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Performance Characteristics of a Single-Cylinder Two-Stroke Diesel Engine using Diesel-RK Software

[1]Olaiya Kamorudeen Adesina, [2]Alabi Ismaila Olanrewaju, [3]Okediji Adebunmi Peter

Abstract—The demand for software that is capable of solving internal combustion engines (ICEs) simulation problems is increasing on a daily basis. Thus, this article presents the application of Diesel-RK in investigating the performance characteristics of a single cylinder two-stroke turbocharged Hesselman's diesel engine with direct fuel injection. Three different engine operating speed 1500, 2000 and 2500 rpm, respectively were utilized and their effects on certain engine performance parameters were investigated numerically. The highest overall specific fuel consumption and engine efficiency of 0.53342 kg/kWh and 0.1588, respectively were obtained at 2000 rpm. More power was delivered at 2000 rpm as a result of the highest value of engine torque obtained at that speed. The cylinder pressure increases significantly as the engine speed increases at ivc (2.0451 bar), evo (3.7801 bar) and tdc (60.141 bar), respectively. This indicates that the pressure developed when the inlet valve closes increases as the piston translate from bdc to tdc on compression, to the value required for combustion and this value dropped as the exhaust valve opens. However, the optimal values corresponding cylinder temperature increases significantly at ivc (717.24 K), evo (1088.5 K) and tdc (1560.8 K), respectively at a speed of 2500 rpm. Also, the engine speed had appreciable impact on the heat exchange parameters, because the values of those parameters increases as the engine speed increases. The effect of engine speed on the ecological, combustion, turbocharged and gas exchange parameters were also studied using Bosch, Hartridge and Strouhal dimensionless numbers and it is evident that engine speed is significant in the study of engine performance parameters and the best values of those parameters were obtained at 2000 rpm.

Index Terms— Hesselman's Diesel Engine, Two-Stroke, Engine Speed, Bosch Number, Performance Parameters.

1 INTRODUCTION

In compression ignition (CI) engines, air alone is inducted into the cylinder. The fuel (in most applications a light fuel oil, through heated residual fuel is used in marine and power-generation applications) is injected directly into the engine cylinder just before the combustion process starts. Load is achieved by varying the amount of fuel injected into each cycle; the air flow at a given engine speed is essentially unchanged. There are great variety of CI engine designs used in a wide range of applications; automobile, truck, locomotive, marine and power generation.

The most difficult challenge for modern 4-Stroke high speed Diesel engines is the limitation of pollutant emissions without penalizing performance, overall dimensions and production costs, the last ones being already higher than those of the corresponding Spark Ignition (SI) engines. An interesting concept to meet up with the conflicting requirements mentioned above is the 2-Stroke cycle combined to Compression Ignition. Such a concept is widely applied to large bore engines, on steady or naval power-plants, where the advantages versus the 4-Stroke cycle in terms of power density and fuel conversion efficiency (in some cases higher than 50%) are required.

In fact, the double cycle frequency allows the designer to either downsize (i.e. reduce the displacement, for a given power target) or "down-speed" (i.e. reduce engine speed, for a given power target) the 2-stroke engine. Furthermore, mechanical efficiency can be strongly improved, for two (2) reasons:

a) the gas exchange process can be completed with piston-controlled ports, without the losses associated to a valve-train;
b) the mechanical power lost in one cycle is about halved, in comparison to a 4-Stroke engine of same design and size, while the indicated power can be the same: as a result, the weight of mechanical losses is lower.

Unfortunately, the 2-Stroke technology used on steady or naval power-plants cannot be simply "scaled" on small bore engines, for a number of reasons. First of all, the increase of engine speed makes combustion completely different, in particular for what concerns the ignition delay; second, small Diesel engines are generally designed according to different targets and constraints (for instance, they have to be efficient and clean on a wider set of operating conditions, they must comply with specific emissions regulations, etc.); third, most of the engine components (such as bearings, connecting rods, piston rings, etc.) are generally different, at least from a structural point of view. As a result, a brand-new engine design is mandatory to develop a successful 2-Stroke high speed CI engine.

Presently, research focus is on development of software for solving internal combustion engines simulation problems and complete the working cycle to solve problems related to combustion in diesel engines [1][2]. The demand for such software is very large, while the prices of software from well-known brands Boost (AVL), Wave (Ricardo), GT-Power (Gamma Technologies) are very high, possibly up to hundreds of thousands of dollars. However, the above software does not allow to study in detail the effects of the certain engine design parameters like combustion chamber shape, spray beam direction and other characteristics on the quality of combustion process [2][3]. To solve the problems listed above, it is necessary to use Computational Fluid Dynamic 3D (CFD) simulation technology. The demand for large computer resources for programs such as KIVA (Los Alamos), FIRE (AVL), STAR-CD (Computational Dynamics), VECTIS (Ricardo) has limited the ability to use them to optimize when solving all technical problems [2][4].

However, software for calculating internal combustion engines developed by Bauman Technical University (Russian Federation) experts called Diesel-RK has been used by many facilities specializing in research, development, and production.

Some other software often used in calculating the working cycle for diesel engines have not investigated the effect of combustion dynamics and formation of diesel engine emissions when changing the above engine design parameters, evaluating and analyzing technical specifications and economic of engines. Therefore, the research on applying Diesel-RK software to teaching and researching marine diesel engines is very practical and meaningful, especially in the trend that sea transport is increasingly involved in "green transport." [5].

Therefore, this article presents the application of Diesel-RK in investigating the performance characteristics of a single cylinder two-stroke diesel engine.

2 DIESEL-RK SOFTWARE

2.1 Features of Diesel RK software

The main features of DIESEL-RK software are similar to known thermodynamic software. However, along with popular features, it has new advanced applications that other programs do not have, such as oriented optimization of the combustion process in diesel engines and internal combustion engine (ICE) analysis and optimization, assuming the same working of all cylinders in the engine, it allows a significant increase in operating speed and makes it possible to optimize the complex diesel engine tasks [2].

