International Research Journal of Engineering and Technology (IRJET) e-ISSN: 2395 -0056
Volume: 03 Issue: 06 | June-2016 www.irjet.net p-ISSN: 2395-0072

EXPERIMENTAL STUDIES ON THE PERFORMANCE OF I C ENGINES USING EXHAUST BLOWER

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Abstract - The present study an experimental work is carried out for the performance parameters of the I C engine using suction blower at the exhaust with variable frequency drive. The study incorporates the device consisting of a Design and the fabrication of a suction blower based on requirements and the variable frequency drive is incorporated for obtaining the optimum performance for the different speed of the suction blower. The result shows the improvement in the break power, specific fuel consumption, and finally increasing in the break thermal efficiency of the engine with the exhaust blower when compared with the normal 4 stroke diesel engine.

Key Words: IC Engines, suction blower, break power, specific fuel consumption, and thermal efficiency.

1.INTRODUCTION

The internal combustion engine is a heat engine that converts chemical energy in a fuel into mechanical energy, usually made available on a rotating output shaft. Chemical energy of the fuel is first converted to thermal energy by means of combustion or oxidation with air inside the engine. This thermal energy raises the temperature and pressure of the gases within the engine, and the high-pressure gas then expands against the mechanical mechanisms of the engine. This expansion is converted by the mechanical linkages of the engine to a rotating crankshaft, which is the output of the engine. The crankshaft, in turn, is connected to a transmission and/or power train to transmit the rotating mechanical energy to the desired final use. For engines this will often be the propulsion of a vehicle i.e., automobile, truck, locomotive, marine vessel, or airplane engines to drive generators or pumps, and portable engines for things like chain saws and lawn mowers. Most internal combustion engines are reciprocating engines having pistons that reciprocate back and forth in cylinders internally within the engine. Other types of IC engines also exist in much fewer numbers, one important one being the rotary engine. These engines will be given brief coverage. Engine types not covered by this book include steam engines and gas turbine engines, which are better classified as external combustion engines i.e., combustion takes place outside the mechanical engine system. Also not included in this book, but which could be classified as internal combustion engines, are rocket engines, jet engines, and firearms.

Reciprocating engines can have one cylinder or many, up to 20 or more. The cylinders can be arranged in many different geometric configurations. Sizes range from small model airplane engines with power output on the order of 100 watts to large Multi-cylinder stationary engines that produce thousands of kilowatts per cylinder. There are so many different engine manufacturers, past, present, and future, that produce and have produced engines which differ in size, geometry, style, and operating characteristics that no absolute limit can be stated for any range of engine characteristics i.e., size, number of cylinders, strokes in a cycle, etc. Early development of modern internal combustion engines occurred in the latter half of the 1800s and coincided with the development of the automobile. History records earlier examples of crude internal combustion engines and self-propelled broad vehicles dating back as far as the 1600s.

This invention relates to internal combustion engines used in combination with exhaust turbines and consists in this, that besides the charging blowers driven by exhaust turbines a plurality of piston pumps driven by the internal combustion engine act as additional charging blowers. By the sub-division of the piston blowers into a plurality of pumps, the delivery of scavenging and charging air to the cylinders can be made more uniform. This is more particularly of importance, when piston blowers are used in conjunction with blowers driven by the exhaust with the object of providing sufficient scavenging and charging air when machines of this kind are being started and

when they are running at low loads, as in this case the blowers driven by exhaust turbines are not very efficient owing to the small energy in the exhaust. This provides the further possibility, when the load of the exhaust turbines increases and they have to deliver more power, of disconnecting the piston blowers at least in part, thus enabling the internal combustion engine to give more power to the outside and obtaining a better efficiency. It also becomes possible to co-ordinate to each combustion cylinder its own piston pump, thus providing a saving in piping and reducing pressure losses. The piston pumps may suitably be arranged in the extension of the axis of the combustion cylinders. In single-acting engines the lower side of the piston may be used as the pump space. In double-acting engines the crosshead guide may be formed into a pump. The air or the combustion mixture delivered may be conveyed through common ports into the combustion cylinders. In the supply pipes of the piston, pumps and of the turbine driven blowers leading to the combustion cylinders, stop valves may be provided. The piston pumps may be provided with means for interrupting their delivery, such that at certain loads they will run idle, that is they will not perform any pumping work. The said interruption of the delivery may be effected by external means or automatically. As when the turbine driven blower commences to deliver heavily, it is preferable to interrupt the delivery of the piston blowers, at least partially this may be affected by means of the pressure of the turbine driven blower. The pipes leading to the piston blowers maybe so arranged that they can draw by suction selectively from the atmosphere or from the delivery pipe of the turbine driven blower.

