Cylinder Bore Deformation Finite Element Analysis Methodology for Inline Engines and Parametric Optimization

Arvind J. Bhosale, Kiran B. Bansode, Ravi P. Kakde, Vijay Y. Dhepe, Sanjay D. Patil

***Abstract*: Cylinder bore deformation plays an important role between piston and cylinder bore. The cylinder bore deformation may happen due to manufacturing process tolerances, assembly loading, thermal loading and dynamic gas loading. It has significant impact on the friction, fuel economy, lubricant oil consumption, reduction in power, increase in emissions, uneven power generation in different cylinders etc.**

This work proposes FEA analysis procedure that could be used to evaluate the cylinder bore distortion due to assembly (clamping) load of cylinder head bolts. Cylinder bore deformation due to head bolt clamp loading is predicted using FEA analysis package MSC NASTRAN for 3 cylinder inline Diesel engine. The effect of various parameters like effect of increase in pretension force on amount of bore deformation are studied. The predicted results are compared with previous experimental results and fair agreement have been observed.

***Keywords* : Bolt Pretension; Cylinder Bore Deformation; Finite Element Analysis; Fast Fourier Transform.**

I. INTRODUCTION

In Today's automotive industry, focus on detailing is ever increasing. Consumers require vehicles with higher level of performance, higher fuel economy, lower emissions, higher reliability and most importantly lower cost. Today's passenger cars are need to be designed to operate at higher rpm with higher temperature loading. The ever increasing demands of IC Engine needs design to continuously move towards development of higher specific power, operate at higher fuel economy, engine downsizing and lower fuel emissions.

It is impossible to manufacture perfect round and straight cylinder bore although it is mostly assumed perfect. But whenever the cylinder bore is subjected to assembly loading, thermal loading of engine due to higher operating temperatures, the cylinder bore is likely to deform. The challenge is to understand how cylinder bore deforms and develop methodology to optimize/reduce it.

Typically, cylinders deform due to three major loading forces viz. Thermal Loading (high operating temperatures), Assembly loading (Cylinder Head, Cylinder block and crankcase clamping) and Dynamic Gas Forces (higher operating pressures). The other deformation is likely to take place while manufacturing but it can be eliminated.

A distorted cylinder bore can influence the lubricating conditions of the piston and piston rings and increase lubricating oil consumption and friction. A cylinder bore deformation calculation method during engine operating was developed. [1]

Reduction in cylinder bore deformation can result in improvement of fuel efficiency and engine friction and blow by emissions. The friction between cylinder bore and piston accounts nearly 23 to 25 % of all engine friction. The cylinder bore deformation measurement and reducing friction are major challenges at stage of engine development. In this research, the analytical and empirical methods for detailed prediction, verification and optimization of bore distortion have been considered for the effective engine operation conditions [2].

For sealing purpose and reduction in friction between cylinder bore and piston, piston rings plays important role. For proper sealing of cylinder bore/ring piston ring confirmability is very important. Here confirmability means capability of the piston ring to confirm to the shape of deformed bore. In this, the clearance between the piston ring and bore should be at zero by elastic deformation of piston ring. Good Ring confirmability ensures reduction of oil passing to combustion chamber and blow-by. Experimental measurements were carried out to verify the simple conformability criteria, and a semi-empirical equation was proposed to calculate the limit of piston ring conformability [3].

Undesirable cylinder bore out-of-roundness can badly affect the sealing functionality and durability of the piston-rings in an engine. The effects of changes in cylinder bore out-of-roundness has been described for the rotation of the top and second compression rings, as well as oil consumption in an open-deck 4.5-L V-8 engine [4].

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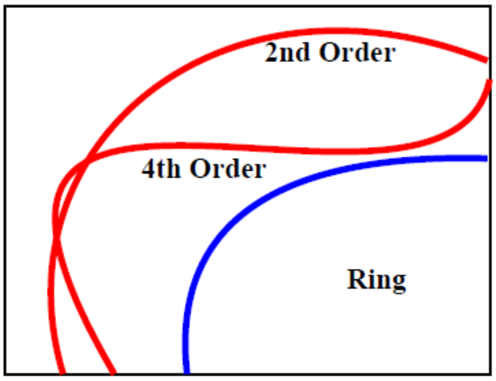

Fig. 1. Ring Confirmability

Rotational dynamics of piston rings are also affected by bore out of roundness. The degrees of piston ring rotation cause either non uniform wear of bore or excessive wear on ring side, if there is continual rotation. Alignment of ring gap is considered to significantly oil consumption.[05]

FEA packages like NASTRAN[6], ABAQUS[7], ANSYS[8] has been used by different authors for cylinder bore deformation analysis. ABAQUS was used as a solver to consider frictional force between cylinder head and block and non-linear spring characteristic of the gasket[1]. Some simplification is done in the geometry of the model in order to minimize the modelling time, processing time and memory required. Instead of the actual cylinder head, a dummy cylinder head was used[9], thickness of which determined by experiment.

