THEORETICAL AND EXPERIMENTAL ANALYSIS ON HEAT RELEASE RATE AND IGNITION DELAY OF COMPRESSION IGNITION ENGINE FUELED WITH EMULSIFIED AND NANO ADDITIVE WCO

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ABSTRACT

Mathematical modeling is a useful tool for any mechanical system design and has the potential to offer a better understanding of the processes involved in the system under analysis. The mathematical model serves as a tool for future predictions of the systems behavior and the stability of the system as well. In the contest of engine research a suitable computer-based modeling can provide adequate means for describing details of the combustion process in engines and can help in reducing the time consuming experiments and the cost for such experiments. The modeling tool could be used for optimizing the engine design for maximizing the performance. In this work, Thermodynamic model was used for analyzing the performance characteristics of four stroke direct injection compression ignition engines. Result shows, the predicted theoretical model of heat release rate (HRR) and Ignition delay (ID) very closed to the experimental values.

Key Words: Modeling, ID, HRR

INTRODUCTION

ENGINE MODELING

The theoretical models used in the case of internal combustion engines can be classified into two main groups viz., thermodynamic models and fluid dynamic models. Thermodynamic models are mainly based on the first law of thermodynamics and are used to analyze the performance characteristics of engines. Pressure, temperature and other required properties are evaluated with respect to crank angle or in other words with respect to time. The engine friction and heat transfer are taken into account using empirical equations obtained from experiments. These models are further classified into two groups namely single-zone models and multi-zone models. On the other hand, multi-zone models are also called computational fluid dynamics models. These are also applied for the simulation of combustion process in the internal combustion engines. They are based on the numerical calculation of mass, momentum, energy and species conservation equations in either one, two or three dimensions to follow the propagation of flame or combustion front within the engine combustion chamber.

Mathematical modeling is a useful tool for any mechanical system design and has the potential to offer a better understanding of the processes involved in the system under analysis. The mathematical model serves as a tool for future predictions of the systems behavior and the stability of the system as well. In the contest of engine research a suitable computer-based modeling can provide adequate means for describing details of the combustion process in engines and can help in reducing the time consuming experiments and the cost for such experiments. The modeling tool could be used for optimizing the engine design for maximizing the performance. Data driven modeling approach is another simple method in predicting the mechanical systems behavior by solving complex non-linear mathematical equations. Data-driven model creates a relationship between the given inputs and outputs, trains and solves the same.

Thermodynamic model is used for analyzing the performance characteristics of four stroke direct injection compression ignition engines. The following fundamental assumptions have been made (Ramadhas et al, 2006). :

I. Cylinder charge is a homogeneous gas mixture of fuel vapor and air.

II. Pressure and temperature inside the cylinder are uniform and vary with crank angle.

III. Specific heats of the gaseous mixture are calculated as a function of temperature.

IV.As the combustion should always be of lean mixture, this leads to temperatures at which dissociation does not have much effect on thermodynamic performance of the engine. Thus the assumptions in dissociation with single zone model are generally acceptable for diesel engine simulations

MATERIAL AND METHODS

THEORETICAL THERMODYNAMIC MODEL (TTM)

The TTM considered the first law of thermodynamics. The processes occurring in the engine such as intake, compression, combustion expansion and exhaust were considered. Sub models were developed for each process. The step by step procedure is presented below.

B – Bore -87.5 mm

S – Stroke- 110 mm

V_{disp} – Displacement volume-

V_C – Clearance volume

V_S – Stroke volume

r – Ratio of compression

V_{bdc} – Volume at bottom dead center

θ – Crank angle

L – Length of the connecting rod

P'max – Previous value of cylinder pressure

dp – Change in pressure

dv- Change in volume

Γ – Ratio of specific heat

S'- previous value of scale factor

θ'- previous value of crank cycle

U- Sensible internal energy of the cylinder contents

mf - Sensible enthalpy of the injected fuel

Apparent Gross heat release rate

The heat transfer rate to the walls

Apparent net heat release rate

S - is the distance between the crank axis and the piston pin axis

a - crank radius

θ - crank angle

l - connecting rod length

V- cylinder volume at any crank position

θ' – previous value of crank angle

dθ- change in crank angle

V'- previous value of cylinder volume

Γ – Ratio of specific heat

$\Gamma = C_P / C_V$

P is the pressure at the end of compression stroke at TDC (bars)

