CHAPTER 2

THEORY AND LITERATURE REVIEWS

2.1 Vegetable Oil

Vegetable oil is also one type of hydrocarbons and the structure of its main components; hydrogen and carbon are shown in Figure2.1. The energy content in the vegetable oil is about 80 to 90% of that in the diesel oil. Table 2.1 shows the physical properties and the calorific value of various vegetable oils.

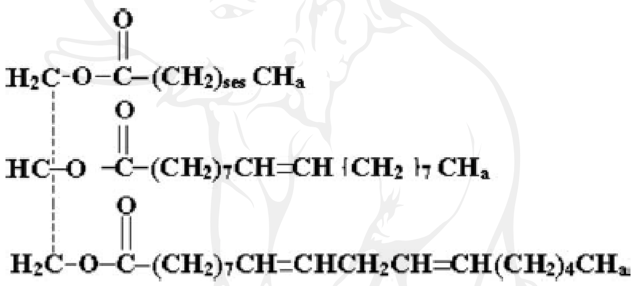

Figure 2.1 Chemical structure of vegetable oil [2]

Table 2.1 Physical and thermal properties of vegetable oils [2].

Oils	Specific Gravity at 21°C	Viscosity at 21°C (cSt)	Heating Value (kJ/kg)
Soybean	0.918	57.2	39,350
Sunflower	0.918	60.0	39,490
Coconut	0.915	51.9	37,540
Peanut	0.914	67.1	39,470
Palm	0.898	88.6	39,550
Palm seed	0.904	66.3	39,720
Jatropha seed	0.915	36.9 @ 38 °C	39,000
Diesel oil	0.845	3.8	46,800

Table 2.2 Chemical composition of fatty acids in general [3].

Name of fatty acid	Chemical name of fatty acids	Structure (*xx*:*y*)	Formula
Lauric	Dodecanoic	12:0	$C_{12}H_{24}O_2$
Myristic	Tetradecanoic	14:0	$C_{14}H_{28}O_2$
Palmitic	Hexadecanoic	16:0	$C_{16}H_{32}O_2$
Stearic	Octadecanoic	18:0	$C_{18}H_{36}O_2$
Arachidic	Eicosanoic	20:0	$C_{20}H_{40}O_2$
Behenic	Docosanoic	22:0	$C_{22}H_{44}O_2$
Lignoceric	Tetracosanoic	24:0	$C_{24}H_{48}O_2$
Oleic	*cis*-9-Octadecenoic	18:1	$C_{18}H_{34}O_2$
Linoleic	*cis*-9,*cis*-12-Octadecadienoic	18:2	$C_{18}H_{32}O_2$
Linolenic	*cis*-9,*cis*-12,*cis*-15-Octadecatrienoic	18:3	$C_{18}H_{30}O_2$
Erucle	*cis*-13-Docosenoic	22:1	$C_{32}H_{42}O_2$

xx indicates number of carbons, and *y* number of double bonds in the fatty acid chain.

Table 2.3 Physical and thermal properties of various vegetable oils [3].

Vegetable oil	Kinematic viscosity at 38 °C (mm^2/s)	Cetane no. (°C)	Heating value (MJ/kg)	Cloud point (°C)	Pour point (°C)	Flash point (°C)	Density (kg/l)
Corn	34.9	37.6	39.5	−1.1	−40.0	277	0.9095
Cottonseed	33.5	41.8	39.5	1.7	−15.0	234	0.9148
Crambe	53.6	44.6	40.5	10.0	−12.2	274	0.9048
Linseed	27.2	34.6	39.3	1.7	−15.0	241	0.9236
Peanut	39.6	41.8	39.8	12.8	−6.7	271	0.9026
Rapeseed	37.0	37.6	39.7	−3.9	−31.7	246	0.9115
Safflower	31.3	41.3	39.5	18.3	−6.7	260	0.9144
Sesame	35.5	40.2	39.3	−3.9	−9.4	260	0.9133
Soya bean	32.6	37.9	39.6	−3.9	−12.2	254	0.9138
Sunflower	33.9	37.1	39.6	7.2	−15.0	274	0.9161
Palm	39.6	42.0	–	31.0	–	267	0.9180
Babassu	30.3	38.0	–	20.0	–	150	0.9460
Diesel	3.06	50	43.8	–	−16	76	0.855

Tables 2.2 and 2.3 show examples of various vegetable oil properties compared with that of diesel. Vegetable oil contains various fatty acids and it is heavier than diesel fuel. It could be noted that high ratio of carbon to hydrogen content results in high viscosity then the vegetable oil viscosity is higher than that of diesel and some wax might be formed. Moreover, high flash point and fire point are

also found. Therefore, injection of vegetable oil into a combustion chamber, the droplet size control is more difficult and incomplete combustion always occurs.

