

EFFECT OF VENTURI DIAMETER OF CARBURETOR ON PERFORMANCE OF SIX-STROKE 125 CC COMBUSTION ENGINE

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ABSTRACT

Performance of six stroke single power combustion engine affected by venturi diameter of carburetor is presented. This study is to begin on development of six-stroke single power of combustion engine (CE). Developing of this six stroke CE is based on addition of duration on both mass- and thermal diffusion, and on mixing time before the combustion. Evaluation of the six-stroke CE performance is done via enlarge venturi diameter of carburetor. Selected venturi diameters are 18 (mm) and 20 (mm). Fuel of the CE is gasoline-based using research octane number (RON) of 92. Interval of each ascending rotation speed is 400 (rpm), and observation was done using a constant throttle aperture 30 (%). The observed performances of the 125 (cc) six-stroke CE is then compared using a conventional four-stroke 125 (cc), as well. It informs that diameter 20 (mm) can increase average of about 21(%) on torsion, 21 (%) on power, 16 (%) on specific fuel consumption (SFC), and 23 (%) on thermal efficiency to that of the diameter 18 (mm). If compared to four-stroke CE, the six-stroke CE has lower in SFC and thermal efficiency, however, it has higher in average value of torsion, power, and engine speed, respectively about 15 (%).

Keywords : Venturi diameter, six-stroke, combustion engine, RON of 92

1. INTRODUCTION

Internal CE is widely used on motor vehicles, industries, and until electric power generator. It is due to the benefits of the engine on performances, economics aspects, durability, easy on operation, and no alternative competitor yet [1]

In present day, almost all of internal CE applies four-stroke Otto cycle. Main reason why the four-stroke is preferably to that of two-stroke Otto cycle is ability on saving fuel. Therefore, it can be built a deduction that the six-stroke CE will be saver in fuel consumption than four-stroke CE.

Siswanto, et al [2] has done on developing internal CE single power using six-stroke cycle based on addition of both mass- and thermal-diffusion. Mass diffusion means diffusion of mass fuel droplet to air volume in-cylinder, whereas thermal diffusion means diffusion of heat from in-cylinder walls to air-fuel mixture volume. The development presents a new concept with adding two strokes before combustion toward four-stroke CE, where the engine has two steps compression on each cycle. This situation increases possibility to both mass- and temperature-homogeneity of air-fuel mixture before burned to obtain a better power expansion. Hence, this six-stroke concept has a high potentiation as a new alternative engine.

Based on the explanation above, this study is aimed to precursor development on internal six-stroke CE based on adding duration of both mass- and thermal-diffusion and mixing. And then, this study did evaluation on performance of the six-stroke internal CE.

By considering a simply evaluated air-fuel supply variation, this study uses a conventional carburetor for generating air-fuel mixture. The carburetor uses venturi system to lift fuel and to mix it with air. To lift and to mix them includes the carburetor or venturi performances. Therefore, evaluation of the engine performances is representatively based on diameter of venturi hole. Selected venturi diameters are 18 (mm), as standard application for four-stroke engine, and 20 (mm), respectively. Modification of hole-

diameter of venturi until 20 (mm) is based on standard modification on four-stroke engine where allowed maximum enlargement diameter is until 2 (mm) [3]. The observed engines, i.e. four-stroke and six-stroke, in this present study use 125 (cc) in-cylinder of combustion volume.

Evaluated engine performances in this study are torsion of crankshaft, mechanical power, effective thermal efficiency, and effective specific fuel consumption.

The six-stroke CE in present study consists of 1) Suction stroke, where air-fuel mixture is sucked by piston to enter in-cylinder. 2) First compression-diffusion-mixing. Besides being mixed to each other between air and fuel, the fuel mass also be diffused to all volume of the air. Together with the mixing and the mass diffusing, the air-fuel mixture also be penetrated by thermal from in-cylinder walls. So in this stroke, the mixing and the both of mass- and thermal-diffusion is carried out while the piston compresses the mixture. 3) Expansion-diffusion-mixing stroke. It is analogue to the compression-diffusion-mixing but, the mixing and the both of mass- and thermal-diffusing is carried out while the piston expanding the mixture. 4) Second compression-diffusion-mixing. As ideal thermo-dynamicaly, this recompression stroke brings pressure P or temperature T sets back to the stroke number-2), however, actually as no-negative entropy change then the pressure and temperature in this stroke is suspected on higher value than in the first compression-diffusion-mixing stroke. 5) After combustion, this stroke is power or expansion stroke. The combusted gasses expand and push the piston to result mechanical power. 6) Exhaust stroke. Residual combusted gasses discharge to go out from the in-cylinder.

