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Evaluation Of Parameters Affecting The Performance Of Spark-Ignition Engine

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ABSTRACT

This paper focused on the performance of a spark-ignition (engine, which is affected by certain parameters at the engine speed and load variations. The parameters are evaluated at three throttle setting positions i.e. half-throttle, two-third throttle and full-throttle , which aims at achieving a good thermal efficiency and a high power output.The parameters include the brake power, measuring the actual power the engine can deliver, the brake mean effective pressure of the engine, specific fuel consumption, which determines the rate of fuel consumption, the brake thermal efficiency which measured the overall efficiency of the engine and the volumetric efficiency. In each throttle setting, all the parameters are evaluated from data obtained from the experiment; results and graph are obtained showing the effect of the parameters on the engine performance.Examination of the behaviour of the parameters showed that, all the parameters affect the performance of a spark-ignition engine and they all have severe effect on the engine performance. Thus, the engine can work efficiently at half-throttle position.

Keywords: Spark-ignition, thermal efficiency, fuel consumption, intake stroke and compression stroke

NOMENCLATURE

Symbol			Unit
A	–	Area	M^2
L	–	Length	m
Pi	–	Indicated mean efficiency pressure	N/m^2
N	–	Sped	rev/min
T	–	Torque	Nm
W	–	Weight	N
R	–	Radius	m
Bp	–	Brake power	w
Mf	–	Mass of fuel consumed	kg/s
Q_{net}	–	Calorific value	kJ/kg
Sfc	–	Specific fuel consumption	kg/kwh
V	–	Volume of induced air	m^2/S
Vs	–	Swept volume	m^3/S
Ma	–	Mass flow rate of air	kg/min
B	–	Atmospheric temperature	N/m^2
Ta	–	Atmospheric temperature	K
H	–	Height	m
X	–	No. of Cylinders	-
N	–	Efficiency	-
R	–	Specific gas constant	J/kgk

1.0 INTRODUCTION

The Internal combustion engine can be classified as the reciprocating and rotary engine. The reciprocating engine works on the piston principles whereby a piston slides back and forth due to high gas pressure developed in the cylinder. The thrust on the piston is transmitted by means of connecting rod to a crankshaft and the angular position of the crankshaft transmits this power to the flywheel and uses some of the power to return the piston to the top of the cylinder. Thus, the backward and forward displacement of the piston is referred to as the reciprocating motion of the piston.The rotary engine is simple, with fewer parts, which cost less and it substitutes a rotary member for the reciprocating piston and is easily balanced. One of its design is the wankel engine consisting of the rotating combustion chamber having a rotor which is constrained to a planetary motion about a gear. The rotor has three sealed sections and drives the output shaft at three times its speed to give one power stroke per shaft revolution.Most rotary engines have certain disadvantages - sealing problems, higher fuel consumption, does not last long. Hence, it is not widely used compared with the reciprocating engine.Generally, the internal combustion engine gives a high thermal efficiency and higher power output and has special importance in the field of land transportation (motor vehicle), industries and locomotives.

1.1 PRINCIPLE OF SPARK-IGNITION ENGINE

The spark-ignition engine requires an amount of fuel-air mixture to affect combustion in the cylinder. The composition of the mixture depends on the speed and load of the engine. This is compressed by an upward movement of the piston which initiates a spark to ignite he combustible mixture. Flame propagation is generated, burning most part of it, which leads to expansion of gases. The pressure development, forces the piston the bottom of its stroke and the force generated is transmitted to the crankshaft through the connecting rod and made to turn through one-half of a revolution at the bottom dead Centre. This cause the motion of the flywheel attached to the crankshaft rotating at high speed and thus producing mechanical used in motion.
However, there are two operating cycles of the engine viz: four stroke cycle and two- stroke cycle. The four-stroke cycle is completed in two revolution of the crankshaft i.e. four strokes of the piston. The order of these strokes is:

i) The intake stroke
ii) The compression stroke
iii) The power stroke
iv) The exhaust stroke

While the two-stroke cycle having the same cycle of events occurring in the four-stroke cycles, is completed in only two stokes of the piston corresponding to one crankshaft revolution. This is described by the position of the piston termed by its extremity. The upper position is termed the TOP DEAD CENTRE (TDC) and the lower position is termed the BOTTOM DEAD CENTRE (BDC).

