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Corn and soybean biodiesel blends as alternative fuels for diesel engine at different injection pressures

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HIGHLIGHTS

- Pre-heat up to 80 °C for biodiesel blends should compensate the high viscosity.
- The increase of injection pressure (IP) improves engine performance parameters.
- At high IP (200 bar), peak pressure for diesel fuel exceeds that for fuel blends.
- The difference in the position of P_{max} for all fuels are in range of 1–2 °C.A.

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ABSTRACT

Experimental study has been carried out using corn and soybean fuel blends at the most recommended blending ratio of 20% biodiesel (C20 and S20) with conventional diesel fuel as alternative fuel for diesel engines. The effect of fuel injection pressure (IP) on diesel engine performance using C20 and S20 blends in comparison with that using neat diesel fuel is studied. Preliminary experiments regarding the analysis of fuel properties indicate that a preheat temperature up to 60–80 °C for these biodiesel fuel blends is necessary to compensate their high viscosity as compared with that of neat diesel fuel. A series of tests are conducted on four-stroke single cylinder air cooled direct injection (DI) diesel engine at different engine speeds, loads and IP of 180, 190 and 200 bar. The investigating parameters include the engine performance parameters (brake thermal efficiency – η_B and brake specific fuel consumption – BSFC) and other necessary parameters (air-to-fuel ratio – A/F ratio, mass of injected fuel – m_f, exhaust gas temperature – T_{exh}, cylinder wall temperature – T_{wall}, in-cylinder dynamic pressure – P_{cyl}, and both value and position of maximum pressure – P_{max} and $\theta_{@Pmax}$, respectively). The properties of corn and soybean blended fuels affect the fuel injection system and cause an increase in the duration of fuel injection to cover more time according to the increase in the amount of injected fuel necessary to overcome the power loss accompanied with the biodiesel low energy content. The major conclusion is that, the increased injection pressure gives better results regarding the engine performance parameters (both BSFC and η_B) in comparison with case of the original injection pressure for all tested fuels, thus the best results are obtained at high injection pressure of 200 bar. At this conditions it is concluded that, the increase of engine η_B and the decrease of BSFC approach 15% (from the original pressure of 180 bar), while the values of P_{max} for diesel fuel are slightly higher than those for blended fuels no matter the engine operating conditions.

1. Introduction

Diesel engines have been received a great attention due to their high power performance, thermal efficiency and low emissions in comparison with gasoline engines. This sector of transportation systems consumes a large portion of non-renewable petroleum fuels. Thus it is urgent to look for a renewable fuel resource that will replace (or at least reduce the consumption of) traditional fuels. One of promising resources is the production of biodiesel from vegetable oils. The physical properties of the biodiesel fuel such as viscosity, volatility and flash point affect the different processes occurring within the diesel engine cylinder; including fuel

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Nomenclature

ATDC	after top dead center	RPM	revolution per minute
A/F	air/fuel mass ratio, kg of air/kg of fuel	P_{cyl}	in-cylinder pressure, kPa
BTDC	before top dead center	P_{max}	maximum in cylinder pressure, bar
BSFC	brake specific fuel consumption, kg/kW h	T_{exh}	exhaust gas temperature, K
BP	brake power, kW	T_{fuel}	inlet fuel temperature, K
C.A.	crank angle, °	T_{wal}	engine wall temperature, K
CO	carbon monoxide, %	S20	fuel blend containing 20% soybean methyl ester biodiesel and 80% neat diesel
CN	cetane number	UHC	unburned hydrocarbon carbon, PPM
C20	fuel blend containing 20% corn methyl ester biodiesel and 80% neat diesel	U_R	uncertainty in the result "R"
IP	injection pressure, bar	X_i	variable, i
m_f	fuel mass flow rate, kg/s	η_B	brake thermal efficiency, %
NO_X	nitric oxide, ppm	η_V	volumetric efficiency, %

atomization, fuel evaporation, fuel mixing with air and fuel burning and thereby engine performance. The injection pressure (IP) plays an important role in metering the desired amount of fuel at correct time depending on engine operating conditions. For diesel engine, the direct injection fuel system is used to achieve a high degree of atomization in order to enable sufficient fuel evaporation in short time and sufficient spray penetration in order to mix fuel effectively with air and so enhance the combustion process. The fuel injection process is influenced not only by the fuel properties, but also by the injection system construction (number and dimensions of nozzles), the injection timing, and the injection operating conditions (as the injection pressure and the in-cylinder air conditions). It will be valuable to maintain engine parameters without (or at least with minimum practical) modifications to save time and efforts spent to develop the current fuel injection systems at safe and effective operating conditions. The biodiesel fuel atomization characteristics are expected to be worsened in comparison with that of neat diesel fuel. This behavior can be owing to the fact that, biodiesel fuels have higher values of density, viscosity, and molecular weight than those of neat diesel fuel. Thus biodiesel properties negatively interfere with the injection process leading to poor fuel atomization, incomplete combustion and excessive carbon deposits on fuel nozzles. In this regard, the regulation of fuel properties in conjunction with re-setting the fuel injection pressure may be acceptable solution to simultaneously regulate the fuel atomization and keep engine with minor modifications. At high the injection pressure, the droplets of the injected fuel become smaller and so better fuel atomization is achieved. Generally, the increase of injection pressure supports the completeness of fuel and air mixing providing better combustion process and thus to improve the engine specific power (kW/liter) [1]. The increase of injection pressure is recorded as one of the basic parameters that tend to reduce particulate matter emissions and fuel consumption in addition to other influences; including the increase of fuel portion burned by premixed combustion, the increase of mixture homogeneity, the increase of local A/F ratio, the decrease of combustion duration, the increase of the in-cylinder peak pressure, and the increase of NOx emissions [2]. For this reason, the improving of diesel combustion and emission characteristics by optimizing the fuel injection strategy received a great concern during the past few years especially those operated on traditional diesel fuels. But for engines operated with neat or blended biodiesels produced from corn and soybean oils there are few available literatures.