DIESEL-RK software simulates the thermodynamic cycle of relatively complete engines. The software is designed to simulate and optimize the working processes of combustion engines for 2-stroke engines and 4-stroke engines with a turbocharger. This Software can simulate the model of the following types of engines [2]:

1) Direct Injection (DI) diesel engine, including PCCI and also biofuel engine;
2) SI gasoline engine;
3) SI gas engines include a pre-combustion chamber system, engines can use different gases: methane, propane-butane, biogas, synthetic gas;
4) Reversible or non-reversing two-stroke engines, reciprocating piston engines (OP motors or Junkers) and OPOC engines;
5) Dual fuel engine (the engine has an independent fuel injection system for different fuels) (Engine with RCCI).

DIESEL-RK includes "Fuel Spray Visualization" code. This code allows the presentation of simulation results in the form of animations, resulting in modelling the interaction between the fuel sprays with the combustion chamber wall, the degree of swirls and the adjacent spray area [6]. Code assists in selecting the best shape of the piston crown and choosing the diameter and number of nozzle holes, direction and angle of spray at a time and a certain swirl intensity level. Simulation results can be saved as a multimedia format of Windows, AVI or GIF animation format. Collected images may include 2D or 3D images.

2.2 Applicability of Diesel-RK software

Software can be applied to solve problems in diesel engines: (1) Predicting torque graphs and engine performance; (2) Predict and optimize fuel consumption; (3) Analysis and optimization of combustion and emission processes;(4) Knock prediction; (5) Optimize working time of valves; (6) Analysis and optimization of EGR system; (7) Optimizing the combination of turbocharged turbines and emissions; (8) Research and optimize fuel spray characteristics of both multi-nozzle system including spray shape and spray position as well as optimize crown piston shape; (9) Convert diesel engine into gas engine; (10) Analysis of engines using dual fuel [7].

2.3 Existing application of Diesel-RK to Marine

[8] analyzed a 6S50MC MAN B&W marine diesel engine which was a slow speed turbocharged two-stroke engine with direct fuel injection. A cross section of the analyzed marine diesel engine 6S50MC MAN B&W is presented in Fig.1. In Fig.1 can be seen all of the housing and cylinder main elements. The engine was built in a diesel engine factory in Split, Croatia, according to the license MAN B&W.

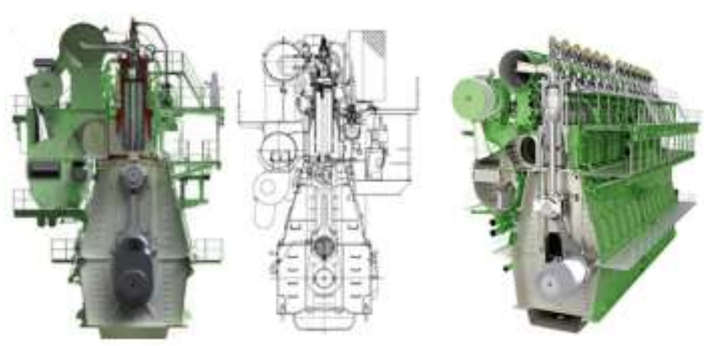


Fig.1. Cross section of marine slow speed two-stroke diesel engine 6S50MC MAN B&W [8]

The effect of engine loads on operating parameters and

- *Olaiya, K.A. is currently The Head, Department of Mechatronics Engineering in The Polytechnic Ibadan, Nigeria, PH-+2348059782951. E-mail: olaiya.kamorudeen@polyibadan.edu.ng*
- *Alabi, I.O. is currently a Lecturer at the Department of Automotive Engineering, Elizade University Ondo State, Nigeria, PH-+2348060849804. E-mail: ismaila.alabi@elizadeuniversity.edu.ng*
- *Okediji, A.P. is currently The Head, Department of Mechanical Engineering in Elizade University Ondo State, Nigeria, PH-+2348030527383. E-mail: adebunmi.okediji@elizadeuniversity.edu.ng*

efficiencies of marine slow speed two-stroke diesel engine 6S50MC MAN B&W was investigated in their research. The highest engine mechanical efficiency of 94.52 % was obtained at engine load 100 % of MCR while the highest indicated engine efficiency of 52.70 % and the highest engine effective efficiency of 49.34 % were obtained at the engine load 75 % of MCR. During the engine load increase, mean effective pressure continuously increases from 7.47 bar at the lowest up to 17.97 bar at the highest observed engine load. The range of available engine effective torque was from 267380 Nm on the lowest up to 643594 Nm on the highest engine load, which is an expected range of developed effective torque for this kind of diesel engine. The range of analyzed engine specific effective fuel consumption was between 171.18 g/kWh and 186.83 g/kWh. Obtained range of specific effective fuel consumption proves the fact that marine two stroke diesel engines have the lowest specific effective fuel consumption of all diesel engines or of all engines in general. In comparison with the other types of internal combustion engines, marine slow speed two-stroke diesel engines have significantly higher effective efficiency which can nowadays reach above 50 %.

However, in this present study the effect of engine speed on performance parameters would be investigated.

3 SIMULATION MODEL SET UP

3.1 Sample engine for simulation

In this present study a Hesselman's marine diesel engine of a slow speed turbocharged two-stroke engine with direct fuel injection was analyzed. The main engine specifications are presented in Table 1.

Table 1. Specifications of Hesselman's marine slow speed two-stroke diesel engine

Content	Value	Unit
Process type	2-stroke Direct injection	
Number of cylinders	6 in-line	
Cylinder bore	100	Mm
Stroke	80	Mm
Ignition sequence	1-5-3-4-2-6	
Air/Fuel equivalent	1.75	kg/kg, on 100% load
Engine speed at MCR	1500, 2000 & 2500	rpm
Pressure ratio of HPC	1.8832	
Mech. Efficiency of HPC	95	%
Specific fuel consumption (with high efficiency turbocharger)	500	g/kWh, on 100% load
Compression ratio	15	

3.2 Diesel-RK simulation process

This is structured into the following:

1) **Engine design:** the selected marine engine design was based on:

a) **Working cycle:** the engine under study operates on a two-stroke cycle. The two stroke diesel engines are generally employed in marine propulsion. it is a type of ICE engine which completes a power cycle with two strokes (up and down movements) of the piston during only one crankshaft revolution.

b) **Fuel and Method of Ignition:** Diesel was selected as the main fuel and compression ignition was taken as the method of ignition. In diesel engines (also called compression ignition engines or C.I engines), only air is supplied to the engine cylinder during suction stroke and it is compressed to a very high pressure, thereby raising its temperature from 600°C to 1000°C. The desired quantity of fuel (diesel) is now injected into the engine cylinder in the form of a very fine spray and gets ignited when comes in contact with the hot air.

c) **Port Design and Location:** the port was designed in such a way that the crankcase can scavenge the exhaust gases easily. So, a crankcase scavenging method was adopted.

