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Thermal Analysis of Piston for the Influence on Secondary motion

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ABSTRACT

The gas force due to the combustion in the cylinder of an IC engine will cause the piston to move with primary motion and secondary motion. The primary motion of the piston from TDC to BDC is linear in nature. This motion is desired for translation of motion of engine components. Secondary motion is due to the transverse motion of the piston while piston moving from TDC to BDC and vice-versa. The secondary motion of the piston is considered as the main source for the piston slap, which in turn causes the impact on the cylinder walls resulting in engine vibration and noise. In the present study, an effort is made to understand the effect of the thermal load, generated by the combustion of fuel inside the cylinder, on the piston deformation and thermal stresses induced in piston. This deformation of the piston inside the cylinder causes the gap between the cylinder and piston to vary and also the piston to move transversely along with impact forces. The transverse motion of the piston in the cylinder is observed experimentally by measuring the gap between piston and cylinder at thrust side load condition. Finite element analysis (FEA) is considered as one of the best numerical tools to model and analyze the physical systems. FEA is carried out to find the piston deformation due to thermal load on the piston for the temperature data obtained from experiments. The three dimensional piston is modeled in CATIA V5 R19 and analyzed in ANSYS 12 solver. The simulation results are used to predict effect of temperature on piston deformation and its secondary motion which are the principal source of engine vibration and noise.

Keywords – Deformation, FEA, Piston, Secondary motion, Simulation

I. INTRODUCTION

As gas forces the piston in IC engine, the piston will move with primary motion and secondary motion. Secondary motion is a transverse motion of the piston while piston moving from TDC to BDC visa-versa. Transverse motion produced is not desired in IC engines because it produces piston slap and twisting movement in the remaining part like pin, connecting rod and crank shaft. When the hot combusted gases impact the piston heat will transfer to the piston and the engine parts. As soon the hot gases impact the piston, the thermal stresses will develop in the piston which results in piston deformation [1]. Figure 1 shows the schematic diagram of engine parts with lateral motion of the parts [2].

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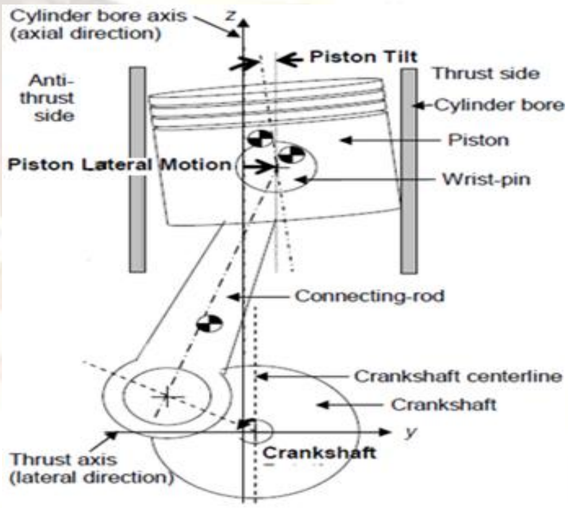

Fig. 1 Schematic of engine parts

Piston Temperature Distribution

The piston crown is exposed to very high combustion temperatures. Figure 2 shows the typical values of temperature at different parts of a cast iron piston. It may be noted that the maximum temperature occurs at the centre of the crown and decreases with increasing distance from the centre. The temperature is the lowest at the bottom of the skirt. [12]

Poor design may result in the thermal overloading of the piston at the centre of the crown. The temperature difference between piston outer edge and the centre of the crown is responsible for the flow of heat to the ring belt throughout the path offered by metal section of the crown. It is therefore necessary to increase the thickness of the crown from the centre to the outer edge in order to make a path of greater cross-section available for the increasing heat quantity.

The length of the path should not be too long or the thickness of the crown cross-section too small for the heat to flow. This will cause the temperature at the centre of crown to build up and thereby excessive temperature difference between the crown and the outer edge of the piston will result. This may even lead to cracking or piston during overload operation.