Reddy et al. [3] studied the effect of changing the IP on combustion and emissions characteristics of diesel engine using cotton seed oil methyl ester blended with diesel fuel. Authors concluded that, as the IP is increased from 170 to 200 bar, the values of brake thermal efficiency (η_B) are increased and those of brake specific fuel consumption (BSFC) are decreased. Kumar et al. [4] studied the effect of compression ratio, fuel atomization, IP, fuel quality, combustion rate, A/F ratio, intake temperature and pressure on engine performance parameters. Authors founded that, the increase in air motion into diesel engine improves the fuel atomization, the heat release rate and reduces the levels of exhausted emissions. Sayin et al. [5] studied the effect of fuel atomization and fuel distribution through combustion chamber using a single cylinder diesel engine operated with canola oil methyl esters (COME) and its blends with diesel fuel. The experimental results showed that, fuel exhibits different combustion and performance characteristics for different IP and engine loads. From their study, Sayin et al. [5] found that, (i) the use of COME instead of diesel fuel resulted in an earlier injection timing, (ii) the maximum in-cylinder pressure (P_{cyl}), the maximum rate of pressure rise and the maximum heat release rate are slightly lower for COME and its blend, (iii) the values of BSFC for COME are higher than those for diesel fuel while values of η_B for COME are lower than those for neat diesel fuel, and (iv) the increase of IP gave good results for BSFC and η_B compared to values obtained at the original IP. Kannan and Udayakumar [6] studied the effect of IP on performance and emissions from diesel engine. Authors concluded that, good performance and low emissions occur at high IP of 200 bar. Canakci et al. [7] observed the decrease of engine mechanical performance parameters (as in-cylinder peak pressure, rate of heat release, and engine efficiency) and the increase of most engine emissions (smoke opacity, UHC and CO) except NOx and CO_2 when the injection pressure becomes lower than the engine original injection pressure. Nagaraju et al. [8] carried out an experimental study to determine the effect of using B20 (fuel blend containing 20% soybean methyl ester biodiesel and 80% neat diesel fuel) on the combustion process, performance and exhaust emissions of diesel engine. Their results indicated that, the emissions of NO_X, CO, UHC and soot for B20 are lower than those for diesel fuel, while BSFC and T_{exh} are higher for B20 than for diesel fuel. Krahl et al. [9] studied the effect of using biodiesel fuels on diesel engine performance and concluded that, the high BSFC and low brake power (BP) obtained with biodiesel are related to the biodiesel low heating value. Song et al. [10] carried out an experimental study on a diesel engine fueled with soybean biodiesel under different engine loads and speeds. The results showed that, the BP, BSFC and torque are increased with the increase of biodiesel portion in the fuel blend. Prasad et al. [11] carried out an experimental study on diesel engine performance (a single cylinder, 4 stroke, naturally aspirated, direct injection, water cooled coupled with eddy current dynamometer kirloskar) fueled by castor non-edible vegetable oil and its blends with diesel fuel at 1500 RPM and variable loads. It is observed that fuel blend containing 25% of neat castor oil mixed with 75% of diesel (B25) is the best suited blend for diesel engine without heating and without any engine modifications. In this case, the values of η_B, BSFC for B25 are 33.45% lower and 54.76% higher than the corresponding values using neat diesel fuel. Bakar et al. [12] performed an experimental study to investigate the effect of IP (varied from 180 to 220 bar) on diesel engine performance. According to their results, the best performance is obtained at 220 bar. Purushothaman et al. [13] investigated the effect of IP on the combustion process and the exhaust emissions of diesel engine fueled with orange skin powder diesel solution. The results indicated that orange skin powder-diesel solution gives superior combustion and emissions characteristics as compared to diesel fuel at IP of 235 bar. Monyem et al. [14] reported that the advance of injection timing of about 2.3 °C.A. is necessary for fueling neat biodiesel fuels in-comparison with that for fueling a diesel fuel using the same fuel injection pump setting. The injection-timing advance is attributed to the physical property differences between biodiesel and diesel fuel; higher viscosity, higher molecular weight and fatty acid contents and so higher bulk modulus for biodiesel than that of diesel fuel [15]. The greater bulk modulus of biodiesels leads to an advance in fuel injection timing. This advance in the injection timing causes the fuel injector to open earlier leading to earlier combustion and an increase in the flame temperature [15,16]. The higher bulk modulus of biodiesel is observed to force the injector needle to open earlier by 1 °C.A. with respect to that caused by fossil diesel fuel [16]. In their work, Jaichandar and Annamalai [17] modified the piston bowl to study the effect of chamber geometry in conjunction with the effect of injection pressure on engine performance. The effect of piston bowl can be wing to the fact that, good bowl should produce better squish and swirling actions resulting in remarkable improve of fuel–air mixing and homogeneity earlier to start of auto-ignition. It is worthy to state that, the optimum injection conditions (that should match combustion chamber geometry to be able to generate sufficient turbulence) are those typically resulted in a fuel spray with fine droplet size (good fuel atomization), long tip penetration (better distribution), narrow spray angle (more aspirated air) and so better air-biodiesel mixing. At high injection pressure, finer atomized fuel droplets are formed and quickly vaporized compared to the larger droplets formed at low injection pressure that slowly vaporize. However the penetration of fine droplets is shorter and so their size distribution may need to be optimized [18]. This optimization can be achieved at the proper match between the IP and the combustion cylinder geometry [17].

The aim of the present work is to investigate the effect of blended fuels (containing 20% of soybean and corn biodiesels and 80% diesel fuel) on the engine performance parameters and the in-cylinder pressure at different engine operating conditions of load, speed, and injection pressure.