With the two diffusion-mixing strokes addition, it theoretically can be suspected that quality of the power or expansion of the six-stroke will have higher values than the conventional four-stroke.

Scheme of six-stroke single power idea is supported by some literatures that address to homogeneity of air-fuel mixture and of temperature of the mixture in-cylinder. Relatively low of the homogeneity especially on higher speed is caused by unadequately diffusion duration of mass fuel to the air [4],

and of thermal from in-cylinder walls to the mixture. Situation above causes local areas of unperfect combustion. The unadequate duration of diffusion also explained by Kovakh [5] that 3000 (rpm) engine speed provides duration only 0.02 (s), then it gains poor mixture quality. Other literature informed some local areas or islands of inhomogeneity of air-fuel mixture. It is explained by Rahman [6] via combustion simulation for hydrogen fuel using computationally fluid dynamics (CFD), as shown on figure 1.

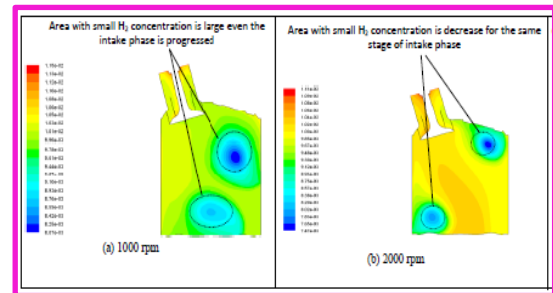


Figure 1. Islands in hydrogen mass fraction on 1000 (rpm), and 2000 (rpm)

Hydrogen is very light molecule, and it can easily diffuse and spread into in-cylinder. With the hydrogen character, however, it still occur some islands in the fuel mixture on each engine speed. It confirms that hydrogen fuel still needs a better handling to reduce the island area, moreover on carbureted gasoline.

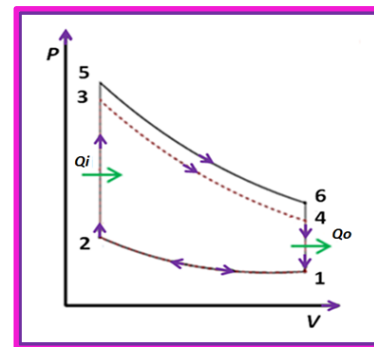


Figure 2. Prediction of pressure-volume or P - V diagram for six-stroke CE

Figure 2 shows it is due to compression stroke (1-2), expansion-diffusion stroke (2-1), and re-compression stroke (1-2), therefore, it makes possibility in adding duration of mass diffusion of fuel to air (1-2-1-2), and also of thermal diffusion into mixture (2-5) as supplement of heat input Q_i for combustion. It

is due to the additional strokes, theoretically resulted enhancement of area of net works (W_{net}) on P - V diagram (continuous line for six-stroke, while dashed line for four-stroke). It means there is additional heat in actual process as compensation of thermal diffusion from in-cylinder walls into air-fuel mixture. It can be suspected that higher reactants temperature before ignition can be obtained than reactants temperature on four-stroke. It also means expansion pressure each cycle on six-stroke has higher quality.

Compression ratio (r) of an engine is ratio of maximum volume (V_{max}) to minimum volume (V_{min}), or bottom death center volume (V_{BDC}) to top death center volume (V_{TDC}) of in-cylinder, as follows.

$$r = \frac{V_{max}}{V_{min}} = \frac{V_{BDC}}{V_{TDC}} \quad (1)$$

Average effective pressure (MEP), while if it operates to piston for power-expansion will result actual net works (W_{net}).

$$W_{net} = MEP \cdot (V_{max} - V_{min}) \quad (2)$$

Where the MEP its self is expressed as follows,

$$MEP = \frac{W_{net}}{V_{max} - V_{min}} = \frac{W_{net}}{V_{BDC} - V_{TDC}} \quad (3)$$

As above explanation, it is due to net works (W_{net}) increase, then obtained a highly average effective pressure (MEP). Therefore, by using the average effective pressure, it can be suspected that the six-stroke CE will result better maximum pressure, torque, and power to that of the four-stroke CE, in a given compression ratio.

