pongamia Biodiesel, Di-Tert Butyl Peroxide, Hydrogen Fuel, Combustion.*

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I. INTRODUCTION

The world's rapidly growing population, the use of energy for transportation, and industrial development particularly in the automotive industry are the primary causes of the demand for fossil fuels over the past 20 years. Its demand is also anticipated to increase during the next three decades. As the source becomes more limited, this reliance on it only grows. The issue is not limited to shortage; burning fossil fuels releases damaging and poisonous pollutants, such as carbon and nitrogen oxides, which cause significant climatic anomalies [1, 2]. Reducing exhaust pollutants and dependence on traditional petroleum fuels have been major issues in the use of compression ignition (CI) engines in recent years. Although the existing engine pollution reduction technologies are almost at the statute's limits, such limits are anticipated to become stricter soon. In order to lower engine emissions without changing the engine, numerous studies have been conducted on the use of renewable and alternative fuels [3]. One of the most crucial measures in this direction is to decrease reliance on fossil fuel-based energy sources by increasing the availability of renewable energy.

However, the development and use of renewable energy sources continue to face numerous technical and financial obstacles, making a full transition away from fossil fuels unlikely anytime soon. Thus, the world's energy needs throughout this time will be met by a combination of fossil and renewable energy sources. Making sure that a seamless transition from conventional technologies based on fossil fuels to renewable technologies can be achieved is equally crucial. As a result, it makes sense to increase the efficiency of our current thermal systems and enable them to use a sufficient quantity of renewable fuels. Since they are two of the biggest users of petroleum oil, alternative fuels for the automotive and decentralized power generation industries have been the focus of much research [4].

Compression ignition engines could benefit greatly from using biodiesel as an alternate fuel. The fact that diesel and biodiesel have similar qualities is one of the main benefits [5, 6]. Because it has a greater boiling point than diesel and it is also easier to handle. Additionally, because it includes 10% oxygen, it is an environmentally beneficial fuel with low smoke emissions. Furthermore, when compared to diesel, biodiesel has a higher specific gravity and kinematic viscosity. Biodiesel's higher cetane number helps reduce NO_x

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emissions during the first combustion process by shortening the ignition delay. Additionally, biodiesel produces fewer exhaust pollutants than diesel, including particulate matter, CO, polycyclic aromatic, hydrocarbons, hydrocarbons, SO_2, and smoke [7–10].

Because of carbonless structure and its high energy content, hydrogen has recently garnered a lot of attention as a fuel for internal combustion engines to help address the global carbon emissions issue. Hydrogen must be mixed with a more volatile fuel in a compression ignition engine because of its extremely high auto-ignition temperature in order to prevent misfiring and guarantee smooth combustion. The majority of studies in the literature employ diesel that is generated from petroleum as the pilot fuel. The combination of hydrogen and diesel frequently results in higher NO_x emissions, despite the fact that it might offer considerable advantages in terms of thermal efficiency and hazardous emissions [11].

Because hydrogen burns at the fastest rate and raises the temperature inside the cylinder, replacing hydrogen in a diesel engine running in dual fuel mode with diesel as pilot fuel results in more NO_x being produced than when diesel fuel is used [12]. O_2 concentration, peak heat release rate, combustion temperature during the pre-mixed combustion phase, and combustion length all affect the production of NO_x. Additionally, it is dependent on the thermo-physical characteristics such as density, cetane number, and viscosity, as well as engine parameters like compression ratio, nozzle opening pressure, and fuel injection time [13]. Di-tert butyl peroxide (DTBP) was the first known efficient cetane number enhancer in the 1940s. Compared to additive 2-Ethyl Hexyl Nitrate (2-EHN), DTBP has a far better chance of lowering NO_x emissions. At certain standard temperature circumstances, the additive (DTBP) exhibits extremely good stable conditions, including thermally and oxidatively, in diesel and biodiesel fuels [14]. Diesel produces more hazardous exhaust gas pollutants as a result of its low cetane number [15].

The rate of ignition delay (ID) gradually falls as the diesel fuel's proportion of additives (DTBP) rises. A thorough analysis of the release heat rate revealed that the additive (DTBP) had an impact on both raising the temperat ure of combustion and hastening the onset of reactions in the early stages of the autoignition process [16]. The rate of heat release during the premixed combustion process was decreased by the use of the additives. In contrast to the Jatropha (B20), the altered blends had an improved rate of heat release [17].

Using high Research Octane Number (RON) fuels, LU¨ et al. [18] have examined the impact on the rate of a homogeneous charge compression ignition (HCCI) engine by additive addition (DTBP) for the emissions and heat release. The RON90 fuel was mixed with 0%, 1%, 2%, 3%, and 4% (by volume) DTBP to conduct the trials. The experimental findings demonstrated that the DTBP addition in RON90 fuel significantly increased the operating range to lower temperatures and lower engine load conditions. With the DTBP addition, the chemical reaction scale time lowers, which causes the ignition timing to advance, duration of the combustion to shorten, and the rates of heat release to increase under overall load conditions of the engine. In the meantime, the DTBP addition also maintained the NO_x emission at a lower level.

When diesel fuel is burned, the traditional diesel combustion phases are seen; however, when diesel fuel with hydrogen are burned together, flame velocity of hydrogen is high causes an combustion phase of non-controllable explosive-type. The knocking of the diesel engine is very high because of the uncontrollable combustion phase caused by the homogenous hydrogen in the cylinder, which results in a superior pressure variation. However, restricting the hydrogen portion to 50% could help reduce this knocking propensity. The results show that, in comparison to diesel fuel operation, the increases in the value of maximum heat release rate by 28.4% to 42.8% with a 25% hydrogen substitution and by 64.7% to 84.6% with a 50% hydrogen substitution [19, 20].

According to V. Gnanamoorthi et al. [21], adding hydrogen to a diesel engine operating in dual fuel mode enhanced exhaust emissions and combustion at various hydrogen mass flow rates. The best results were obtained with a hydrogen addition rate of 30 LPM. At its maximum power output, NO_x generation rose by 7.3% in comparison to diesel fuel. Because hydrogen had a faster flame, the combustion time was also decreasing. Researchers looked into how adding hydrogen affected the emissions and combustion properties of diesel engines operating in dual fuel mode. On an energy basis, a 25%–50% substitution of hydrogen results in a notable rise in nitrogen oxides [22–23]. Furthermore, as the hydrogen fractions grow, so did the maximum in-cylinder gas pressure and peak heat release rate.

In the present paper, we investigate the variation of net heat release (NHR) rate verses crank angle (deg) and also shows the variation of mean gas temperature (MGT) verses crank angle (deg) at various load (%) conditions with different substitution of biodiesel of karanja oil (BKO) and hydrogen as secondary fuel.

II. METHODS AND MATERIALS

The experiment in dual-fuel mode (Fig. 1(a)-(b)) was carried out on a modified Kirloskar TV1 engine, which is four-stroke, a single-cylinder, diesel engine with a power rating of 3.5 kW at 1500 rpm. The modifications included changes to the compression ratio. The engine was fitted with an eddy current dynamometer, which was configured for loading. The experimental investigation was conducted using a manometer, a fuel tank, a panel box, a fuel measuring device, a digital indicator, and a digital temperature indication. Data for the engine performance analysis was collected using both Microsoft Excel and IC engine simulation software. Table 1 provides the technical specifications of the engine. The hydrogen cylinder setup includes a pressure regulator, flashback arrester, rotameter with a one-way non-return valve, and flame arrester. The

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impact of Karanja oil on biodiesel was examined experimentally in this work while the engine was operating at a constant speed of 1500 rpm and the dynamometer's knob turned in response to changes in engine load, which ranged from 0.2 kg to 18 kg. Piezo signals were also used to send the powering unit/AX-409 used for additional analysis, and the engine also prepared a piezo sensor to measure the cylinder pressure. A crank angle encoder was utilized to measure angles with an accuracy of 1°.

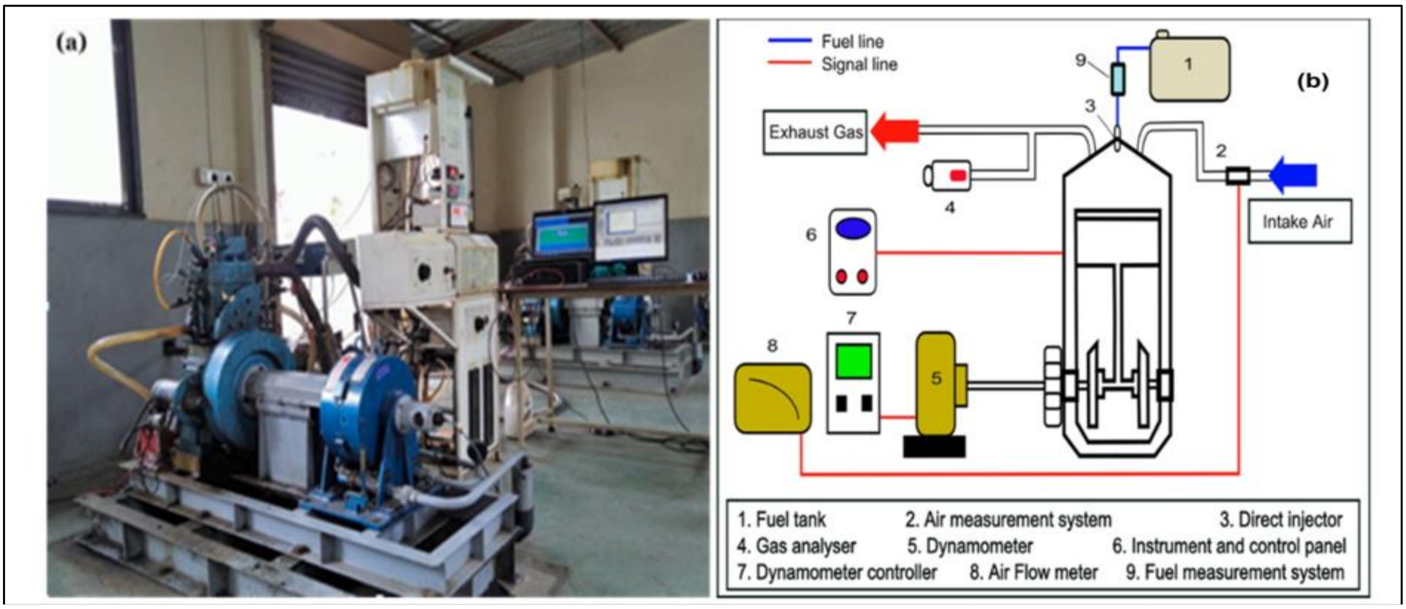

Fig 1: (a) Experimental Photographic View (b) Experimental Set-up Schematic View

After five minutes of engine operation, pressure data was collected for an average of 100 cycles under stable conditions. The mass flow rate of hydrogen was measured in kilograms per minute. Moreover, the properties of biodiesel of karanja oil and comparison between the characterstics of hydrogen and diesel fuel are indicated in table 2 and table 3. For the first step of the trials, primary diesel fuel (case 1) was utilized, followed by a mixture of diesel and biodiesel made from Karanja oil (case II). H_2 was then used as a secondary fuel for cases 3 and 4, and mixed fuel with H_2 which is indicated in table 4.

Table 1: Scientific Capacity of the Engine [24]

S. No.	Factors	Measurements	Units
1	Swept volume	661.45	cc
2	Rated speed	1500	RPM
3	Bore × Stroke	87.50 × 110	Mm × mm
4	No. of cylinder	1	-
5	Engine type	Single cylinder four-strokes, CI engine etc.,	-
6	Make and model	Model Kirlosker, TV1	-
7	Inlet temperature	300	K
8	Inlet pressure	1.03	bar
9	Rated power	3.5	KW
10	Injection timing BTC	19	°C
11	Injection pressure	224.11	bar
12	Compression ratio	18	-

Table 2: Properties of Biodiesel of Karanja Oil (B100) [24]

Properties	Unit	Biodiesel of Karanja oil
Fire point	°C	74
Density at 25°C	kg/m^3	885
Kinematic Viscoisty @40°C	cSt	5.31
Dynamic Viscosity @40°C	cP	4.7
Flash Point	°C	68
HCV Calorific Value	Cal\gm	9414
LCV Calorific Value	Cal\gm	8828

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Table 3: Comparison between the Characterstics of Hydrogen and Diesel Fuel [24].

Properties	Unit	Hydrogen	Diesel
Density At 25◦C	(kg/m3)	0.09	818
LCV Calorific Value	(Cal/gm)	120	9877
HCV Calorific Value	(Cal/gm)	858	10463
Flash Point	◦C	4-7.5	51
Fire Point	◦C	0.02	58
Kinematic Viscoisty @40◦C	cSt	0.64	2.09
Dynamic Viscosity @40◦C	cP	0.61	1.73

Table 4: The Experimental Test Matrix has been Summarized

Case No.	Load (%)	Primary Fuel	Auxiliary Fuel	Biodiesel (%)	DTBP (Additive)
1	2,36, and 69	Neat diesel	-	-	-
2	2,36, and 69	Neat diesel	-		DTBP (Additive)
3	2,36, and 69	Neat diesel	-	Biodiesel of Karanja oil	-
4	2,36, and 69	Neat diesel	H2 fuel	-	-
5	2,36, and 69	Neat diesel	H2 fuel	Biodiesel of Karanja oil	-
6	2,36, and 69	Neat diesel	H2 fuel	-	DTBP (Additive)

Table 5: The Experimental Investigation of the Test Template

Trials No	Base Fuel	Alternative Fuel	At Each Load Circumstances with Hydrogen Fuel (%) along with Base Fuel (Diesel) Substituted.
1	Pure diesel	-	-
2	Diesel	DTBP (Additive)	A-1%,2%,3%, 4%,5%
3	Diesel	Biodiesel of Karanja oil	B-10%,B-20%,B-30%,B-40%.
4	Diesel	H2 fuel	H-7%,11%,16%,20%,25%
5	Diesel	H2 + Biodiesel of Karanja oil	{D-90%+B-10%+H-7%, D-90%+B-10%+H-11%, D-90%+B-10%+H-16%, D-90%+B-10%+H-20%, D-90%+B-10%+H-25%}. {D-80%+B-20%+H-7%, D-80%+B-20%+H-11%, D-80%+B-20%+H-16, D-80%+B-20%+H-20% D-80%+B-20%+H-25%}. {D-70%+B-30%+H-7%, D-70%+B-30%+H-16%, D-70%+B-30%+H-11%, D-70%+B-30%+H-20%, D-70%+B-30%+H-25%}. {D-60%+B-40%+H-7%, D-60%+B-40%+H-11%, D60%+B-40%+H-16, D-60%+B-40%+H-20% D-600%+B-40%+H-25%}.

Figure 2-11 shows the distinction between the net heat release (NHR) versus crank angle with different load conditions with the diesel, di tert butyl peroxide (1%-5%) as additive, hydrogen (7%-25%) as a secondary fuel, and biodiesel of Karanja oil (10%-40%) Table-5. As seen in Figs. 2-11, two combustion phases—diffusion and premixed were clearly distinguished. Two peaks in the heat release rate curve indicate the region of diffusion and premixed combustion respectively [25-26].