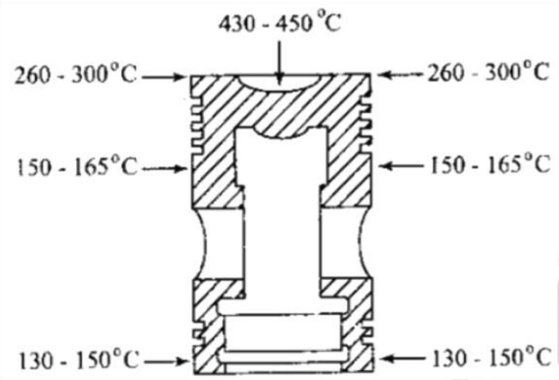

Fig. 2 Piston Temperature distributions

There are many technical contribution published in IC engines field, mainly for the piston and piston ring dynamics. The modeling of piston and piston ring for primary and secondary motion, especially FE models with 3D approach have been used to study the secondary motion.

Ouk Sub Lee, Hyebin Choi and HongMin Kim [1] has carried the work to investigate the dynamic deformation of two aluminum alloy, results obtained by considering stresses due to temperature at front-end and back-end of the specimen. The effect of contact time at high temperature is noted early deformation. The strain rate and the high temperature significantly affect the plastic flow behavior. The effect of temperature on plastic flow increases at high temperatures and effects seem to be more pronounced than that of the increases strain rates.

The Dongfang Bai [2] in Sloan Automotive Laboratory, Massachusetts Institute of Technology has been involved in solving piston secondary motion of IC engine by developing a numerical model, with both cavitations & piston skirt deformation taken into consideration. As the load and correspondingly the side force are large, the piston skirt will have very large deformation and hence result in a lateral position. Hirotaka Murakami, Narutake Nakanishi, Naoto Ono and Tomoharu Kawano, [3] have developed a new piston secondary motion analysis that accurately predicts piston strength and piston slap, that occurs when the engine is running. Z. Geng and J.Chen [4] has been involved in the investigation of piston slap induced vibration for engine condition. P**.** Gustof, A. Hornik [5] has been involved to find out the influence of the engine load on value and temperature distribution in the piston of the turbocharged Diesel engine. The results of calculations of the temperature distribution in the piston of the turbocharged diesel engine in dependence from the engine loads were received by means of the two – zone combustion model and the finite element method. Zuoqin Qian, Honghai Liu, Guangde Zhang, and David J. Brown[6] has involved in Temperature Field Estimation for the Pistons of Diesel Engine 4112 using the combination of experimental measurement and finite element analysis (FEA). The experimental results show that the temperatures of pistons and liners on the same circle are different as the circle area changes, which is caused by their local over-high heat loads. Dr. S.N Kurbet [7] et al was presented the results by finite element study of piston ring, under assembly load in terms of induced stress and ring gap. The study included the stress analysis at the interface between the coating and substrate of ring for various lay design. Information from the analysis would serve to reduce the design performance testing cycle time & be useful in the development of coating techniques. The finite element and piston transverse movement calculation technique is satisfactorily used to predict engine vibrations and noise due to piston slap. C.D. Rakopoulos, D.C. Rakopoulos, G.C. Mavropoulos, E.G. Giakoumis [8] have been involved in estimation of cylinder wall transient temperature cylinder walls of a diesel engine at various operating conditions. Ali Sanli, Ahmet N. Ozsezen, Ibrahim Kilicaslan, Mustafa Canakci [9] have studied the heat transfer characteristics between gases and in-cylinder walls at fired and motored conditions in a diesel engine were investigated by using engine data obtained experimentally. Dr. Ahmed A, Dr. Basim [10] has worked to study the Thermal effects on diesel engine piston and piston compression ring.

The conclusion has came to know from above literature survey are as, many works have been carried out to study the causes of piston slap and piston distortion. The result analysis gives the force exerted by the gas over the piston is the major source which causes the piston slap. Also many works have been carried to study effect of piston slap on lubrication. Due to combustion of fuel enormous heat will be released in engine cylinder. This thermal load will be absorbed by the piston and other components of the engine. The effect of the engine operating conditions, viz., load on the engine keeping speed of the engine constant, the heat release in the cylinder, heat absorbed by the piston, have been discussed and found that increase in engine load at constant speed has a major effect on the peak heat fluxes and heat transfer coefficients over the combustion chamber wall surfaces. The effect of contact time at high temperature is noted early deformation. The strain rate and the high temperature significantly affect the plastic flow behavior.

