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Computer Modelling For 4-Stroke Direct Injection Diesel Engine

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Abstract. Study on computational modeling of 4-stroke single cylinder direct injection diesel engine is presented. The engine with known specification is being modeled using one dimension CFD GT-Power software. The operational parameters of the engine such as power, torque, specific fuel consumption and mean effective pressure which are dependent to engine speed are being discussed. The results from the simulation study are compared with the theoretical results to get the true trend of the results.

Introduction

Computer modelling in solving fluid dynamic problem in engineering is rapidly grown. The 1D-CFD engine simulation tools commonly used in the entire engine development (e.g., GT-Power, Wave, PROMO, Boost) compute the flow field in internal combustion engines based on one-dimensional fluid dynamics, thus representing the flow and the heat transfer in the piping and the other components of an engine system [1]. These tools can resolve flow effects only in the main direction (the pipe axis). They assume uniformity of the flow field in each perpendicular plane.

A study of computational method to simulate a four stroke single cylinder direct injection diesel engine has been performed using one dimension CFD GT-Power software. In this research, the engine performance parameters such as power, torque, specific fuel consumption and mean effective pressure are studied.

Power

The engine power is divided into two types; gross and net power. The gross power is from an engine without the cooling fan, muffler and tail pipe; whereas the net power is from the complete engine. The energy through the engine is expressed in three distinct terms; *ip*, *fp* and *bp*. The *ip* can be measured from the measurement of forces in the cylinder and *bp* can be computed from the measurement of forces at the crankshaft of the engine. The difference between the *ip* and the *bp* of an engine is known as *fp*. It is shown in Eq. 1.

$$fp = ip - bp \tag{1}$$

Power is defined as the rate of doing work [4]. This may be converted to power by multiplying by the mass flow rate of air through the engine in kg per unit time. The net work obtained from the p-V diagram is the net work produced in the cylinder as measured by an indicator diagram, the power based on there is shown in Eq. 2.

$$ip = \dot{m}_a \times \text{ net work of cycle} \tag{2}$$

Nomenclature

A	area of piston (m^2)
B	net brake load (kg)
b	distance (m)
bmep	mean effective pressure (kPa/cycle)
bp	brake power (kW)
bsfc	brake specific fuel consumption (kg/kWh)
d	effective brake diameter (m)
F	force (N)
fmep	friction mean effective pressure (kPa/cycle)
fp	friction power (kW)
g	gravity constant (m/s^2)
imep	indicated mean effective pressure (kPa/cycle)
indicated net work/cycle	$= p_{im} V_s$
ip	indicated power (kW)
isfc	indicated specific fuel consumption (kg/kWh)
K	number of cylinders
L	length of the stroke (m)
$\dot{m}_a$	mass flow rate (kg/s)
mep	mean effective pressure (kPa/cycle)
N	speed in revolution per minute (rpm)
n	number of power strokes per minute = $N/2$ (for a four stroke engine)
net work	$= p_{im} \times (V_1 - V_2)$ (kJ/kg of air)
p_{im}	indicated mean effective pressure (N/m^2)
sfc	specific fuel consumption (kg/kWh)
T	tork (Nm)
V_d	displacement volume (m^3)
V_1	volume at point 1 in p-V diagram (m^3)
V_2	volume at point 2 in p-V diagram (m^3)
W	work (kJ)
$\dot{W}_b$	brake power (kW)
τ	engine torque (Nm)

In working with actual engines, it is often desirable to compute *ip* from a given p_{im} and given engine operating conditions as shown in Eq. 3.

$$ip = \frac{p_{im} V_s n K}{1000 \times 60} = \frac{p_{im} L A n K}{60000} \tag{3}$$

