An investigation of the relation between premixed burn fraction and emissions Characteristics of a common-rail HSDI diesel at different loads

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Abstract

This study investigates the effect of combustion phase (premixed and diffusion phases) duration on the emissions emitted from a common-rail high speed direct injection (HSDI) diesel engine fueled with ultra-low sulfur diesel fuel(ULSD) and runs at constant speed (1500 rpm) and at constant fuel injection pressure (800 bar), fuel injection timing (-9 ATDC) with varying loads (0,20,40,60,80 N.m) . In-cylinder pressure was measured and then analyzes this pressure using LABVIWE program and calculation program in MATLAB software to extract the apparent heat release rate, the ignition delay, combustion duration and the amount of heat released during the premixed and diffusion combustion phases . The influence of load on the exhaust emissions such as carbon monoxide (CO), total hydrocarbons (THCs), nitric oxides (NOx), smoke number (SN) and fuel consumption were also investigated. A result referring to that Increase of load lead to a decrease in the ignition delay which leads to reduce the premixed burn fraction, which plays a key role in the combustion and emission characteristics. But at high load Nox and soot emissions increase due to high temperature and increase of the diffusion burn fraction respectively.

بحث العلاقة بين جزء الاحتراق المسبق الخلط وخصائص الانبعاثات لمحرك ديزل عالي السرعة مباشر الحقن عند احمال مختلفة

مدرس حيدر عبد ضهد [1] ، استاذ مساعد دكتور محمد علي عبد الهادي[1] ، استاذ مساعد دكتور اخلاص محمد فياض[1] ، استاذ دكتور ثانوس ميكاريتس[2]

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الملخص

هذه الدراسة بحثت تاثير طور الاحتراق (طور الاحتراق المسبق الخلط والاحتراق الانتشاري) على الانبعاثات العادم من محرك ديزل عالي السرعة مباشر الحقن ، اجريت التجارب عند سرعة ثابتة (1500rpm) وضغط حقن

ثابت (800 bar) وتوقيت حقن (ATDC 9-) مع تغيير الحمل (0,20,40,60,80 N.m) . تم قياس الضغط داخل غرفة الاحتراق ومن ثم تحليل هذا الضغط باستخدام برنامج LABVIWE وبرنامج حسابي باستخدام برنامج MATLABلحساب معدل الحرارة المنطلقة و تاخر الاشتعال و مدة الاحتراق وكمية الحرارة المنطلقة في طور الاحتراق المسبق الخلط وكذلك في طور الاحتراق الانتشاري . تاثير الحمل على انبعاثات غازات العادم مثل اول اوكسيد الكاربون (CO) الهادروكابون غير المحترق الكلي (THC) واكاسيد الناتروجين (NOx) ورقم الدخان (SN)ومعدل استهلاك الوقود . النتائج اشارة الى ان زيادة الحمل نتج عنه نقصان في تاخر الاشتعال وبالتالي نقصان في جزء الاحتراق المسبق الخلط والذي يلعب الدور الاساسي في تحديد خصائص الاحتراق والانبعاثات ، ولكن عند الحمل العالي فان كل من NOx و السناج تزداد نتيجة لدرجة الحرارة العالية وزيادة جزء الاحتراق الانتشاري على التوالي .

1. Introduction

Diesel technologies play a significant role as a power source in many industries, from farming to road-building, to generating electricity, to powering everything from boats to trucks to automobiles. Diesel engines are typically chosen for these applications because they are significantly more fuel efficient and provide a longer service life compared to spark ignited gasoline engines [1] . New emission legislation and standards concerning environmental protection demand further improvements in internal combustion (IC) engine performance, emissions and fuel consumption. The two major pollutants, particulate matter (PM) and oxides of nitrogen (NOx), have been significantly reduced by 90% and 70% respectively over the last 10 years. Engine manufacturers have redesigned their engines to utilize higher fuel injection pressure, higher peak cylinder pressures and sophisticated electronic engine controls have replaced mechanical engine controls[2]. Furthermore, diverse combustion and injection strategies (Low Temperature Combustion LTC, Homogenous Charge Compression Ignition HCCI, and Premixed Charge Compression PCCI) have been studied among many research groups to reduce the exhaust emission and simultaneously increase the thermal efficiency of engine [3-6].