The objective of the present work is to find out the contribution of thermal load in piston slap and piston deformation. For the purpose it is necessary to conduct an experiment to find out the piston position inside the cylinder and also to estimate the piston cylinder gap. Analytical calculations are done to find out the piston crown top surface temperature for which the experimental data is used. The other intention is to know the temperature distribution over the piston and to find out the piston deformation from the finite element analysis to correlate the effect of temperature on piston deformation. To find out the lateral motion of piston in terms of the gap between cylinder and piston, to predict the effect of piston deformation for piston slap and hence the vibration of the engine.

II. Description of IC Engine

The specifications of the single cylinder vertical diesel engine are shown in Table 1.

Table 1. Diesel engine specifications

Sl. No	Description	Specification
1	Engine make	Kirloskar diesel engine
2	Bore	80 mm
3	Stroke	110mm
4	Engine speed taken for study	1500 rpm
5	Compression ratio	16.5:1
6	Test condition/Type	Water cooled direct injection diesel single cylinder engine
7	Max pressure at study rpm	54 bars

The material properties of the piston of the cylinder are shown in the Table 2. The observations of test rig are made and are listed in Table 3.

Table 2. Material properties

Part	piston
Material	Aluminum alloy
Density (Kg/m^3)	2630
Youngs modulus (MPa)	72400
Coefficient of Thermal Expansion (/^{o}C)	2.3×10^{-5}
Poison's ratio	.31

Load	11.26 N-m
Speed	1464 rpm
Fuel Rate	1.38 kg/Hr
Air Rate	22 m^3/Hr
Water Flow	37.7 cc/sec
t_{w1}, water inlet to Calorie meter	39.7 °C
t_{w2}, water outlet from Engine Jacket	34.7 °C
t_{w3}, water outlet from Calorie meter	55.9 °C
t_{e4}, exhaust Gas inlet to Calorie meter	199 °C
t_{e5}, exhaust Gas outlet from Calorie meter	190.8 °C
t_1, ambient Temperature	36.1 °C
Air Fuel Ratio, AFR	17.87
Load	11.26 N-m
Speed	1464 rpm

Table 3. Test rig observations

III. Analytical calculations:

1) Air is considered as ideal gas, from ideal gas the isentropic compression is given by [13]

$$\frac{T_2}{T_1} = \left(\frac{v_2}{v_1}\right)^{\gamma-1}$$

2) Heat release by the combustion is given by, Q = m_f * LHV (low heat value of fuel) [8, 11]

3) Stoichiometric combustion equation of fuel [8.11]

$$C_{12}H_{23} + 18.5(O_2 + \frac{0.7}{0.3} N_2) \rightarrow 12CO_2 + 13H_2O + 43.167N_2$$

4) Mass fraction,[8,11]

$$Y_m = \frac{molecular\ weight\ of\ fuel}{Molecular\ weight\ of\ fuel + molecular\ weight\ of\ oxygen + molecular\ weight\ of\ nitrogen}$$

5) After combustion to find the gas temperature,[8,11]

$$Y_m \times Q = m_a \times c_p \times (T_g - T_2)$$

6) Woshni's equation for heat transfer coefficient to calculate the cylinder wall temperature,[9,12]

$$h\left(\frac{w}{m^2K}\right) = 3.26 \times P(k\,Pa)^{.8} \times \frac{w\left(\frac{m}{s}\right)^{.8}}{B(m)^{.2} \times T(K)^{.55}}$$

The average gas velocity,

$$w = C_1 S_p + C_2 V_s T_r (P - P_m)/P_r V_r$$

Where C_1 and C_2 are constants, C_1 = 2.38 and C_2 = 3.24×10^{-3}

Velocity of the piston is given by, $S_p(\frac{m}{s}) = \frac{2LN}{60}$

7) To estimate cylinder wall temperature,

$$Q = h \times A\,(T_g - T_w)$$

8) The piston crown top surface temperature is given by, $t_p = t_g - t_w - t_{exh}$

The 3D geometric model of the piston is developed using the dimensions of the Kirloskar engine piston using CATIA V5 modeling software and is shown in Figure 3. This model is then imported to ANSYS software to carry out finite element analysis.