2.2 Emulsion and Emulsifier

Emulsion is a type of colloid caused by a mixing of two immiscible liquids by adding an additive called surfactant. A surfactant can reduce the oil and water surface tension, activate their surface, and maximize their superficial contact areas to make oil in water or water in oil two phase emulsions. A surfactant is also called an emulsifier because it can stabilize emulsions when it exits along the interface between water and oil. An ion surfactant can increase the electric charge in the disphere phase droplets and increase the droplets from merging. If the surfactant forms an absorbing layer outside the disperse phase it can also prevent droplet merging [4]. Table 2.4 shows properties of an emulsifier.

Table 2.4 Properties of Span 80 [5].

Code name	Span 80
Synonyms	Sorbitan monooleate, (Z)-Sorbitan mono-9-octadecenoate
Formula Hill	$C_{24}H_{44}O_6$
Molar mass	428.68 g/mol
Density	1.0 g/cm^3 (20 °C)
Boiling point	> 100 °C
Flash point	> 149 °C
Vapor pressure	< 1.4 kPa (20 °C)
pH	5-7
Saponification value	149 - 160

Emulsified oil droplets spraying into a combustion chamber and as a result of the violent transformation of the water content into steam, it shatters the petroleum surrounding the water into much smaller droplets. It can reduce some types of exhaust gases. Combustion phenomenon of emulsified oil is called micro-explosion. When this type of emulsion is sprayed on a hot combustion chamber, heat convective on the surface of the fuel droplet. As water and diesel have different boiling temperatures, the evaporation rates of these two liquids will be different. As a result, the water molecule will reach its superheated stage faster than diesel creating vapor expansion breakup [6] as shown in Figure 2.2.

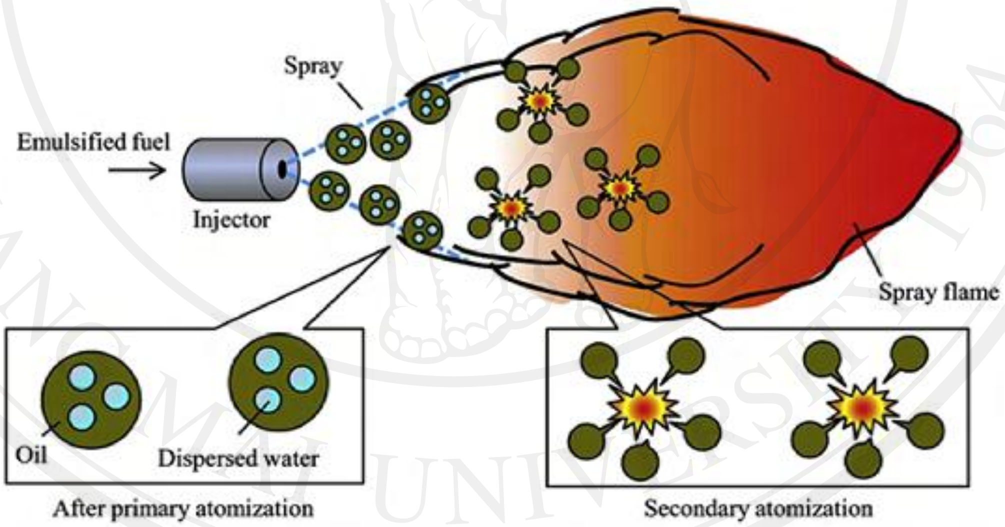

Figure 2.2 Emulsified Oil Combustion [6].

Smaller droplets have a much greater surface area, significantly improving the efficiency of combustion. A secondary effect of water transforming into steam is that peak combustion temperature could be reduced, resulting in the formation of significantly fewer smog-forming NOx emissions. The changes in combustion kinetics also significantly reduce PM emissions that result from incomplete combustion [6].