 In diesel engines, NOx is formed primarily by the thermal (Zeldovich) mechanism, wherein production rates increase exponentially with temperature [7-9]. Accordingly, recent research into in-cylinder strategies for NOx emissions reduction has largely focused on strategies to reduce incylinder combustion temperatures. Such strategies can be broadly classified as low-temperature combustion (LTC). In all LTC strategies, the combustion temperatures are reduced by dilution of the in-cylinder combustible mixtures, either with excess charge gas to create mixtures that are more fuel-lean than stoichiometric, or with moderate to high levels of EGR. In either case, the diluent gases increase the fuel specific heat capacity of the combusting mixtures, thereby reducing the combustion temperatures. Soot formation has been extensively investigated with regard to combustion characteristics [10-14] and application of alternative fuels [15-17]. The model of soot formation in diesel spray has been proposed by Dec [18] , This model predicts that primary soot particles initially form at the leading edge of the liquid fuel jet where rich premixed combustion takes place due to initial fuel mixing process. However, as the vapor jet progresses across the cylinder, secondary soot particles, accountable for the majority of soot emissions produced by diesel engines, form downstream of the jet particularly around the jet periphery whereby mixing controlled diffusion combustion occurs. Recent studies on NOx and soot formation involved detailed analysis of the effect

of alternative fuels [19-26], fuel injection timing [27,28], injection strategy [29,30], and Exhaust Gas Recirculation (EGR) [31-33], fuel injection pressure [34-37].

Engine load is one of the most important factors affecting the performance of the engine and emissions, A large number of researchers studied the effect of the load on the engine performance and emissions

The main focus of this work was to investigate the effects of load on the combustion and emission characteristics of ultra-low sulfur diesel fuel in a high speed direct injection diesel engine HSDI and that use of the modern fuel injection system type common rail injection system.

2- METHODOLGY

2-1- Experimental setup and test conditions

Experiments were carried out in a 2.0 l, 4 cylinders, 16 valves, direct Injection Ford's Duratorq (Puma) diesel engin , coupled to a Schenck eddy current dynamometer. The schematic of the experimental setup is shown in Fig. 1 and the main specifications of the engine are provided in Table 1.
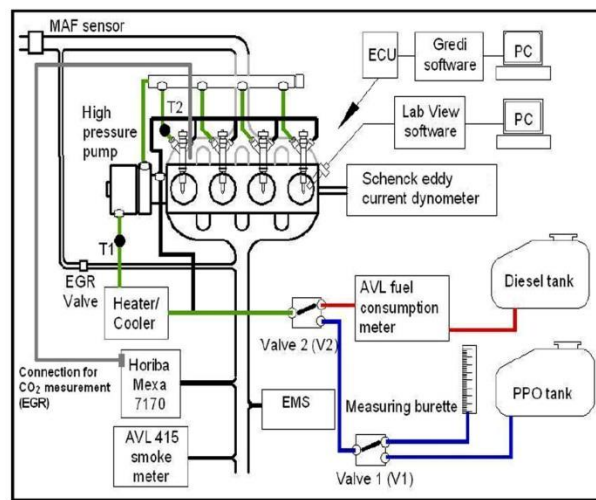

Fig.1. The schematic of experimental

Table 1
Engine specifications.

Multi-cylinder turbo-charged DI diesel engine	
Displacement (cm^3)	1998.23
Cylinder number	4
Compression ratio	18.2:1
Bore (mm)	86
Stroke (mm)	86
Cod-Rod length (mm)	155

In this investigation the engine was operated under naturally aspirated mode. The engine is fully instrumented, which enables the measurement of in-cylinder pressure and exhaust gas emissions under steady-state engine operating conditions. The in-cylinder pressures were measured using a Kistler pressure transducer fitted into the first cylinder of the engine. The signal from pressure transducer was amplified by the charge amplifier and then recorded by the LabView software in conjunction with the shaft encoder. In-cylinder pressure data were collected over 100 engine cycles per measurement, and the measurement was repeated 5 times for each point in the experimental matrix .The in-cylinder pressure data was averaged from100 cycles Output from the LabVIEW program are showed inFig.2.A common rail fuel injection system with six holes injector of0.154 mm in diameter each , and a spray-hole angle of 154 degree was used in this investigation.
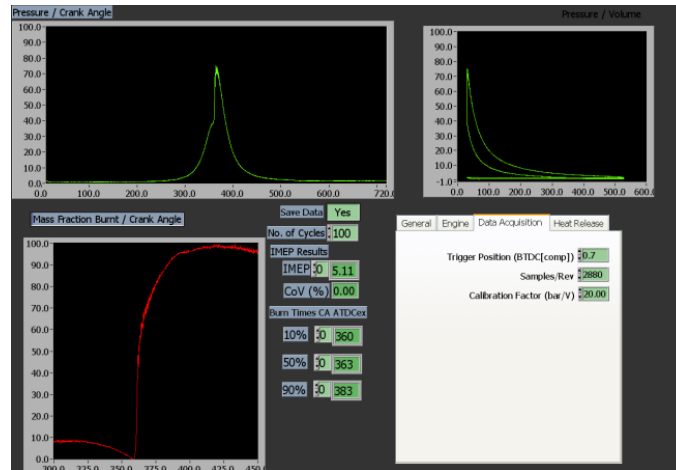