Thermo-vision system has been used[10] to measure the temperature at different portion of the liner and engine block. The measured temperature matched with the temperature calculated using Woschni's universal heat transfer coefficient model and from the heat flux measurement.

II. CAUSES OF BORE DEFORMATION

2.1 Manufacturing tolerances

It is not possible to manufacture the component with zero tolerance value. Some tolerances are always provided at design stage only. No doubt these tolerances are at micron level, but it results in deviation from ideal circular shape of bore.

2.2 Assembly loads

Pretensioning of cylinder head bolt is required to avoid gas leakage during operation of engine. The ratio of clamp load to combustion forces varies from 2 to 3 times, depending upon the sealing surface temperature and the land between the cylinders. The forces applied to the liner through the block and the gasket does not transfer evenly. This unbalance of the forces causes some deformation in the liner shape. This is also called as static cylinder bore distortion.

2.3 Thermal loading

During operation of engine, due to increased temperature of cylinder block, head causes to expansion of material. Now a days in order to reduce overall length and weight of the engine, mostly all engines have siamesed type construction without cooling passages between adjacent cylinder bore. In this type of construction very high temperature is generated at the land between cylinders at operating condition. Due to construction of cooling water passage and assembly of head and other accessories the material is not free to expand in all direction evenly. This causes the cylinder to deviate from circular shape.

2.4 Dynamic gas forces

During operating condition of engine very high pressure is acting on the cylinder head and the area near TDC. This gas forces also plays a role in creating the deformation of cylinder bore.

III. METHODOLOGY USED

For predicting the cylinder bore deformation at early stage of engine development following methodology has been used.

3.1. Geometry Modelling

Engine CAD model was developed as per design in Pro-E software package.

3.2. Assembly Load Calculation

Assembly load (Bolt Pretension) calculations were done based on engine design.

3.3. Finite Element Analysis

1. Import CAD model in Altair Hypermesh.
2. Mesh generation (Tetra Meshing)
3. Application of Material properties and boundary conditions. (MSC PATRAN)
4. Finite element analysis using MSC NASTRAN (Solution 400- Non Linear Static and Dynamic Analysis)
5. Post Processing in Altair SimLab Software Package.

3.4. Calculation of different parameters

3.5. Analysis of data

1. Analyzing different parameters affecting bore deformation

3.6 Parametric Optimization study.

2. Cylinder Bore Deformation Finite Element Analysis

IV. FE ANALYSIS

4.1 SOLUTION 400 (NON-LINEAR STATIC AND DYNAMIC ANALYSIS)

SOL 400 analysis allows for seven analysis type combinations (nonlinear single physics, nonlinear chained physics, nonlinear coupled physics, linear perturbation analysis, standard linear physics, nonlinear chained analysis with mesh/time change physics, and nonlinear response optimization with ESLNRO (Equivalent Static Loads Nonlinear Response Optimization).

SOL 400 is widely used in industries for analysis like heat transfer analysis, deformation analysis, contact analysis etc.

4.2 FE Model Mesh Generation

FE Package Altair Hypermesh was used for meshing. In this work, second order mesh is preferred for increasing accuracy and precision. These types of elements offer the benefits of ease of modelling and a higher degree of accuracy per element. The following figure shows the meshed engine model with cylinder block, cylinder head and gasket and bolts. In case of gasket, hexahedral mesh was preferred due to solver software (MSC NASTRAN) requirements. Because in SOL 400, MATG card considers the gasket as solid continuum composite and due to thickness direction anisotropy, hexahedral mesh is supported for gasket behaviour. The figure below shows the tetra meshed model considered for this study.

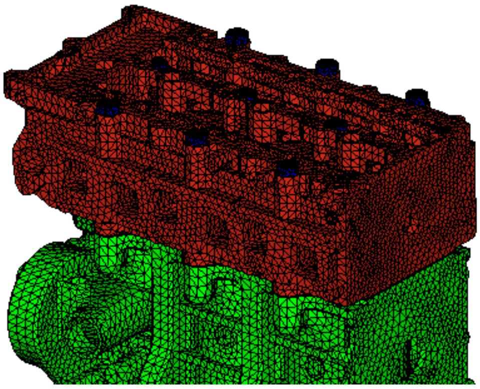

Fig. 2. FEA model for cylinder bore distortion analysis

4.3 Bolt Pretension Load Calculation

During the assembly of the cylinder head, cylinder block and gasket, torque is applied on the bolts for holding the assembly, which creates a tensile load on the bolts, this is called as "Bolt Preload".