1.2 METHODOLOGY

The method involved in data collection is through a laboratory experiment carried out on a heat engine (one cylinder) in order to evaluate the parameters affecting the engine performance at three throttle positions.Firstly, the throttle was set at half throttle position with a fixed compression ratio of 6:1 used throughout the experiment. The switch lever on the control box was raised and the engine starts to run with the dynamometer operating as an electric motor when the engine is fired, the lever was dropped down cutting off power from the dynamometer which then acted as the load and absorbed the power developed by the engine. In this case the engine now drives the dynamometer.The initial speed of the engine was set at l000rpm and the data receded were; The brake load indicated by the spring balance, the monometer reading, the time taken for the volume of petrol (17.5ml) to be consumed indicated by the stop watch and the atmospheric pressure and temperature indicated by the barometer and thermometer respectively. The procedure was repeated for other speeds at 200rpm intervals up to 1800 rpm and also for two-third (2/3) throttle position and full throttle position.

2.0 LITERATURE REVIEW

2.1 HISTORICAL BACKGROUND

Incentive for the development of practical engine has been widespread throughout history and the most applicable of the engines have been in transportation. The internal combustion engine in the 18^{th} century is as result of greater scientific understanding and a search by scientist for a substitute in coal power generation. The most readily available fuel for the earliest engine developed by Christian Huyghens (1980) and Denis Papin (1690) were gunpowder and coal. Gunpowder, which burns more rapidly than coal, became the earliest recoded internal combustion engine (Obert, 1969). Though an extensive literature of experimental findings on investigation carried out on internal combustion engine abound in the works of various authors. Thus/ it was concluded that the engine had no practical application because of failure to produce adequate power output.

It was not until the end of 18^{th} century that a practical internal combustion engine was develop. Robert Street, an Englishman invented such an engine In 1794, which was extremely clumsy in operation. Air was pumped into the cylinder, forcing the heavy piston halfway up its stroke, liquid fuel then flowed in and was ignited by the heated cylinder head. The resulting combustion and hating of gases drove the piston further up its stroke and after expansion; the gases were cooled and forced to contract thereby lowering the piston. An investigation carried out showed that the engine was too tedious because the operation of the engine was done manually (Heywood, 1988).

However, in 1857, two Italian: Barsanti and Mateuci, built a free-piston engine, which illustrated the motion of the piston, controlled by gas pressure. The piston was capable of sliding back and forth in a reciprocating motion during combustion in the cylinder. At the bottom stroke, it engages a ratchet connected to a shaft on repeated process, the same work done by (Heywood, 1988) showed that the engine was doomed to failure because of low power generation.

In 1860, Joseph Lenoir developed what was to become the first practical gas engine. It was build without compression and on the intake stroke, the air and gas were ignited by an electric spark. Due to this effect, the fuel consumption was very high and was concluded that the engine could not achieve a high power output due to low fuel economy (Obert, 1968).

Beau de Roches, a scientist, stated in 1862, that a sequence of operations must follow from the first phase of engine developed. He went further and devised cycle of which his principle could be applied, which are: induction of combustible mixture during the downward stroke of the piston, compression of mixture on the upward stroke, ignition and combustion of mixture on the upward stroke; expansion of gases during the downward stroke and burned mixture expelled at the upward stroke.

In 1867, Nikolaus Otto, a German engineer worked on this principle and developed a four-stroke circle engine which became highly successful and the name of the cycle of events occurring in the engine is known as the Otto cycle. This is typical of most spark-ignition engines, even today. The work on this engine shows that the performance of the engine produce higher speed, high power output and a good maximum efficiency (Heywood, 1988).