Other advantage that may be obtained is power band of six-stroke CE will be wider. It is theoretically the CE tends to work at high crank rotation, therefore, range of torsion peak and of power peak will spread widely and evenly.

Based on the explanation, it can be concluded that this six-stroke single-power CE has highly potential to develop as new alternative for future CE technology.

2. METHOD

Method in this study is true experimental research, to observe influence of venturi diameter of carburetor towards performance of six-stroke single-power CE. Installation of observation is shown on figure 3.

Independent variable of this study is diameter of venturi carburetor and also crankshaft rotation, with interval 400 (rpm). Whereas, dependent variable of this study is performance of CE, namely torsion, power, effective thermal efficiency, and specific fuel consumption. Controlled variable for this study is aperture of 30 (%) throttle valve of carburetor. Fuel used to generate the power is gasoline based with research octane number (RON) 92 with trade mark Pertamina.

Apparatus and equipments applied in this research are a unit prototype six-stroke single-power CE, two sets carburetor assy, a unit Prony disk brake, a unit tachometer, a unit stopwatch, a unit measurement glass, a unit gas analyzer, a unit intake manifold, a unit fan for cooler, and a unit hot wire anemometer.

Performances evaluation processes is done by using Prony disk brake, where a free brake caliper is joined with a spring balance thus brake load of the disk can be measured via the balance. Atmospheric condition while observation is relative humidity at (ϕ) 69(%), room temperature at 24 ($^{\circ}$ C), and room pressure on (Ps) 949(kPa).

By combining the brake load and rotation speed of disk brake, it can be calculated the power and torsion of disk brake. And then, power and torsion of crankshaft are determined from transmission reduction of rotation speed between disk brake and crankshaft. Transmission reduction for gear-1, gear-2, gear-3, and gear-4, are respectively as (1:2.7), (1:18.7), (1:14.7), and (1:11.5).

Measurement of performance of CE is done by using observation procedure with ascending brake load. The brake load (kg) is obtained by controlling strength of breaking on dynamometer till gain desired rotation engine (rpm), with interval descending rotation of 400 (rpm) and fuel consumption on every 1 (ml). The above procedure is done respectively on each carburetors and engines.

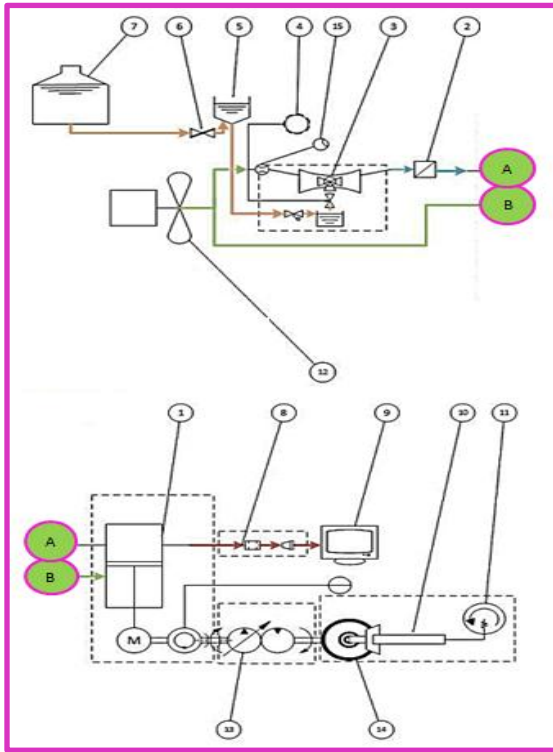


Figure 3. Sceme of installation of observation

The caption of the figure 3, where the apparatus is provided in detail, are : 1) six-stroke engine, 2) intake manifold, 3) carburetor assy, 4) throttle valve control, 5) glass for measurement, 6) fuel faucet, 7) fuel tank, 8) exhaust muffler assy, 9) gas analyzer, 10) dynamometer, 11) balance, 12) cooler fan, 13) power transmission, 14) disk brake, and 15) hot wire anemometer.

