

Prediction and Reduction of Cylinder Liner Bore Deformation for a Two Wheeler Single Cylinder Gasoline Engine

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ABSTRACT

India is considered to have one of the maximum two wheeler density in the world. Hence, all the scooter and motorcycle manufacturers are striving to keep their market share by quickly bringing quality products with high mileage at a cheap price tag. As emission norms are becoming stringent day by day, these OEMs have to take care of every detail in the engine that is driving their vehicles. So today's engineers are obliged to refine and improve the technologies they use, faster and with greater accuracy than ever before.

This paper focuses on reducing Cylinder Bore deformation for a two wheeler engine through CAE simulation. In standard testing conditions this deformation was observed to cause engine seizure. Bore deformation is of great significance to the overall performance of an engine. It can increase oil consumption, blowby and emissions and may influence piston dynamics to a great extent. These deformations are however, difficult to measure and being in micron level, are often prone to misinterpretation. Moreover, making physical prototypes with frequent design changes for reducing bore deformation is a highly time consuming and costly exercise since it involves most critical engine parts. These parts are often long lead time items in engine development cycle. This, with the added constraints on packaging and styling in two wheelers, leaves very little variations to be done on the design. Therefore, to cut down the design cycle time, virtual simulation methods are preferred.

A high cubic capacity two wheeler gasoline engine is studied to evaluate and reduce deformation of cylinder liner. The cylinder head assembly comprising of major components like head, block, liner, etc. was meshed with second order elements and the analysis model was set up in a commercially available numerical code. The base design was then subjected to bolt pretension loads for simulating liner deformation. Based on this finite element analysis, the scope of reducing deformation was observed. Subsequently, a number of design modifications on liner, block and crankcases were made and thoroughly simulated under same loading conditions. This way a final proposal giving acceptable deformation was reached. The results of the final proposal, when compared to base design, indicated a useful reduction in deformation in the localized liner area without affecting stress and stiffness in other areas. The effect of modifying the cylinder head design on liner deformation was also studied. This paper may serve as a good reference for engineers who are targeting to reduce liner deformation in two wheeler engines.

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INTRODUCTION

The deviation of a cylinder bore from an ideal cylindrical shape is termed as bore deformation and is a major influence on the piston dynamics. The piston and the ring pack is made to run in a perfect cylinder, however, in practice the cylinder bore gets distorted due to application of bolt clamping, thermal loading, manufacturing deviations etc. To simplify the nature of bore distortion, it is mathematically understood in terms of harmonics. A perfect circle with change in diameter is a 0th order harmonic, a shift in the center of circle is 1st order deviation, an elliptical shaped bore is termed 2nd order, a three lobed circle is known as 3rd order deviation and so on. These distortions occur in harmonic form and each of these orders represents a sinusoidal curve. The deformation found in an engine is generally a combination of several distinct orders and can be

calculated based on the contribution of each of the order in terms of magnitude and phase using Fourier transforms [1]. Figure 1 shows various patterns of bore deformation in a discrete manner.

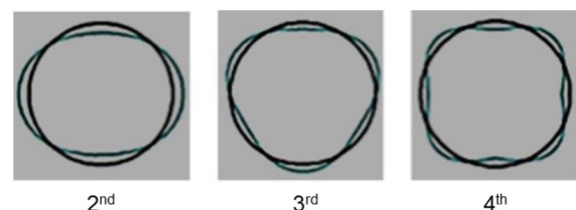


Figure 1. Distinct orders of Bore Deformation

In an engine, there are several bores which are prone to deformation and are critical for its proper functioning, especially.

1. Cylinder Liner Bore (Figure 2 (a))
2. Connecting Rod Big/Small Ends Bore (Figure 2 (b))
3. Crank Shaft Bearing Insert Bore (Figure 2 (c))

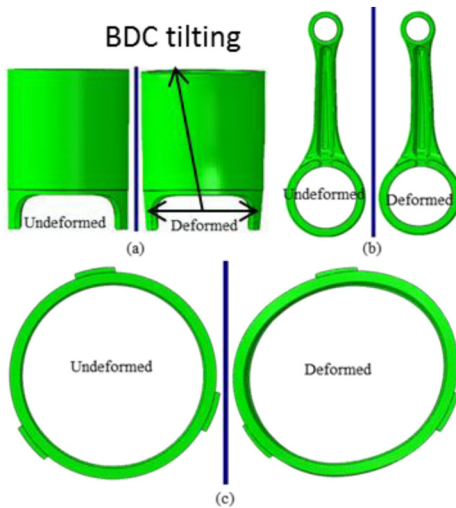


Figure 2. Example of Bore Deformation on Various Engine Subsystems

Here, in this paper the discussion is restricted to liner bore deformation which is considered to be very essential for smooth operation of an engine. It may cause blowby, piston scuffing and excess oil consumption if left unattended [2].