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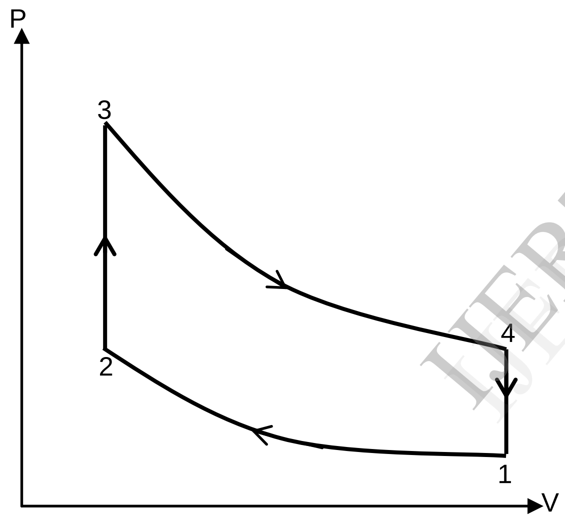
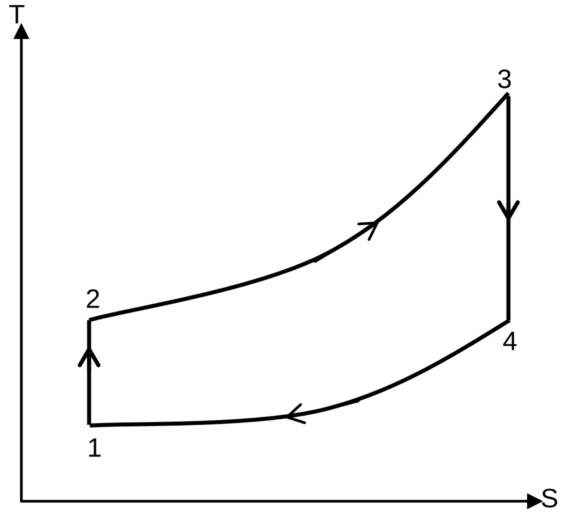
When oil was discovered in 1859, liquid fuels became available and with the invention of Otto engines and the Pneumatic tyre industry by John B. Dunkop in 1888, an automotive industry was set up Association of Licensed Manufacturers was formed and vehicles were manufactured to be used on roads.

2.2 THE AIR-STANDARD CYCLE

This is the theoretical cycle for the spark-ignition engine which involves four processes taking place simultaneously during combustion.
Isentropic compression
Heat addition at constant volume
Isentropic expansion
Heat rejection at constant volume

These process are represented on P-V and T-S diagram shown below;





T- S Diagram

T-S Diagram

The thermodynamic equations of the process are given by
$W_{12} = C_v (T_1 - T_2)$

$$Q_{23} = C_v (T_3 - T_2)$$

$$Q_{41} = C_v (T_4 - T_1)$$

Hence. The thermal efficiency of the air-standard cycle can be written as;

$\eta = \frac{Network}{Heat\ supplied} \frac{Q_{23} - Q_{41}}{Q_{23}}$ …………………………………………… (2.3.1)

$$\eta_1 = \frac{C_V(T_3 - T_2 - C_V(T_4 - T_2)}{C_V(T_3 - T_2)}$$

Therefore,

$$\eta_1 = 1 - \frac{T_4 - T_1}{T_3 - T_2}$$

Since process 1 – 2 and 3 – 4 are Isentropic, it can be shown that;

$\frac{T_2}{T_1} = \left[\frac{V_1}{V_2}\right]^{y-1} and \frac{T_3}{T_2} = \left[\frac{V_4}{V_3}\right]^{y-1}$ ………………………………………………………… (2.3.2)