Figure 2 shows the distinction between the net heat releases (NHR) versus crank angle with different load conditions in the pure diesel pathway (case 1). The maximum enhancement in the net heat release was observed to be 29.49, 35.87, and 51.16 j/deg at 2%, 36%, and 69% load circumstances respectively. Among these loads, the maximum value of NHR was observed at 69% of load circumstances. Because of the rise in fuel mass flow rate, the net peak heat release rises as the load increases and moves to the left, reducing ignition delay and improving fuel mixing under higher load conditions.

Figure 3 shows the distinction between the net heat releases (NHR) versus crank angle with different load conditions by adding di tert butyl peroxide in the pure diesel pathway (case 2). The diminishment in the peak of the NHR with the increasing fraction of the additive at 2%, 36%, and 69% load circumstances. At higher load circumstances (69%), the maximum value of NHR was observed to be 18.21 j/deg by the addition of 5% DTBP with pure diesel as compared to 51.16 j/deg of pure diesel pathway due to the inclusion of DTBP resulted in a shift in the net heat release curve. As a cetane booster, DTBP affected the combustion process, especially during the early "premixed burning" phase. It also led to an earlier peak in net heat release associated with the combustion process. As the cetane number of the fuel increased, the magnitude of heat release decreased when the additive fraction rose from 1% to 5% [27]. A higher cetane number shifted the net heat release

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curve, affecting the amount of fuel consumed during the premixed combustion phase. The initial heat release rate declined because the first phase of combustion was shorter [27]. An increase in cetane number also resulted in a shorter ignition delay, which extended the duration of mixing-controlled combustion [28]. This reduced ignition delay, combined with less efficient combustion, and explained the observed trend.

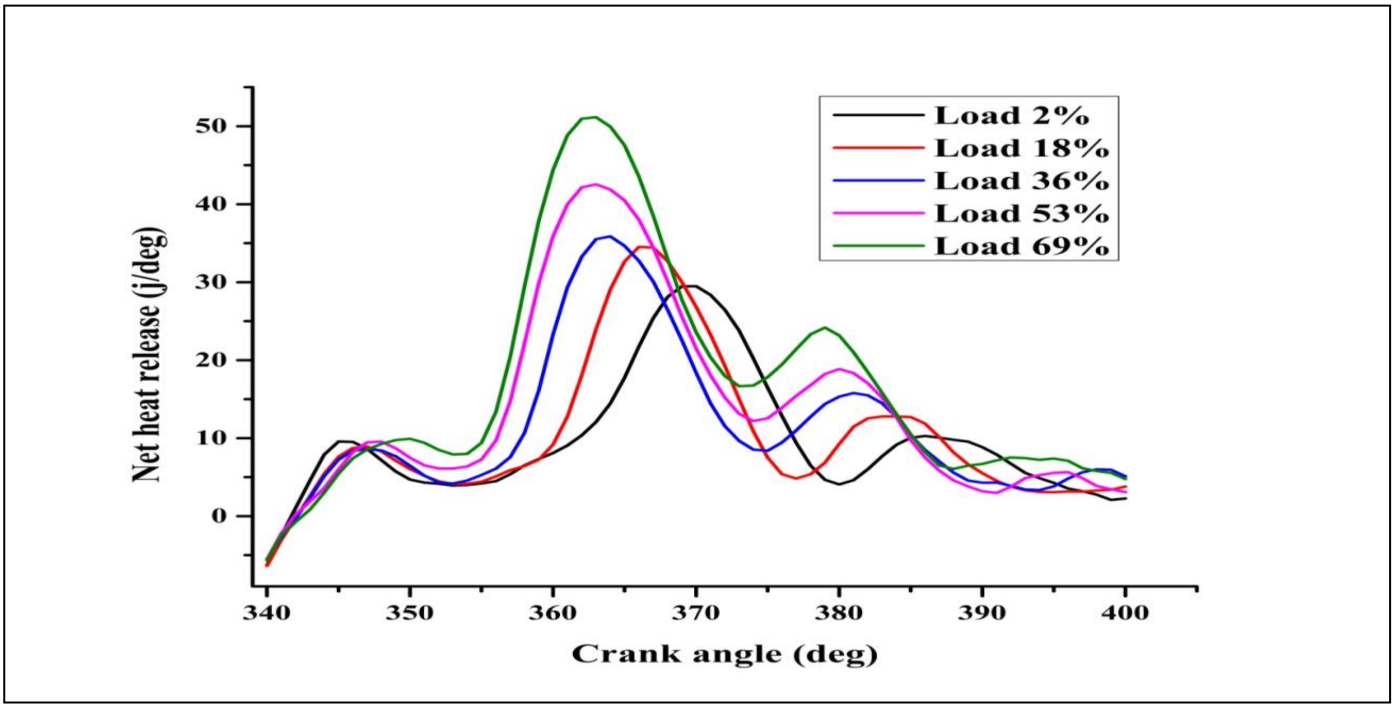

Fig 2: Variation of Net Heat Release (NHR) Rate Verses Crank Angle (deg) at Various Load (%) Conditions

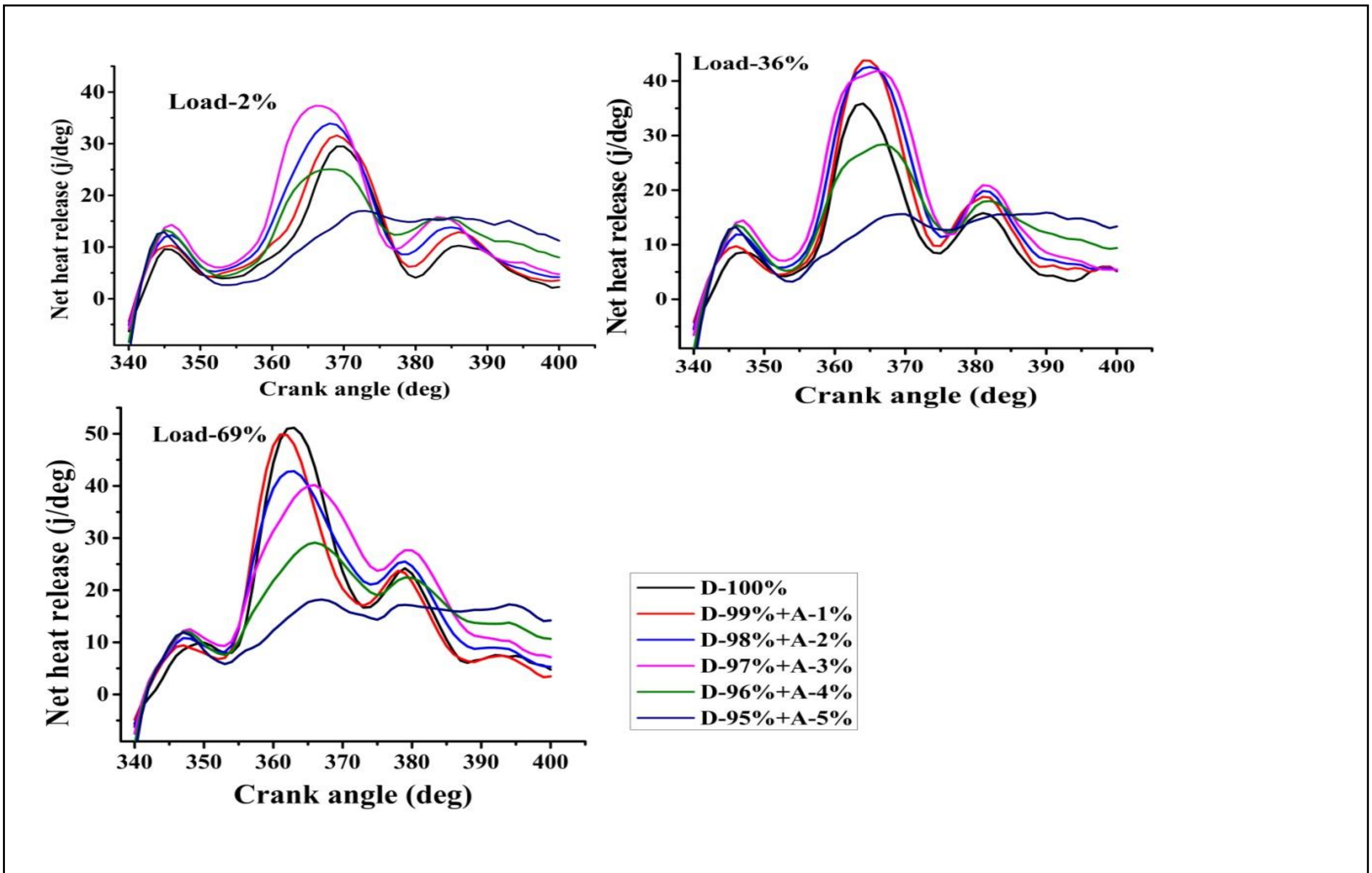

Fig 3: Variation of Net Heat Release (NHR) Rate Verses Crank Angle (deg) at Various Load (%) Conditions with Different Substitution of DTBP

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Figure 4 shows the distinction between the net heat releases (NHR) versus crank angle with different load conditions by adding BKO in the pure diesel pathway (case 3). The diminishment in the peak of the NHR with the increasing fraction of the adding BKO at 2%, 36%, and 69% load circumstances. At lower load circumstances (2%), the maximum value of NHR was observed to be 15.83 j/deg by the addition of 10% BKO with pure diesel as compared to 29.49 j/deg of pure diesel pathway. This occurs as a result of the biodiesel's surplus oxygen content from a former combustion cycle continuing to burn in a later step. This occurs because the surplus oxygen content in biodiesel from a previous combustion cycle continues to burn in a subsequent step. According to the combustion analysis, adding biodiesel to regular diesel fuel reduced the premixed combustion's heat release rate as well as time of delay [29].

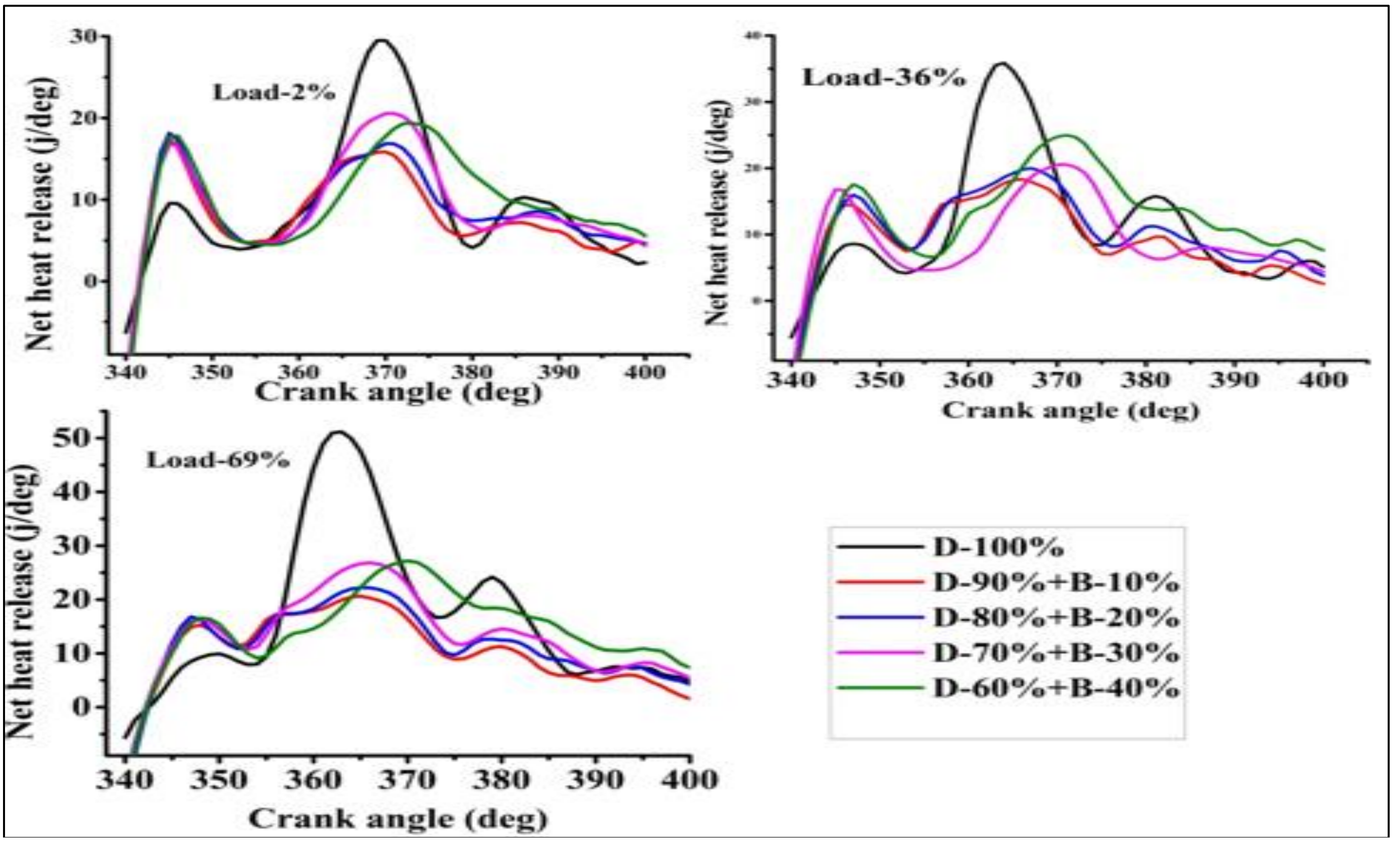

Fig 4: Variation of Net Heat Release (NHR) Rate Verses Crank Angle (deg) at Various Load (%) Conditions with Different Substitution of Biodiesel of Biodiesel of Karanja Oil (BKO)

Figure 5 shows the distinction between the net heat release (NHR) versus crank angle with different load conditions by adding hydrogen as secondary fuel in the pure diesel pathway (case 4). The diminishment in the peak of the NHR with the 20% of hydrogen fuel substitution at medium load circumstances. At medium load circumstances, the maximum value of NHR was observed to be 21.55 j/deg by the addition of 20% hydrogen with pure diesel as compared to 35.87 j/deg of pure diesel pathway.

The ability of diesel injection to ignite the hydrogen-air mixture plays a crucial role in determining hydrogen (H_2) emissions, which are a key measure of combustion quality. In conventional engines, increasing the proportion of hydrogen substitution results in a significant and undesirable increase in hydrogen emissions under low load conditions. This issue arises from diminished engine performance and incomplete combustion caused by the inability of diesel injection to ignite the mixture effectively. Under high load conditions, improved combustion quality ensures thorough flame propagation throughout the combustion chamber, leading to a substantial reduction in hydrogen emissions. Furthermore, at higher loads, the constant volume combustion phase does not noticeably affect hydrogen emissions, as diesel injection alone provides sufficient ignition without requiring the additional heat from this phase [30].