3. RESULTS AND DISCUSSIONS

- **Relation between crankshaft rotation and torsion**

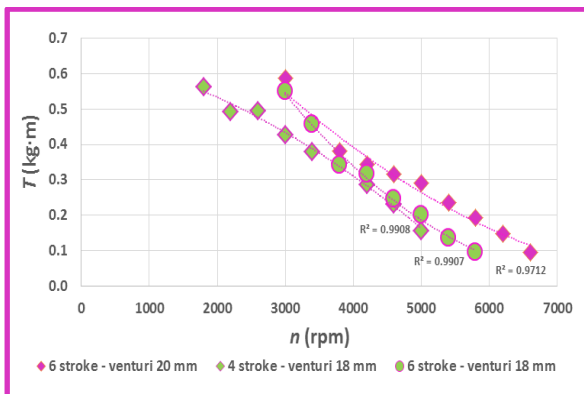


Figure 4. Relation between crankshaft rotation and torsion

Figure 4 depicts relation between crank rotation speed and torsion. It can be seen that torsion decreases with increasing crankshaft rotation. It is due to the crank rotation is descended by ascending the brake load till the desired rotation speed. It can be concluded that as increasing brake load then magnitude of the torsion also increases. Relation between the brake load F and the torsion T can be written as follows.

$$T = FxL \quad (4)$$

Where T is the resulted torsion (kg·m), F is magnitude of brake load (kg), and L is length of dynamometer arm (m).

Based on the equation (4), it can be seen that magnitude of the torsion is directly proportional to magnitude of brake load subjected to the shaft.

On the other side, as increasing crank rotation, moving of piston translation and acceleration also increase and it causes shear between piston and cylinder wall enhances, hence mechanical loss due to the shear increases.

Increasing crank rotation also causes moving of open-close of intake valve increases, then mixture mass of air-fuel entering in-cylinder each cycle decreases and it also makes effective pressure of resulted combustion decreases. It makes pressure energy pushing the piston to the before top death center (BDC) position in expansion stroke decreases, and it results magnitude of torsion also decreases.

The highest torsion of 0.68 (kg·m) using venturi diameter 20 (mm) can be achieved by the six-stroke engine at crankshaft rotation 3000 (rpm), whereas the lowest torsion of 0.07 (kg·m) can be achieved at crankshaft rotation 6600 (rpm).

In the given venturi diameter 18 (mm), the highest torsion of six-stroke engine is 0.65 (kg·m) at 3000 (rpm), whereas the lowest torsion is 0.08 (kg·m) at 6200 rpm. For four-stroke engine, the highest torsion is 0.63 (kg·m) at 1800 rpm, where as the lowest torsion is 0.10 (kg·m) at 5000 (rpm).

• **Relation between engine rotation and effective power**

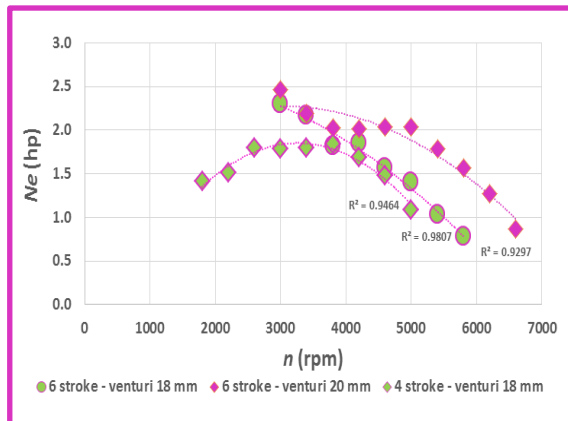


Figure 5. Relation between engine rotation and effective power

Figure 5 shows that shape of curve tends to a half of parabolics facing downward. Globally, decreasing effective power (Ne) is with increasing engine rotation. It is due to effective power is directly proportional to resulted torsion (T) and engine rotation (n), as suitable with this equation,

$$Ne = T \cdot \omega = \frac{T \cdot 2\pi \cdot n}{60 \cdot 75} = \frac{T \cdot n}{716,5} \quad (5)$$

Where Ne is effective power (hp), T is torsion of engine (kg·m), ω is angular velocity of crankshaft ($\text{rad} \cdot \text{s}^{-1}$), and n is crankshaft rotation (rpm).