And

$$V_1 = V_4, V_2 = V_3$$

And

$$\frac{T_2}{T_1} = \frac{T_3}{T_4}\ or \frac{T_4}{T_1} = \frac{T_3}{T_2}$$

Hence,

$$\eta_1 = 1 - \frac{T_1}{T_2}$$

Or

$$\eta_1 = 1 - \frac{1}{[T_2/T_1]} \quad \ldots\ldots (2.3.3)$$

From equation (2.3.1)

$$\frac{T_2}{T_1} = \left[\frac{V_1}{V_2}\right]^{y-1}$$

Where

$$\frac{V_1}{V_2} r_v \; (Compression\,ratio)$$

Therefore,

$$\eta_1 = 1 - \frac{1}{[r_v]^{y-1}} \quad \ldots\ldots (2.3.4)$$

2.3 FACTORS AFFECTING THE PERFORMANCE OF S,I ENGINE

The air-standard Otto cycle showed that with an increase in compression ratio, the efficiency increase together with the brake thermal efficiency of S.I. Engines.

However, it has an upper limit to which the compression ratio can be applied, noting that liquid compressed in the cylinder is a mixture of fuel and air. The temperature of the combustible mixture increases on compression and if the compression ratio is too high, it is possible for self-ignition to occur before spark ignition. But if the compression ratio is sufficiently low, it produces a much higher efficiency for which the engine can withstand and pre-ignition is avoided.

3.0 THEORETICAL ANALYSIS

3.1 PERFORMANCE CRITERIA

An engine is selected to suit a particular application, usually based on its power and speed characteristics. These characteristics are however, governed by some other quantities when running at different loads. These include mixture strength, ignition timing, throttle opening, compression ratio and air-fuel ratio. Evolution of the criteria are derived from these quantities from the testing of the S.I engine.

3.1.1 Indicated power (ip)—Defined as the rate of work done by the gas on the piston as evaluated from the indicated diagram obtained from the engine, considering one engine cylinder,

Work done per cycle = Pi x A x L........ (3.1)

A—Area of piston (m2)

L—Length of stroke (m)

Therefore,

Power output/unit time = work done per cycle x cycle/minute

Or ip=PiAL x (cycle/unit time).. .... (3.2)

The number of cycles per unit time depends on the type ofengine

For four -stroke engine, the number of cycles / unit time is $\frac{N}{2}$

For two-stroke, the number of cycles / unit time is N Where N is the engine speed (rev/min) And x stands for number of cylinders in the engine Therefore,

$$\left.\begin{aligned} IP &= \frac{PiALNx}{2} \text{ [for four-stroke engine]} \\ iP &= PiALNx \text{ [for two-stroke engine]]} \end{aligned}\right\} \ldots\ldots\ldots\ldots\ldots\ldots \quad (3.3)$$

3.1.2 Brake power (bp)-This is the measured output of the engine. The engine is connected to a brake or dynamometer, which can be loaded in such a way that the torque exerted by the engine can be measured. The brake power is given by the engine can be measured.

The brake power is given by

$$\boldsymbol{bp \quad - Z\pi NT} \ldots\ldots\ldots\ldots\ldots\ldots \quad (3.4)$$

Where T is the torque evaluated as

T - WR (Nm)

W - Load (kg) x Acceleration due to gravity (m/s^2)

R - Brake arm of the dynamometer (m)

3.1.3 FRICTION POWER AND MECHANICAL EFFICIENCY

The difference between the ip and bp is the friction power, and is that power required to overcome the frictional resistance of the engine parts.

$$Fp \quad = \quad ip - bp \ldots\ldots\ldots\ldots\ldots\ldots \quad (3.5)$$

Mechanical efficiency of the engine is defined as

$$\boldsymbol{\eta_m} \quad = \frac{\boldsymbol{bp}}{\boldsymbol{ip}} \ldots\ldots\ldots\ldots\ldots\ldots \quad (3.6)$$

Or

$$\boldsymbol{\eta_m} \quad = \frac{\boldsymbol{bmep}}{\boldsymbol{imep}} \ldots\ldots\ldots\ldots\ldots\ldots \quad (3.7)$$

3.1.4 Brake Mean Effective($bmep$). Thermal Efficiency And an specific fuel consumption

The brake mean effective pressure is defined as the theoretical constant pressure, which can be exerted during each power stroke of the engine to produce power equal to the brake power.