2.3 Combustion Reaction

Fuel could react with O_2 in an appropriate ratio for combustion and there is heat release with combustion products such as carbon dioxide (CO_2) and water (H_2O). The combustion equation could be written as

$$C_uH_vO_wN_xS_y + \left(u + \frac{v}{4} - \frac{w}{2} + y\right)(O_2 + 3.76N_2) \rightarrow$$

$$uCO_2 + \frac{v}{2}H_2O + ySO_2 + \left(3.76(u + \frac{v}{4} - \frac{w}{2} + y) + \frac{x}{2}\right)N_2$$

In general, O_2 from the air is used in combustion. Generally, the compositions of O_2 and N_2 in the air are 23.2% and 76.8% by mass, respectively. For complete combustion (Stoichiometric combustion), the air-fuel ratio (A/F) could be calculated from

$$(A/F)_s = \frac{4.3211 \times 15.999\left[2 + (0.5)\left(H/C\right) - \left(O/C\right) + 2\left(S/C\right) + \left(N/C\right)\right]}{\left[12.011 + 1.0079\left(H/C\right) + 15.999\left(O/C\right) + 32.066\left(S/C\right) + 14.007\left(N/C\right)\right]} \qquad (2.1)$$

where $\left(H/C\right)$ = atomic hydrogen to carbon ratio,

$\left(O/C\right)$ = atomic oxygen to carbon ratio,

$\left(N/C\right)$ = atomic nitrogen to carbon ratio,

$\left(S/C\right)$ = atomic sulfur to carbon ratio.

2.4 Fuel Droplet Combustion

The combustion of fuel droplets in diesel engines is very sophisticated and depends on several mechanisms. Models of droplet combustion have been developed by many researchers. Godsave et al considered combustion phenomena of a burning fuel droplet in the atmosphere with enough O_2 as shown in Figure 2.3. [1]

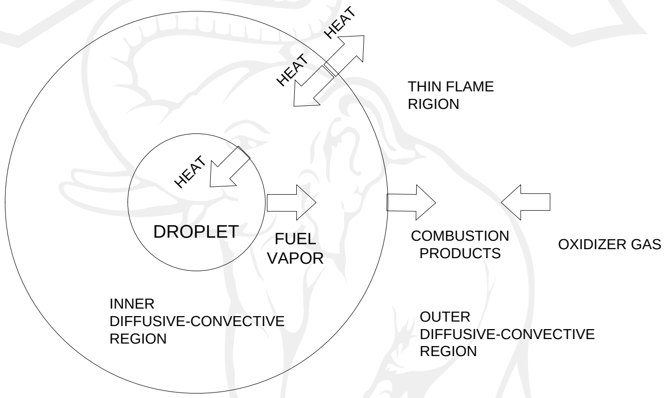

Figure 2.3 The pattern of fuel droplets combustion. [1]

From Figure 2.3, as the fuel temperature drops below the ambient temperature, heat from the air transfers into the fuel droplets. The fuel temperature is up until around the fuel droplets are filled with fuel vapor that evaporates from the fuel droplets and the droplets will be smaller. When combustion occurs, heat of reaction spreads into the surrounding ambient air and the fuel droplets. Spalding [7] proposed a model for evaluating mass transfer as

$$m'' = g \ln(1+B) \tag{2.2}$$

where m” = mass transfer flux (kg/m^{2}s), B = mass transfer driving force,

g = mass transfer conductance (kg/m^2s).

For steady state combustion, Spalding transfer number can be obtained from

$$B = \frac{C_p(T_G - T_{BP}) + m_{ox,G}\frac{H}{r}}{L + C_{p_{fu}}(T_{BP} - T_{fu})} \tag{2.3}$$

where C_p = specific heat of air (kJ/kg K),

T_G = the ambient temperature (°C),

T_{bp} = boiling poing temperature (°C),

H = heating value (kJ/kg),

$m_{ox,G}$ = mass ratio of O_2 in the ambient air = 0.232,

r = ratio of O_2 (Oxidant) to fuel for complete combustion,

L = latent heat of evaporation of fuel (kJ/kg),

Cp_{fu} = specific heat capacity of fuel (kJ/kg K),

T_{fu} = temperature of the fuel in the tank. (°C).

For fuel having a formula of $C_uH_vO_wN_xS_y$, the r value could be evaluated from

$$r = \frac{32\left(u + \frac{v}{4} - \frac{w}{2} + y\right)}{(12u + v + 16w + 14x + 32y)} \tag{2.4}$$

where r = ratio of oxygen to fuel,

u = mole number of carbon,

v = mole number of hydrogen,

y = mole number of sulfur,

w = mole number of oxygen,

x = mole number of nitrogen.