In lower rotation, combustion processes at in-cylinder has much more duration time due to lower reciprocating of piston to that of in higher rotation. This condition causes more perfectly combustion and results larger power. However, while engine rotation go on enhancing, the decreasing torsion does not balance with increasing engine rotation and mechanical losses caused by increasing of reciprocating movement of piston. Besides it, possibility of unperfectly combustion is higher due to limitation of duration for combustion time. It causes decreasing of resulted energy and affects to decrease to effective power. The mechanism is suitable with equation (6), as follows.

$$Ne = Ni - Nm \quad (6)$$

Where Ne is effective power (hp), Ni is indicative power (hp), and Nm is losses of mechanical power (hp)

Based on the equation (6), effective power is generated by indicative power resulted by reaction of combusted gas pushing the piston to BDC, where part of this power is used to against a mechanical friction.

The largest power can be achieved by six-stroke engine is of 2.84 (hp) at 3000 (rpm) at venturi diameter 20 (mm), whereas the lowest power on the venturi diameter is on 0.74 (hp) at 6600 (rpm). Furthermore, it is for the venturi diameter 18 (mm) for six-stroke and four-stroke engine, the largest power is obtained by six-stroke engine as 2.73 (hp) at 3000 (rpm), whereas the lowest power is 0.6 (hp) at 6200 (rpm). For four-stroke engine, the largest power is 2.25 (hp) at 3800 (rpm), whereas the lowest power is obtained as 0.69 (hp) at 5000 (rpm).

• **Relation between engine rotation and specific fuel consumption**

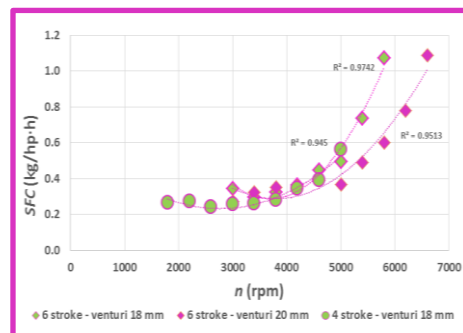


Figure 6. Relation between engine rotation and specific fuel consumption (SFC)

Figure 6 shows trend of effective specific fuel consumption SFC from starting rotation to ending rotation increases significantly. It is due to the higher rotation then number of cycle also higher, and it consumes much more fuel FC , where fuel flow velocity increases but amount of combusted fuel decreases, therefore resulted energy decreases as well.

Specific fuel consumption SFC is amount of fuel needed to result effective power of 1 (hp) for 1 (hour). The SFC is directly proportional to fuel consumption FC and inversely proportional to the resulted effective power Ne . It means in the given FC , the higher

N_e causes the lower SFC . Relation of the parameters are formulated as follows,

$$SFC = \frac{F_c}{N_e} \quad (7)$$

Where SFC is effective specific fuel consumption ($\text{kg}\cdot\text{hp}^{-1}\cdot\text{h}^{-1}$), F_c is fuel consumption ($\text{kg}\cdot\text{h}^{-1}$), and N_e is effective power (hp).

It for the six-stroke engine, result of observation shows that the lowest specific fuel consumption SFC of six-stroke engine is 0.20 ($\text{kg}\cdot\text{hp}^{-1}\cdot\text{h}^{-1}$) at low engine rotation, and gained at venturidiameter of carburetor 18 (mm). The SFC value increases until 1.45 ($\text{kg}\cdot\text{hp}^{-1}\cdot\text{h}^{-1}$). And then, for the venturi diameter 20 (mm), it is obtained SFC of 0.24 ($\text{kg}\cdot\text{hp}^{-1}\cdot\text{h}^{-1}$), and then increases with increasing engine rotation until 1.16 ($\text{kg}\cdot\text{hp}^{-1}\cdot\text{h}^{-1}$). Difference of the lowest value of the average can also be caused by fluctuation of brake load of six-stroke engine observation on range 3000 to 4600 (rpm), it is due to as ideally better SFC is obtained from venturi diameter 20 (mm).

Furthermore, on comparison SFC between four-stroke and six-stroke engine, the lowest value of the SFC is on four-stroke engine of 0.37 ($\text{kg}\cdot\text{hp}^{-1}\cdot\text{h}^{-1}$), whereas the value increases to achieve 0.76 ($\text{kg}\cdot\text{hp}^{-1}\cdot\text{h}^{-1}$).