From equation (3.6) the bp of an engine can also be written as;

$$\boldsymbol{B_p \qquad \eta_m \quad \times \quad \frac{PIALNx}{2}} \ldots\ldots\ldots\ldots\ldots\ldots \quad (3.8)$$

From equation (3.7)

$$\boldsymbol{\eta_m} \quad = bmep \ldots\ldots\ldots\ldots\ldots\ldots \quad (3.9)$$

$imep$ - indicated mean effective pressure Substituting equation (3,9) into (3.8)

$$B_p = \frac{bmepALNx}{\mathbf{2}}$$

Therefore, the brake mean effective pressure is given as

$bmep \quad \frac{2bp}{ALNx} \quad (N/M2)$.. (3.10)

The power output of the engine is obtained from the chemical energy of the fuel supplied. The overall efficiency of he engine also known as the brake thermal efficiency η_{bt}

$$\eta_{bt} = \frac{\textbf{Brake Power}}{\textbf{Energy Supplied}}$$

Or

$\boldsymbol{\eta_{bt}} = \frac{\boldsymbol{bp}}{\boldsymbol{m_f \times Q_{net}}}$ .... (3.11)

Where,
M_f - The mass of fuel consumed per unit time (kg/s)

$\boldsymbol{Q_{net}}$ - The net colorific value of the fuel (KJ/kg)
$\eta_{bt} = \eta_{it} \times \eta_m$

Also, the indicated thermal efficiency is defined as

$\eta_{it} = \frac{ip}{m_f \times Q_{net}}$ .... (3.12)

SPECIFIC FUEL CONSUMPTION (Sfc)
Specific fuel consumption is the mass flow rate of fuel consumed per unit power output and is a criterion for economical power production.
i.e.

$Sfc = \frac{m_f}{bp} \times 3{,}600\ (kg/kwh)$ .... (3.13)
Where,

m_f - mass of fuel consumer per hr
3.1.5 Volumetric Efficiency
The power output of an internal combustion engine depends directly upon the amount of charge which can be inducted into the cylinder. Therefore, the volumetric efficiency is he ratio of the volume of air induced, measured at the free air conditions to the swept volume of the cylinder.

$\boldsymbol{i.e. \eta_v} = \frac{\boldsymbol{V}}{\boldsymbol{V_5}}$.. (3.14)
Where,

$$\boldsymbol{V - The\ volume\ of\ air\ induced}$$

$$\boldsymbol{V_{5-}\ Swept\ volume}$$

4.0 DATA PRESENTATION AND ANALYSIS

4.1 DATA PRESENTATION 4.1.1 INTRODUCTION

The apparatus used for the experiment is the Peter Petrol engine at full and part throttle positions. The engine was developed on the principles of the four stroke Otto cycles, which allows a limited quantity of fuel and an adequate combustion chambers space combustion.

The engine capacity is adequate large in additional to a small cylinder size and provide a consisted performance for a good thermal efficiency and power output at its speed and load variations. The engine speed can be operated up to 2,000 r.p.m for spark-ignition having a fixed compression ratio.

4.1.2 ENGINE DESCRIPTIVE (PETTER SPARK-IGNITION ENGINE

The S.I engine is a single-cylinder, water cooled, four stroke unit of $468.67cm^3$ swept volume, having a bore of 84.988mm

4.1.3 EXPERIMENTAL PROCEDURE

To evaluate the parameters affecting the performance of S.Iengine the following procedures are used:

The throttle was set at half-throttle position with a fixedcompression ratio of 6:1 used throughout the experiment. Theswitch lever on the control box was raised and the enginestarts to run with the dynamometer operating as an electricmotor. When the engine is fired, the lever was dropped downcutting-off power from the dynamometer which then acted asthe load and absorbed the power developed by the engine. Inhis case, the engine now drives the dynamometer.