In smaller engines distortion of 2nd, 3rd and 4th order are found (Figure 1), but the bores of bigger engines with multiple bolts, e.g. 6 to 12 in number may experience even higher orders particularly with flexible liner design. Generally, the bore is not an ideal circular cylinder due to the following conditions:

1. In manufacturing it is difficult to make a completely circular bore (which may induce 2nd order deformations - ovality)
2. Finishing and machining processes such as honing may induce another form of deformation (3rd order deformations)
3. Deformations induced during fitment; e.g. fastening cylinder head bolts in smaller engines (4th order deformations)
4. Other higher order mechanical deformations may be induced during operation when subjected to dynamic stresses
5. Deformations induced by thermal stresses

Diesel engines with high bolt torquing and greater in cylinder combustion pressure are often affected by bore deformation. However, parts of a diesel engine are inherently heavier and stiffer and are mostly used for larger vehicle applications where space restriction and thus material addition may not be an immediate constraint. Two wheelers, on the other hand, are in general driven by fast gasoline engine with lighter parts and comes with the added limitation on space and packaging. Hence, any kind of excessive bore distortion in two wheelers asks for a smart and effective design change which will reduce the deformation without increasing the engine weight excessively. This paper is aimed to provide a holistic overview on the iterative design methodology undertaken to reduce excessive liner deformation for a high cubic capacity gasoline engine.

THEORY

Tomanik [1] has explained if Δr is radial deformation from the circular bore, A_i , B_i are the Fourier co-efficients, θ is the angular position and i denotes the order of deformation; then mathematically the deviation from a circular shape of the bore can be defined by Fourier series:

$$\Delta r = A_0 + A_1 \cos(\theta) + A_2 \cos(2\theta) + \dots + A_i \cos(i\theta) + B_1 \sin(\theta) + B_2 \sin(2\theta) + \dots + B_i \sin(i\theta) \quad (1)$$

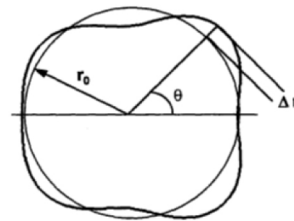


Figure 3. Cross Section of a Deformed Bore [1]

For every i^{th} order, the maximum deformation of the cylinder u_{bi} and the phase angle ϕ_{bi} (Figure 3), where it occurs, can be calculated using the Fourier coefficients:

$$u_{bi} = 2\sqrt{A_i^2 + B_i^2} \quad (2)$$

$$\phi_{bi} = (1/i) \arctan\left(\frac{B_i}{A_i}\right) \quad (3)$$

Then, using a harmonic series the radial cylinder deformation (Δr) can be described as

$$\Delta r = A_0 + \sum_{i=1}^n \left[\frac{u_{bi}}{2} \cos\{i(\theta - \phi_{bi})\} \right] \quad (4)$$

EXPERIMENTAL SETUP

Over the years, experimental studies have been conducted on measuring bore deformation in the test bed. Fujimoto et. al. [3], for instance, used a piston with a gap sensor turned by an ultra-low speed gear mechanism to measure the ever changing gap between the sensor and cylinder bore. Tomanik [1] measured bore deformation using a PAT Incometer equipment. More recently, Patwardhan et. al. [4] has also measured bore deformation in similar terms, and achieved good correlation with their FEA model. All of these studies, however, were based on diesel engines. Experimental examination of the effect of different surface morphology of cylinder bore on bore distortion and scuffing has been conducted by Uhara et al. [5]

The high cubic capacity two wheeler gasoline engine considered for this study underwent several endurance tests in the engine dynamometer and the liner and piston were found to have consistent BDC tilting (Figure 2 (a)) and scuffing marks which eventually caused seizure. This was found to happen within a very small run time for even all new engines when operated under high load.

condition. Further investigation showed that even in engine assembly non-firing condition, the liner was deforming higher than expected. Hence, it was decided to initially reduce the distortion for bolt pretension loads only.