From heat and mass transfer analogy, in case of flow over a sphere, the mass transfer conductance could be calculated from

$$g = \left(\frac{\mu}{d \cdot Sc}\right)(2 + 0.6Re^{0.5} \cdot Sc^{0.33}) \quad (2.5)$$

where μ = viscosity at average temperature of surrounding air and droplet (kg/m s),

d = diameter of droplet oil (m),

Sc = Schmidt Number,

Re = Reynolds Number.

2.5 Fuel Properties

2.5.1 Specific Heat Capacity of Fuel (Cp_{fu})

Specific heat of vegetable oil could be given by [8]

$$Cp_b = 0.47 + 0.00073T \quad (2.6)$$

where T = temperature (°C),

Cp_b = specific heat capacity of vegetable oil (kcal/kg°C).

For emulsified oil which is a blend of diesel oil, vegetable oil and water the average specific heat capacity could be calculated by

$$Cp_m = (X_d \cdot Cp_d + X_b \cdot Cp_b + X \cdot Cp_w)/X_m \quad (2.7)$$

where X_d = mass fraction of diesel oil,

X_b = mass fraction of vegetable oil,

X_w = mass fraction of water,

X_m = total mass of emulsified oil.

2.5.2 Heating Value of Fuel (HV)

Heating value of liquid fuel (HV) is usually obtained by burning fuel in a bomb calorimeter. After combustion, if there is liquid water contained in the equipment, the heat obtained is called higher heating value (HHV) and if not, lower heating value (LHV) is the result. The difference between the two types of heating values is the amount of latent heat of water during the combustion reaction which is

$$LHV = HHV - 21.98H \quad (2.8)$$

where H= percent by weight of hydrogen in fuel.

2.5.3 Latent Heat of Fuel (L)

Latent heat of diesel oil blended with vegetable oil could be estimated from [1]

$$L = (110.9 - 0.09T_{bp})\frac{2.328}{spgr} \quad (2.9)$$

where L = latent heat of blended diesel oil and vegetable oil (kJ/kg),

spgr = specific gravity of blended oil at 15°C,

T_{bp} = boiling poing temperature (°C).

If the oil is blended with water then the total latent heat could be

$$L_{,m} = (X_b \cdot L_{,b} + X \cdot L_{,w})/X_m \quad (2.10)$$

where

X_b = mass fraction of diesel oil blended with vegetable oil,

X_w = mass fraction of water,

X_m = total mass of emulsified oil.

2.5.4 Boiling Point of Liquid Fuels (T_{bp})

Boiling point of liquid fuel could be tested from Automated Distillation Tester Tanaka Model AD-6 following ASTM D86

2.5.5 Schmidt Number (Sc)

Schmidt number of fuel vapor in air could be estimated from [9]

$$Sc = 0.145M^{0.556} \tag{2.11}$$

where M = molecular weight of fuel.

For emulsified oil, the molecular weight could be

$$M = \Sigma_{allj}\, x_j M_j \tag{2.12}$$

where M is the molecular weight of the mixture, M_j is the molecular weight of chemical compound j and x_j indicates the mass fraction of each term of the chemical compounds in the mixture.

2.6 Application of Mass Transfer Theory in Engine Performance Prediction

Single droplet combustion could be performed experimentally by feeding fuel oil to cover a sphere. With combustion in an open space having air flow at various velocities, the mass transfer rate could be evaluated. Single droplet combustion mass transfer theory could be used to predict engine performance. Reynolds number is one main parameter to transform the data obtained from the single drop combustion model into the engine cylinder. Reynolds number of a single droplet combustion could be

$$R_e = \frac{\rho_a V_a d}{\mu_a} \tag{2.13}$$

where V_a = velocity of air flow over a sphere (m/s)

d = droplet diameter of fuel (m)

μ_a = viscosity of air at average temperature between combustion and air intake (N-s/m^2)

ρ_a = density of air (kg/m^3).

For combustion in an engine cylinder, a similarity will be undertaken by considering the same value of Reynolds number as that obtained in the experimental single droplet. Before combustion starts in the engine cylinder, the piston is close to the top dead center in the cylinder and the fuel droplets are injected into the cylinder. The relative velocity of oil droplets could be taken as the value for calculating the Reynolds number. The average speed of the oil droplets could be estimated [10] from

$$V_f = C_d \sqrt{2\frac{\Delta P}{\rho_f}} \tag{2.14}$$

where C_d = Discharge Coefficient of Nozzle = 0.6 ,

ΔP = pressure difference in nozzle injection and combustion chamber (Pa),

ρ_f = density of fuel (kg/m^3).

