Investigation of Diesel Engine Performance with Design Modifications in Piston: Inducing Turbulence by Swirl

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***Abstract*—The research work carried out with the aim of improving the performance of diesel engine by modifying the piston head in view of promoting the uniform combustion by inducing turbulence in the incoming charge. The experiments are conducted at constant speed with 20.1 compression ratio and 300 bar injection pressure at three different injection timings. A stirrer is introduced at the top of the piston so as to create more turbulence in the incoming charge for enhancing the vaporization rate. Whirling motion is created in a combustible mixture by providing rotating blades in the cavity/bowl on the reciprocating piston head. The oscillatory motion of the connecting rod will cause to rotate the blade by an angle of 60°. The final experimental result shows that, the modified piston performance and emission results are achieved better compared to normal piston at all injection timings.**

***Index Terms*— diesel engine, piston, turbulence, diesel**

I. INTRODUCTION

Internal combustion engines have been a relatively inexpensive and reliable source of power for applications ranging from domestic use to large scale industrial and transportation sectors in most of the twentieth century. Direct-injection diesel engines, having the evident benefit of higher thermal efficiency than all other engines, hence it served for both light- duty and heavy-duty engines. Motion of charge in cylinder of internal combustion engines is a key factor that governing the fuel-air mixing and burning rates in combustion of diesel engines. Turbulence is created during the inducing stage of charge and developed during the compression stroke in prior to combustion of internal combustion engines. Therefore, a better understanding of fluid motion during the induction process is critical for developing engine designs to attain the desirable operating and emission characteristics.

S.L.V. Prasad *et al.* [1-3] studied the influence of the air swirl in engine cylinder on the performance and emission of single cylinder diesel direct injection engine. Piston crown is modified, in order to achieve the different swirl intensities in the cylinder, three design parameters have been changed in cylinder head, piston crown, and inlet duct. i.e. alteration of combustion chamber to enhance the turbulence in the cylinder. The swirl is intensified by cutting grooves on the piston crown. J Benjacs *et. al.* [4] studied the performance of 1.8 L single-cylinder diesel engine with an additional attachment of the swirl rate controller. This attachment allowed variation of the mean swirl ratio. Swirl level has a significant influence on the combustion process and exhaust emissions. When the swirl ratio increased to certain level, more intense premixed combustion phase is improved. By increasing the swirl ratio, pressure and temperature of the charge are increased due to premixed and controlled combustion in the engine cylinder. Which

Manuscript received August 27, 2017; revised August 15, 2018.

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doi: 10.18178/ijmerr.7.5.495-499

results the promotion of NOx formation, also reduce the soot production and increase the soot oxidation. The overall combustion duration was also shortened with reduced fuel consumption. Somender Singh et al. [5] studied the effect of design change by forming grooves or channels, or passages through the squish areas to enhance in-cylinder turbulence. This provides a faster and efficient burn, with less loss of heat, through design to improve turbulence in combustion chambers. Yang-Liang Jeng *et al.* [6] experimentally investigated the quality of the tumbling motion, especially for the engine with a bowl in the piston. It is observed that, a small-scale vortex will be reserved inside the bowl in the piston. In their further investigation the quality of the vertical flow in the axial plane with the generation of turbulence during the compression stroke is strongly recommended. Jeong-Eue Yun *et al.* [7] Studied the combined effect of swirl and tumble flow of the intake port system in cylinder head. Since both swirl and tumble flows are together induced to the cylinder, in-cylinder flow pattern becomes very complicated so that it is difficult to decide this flow as one major pattern like a swirl or tumble. Their study results are to find a new evaluation index for in-cylinder flow characteristics instead of current swirl or tumble coefficient using a steady flow test rig on intake port systems. Yoshihiro Suzuki [8] proposed baseline modifications in piston and cylinder liners to enhance mechanical and thermal loading on vital engine parts. Hard-anodizing the piston head, reinforcing the piston head with SiC - whiskers. He proposed a new process for improving the surface lubrication by introducing numerous finely distributed micro-pits. B.V.V.S.U. Prasad *et al.* [9] Studied the in-cylinder, air motion in number of combustion chamber geometries which produced the highest in-cylinder swirl and Turbulence Kinetic Energy (TKE) around the top dead center. The optimal nature of this re-entrant piston bowl geometry is confirmed by detailed combustion simulations and emission predictions. Lalvani et al. [10] has investigated the combustion, performance and emission characteristics of a diesel engine with modified piston and their results showed that reduction in brake thermal efficiency that may be due to poor air fuel mixing and higher viscosity of the fuel blends. This incited the research towards improvement of turbulence for air fuel mixing and better brake thermal efficiency, combustion characteristics and reduced emissions with the modification of piston in diesel engines.