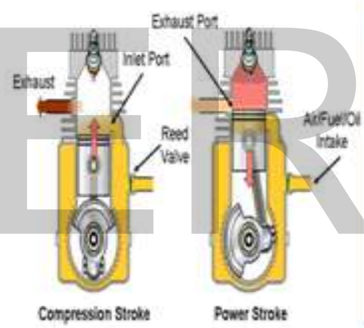


Fig. 2: Two-stroke engine cycle **Fig. 3: Engine Design**

2) **General engine parameters:** The following parameters define the basic geometry of a reciprocating engine:

a) **Geometry properties:** The function of a cylinder is to retain the working fluid and to guide the piston. In designing a cylinder for an I. C. engine, it is required to determine the following values:

i. **Cylinder bore (B) and stroke (*l*):** cylinder bore is the inner diameter of the cylinder while its stroke is the distance between the Top Dead Center (TDC) and Bottom Dead Center (BDC). The bore and stroke maybe determined as presented below: We know that the power produced inside the engine cylinder, i.e. indicated power,

$$\boldsymbol{I.P.} = \frac{p_m l A n}{60}\ \boldsymbol{watts} \quad \textbf{(1)}$$

Let p_m =Indicated mean effective pressure in Nmm^{-2}, D =Cylinder bore in mm,

A = π $D^2/4$, Cross-sectional area of the cylinder in mm^2

L =Length of stroke in metres,

N =Speed of the engine in rpm., and

n =Number of working strokes per min

= N, for two stroke engines

= N/2, for four stroke engines.

Since there is a clearance on both sides of the cylinder, therefore length of the cylinder is taken as 15 percent greater than the length of stroke.

ii. **Cylinder bore to stroke ratio:** The cylinder bore to piston stroke ratio required is

$$\boldsymbol{R_{ba} = \frac{B}{L}} \quad (2)$$

B/L = 0.8 to 1.2 for small- and medium-size engines, decreasing to about 0.5 for large slow-speed CI engines.

iii. **Compression ratio (r_c):** This is the ratio of the maximum engine cylinder volume to the clearance (minimum cylinder) volume.

$$\boldsymbol{r_c = \frac{v_d + v_c}{v_c}} \quad (3)$$

Where v_d is the displaced or swept volume and v_c is the clearance volume. Typical values of these parameters are: r_c = 8 to 12 for SI engine and r_c = 12 to 24 for CI engines.

iv. **Connecting rod length to crank radius:** This is the ratio of the connecting rod length to the crank radius.

$$\boldsymbol{R = \frac{l}{a}} \quad (4)$$

In addition, the stroke and the crank radius are related by

$$\boldsymbol{L = 2a} \quad (5)$$

R = 3 to 4 for small- and medium- size engines, increasing to 5 to 9 for large slow-speed CI engines.

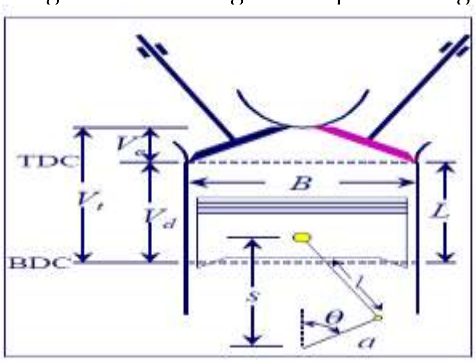


Fig. 4: Geometry of cylinder, piston, connecting road, and crankshaft. *Where B = bore, L = stroke, l = connecting rod length, a = crank radius, ϴ = crank angle.*

v. **Mean piston speed:** An important characteristic speed is the mean piston speed:

$$\boldsymbol{\overline{S_p} = 2LN} \quad (6)$$

Where, N is the rotational speed of the crankshaft. Mean piston speed is often a more appropriate parameter than crank rotational speed for correlating engine behaviour as a function of speed.

b) **Cylinder head:** Usually, a separate cylinder head or cover is provided with most of the engines. It is, usually, made of box type section of considerable depth to accommodate ports for air and gas passages, inlet valve, exhaust valve and spark plug (in case of petrol engines) or atomizer at the centre of the cover (in case of diesel engines). The cylinder head may be approximately taken as a flat circular plate whose thickness (t_h) may be determined from the following relation:

$$\boldsymbol{t_h = D\sqrt{\frac{C.p}{\sigma_c}}} \quad (7)$$

Where, D = Cylinder bore in mm, p =Maximum pressure inside the cylinder in Nmm^{-2}, σ_c = Allowable circumferential stress in MPa or N/mm^2. It may be taken as 30 to 50 MPa, and C = Constant whose value is taken as 0.1.

Thus, the mean cylinder head wall temperature was calculated by solving the heat conduction problem for multilayer wall. Meanwhile, cast iron was considered for the cylinder head.

c) **Piston and Rings:** The main function of engine piston is to receive the impulse from the expanding gas and to transmit the energy to the crankshaft through the connecting rod. It must also disperse a large amount of heat from the combustion chamber to the cylinder walls. The piston rings are used to seal the cylinder in order to prevent leakage of the gas past the piston.

The piston head or crown is designed keeping in view the following two main considerations, *i.e.*

1) It should have adequate strength to withstand the straining action due to pressure of explosion inside the engine cylinder, and

2) It should dissipate the heat of combustion to the cylinder walls as quickly as possible.