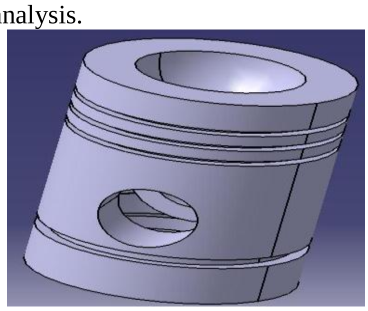

Fig. 3 Piston model

IV. Result and discussions:

The Table 4 shows the result obtained by the analytical calculations using empirical relations used in calculation in above section.

Table 4. Analytical results

Mass flow rate of fue for one cyclel, m_f	3.006×10^{-5} kg/s
Volumetric flow rate of air	22 m^3/hr
Mass flow rate of air for one cycle , m_a	5.69×10^{-4} kg/ s
Temperature at the end of compression stroke, T_2	948.31K
Heat release by the combustion for one cycle, Q	Q = 1303 J/ s
Temperature of the gas, T_g	T_g = 1142.82K
Heat transfer coefficient for one cycle, h	97.30 w/m^{2}K
Cylinder wall temperature, T_w	657.65 K
Piston crown temperature, T_p	567.52K

Experiment has been carried on the single cylinder diesel engine for load condition of 11.6 N and observed for 10 minutes. The Figure 4 shows pressure (P) verses crank angle in degrees (θ). The peak pressure reached to 36 bar at 367° crank angle. Pressure in the suction stroke is remain zero and when the piston starts to move in compression stroke the pressure at 336° crank angle starts to increase. Around 352° crank angle the pressure start to decrease and later at 360° crank angle again pressure starts to increase. This is due to preparation period for combustion after the fuel injection called delay period. Therefore the motoring pressure is considered as 16 bar around 352° crank angle. When the piston starts to move in expansion stroke the will decreases 368° crank angle and reaches to initial pressure zero.

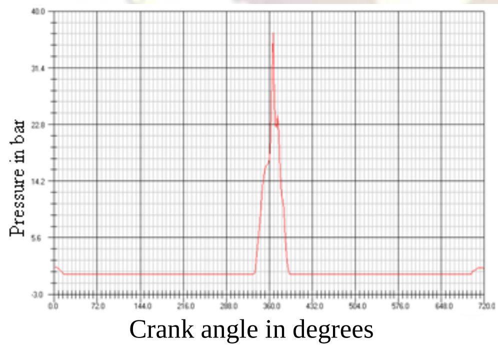

Fig. 4 Pressure verses crank angle graph

From the P-θ graph (Figure 4) it is noted that, the peak pressure, P = 36 bar and motoring pressure, P_m = 16 bar are at 352° crank angle. Hence change in pressure, $\Delta P = P - P_m$. The experiment has been carried out by applying the load of 11.26 N-m by using the electric dynamometer. The first data has been logged after running the engine for first five minutes and the second data has been logged after running the engine for ten minutes. The piston position and piston-cylinder gap inside the cylinder is noted by plotting the piston-cylinder gap verses time graph as shown in Figure 5 (a) and (b).

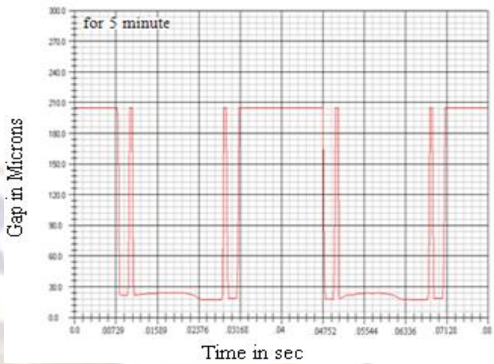

Fig. 5 (a) Piston cylinder gap (for 5 min)