It is observed from the experimental work that the bowl geometry has a negligible effect on the airflow during the intake stroke and early part of the compression stroke. But when the piston moves towards Top Dead Centre (TDC), the bowl geometry has a significant effect on air flow for creating proper turbulence thereby resulting better mixing and better combustion.

Turbulence plays a vital role in combustion phenomenon. In combustion the flame speed is very low in non-turbulent mixtures. A turbulent motion of the mixture intensifies the processes of heat transfer and mixing of the burned and unburned portions in the flame front, which practically increase in proportion to the turbulence velocity. This turbulence can be increased at the end of the compression by suitable design of the combustion chamber, which involves the geometry of cylinder head and piston crown. The degree of turbulence increases directly with the piston speed. However, excessive turbulence is also undesirable. The effects of turbulence can be summarized as, turbulence accelerates chemical action so that the combustion time is reduced and hence minimizes the tendency to detonate. Turbulence increases the heat flow to the cylinder wall and in the limit excessive turbulence may extinguish the flame. In the present work experiments are conducted at constant speed, with and without piston modification, 300 bar injection pressure and varies injection timings at four different loads levels viz., 20%, 40%, 60% and 80% of full load with 20.1 compression ratio.

II. Experimental Setup

Table I and Fig. 1 show the engine specifications and schematic diagram of the experimental setup for determining the effects of squish and tumble on the performance of CI engine. It is a computerized single cylinder, four stroke, water cooled CI engine with an eddy current dynamometer. The engine is provided with Chromium-Aluminum thermocouples to measure the jacket water inlet and outlet (T1 & T2), calorimeter water inlet and outlet (T3 & T4), exhaust gas inlet and outlet (T5 & T6) temperature.

TABLE I. Engine Specifications

Engine Parameters	Specifications
Engine type	TV1 (Kirloskar, 4-Stroke)
Number of cylinders	Single cylinder
Bore	87.5 mm
Stroke	110 mm
Swept Volume	661 cc
Compression Ratio	20.1:1
Rated Speed	1500 RPM
Dynamometer	Eddy Current Dynamometer
Type of Ccooling	Water Cooling
Fuel injection pressure	300 bar
Fuel	Diesel

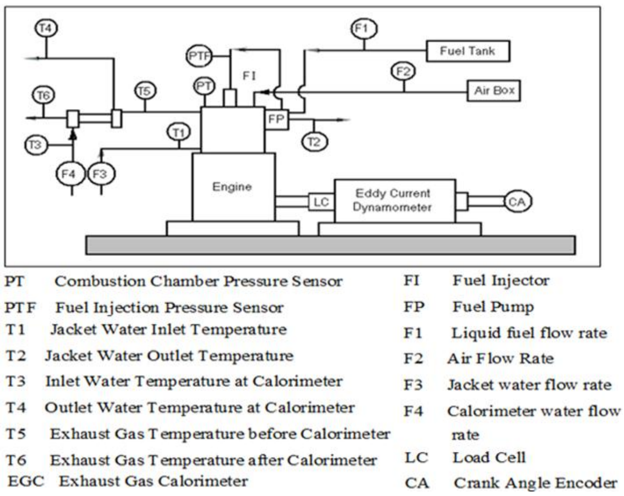

Figure 1. Schematic Diagram of the Experimental Set-up.

This engine also provided with pressure sensors, the dynamic pressure with water cooled piezo sensor, combustion gas pressure with differential pressure transducers and fuel injection pressure with differential pressure unit. Cooling water flow with a calibrated Rota meter with stainless steel float. An encoder is fixed for crank angle record. The signals from all these sensors interface with a computer to display P-Ө, P-V and FIP-Ө plots. The provision is also made in the measurement of volumetric fuel flow. A built-in program in the system calculates indicated power, brake power, thermal efficiency, volumetric efficiency and heat balance.