The *ip* is based on indicated net work and is thus a measurement of the forces developed within the cylinder. More practical interest is the rotational force available at the delivery point, at the engine crankshaft, and the power corresponding to it. This power is called *bp*. This power can be measured by attaching a power absorption device to the drive-shaft of the engine [4]. Meanwhile the *bp* is given by Eq. 4;

$$bp = \frac{W\pi Nd}{60000} = \frac{BgNd}{60000} \tag{4}$$

Torque

Torque is a measurement of an engine's ability to do work. Normally it is measured with a dynamometer. The rotor is coupled electromagnetically, hydraulically or by mechanical friction to a stator which is supported in low friction bearings. The stator is balanced with the rotor stationary. The torque exerted on the stator with the rotor turning is measured by balancing the stator with weights, springs or pneumatic means [3]. Using the notation in Figure 1, the torque exerted by the engine is shown in Eq. 5. The power delivered by the engine and absorbed by dynamometer is the product of torque and angular speed, as shown in Eq. 6. It can be simplified as in Eq. 7.

$$T = Fb \tag{5}$$

$$P = 2\pi NT \tag{6}$$

$$T = \frac{P \times 60000}{2\pi N} \tag{7}$$

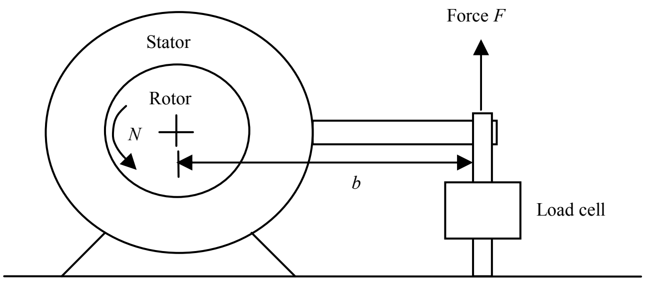

Figure 1: Parameters involved in dynamometer [3]

Specific fuel consumption

The parameter that indicates how good the engine performance is *sfc*. The lower *sfc* means the better the engine. This parameter is expressed in term of specific fuel consumption in kilograms of fuel per kilowatt-hour. There are two types of *sfc*; *bsf* and *isfc* [2,4]. *Bsfc* and *isfc* are the specific fuel consumptions base on *bp* and *ip* respectively.

$$sfc = \frac{\text{Fuel consumption per unit time}}{\text{Power}} \tag{8}$$

Mean effective pressure

The parameter that measure the work done per unit displacement volume is *mep* [2]. It increases as manifold pressure increases. There are two important parameters for *mep*; *imep* and *bmep*. The *imep* is the net work per unit displacement volume done by the gas during compression and expansion. It may be considered to consist of *fmep* and *bmep* as shown in Eq. 9. *fmep* is the portion of friction losses. Meanwhile, the *bmep* is the external shaft work per unit volume done by the engine.

$$imep = bmep + fmep \tag{9}$$

The *mep* is the average pressure that results in the same amount of indicated or brake work produced by the engine. Based on torque for four stroke engine, the *bmep* is shown in Eq. 10 and in term of power the *bmep* is shown in Eq. 11.

$$bmep = \frac{4\pi\tau}{V_d} \tag{10}$$

$$bmep = \frac{2\dot{W}_b}{V_d N} \tag{11}$$

Methodology

In this study the computer simulation method using GT Power is used. The engine specification is shown in Table 1. This engine being modeled in GT Power software is shown in Figure 2. Air with compositions of 76.7% N_2 and 23.3% O_2 at pressure 1 bar and temperature 26.85°C is being modeled as env-01. The properties for env-02 are similar with env-01. The intake system consists of intrunner_airfilter (intake runner for air filter), airfilter-01 (airfilter box), intrunner-01 (intake runner) and intport-01 (intake port). These parts represent the pipes with specific parameters as shown in Table 2. The exhaust system consists of exhport-01 (exhaust port), exhrunner-01 (exhaust runner), muffler-01 (muffler) and exhrunner-exit-01 (exhaust tail). These parts also represent the pipes with specific parameters as shown in Table 3. Intvalve-01 represents intake valve and exhvalve-01 represents exhaust valve. The parameters of these valves are shown in Table 4.