The initial speed of the engine was set at 1,000 r.p.m and the following data were recorded. The brake load, indicated by the spring balance, the manometer reading, the time taken for the volume of petrol (17.5ml) to be consumed, indicated by the stop watch and the atmospheric pressure and temperature indicated by the barometer and thermometer respectively. The procedure was repeated for other speeds at 200 r.p.m Intervals to 1,800 r.p.m and also for two third (2/3) throttle position and full throttle position.

4.1.4 RESULT AND CALCULATIONS

4.1.4 HALF THROTTLE POSITION

TABLE 4.1.4 (a)

Speed rev/m in	Load (kg)	Manometer reading (m)	Time (s)	Atmosphere temp. (oC)
1,000	8.7545	0.0122	60.0	29.0
1,200	8.6411	0.0135	59.0	29.0
1,400	8.4596	0.0148	57.0	29.0
1/600	8.1421	0.0174	54.0	29.0
1,800	7.8473	0.0177	49.8	29.0

TABLE 4.1.4 (b)

Speed rev/min	Mass flow rate of fuel (kg/s)	Torque (Nm)	Brake power (KW)	bmep (bar)	Bake thermal efficiency	Sfc (kg/kwk)	Volumetric efficiency
1,000	2.1583×10^{-4}	27.46	2.87	6.61	30.4	0.270	32.6

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1,200	2.18944x10^{-4}	27.14	3.41	7.3	35.6	0.230	28.4
1,400	2.2719 x10^{-4}	26.37	3.89	7.13	39.2	0.210	25,6
1,500	2.398 x $l0^{-4}$	25.58	4.28	6.84	41.0	0.201	24.0
1,800	2.600 x 10^{-4}	24.65	4.65	6.61	40.8	0.209	21.7

From the table above, graph of torque, volumetric efficiency brake thermal efficiency, brake power, bmep plotted against speed for half-throttle position is shown in fig. 4.4.1

4.1.5 Two-third (2/3) throttle position

The procedure was repeated for (2/3) throttle position and hresult are as shown in the table below

Table 4.1.5 (a)

Speed rev/m in	Load (kg)	Manometer reading (m)	Time (s)	Atmosphere temp. (oC)
1,000	9.5708	0.0207	49.5	29.0
1,200	9.7069	0.0247	43.8	29.0
1,400	9.5708	0.0279	39.7	29.0
1,600	9.2986	0.0332	38.2	29.0
1,800	8.7316	0.0385	37.8	29.0

From the evaluation of the parameter, the result of are tabularin the table below.

From the table above, graphs of torque, volumetric efficiency, brake thermal efficiency, brake power, bmep plotted against sped for two-third throttle position is shown in fig. 4.1.5

4,1,6 FULL THROTTLE POSITION

The procedure was repeated for full throttle position and the results are as shown in the tables below:

Table 4.1.6 (a)

Speed rev/mm	Load (kg)	Manometer reading (m)	Time (s)	Atmosphere temp. (°C)
1,000	9.8880	0.0225	43.0	29.0
1,200	9.7524	0.0279	38.2	29.0
1,400	9.7524	0.0371	35.6	29.0
1,600	9.5256	0.0470	34.8	29.0
1,800	8.7545	0.0559	33.5	29.0

From the evaluation of the parameters, the results are tabulated in the table below:

Table 4.1.6 (b)

Speed rev/min	Mass flow rate of fuel (kg/s)	Torque (Nm)	Brake power (KW)	BMEP N/m^3 X 10^5	Bake thermal efficiency	Sfc (kg/kwk)	Volumetric efficiency
1,000	3.012 x 10^{-4}	31.06	3.25	8.34	24.70	0.33	43.60
1,200	3.39 x $10^{'4}$	30.63	3.85	8.22	26.00	0.317	41.00
1,400	3.638 x 10^{-4}	30.63	4.49	8.22	28.00	0.292	40.30
1,400	3.721 x 10^{-4}	30.27	4.25	8.25	29.50	0.268	40.30
1,800	3.866 x 10^{-4}	27.50	5.18	7.38	30.70	0.268	38.45