Bracco [9] described a mechanism of fuel spray injected from the nozzle in a diesel engine. The size of an injected fuel droplet could be

$$d = C\,\frac{2\pi\sigma}{\rho_g V^2}\,\lambda \tag{2.15}$$

where σ = surface tension of fuel (N/m),

ρ_g = density of air in combustion chamber (kg/m^3),

V = velocity of fuel injection in combustion chamber (m/s),

C = constant of the initial drop size equation is $\sqrt[3]{\frac{3}{\pi A_\theta}}$

where A_θ is constant of nozzle; $A_\theta \cong 0.01$,

λ = dimensionless wavelength of fastest growing surface wave in Taylor's theory of droplet formation which is a function of Ta. $Ta=\rho_l\sigma^2/\rho_g\mu_l^2V^2$.

The dimensionless wavelength of the fastest growing of the unstable surface perturbation waves could be estimated from the value of Ta, where $Ta=\rho_l\sigma^2/\rho_g\mu_l^2V^2$ in Figure 2.4. With the value of Ta, the terms ω^* and f^* could be estimated including the term λ since $f^* = \lambda.\omega^*$.

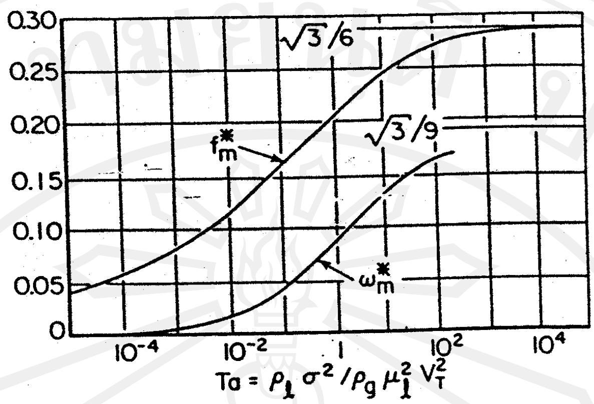

Figure 2.4The values of f* and ω* which are functions of Ta [9].

With the velocity and the droplet diameter, during combustion, the combustion heat rate from the single fuel droplet could be

$$\dot{Q}_{single} = \frac{H(\pi d \mu)}{Sc}(2 + 0.6Re^{0.5} \cdot Sc^{0.33}) \ln(1 + B). \qquad (2.16)$$

where H is heat of combustion or heating value of fuel.

If the combustion rate of the emulsified oil was obtained with the same droplet size, then the power ratio of the heat rate of the blended oil and that of the diesel oil generated by the engine could be set in a form of

$$\frac{P_b}{P_d} = \left(\frac{g_b}{g_d}\right)\left(\frac{ln(1+B)_b}{ln(1+B)_d}\right)\left(\frac{d_b^2}{d_d^2}\right)\left(\frac{H_b}{H_d}\right) \qquad (2.17)$$

Subscripts b and d refer to blended emulsified oil and diesel oil, respectively.

In practice, P could also be calculated by

$$P_{in} = \dot{m_f} H \qquad (2.18)$$

where $\dot{m_f}$ is fuel mass flow rate.

In a diesel engine, the output power or brake power is normally less than the total heat rate from the fuel. Wibulswas et al [11] commented a relation of $P_{bp} = kP_{in}/N^a$; k and a were constants, and N is engine speed. Since P_{bp} could be evaluated experimentally from a dynamometer then the values of k and a could be estimated by plotting the values of $log(P_{bp}/P_{in})$ versus $log\ N.$ k is the intercept on the axis of $log(P_{bp}/P_{in})$ and a is the slope of the fitted curve.

When substitute $P_{in} = P_{bp}N^a/k$ into eqn (2.18) then

$$\frac{P_{bp,B}}{P_{bp,D}} = \left(\frac{H_B}{H_D}\right)\left(\frac{Sc_D}{Sc_B}\right)\frac{\ln(1+B_B)}{\ln(1+B_D)}\ \frac{({N^a}/{k})_D}{({N^a}/{k})_B}. \qquad (2.19)$$

${P_{bp,B}}/{P_{bp,D}}$ is the ratio of the engine output power when the fuels are emulsified blended oil and diesel oil, respectively.

Eqn (2.19) is very useful. If the engine performance running with diesel oil is known, then the output power of the engine with emulsified blended oil could be evaluated directly.