Table 1: Engine specification

Item	Technical Specification
Type	Single cylinder, vertical, air-cooled, direct injection, 4 stroke
Bore (mm)	86
Stroke (mm)	70
Displacement (L)	0.406
Initial angle of fuel delivery (CA)	-22º ± 1º BTDC
Intake valve clearance (mm)	0.1~0.15 (cold state)
Exhaust valve clearance (mm)	0.1~0.15 (cold state)
Exhaust temperature (ºC)	≤ 480
Pressure of injection (MPa)	≤ 19.6 ± 0.49
Mean effective pressure (kPa)	561.6 @ 3000 rpm, 543.5 @ 3600 rpm
Consumption rate of fuel (g/kW)	≤ 273.5 @ 3000 rpm, ≤285.7 @ 3600 rpm

Table 2: Parameters for intake system

	Intrunner_airfilter	Airfilter-01	Intrunner-01	Intport-01
Diameter at inlet end (mm)	44.88	159.632	40.44	40.6973
Diameter at outlet end (mm)	62.13	159.632	40.1	32.78
Length (mm)	80	69.64	59.7	55.2
Discreatization length (mm)	34.4	34.4	34.4	34.4
Wall temperature (ºC)	28.85	28.85	76.85	176.85

Table 3: Parameters for exhaust system

	exhport-01	exhrunner-01	muffler-01	exhrunner-exit-01
Diameter at inlet end (mm)	26.38	27.86	138.88	34.6
Diameter at outlet end (mm)	29.82	27.86	138.88	34.6
Length (mm)	40.4	98	283.4	25.6
Discreatization length (mm)	47.3	47.3	47.3	47.3
Wall temperature (ºC)	480	480	480	480

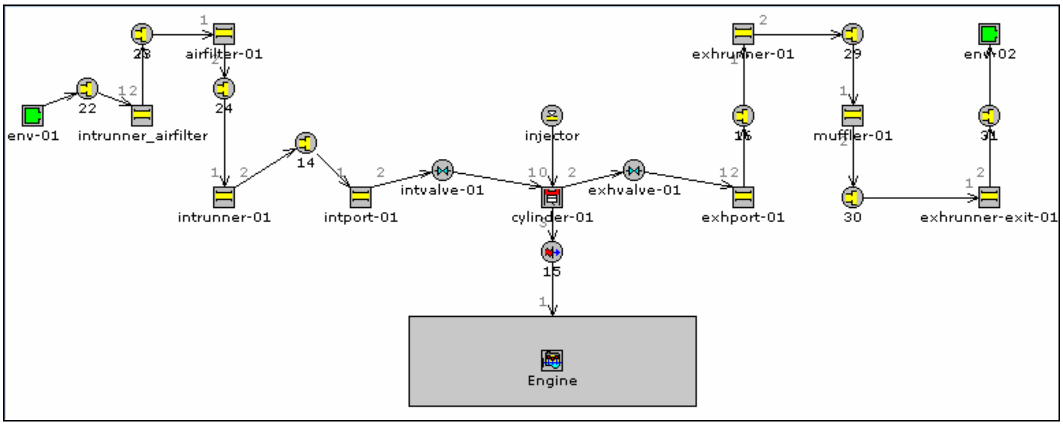

Figure 2: Engine modeling using GT Power software

Table 4: Parameters for intake and exhaust valves

	Intvalve-01	Exhvalve-01
Valve diameter (mm)	35.54	29.04
CAM timing angle (°CA)	462.5	214.6
Valve lash (mm)	0.125	0.125

Results and discussion

For power output, only indicated power is being discussed. This is due to data specification from engine manufacturer is not enough for calculation of brake power. Indicated power gained from simulation is plotted against rpm as shown in Fig. 3. From the graph, the indicated power value is selected at 3000 and 3600 rpm as a comparison to theoretical value. By using the simulation software the results are 5.63 kW at 3000 rpm and 6.54kW at 3600 rpm, meanwhile the results by using Eq. 3 are 5.72kW at 3000 rpm and 4.93kW at 3600rpm. By comparison the results at 3000 rpm are significant.

Indicated torque by simulation is shown in Fig. 3. Similar to indicated power, the indicated torque that has been selected for comparison for indicated torque are 5.72kW at 3000 rpm and 4.93kW at 3600 rpm. By referring to Eq. 6 and indicated power obtained by calculation at specific rpm, the indicated torque theoretically can be calculated. The results are 5.63kW at 3000 rpm and 6.54kW at 3600 rpm. By comparison the results at 3000 rpm are significant.