The injection of H_2 at a 15% load significantly reduced the rate of heat release in the premixed combustion phase. In contrast, the diffusion combustion phase was slightly prolonged and improved. The engine's performance decreased due to the extended diffusion combustion and the degraded premixed combustion, which caused the overall combustion process to last longer [31]. At high load conditions, as the hydrogen substitution increased, the share of energy released during the initial premixed burn phase decreased, while the peak heat release rate during the second combustion phase increased with more than 20% hydrogen substitution. Additionally, it was noted that as the level of hydrogen addition increased, the second peak heat release rates occurred earlier compared to diesel fuel combustion [32-35].

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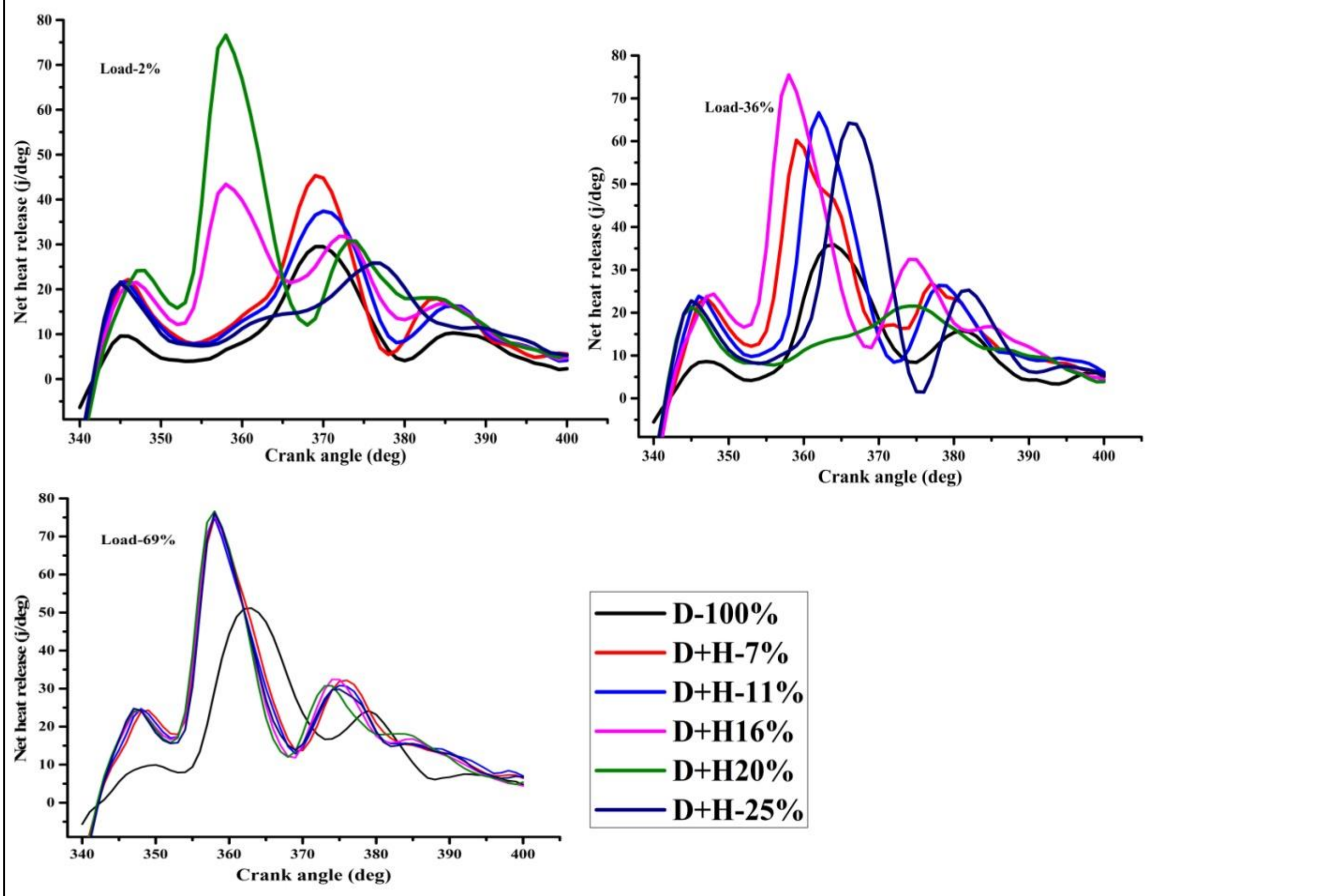

Fig 5: Variation of Net Heat Release (NHR) Rate Verses Crank Angle (deg) at Various Load (%) Conditions with different Substitution of Biodiesel of Gaseous Fuel (H_2).

Figure 6 shows the distinction between the net heat release (NHR) versus crank angle with different load conditions by the addition of BKO and hydrogen with the diesel fuel (case 5). The diminishment in the peak of the NHR with the addition of 10% BKO with 7% hydrogen substitution in a dual-fuel diesel engine at lower load circumstances. At lower load circumstances, the maximum peak of the NHR with the addition of 10% BKO with 7% hydrogen substitution in a dual-fuel diesel was observed to be 13.41 j/deg diesel as compared to 29.49 j/deg of pure diesel pathway.

Figure 7 shows the distinction between the net heat release (NHR) versus crank angle with different load conditions by the addition of BKO and hydrogen with the diesel fuel (case 5). The diminishment in the peak of the NHR was observed with the addition of BKO with hydrogen substitution in a dual-fuel diesel engine at low and medium load circumstances. At medium load circumstances, the maximum peak of the NHR with the addition of 20% BKO with 7% hydrogen substitution in a dual-fuel diesel was observed to be 20.09 j/deg diesel as compared to 35.87 j/deg of pure diesel pathway. However, at lower load circumstances, the maximum peak of the NHR with the addition of 20% BKO with 25% hydrogen substitution in a dual-fuel diesel was observed to be 15.05 j/deg diesel as compared to 29.49 j/deg of pure diesel pathway.

Figure 8 shows the distinction between the net heat release (NHR) versus crank angle with different load conditions by the addition of BKO and hydrogen with the diesel fuel (case 5). The diminishment in the peak of the NHR was observed with the addition of BKO with hydrogen substitution in a dual-fuel diesel engine at low and medium load circumstances. At medium load circumstances, the maximum peak of the NHR with the addition of 30% BKO with 25% hydrogen substitution in a dual-fuel diesel was observed to be 18.35 j/deg diesel as compared to 35.87 j/deg of pure diesel pathway. However, at lower load circumstances, the maximum peak of the NHR with the addition of 30% BKO with 16% hydrogen substitution in a dual-fuel diesel was observed to be 15.86 j/deg diesel as compared to 29.49 j/deg of pure diesel pathway.

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Figure 9 shows the distinction between the net heat release (NHR) versus crank angle with different load conditions by the addition of BKO and hydrogen with the diesel fuel (case 5). The diminishment in the peak of the NHR was observed with the addition of BKO with hydrogen substitution in a dual-fuel diesel engine at low, medium and high load circumstances. At high load circumstances, the maximum peak of the NHR with the addition of 40% BKO with 11% hydrogen substitution in a dual-fuel diesel was observed to be 22.71 j/deg diesels as compared to 51.16 j/deg of pure diesel pathway. At medium load circumstances, the maximum peak of the NHR with the addition of 40% BKO with 16% hydrogen substitution in a dual-fuel diesel was observed to be 22.78 j/deg diesel as compared to 35.87 j/deg of pure diesel pathway. However, at lower load circumstances, the maximum peak of the NHR with the addition of 40% BKO with 20% hydrogen substitution in a dual-fuel diesel was observed to be 16.19 j/deg diesel as compared to 29.49 j/deg of pure diesel pathway. The use of hydrogen improves the combustion phasing (timing of heat release), spreading the heat release more evenly across the combustion cycle. This reduces the sharp spikes in NHR. In dual-fuel engines, the introduction of BKO and hydrogen alters the air-fuel mixing process, often resulting in reduced premixed combustion. Premixed combustion is typically responsible for higher NHR peaks in diesel engines.

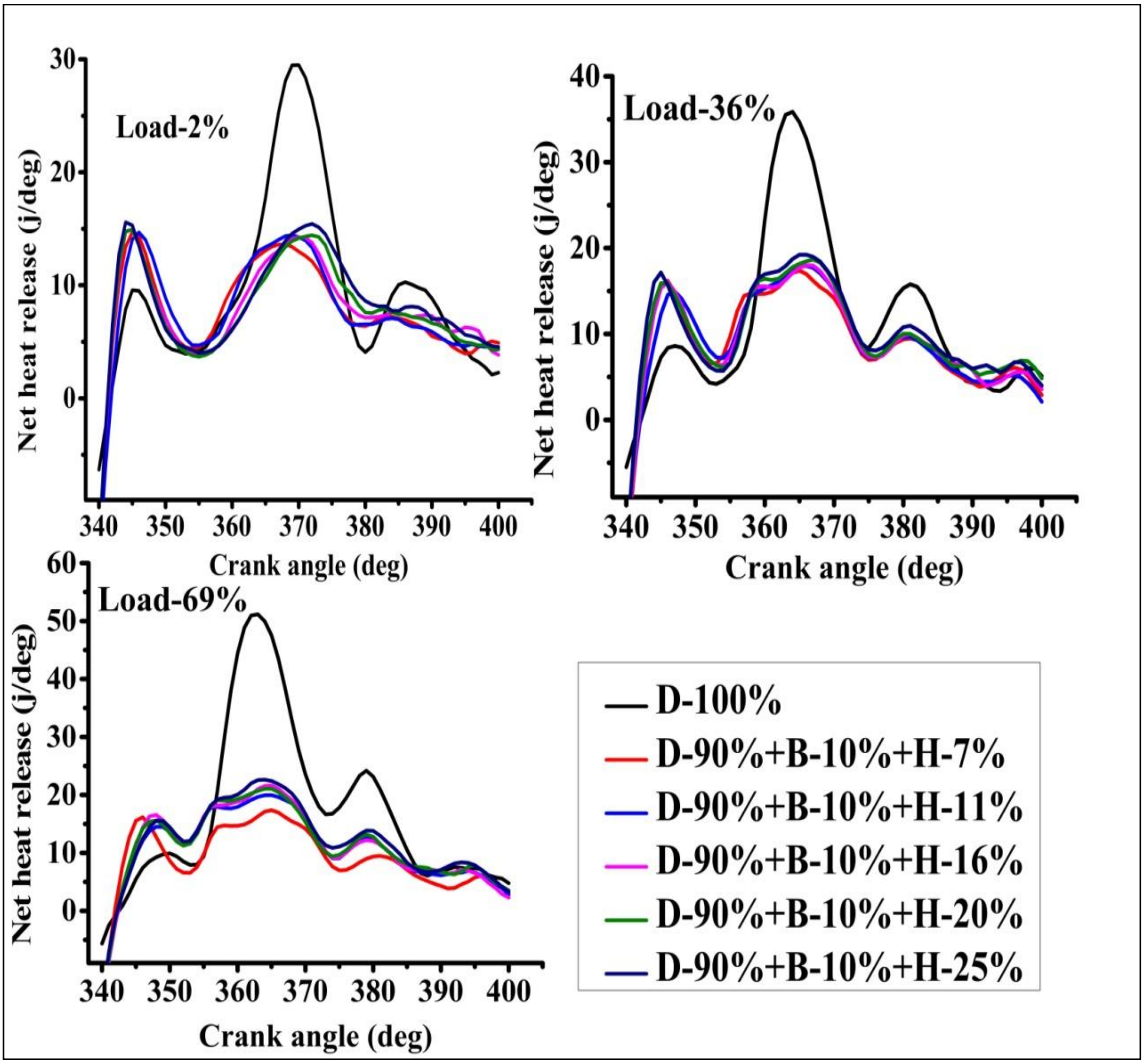

Fig 6: Variation of Net Heat Release (NHR) Rate Verses Crank Angle (deg) at Various Load (%) Conditions with Different Substitution of Biodiesel of Karanja Oil (BKO) and Hydrogen as Secondary Fuel

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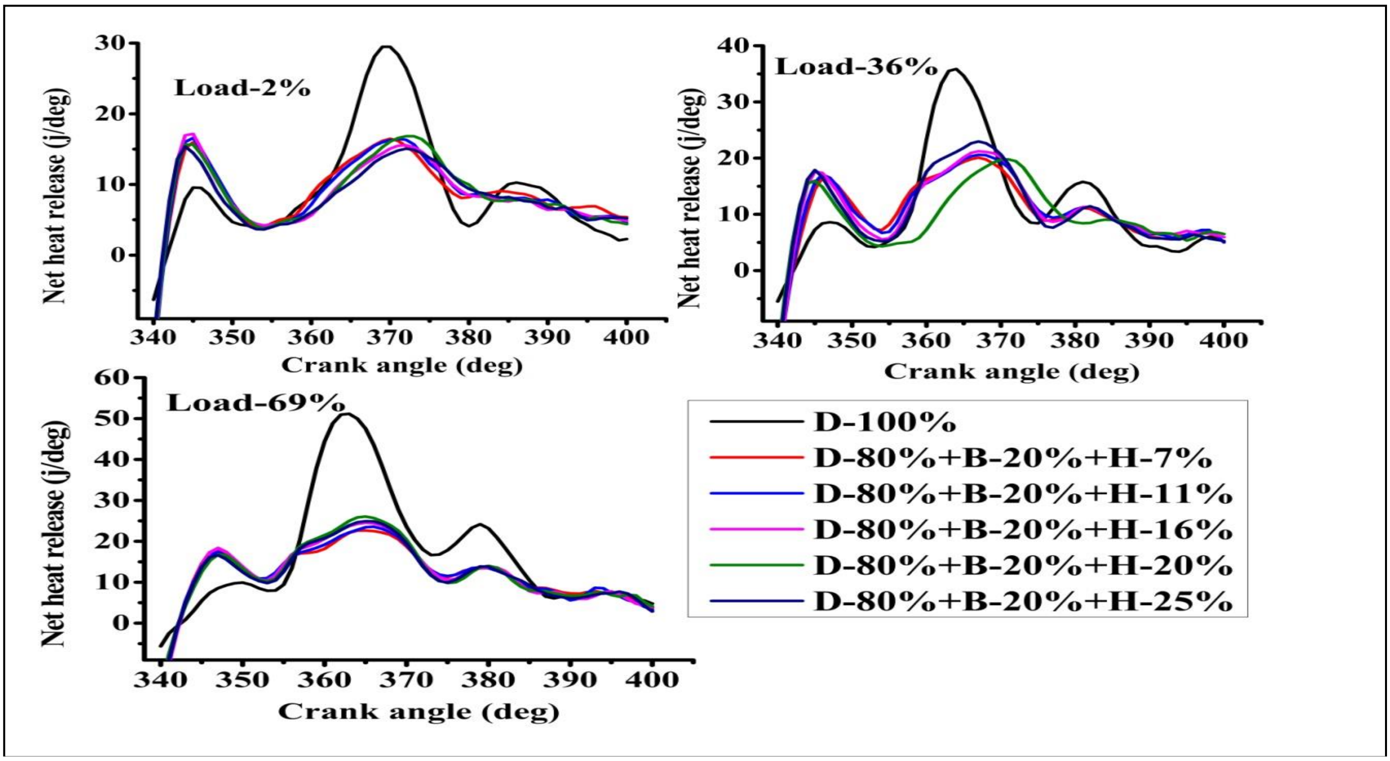