2.7 Engine Performance

Engine performances are normally presented in terms of brake power, specific fuel consumption and fuel conversion efficiency.

2.7.1 Brake Power of engine (P_b)

Engine brake power is the useful power output from engine. It could be calculated from

$$P_{b=\frac{2\pi TN}{60}} \qquad (2.20)$$

where P_b = brake power of engine (W),

N = speed of engine (rpm),

T = torque of engine from dynamometer (N-m).

2.7.2 Brake Specific Fuel Consumption (bsfc)

Brake specific fuel consumption is the rate of fuel consumption per unit of engine brake power and it could be calculated from

$$bsfc = \frac{m'_f}{P_b} \tag{2.21}$$

where m_f' = fuel consumption (g/hr),

bsfc = brake specific fuel consumption (g/kW hr).

2.8 Literature Review

2.8.1 Mass Transfer

Wibulswas et al [8] initiated the theories of mass transfer to predict the rate of burning vegetable oil such as palm and coconut oil. The model is used by introducing a sphere of 2.5 and 5 cm in diameter covered with a thin film of vegetable oil, then heat up until the combustion occurred. The results showed that the calculated combustion rate agreed well with the experimental results.

Wibulswas et al [11, 12] also applied the theory of mass transfer to predict the small diesel engine performances running with blended diesel/vegetable oil. The vegetable oils were palm oil, palm oil methyl ester, sunflower oil, soybean oil and rice bran oil. It could be noted that the developed model could predict the output power ratio of the engine with the blended oil to that of the diesel oil very well compared with the experimental data. The maximum error was less than 15 %.

Samec et al [13] studied the numerical and experimental were made on some of the chemical and physical properties of water/oil emulsified fuel (W/OEF) combustion characteristics. The injection and fuel spray characteristics are analyzed numerically also in order to study indirectly the physical effects of water present in diesel during the combustion process. It has been evaluated by analyzing in cylinder pressure, and the rate of heat release time histories. In relation to the net diesel fuel combustion, the ignition delay became longer by about 10 % and the gradient of heat release rate during premixed burning increased up to 26 % when used W/OEF as fuel. The experimental tested of water/oil emulsified fuel combustion in the DI diesel engine under several loads and speeds, using 10% and 15%. The results of engine tested were reduced NO_x by nearly 20% and concentration of soot by up 50% with the regular diesel oil.

Kadota et al [14] studied on the fundamental mechanism relevant to the micro-explosion phenomena leading to the secondary atomization which is not common to the combustion of pure fuel. The kinetic model and the probability model for predicting the nucleation of vapor bubbles in the liquid phase, and the measured and predicted results of the superheat limit of hydrocarbons and water beyond which the liquid phase cannot exist. The evaporation and the combustion of emulsion droplet are the phenomenological burning processes, the burning rate constant, the ignition process, the flame phenomena including soot concentration profile in the droplet flame and the spherical evaporation on a hot surface. Also mentioned are the in-droplet transfer processes including the phase separation, the micro-explosion phenomena and the conditions for the micro-explosion to occur and the empirical equation for the rate of micro-explosion based on the probability model.

Broniarz-Press et al [15] studied on atomization of the emulsions flowing through twin-fluid atomizers obtained by the use of the digital microphotography method. The photographs were taken by a digital camera with automatic flash at exposure time of 1/8000 s and subsequently analyzed using Image Pro-Plus. The oils used were mineral oils 20–90, 20–70, 20–50 and 20–30. The studies were performed at flow rates of liquid phase changed from 0.0014 to 0.011 (dm3/s) and gas phase changed from 0.28 to 1.4 (dm3/s), respectively. The experimental results showed that the changes in physical properties of a liquid phase lead to the significant changes in the spray characteristics. The analysis of the photos of water and emulsions atomization process showed that droplets are bigger in air-emulsion system (at the same value of gas to liquid mass ratio). The values of Sauter mean diameter (SMD) increased with increase of volume fraction of oil in emulsion. The droplet size increased with emulsion viscosity.

2.8.2 Blended Vegetable Oil/Diesel Oil/Water

Pramanik [16] tested the use of Jatropha oil mixed with diesel fuel in a single cylinder open combustion chamber. The results showed that the mixed fuel could be used as a fuel to replace diesel oil.

Zubr et al [17] tested oil from Camelina seeds (Camelina sativa) with a ratio of 5, 10% mixed with diesel oil in a 5.1 kW fuel engine (Fairy Man 18D 430 Type 1001 Model). The test engine was running at 4.0 kW 3,260 rpm for 250 hr to see the conditions inside the cylinder parts.