Specific fuel consumption is one of the important parameters to be considered whether the engine is good or not. For this research, only the indicated specific fuel consumption is studied. Fig. 4 shows the indicated specific fuel consumption result for simulation. The values of result are 275.115 and 383.007 g/kWh at 3000 and 3600 rpm respectively. The results from engine manufacturer are 273.5 and 285.7 g/kWh at 3000 and 3600 rpm respectively. By comparison the results at 3000 rpm are significant. Indicated mean effective pressures as mention by engine manufacturer are 5.616 and 5.435 bar at 3000 and 3600 rpm respectively.

Indicated mean effective pressure result for simulation is shown in Fig. 4. From the graph, the indicated mean effective pressures are 5.632 and 4.045 bar at 3000 and 3600 rpm respectively. By comparison the results at 3000 rpm are significant.

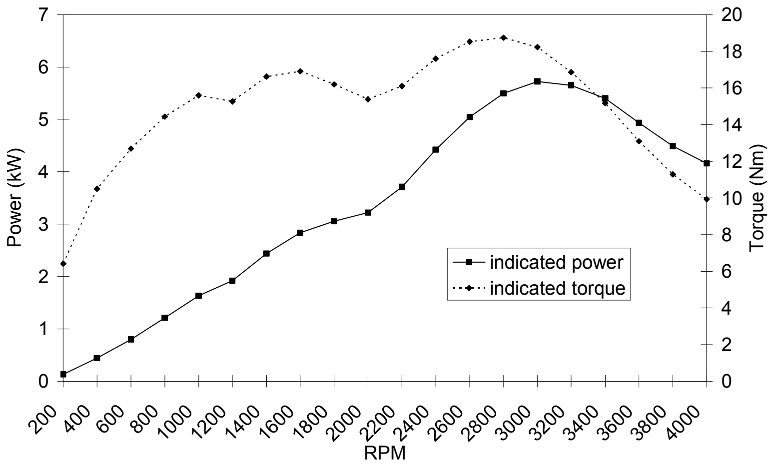

Figure 3: Indicated power and torque graph for simulation.

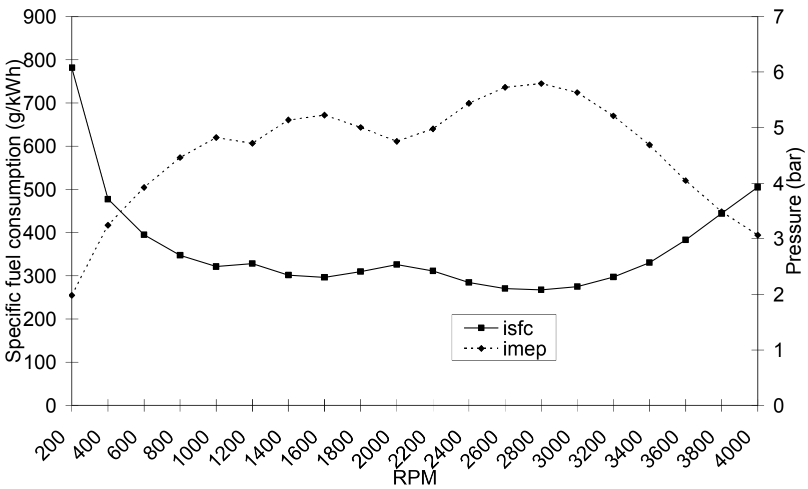

Figure 4: Indicated specific fuel consumption and mean effective pressure graph for simulation.

Conclusion

Comparison of parameters such as indicated power, indicated torque, indicated specific fuel consumption and indicated mean effective pressure have been successfully validated with the results from engine manufacturer and theoretical equation. The results at 3000 rpm are significant compared to results at 3600 rpm. Although the results for 3600 rpm is not significant but the trend of the graph shows the similar pattern with other researchers. For further research, experimental study will be conducted to validate the simulation results.

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