2. Experimental setup

The present study is conducted on a single cylinder diesel engine of specifications presented in Table 1a; the schematic diagram of the whole test facility is shown in Fig. 1. The engine injection system comprises of a Jerk type fuel injection pump that starts fuel injection at 24 C.A. deg. BTDC according to the engine specifications. The injector opening pressure has been set at a specific value to meet specific experimental condition of injection pressure to be able to achieve the current study objective; influencing the injection pressure on engine fueled by biodiesel fuel blends. Biodiesel fuels are prepared from corn and soybean oils and their properties are determined according to standard the ASTM testing methods by Egyptian Petroleum Research Center and the results are given in Table 1b. The test facility includes an electrical Direct Current (DC) dynamometer electric model MEZ-BURNO with maximum electric power output 10.5 kW) coupled to the engine output shaft. An external excitation voltage is used to generate the necessary magnetic field around the rotating rotor to convert the mechanical power into electric power depending on both the rotor speed and the excitation voltage. The rotor speed is the same as the engine shaft rotating speed. While the excitation voltage measured by a digital Alternating Current (AC) voltammeter (Radio-Shack of 750 V measurement range and 1 V resolution) is controlled and adjusted by the AC autotransformer and the supplied voltage is converted to DC excitation voltage via a rectifier bridge. The DC generator has been cooled to maintain its body at constant temperature of 25 °C to avoid the generator internal losses and so maintain its accuracy for mechanical power conversion into electric power. The electric power output from the DC generator is consumed by heating of flowing water throughout a water tank; thus the resistance of electric heaters is kept constant. The present system provides a facility to conduct engine performance tests at different values of engine loads. The engine is said to be working at a certain load ratio when the excitation voltage applied on the DC generator is adjusted to produce the prescribed ratio of engine output power relative to the engine full power at the engine rated speed. The cylinder pressure is measured using piezoelectric pressure transducer model Kistler 6123 (having pressure range up to 200 bar with sensitivity of 16.5 pc/bar) coupled with charge amplifier NEXUS 26290AOS1. The signals from the pressure transducers, proximity and thermocouples are converted from analog to digital data and recorded in PC with the help of data acquisition card (model CIO-DAS1602/12, 12-bit, 32 channel single-ended 16 differential channels) with typical sampling rate of 10^4 sample/s during 10 s then digitized data are saved in Excel sheet for later analysis before plotting the results using Sigma Plot software. For each test, the exhaust gas temperature, cylinder wall temperature, inlet air temperature, differential pressure across orifice plate at inlet air box, engine cylinder pressure, volumetric fuel consumption are measured. For studying effect of specific fuel on engine performance, the engine is allowed to operate until steady state (as the oil temperature is fixed usually this attained after 30 min) before starting the measurements to ensure the stability and the elimination of all previously studied fuel from fuel system. Then the test conditions are adjusted to explore the engine performance at the pre-defined operating conditions according to the experimental program summarized in Table 1c. Generally, the accuracy of the measured parameters is within 3–5% as stated in the next section.

Table 1a
Engine technical specifications.

Type	DEUTZ F1L511
Injection	Direct injection
Cooling type	Air cooled
Number of cylinders	1
Number of stroke per cycle	4
Bore, mm	100
Stroke, mm	105
Compression ratio	17:1
Rated brake power, kw	5.775 at 1500 RPM
Advance angle	24° BTDC
Injector pressure, bar	180
Number of nozzle holes	1

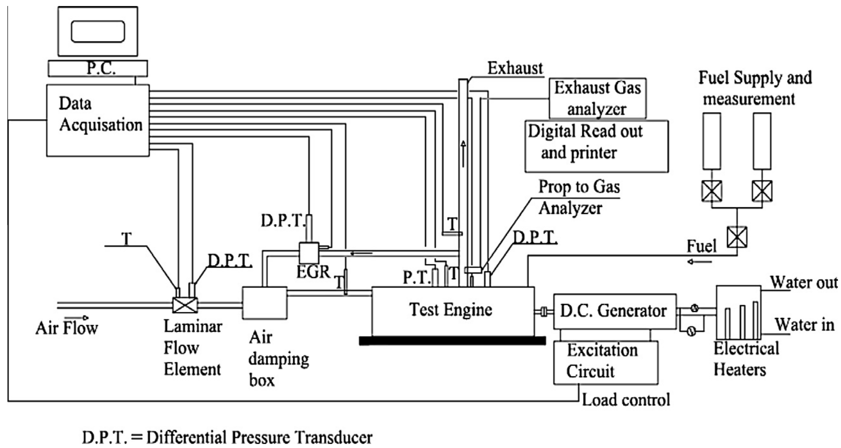

Fig. 1. Experimental setup.

Table 1b
Physical and chemical properties of diesel, vegetable oils and their biodiesel fuels.

Test properties	Test method	Diesel fuel	Soybean vegetable oil	Corn vegetable oil	S100 biodiesel	C100 biodiesel	S20 blend	C20 blend
Chemical formula		$C_{14.09}H_{24.78}$	$C_{56}H_{102}O_6$	$C_{56}H_{103}O_6$	$C_{18.74}H_{34.43}O_2$	$C_{17.89}H_{32.88}O_2$		
Cetane number	ASTM D613	42.7	38	37.6	51.8	52.5	44.52	44.66
Flash point (°C)	ASTM D93	52	254	277	158.8	165.7	71.6	74.74
Cloud point (°C)	ASTM D2500	−18	−3.9	−1.1	−2	−3	−14.8	−15
Pour point (°C)	ASTM D97	−32	−12.2	−40.0	−6	−5.1	−26.8	−18
Cold filter plugging point (°C)	ASTM D2500	−18	−2	−5	−3.6	−7.5	−5.12	−6.62
Lower heating value (MJ/kg)	STM D240	45.9	39.623	35.1	37.75	38.48	44.27	44.42
Density at 40 °C, (kg/m^3)	TMDA 1298	815	914	915	855	880	823	828
viscosity at 40 °C, (mm^2/S)	ASTMD445	3.1	33.1	35.1	4.29	4.5	3.338	3.38
Sulfur content (wt%)	ASTMD545	0.22	0.01	0.01	0.0	0.0013	0.001	0.001
Ash content (wt%)	ASTMD482-91	0.0055	0.006	0.01	0.006	0.01	0.006	0.01
Carbon residue (wt%)	ASTM D4530	0.1780	0.24	0.22	0.244	0.21	0.19	0.18
Distillation temp. (°C) 90%	STM D86	185–345	250–445	155–365	345	345.8	345	345.2
Carbon content (wt%)	ASTM D5291	86.7	78	77.15	77.03	76.71	84.77	84.7
Hydrogen content, (wt%)	ASTM D5291	12.71	11	11.82	11.9	11.52	12.55	12.47
Oxygen content, (wt%)	ASTM D5291	0.0	11	11.02	10.95	10.98	2.19	2.2
Carbon/hydrogen ratio		6.82	6.58	6.52	6.53	6.53	6.67	6.67
Iodine value (g) I/100 g	EN 14111	0	120–141	103–128	126	120	25.2	24
Sulfur content (ppm)					2.7	3		

Table 1c
Engine test conditions.