T is the temperature of the charge at TDC (Kelvin)

Sp is the mean piston speed of the engine (meter per second)

STEP: 1

$$V_{disp} = \frac{\pi}{4} B^2 S$$

Where

B – Bore

S – Stroke

V_{disp} -Distance volume

STEP: 2

$$V_{bdc} = \left(\frac{\gamma}{\gamma-1}\right) V_{disp} = Vs + V_c$$

Where,

V_{bdc} – Volume at bottom dead center

r – Ratio of compression

STEP: 3 $V_{tdc} = \left(\frac{\gamma}{\gamma-1}\right) V_{disp} = V_c$

Where

V_{tdc} - Volume at top dead center

STEP: 4

$$V(\theta) = V_{disp} \left[\frac{r}{r-1} - \frac{1-\cos\theta}{2} + \frac{L}{S} - \frac{1}{2}\sqrt{\left(\frac{2L}{S}\right)^2 - \sin^2\theta}\right]$$

Where

θ- Crank angle

L-length of the connecting rod

L- 2S

STEP: 5

$$V'(\theta) = \frac{dv}{d\theta} = \frac{V_{disp}}{2}\left(\frac{1}{2}\frac{\sin 2\theta}{\sqrt{\left(\frac{2L}{S}\right)^2 - \sin^2\theta}} - \sin\theta\right)$$

Where,

$\frac{dv}{d\theta}$ - Rate of volume

STEP: 6

$$\frac{dQ_n}{d\theta} - p\frac{dv}{d\theta} + m_f * h_f = \frac{dU_s}{d\theta} - - - - - - - - (1)$$

Here $h_{s,f} \approx 0$

$$\frac{dQ_n}{d\theta} - p\frac{dv}{d\theta} + 0 = \frac{dU_s}{d\theta}$$

$$\frac{dQ_n}{d\theta} = p\frac{dv}{d\theta} + \frac{dU_s}{d\theta}$$

Where

U- Sensible internal energy of the cylinder contents

m_f- Sensible enthalpy of the injected fuel

$$\frac{dQ_n}{d\theta} = p\frac{dv}{d\theta} + mC_v\frac{dT}{d\theta} - - - - - - - - - (2)$$

= The rate at which the work done on the piston + Rate of change of sensible internal enthalpy of the cylinder contents

But,

$$\frac{dQ_n}{d\theta} = \frac{dQ_{ch}}{d\theta} - \frac{dQ_{ht}}{d\theta} - - - - - - - - - - - - (3)$$

Where

$\frac{dQ_{ch}}{d\theta} \rightarrow$ Apparent Gross heat release rate

$\frac{dQ_{ht}}{d\theta} \rightarrow$The heat transfer rate to the walls

$\frac{dQ_n}{d\theta} \rightarrow$Apparent net heat release rate

Equation (2) and (3) are equal

$$\frac{dQ_n}{d\theta} = \frac{dQ_{ch}}{d\theta} - \frac{dQ_{ht}}{d\theta} = p\frac{dv}{d\theta} + mC_v\frac{dT}{d\theta} - - - - - - - - - - (4)$$

According to ideal gas law,

pv= mRT

Taking log to both sides,

log p + log v = log m + log R + log T

Taking differential to both sides,

$$\frac{dp}{p} + \frac{dv}{v} = 0 + 0 + \frac{dT}{T}$$

$$\frac{dp}{p} + \frac{dv}{v} = \frac{dT}{T} - - - - - - - - - - - (5)$$

$$\frac{v.dp + p.dv}{pv} = \frac{dT}{T}$$

$$v.dp + p.dv = pv.\frac{dT}{T}$$

$$= mRT.\frac{dT}{T}$$

$$v.dP + p.dv = mR.dT$$

Taking differential with respect to ‘θ’

$$v.\frac{dp}{d\theta} + p.\frac{dv}{d\theta} = mR.\frac{dT}{d\theta}$$

$$\frac{dT}{d\theta} = \frac{v.\frac{dp}{d\theta} + p.\frac{dv}{d\theta}}{mR} - - - - - - - - (6)$$