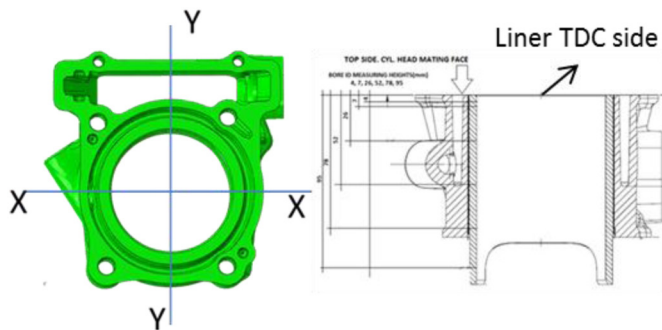


Figure 4. Testing Measurements

Accordingly, liner distortion was measured by probing the liner inner periphery using a bore gauge. The measurements were taken at different sections (4, 7, 26, 52, 78 and 95 mm height) starting from the liner top and in two perpendicular directions (shown in Figure 4 as XX - thrust-anti thrust side and YY - chain sprocket side).

For each block and liner sample, measurements were taken three times. Firstly, before assembly the liner bore was checked for deformation and first set of reading was noted. The engine subsystems comprising of all components such as head, block, liner, gaskets, crankcases etc. were then assembled. Crankcases were cut into half for easy access of the probe into the liner in assembly condition. Four studs and two bolts connecting the head, through the block to the crankcases were given a tightening torque equivalent to 5.2 kgm and 1.1 kgm respectively. Once assembly was done, the second deformation reading was probed and noted. Subsequently, a third bore deformation measurement was taken after disassembling the engine again.

The difference between first two set of measurement is attributed to engine clamp load and is of area of interest in this study. This difference has been plotted in Figure 5. The third set of reading is matched up against the first measurement to ascertain the elasticity of deformation induced in the liner due to bolt clamping. This method was particularly used to eliminate the manufacturing deviations, and confirm the permanent nature of the deformation. This process was repeated for a large number of block and liner samples and similar data was collected. The test data of deformation for a block-liner sample due to bolted engine assembly alone, at various liner depths are as shown in Figure 5. The deformation is plotted from the liner top to bottom with liner TDC side marked as zero.

The measurement revealed maximum contraction of 35 μm at the bottom end and elongation of almost same magnitude at the top end of the liner along XX plane. These values are at par with the block to liner cold clearance provided for this engine, which is in the range of 30 μm to 55 μm . This, in conjunction with thermal loads may have caused or worsened the BDC tilting marks on the piston on its thrust-anti thrust face. Therefore, the magnitudes of the measurement results indicated excessive bore deformation and demanded for a

quick solution to the problem with minimum number of prototyping. To facilitate the same, it became vital to use simulation techniques to work out a design improvement.

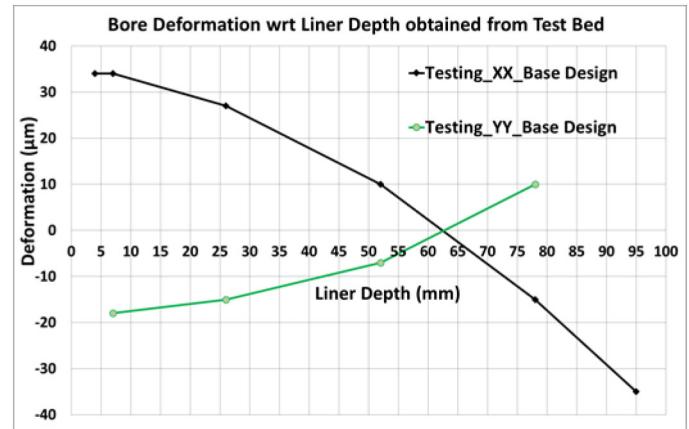


Figure 5. Test Data for Bore Deformation of a Base Block - Liner Sample

FE ANALYSIS OF ENGINE BASE DESIGN

FE analysis has always been useful for simulating large number of design iteration within a short time span. Abaqus, a well-known FEA numerical code was used to analyze all the design proposals referred in this paper. The prime advantage of using ABAQUS is that its robust algorithms are capable of handling geometric, material and contact non-linearities in FEA model. At first, the base engine model was prepared in CAD modelling software with accurate tolerances (Figure 6). Crankcases were cut in half and considered in simulation to replicate exact test bed conditions.

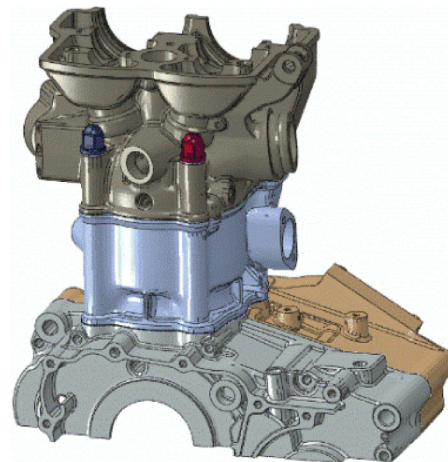


Figure 6. CAD Model

Numerical Discretization or Meshing

Fine meshing (Figure 7) required to capture micron level bore deformation was done with second order 10 node quadratic tetrahedral elements for all components such as head, block, liner, crankcases, studs, nuts and bolts except the gaskets. Gaskets between head and block and between block and crankcases were carefully meshed with 6 node linear triangular prism elements and 8 node linear wedge elements. This was done to closely match the gasket