A. *Piston Modification*

Figure 2. Normal piston.

Figure 3. Modified piston.

Fig. 2 and Fig. 3 show the normal piston and modified piston respectively**.** Normal piston is having a simple bowl shaped structure on the crown of it. But the modified piston (Fig. 4) is made with three blades at 120^0 to each other. The same aluminum alloy material is used in fabrication of chamber and 2mm thick small strips are used to make the chambers.

B. *INDUS Model PEA205 Five Gas Analyzer*

Emission analysis is carried out by INDUS model PEA205 as shown in Fig. 5, it is a 5-gas analyzer for monitoring CO, CO_2, HC, O_2 and NOx in automotive exhaust. It meets OIML Class-I specifications. CO, CO_2 and HC (Hydrocarbon residue) are measured by NDIR technology and O_2 and NOx are measured by electrochemical sensors.

Figure 4. Modified piston arrangement in cylinder.

Figure 5. INDUS model PEA205 five gas analyzer.

III. Results and Discussions

A. *Performance Characteristics*

The experiments were conducted on a DI diesel engine for various loads with diesel. The variation of brake specific fuel consumption with respect to load for modified piston and normal piston shown in Fig. 6. It can be observed that the brake specific fuel consumption of modified piston is reduced compared to normal piston. A similar trend in brake specific fuel consumption can be observed at all the loads considered in the present work. From figs. the brake specific fuel consumption is minimum at 20.1 CR, 300bar Pressure and advanced timing for swirl piston at 80% load 0.258 kg/kw-hr and it is maximum at 20.1 CR, 300bar Pressure, retard timing for swirl piston at 20% load 0.543 kg/kW-hr. So here brake specific fuel consumption varies between 0.258 kg/kW-hr and 0.543 kg/kW-hr. The variation of brake thermal efficiencies with respect to load for modified piston and normal piston shown in Fig. 7. It can be observed that the brake thermal efficiency of modified piston increases with increase in load at all injection timings compared to normal piston due to homogeneous combustion by increasing the turbulence of the air. A similar trend in brake thermal efficiency can be observed at all the loads. From figs. the brake thermal efficiency is minimum 13.01% at compression ratio of 20.1, 300 bar injection pressure with retard injection timing for the normal piston at 20% load.

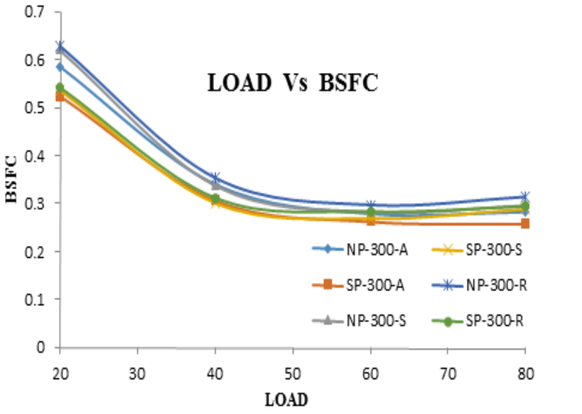

Figure 6. Load Vs Brake Specific Fuel Consumption for 20.1 CR at 300bar.

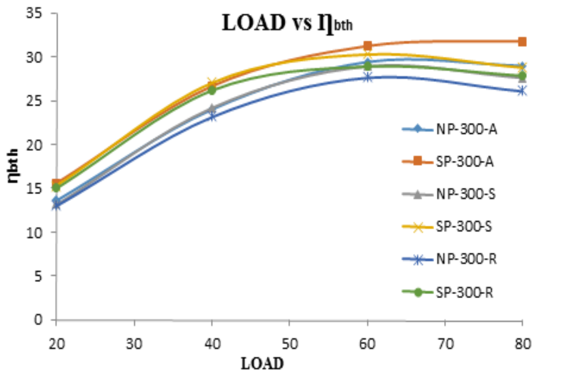

Figure 7. Load Vs Brake Thermal Efficiency for 20.1 CR at 300bar.