2. LITERATURE REVIEW

In two-stroke engines, burned gases have still some energy left in them when the exhaust port opens. The burned gases have to get past the ports to enter the exhaust pipe and from thereon the flow is governed by laws pertaining to pipe flow. The inlet of the charge into the cylinder depends very much on the trapped compression ratio and the delivery ratio of the engine which in turn depends on the geometry of the port openings, the roughness factors and the coefficient of discharge through the port openings. Many researchers have focused their attention to this problem of fluid flow behavior inside the cylinder after the uncovering of the exhaust port in a crank case scavenged two-stroke engine peculiar to a two-stroke engine. In the following sections the literature on performance enhancement in general and experimental as well as mathematical methods to study and improve the complex phenomenon of gas exchange process in particular are outlined. Some of the methods to remove the basic deficiency of short-circuiting fresh charge into exhaust port in a crankcase scavenged naturally aspirated two stroke engine by new approaches in technology are also outlined for comparison.

2.1 EXPERIMENTAL WORKS

Tobis (1994) have used in-cylinder gas composition analysis to measure trapping and scavenging efficiencies of an Orbital prototype crankcase scavenged 1.2L, 3-cylinder two-stroke 84mm bore x 72mm stroke engine with a geometric compression ratio of 10.5 and effective compression ratio of 6. It has been reported that the cylinder contents were sampled before and after combustion process by an electromagnetic exhaust sampling valve. It has also been reported that the scavenging process following a motored cycle was less effective than one following a firing cycle. It has been concluded that the predictions from a simple semi-empirical mathematical model based on modification of non-isothermal perfect-mixing model matched well with experimental findings.

Ekenberg (2001) have studied the effect of injecting pure air late in the compression stroke using Particle Image Velocimetry (PIV) in an engine fitted with two optical access ways to the cylinder. It has been reported that during the PIV measurements only air was let in through the injector and that for recording pressures in the fired engine port fuel injection was used along with air injected through the direct injector. It has been further reported that the experiments were also used to study the concept of High dilution Stoichiometric Combustion Concept (HDSCC) in which large amounts of exhaust gas recirculation (EGR) was used. It has been concluded that the air blast from the injector resulted in increased in-cylinder velocities at the time of ignition.

2.3 Mathematical Approach

Basha (2009) have investigated the progress of CFD in the area of in-cylinder flow field analysis, turbulence and spray characteristics of engines. It has been reported that re-normalized group (RNG) model was the best turbulence model for engine analysis. It has been stated that the capability of KIVA CFD code in resolving complex geometries was limited and that commercial codes such as STAR-CD, FLUENT etc. possessed superior mesh generating capabilities. It has been concluded that low load experimental and low speed simulated data on combustion models were not available for reference.

Verhelst and Sheppard (2009) have investigated the necessity of sub-models in combustion analysis of engines. It has been stated that choice of multi-zone and multi-dimensional models were dictated by the

International Research Journal of Engineering and Technology (IRJET) e-ISSN: 2395 -0056
Volume: 03 Issue: 06 | June-2016 www.irjet.net p-ISSN: 2395-0072

application. It has been further stated that sub-models were necessitated by additional information required to close the equations for cylinder pressure and temperature. It has been concluded that a unified framework used for comparing all sub-models on the same basis was essential with particular emphasis on turbulent combustion models to provide clarity to combustion analysis.