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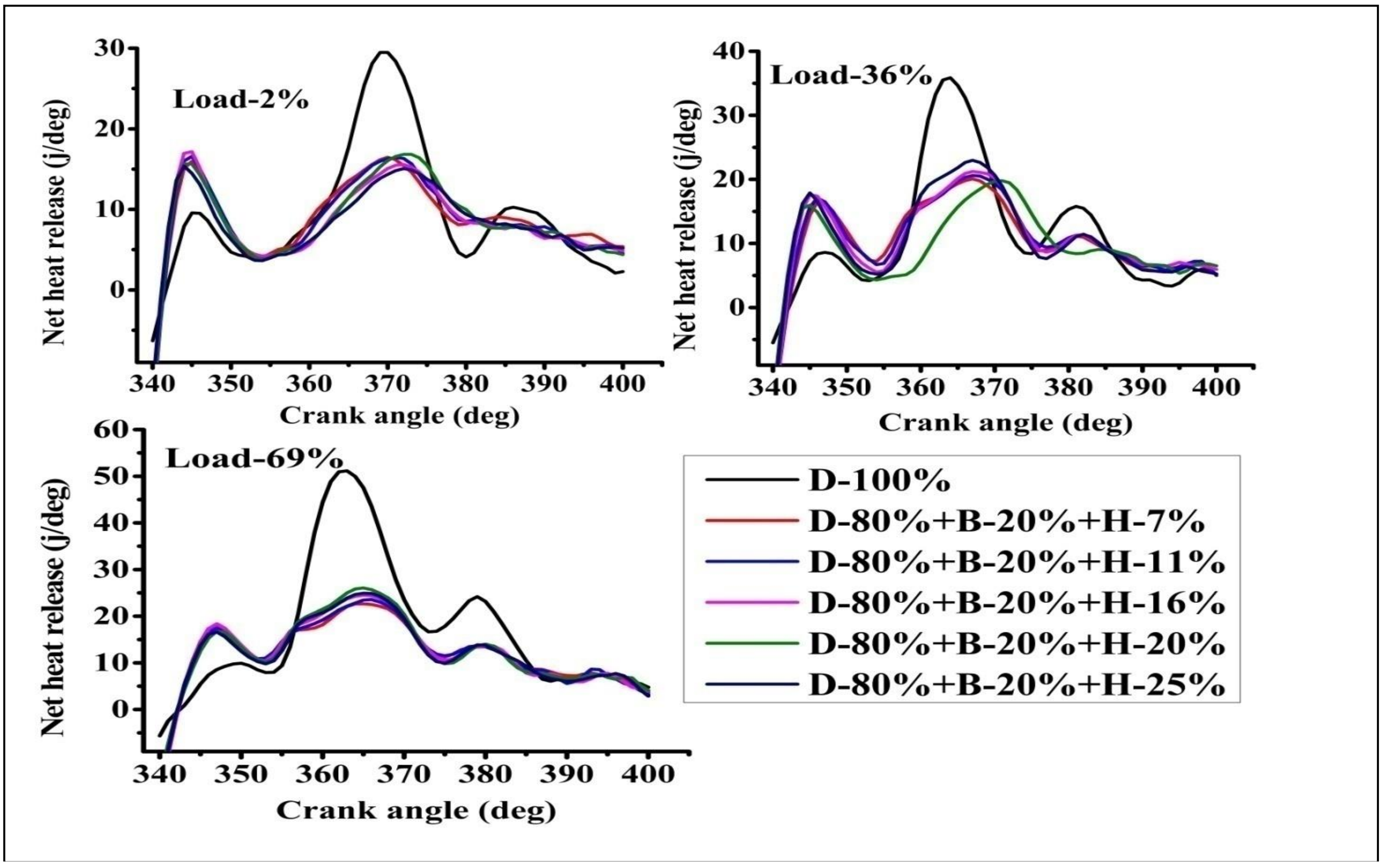

Fig 8: Variation of Net Heat Release (NHR) Rate Verses Crank Angle (deg) at Various Load (%) Conditions with Different Substitution of Biodiesel of Karanja Oil (BKO) and Hydrogen as Secondary Fuel.

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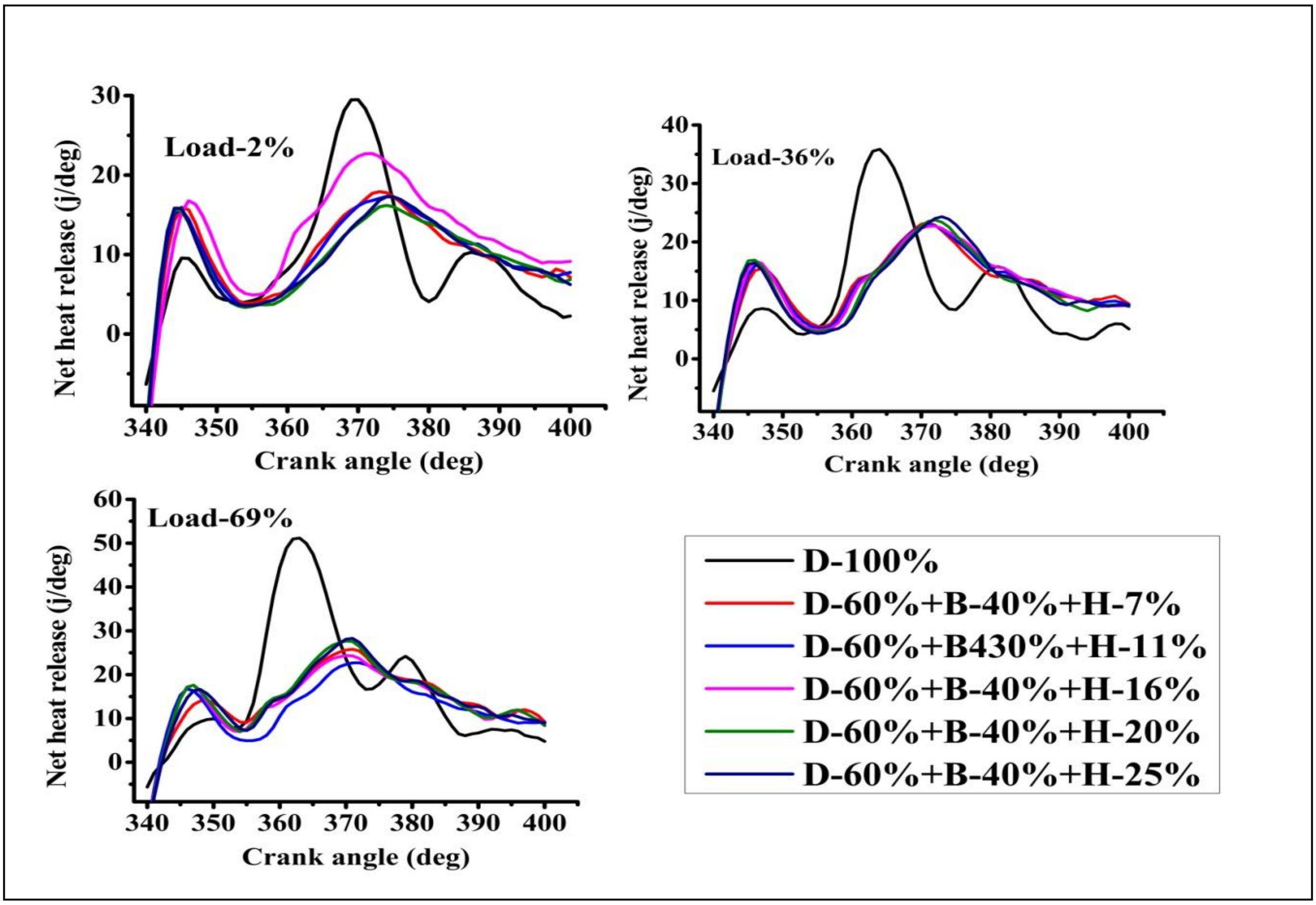

Fig 9: Variation of Net Heat Release (NHR) Rate Verses Crank Angle (deg) at Various Load (%) Conditions with Different Substitution of Biodiesel of Karanja Oil (BKO) and Hydrogen as Secondary Fuel

Figure 10 shows the distinction between the net heat releases (NHR) versus crank angle with different load conditions by the addition of DTBP as an additive and hydrogen as gaseous fuel with the diesel fuel (case 6). The diminishment in the peak of the NHR was observed by the addition of DTBP with hydrogen substitution in a dual-fuel diesel engine at low and medium load circumstances. At medium load circumstances, the maximum peak of the NHR with the addition of 1% DTBP with 20% hydrogen substitution in a dual-fuel diesel was observed to be 27.94 j/deg diesel as compared to 35.87 j/deg of pure diesel pathway. However, at lower load circumstances, the maximum peak of the NHR with the addition of 1% DTBP with 25% hydrogen substitution in a dual-fuel diesel was observed to be 25.95 j/deg diesel as compared to 29.49 j/deg of pure diesel pathway.

Figure 11 shows the distinction between the net heat release (NHR) versus crank angle with different load conditions by the addition of DTBP as an additive and hydrogen as gaseous fuel with the diesel fuel (case 6). The diminishment in the peak of the NHR was observed by the addition of DTBP with hydrogen substitution in a dual-fuel diesel engine at high, medium and low load circumstances. At high load circumstances, the maximum peak of the NHR by the addition of 3% DTBP with 25% hydrogen substitution in a dual-fuel diesel was observed to be 20.95 j/deg diesel as compared to 51.16 j/deg of pure diesel pathway. At medium load circumstances, the maximum peak of the NHR with the addition of 3% DTBP with 16% hydrogen substitution in a dual-fuel diesel was observed to be 22.55j/deg diesel as compared to 35.87 j/deg of pure diesel pathway. However, at lower load circumstances, the maximum peak of the NHR by the addition of 3% DTBP with 7% hydrogen substitution in a dual-fuel diesel was observed to be 19.56j/deg diesel as compared to 29.49 j/deg of pure diesel pathway.

Figure 12 shows the distinction between the net heat releases (NHR) versus crank angle with different load conditions by the addition of DTBP as an additive and hydrogen as gaseous fuel with the diesel fuel (case 6). The diminishment in the peak of the NHR was observed by the addition of DTBP with hydrogen substitution in a dual-fuel diesel engine at low and medium load circumstances. At high load circumstances, the maximum peak of the NHR with the addition of 5% DTBP with 25% hydrogen substitution in a dual-fuel diesel was observed to be 12.83j/deg diesel as compared to 51.16j/deg of pure diesel pathway. However, at low and medium load circumstances, the maximum peak of the NHR by the addition of 5% DTBP with 7% hydrogen substitution in a dual-fuel diesel was observed to be 12.29 j/deg diesels as compared to 49.35 j/deg of pure diesel pathway due to the addition of DTBP to the fuel blend enhanced its ignition properties. The blend now has a higher

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cetane number, resulting in a shorter ignition delay time [27]. DTBP is known to decompose rapidly at lower temperatures. At 500 K and atmospheric pressure, its half-life is 10 milliseconds, and at 700 K, it decreases to 0.1 milliseconds [36]. Changes in net heat release can be analyzed to assess in-cylinder combustion characteristics. The combustion process consists of three stages: premixed combustion, mixing-controlled combustion, and late combustion [32]. The initial heat release rate is influenced by factors such as ignition delay, combustion rate, and mixture formation in the combustion chamber [33].

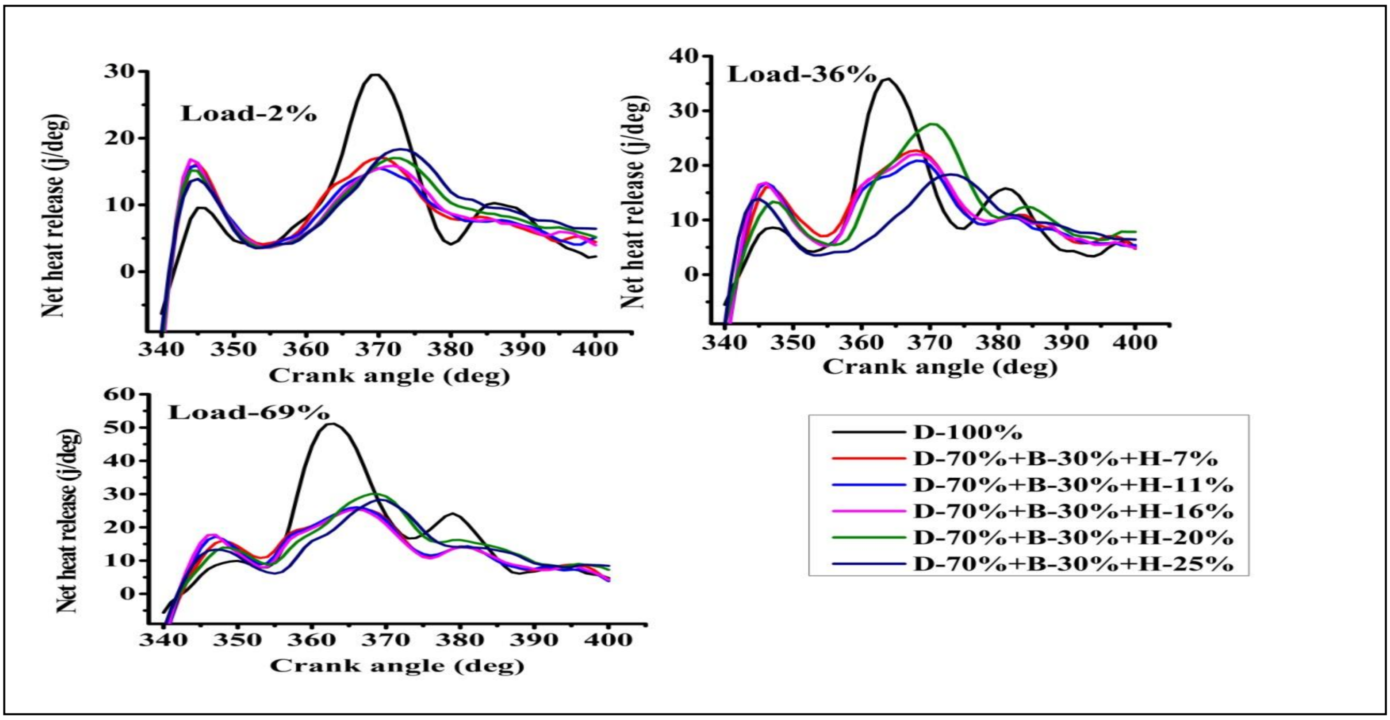

Fig 10: Variation of Net Heat Release (NHR) Rate Verses Crank Angle (deg) at Various Load (%) Conditions with Different Substitution of Biodiesel of Karanja Oil (BKO) and Hydrogen as Secondary Fuel