At the design stage, the bolt preload is calculated in such a way that, there is always a minimum clamp force in the joint to prevent slipping or separating at the interfaces of the joint members when the operation force(in our case combustion gas force) is applied. Since an applied torque preloads most of fasteners, the externally threaded fastener is subject to tensional stress. The input torque applied to the bolt head or not (whichever is turned) results in three different reaction torques. These are associated with bearing face friction, thread friction and bolt torques. The proportions are typically defined as 50%, 40% and 10% respectively.

The following formula has long been used to calculate the tightening torque for fasteners.

$$T = \frac{KDF}{1000} \qquad \text{...eq. (1)}$$

The simplified base equation (1) can be applied to linear elastic clamping zone of the assembly tightening process. The prevailing torque during initial rundown must be subtracted from the total applied torque, a procedure that is usually accurate for all practical purposes, if sufficient torque is applied to tighten beyond the alignment zone, the tension estimated for a given assembly will still be correctly predicted if the overall K factor has been accurately estimated. There are also published tables of K factors for various combinations of materials, surface finishes, coating and lubricants

Table I. The considered loading parameters for study.

Torque Range	Min	138000	Nmm	
	Max	148000	Nmm	
Minimum Preload applied for the given application	Min	56636.61	N	
	Max	61920.42	N	
Minimum Residual load on each bolt		41483.25	N	
Maximum Residual Load on each bolt		45667.06	N	acceptance criteria

4.4 Loading and Unloading curve for cylinder head gasket

Fig. 3 shows the characteristics of typical cylinder gasket loading and unloading curves which are used in FE analysis. Multi-Layer Steel gaskets (MLS) are typically made of 3 or 4 embossed steel layers with or without a stopper. A 3 layer non-stopper design is typically used when there is a stiff head and for lower peak combustion pressures. As the flexibility of the cylinder head increases and/or as the peak cylinder pressure increases, gasket designer is forced to choose a 4 layer or a 3 layer stopper design. Adding a stopper to the design, provides a combustion gas leak-free seal but it also increases cylinder bore distortion as more of the bolt load goes through the stopper. So an optimized design would have to account for these two conflicting design requirements, sealing requirement versus bore distortion requirement.

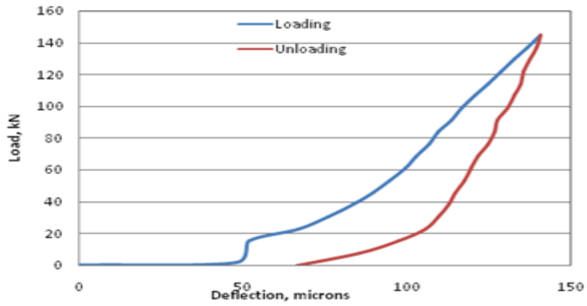

Fig. 3. Loading/unloading curve for MLS gasket.

V. RESULTS AND DISCUSSION

The FEA structural analysis calculates bore surface displacements at each node. To extract bore distortion data from FEA results, the first step is to calculate the least square circle for each level of the bore. The block bore distortions will be analyzed on a layer-by layer basis. The original deformation plots are shown figures (Fig. 4 to 6) below are for Cylinder bore 1 to 3 respectively. These deformation values are calculated at various down bore levels of the cylinder bore such as top of bore, at 32 mm and 60 mm from TDC for each cylinder.

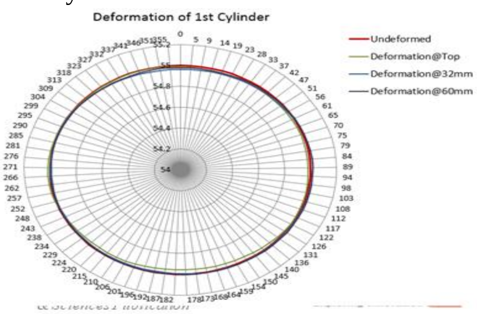

Fig. 4. Deformation plot of cylinder bore 1.

These values calculated at different down bore levels will be used to calculate the harmonic orders coefficients.

These plots unveil valuable information regarding behavior of the cylinder liner to the cold static deformation and effect of different design parameters on the deformation pattern.