B. *Emission Characteristics*

Major constituents of engine emissions are CO, O_2, CO_2, unburned HC, NO_x and particulate matter. The variation of hydrocarbons with respect to load for normal piston with modification is depicted in Fig. 8. From the results, it can be noticed that the unburnt hydrocarbon is minimum at 20.1 CR, 300bar Pressure, retard timing for swirl piston at 20% load 5ppm and it is maximum at 20.1 CR, 300bar Pressure, advanced timing for normal piston at 80% load 45ppm.

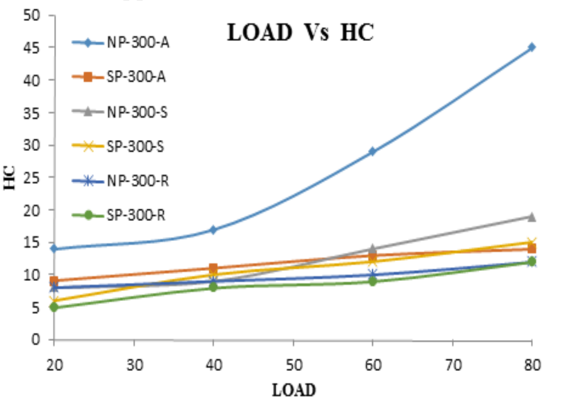

Figure 8. Load Vs Hydrocarbons for 20.1 CR at 300bar.

The variation of oxides of nitrogen with respect to load can be observed for normal piston and modified piston is depicted in the Fig. 9. NO_x emissions are increasing with uniformly increase in the peak cylinder temperature by an increase in load. The results show that NO_x is minimum at 20.1 CR, 300 bar pressure, retard injection timing for normal piston at 20% load 95 ppm and it is maximum at 20.1 CR, 300 bar pressure, advanced injection timing for swirl piston at 60% load 483ppm.

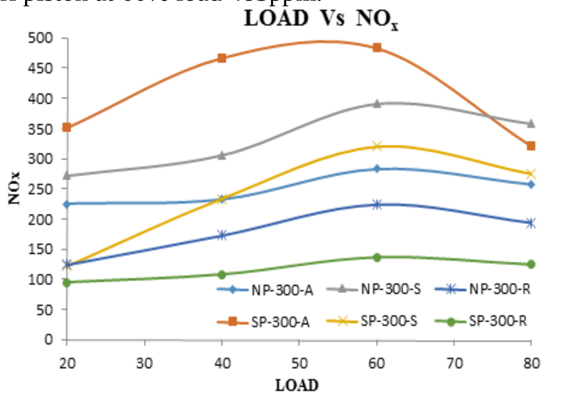

Figure 9. Load Vs Oxides of Nitrogen for 20.1 CR at 300bar.

IV. Conclusions

In order to improve the performance of the engine several active and passive alternative techniques are available. Piston modification is a one of the active technique to induce more turbulence and minimize the emissions. In this connection, the geometry of the piston is modified by accommodating rotating blades in the piston crown to induce turbulence by means of swirl motion of charge.

- The brake specific fuel consumption is found to be 0.282 kg/kw-hr for normal piston and 0.258 kg/kw-hr for swirl piston at advanced injection timing. From the results, 8.5% decrement of the brake specific fuel consumption observed for swirl piston compared to normal piston with 300 bar pressure and advanced injection timing at 80 % load.
- The brake thermal efficiency with 300 bar pressure and advanced injection timing at 80 % load is found to be maximum value of 31.75 % for swirl piston and it is 28.99 % of normal piston which is an approximate rise of 2-3 % at every load and injection timings.
- The HC emission with the use of swirl piston, there has been a considerable decrement was noticed compared to normal piston even at 80% load with advanced injection timing.
- The NOx emissions are 275 ppm for swirl piston and 358 ppm for normal piston with 300 bar pressure and standard injection timing at 80 % load. Hence, with the use of swirl piston there has been a considerable decrement in NOx emissions.

From the above, it can be concluded that with the use of swirl piston there has been an improvement in brake thermal efficiency and decrease of brake specific fuel consumption and have a remarkable decrease in exhaust emissions of HC, NO_x. Also the results obtained for advanced injection timing performance are better compared to other injection timings.

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