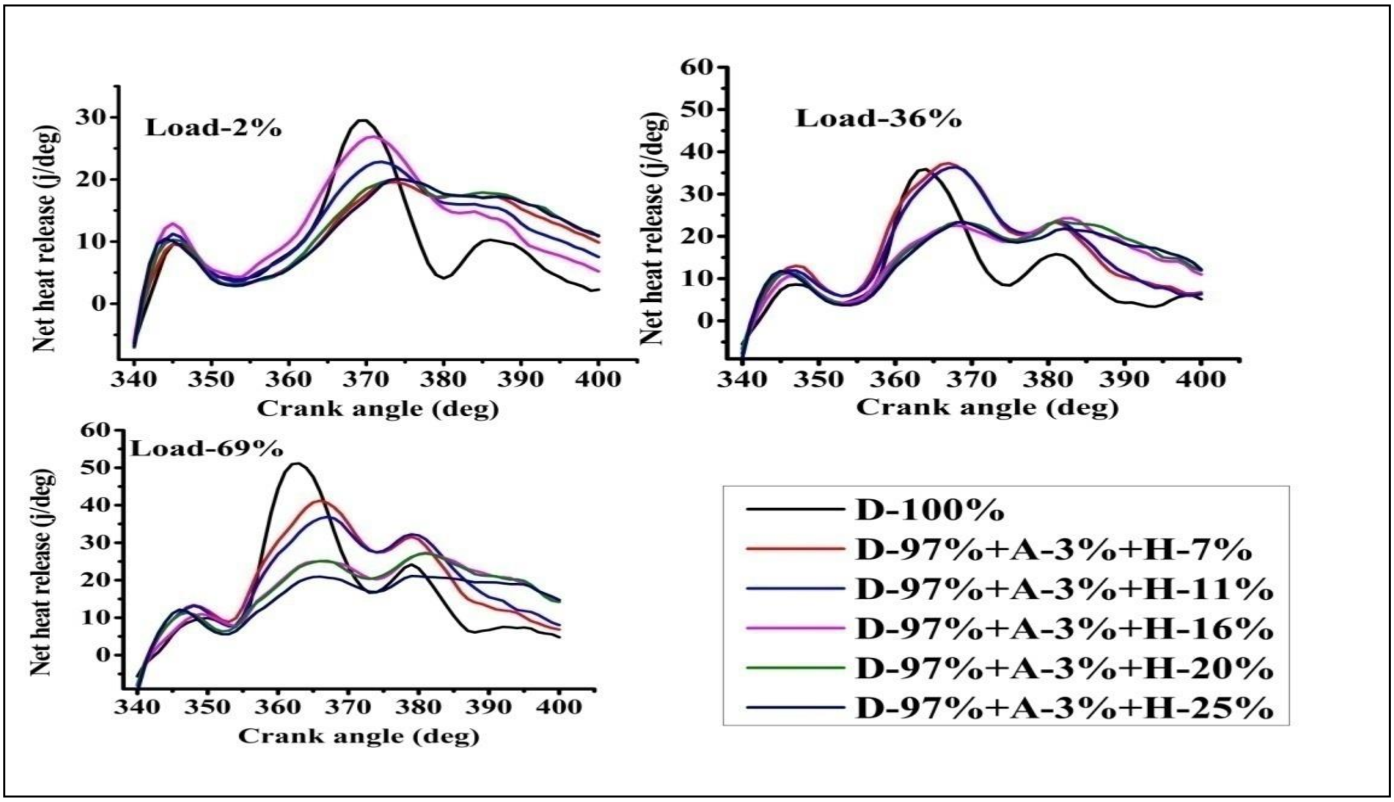

Fig 11: Variation of Net Heat Release (NHR) Rate Verses Crank Angle (deg) at Various Load (%) Conditions with Different Substitution of Biodiesel of Karanja Oil (BKO) and Hydrogen as Secondary Fuel

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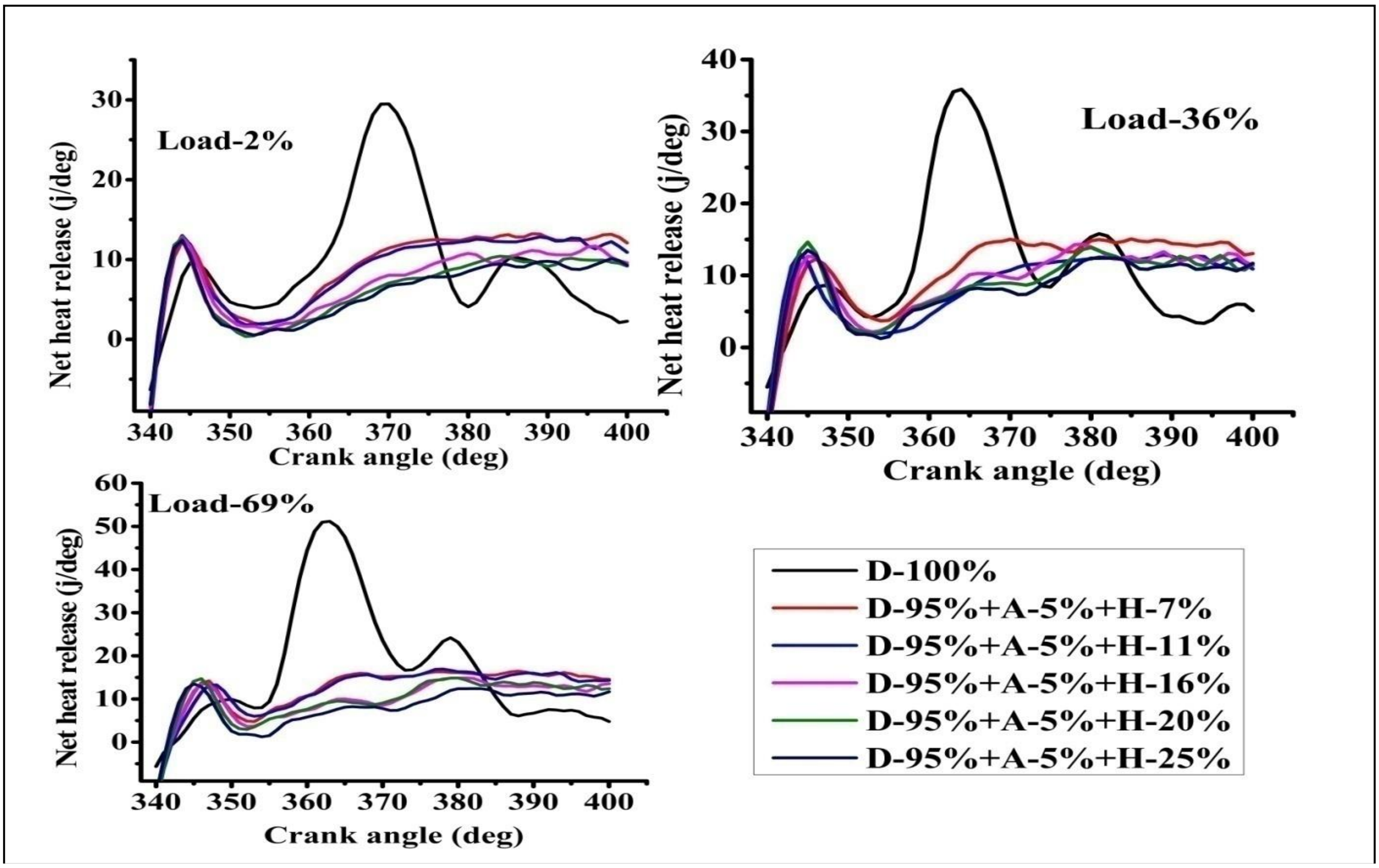

Fig 12: Variation of Net Heat Release (NHR) Rate Verses Crank Angle (deg) at Various Load (%) Conditions with Different Substitution of Biodiesel of Karanja Oil (BKO) and Hydrogen as Secondary Fuel

Figure 13-23 shows the distinction between the mean gas temperature (MGT) in cylinder versus crank angle with different load conditions with the diesel, di tert butyl peroxide (1%-5%) as an additive, hydrogen (7%-25%) as a secondary fuel, and biodiesel of Karanja oil (10%-40%) Table-5.

Figure 13 shows the distinction between the mean gas temperatures (MGT) versus crank angle with different load conditions in the pure diesel pathway (case 1). The maximum values of the mean gas temperature was observed to be 781.28, 853.33, 947.68, 1156.9 and 1380.26 ºC at 2%, 18%, 36%, 53% and 69% load circumstances respectively. Among these loads, the maximum value of mean gas temperature was observed at 69% of load circumstances. Because of the rise in fuel supply, the mean gas temperature rises as the load increases. The mean temperature of the cylinder rose as the engine's load increased because the diesel fuel supply increased along with the engine's load.

Figure 14 shows the distinction between the MGT versus crank angle with different load conditions by adding di tert butyl peroxide in the pure diesel pathway (case 2). The diminishment in the peak of the MGT with the increasing fraction of the additive at 69% load circumstance. At higher load circumstance (69%), the maximum value of MGT was observed to be 958.91 ºC by the addition of 5% DTBP with pure diesel as compared to 1380.26 ºC of pure diesel pathway. This may be due to a reduction in the heat release rate. As the proportion of the additive in the diesel fuel increased, the cetane number of the blend also increased, which in turn raised the average cylinder temperature. However, this temperature was still lower compared to when only diesel fuel was used. This could be a result of the more efficient combustion of the fuel when DTBP is added. Similarly, the earlier occurrence of net heat release before top dead center also impacts the temperature increase in the cylinder [37].

Figure 15 shows the distinction between the MGT versus crank angle with different load conditions by adding biodiesel of karanja oil (BKO) in the pure diesel pathway (case 3). The diminishment in the peak of the MGT with the increasing fraction of the BKO at 2%, 36% and 69% load circumstances. At 2%, 36% and 69% load circumstances, the maximum value of MGT was observed to be 561.16, 672.26, and 759.98 ºC by the addition of 10% BKO with pure diesel as compared to 781.28, 947.68, 1380.26ºC of pure diesel pathway. It can be seen that by the end of the compression phase, the air temperature is sufficiently high for the fuel droplets to vaporize and ignite as they enter the cylinder. The gas temperature during combustion is higher with diesel fuel, leading to increased pressure during the combustion phase [38].

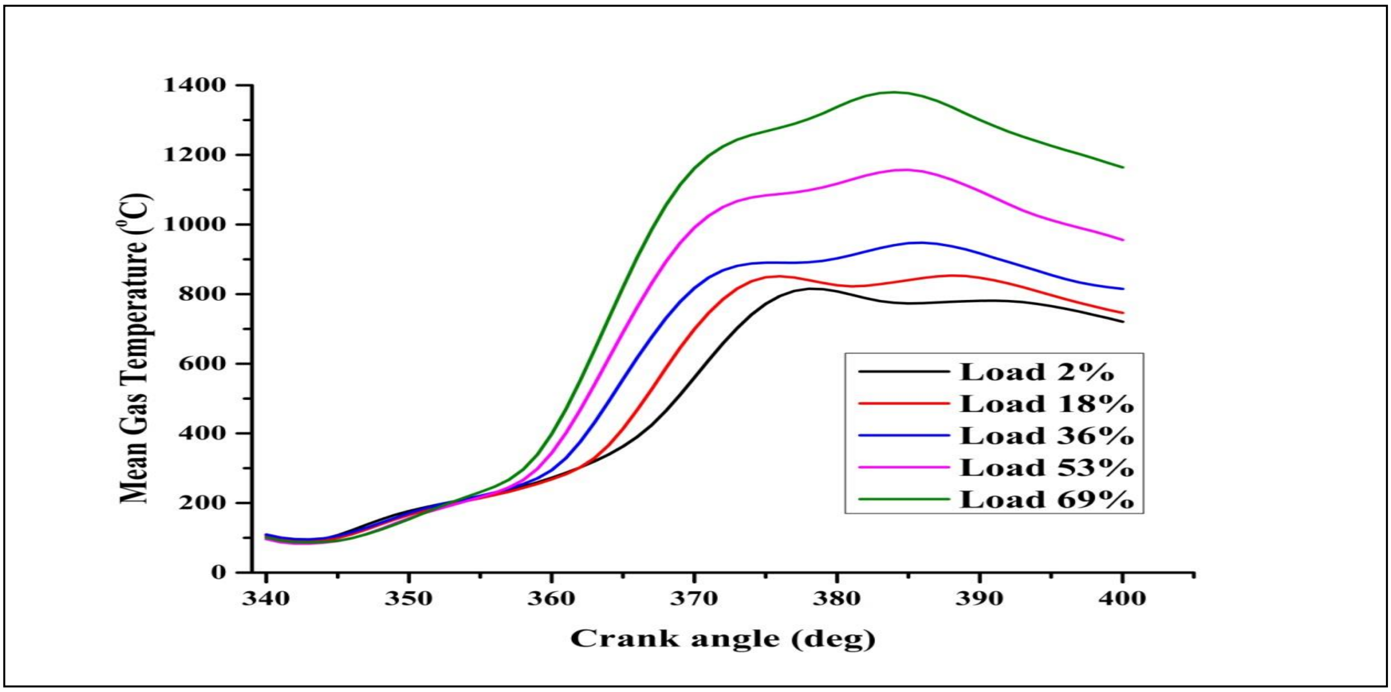

Fig 13: Variation of Mean Gas Temperature (MGT) Versus Crank Angle (deg) at Various Load (%) of Pure Diesel

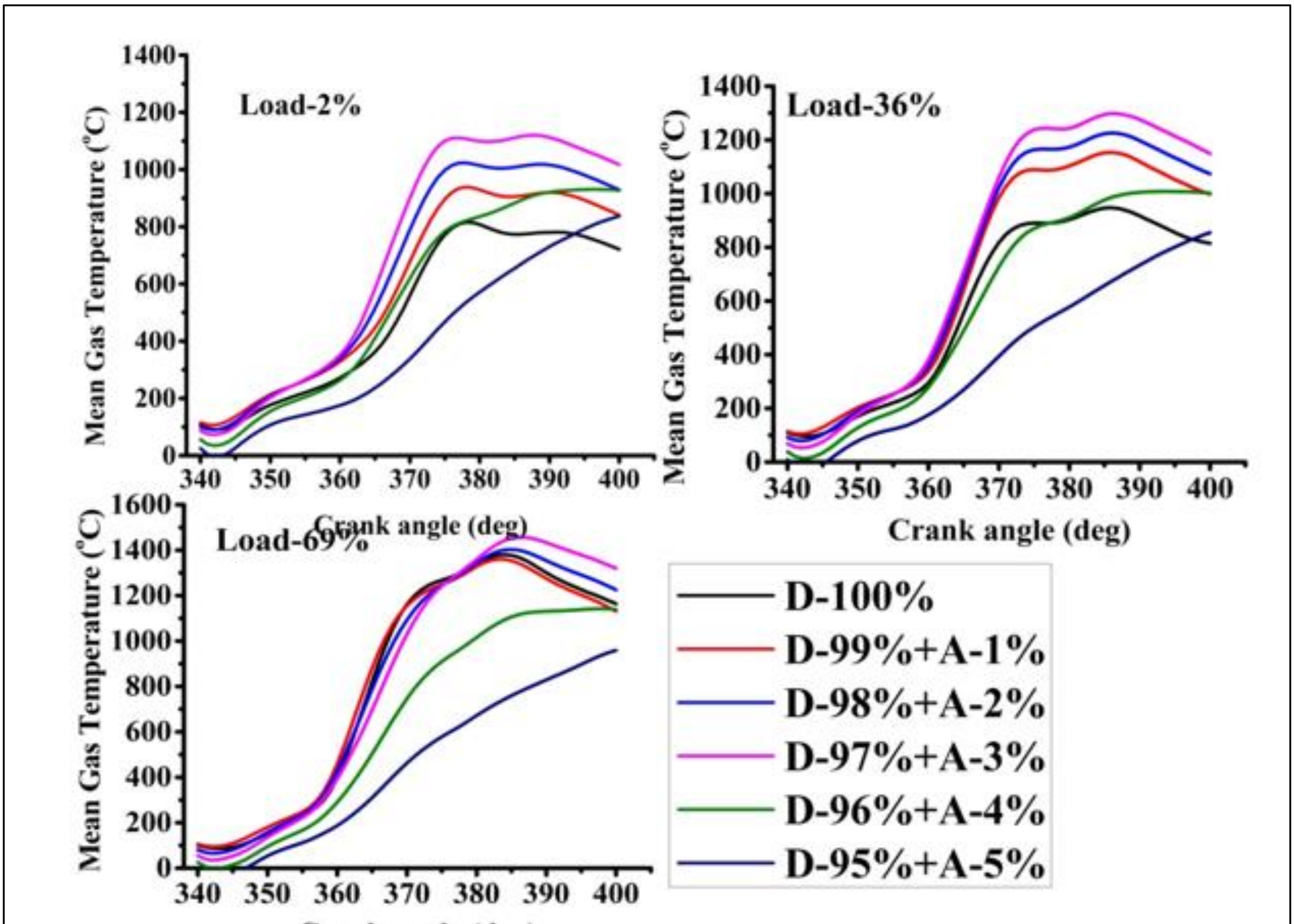

Fig 14: Variation of Mean Gas Temperature (MGT) Verses Crank Angle (deg) at Various Load (%) Conditions with Different Substitution of Additive (DTBP)