Regulating parameter		Measured/analyzed parameter
Item	Range of regulation	
Injection pressure	180, 190, 200 bar	• Fuel and air mass flow rates/air to fuel ratio
Fuel	Diesel (D100), blended fuels of C20 and S20	• BSFC and η_B
Engine speed	From 700 to 1500 RPM with step of 100 RPM	• Exhaust gas temperature
Engine load	50%, 67%, and 83% of the engine full load	• In-cylinder pressure

3. Experimental errors analysis

The difference between measured and true value of measured quantity is known as an error. The uncertainty in each individual measurements (X_i) leads to resultant uncertainty for any variable "R" that is computed from n independent measurements by the following relation [19]:

$$R = R(X_1, X_2, \ldots, X_n)$$

If the uncertainty in each independent measurement X_i is known from the measuring device specifications to be U_{Xi}, then the absolute resultant uncertainty in in the computed variable R is given by the following equation:

$$U_R = \sqrt{\left(\frac{\partial R}{\partial X_1}\right)^2 U_{X1}^2 + \left(\frac{\partial R}{\partial X_2}\right)^2 U_{X2}^2 + \cdots + \left(\frac{\partial R}{\partial X_n}\right)^2 U_{Xn}^2}$$

where the partial derivative $\left(\frac{\partial R}{\partial X_i}\right)$ is a measure of the sensitivity of the result to a single variable X_i.

From the previous relation, the relative uncertainty if R can be computed to receive non-dimensional value by dividing each term by R^2 and multiplying each term on the right-hand side by $(X_i/X_i)^2$, to obtain:

$$\frac{U_R}{R} = \sqrt{\left(\frac{X_1}{R}\frac{\partial R}{\partial X_1}\right)^2 \frac{U_{X1}^2}{X_1^2} + \left(\frac{X_2}{R}\frac{\partial R}{\partial X_2}\right)^2 \frac{U_{X2}^2}{X_2^2} + \cdots + \left(\frac{X_n}{R}\frac{\partial R}{\partial X_n}\right)^2 \frac{U_{Xn}^2}{X_n^2}}$$

Table 2
The relative uncertainties for measured or calculated variables used in this study.

Item	T_{exh}	T_{wall}	m_{air}	m_f	RPM	BP	BSFC	η_B	η_V	P_{cy}	μ	C.A. deg.
U_R/R (±%)	2	3	4	3	3	2	2	1	1.5	3	2	3

Applying this relation to compute the relative uncertainty in the determined variables used throughout this study from the independent measurements, the following specific uncertainty for each are obtained as collected in Table 2.

4. Results and discussion

4.1. Fuel properties

The properties of base diesel fuel ''Egyptian diesel fuel – No. 2 D'' in comparison with those of raw and biodiesel vegetable oils and their blends (supplied by the Egyptian international research center) are collected in Table 1b. It can be noticed that, the major properties of biodiesel fuels affecting the atomization, evaporation, and combustion processes such as heating value, viscosity, volatility and flash point completely differ from those of diesel fuel, thereby engine performance will be influenced by blending the base fuel with the tested biodiesels. Biodiesels almost have a 10% lower energy content than diesel fuel on mass base and so a greater volume of fuel shall be injected to maintain engine power. The high density of biodiesels over that of diesel fuel may partially compensate the low energy content on weight basis. This is importance for diesel engines because fuel is metered volumetrically. The greater bulk modulus of biodiesels leads to an advance in fuel injection timing causing earlier combustion and so an increase in the flame temperature. This behavior has been confirmed by Lahane and Subramanian [20], who reported an advance in the injection timing at all loads for different blending ratios (at engine rated load this advance varied by about 0.7 C.A. deg. BTDC and 1.5 C.A. deg. BTDC with B20 and B100, respectively, with respect to that of neat diesel fuel). As the biodiesels have a number of oxygen atoms in their chemical structure (commonly called oxygenated fuels), their burning will be proceed in a faster reaction rates and so faster rate of pressure rise. The combustion process in diesel engine occurs partially with premixed mode and mainly with diffusion mode. More premixed combustion means a high initial rate of pressure rise and high gas temperature prior the fuel combustion. Cetane number and fuel volatility are the two most important fuel properties that determine the amount of the fuel burned via premixed combustion mode and thus the combustion rate. Biodiesel's high cetane number is expected to shorten the ignition delay period and thus lower the amount of fuel that is involved with the premixed combustion portion of the biodiesel combustion, lowering flame temperature. Biodiesel's lower volatility (indicated by the high flash point) decreases the amount of fuel vaporized during the delay period and, therefore, also decreases the portion of premixed combustion. As the increase of injection pressure improves the fuel atomization, the ignition delay period is decreased with the increase of the injection pressure [21]. The carbon residue of biodiesel fuels is higher than diesel fuel due to difference in chemical composition and molecular structure, and so there exists a greater chance to increase the carbon deposition in the combustion chamber. The aromatics, a class of hydrocarbon compounds, are lower in biodiesel fuels than that in diesel fuel (in range from 25% to 35%) that are characterized by stable chemical structure but have great influence on the poly- and mono-aromatic hydrocarbon emissions the major potential for cancer causing. Viscosity is one of the most important properties of any fuel as it affect mainly the fuel system (by increasing the pumping energy and increasing the losses in fuel line) and the fuel spray pattern. The fuel viscosity is increased with the increased chain length (number of carbon atoms) and degree of fuel unsaturation; but the effect of structure degree of unsaturation is lower [13]. The high viscosity of biodiesel decreases the fuel leakage in a plunger pair and changes the parameters of the fuel supply process: larger quantity of injected fuel, advance of injection timing, and spray pattern (having smaller spray cone angle, larger droplet size, and larger fuel penetration length). Correspondingly, the biodiesel is usually mixed with conventional fuels as the previous influences regarding spray worsening mainly depend on the biodiesel fraction within the fuel blend. It is observed that, the increase of biodiesel fraction over 30% has a remarkable effect on spray evolution process which leads to the increase of Sauter droplet size, the decrease of droplet number density, and the increase of spray cone angle [20]. The formation of larger fuel droplets within intensified spray may cause the impingement of fuel spry into the relative cold wall of engine cylinder that interrupts the fuel combustion and so blue smoke is formed and the cylinder wall deposits are increased. Moreover, the high biodiesel fraction (greater than 25%) may promote the fuel spray to impinge the cylinder wall [20]. The variation of values of viscosity and density for the studied fuels versus temperature are shown in Fig. 2; fuel density is linearly decreased while viscosity is exponentially decreased with the increase of fuel temperature. It is noticed that, the viscosity of diesel, biodiesel, and blended fuels is decreased as the temperature is increased as the intermolecular attraction between different layers of the fuel becomes weak [13]. As observed from Table 1b and Fig. 2, the properties of raw vegetable oils fully differ from those of trans-esterified oils forming biodiesels. Viscosity and density of diesel fuel are lower than those of soybean and corn biodiesel fuels due to the removal of glycerin and heavy waxes within raw oils. The maximum difference between viscosities of C20, S20 and D100 are obtained at 40 °C. In order to bring physical properties of C20 and S20 biodiesels close to diesel fuel at normal temperature of 30 °C, biodiesels should be heated to 60–80 °C. Fuel density is decreased with the increase of temperature. The density of C20 and S20 biodiesels blends are closed to that of neat diesel fuel. These findings about the effect of preheat temperature on the viscosity and density of neat and blended fuels are comparable with those cited in the literature [22,23].