Fig.2 LabVIEW program screen

In this study, The Gredi software allowed the control and change of these parameters by programming the ECU in real time. , injection timing could be directly controlled through the software. The gaseous exhaust emissions were acquired using a Horiba-Mexa 7170DEGR gas analyser. A non-dispersive infrared method has been used for measuring the CO and CO2 emissions. The NOx emissions have been measured using chemiluminescence technique whereas the total unburnt hydrocarbons (THC) were measured using the flame ionization detection technique, Figure 3 show sample of these measurements.

Fig.(3) Horiba measurement screen

Emissions data were recorded over 180 s intervals, twice for each point in the experimental matrix. Again, this process was repeated for confirmatory purposes.

The engine exhaust smoke emissions were measured using the AVL – 415 smoke meter while the diesel fuel consumption was measured using an AVL fuel consumption meter, which is based on gravimetric measurement principle .

Measurements were carried out using the standard ultra-low sulfur diesel fuel. Table 2 provides an overview of all tested engine operating conditions .The fuel injection pressure , injection timing and speed conditions have been chosen to represent the commonly used operating condition for   the stationary driving cycle for automotive diesel engines.

Table (2) experiment matrix

Injection timing (ATDC)	Injection pressure bar	Engine speed rpm	Load N.m
-9	800	1500	0
			20
			40
			60
			80

After start-up, the engine was allowed to warm up until hydrocarbon emissions (the slowest pollutant to stabilize) had settled to an apparent steady state; this typically took around 90 min under an80 Nm load. At each operational condition data collection was postponed until all emissions had reached apparent consistency when viewed over a 180 s duration.

2-2 Data Analysis

Raw data collected using LabView was loaded into MATLAB and batch processed to retrieve the pertinent information. In-cylinder pressure was processed to extract relevant data-peak pressure, angle of peak pressure, angle between start of combustion (SOC) and peak pressure- and to calculate apparent heat release rate (AHRR) data using the traditional first law heat release model(38) without any modeling of heat transfer or crevice effects, and using an assumed constant specific heat ratio of 1.35. This is a very basic approach, but it is held to be sufficient for the purposes of comparison.

Approximations of cp/cv were made from the logarithm of pressure versus logarithm of volume charts in order to ensure that a constant value would represent fuel equally well. The calculated ratios of specific heats were found to be essentially consistent for ULSD under varying conditions, and so an assumed value of 1.35 was found to be adequate.

The definitions on which calculations of AHRR related parameters were based are illustrated in Figure (4) and explained in the accompanying text. Each 100 cycle pressure data set was used to generate a single average pressure trace, in order to reduce noise

while maintaining the essential characteristics of combustion. It should also be noted that although pressure data was only logged by the data acquisition system once per crank angle degree, all values were interpolated to one decimal place by the MATLAB code. All heat release parameters were calculated from the AHRR curve without filtering or averaging, except for the end of combustion, which was defined on the basis of the moving average of AHRR in order to improve consistency, and the end of premixed burn, which was calculated from the second derivative of AHRR, The following is a definition of the combustion characteristics in the fig.(4)

1. Start of injection (SOI) was defined from the commanded SOI set within the engine management software. Any potential difference between commanded and actual SOI, due to solenoid delay for instance, should be consistent between measurements, since engine speed was held constant.

2. Ignition delay (ID) was defined as the difference between commanded SOI and calculated SOC.

3. Start of combustion (SOC) was defined as the point at which the AHRR curve crossed the x axis; that is, the heat release rate became positive.

4. Premixed burn fraction (PMBF) was defined as the integral of the AHRR curve between SOC and EOPMB, divided by the integral of the AHRR curve between SOC and EOC.