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Figure 16 shows the distinction between the mean gas temperature (MGT) versus crank angle with different load conditions by adding hydrogen as gaseous fuel in the pure diesel pathway (case 4). The enhancement in the peak of the MGT by adding different percentages of hydrogen as gaseous fuel at 2%, 36% and 69% load circumstances. However, the diminishment in MGT was observed to be 375.11 ºC by the addition of 16% hydrogen at 2% load with pure diesel as compared to 781.28 ºC of pure diesel pathway. At low load, when the hydrogen substitution in the dual-fuel engine increased, the amount of diesel fuel supplied decreased automatically, but it was not entirely eliminated. As the diesel supply was reduced, the average temperature within the cylinders may have decreased. Since hydrogen has a higher auto-ignition temperature (858 K) than diesel fuel (473 K), some hydrogen molecules might remain unburned in the exhaust as unburned hydrocarbons at higher levels of hydrogen substitution. However, under medium and high load conditions, the mass flow rate of diesel fuel increased compared to low load conditions, which influenced the engine's peak in-cylinder temperature [37].

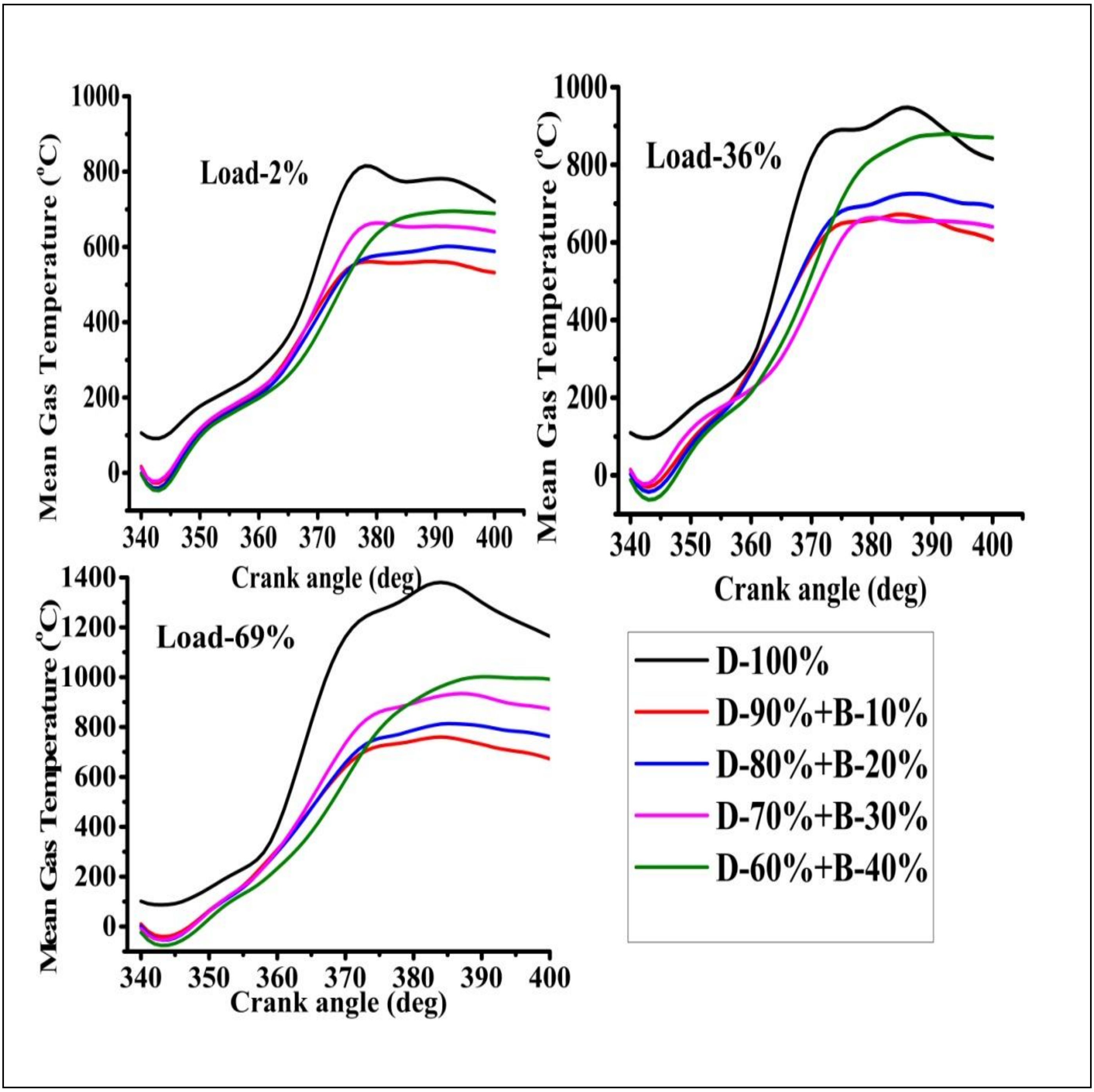

Fig 15: Variation of Mean Gas Temperature (MGT) Verses Crank Angle (deg) at Various Load (%) Conditions with Different Substitution of Biodiesel of Karanja Oil (BKO).

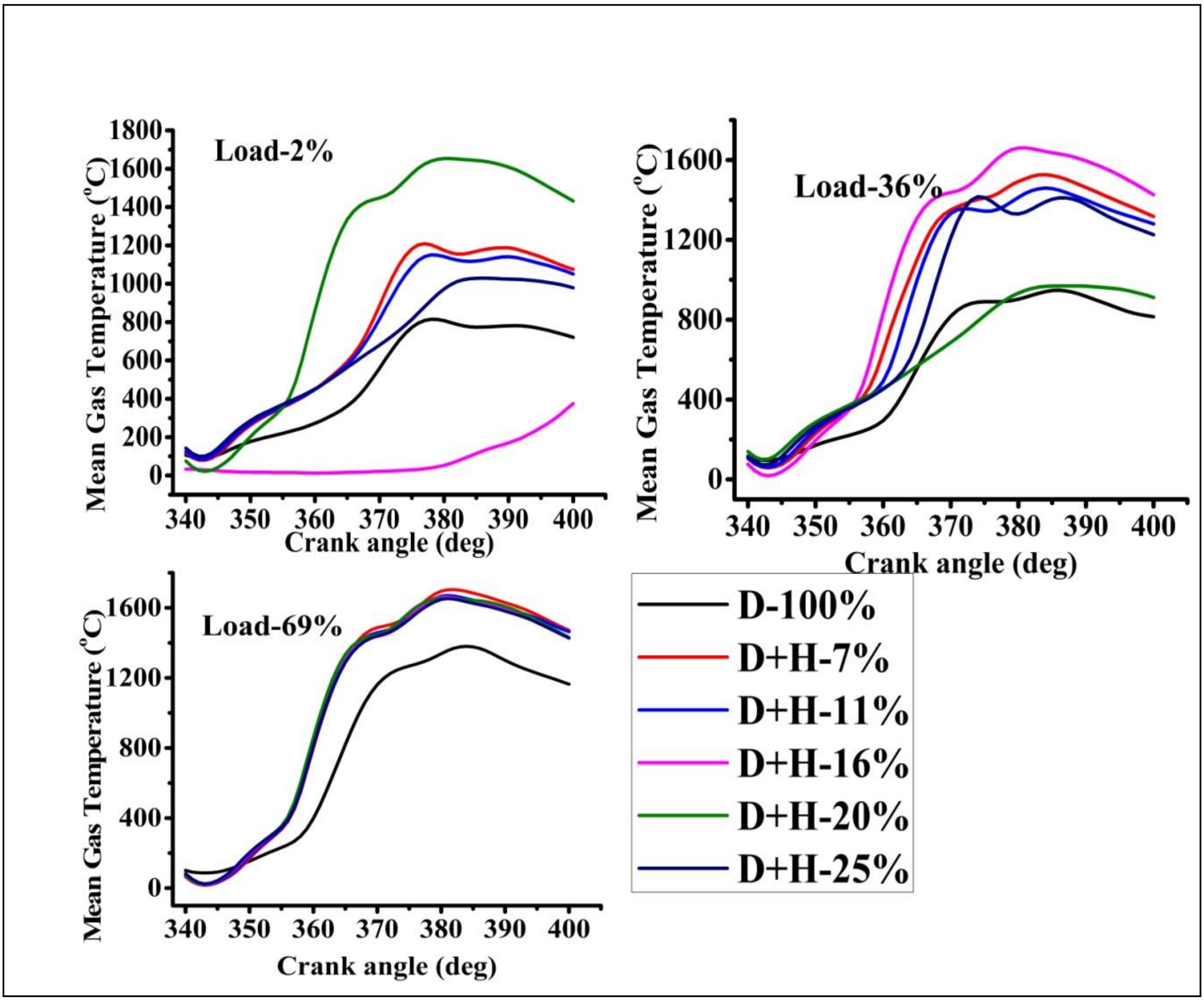

Fig 16: Variation of Mean Gas Temperature (MGT) Verses Crank Angle (deg) at Various Load (%) Conditions with Different Substitution of Biodiesel of Karanja Oil (BKO) and Hydrogen as Secondary Fuel.

Figure 17 shows the distinction between the mean gas temperature (MGT) versus crank angle with different load conditions by the addition of BKO and hydrogen with the diesel fuel (case 5). The diminishment in the peak of the MGT with the addition of 10% BKO with different percentages of hydrogen substitution in a dual-fuel diesel engine at 2%, 36%, and 69% load circumstances. The diminishment in the peak of the MGT with the addition of 10% BKO with 7% hydrogen substitution at 2% and 69% load circumstances was observed to be 541.03 and 652.92 ºC respectively as compared to 781.28 and 1380.26 ºC of pure diesel pathway. At medium load circumstances, the maximum peak of the MGT with the addition of 10% BKO with 11% hydrogen substitution was observed to be 646.77 ºC diesel as compared to 947.68ºC of pure diesel pathway. Moreover, the enhancement in the peak of the MGT with the addition of different percentages of hydrogen substitution at 2%, 36% and 69% load circumstances.

Figure 18 shows the distinction between the mean gas temperature (MGT) versus crank angle with different load conditions by the addition of BKO and hydrogen with the diesel fuel (case 5). The diminishment in the peak of the MGT with the addition of 20% BKO with different percentages of hydrogen substitution in a dual-fuel diesel engine at 2%, 36%, and 69% load circumstances. The diminishment in the peak of the MGT with the addition of 20% BKO with 16%, 20%, and 7% hydrogen substitution at 2%, 36%, and 69% load circumstances was observed to be 589.1, 671.72, and 837.35 ºC respectively as compared to 781.28, 947.68, and 1380.26 ºC of pure diesel pathway. Moreover, the diminishment in the peak of the MGT with the addition of different percentages of hydrogen substitution at 2% except 36%, and 69% load circumstances.

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Figure 19 shows the distinction between the mean gas temperature (MGT) versus crank angle with different load conditions by the addition of BKO and hydrogen with the diesel fuel (case 5). The diminishment in the peak of the MGT with the addition of 30% BKO with different percentages of hydrogen substitution in a dual-fuel diesel engine at 2%, 36%, and 69% load circumstances. The diminishment in the peak of the MGT with the addition of 30% BKO with 7%, 25%, and 16% hydrogen substitution at 2%, 36%, and 69% load circumstances was observed to be 587.72, 681.53, and 573.91 ºC respectively as compared to 781.28, 947.68, and 1380.26 ºC of pure diesel pathway. Moreover, the diminishment in the peak of the MGT with the addition of different percentages of hydrogen substitution at 2%, 36%, and 69% load circumstances.

Figure 20 shows the distinction between the mean gas temperature (MGT) versus crank angle with different load conditions by the addition of BKO and hydrogen with the diesel fuel (case 5). The diminishment in the peak of the MGT with the addition of 40% BKO with different percentages of hydrogen substitution in a dual-fuel diesel engine at 2%, and 36%, and 69% load circumstances. The diminishment in the peak of the MGT with the addition of 40% BKO with 20% hydrogen substitution at 2%, and 36% load circumstances was observed to be 682.72, and 545.51 ºC respectively as compared to 781.28, and 947.68 ºC of pure diesel pathway. At high load, the maximum peak was observed to be 871.27 ºC with 20% hydrogen substitution as compared to 1380.26 ºC of pure diesel pathway. Moreover, the diminishment in the peak of the MGT with the addition of different percentages of hydrogen substitution at 2%, 36%, and 69% load circumstances due to the use of BKO and hydrogen substitution creates a synergistic effect that reduces the peak MGT across different load conditions by altering the combustion dynamics.