4.2. Mass of fuel injected and air/fuel ratio for different operating conditions

The fuel should be introduced into the combustion chamber at a precisely injection timing with a desired rate of injection that results in a desired heat release pattern. The spray-pattern should result in rapid mixing of fuel and air at all operating conditions of engine speed and load. Figs. 3a and 3b show the mass of injected fuel per cycle versus engine speeds for different fuels at different engine loads and IP. For all tested fuels, mass of injected fuel is significantly increased with the increase of engine speed and load but it is slightly increased with the increase of IP. Generally, the use of high injection pressure leads to the increase of the injected fuel velocity and so the quantity of injected fuel is increased [1]. Mass of injected fuel for both C20 and S20 is higher than that for neat diesel fuel at all engine loads and speeds to compensate the effect of lower heating values while maintaining specific engine power

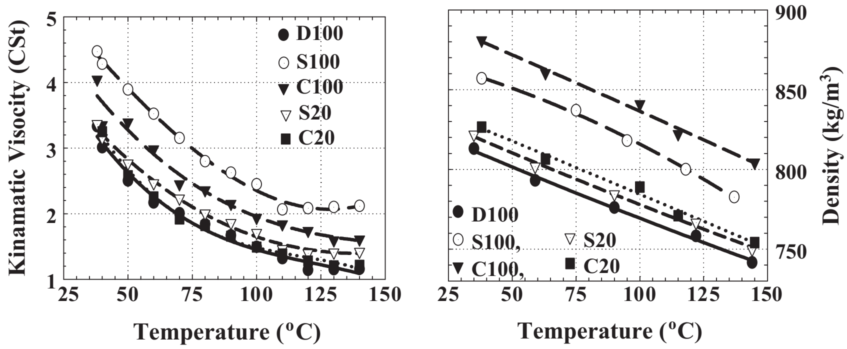

Fig. 2. Viscosity and density of different fuels versus temperature.

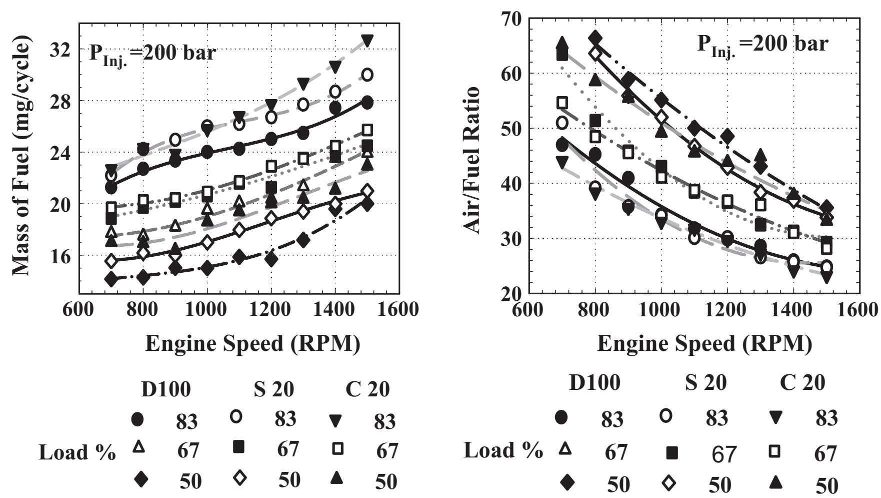

Fig. 3a. Mass of fuel injected and air/fuel ratio for different fuels and loads with injection pressure of 200 bar.

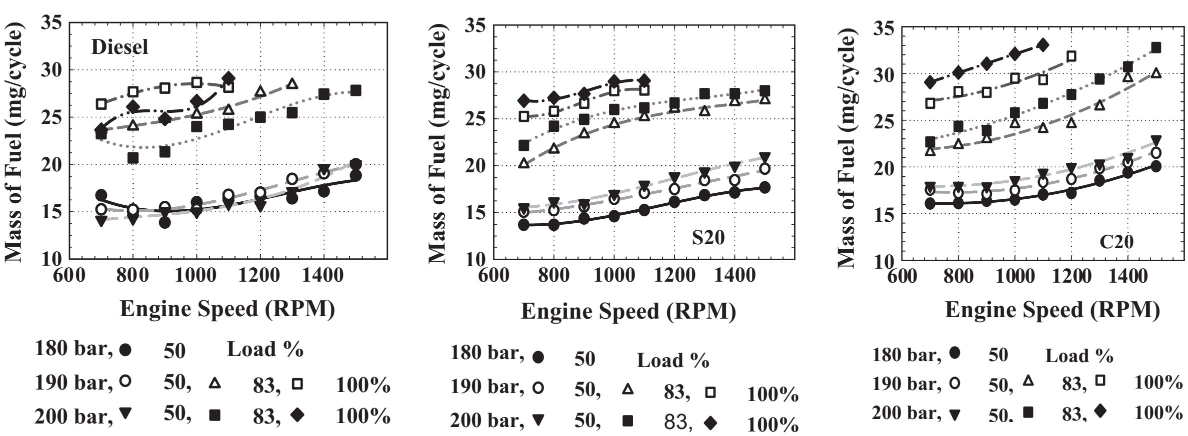

Fig. 3b. Mass of fuel injected with different fuels, loads and injection pressures.

and speed. The injected mass of C20 is higher than that of S20, that may be due to the following facts (1) its higher viscosity and density; (2) the lower fuel leakage throughout pump plunger; (3) the greater advance injection timing and longer duration of injection process. Mass of fuel injected is arrangement in descending order as C20, S20 and neat diesel fuel respectively according to their viscosity and density.