3. EXPERIMENTAL SPECIFICATION

3.1.Selection of forced draft fan or Blower:

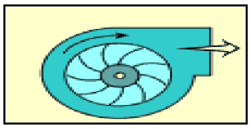


Fig 3. 3: **Forward curved air blower.**

$$\text{Actual blower Power} = \frac{\text{Theoretical blower Power}(P_b)}{\text{Efficiency }(\eta)} \quad \text{-------------------- (3.3)}$$

We have

$$\text{Theoretical blower Power}(P_b) = \frac{\rho_a Q_a g\, He}{60 \times 1000} \text{ KW} \quad \text{------------------------ (3.4)}$$

Where,

P_b = Theoretical blower power (kW)

Q_a = Flow capacity of Air (m³/min)

ρ_a = density of air (kg/m³)

g = gravity (9.81 m/s²)

H_e = equivalent head (m)

The air flow rate is calculated by using the relation.

$$Q_a = \frac{a_1 X a_2}{\sqrt{a_1^2 - a_2^2}} \sqrt{2gH_e}$$

Where ,

a1 = Minor flow area in m2

a2 =Major flow area in m2

The equivalent head is calculated by

$$H_e = \left[\frac{\rho_w}{\rho_a} - 1\right] h_w$$

h_w= manometer reading in mm of water column

= 75mm of water column

$$H_e = \left[\frac{1000}{1.18} - 1\right] 0.075$$

H_e = 63.48m

$$H_e = \left[\frac{\rho_w}{\rho_a} - 1\right] h_w$$

h_w= manometer reading in mm of water column

= 75mm of water column

$$H_e = \left[\frac{1000}{1.18} - 1\right] 0.075$$

H_e = 63.48m

Also we have

d_1 = 80 mm

d_2 = 49 mm

$$a_1 = \frac{\pi d_1^2}{4} = \frac{\pi}{4} x\ (0.08)^2$$

= 0.05029 m²

$$a_2 = \frac{\pi d_2^2}{4} = \frac{\pi}{4} x\ (0.049)^2$$

= 0.001887 m²

Thus,

$$Q_a = \frac{0.05029\ X\ 0.001887}{\sqrt{0.05029^2 - 0.001887^2}} \sqrt{2x\ 9.81x63.48}$$

= 4.308 m³/min

But we have the maximum flow in our experimental conduct in order to maintain the L/G ratio up to a maximum of 1:5 is,

ρaQa = 1.18 x 4.308 kg/min

ρaQa = 5.0834 kg/min

≈ 5 kg/min

Substituting the values ofρaQa, g and h in the equation (3.4).we have,

$$\textbf{Theoretical blower Power(Pb)} = \frac{\rho_a Q_a g\, He}{60 \times 1000} \textbf{ KW}$$

$$= \frac{5 \times 9.81 \times 63.48}{60 \times 1000} \textbf{ KW}$$

$$= \textbf{0.052 KW}$$

Considering the factor of safety as 3

$$\text{Theoretical blower Power } (P_b) = 0.052 \times 3 \textbf{ KW}$$

$$= 0.1556 \textbf{ KW}$$

Considering the $\text{Efficiency}\ (\eta)$ **of blower as 0.45 and substituting in the eqn (3.3) we have,**

$$\text{Actual blower Power} = \frac{0.1556}{0.45}\ \text{KW}$$

$$\text{Actual blower Power} = 0.3459\ \text{KW}$$

Or

$$\text{Actual blower Power} = 0.512\ \text{H P}$$

The easily available blower in the market with best economy is 1HP is taken which is more than that of the calculated Actual blower power 0.512HP.hence, the design is safe.

The easily available blower in the market with best economy is 1HP is taken which is more than that of the calculated Actual blower power 0.512HP.hence, the design is safe.

Note : 1 hp blower is considered from the economical point of view and the flow rate is controlled to the required rate by using flow control valve.

4. RESULTS AND DISCUSSIONS

In order to obtain the various performance parameters experiments were conducted on normal kirloskar engine and as well as the modified engine with fabricated suction blower. The engine has constant Speed 1500 N and experiments are conducted at various loads on brake drum varying from 0 to 12(kg). Spring mass constant are considered which vary according to the loads applied to kirloskar engine and modified setup.

Time taken by the normal engine and as well as for modified engine for consumption of 20ml fuel at various loads is observed and are noted down. Manometer readings of the normal engine and as well as for modified engine are observed and are noted down. The values of loads on the brake drum, Time for 20ml and Manometer readings are noted down in order to get various performance parameters such as Brake Power, Indicated Power, Brake Specific Fuel Consumption, Overall Efficiency, Brake Thermal Efficiency. Observations made on both normal engine and the modified engine are useful in order to formulate and produce the various performance parameters respectively.