5. End of premixed burn (EOPMB) was defined as the first point at which the second differential of heat release rate reached a local maximum following the global minimum. Under most conditions, this approximately corresponds to the position at which the AHRR reaches a first local minimum following the global maximum, but the second differential was used instead because under low loads there local minimum in the AHRR curve is not always clear, as can be seen in Figure 4.

6. End of combustion (EOC) was defined as the first point at which the moving average of heat release rate dropped below zero. A moving average was used to minimize problems due to noise, while still being representative of the general tendencies of the data.

Other values calculated from the in-cylinder pressure data including total apparent heat release, peak AHRR, PMBF, 10−90% burn fraction intervals, duration of partial burn fraction intervals, and average burn rates through partial intervals. Emissions data were averaged over the180 s durations recorded.
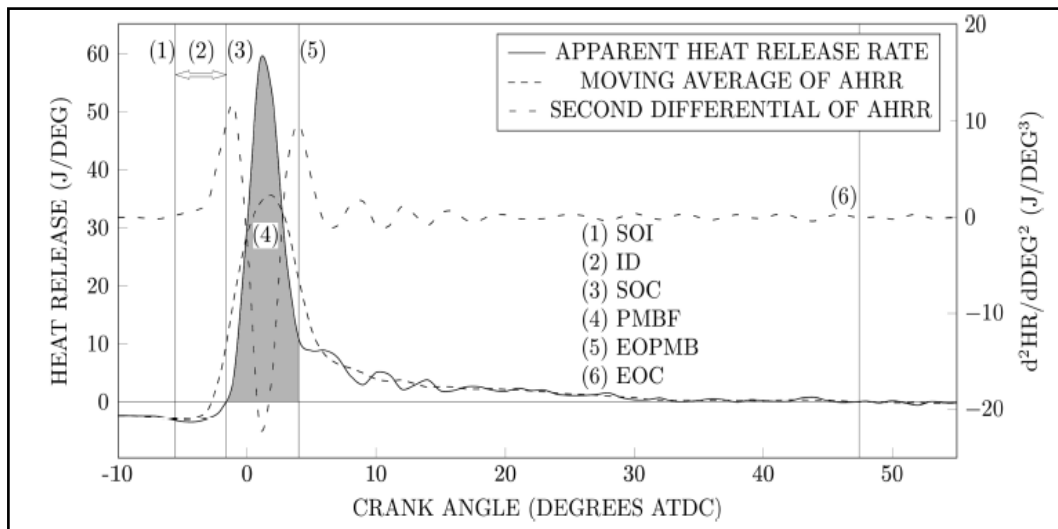

Fig.(4) Labeled plot of heat release and the derivatives used to calculate combustion criteria

3-Results and discussions
3-1 Combustion characteristics

Figure 5 shows the variations of the in-cylinder pressure with crank angle at different loads (0N.m= 0 bar BMEP, 20 N.m = 1.25 bar BMEP, 40 N.m = 2.5 bar BMEP, 60N.m = 3.75 bar BMEP and 80 N.m = 5 bar BMEP) at fuel injection pressure (800 bar) and injection timing (-9 ATDC) , it can be seen that as the load increases from 0N.m to 80 N.m the combustion starts earlier. In addition to this, the in-cylinder pressure peak increases with higher load. The increase in the pressure peaks with the increase of load are shown in Figure (6),