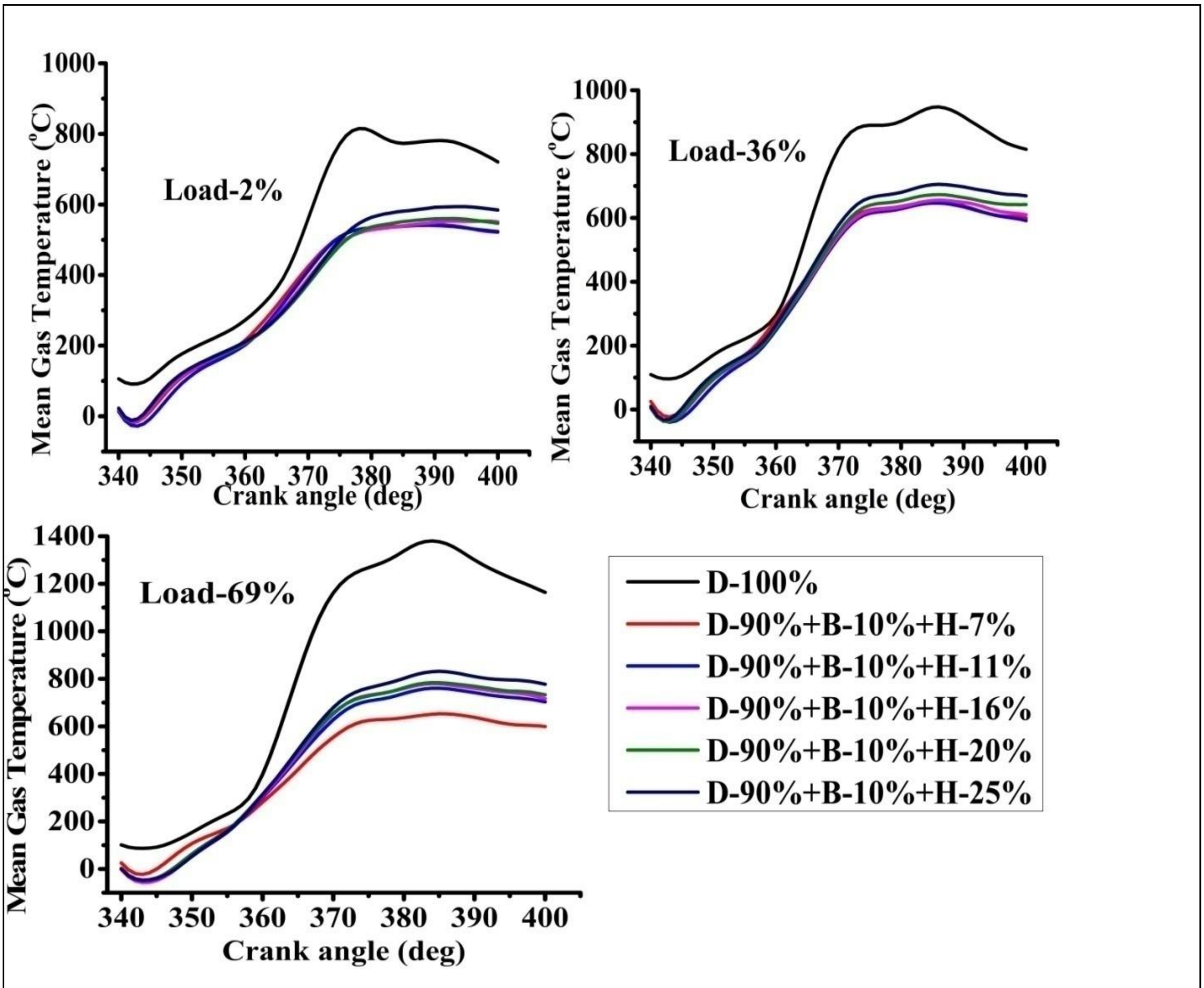

Fig 17: Variation of Mean Gas Temperature (MGT) Verses Crank Angle (deg) at Various Load (%) Conditions with Different Substitution of Biodiesel of Karanja Oil (BKO) and Hydrogen as Secondary Fuel

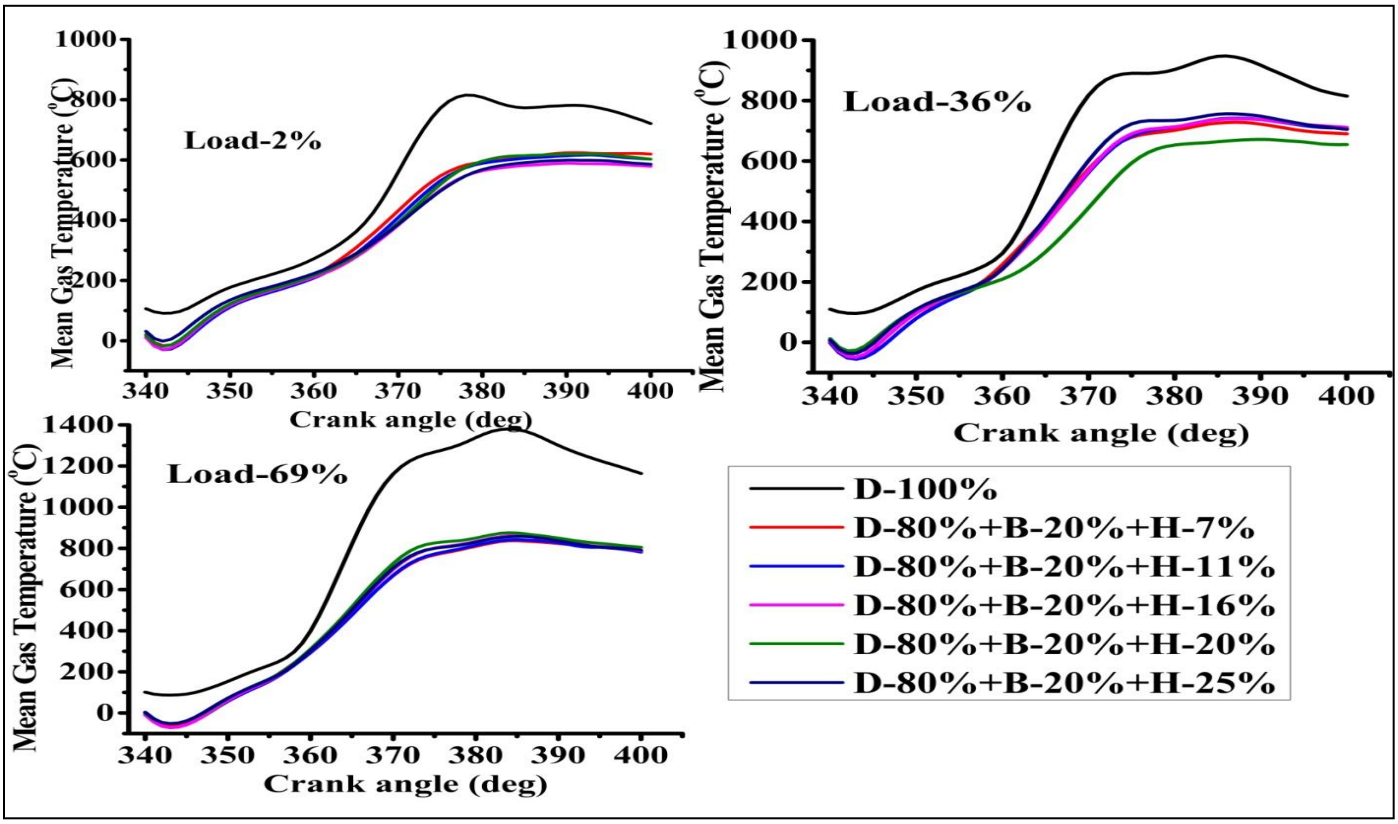

Fig 18: Variation of Mean Gas Temperature (MGT) Verses Crank Angle (deg) at Various Load (%) Conditions with different Substitution of Biodiesel of Karanja Oil (BKO) and Hydrogen as Secondary Fuel

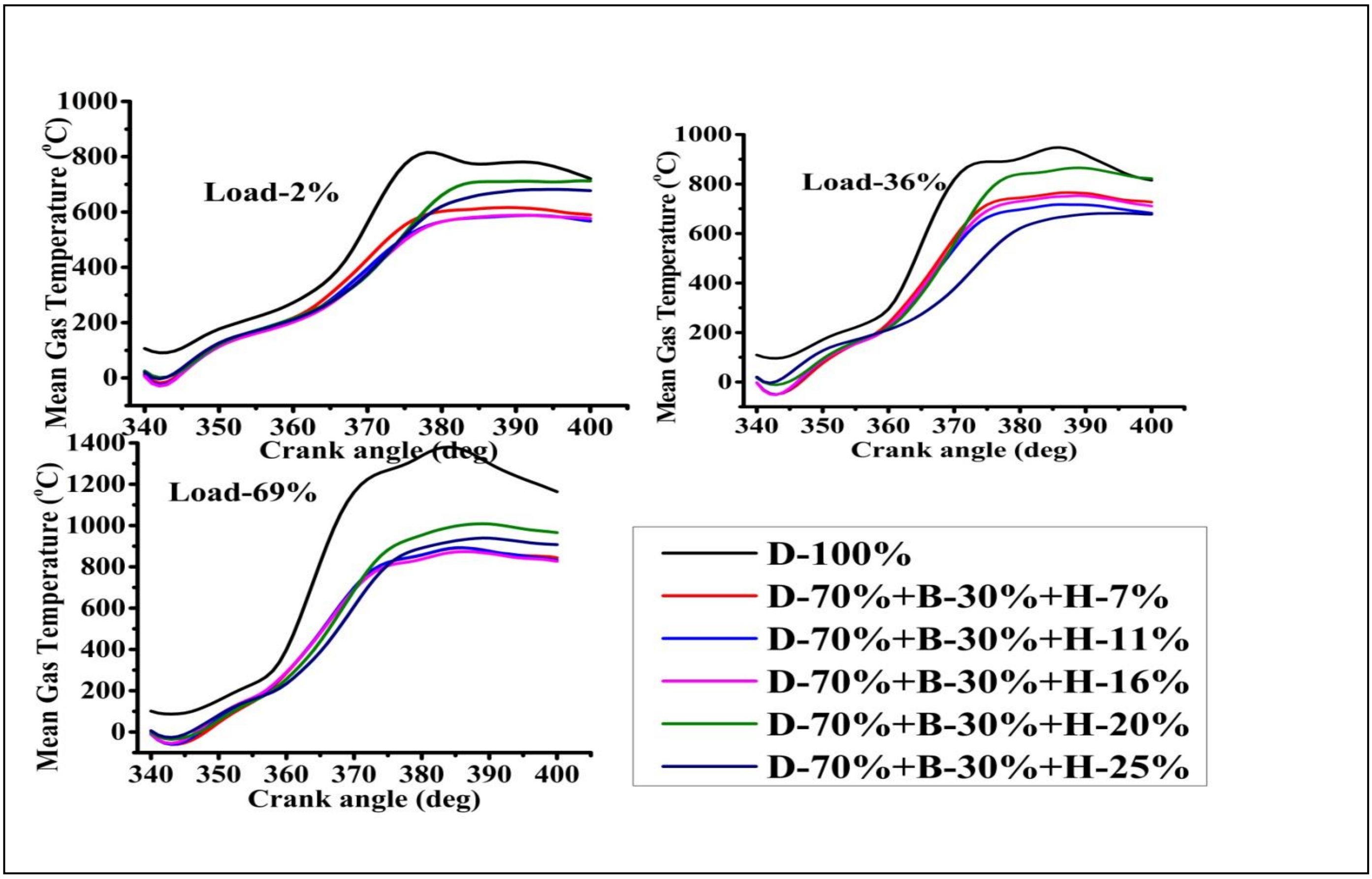

Fig 19: Variation of Mean Gas Temperature (MGT) Verses Crank Angle (deg) at Various Load (%) Conditions with Different Substitution of Biodiesel of Karanja Oil (BKO) and Hydrogen as Secondary Fuel

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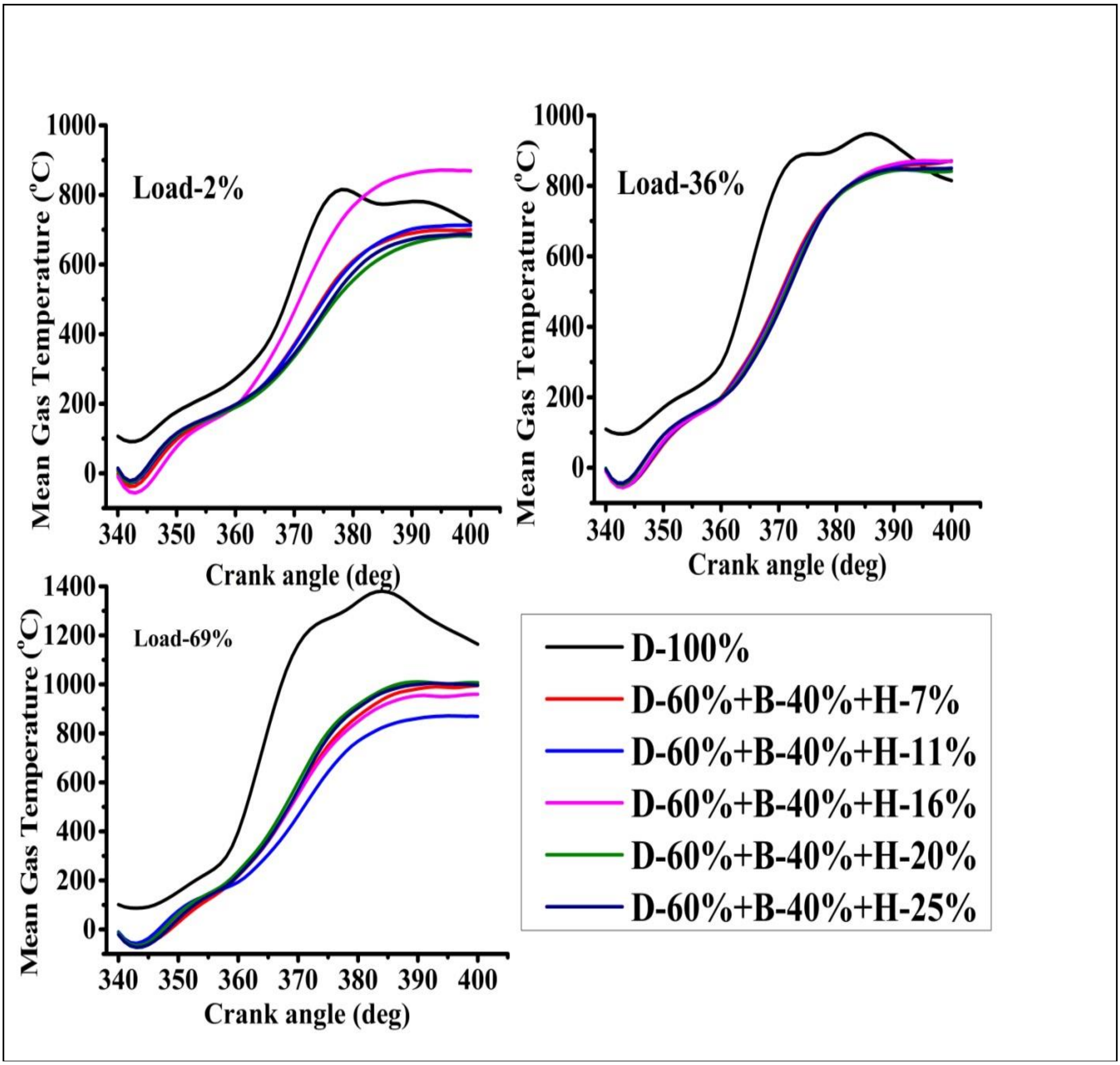

Fig 20: Variation of Mean Gas Temperature (MGT) Verses Crank Angle (deg) at Various Load (%) Conditions with different Substitution of Biodiesel of Karanja Oil (BKO) and Hydrogen as Secondary Fuel.