After the various performance parameters of the normal engine and modified engine are produced using the observations they can compared with each other. Comparison study of the performance parameters of both the engines helps in getting results. Positive performance parameters of the modified engine can be analyzed and the shortcomings in the performance of the modified engine can be analyzed. Comparison study gives a brief outcome of performance of the modified engine and helps in easily determining the advantages and disadvantages of the modified system. Performance variations in the modified engine can be easily noted down. Comparison Analysis of the various performance parameters noted can be used in further and better modification of suction blower assembly. Results can be used to analyze performance parameters and as well as for further Research Work on extracting the exhaust using suction blower.

Table 1.1: observations of baseline experimental setup

SI .No	Speed N	Load on the Brake drum (Kg)			Time for 20ml	Manometer reading		
	RPM	W	S	W-S		H1	H2	H1-H2
1	1500	0	0	0	185	13	-54	67
2	1500	3	0.2	28	138	13	-54	67
3	1500	6	0.4	56	123	13	-54	67
4	1500	9	0.6	84	110	13	-54	67
5	1500	12	0.8	112	95	13	-54	67

These are the observations of the baseline i.e. before modification experimental setup. The experiment was conducted with constant speed with variations in loading. The load is applied on the rope brake dynamometer. The consumption of the fuel with respect to time is observed in this table..

Observations of modified experimental setup

Half valve opening of the suction blower

Table 5. 2:observations of half valve open

SI. No	Speed N	Load on the Brake drum (Kg)			Time for 20ml	Manometer reading		
	RPM	W	S	W-S		H1	H2	H1-H2
1	1500	0	0	0	206	13	-54	67
2	1500	3	02	28	158	13	-54	67
3	1500	6	04	56	134	13	-54	67
4	1500	9	06	84	115	13	-54	67
5	1500	12	08	112	102	13	-54	67

Full valve opening of the suction blower

Table 5.3: observations of full valve open

SI. No	Speed N	Load on the Brake drum (Kg)			Time for 20ml	Manometer reading		
	RPM	W	S	W-S		Hl	H2	Hl-
1	1500	0	0	0	196	6	-61	67
2	1500	3	02	28	148	6	-61	67
3	1500	6	04	56	134	6	-61	67
4	1500	9	06	8.4	120	6	-61	67
5	1500	1	08	11.2	106	6	-61	67

Performance Parameters Calculation and Observation Table 5.4.1 For Baseline Engine

Table 5.4: calculations of baseline engine

Slaw	W net Kg	mf Kg/sec	B.P kW	I.P kW	BSFC Kg/kWh	ηm %	ηbth %	ηith %	A\F %
1	0	8.94x10⁻⁵	0	2.3	0	0	0	56.46	84.31
2	2.8	1.06x10⁻⁴	0.84	3.14	0.4542	26.75	17.39	65	71.19
3	5.6	1.34x10⁻⁴	1.68	3.98	0.2870	42.25	27.50	65	56.31
4	8.4	1.504x10⁻⁴	2.52	4.82	0.2100	52.28	36.70	70	50.17
5	11.2	1.742x10⁻⁴	3.36	5.66	0.1866	59.36	42.33	71	43.32

Various observation of the baseline engine obtained are noted down and are formulated in order to obtain the various performance parameters. The performance parameters are formulated on the basis varying loads. Fuel consumption in the baseline engine increases with a increase in load. Brake Power developed increases with a increase in loads. Brake Specific Fuel Consumption decreases with increase in loads applied on the brake drum. Mechanical Efficiency, Brake Thermal efficiency, indicated Thermal Efficiency varies according to the loads. Mechanical efficiency of the normal engine ranges from 26%-59%. Air Fuel Ratio is mass of air to mass of fuel. Air Fuel Ratio determines the combustion of the fuel.