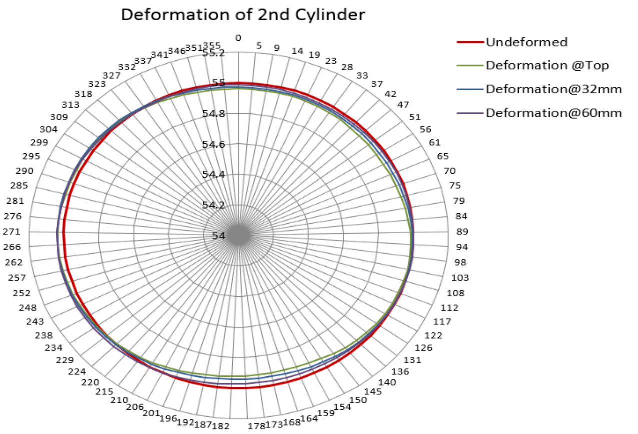

Fig.5. Deformation plot of cylinder bore 2

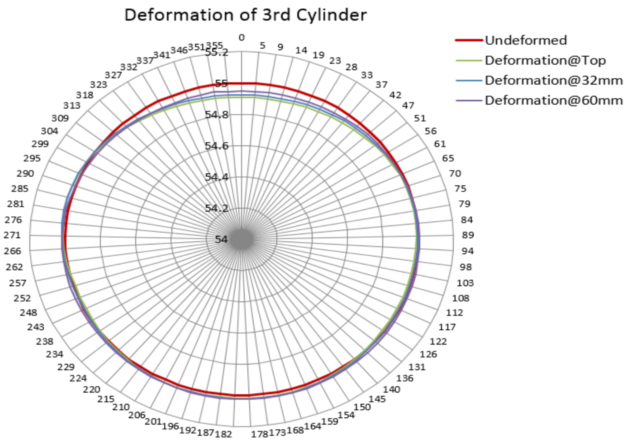

Fig.6. Deformation plot of cylinder bore 3.

The 0th order and 1st order can have little influence on piston ring conformability since they only represent the expansion (or contraction) and rigid body motion of the bore.

The 2nd, 3rd, and 4th orders are of the most important terms regarding ring conformability. In the Fig.7. below, 2nd harmonic deformation order is shown.

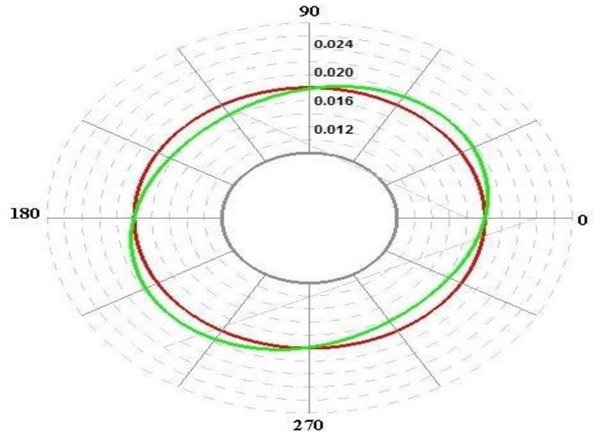

Fig.7. 2nd harmonic order for the top layer of the cyl.bore 1

It can be observed from the figure above, deformation shape is oval and the maximum deformation is about 20 micron. The 2nd order deformation is primarily caused due to bearing carriers used in the cylinder block. In cylinder bore deformation analysis 2nd and 4th order coefficients are predominant.

The following Fig.8 shows the 2nd harmonic order deformation at different levels of the cylinder bore 3 from top.

It can be seen from the figure below that, the deformation at each level is different. This may be due to presence of cylinder water jackets in the body of cylinder block which in turn reduces the strength of the block.

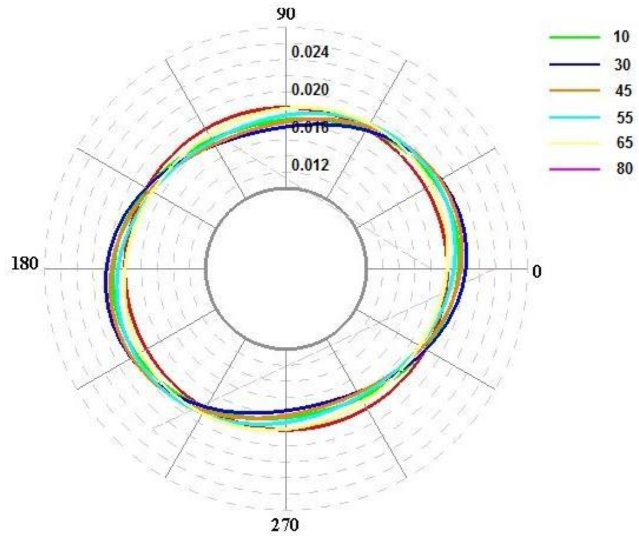

Fig.8 2nd harmonic order at diff. layers of the cyl. bore 3.