Substituting equation (6) in equation (2)

$$\frac{dQ_n}{d\theta} = p.\frac{dv}{d\theta} + \frac{mC_v}{mR}\left(p.\frac{dv}{d\theta} + v.\frac{dp}{d\theta}\right)$$

$$= p.\frac{dv}{d\theta} + \frac{C_v}{R}p.\frac{dv}{d\theta} + \frac{C_v}{R}.v\frac{dp}{d\theta}$$

$$\frac{dQ_n}{d\theta} = (1 + \frac{C_v}{R})p.\frac{dv}{d\theta} + \frac{C_v}{R}.v\frac{dp}{d\theta}$$

$$R = \gamma C_v - C_v$$

$$R = C_v(\gamma - 1)$$

$$\frac{dQ_n}{d\theta} = \left(1 + \frac{C_v}{C_v(\gamma - 1)}\right)p.\frac{dv}{d\theta} + \frac{C_v}{C_v(\gamma - 1)}.v\frac{dp}{d\theta}$$

$$\frac{dQ_n}{d\theta} = \left(1 + \frac{1}{(\gamma - 1)}\right)p.\frac{dv}{d\theta} + \frac{1}{(\gamma - 1)}.v\frac{dp}{d\theta}$$

$$= \left(\frac{\gamma - 1 + 1}{(\gamma - 1)}\right)p.\frac{dv}{d\theta} + \frac{1}{(\gamma - 1)}.v\frac{dp}{d\theta}$$

$$\frac{dQ_n}{d\theta} = \left(\frac{\gamma}{(\gamma - 1)}\right)p.\frac{dv}{d\theta} + \frac{1}{(\gamma - 1)}.v\frac{dp}{d\theta}$$

Test Engine

A complete experimental setup was developed with all required equipment to evaluate the performance, emission, and combustion characteristics of the test engine. The engine experimental setup consisted of a single-cylinder diesel engine (3.54 kW and constant speed of 1500 rpm) coupled with an eddy current dynamometer for power variation and torque measurement (shown in the figure. 1). The detailed specification of the test engine is given in

Table 1. The experimental setup has a standalone panel box consisting of an airbox, a twin fuel tank, a manometer, fuel measuring unit, and the fuel flow was measured using a burette and timed with a stopwatch. Rotameters were used for cooling water and calorimeter water flows measurement. For electronic fuel injection, Open ECU, Throttle position sensor (TPS), fuel pump, fuel spray nozzle, and a trigger sensor were used. The piezoelectric sensor fitted into the combustion chamber acquires the combustion pressure, and temperature, and the signal is sent to data acquisition (DAQ) which is coupled to the personal computer. Performance and combustion characteristics were studied using engine soft, a Lab view-based software. The engine had a conventional fuel injection system. The fuel injector opening pressure was set at the rated value of 200 bars. The governor of the engine was used to control the engine speed by adjusting the flow rate of the fuel.

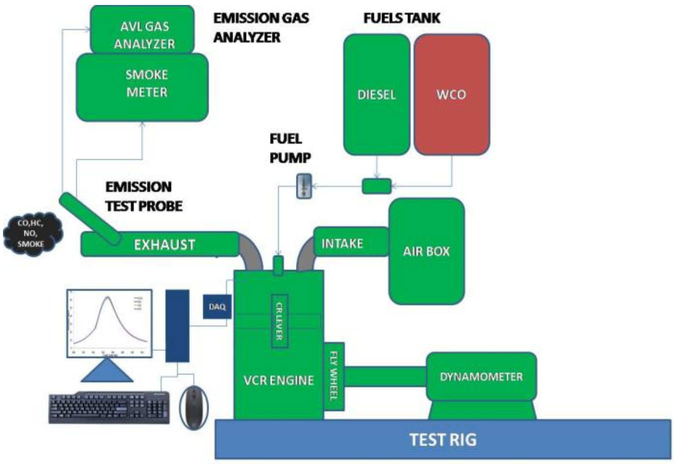

Figure 1. Schematic View of the Test Engine

Table 1. Test Engine specification

Make	Kirloskar
Type	Four stroke, single-cylinder, C.I engine, water-cooled, Direct Injection
Cubic Capacity	661cc
Bore × Stroke in mm	87.5 × 110
Compression Ratio	18 :1
Power rating	3.54kW
Rated Speed	1500 rev/min
Engine Loading type	Eddy Current type Dynamometer
Injection Timing (BTDC)	23°

Result and Discussion

The Heat Release Rate was calculated using thermodynamic theoretical modeling with diesel, WCO, WCO emulsion and WCO nano emulsion at maximum load and maximum compression ratio (18) of the engine (Shown in Fig.2 to 6).