5.4.2 For Modified Engine

Half valve open of suction blower

Table 5.5: calculations of half valve opening

Sl.no	W net Kg	Mf Kg/sec	B.P kW	LP kW	BSFC Kg/kWh	ηm %	η b.th %	ηith %	A/F %
1	0	8.030×10^{-5}	0	2.60	0	0	0	71	95.15
2	2.8	1.040×10^{-4}	0.8436	3.443	0.4438	24.49	17.80	72.6	73.16
3	5.6	1.235×10^{-4}	1.687	4.287	0.263	39.35	29.90	72.0	61.61
4	8.4	1.439×10^{-4}	2.530	5.13	0.2047	49.31	38.58	78.2	52
5	11.2	1.620×10^{-4}	3.374	5.974	0.1728	56.46	45.68	80.8	46.97

Various observation of the modified engine with half valve opening obtained are noted down and are formulated in order to obtain the various performance parameters. The performance parameters are formulated on the varying loads. Fuel consumption in the modified engine with half valve opening increases with increase in load. Fuel consumption in the modified engine with half valve opening is less than the baseline model. Brake Power developed increases with increase in loads. Brake Power developed by the modified engine with half valve opening is more than the normal engine. Brake Specific Fuel Consumption decreases with increase in loads applied on the brake drum. Brake Specific Fuel Consumption of modified engine with half valve opening is less than the normal engine or baseline model.

Mechanical Efficiency, Brake Thermal efficiency, indicated Thermal Efficiency varies according to the loads. Brake Thermal efficiency, Indicated Thermal Efficiency of the of modified engine with half valve opening is less than the normal engine. Mechanical efficiency ranges from 24%-56% due to excess vibration.

Full valve open of suction blower

Table 5.6: calculations of full valve opening

Sl.no	Wnet Kg	nu Kg/sec	B.P kW	I.P kW	BSFC Kg/kwh	tum %	llb,th %	tiLth %	A/F %
1	0	8.443×10^{-5}	0	2.6	0	0	0	67.5	90.1
2	2.8	1.054×10^{-4}	0.8435	3.4435	0.4498	24.49	17.56	71.6	72.1
3	5.6	1.149×10^{-4}	1.6870	4.287	0.2452	39.35	32.21	81.8	66.2
4	8.4	1.253×10^{-4}	2.592	5.192	0.1741	49.92	45.37	90.8	60.6
5	11.2	1.451×10^{-4}	3.378	5.978	0.1547	56.50	51.06	90.3	52.4

Various observation of the modified engine with half valve opening obtained are noted down and are formulated in order to obtain the various performance parameters. The performance parameters are formulated on the varying loads. Fuel consumption in the modified engine with full valve opening increases with increase in load. Fuel consumption

in the modified engine with full valve opening is less than the baseline model and half valve. But during low loads half valve should be maintained and full valve should be maintained for less fuel consumption. Brake Power developed increases with increase in loads. Brake Power developed by the modified engine with full valve opening is more than the normal engine and the half valve. Brake Specific Fuel Consumption decreases with increase in loads applied on the brake drum. Brake Specific Fuel Consumption of modified engine with full valve opening is less than the normal engine or baseline model and half valve mechanism. Mechanical Efficiency, Brake Thermal efficiency, indicated Thermal Efficiency varies according to the loads. Brake Thermal efficiency, Indicated Thermal Efficiency of modified engine with full valve opening is less than the normal engine or baseline model and half valve opening. Brake Thermal efficiency of modified engine with full valve opening varies from 18%-52%. Mechanical efficiency ranges from 24%-56% due to excess vibration. Air fuel Ratio is mass of air by mass of fuel. Air Fuel Ratio determines the combustion of the fuel.

Velocity of Suction blower & Engine

Fig 5.1: velocity at full valve open

Fig 5.2: velocity at half valve open

» Full valve open

The velocity of the metallic suction blower as shown in fig 5.1 is 4.5 m/s when the contro valve is fully open.

» Half valve open

Fig 5.2 shows the velocity i.e the suction pressure, when the control valve is restricted to half. The velocity obtained here is 2 m/s.

5.5.2 Velocity of Engines exhaust

Fig 5.3: Engine exhaust velocity

The velocity of the engines exhaust was measured using anemometer. The exhaust velocity obtained here is 15.8 m/s. The above figure shows the exhaust velocity.