Additionally, the mean temperature of the cylinder dropped when an additive was added to the diesel fuel and hydrogen was utilized as a secondary fuel, in contrast to Case IV, where diesel and hydrogen were combined in the dual fuel diesel engine. The temperature of mean cylinder with crank angle for Case VI at 1%, 3%, and 5% additive replacement is shown in Figs. 21-23. In contrast, at 69% load conditions, the temperature of mean cylinder was found to be 600.29 °C when 5% additive was added to the diesel fuel and 25% hydrogen was used as secondary fuel in the diesel engine (Case IV). At high loads, increasing the addition of DTBP and hydrogen substitution in the dual-fuel engine raised the cetane number of the blend. As a result, the mean temperature in the cylinder increased, although it was still lower than when only diesel fuel was used, and the supply of diesel fuel was automatically reduced [37]. With the reduction in diesel fuel supply, the average temperature in the gas cylinders may have decreased. Since hydrogen has a higher auto-ignition temperature (858 K) than diesel fuel (473 K), some hydrogen molecules are likely to remain unburned in the exhaust as unburned hydrocarbons when hydrogen substitution levels are high. However, at high load conditions, the mass flow rate of diesel fuel increased compared to low load conditions, which influenced the engine's peak in-cylinder temperature [37].

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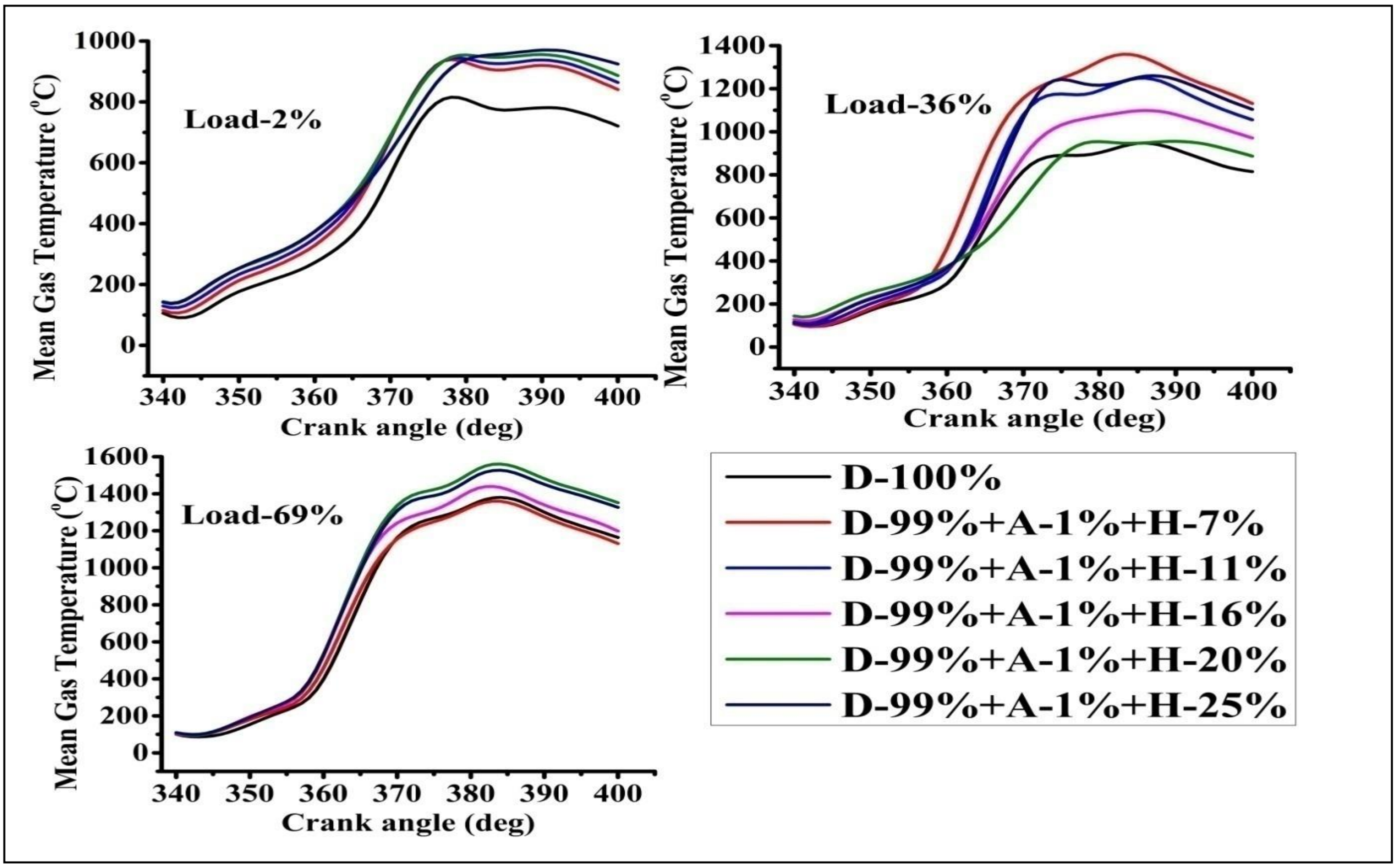

Fig 21: Variation of Mean Gas Temperature (MGT) Verses Crank Angle (deg) at Various Load (%) Conditions with different Substitution of Biodiesel of Karanja Oil (BKO) and Hydrogen as Secondary Fuel

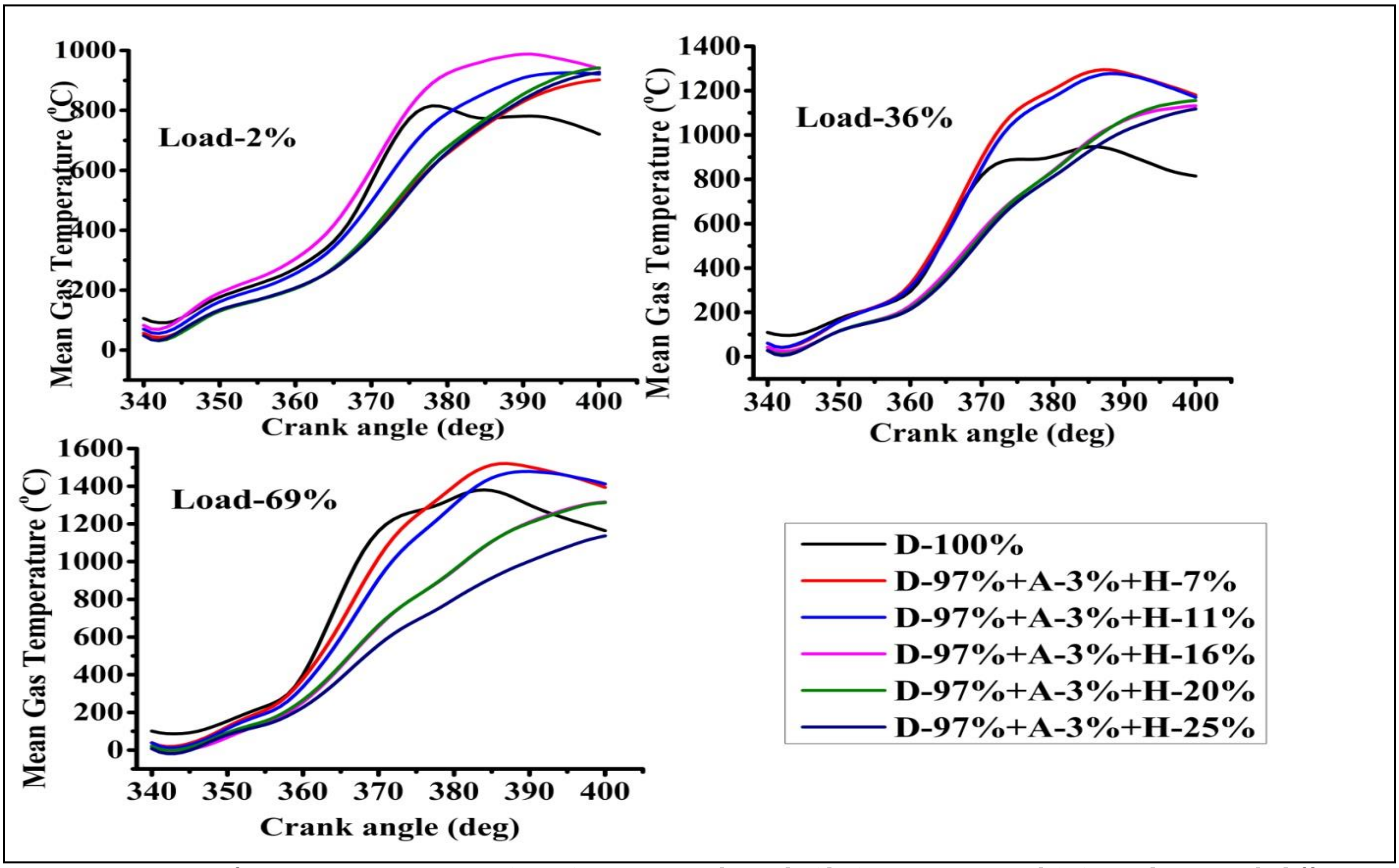

Fig 22: Variation of Mean Gas Temperature (MGT) Verses Crank Angle (deg) at Various Load (%) Conditions with different Substitution of Biodiesel of Karanja Oil (BKO) and Hydrogen as Secondary Fuel

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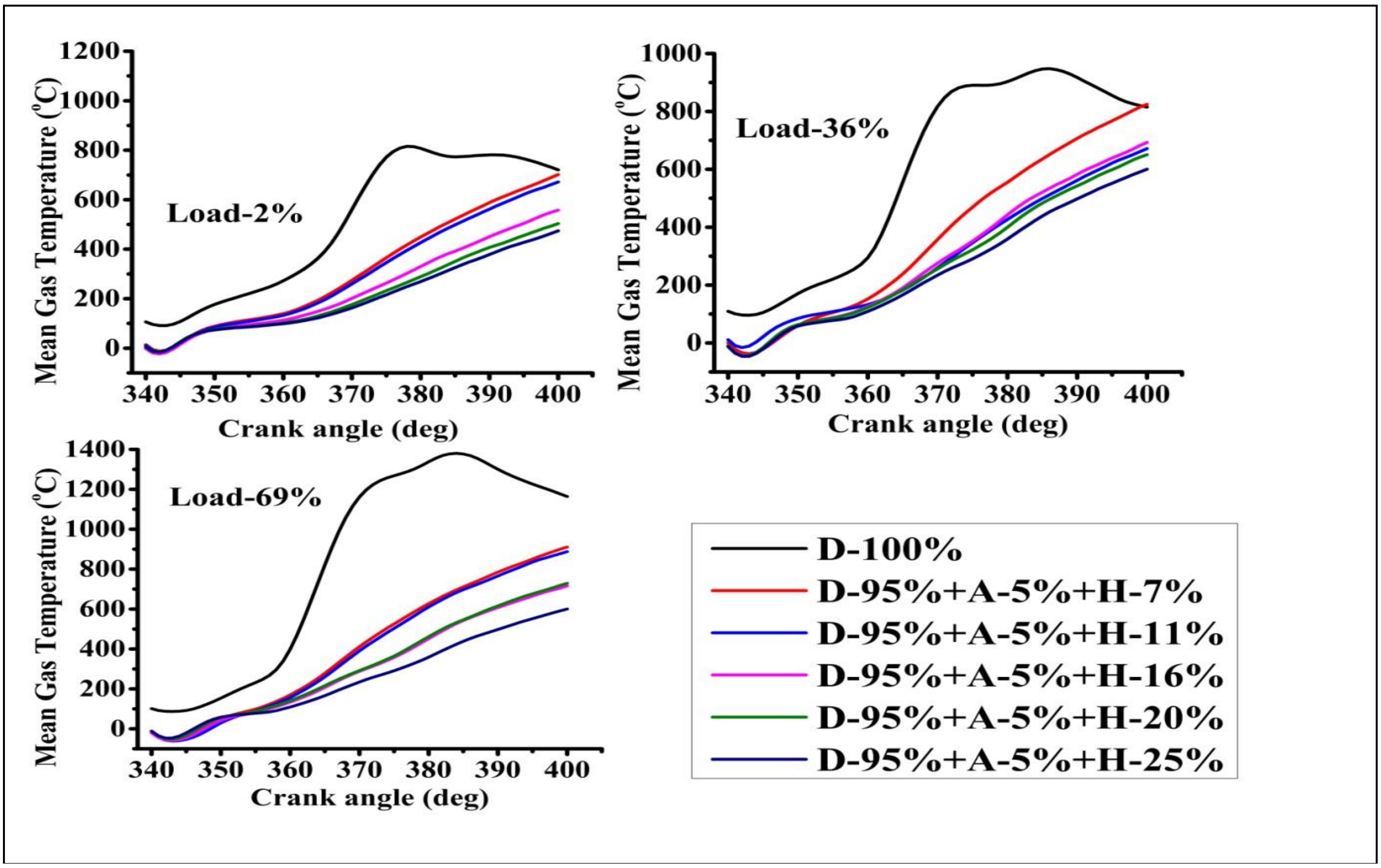

Fig 23: Variation of Mean Gas Temperature (MGT) Verses Crank Angle (deg) at Various Load (%) Conditions with Different Substitution of Biodiesel of Karanja Oil (BKO) and Hydrogen as Secondary Fuel

III. CONCLUSION

- At high load, the maximum value of net heat release (NHR) and the peak of mean gas temperature (MGT) in cylinder were observed to be 18.21 j/deg and 958.91 ºC by the addition of 5% DTBP.
- At low load, the maximum value of net heat release (NHR) and the peak of mean gas temperature (MGT) in cylinder were observed to be 15.83 j/deg and 561.16 ºC by the addition of 10% BKO.
- At all load, the value of net heat release (NHR) and the peak of mean gas temperature (MGT) in cylinder were increased by the substitution of 7%-25% hydrogen as a secondary fuel.
- At low load, the maximum value of net heat release (NHR) and the peak of mean gas temperature (MGT) in cylinder were observed to be 13.41 j/deg and 541.03 ºC by the addition of 10% BKO with substitution of 7% hydrogen as a secondary fuel.
- At high load, the maximum value of net heat release (NHR) and the peak of mean gas temperature (MGT) in cylinder were observed to be 12.83 j/deg and 600.29 ºC by the addition of 5% DTBP with substitution of 25% hydrogen as a secondary fuel.

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AUTHOR CONTRIBUTION

- Sunil Mahto (Conceptualization; Data curation; Investigation; Software; Visualization; Writing – original draft; Writing – review & editing)
- Ashish Kumar Saha (Supervision)
- Satish Saw (Investigation; Writing – review & editing)
- Chandra Bhusan Kumar – Conceptualization

➢ *Conflict of Interest*

No conflict of interest exists (Not applicable).

➢ *Declaration of Competing Interests*

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➢ *Ethics Declaration:*

- Not applicable.

➢ *Data and Code Availability*

- Not applicable

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➢ *Supplementary Information*

- Not applicable

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