For diesel engine at a given speed no matter engine load, a constant air is supplied into engine cylinder and so the Air-to-Fuel (A/F) ratio changes based on mass of the injected fuel per cycle. As shown in Fig. 3c, the A/F ratio is decreased with the increase of engine speed and load due to the increase of injected fuel mass per cycle. For all tested fuels, the A/F ratio is decreased with increase of IP due to the increase of pressure difference across the fuel nozzle orifice which provides better chance for fuel velocity to be increased and so the mass of injected fuel is increased. For the same engine speed and load, C20 and S20 have A/F ratio lower than diesel fuel due to high mass of fuel injected. A/F ratio is

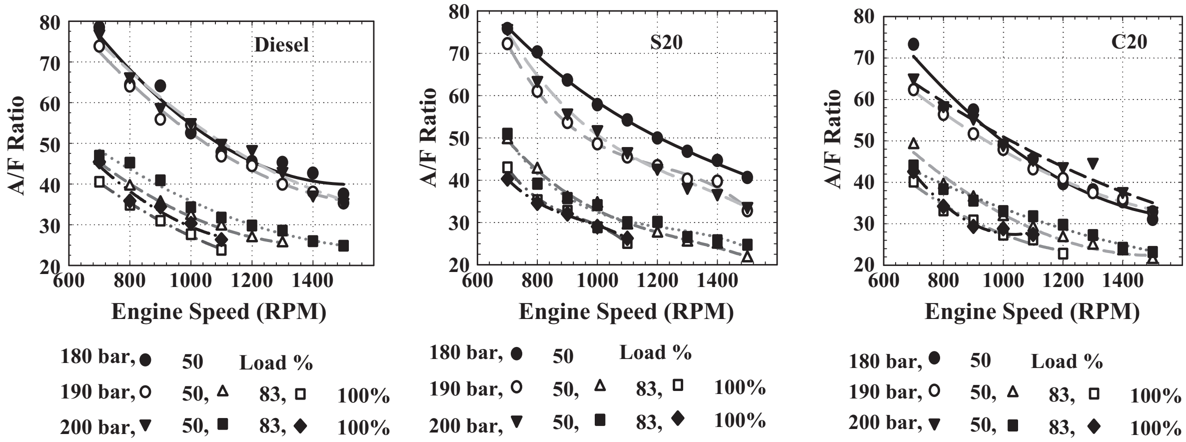

Fig. 3c. Air/fuel ratio with different fuels, loads and injection pressures.

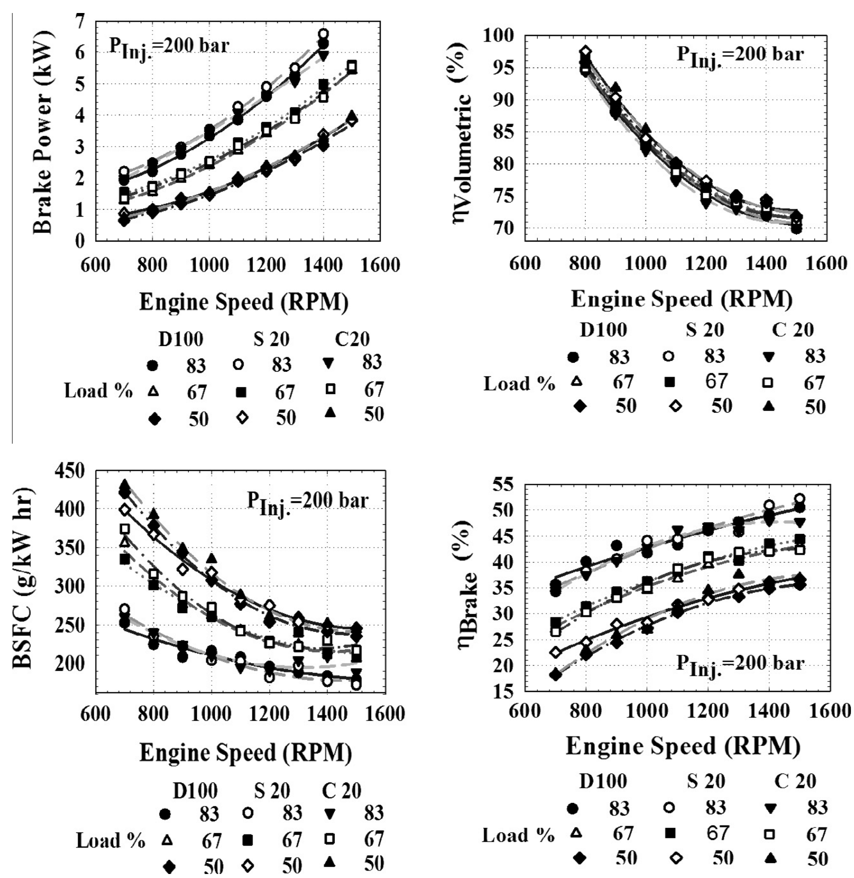

Fig. 4. Engine performance parameters versus engine speeds for different fuels and loads with ip of 200 bar.

arranged in descending order as diesel, S20 and C20 respectively. Similar corresponding results were observed by Agrawal et al. [18] while discussing the reasons of increasing the particulate size–number distribution with the increase of IP due to the increase of inject fuel and so the formation fuel-rich zones. However, as at high IP the fuel–air mixing is improved, the total particulate number was observed to be decreased with the increase of IP [18].