Experimental Investigation

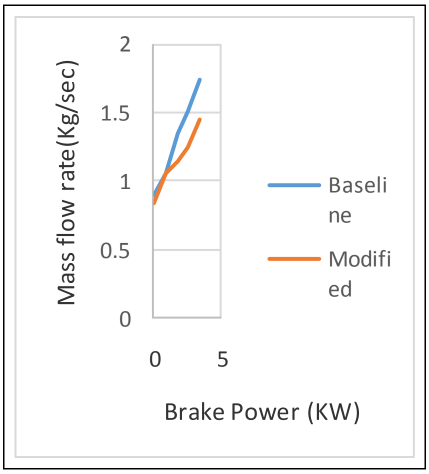
For the obtaining the results various observations noted down through the experiments were formulate in order to the performance parameters. Various parameters obtained such as mass flow rate at different loads, different brake powers are compared. Comparison Study between performance parameters of the normal engine and as well as performance parameters of modified engine delivers a clear picture of the positives the drawbacks of the modified engine. This part of the report deals with comparison study between the normal engine and as well as the modified engine.

According to the observations and calculations obtained from the experiments as seen in chapter 4. The various performance parameters of the baseline model and as well as the modified engine with fabricated metallic suction blower were calculated. The performance parameters are being compared in this chapter in order to obtain the results.

The following performance parameters of the baseline model and the modified engine with fabricated metallic suction blower are compared for investigation and results.

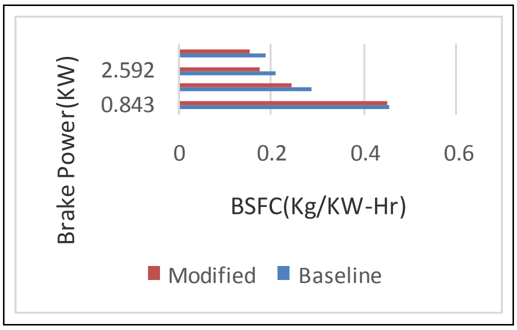
Mass Flow rate in Kg/sec V/S Brake Power in kW. Brake Specific Fuel Consumption in Kg/kWh V/S Brake Power in kW. Brake Thermal Efficiency in % V/S Load in Kg.

5.6.1 Full Valve Opening



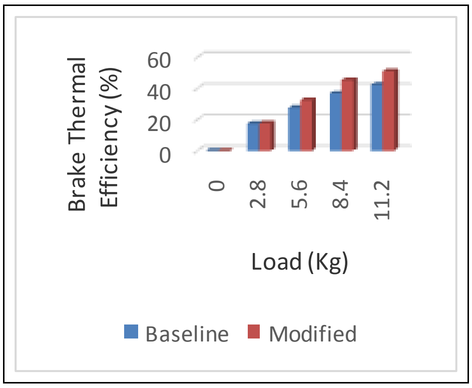
After comparing the mass flow rate of fuel for baseline model at full valve and modified model, it is observed that fuel consumption in Kg/sec in the modified model is decreased as compared to the baseline model as shown above It is observed that as the power goes on increasing, the difference in the mass flow rate of the fuel is increasing between the baseline and modified model. Hence Fuel consumed by modified engine at full valve is less as compared to normal engine at various Brake Powers.

Comparing the Brake Specific Fuel Consumption



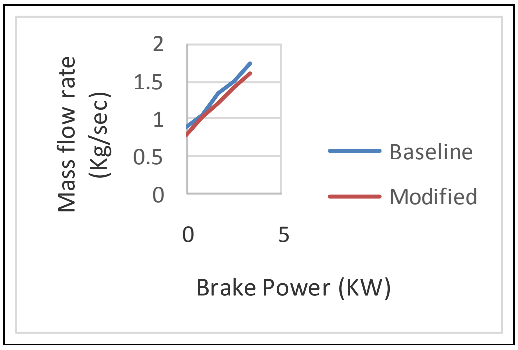
The graph shows that, the fuel consumption of the modified model at full valve opening is decreased as the brake power is increasing. The brake power is varying according to the load applied on the rope brake dynamometer, and therefore the graph shows that as compared to the baseline model as the load goes on increasing the fuel consumption is decreasing.

Comparison of Brake Thermal Efficiency V/S Load



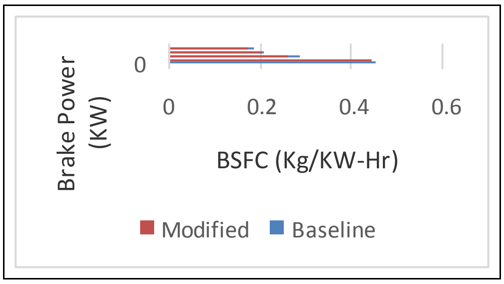
Brake thermal efficiency is defined as the amount of heat energy used from the fuel burnt in order to convert it in mechanical energy. Here the comparison of baseline and modified model at full valve opening shows that as the load goes on increasing the brake thermal efficiency of the engine is increasing. There is an increase of 8. 73% in brake thermal efficiency of modified model as compared to baseline model. In this at high loads more heat is converted into mechanical energy and hence the fuel consumption is less.