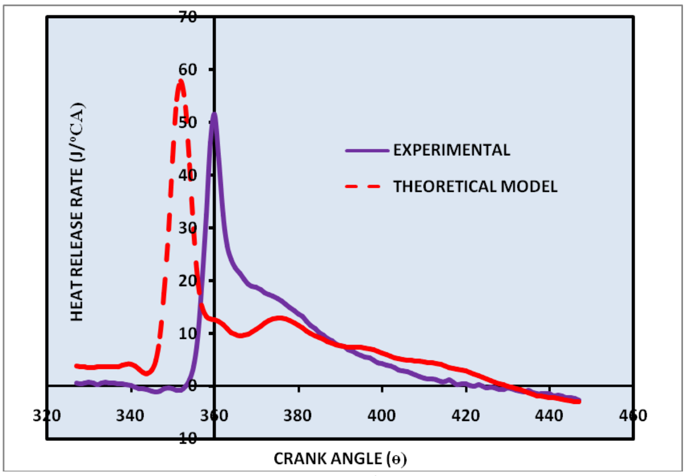

Fig. 2: HRR of Diesel

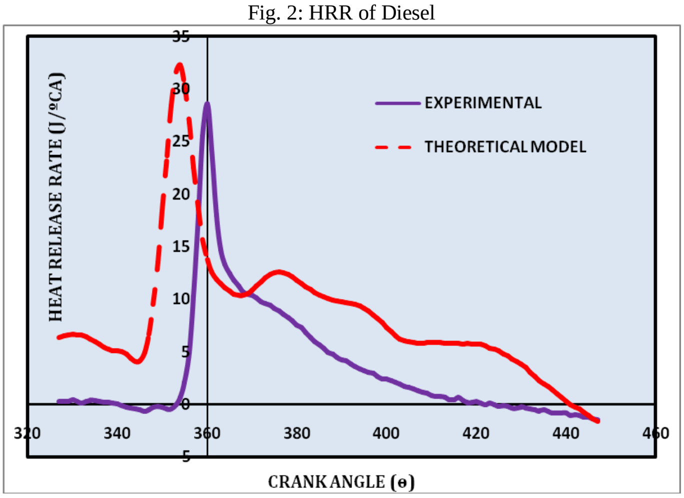

Fig. 3: HRR of WCO

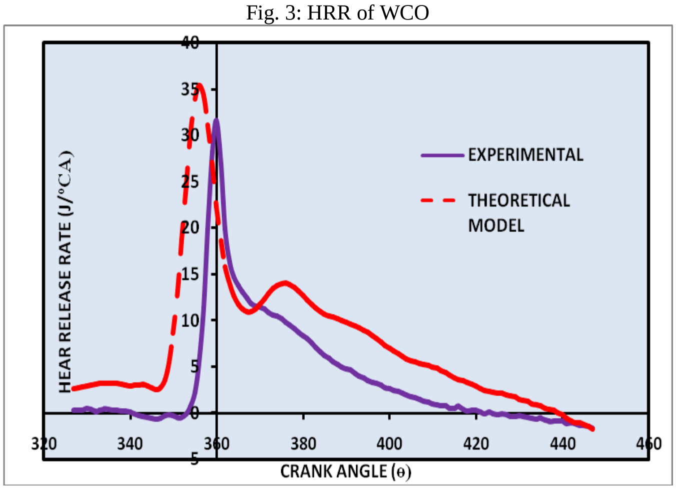

Fig. 4: HRR of WCOE

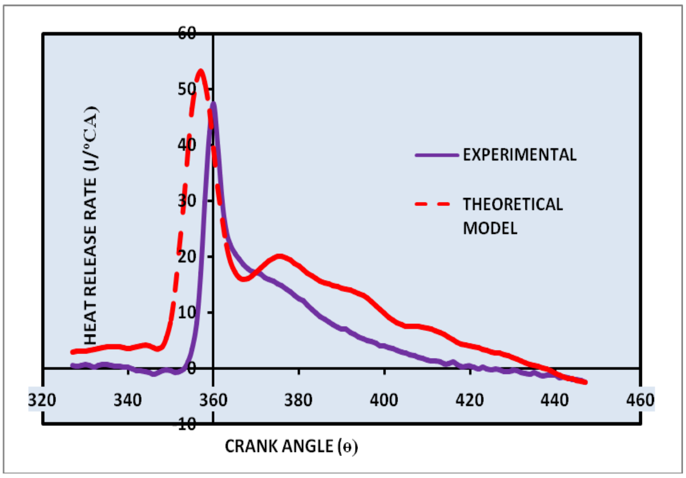

Fig. 5: HRR of WCONE

Ignition Delay

Ignition delay is defined as time interval between the start of injection and start of combustion. During the delay period, injected fuel gets atomized to finer droplets and evaporates by acquiring the temperature of hot compressed air. Evaporated fuel droplets mixes with the air and form combustible charge in the final phase of the delay period. Delay period was found to be one of the factors which decided the occurrence of peak pressure and temperature. An empirical formula for predicting the duration of the ignition delay period in DI engines (shown in Fig. 5 to 8). This formula gives the ignition delay (in crank angle degrees) in terms of charge temperature T (kelvins) and pressure p (bars) during the delay .The delay period was calculated using the Hardenberg's equation (7) listed in (Heywood, 1988).

$$\tau id(CA) = (0.36 + 0.22{*}Sp)exp\left[EA\left(\frac{1}{RT} - 5.81 * 10^{-5}\right)\left(\frac{21.2}{P-12.4}\right)^{0.63}\right] \rightarrow 7$$

Where,