4.3. Engine performance parameters

The variation of engine performance parameters with the engine speed at different engine loads using different fuels at the maximum attainable IP (200 bar) are shown in Fig. 4. For all tested fuels, the engine Brake Power (BP) is increased with the increase of engine speed and engine load ratio due to the increase of the amount of injected fuel and so the energy released within engine cylinder is increased. According to the engine setting and the test conditions, there are no major differences between BP when fuel blend is changed. However, at specific engine load ratio and engine speed the main differences occur in engine fuel mass consumptions. The diesel engine volumetric efficiency depends mainly on the engine speed as well as the engine load ratio. The volumetric efficiency (η_V) is decreased with the increase of engine speed due to the increase of mass of air entering the engine cylinder. However, when the fuel blend is changed, η_V is slightly changed which may be due to the difference in the latent heat of vaporization resulting in different cooling effect of air/fuel mixture compared to that of neat diesel fuel. In this case, the overall change in the cylinder head thermal characteristics would slightly change the coefficient of fresh charge admission. It is also observed that, the Brake specific fuel consumption (BSFC) for all fuels is decreased with the increase of engine speed and engine load due to the continuous improve of in-cylinder combustion up to the engine rated conditions where combustion quality is the maximum. At high and medium engine loads (83% and 50%), for high speed the BSFC due to C20 is higher than that for S20 and neat diesel, but at low engine speed the BSFC due to S20 is higher than that for C20 and neat diesel fuel. This behavior can be owing to the combined effects of high viscosity, density and low heating value of C20 which leads to the largest fuel consumption in order to release the same energy as that for diesel fuel. This is consistent with the fact that BSFC is increased with the decrease of fuel heating value, this is why the BSFC for both S20 and C20 is observed to be higher than that for neat diesel fuel. While, C20 fuel has globally higher values of BSFC than for S20 due to its slightly lower heating value, its higher viscosity and density compared to the corresponding values of S20. From Fig. 4, it is observed that, the Brake thermal efficiency (η_B) is increased with the increase of engine speed and engine load as the combustion quality is improved and so the fuel burning becomes more economic. At high engine loads, the combustion chamber temperature is relatively high and this assists vaporization of fuel and improves η_B [24]. The BSFC and η_B have the inverse behavior; i.e. when BSFC is reduced that means the η_B is increased and vice versa. For high engine speed, no matter the engine load, the value of η_B for S20 and C20 blended fuels is higher than that for neat diesel fuel due to the better combustion characteristics of the oxygenated fuels over that of neat diesel fuel. The S20 gives higher values of η_B than that received when C20 is used. The difference between values of η_B for S20 and C20 at low engine speeds is lower than the difference at high engine speeds. Finally it is true to state that, the engine performance parameters due to use of blended fuels of S20 and C20 are generally better than those received when neat diesel fuel is used especially at high engine speeds, high engine loads and when high IP is used.

4.4. Exhaust and wall temperatures

Exhaust gas temperatures (T_{exh}) due to the use of all tested fuels are shown in Fig. 5. It is noticed that, the values of T_{exh} are increased with the increase of engine speed and load for all tested fuels due to the increase of the released energy. Increasing the engine load has a direct impact on increasing the mass of injected fuel so more heat is generated during the combustion process [18,25]. Moreover, at high engine speeds, the combustion is enhanced due to the increase of turbulence intensity with engine speed and so better fuel air mixing leading to complete combustion and so the T_{exh} is increased. Due to lower energy contents and better combustion quality of biodiesel burning, the values of T_{exh} for C20 and S20 are slightly lower than those for diesel fuel. As the combustion process is more completed when the oxygenated fuels are used, the wall temperature (T_{wall}) for S20 and C20 are higher than that is observed when diesel fuel is used (see Fig. 5). This action may also, be attributed to the high BSFC that results in increasing the heat lost to the cylinder wall.

4.5. Engine cylinder pressure analysis

The in-cylinder pressure (P_{cyl}) versus engine crank angle for different fuels at high and medium loads and at different IP are shown in Fig. 6a. As observed in this figure, the high viscosity, poor atomization and low heating values for C20 and S20 make P_{cyl} lower than that for diesel fuel. At high IP, the peak pressure within the engine cylinder is increased and its location is slightly retarded as the amount of the injected fuel is increased. For all tested fuels, P_{cyl} increases with the increase of engine load and IP due to the increase of heat release rate, mass of injected fuel and the A/F ratio goes to stoichiometric condition. From the measured indicator diagrams at different engine conditions, the values of the maximum pressure and the corresponding crank angle are recorded and collected as that shown in Fig. 6b. The effect of engine load on the maximum cylinder pressure (P_{max}) at 1200 RPM using the different tested fuels at different IP is shown in Fig. 6b. For different fuels, the values of P_{max} are increased with the increase of engine load and IP due to the increase of injected fuel mass that results in increasing the amount of heat released. For diesel engine, the P_{max} depends mainly on the part of energy released via premixed combustion mode, which is governed by the delay period [26]. For the same engine conditions at IP of 190 bar, the values of P_{max} for C20 are higher than those for neat diesel and S20 blended fuel due to its higher CN and so shorter delay period. For IP of 200 bar, the neat diesel fuel exhibits higher values of P_{max} than those obtained for C20 and S20 due to its lower viscosity, better mixing, and higher heating value. The crank angle positions at which occurring the P_{max} are plotted versus engine loads for all tested fuels with different IP as shown in Fig. 6c. The location of P_{max} for C20 at different engine loads are more retarded than those for neat diesel and S20 fuels due to its higher viscosity, greater density and larger

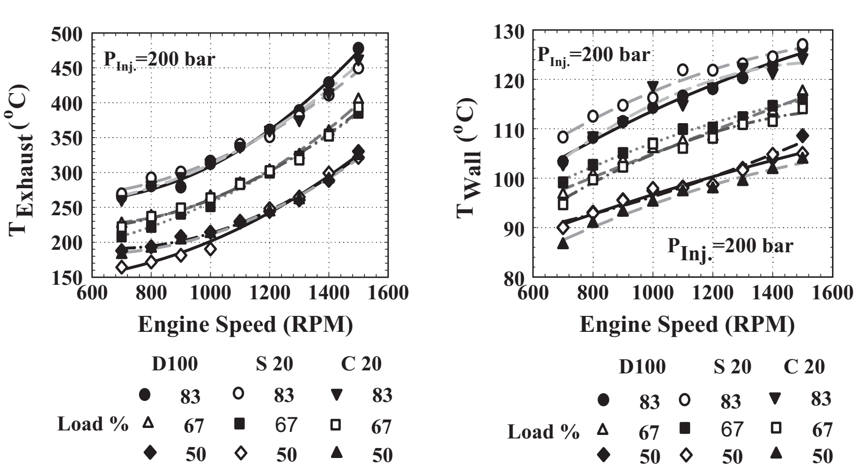

Fig. 5. Exhaust gas temperature and wall temperature versus engine speed for different load and fuels at injection pressure of 200 bar.