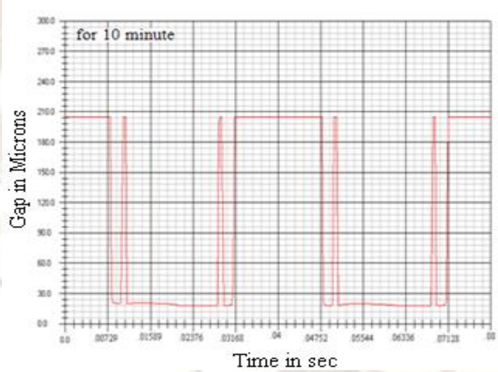

Fig. 5 (b) Piston cylinder gap (for 10 min)

It is clear from the Figure 5. that the piston slap occurs in between 0.0099 sec and 0.0114 sec, 0.02844 sec and 0.0297 sec, 0.0494 sec and 0.0537 sec and 0.0666 sec and 0.0262 sec. About 18 microns to 24 microns there is film lubrication of oil from the cylinder wall and when the piston is at centre line axis then 204 microns of gap between the piston and the cylinder is noted. The finite element analysis is carried out to know the piston behavior due to the thermal load acted upon it by the combusted gas. The Figure 6 shows the temperature distribution over piston.

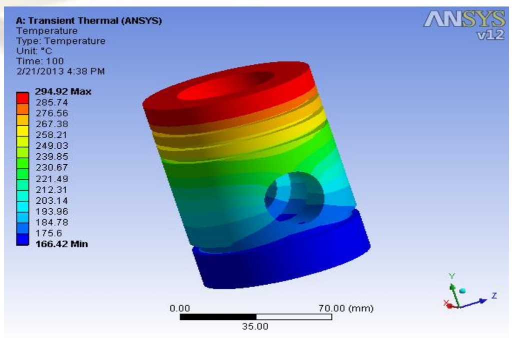

Fig. 6 Temperature distribution

The highest temperature distribution is observed ranging from 285.74°C to 294.9°C at the crown and the skirt temperature distribution is ranging from 166.42°C to 258.21°C. So the piston crown absorbs more heat from the gas, which is directly in contact with the hot gases.

The thermal load is imported in structural analysis for analyzing the piston directional deformation and the stresses developed due to the thermal load. The piston deformation is analyzed in z-direction of piston thrust side. It has been noted in the Figure 7, that the maximum deformation value 263 microns is at the piston crown and the minimum deformation at the skirt is 28 microns in z-direction. The Figure 8 shows the distribution of thermal stresses over the piston. The maximum thermal stress distribution is ranging from 62MPa to 69.788MPa and 54.323MPa to 62MPa noted at inner side oil ring land and at the centre of the piston bowl. The second and third piston land is much exposed to the thermal stresses, but the piston skirt has minimum value thermal stresses ranging from .19MPa to 16.661MPa.

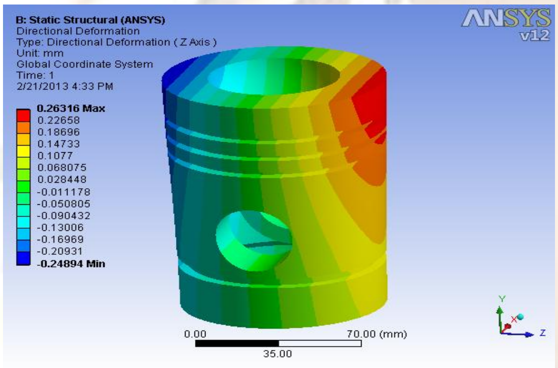

Fig. 7 Piston deformation in z-direction

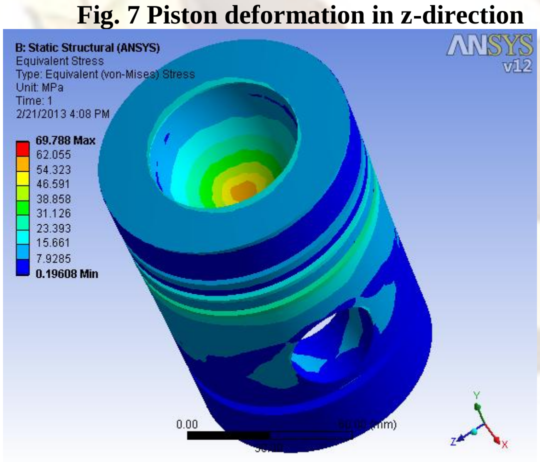

Fig. 8 Thermal stresses over the piston

The piston cylinder gap verses time graphs for five minutes engine run and for ten minutes engine run are combined and plotted with reference to gap in microns and crank angle in degrees is shown in the Figure 9. When the piston starts to move between TDC to BDC due to eccentricity of crank shaft piston will tilt and slap to the one side of cylinder, which will be on thrust side.