Fig.(5) In-cylinder pressure at different loads

Fig.(6) In-cylinder pressure peak at different loads

When increase the load , the amount of fuel burned increase , thus increasing energy released from fuel, which leads to increase in-cylinder pressure .The increase of the heat released with the increase in load are shown in Figure (7), In addition, the heat released peak moving forward because earlier starts of combustion. Figure (8) shows the difference between heat released rate peak at different loads , It is observed that AHRR peak at high load is higher than that at low load .This may be because operation temperatures are higher at high load.Vaporization of fuel will occur more readily than at lower temperatures, reducing the physical component of ID time. Therefore, it is probable that the observed reduction in ID at higher load is related to the impact of temperature change upon the physiochemical properties of fuel (its viscosity, density, heat capacity, and surface tension, vapor pressure, etc.)., which make the fuel generally at high load less resistant to vaporization as shown in figure (9). The ignition delay at low load was taller than at high load, Because of the relationship between the ignition delay and temperature, as explained above.
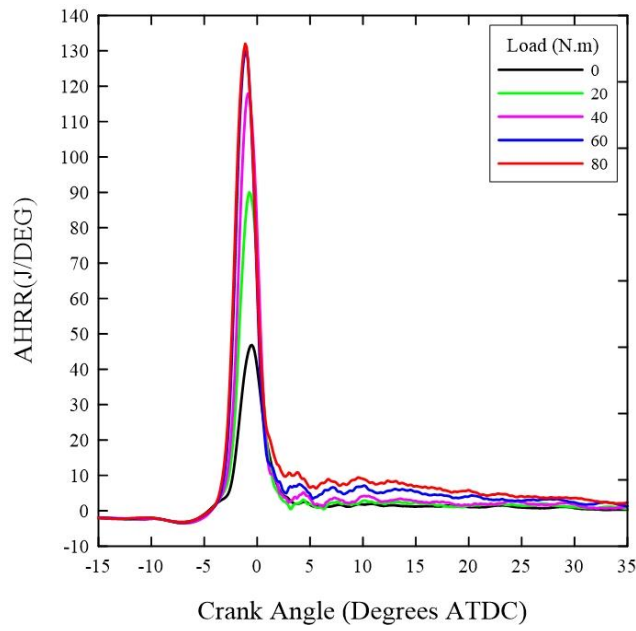
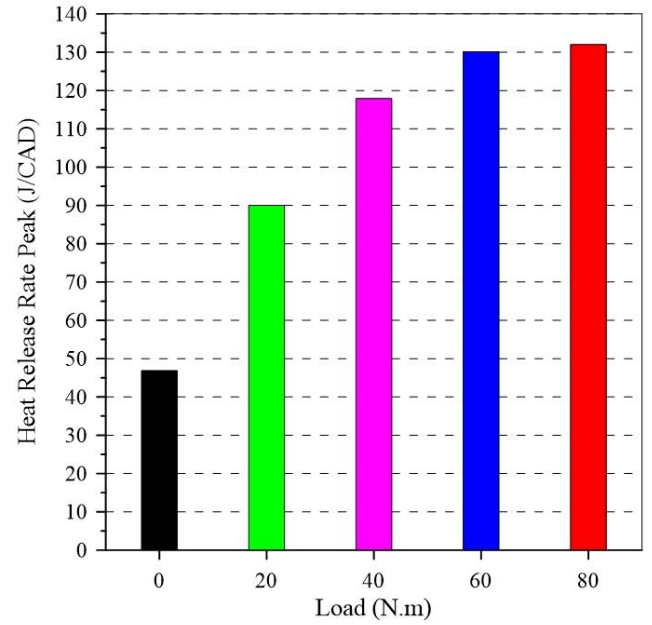**Fig.(7) apparent heat release rate at different loads**

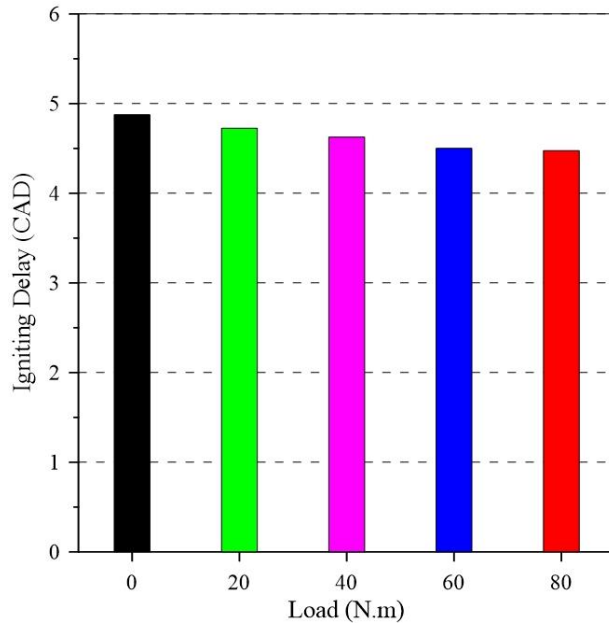**Fig.(8) Heat release rate peak at different loads**