Table 4.1.7 Result f Evaluated Torque at the Three Throttle Position

Speed (re/min	**Torque (Nm)**		
	1/2 Throttle	**2/3 Throttle**	Full Throttle
1,000	27.46	30.6	31.06
1,200	27.14	30.49	30.63
1,400	26.37	30.06	30.63
1,600	25.58	29.21	29.92
1,800	24.65	27.43	27.50

4.2 DATA ANALYSIS

The experiment carried out to evaluate the parameters demonstrates the effect each has on the engine performance and showing the relationship that exist between them. Three throttle positions (half, two-third, full) are varied with variation in speed during the experiment.

The values obtained from the evaluation of the brake power shows the maximum power output attained on the engine as represented in figure 4,1.4 for al half throttle position. This figure also represents the variation of the other parameters, which are plotted against speed/ and shows the typical curves obtained from the evaluation of the parameters. As the engine speed increases, the brake power also increase until it reaches its maximum value of 4.56 KW at the speed of 1,800 r.p.m. This is also true for the two-third and full -throttle positions as represented in the graphs of figure 4.1.5 and figure 4.1.6 respectively. Except that, the maximum value attained at that speed is 5.18 KW at full throttle position.

The brake means effective pressure as represented on the same graph of figure 4.1.4 at half throttle position, shows the pressure developed with an increase in engine speed. The optimum pressure attained is 7.13 x10 N/m^2 and then dropped to a lower value of 6.61 x $10^{-4}KN/m^2$ at the maximum speed. This is due to the pressure generated at the end of the exhaust stroke. The result also follows for the two-third and full-throttle position as shown on the same graph of the 4.1.5 and figure 4.1.6, The torque-speed relationship represented on the same graph at half-throttle position shows that as the speed increases, the torque developed by the dynamometer acting as load on the engine decreases. This resulted to friction lost and pumplost on the engine, as the torque is proportional to the load. For the two-third and full throttle positions, the same result holds.

The specific fuel consumption shows the relationship with the engine speed as shown on the curve obtained in the graph at half-throttle position. The Sfc decreases to 0.209kg/kwh at 1,600 r.p.m and as h speed increases to its maximum at 1,800 r.p.m the consumption rate increases slowly to 0.21 kg/kwh. . Thus, on comparison with the theoretical curve for a real engine, the curves are identical and is also true for two-third and full-throttle position.

The volumetric efficiency of the engine attained for a full-throttle position is higher (43%) as compared to the other throttle positions. This decreases as the engine speed increases to its maximum value of 1,800 r.p.m. Since the engine is a single-cylinder the highest value obtained is not the same with the multi-cylinder engine having an efficiency of 73%.

The brake thermal efficiency a half-throttle position increases to 42% and then dropped slightly as the sped increases. This is due to overheating in the combustion chamber at the optimum engine speed of 1,800 r.p.m as shown on the graph of figure 4.1.4. The two -third and full throttle positions also shows the same result.

5.0 CONCLUSION AND RECOMMENDATIONS

5.1 CONCLUSION

From the experiment carried out, it can be concluded that the engine parameters affecting the engine performance at the three throttle positions has effect on the engine. The brake power, brake thermal efficiency increase, while the torque volumetric efficiency, specific fuel consumption and brake means effective pressure decreases as the engine speed increases for half throttle position, two-third throttle position and full throttle. As it can observed from the resulting graph for the half-throttle position, a good volumetric efficiency and brake power output is achieved. The maximum valued attained is not exceeding the performance limit of the engine at it's maximum speed. Hence, the engine can perform effective at this throttle position as compared to the full-throttle positions, where the power output is higher than the actual rating of the engine.

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