Half Valve Open

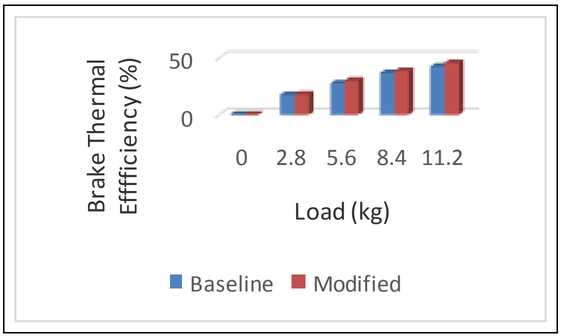


As per the previous results obtained for full valve opening, for initial load the mass flow rate was slightly more in half opening than that in full valve opening.

But as the load goes on increasing the mass flow rate is more than that in full valve opening. Hence the half valve should be used at low loads and full valve during heavy loads

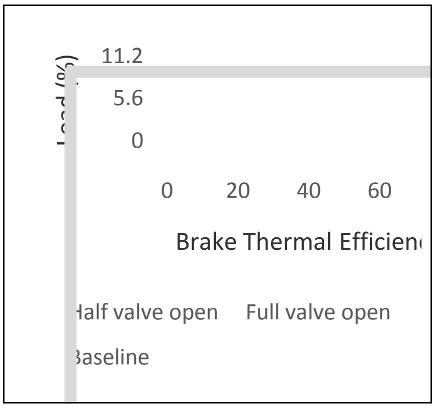


As compared to full valve, at low loads half valve is more efficient than full valve opening. At 0.8433kW power the fuel consumption is decreased by 0.0104 kg/kW which is greater than that of full valve opening. And as the load goes on increasing according to the results obtained, full valve opening proved more efficient than that of half valve opening.

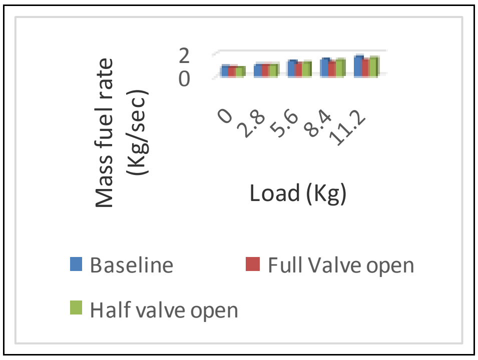


According to the previous graphs for half valve, it is know that at low loads half valve is more efficient. And the above graph shows the brake thermal efficiency which means the conversion of total heat energy obtained from combustion of the fuel to the mechanical energy. In this at low loads more heat is converted into mechanical energy and hence the fuel consumption is less.

Overall comparison of the experiment



Overall comparison study is done in order to achieve or select the best control valve opening. As seen in Fig 5.10 brake thermal efficiency is compared with loads. The graph shows that modified engine setup with full valve opening delivers better brake thermal efficiency than the modified half full valve engine setup. But half valve brake thermal efficiency is less more than the normal engine. Hence full valve opening should be used for better thermal efficiency.



The Fig 5.11 above shows comparison of the fuel rate with the various loads. The graph shows that fuel consumption is less when half valve is maintained at the low loads. But as the load goes on increasing the valve opening should be maintained at full valve.

As at full valve the fuel consumption is decreasing considerably. Hence at low loads half valve should be maintained and at heavy loads full valve opening should be maintained for the better and less fuel consumption.

4. CONCLUSIONS

- **Due to better combustion inside chamber, more amount of air to the fuel is supplied due to which the thermal efficiency increases and hence overall efficiency is increases.**
- **Lesser fuel consumption is achieved while comparing the baseline model that is normal engine with the model fabricated with suction blower that is modified engine.**
- **Brake specific fuel consumption has drastically reduced for fabricated model as observed.**
- **Fuel consumption is less for low load in half valve opening has compared to full valve opening.**

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