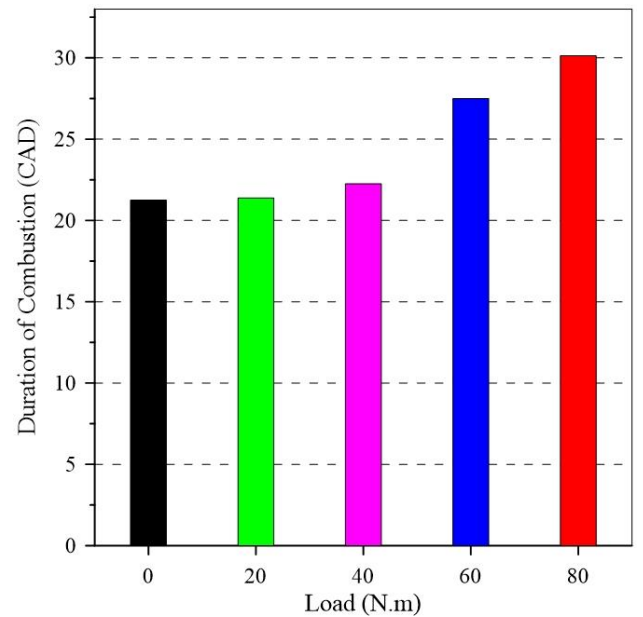**Fig (9) Ignition delay at different loads**

Fig (10) effect load on duration of combustion

Fig. (10, 11) shows the duration of combustion during the premixed and the diffusion combustion phases for different loads. The overall DOC is the sum of premixed and diffusion combustion durations. Within this sum, the change in the premixed DOC remained almost short, therefore the overall DOC inherits the trend of diffusion combustion duration for all cases. It is noticeable that duration of combustion for higher load longer than duration of combustion for lower load.
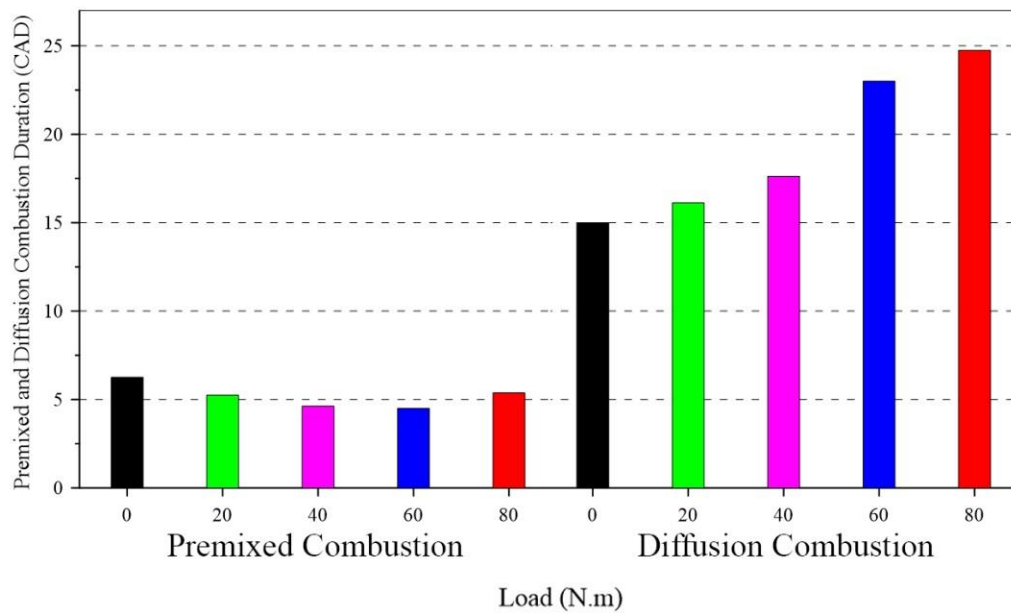
Fig (11) effect load on duration of premixed and diffusion combustion