Based on strength, the thickness of the piston head (t_H), according to Grashoff's formula is given by

$$\boldsymbol{t_H = \sqrt{\frac{3p.D^2}{16\sigma_t}}(mm)} \quad (8)$$

On the basis of second consideration of **heat transfer**, the thickness of the piston head should be such that the heat absorbed by the piston due combustion of fuel is quickly transferred to the cylinder walls. Treating the piston head as a flat circular plate, its thickness is given by

$$\boldsymbol{t_H = \frac{H}{12.56k(T_C - T_E)}(mm)} \quad (9)$$

Where, *p* = Maximum gas pressure or explosion pressure in N/mm^2, *D* = Cylinder bore or outside diameter of the piston in mm, and σ_t = Permissible bending (tensile) stress for the material of the piston

in MPa or N/mm². H = Heat flowing through the piston head in kJ/s or watts, k = Heat conductivity factor in W/m/°C. The temperature difference (TC – TE) may be taken as 220°C for cast iron and 75°C for aluminium.
The heat flowing through the piston head (H) may be determined by the following expression, *i.e.,* $\boldsymbol{H = C \times HCV \times m \times B.P.}$ (in kW), **where C = 0.05 (Constant representing that portion of the heat supplied to the engine which is absorbed by the piston).**
However, the way of calculating mean piston wall temperature is by computing under the formular

$$\boldsymbol{T_{w\_pist} = C.T_{w\_head}} \qquad \textbf{(10)}$$

Meanwhile cast iron was considered for the piston and rings, because; (i) its mechanical strength is good at high temperatures, unlike aluminium that lose about 50% of their strength at temperatures above 325°C. (ii) it has good wearing properties and (iii) it retains spring characteristics even at high temperatures.

d) **Friction:** This deals with the opposing force that resists the movement of a body that slides relative to the other. It may occur as a result of the reciprocating movement of the piston in the cylinder which results into heat generation, wear and tear, poor metal surface finish and so on due to metal to metal contact.

Thus, a friction pressure model utilized is:

$$\boldsymbol{P_{fr} = A.C_m + B.\overline{p}} \qquad \textbf{(11)}$$

Where, C_m is a mean piston speed (m/s);p is mean cylinder pressure (bar). A and B are friction coefficient as developed by the model.

e) **Heat transfer and cooling system:** The heat transfer coefficient approximation for engine cylinder and corresponding scaling factor 'a_s' was obtained using the original Woschni's formula.

Woschni accounts for the increase in the gas velocity in the cylinder during combustion [9]. The fundamental Woschni's correlation without swirl (12), (13), and (14) is described in detail in [10]. Coefficients C_1 and C_2 in the relation of in-cylinder gas velocity depend on the engine cycle phase (13).

$$h = zB^{-0.2}p^{0.8}T^{-0.53}w^{0.8} = A_1B^{A_2}p^{A_3}T^{A_4}w^{A_5} \qquad \textbf{(12)}$$

$$w = C_1S_p + C_2\frac{v_dT_{ref}}{p_{ref}v_{ref}}(p - p_m) = A_6S_p + A_7\frac{v_dT_{ref}}{p_{ref}v_{ref}}(p - p_m) \qquad \textbf{(13)}$$

$$p_m = p_{ref}\left(\frac{v_{ref}}{v}\right)^k = p_{ref}\left(\frac{v_{ref}}{v}\right)^{A_8} \qquad \textbf{(14)}$$

Gas exchange period: C_1 = 6.18; C_2 = 0; compression period: C_1 = 2.28; C_2 = 0; combustion and expansion period: C_1 = 2.38; C_2 = 3.24.10⁻³; then z = 0.01297829376

In addition, air cooling system was considered. The following parameters were also considered for the heat transfer in the crank case: scavenging open area, exhaust manifold, exhaust porting, exhaust open area, intake manifold, crankcase and scavenging porting.

3) **Fuel injection system and combustion chamber:** In this window, it is necessary to enter the parameters for the calculation of mixture formation and combustion in diesel engines (and forced combustion engines). The injection timing of the operating model table as well as the swirl ratio were computed for cylinder at inlet valve closed (IVC) and at full capacity automatically by V.Galgovski's method. Other parameters considered include: injector profile, particulate matter (PM) and oxides of Nitrogen (NOx) emissions, R-K model settings, injector design and piston bowl design.

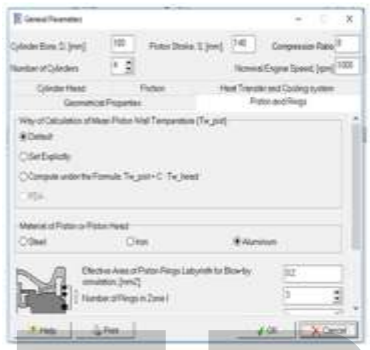


Fig. 5: General Parameters

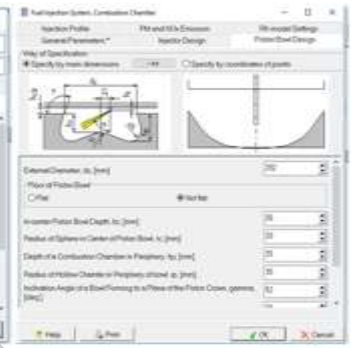


Fig. 6: Fuel Injection System and Combustion Chamber

4) **Gas exchange system:** This was as set under heat transfer and general cooling system in general engine parameters. It is necessary to set the basic parameters of the charging / discharging system: the time of opening and closing the valve, the design of the intake and exhaust ports, designing the inlet and outlet pipes.

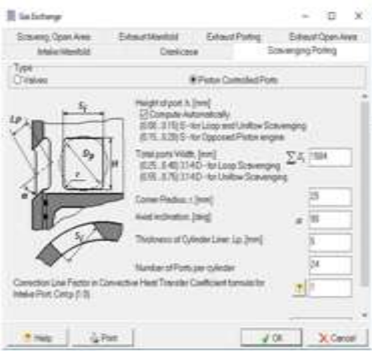
5) **Super- and Turbocharging:** Engines may be naturally aspirated (NA), supercharged or turbocharged.

   i. Naturally Aspirated Engines (NA): here, the atmospheric air is inducted.

   ii. Turbocharged Engines: here the inlet air is compressed by an exhaust-driven turbine-compressor combination,

   iii. Supercharges Engines: here, the air is compressed by a mechanically driven pump or blower.