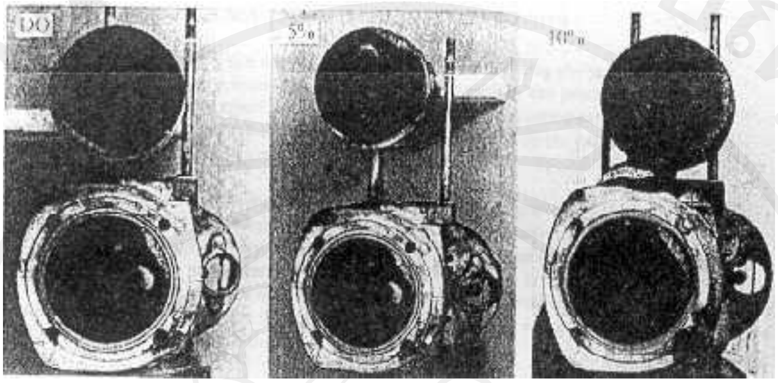

Figure 2.5 Cylinder conditions after 250 hr operation[14].

Figure 2.5 shows pictures of the soot formation on the piston and cylinder of the tested engine. It could be noted that the cylinder is still clean compared with diesel oil.

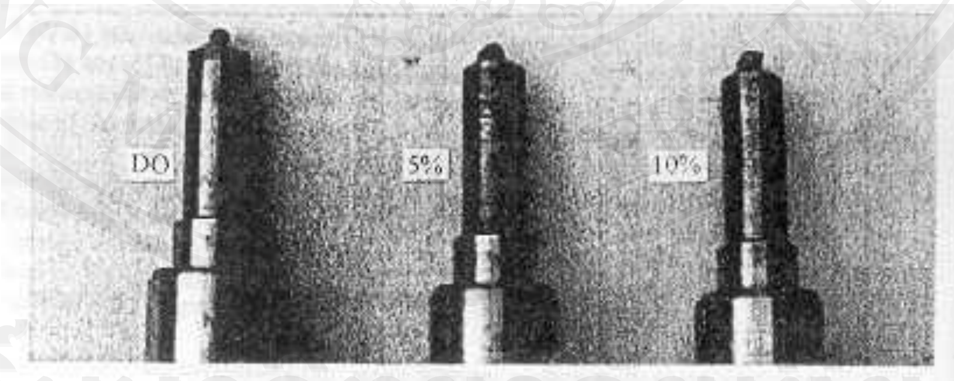

Figure 2.6 The condition of the fuel injectors after 250 hr operation.

Figure 2.6 shows the condition of the engine nozzle after 250 hr operation. The formation of soot, with the blended oil, was found to be slightly higher than that of the diesel oil. The measured rate of specific fuel consumption values were 271.6, 273.4 and 277.1 g / kW.hr when using diesel fuel, and the blended oil with Camelina oil of 5% and 10% respectively.

Abu-Zaid [18] used diesel fuel mixed with water at the ratios of 0, 5, 10, 15 and 20% by volume. Span 80 and Tween 80 2% were used as a surfactant to determine the performance and emissions from a 659 cc DI single cylinder engine (PETTER PHIW). The tested engine speed was in a range of 1200 - 3300 rpm. The results showed that the use of diesel fuel mixed with water increased engine power and thermal efficiency is higher than diesel. For the exhaust temperature of engine at various engine speeds, increase of the percentage of water blended in the fuel, the value tended to be less.

Kalam [19] used CPO oil mixed with 5 and 10% water to run a diesel engine in a short period (20 hours). It was found that the values of NO_X and CO_2 emissions were higher than those with petroleum diesel while the other emissions were less. In this study a DI single cylinder engine was tested its long-term performance with palm water emulsion having 1%, 2% and 3% water by volume (CPO99, CPO98 and CPO97) and compared with refined palm oil (CPO) and diesel (OD). The experiment was performed at an engine load of 5.5 kW at 2700 rpm and fixed for 100 hours to study the sediment left behind the piston and cylinder. The results showed that the oil viscosity, the heating value and the cetane number of the emulsified oil decreased with the composition of water in the oil. The pour point and the fired point of the emulsified oil increased which resulted in longer ignition delay.

The CO emissions from the engine with the emulsified oils were slightly higher than that with diesel. However, the use of CPO alone gave lower result. Vice versa for the results of NO_x. For the PM measurements, the CPO gave slightly lower value than that of OD but the emulsified oils showed the higher results.