It is noticeable that duration of combustion for high load longer than duration of combustion for low load.In order to understand this, Figure (11) provides information about the duration of combustion in the premixed phase as well as in the diffusion phase. The premixed and diffusion combustion duration are distinguished using the second differential of the heat release rate.There is a significant difference in diffusion combustion duration with increase of load , while there are no noticeable differences between the premixed combustion duration at high load and premixed combustion duration at low load. Despite shorter premixed burn duration, the magnitude of heat that is released during this phase (mass burn fraction) is higher compared to the magnitude of heat that is released in the diffusion phase fig. (12), As a result of the complete combustion resulting from combustion of premixed fuel –air mixture. In addition , It can be seen decreasing premixed burn fraction with the increasing of load and increasing diffusion burn fraction, Because of the increased amount of fuel burned during diffusion combustion and increased the duration of diffusion combustion with increasing of load.
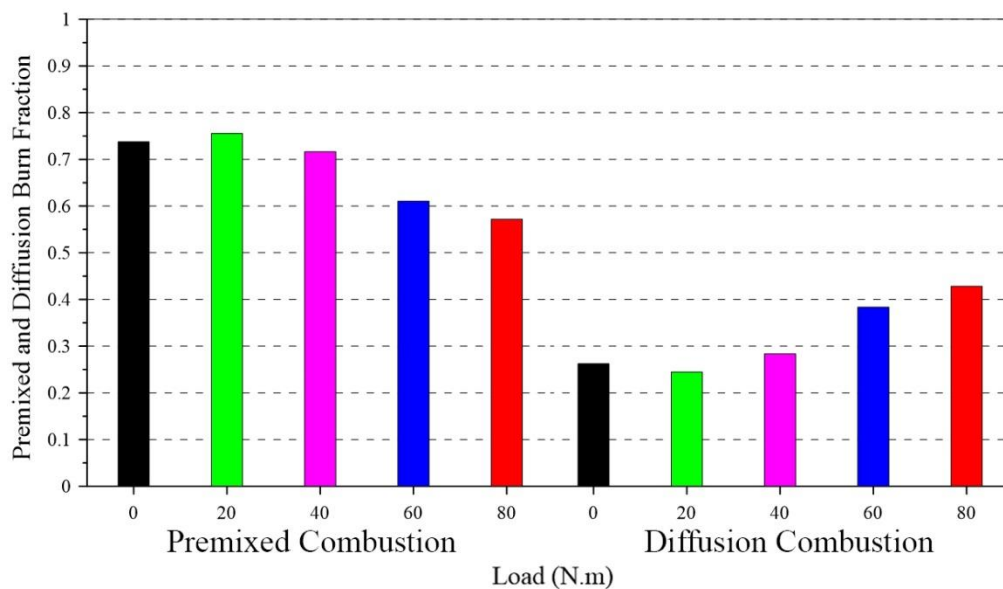

Fig (12) effect of load on   premixed and diffusion burn fraction

The Brake Specific Fuel Consumption (BSFC) for different loads are shown in fig. (13), the increase in BSFC at low load is a result of the incomplete combustion of fuel, Therefore, the engine running at low load is considered uneconomic.
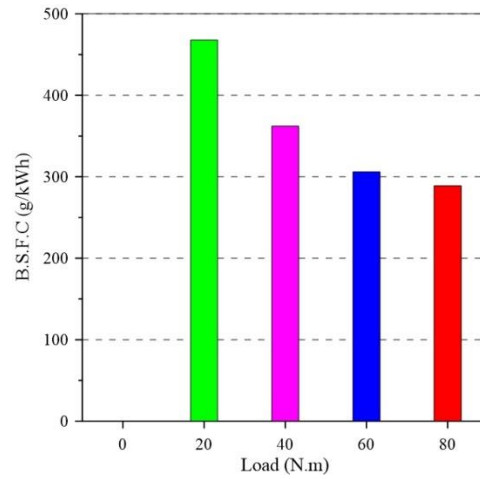

Fig (13) Effect of load on The Brake Specific Fuel

No value for B.S.F.C at zero load, because the calculation value will be indefinitely, due to the power generation equal to zero.

3-2 Emission characteristics

Figures (14, 15, 16, 17) show the variation of the nitrogen oxides emissions (NOx), the smoke number (SN), the carbon monoxide emissions (CO) and the total unburnt hydrocarbons emissions (THC) with load. In fig. (14) emissions characteristics of NOx for different loads are shown, it can be seen that an increase in the load leads to higher NOx emission. This increases due to the difference of temperature in combustion chamber . The NOx formation during combustion mainly depends on the Zeldowich mechanism where formation reactions are more intensive in a high temperature environment leading to higher NOx emissions. The highest cylinder temperature occurs during the premixed combustion phase where the formation of NOx is dominant.
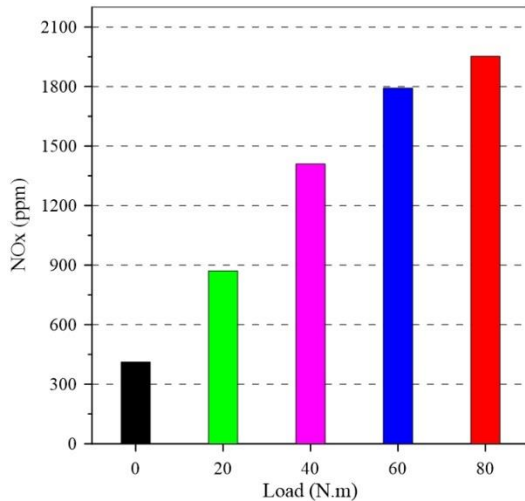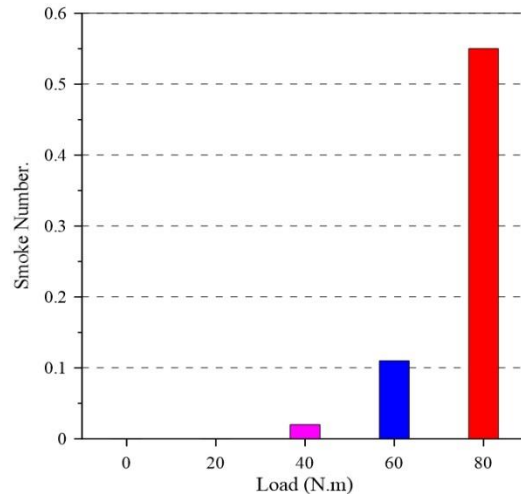