Turbocharging and supercharging increase engine output by increasing the air mass flow per unit displaced volume, thereby allowing an increase in fuel flow. These two methods are used, usually in larger engines, to reduce engine size and weight for a given power output. Except in smaller engine sizes, the two-stroke cycle is competitive with the four-stroke cycle, in large part because, with the diesel cycle, only air is lost in the cylinder scavenging process. In this case a supercharged engine was considered



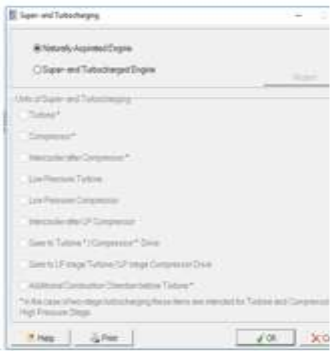


Fig. 7: Gas Exchange **Fig. 8: Super and Turbocharged**

6) **Fuel:** This contains the fuel parameters group that allows input fuel properties and to choose the type of fuel.

7) **Operating mode:** This parameter group is used to declare the parameters of calculation mode such as speed mode, amount of fuel supplied for a cycle (or air residue coefficient), early injection angle, pressure, ambient temperature etc.

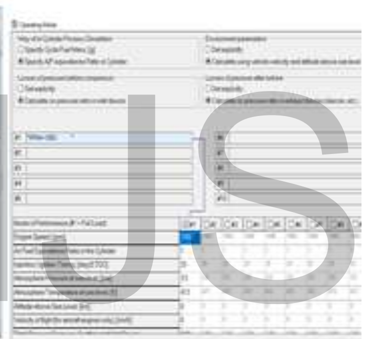


Fig. 9: Fuel **Fig. 10: Operating Mode**

4 RESULTS AND DISCUSSION

The results obtained from the simulation is presented below

4.1 Effect of engine speed on engine parameters

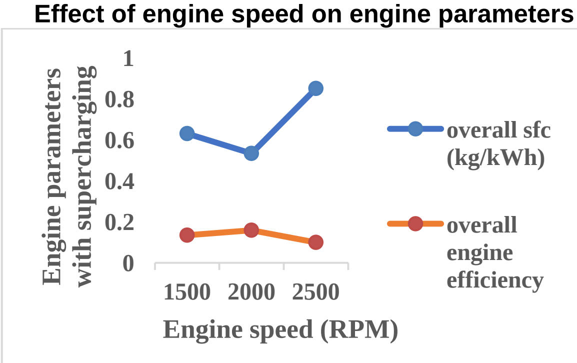


Fig. 11: Engine parameters against engine speed

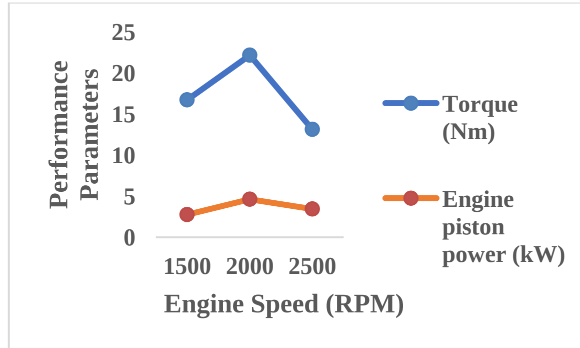


Fig. 12: Performance parameters against engine speed

Figure 11 shows the effect of engine speed on engine parameters with supercharging. The overall specific fuel consumption is lower at 2000 RPM and the overall engine efficiency also had appreciable value at a speed of 2000 RPM. This is an indication that the engine performed optimally at any speed below 2500 RPM.

Figure 12 shows the effect of engine speed on engine performance parameters. More torque is required at 2000 RPM and this resulted into the highest engine power. That is, more power was delivered at a speed of 2000 RPM.

4.2 Effect of engine speed on cylinder parameters

Figure 13 shows the effect of engine speed on engine cylinder pressure. The cylinder pressure increases significantly as the engine speed increases at inlet valve closed (ivc), exhaust valve open (evo) and top dead centre (tdc), respectively. But these values were optimal at a speed of 2000 rpm. That is, 2.0451, 3.7801 and 60.141 K at ivc, evo and tdc, respectively. The implication of these results is that the pressure developed when the inlet valve closes increases as the piston translate from bdc to tdc on compression, to the value required for combustion and this value dropped as the exhaust valve opens.

The effect of engine speed on engine cylinder temperature is as shown in fig. 14. The cylinder temperature increases significantly as the engine speed increases at inlet valve closed (ivc), exhaust valve open (evo) and top dead centre (tdc), respectively. But these values were optimal at a speed of 2500 rpm. That is717.24, 1088.5 and 1560.8 bar at ivc, evo and tdc, respectively. The implication of these results is that the temperature developed when the inlet valve closes increases as the piston translate from bdc to tdc on compression, to the value required for combustion and as well as when the exhaust valve opens.

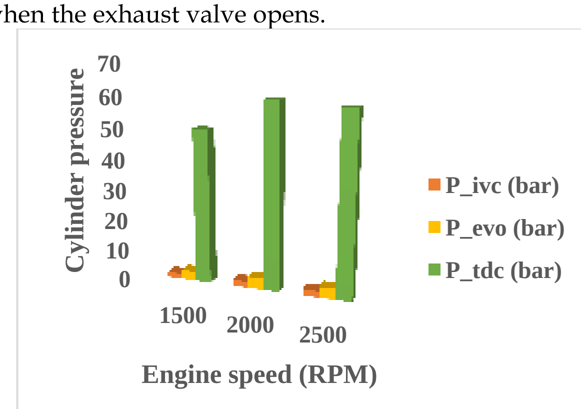


Fig. 13: Cylinder pressure against engine speed

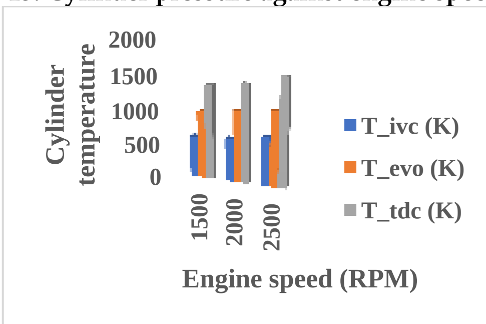


Fig. 14: Cylinder temperature against engine speed

4.3 Effect of engine speed on heat exchange in the cylinder

Table 2 showed that engine speed had appreciable impact on the heat exchange parameters, because the values of those parameters increases as the engine speed increases.