EA = 618840/(CN+25)

R is the universal gas constant (8.314 J/mol.K)

EA is the apparent activation energy (joules per mole)

CN is the cetane number of the fuel.

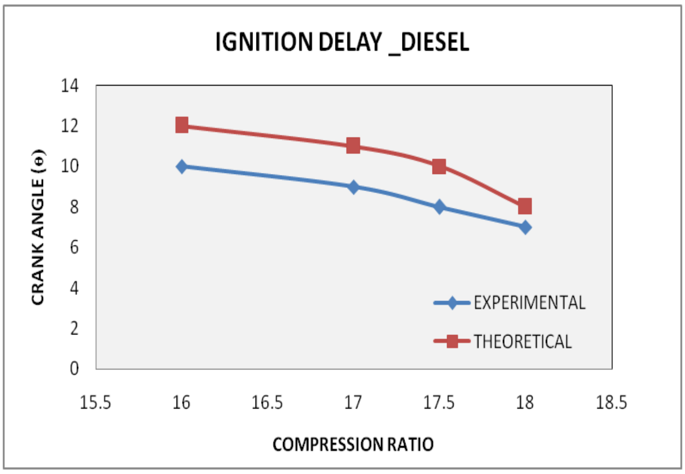

Fig. 6: Ignition Delay of WCO with respect to different compression ratios

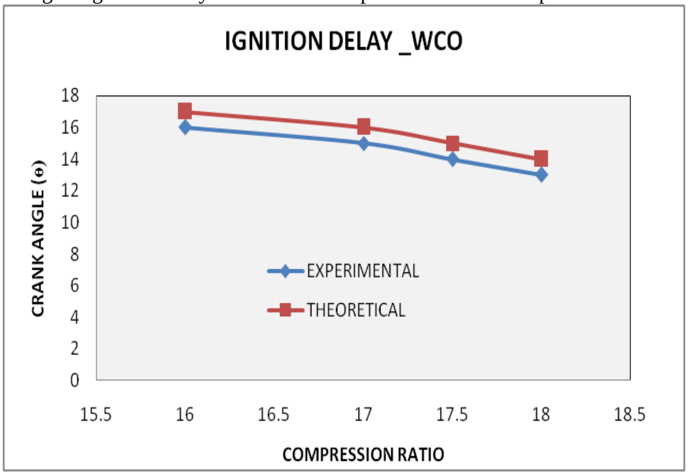

Fig. 7: Ignition Delay of WCO with respect to different compression ratios

The gaseous mixtures are compressed to high pressure in the compression stroke by increasing the compression ratio of the engine. The higher the pressure of the mixture the higher its temperature also. In this analysis, with increasing compression ratio of the engine the ignition delay is decreases for all the fuels at the peak power output. This was due to the influence of cylinder gas temperature within the ignition delay period. The gas temperature is higher at high engine loads than that at low loads. The higher temperature helped in accelerating the air fuel mixture preparation and reduced the physical and chemical processes involved during the ignition delay period. It was noted that the ignition delay was more with NWCO as compared to all other fuels at maximum power output and maximum compression ratio (Ramadhas et al, 2006).