Kerihuel [20] studied emulsified oil from duck oil blended with water and alcohol with various compositions. The surfactants were Sorbitan Sesquiolate (Span 83) and Sobitan Monooleate (Span 80). The animal oils, namely AF1 $(C_7H_{16}0)_n$ and AF2 $(C_4H_80)_n$, lower heating values (LHV) were 38,300 kJ/kg and 28,200 kJ/kg compared with 42,500 kJ/kg of diesel oil. The appropriate composition of the blended oils were 15% of water, 15% of methanol and 2 % of surfactant.

Armas [21] tested oil-water emulsion in an IDI diesel engine (Renault F8Q with turbocharger and intercooler). It could be found that the mixture viscosity affected the fuel injection and the heating value of fuel affected the engine performance. The emissions of NO, HC, PM and soot from engine with the blended oil were lower than those with the diesel.

Lin et al [22] tested two phase emulsions (W/O) and three phase emulsions (O/W/O) in a marine diesel engine (UMBDI) model of Isuzu with in-line four cylinders, four stoke, direct injection. It could be found that the heating values for W/O (10 % W) and O/W/O (10 % W) were 40.85 MJ/kg and 40.55 MJ/kg, respectively compared with 45.80 MJ/kg of regular diesel oil. The amount of emulsifying agent added into the O/W/O three and W/O two phase emulsions was 2 % of the oil and water mixture volume. The engine torque and the brake specific fuel consumption (bsfc) when the engine speed was fixed at 1800 rpm which were higher than those with the diesel. The emissions of NOx, CO and smoke opacity from engine with the W/O and O/W/O were lower than those with the regular diesel oil.

Lin et al [23] studied emulsified oil from soybean biodiesel oil blended with water. Aqueous ammonia of 5% by wt. was mixed with the surfactant Tween 80 and distilled water to prepare the O/W emulsion. The preparation procedure in the second stage was the same as that for preparing the O/W/O biodiesel emulsion. The engine performance and emission characteristics of the biodiesel and three-phase O/W/O biodiesel emulsions were tested by a diesel engine in combination with an eddy-current dynamometer. The engine was an UMBDI model of Isuzu with in-line four cylinders, four stoke, direct injection with a displacement volume of 3856 cm^3. It could be found that the O/W/O emulsion had the lowest carbon dioxide (CO_2) emissions, exhaust gas temperature, heating value and the largest brake specific fuel consumption, fuel consumption rate. The increase of engine speed respected in higher, exhaust temperature and CO_2 emission but decrease of NO_x emissions.

Lin et al [24] used ultrasonic emulsification method was applied to prepare two-phase water-in-oil (W/O) and three-phase oil-in-water-in-oil (O/W/O) emulsions were tested the engine performance and the pollutant emission characteristics of a diesel engine. The emulsions produced lower NO emission, lower soot concentration and lower black smoke opacity, while creating a larger brake specific fuel consumption (bsfc) and a larger CO emission compared with that of an engine using neat diesel fuel. A comparison with the characteristics of the two-phase W/O emulsion, the three-phase O/W/O emulsion was found to have a larger CO emission, larger soot particles and larger bsfc while producing a lower brake thermal efficiency and a lower black smoke opacity.

Nadeem et al [25] studied emulsified formulations. Water/diesel (W/D) are reported to reduce the emissions of NOx, SOx, CO and particulate matter (PM).

Diesel engine performance and exhaust emission was also measured and analyzed with these indigenously prepared emulsified fuels. A comparative study involving torque, engine brake mean effective pressure (BMEP), specific fuel consumption (SFC), particulate matter (PM), NOx and CO emissions is also reported for neat diesel and emulsified formulations. It was found that the biggest reduction in PM, NOx, CO and SOx emission was achieved by the emulsion stabilized by gemini surfactant containing 15% water contents.

Maiboom et al [26] tested water-in-diesel emulsion (WDE) in a modern automotive 1.5 l HSDI Diesel. Four injection strategies are considered with and without pilot injection, with two levels of injection pressure. First, the injection of WDE is compared to diesel-fuel in terms of combustion and NOx and PM emissions without using exhaust gas recirculation (EGR). Depending on the WDE fuelling rate and injection strategy (with or without a pilot injection before main injection), NOx emissions are most often reduced (of up to 50%), and PM emission are most often decreased as well (the maximum relative reduction being 94%).

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