Table 2: Effect of engine speed on heat exchange in the cylinder

	1500 (rpm)	2000 (rpm)	2500 (rpm)
Cylinder average factor of heat transfer (W/m²K)	227.35	297.26	311.96
Average factor of heat transfer from head cooled surface to coolant (W/m²K)	8672	10037	10338
Average Piston Crown Temperature (K)	464.76	483.24	490.69

4.4 Effect of Engine Speed on Ecological Parameters

The ecological parameters are based on the level of smoke and other exhaust emission that are dispersed to the environment. These are evaluated based on Hartridge Smoke Level, Bosch Smoke Number and specific values of Particulate Matter (PM), Carbon dioxide (CO_2), Oxides of Nitrogen (NOx) and Sulphur dioxide (SO_2) emissions. Thus, the effect of engine speed on the ecological parameters are as presented in fig. 15 and fig. 16.

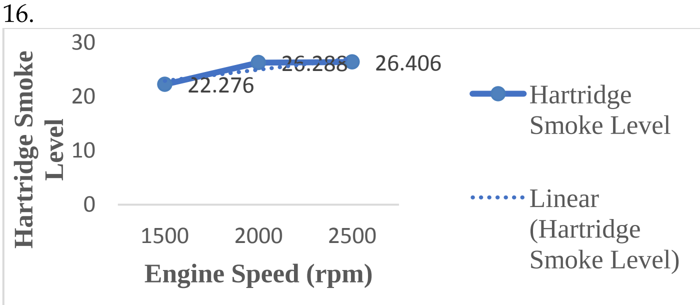


Fig. 15: Plot of Hartridge smoke level against engine speed

Emissions inspection procedures for diesel vehicles usually focus on smoke emissions. Smoke can be measured in a number of ways, including the Bosch and Hartridge methods or with a full-flow opacity meter [11]. High level of vehicular **pollution** in cities is reason behind the state-wide drive. In case of diesel vehicles, the smoke density is checked. If the density of smoke emitted by the vehicle is less than 65 Hartridge Smoke Unit (**HSU**), the vehicle is certified as non-polluting. While '0' **HSU** means smoke is invisible, 100 **HSU** means smoke is thick and opaque [12].

However, the result in fig. 15 showed that the engine speed greatly influenced the Hartridge smoke level.

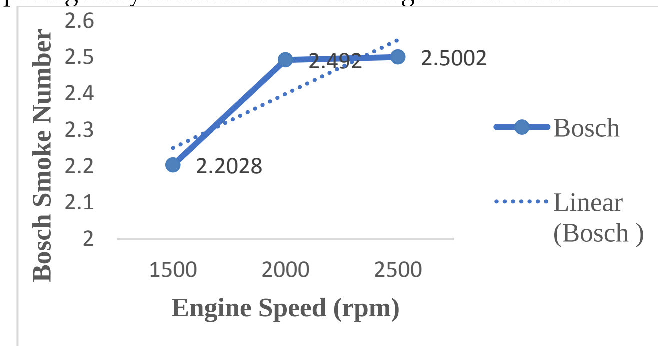
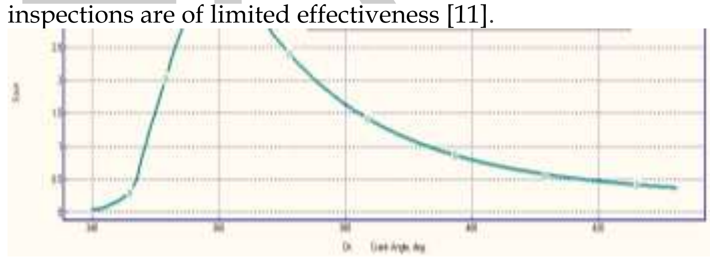


Fig. 16: Plot of Bosch smoke number against engine speed

The Bosch method provides an accurate measure of soot and other dark material in the smoke, but it responds poorly, if at all, to smoke particles that are not black.

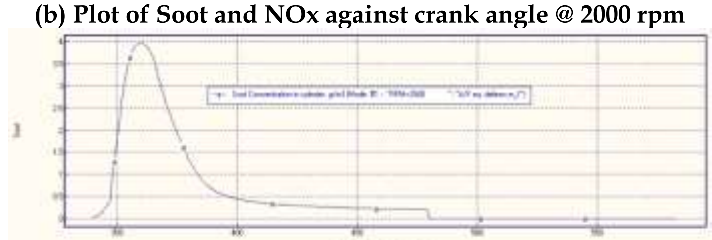
Soot concentration increases as the engine speed increases, because the higher the Bosch smoke number the more the soot concentration produced in the exhaust emissions.

Teams from the Department of Pollution Control and the Bangkok Police carry out roadside checks of carbon monoxide emissions from passenger cars, hydrocarbons from motorcycles, and smoke from diesel vehicles. The motorcycle hydrocarbon standard is 14,000 ppm, which means that most two-stroke motorcycles can pass the test. The smoke opacity limit is Bosch number 5, measured in a free-acceleration test. Commercial vehicles are also subject to annual checks of smoke opacity (for diesels) or carbon monoxide (for spark-ignition vehicles) as part of the licensing process, but these inspections are of limited effectiveness [11].