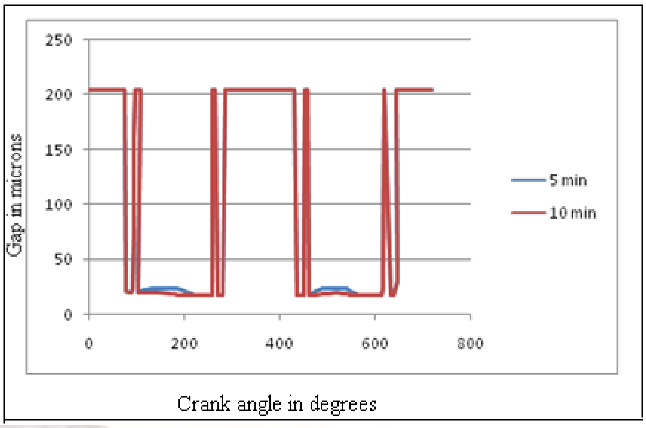

Fig. 9 Piston cylinder gap comparison graph 3

It has been noted clearly that piston slap occurs in-between the 77.4 and 99 degrees of crank angle in the suction stroke, 258 and 280.8 degrees of crank angle in the compression stroke, 432 and 453.6 degrees of crank angle in the expansion stroke and 617.1 and 635 degrees of crank angle in the exhaust stroke. It is observed, when piston axis coincides with the cylinder axis the gap between piston and cylinder is 204 microns and when piston tilts the gap is about 18 microns to 24 microns which accounts the film thickness of the lubricating oil between the cylinder wall and piston skirt. This will repeat for the remaining cycles.

In the Figure 9, it is observed that after running the engine for 10 minutes there is change in piston diameter about 6 microns, this is due to the load absorbed by the piston for combustion. When piston will absorb, the geometry diameter has increased and it has confirmed by the experimental investigation on the present engine.

The FE analysis is done to analyze the piston deformation due to the thermal load. The analysis is done for 100 seconds and the deformation is noticed in the z-direction towards the thrust side of piston. . The maximum deformation value 263 microns is at the piston crown and the minimum deformation at the skirt is 28 microns in z-direction. The designed piston cylinder gap is 204 microns and the change in piston diameter is about 6 microns. So that the designed piston considered for the analysis has lesser deformation values than the FE analysis model and so the designed piston for the analysis of piston deformation and piston slap is correct. The directional deformation in the piston for the given gas pressure is proportional to the variation of the gap between the cylinder liner and the piston. The secondary motion of the piston is considered by the z-directional deformation which is measured here by the gap between cylinder liner and the piston.

V. Conclusions

The secondary motion of the piston is considered as the main source for the piston slap, which in turn causes the impact on the cylinder walls resulting in engine vibration and noise. In the present study, the geometric three dimensional model of the piston is developed and is used for the FE analysis for the thermal boundary conditions which are calculated by using the experimental data of the engine in running condition and by using the empirical relations.

Thermal stresses and deformation in the piston reduces the gap between the cylinder liner and the piston, which increases in the diameter of the piston and influences impact on the cylinder. Gap between the cylinder liner and the piston which indicated the lateral motion of the piston for different crank angle motion of the piston from TDC to BDC is measured.

The FE analysis of the piston revealed that the thermal stresses induced in the piston are proportional to the directional deformation. The z-directional deformation is proportional to the lateral motion of the piston, which is represented by the gap between the cylinder liner and the piston. This result agrees with the results obtained from the FE analysis and hence the FE results can considered conveniently for the prediction of the engine vibration and noise.

Piston material and geometry can be optimized to lower the NVH, by maintaining the thermal efficiency unaltered. This approach can considerably cut down the time and cost of experiments in the design phase of the engine to optimize the engine parameters for low environmental pollution.

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