• **Relation between engine rotation and effective thermal efficiency**

On figure 7, it can be seen that to higher engine rotation then generally effective thermal efficiency tends to decrease. It is due to the effective thermal efficiency is inversely proportional to value of SFC and of low heating value LHV of fuel. In this case fuel of pertamax has a constant LHV . Relation between effective thermal efficiency η_{te} , specific fuel consumption SFC and low heating value of fuel LHV_{bb} can be written as follows,

$$\eta_{te} = \frac{Q_e}{Q_b} \frac{632 \cdot N_e}{F_c \cdot LHV_{bb}} = \frac{632}{F_c / N_e \cdot LHV_{bb}} \quad (8)$$

$$\eta_{te} = \frac{632}{SFC \cdot LHV_{bb}} \times 100\% \quad (9)$$

Where 1 (hp) is equal to 632 ($\text{kcal}\cdot\text{h}^{-1}$), η_e is effective thermal efficiency (%), LHV_{bb} is low heating value of fuel ($\text{kcal}\cdot\text{kg}^{-1}$), F_c is fuel consumption ($\text{kg}\cdot\text{h}^{-1}$), and N_e is effective power (hp).

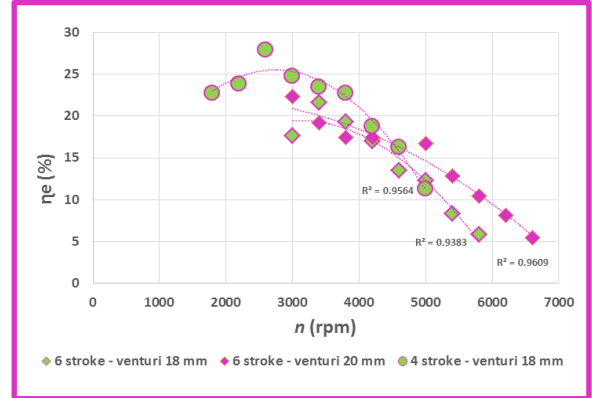


Figure 7. Relation between engine rotation and effective thermal efficiency

Observation on 18 (mm) of venturi diameter on six-stroke engine, the highest effective thermal efficiency is on 29.78 (%) gained at engine rotation of 3400 (rpm) and decrease till 4.12 (%) with increasing engine rotation. Observation on 20 (mm) of venturi diameter is obtained with higher average of effective thermal effective on 24.64 (%) at 3000 (rpm), and then decrease to about 5.15 (%). It is suspected due to higher fluctuation at maximum load while the observation of six-stroke engine, hence, although average of higher thermal efficiency is belonged to six-stroke engine with venturi diameter 20 (mm), the fluctuation causes less meticulous count of torsion, power, and thermal efficiency on low engine rotation.

On comparison four-stroke and six-stroke engine using 18 (mm) of venturi diameter, it is obtained that the highest efficiency on low engine rotation is on four-stroke engine, it is in 32.88 (%), and then decrease to 7.84 (%).

4. CONCLUSION

Firstly, on the performance of the six stroke engine, selection of venturi diameter of 20 (mm) and 18 (mm) give influences as follows,

1. Venturi diameter 20 (mm) gives enhancement on average torque value of

21(%), where the highest torque achieved on 0.68 (kgm) at engine speed 3000 (rpm).

2. Venturi diameter 20 (mm) gives enhancement on average effective power of 21 (%), where the highest effective power on 2.84 (hp) at engine speed 3000 (rpm).
3. Venturi diameter 18 (mm) gives decrement of *SFC* in average value of 16 (%), where the lowest *SFC* is obtained on the lowest engine speed, that is $0,20(\text{kg}\cdot\text{hp}^{-1}\cdot\text{h}^{-1})$.
4. Venturi diameter 18 (mm) gives enhancement on average effective thermal efficiency of 23 (%), where the highest thermal efficiency is on 29.78 (%) at engine speed 3400 (rpm).

Secondly, in the given venturi diameter, namely 18 (mm), application of the carburetor on six-stroke engine has both advantages and disadvantages performances if compared with the four-stroke engine. They are as follows,

1. Six-stroke engine has increase on torque value at average of 15 (%).
2. Six-stroke engine has increase on effective power at average of 15 (%).
3. A wider range of engine speed is obtained with increasing engine speed of 15 (%) for a given throttle position.
4. Specific fuel consumption of engine decreases at average of 14 (%).
5. Thermal efficiency of engine decreases at average of 12 (%).

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