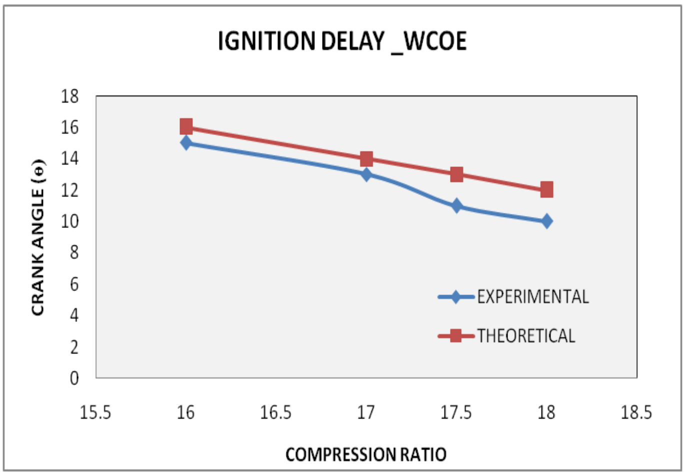

Fig. 8: Ignition Delay of WCOE with respect to different compression ratios

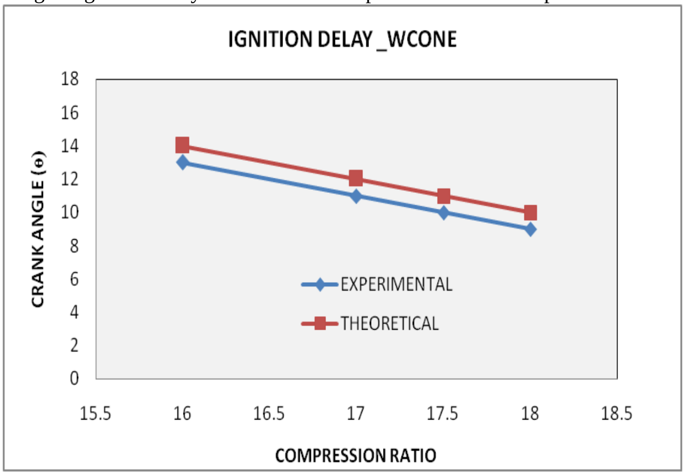

Fig. 9: Ignition Delay of WCONE with respect to different compression ratios

Conclusions

- The Heat Release rate of modeling results were closed to experimental results, however, the experimental results of WCONE increases as 10-15% as compared to WCO.
- The Ignition Delay of modeling results were closed to experimental results, however, the experimental results WCONE increases as 3-5% as compared to WCO.

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Definitions/Abbreviations

TTM	Theoretical Thermodynamic Model
ATDC	After top dead center
BP	Brake power

BDC	Bottom dead center
BTE	Brake Thermal Efficiency
CC	Cubic Capacity
CI	Compression ignition
CNT	Cabon nano tube
CO	Carbon monoxide
CO_2	Carbon dioxide
cSt	Centistokes
CuO	Copper oxide
DAQ	Data acquisition
DFM	Dual fuel Mode
DI	Direct injection
EFI	Electronic Fuel Injection
EES	Ethanol energy share
EG	Ethylene glycol
HLB	Hydrophilic–lipophilic balance
HRR	Heat release rate
kg/kW.hr	Kilogram per kilo watt hour
LHR	Low heat rejection
ND	Neat diesel
NO	Nitric oxide
NWCO	Neat waste cooking oil
mV	milli volt
OPEN ECU	Open electronic control unit
TPS	Throttle Position Sensor
rev/min	Revolution per minute
UCO	Used cooking oil
UHC	Unburnt hydrocarbon
VCR	Variable compression ratio
WCO-NF-EM	Waste cooking oil nano fluid emulsion
J/ºCA	Joule pet degree crank angle
ppm	Parts per million
wt.%	Weight in percentage
%	Percentage
% vol.	% Volume