Fig (14) Effect of load on NOx emissions

Fig (15) Effect of load on smoke

Fig. (15) Shows the measured smoke number at different injection timings, we can clearly note the huge difference in soot emissions between high load and low load, This difference results from the large difference in the amount of diffusion combustion (diffusion burn fraction) between high and low-load and there's durations as shown in Figure (11) and figure (12) respectively .The heat release rate analyses are helpful in understanding NOx and soot emissions variations for different conditions. A lower magnitude of heat released in the premixed phase, may lead to lower NOx emissions. On the other hand, longer duration of the diffusion combustion phase leads to higher soot emissions. Figs. (16) and (17) show the variation of CO and THC emissions at different loads .In fig.(16) it can be seen that when load increases the CO emission decreases. Fig. (17) Shows the variation of   the THC emissions at different loads. The THC emissions decrease with higher load. However, the general emission trends for both THC and CO are similar. Both   CO and THC emissions are the products of incomplete combustion and they tend to decrease with higher load . The unburnt hydrocarbon emissions are generally formed as a result of flame quenching. The formation of higher CO and THC emissions are strongly related to the viscosity of fuel. Higher viscosity also leads to longer spray penetration. As a result, wetting of the cylinder walls eventually leads to the formation of higher CO and THC emissions by incomplete combustion.
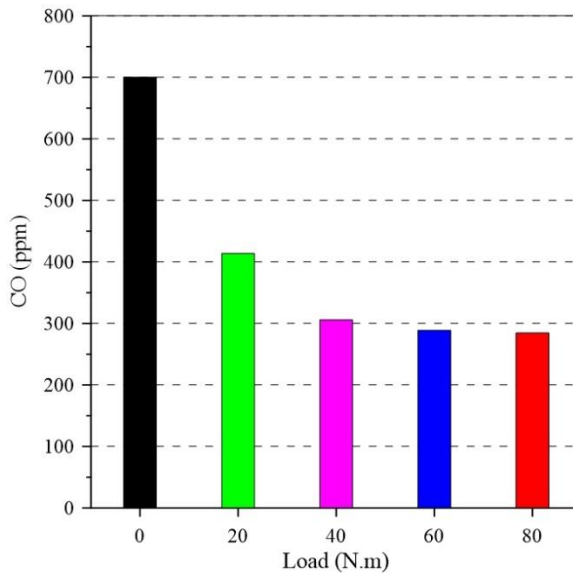

Fig (16) Effect of load on carbon monoxide emissions

Fig (17) Effect of load on (THC) emissions

4- Conclusions

1-   Increase of load lead to a decrease in the ignition delay which leads to reduce the premixed burn fraction, which plays a key role in determining the characteristics of combustion and emissions.

2- Increase premixed burn fraction lead to increase in temperature and thus increasing NOx emissions and reducing soot emissions. while with the increase of load the

premixed burn fraction will decrease , but Nox and soot emissions increase due to high temperature and increase of the diffusion burn fraction respectively.

3- The diffusion burn fraction at high load is higher than that at low load, which , leads to significant increase of soot emissions compared to those at low load.

4- (CO) and (THC) emissions are formed as a result of incomplete combustion, mainly due to the combustion taking place at low temperature in low load.

5- The Brake Specific Fuel Consumption (BSFC) shows high increase at low load as a result of the incomplete combustion of fuel, therefore, the engine running at low load is considered uneconomic.

Acknowledgment

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