(a) Plot of Soot against crank angle @ 1500 rpm

(b) Plot of Soot and NOx against crank angle @ 2000 rpm



(c) Plot of Soot against crank angle @ 2500 rpm

Fig. 17: Effect of engine speed on soot concentration in cylinder

Fig. 17 above showed the plots of soot concentration in the cylinder against the crank angle at the selected engine speeds. The results obtained showed that the soot concentration increases slightly from 3.5, 3.7 to 4 g/m^3 at 1500, 2000 and 2500 rpm, respectively. Cranking also affect the soot concentration significantly. At 1500 rpm, the soot concentration was at the peak when the crank angle was 360^0 and immediately after that it dropped marginally as the cranking increases. However, at 2000 rpm, the soot concentration was at the peak when the crank angle was 366^0 before it started dropping marginally and the cranking increases. In addition, at 2500 rpm, the soot concentration was at the peak when the crank angle was 370^0 before it started dropping marginally and the cranking increases. This is an indication that the more the engine speed, the more the cranking.

4.5 Effect of Engine Speed on Engine Emissions

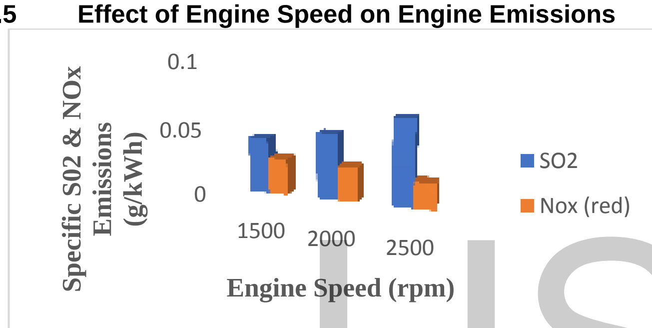
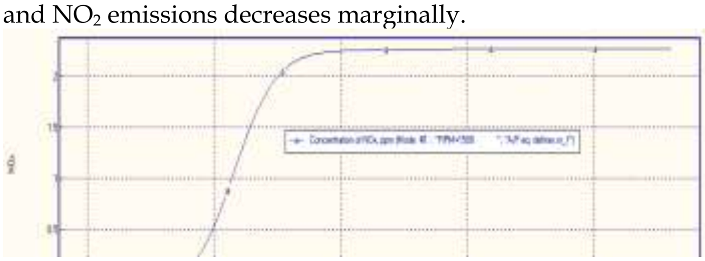


Fig. 18: Plot of specific SO_2 & NOx emission against engine speed

Fig. 18 showed that engine speed significantly affects the specific SO_2 and NOx emissions. The more the engine speed, the higher the specific SO_2 emissions but the reverse is the case in specific NOx emissions. This affirmed the research work carried out by [13] that as the engine speed increases, both NO and NO_2 emissions decreases marginally.



(a) Plot of NOx against crank angle @ 1500 rpm

(b) Plot of NOx against crank angle @ 2500 rpm

Fig. 19: Effect of engine speed on NOx concentration in cylinder

Fig. 17 (b) and Fig. 19 showed the plots of NOx concentration against the crank angle at the selected engine speeds. The results obtained showed that the NOx concentration decreases slightly from 2.56, 2.49 to 1.98 ppm at 1500, 2000 and 2500 rpm, respectively. These affirmed the research work carried out by [14] where NOx formation was found to be mainly caused by combustion chamber temperature, amount of O_2 and chemical reaction time in the combustion chamber. The paper concluded that the NOx emission decreases sharply at the engine speed below 2745 rpm but increases sharply between 2994 and 3500 rpm, then, beyond a speed of 3500 rpm, NOx emission decreases again. Cranking has no effect on the NOx concentration.

Table 3: Effect of engine speed on specific PM and CO_2 emissions

Engine Speed	PM (g/kWh)	CO_2 (g/kWh)
1500	0.96307	1718.8
2000	1.2949	2030.1
2500	1.7977	2740.1

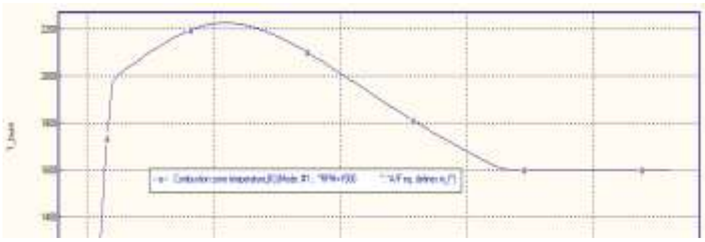
Table 3 showed the effect of engine speed on specific particulate matter (PM) and carbon dioxide emissions (CO_2). This result showed that both the specific PM and CO_2 emissions increases marginally as engine speed increases.

Table 4: Effect of engine speed on Combustion parameters

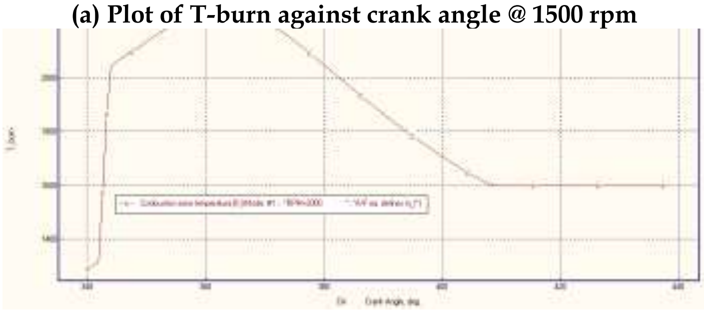
Engine Speed	A/F_eq	Ring_intn	ϕ_z (Delay), Deg.	ω_swirl @ R=23
1500	1.75	0.51798	92.4	6.6328
2000	1.75	0.85586	98.6	8.7409
2500	1.75	1.4096	243.2	10.474

Table 4 showed the effect of engine speed on combustion parameters. The result showed that at constant Air/Fuel Equivalent Ratio in the Cylinder (A/F_eq) all the selected engine combustion parameters like Ringing / Knock Intensity, MW/m^2 (Ring_intn); Combustion duration, deg. (ϕ_z) and Max. Air Swirl Velocity, m/s (ω_swirl) at cylinder R=23; increases spontaneously as engine speed increases. The result of the knock intensity showed that it is not ideal to have an engine with higher speed as it speeds up the rate at which ringing of the engine is required or the rate at which engine knocks. In addition, increase in engine speed contributes to the delay in combustion. Increase in engine speed also speed up the air inflow which is required in oxidizing combustion as observed in the air swirl velocity results. Meanwhile, all these were taken at constant environmental parameters like ambient temperatures and pressures.