compression behavior and avoid any kind of mesh interference between the mating components. Owing to high nonlinearity of the problem, mapped mesh was ensured between all the mating parts to avoid penetration error while solving. Mesh element size and number of elements on the liner was kept in such a way that the XX and YY plane and various liner depths in the simulation model exactly corresponds to test measurement planes and heights; thus enabling a direct comparison between the two methods.

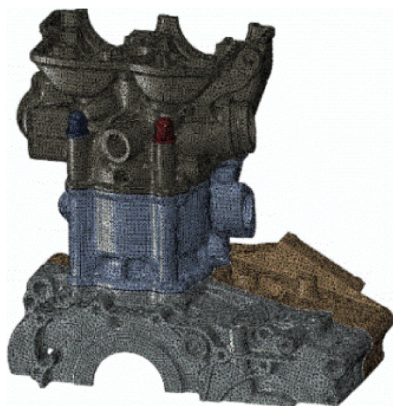


Figure 7. Meshed Model

Material Properties

The completely meshed engine model was fed into the FEA simulation code. Material properties were defined in the pre-processor as per Table 1. Material non-linearity was considered for ADC12 and Steel components as per various standards and available in-house metallurgical testing data. It was tabulated as a plastic stress versus strain curve and fed into the materials database of the FEA code. Table 2 represents the material assigned to each component as per design requirement.

Table 1. Material Properties

Serial Number	Materials	Properties			
		Density (tonnes /mm ³)	Elasticity		Plasticity
			Young's Modulus	Poisson Ratio	
1	ADC12	2.7e-09	71000	0.33	Defined
2	Steel	7.8e-09	210000	0.3	
3	Cast Iron	7.2e-09	130000	0.3	Not Defined
4	CAF 1111	7.0e-10	4800	0.45	

Table 2. Properties assigned to different components

Serial Number	Components	Material
1	Cylinder Head, Cylinder Block, Crankcases	ADC12
2	Bolts (2 in numbers); Studs, Nuts and Washers (4 in numbers), Gasket between Cylinder Head and Cylinder Block	Steel
3	Cylinder Liner	Cast Iron
4	Gasket between Cylinder Block and Crankcases	CAF 1111

Contact and Boundary Conditions

Two forms of contact, namely interactions and ties were used to define the model. Interaction is defined as a contact which allows relative movement between two mating surfaces, whereas, ties are contacts in which there is no relative motion and the mating surfaces are locked to each other. An algorithm with a strain free adjustment for all nodes was used to define interaction properties. Bolt flange was modeled with surface to surface sliding interaction associated with a judicious selection of co-efficient of friction to better evaluate the flange pressures at the bolt locations. The bolt threads were made to tie with the corresponding mating part. Sliding interaction was defined between gasket layers and head, block and crankcases with surface to surface formulation (Figure 8).

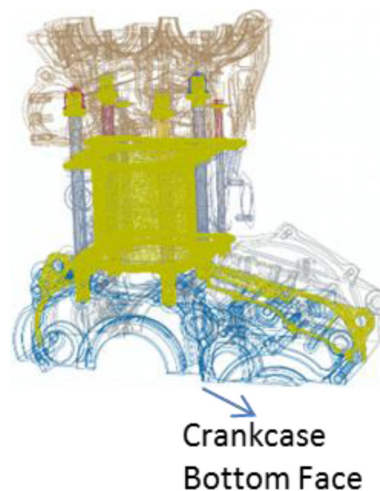


Figure 8. Contact Definitions

Soft springs with a stiffness value of 1 unit were defined in all components so that the simulation solver can achieve initial stabilization. The crankcases were held from a set of nodes at their bottom face and all degrees of freedom were arrested to give fixed Boundary conditions only to those set of nodes.

Load Case

A single static load case with bolt pretension load was used for the analysis. The four M10 studs were given a force equivalent to 5.2 kgm torque at their midsections [6]. Similarly, two M6 bolts on the chain sprocket side had a force equivalent to 1.1 kgm applied on them (Figure 9).

RESULTS FOR THE BASE DESIGN

The stress on block and liner was within the yield stress limit. Hence, they were thoroughly checked for deformation alone. As mentioned previously, the liner was meshed in a fashion so as to ensure accurate measurement of bore deformation at nodes lying exactly on XX and YY planes as was considered in testing.