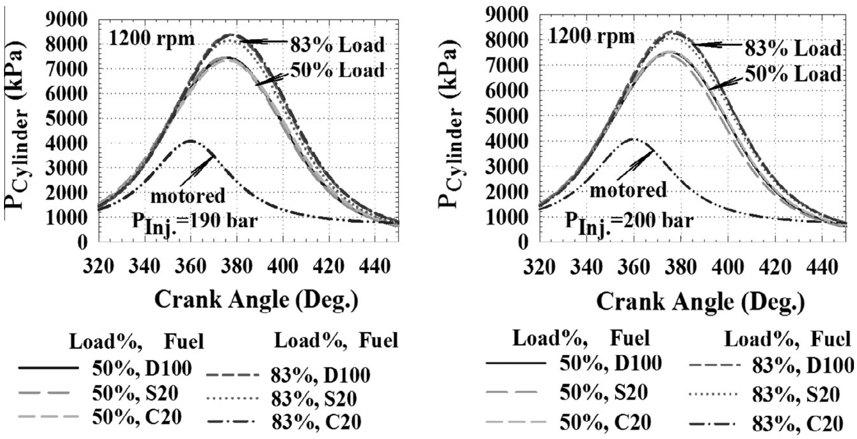

Fig. 6a. Cylinder pressure at 1200 RPM for different fuels, injection pressures and loads.

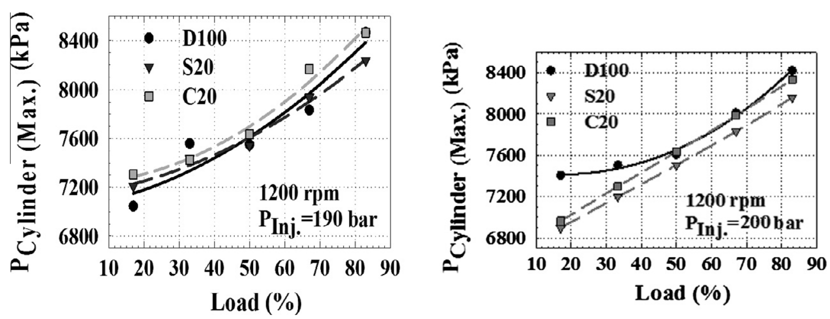

Fig. 6b. Maximum cylinder pressure at 1200 RPM for different fuels, injection pressures and loads.

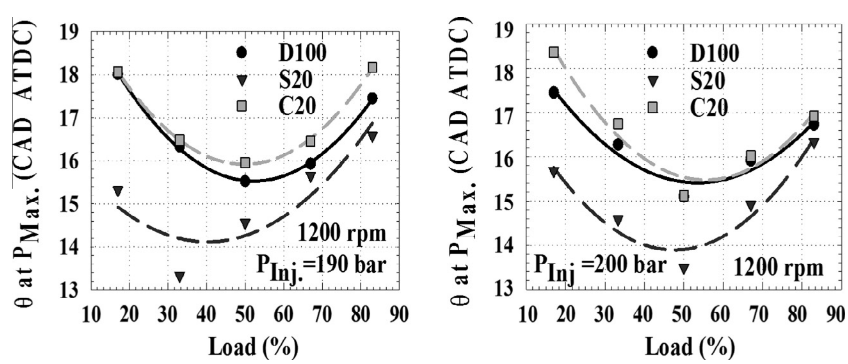

Fig. 6c. Position of maximum cylinder pressure at 1200 RPM for different fuels, injection pressures and loads.

injected mass. The position of P_{max} is arranged in descending order as S20, neat diesel and C20 respectively. For three fuels, position of P_{max} as C.A. deg. decreases with the increase of engine load up to 50% after that the trend is reversed. The inflection point of P_{max} position is attained within 14–18 C.A. deg. ATDC for all fuels no matter engine load and/or IP; with difference of 1–2 C.A. deg. for different tested fuels. Furthermore as IP is increased from 190 to 200 bar, the position of P_{max} as C.A. deg. ATDC for different fuels is advanced in case of 200 bar in comparison with that at IP of 190 bar, especially at high engine loads. The position of P_{max} as C. A. deg. ATDC has the arrangement in descending order as C20, neat diesel and S20 respectively.

The previous results are found to be in good agreement with the corresponding one cited in the literature, for example, Jaichandar and Annamalai [17] concluded that, the increase of IP leads to the decrease of the delay period and the increase of both the in-cylinder peak pressure and the maximum rate of heat release. Ryu [27] studied the influence of using pilot injection of biodiesel fuel on the combustion stability of a single cylinder diesel engine operated with dual fuel (compressed natural gas – CNG). It was concluded that, the combustion stability of biodiesel–CNG dual fuel is improved with the increase of pilot injection pressure. While investigating the influence of injection pressure on the performance of diesel engine with common-rail system, Liu et al. [21]

concluded that with the increase of the injection pressure, the combustion duration is shortened, the in-cylinder peak pressure and the peak value of heat release rate are increased, and the crank angle where achieved 50% of the total heat release becomes close to the TDC.

5. Conclusions

A successful stable operation of diesel engine fueled by blended fuels of C20 and S20 biodiesels with diesel fuel over a wide range of engine speeds, loads and IP without any hardware engine modification. The study has been performed as a function of engine brake thermal efficiency, brake specific fuel consumption, wall and exhaust gas temperature, and in-cylinder pressure. The following results have been concluded:

1. To fully compensate the effect of high viscosity of S20 and C20 as compared with that of neat diesel fuel at normal temperature, biodiesel blends should be preheated up to 60–80 °C.
2. The brake thermal efficiency (η_B) for diesel, S20 and C20 fuels are arrangement inversely according to their viscosity/density and oxygen content while their arrangement in descending ordering regarding the heating values.
3. For S20 and C20 optimum IP is 200 bar where the highest η_B and the lowest BSFC are obtained, where these performance parameters are improved by about 15% (from those at original IP of 180 bar) over that of diesel fuel.
4. With IP of 200 bar, the in-cylinder peak pressure (P_{max}) for diesel fuel is higher than that for C20 and S20 blended fuels no matter the engine conditions.
5. The position at which the maximum in-cylinder pressure (P_{max}) is attained within 14–18 °C.A. ATDC for all fuels at different IP, speeds and loads. The difference in the position of P_{max} for all fuels are in range of 1–2 °C.A.

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