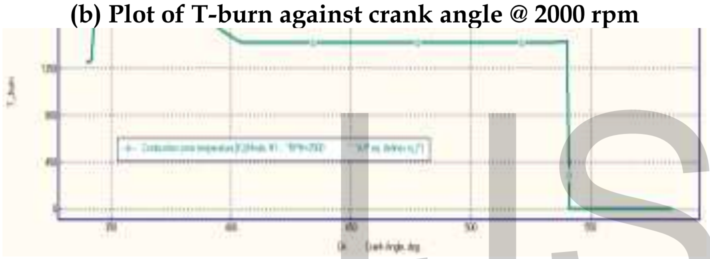
4.6 Effect of Engine Speed on Combustion Parameters



(a) Plot of T-burn against crank angle @ 1500 rpm



(b) Plot of T-burn against crank angle @ 2000 rpm



(c) Plot of T-burn against crank angle @ 2500 rpm

Fig. 20: Effect of engine speed on combustion zone temperature

Fig. 20 showed the plots of combustion zone temperature against the crank angle at the selected engine speeds. The results obtained showed that the combustion zone temperature increases slightly from 2205, 2215 to 2250 K at 1500, 2000 and 2500 rpm, respectively. These results signified that engine speed raises the rate at which the combustion chamber is heated over the cranking of the engine.

Table 5: Effect of engine speed on Turbocharging and Gas Exchange Parameters

Engine Speed	η_{vol}	ϕ_{sc}	η_{TC}
1500	0.33199	0.76931	0.7923
2000	0.38454	0.77398	0.79325
2500	0.33286	0.73164	0.79292

Table 5 showed the effect of engine speed on Turbocharging and Gas Exchange Parameters. The result showed that all the selected Turbocharging and Gas Exchange parameters like Volumetric Efficiency (η_{vol}); Turbocharger Efficiency (η_{TC}); and Coefficient of Scavenging (Delivery Ratio / η_{vol}) (ϕ_{sc}); increases spontaneously as engine speed increases from 1500 rpm to 2000 rpm, and these values dropped at 2500 rpm.

Table 6: Effect of engine speed on Engine Intake System

Engine Speed	hc_int, W/(m^2*K)	hc_int.p, W/(m^2*K)
1500	73.124	783.69
2000	71.953	988.95
2500	71.732	1181.9

Table 6 showed the effect of engine speed on engine intake system. The result showed that engine speed has great influence on the selected engine intake system parameters. The heat transfer coefficient in intake manifold (hc_int), W/(m^2*K) reduces as the engine speed increases. The heat transfer coefficient in intake port (hc_int.p), W/(m^2*K) increases as the engine speed increases.

Table 7: Effect of engine speed on Engine Exhaust System

Engine Speed	hc_exh, W/(m2*K)	hc_exh.p, W/(m2*K)	Sh = a*Tau/L
1500	90	306.09	23.381
2000	90	411.5	18.135
2500	90	430.76	15.077

Table 7 showed the effect of engine speed on engine exhaust system. The result showed that engine speed has great influence on the selected engine exhaust system parameters. The heat transfer coefficient in exhaust manifold (hc_exh), W/(m^2*K) remain unchanged because the exhaust system conditions would have been neutralized or attained thermal equilibrium state while in the exhaust manifold. However, the heat transfer coefficient in exhaust port (hc_int.p), W/(m^2*K) increases as the engine speed increases. In addition, the engine exhaust system heat transfer coefficient can be evaluated by utilizing a dimensionless number called, Strouhal number (Sh). The more the value of Sh, the more the rate of heat transfer in the exhaust port. Thus, the value of Sh increases as the engine speed increases.

5 CONCLUSIONS

The effect of engine speed on certain engine performance parameters cannot be underestimated when it comes to the production of diesel engines on a commercial scale. Thus, this article presents the application of Diesel-RK in investigating the performance characteristics of a single cylinder two-stroke turbocharged Hesselman's diesel engine with direct fuel injection. Three different engine operating speeds 1500, 2000 and 2500 rpm, respectively were utilized and their effects on certain engine performance parameters were investigated numerically. The highest overall specific fuel consumption and engine efficiency of 0.53342 kg/kWh and 0.1588, respectively were obtained at the 2000 rpm. More power was delivered at a speed of 2000 rpm as a result of the highest value of engine torque obtained at that speed. The cylinder pressure increases significantly as the engine speed increases at ivc (2.0451 bar), evo (3.7801 bar) and tdc (60.141 bar), respectively. This indicates that the pressure developed when the inlet valve closes increases as the piston translate from bdc to tdc on compression, to the value required for combustion and this value dropped as the exhaust valve opens.

However, the optimal values of corresponding cylinder temperature increase significantly at ivc (717.24 K), evo (1088.5 K) and tdc (1560.8 K), respectively at a speed of 2500 rpm. Also, the engine speed had appreciable impact on the heat exchange parameters, because the values of these parameters increases as the engine speed increases. On the ecological parameters, the Bosch number and Hartridge smoke level were significantly influenced as the engine speed increases which is an indication that the smoke level and the soot concentration in the engine increases as the engine speed increases. Engine speed also had greater influence on the combustion parameters. The results obtained signified that engine speed raises the rate at which the combustion chamber is heated over the cranking of the engine. Turbocharging and Gas Exchange parameters also increases spontaneously as engine speed increases from 1500 rpm to 2000 rpm, and the values dropped at 2500 rpm. The heat transfer coefficient in the intake manifold reduces as the engine speed increases while that of the intake port increases as the engine speed increases. Engine speed also influences the exhaust system parameters, as the heat transfer coefficient in exhaust manifold remain unchanged because the exhaust system conditions had attained thermal equilibrium state while in the exhaust manifold. However, the heat transfer coefficient in exhaust port increases as the engine speed increases. This result also brings about the increase in value of Strouhal number. It is now evident that engine speed is significant in the study of engine performance parameters and the best values of those parameters would be obtained at a speed of 2000 rpm.

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