In Fig. 9. 3rd harmonic deformation order is shown. From figure, it can be observed that 3rd order deformation produces three lobed profile. The unsymmetrical supports on the outside of bore of cylinder block cause 3rd order deformation. The 3rd harmonic order deformation value is ranging from 5 to 15 microns.

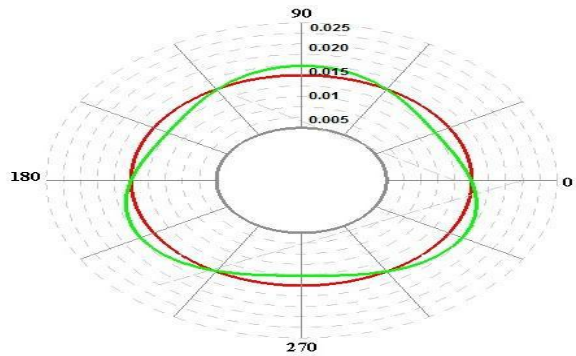

Fig. 9. 3rd harmonic order for the top layer of the cyl. bore 1

The cylinder bores are usually surrounded by four symmetrically placed bolts parallel to the bore axis. So when these bolts are tightened, they tend to produce a four lobe distortion pattern, which is shown in Fig.10. The 4th harmonic order deformation value is ranging from 4 to 6 microns

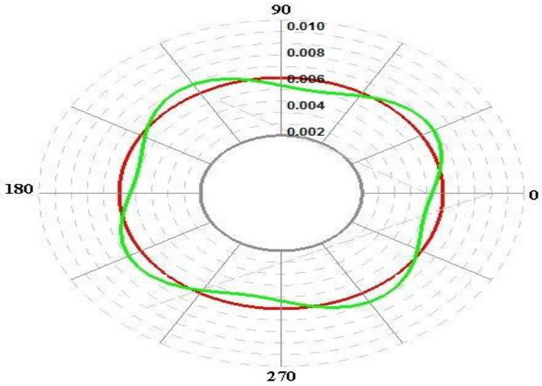

Fig.10. 4th harmonic order for the top layer of the cyl. bore 1

Similarly, the 6th harmonic order is shown below in Fig. 11. For a six bolt pattern in heavy duty diesel engine, will generally create 6th harmonic deformation order.

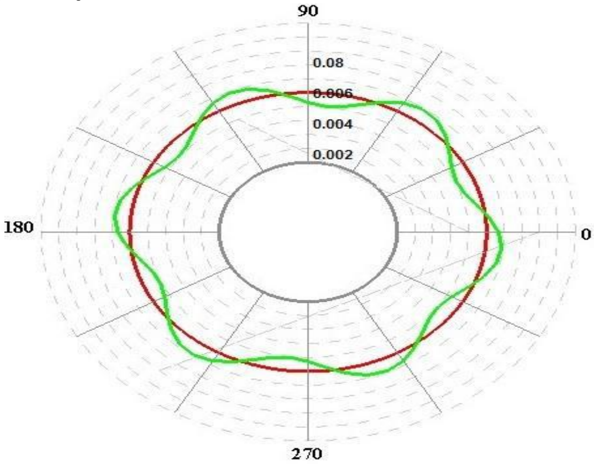

Fig. 11. 6th harmonic order for the top layer of the cyl. bore 1

Likewise, deformation orders for cylinder bore 2 and cylinder bore 3 were obtained. But the deformation values for different cylinders were different. Also, deformation orders for different levels of cylinder bore were obtained from FFT. The following Fig. 12. shows the three dimensional view of 2nd order bore deformation of cylinder bore 2.

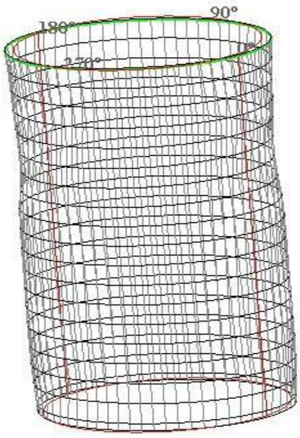

Fig.12. 3D view of 2nd harmonic order of the cyl. bore 2.

From the following Fig.13, the effect of increase in pretension force on 2nd deformation order can be seen. The 2nd order deformation at top layer of the cylinder bore is considerably dominant. The deformation decreases as the distance from top of the cylinder bore increases. So from figure it can be concluded that, the increase in pretension force causes to increase in deformation values till certain level. After certain height deformation amount decreases.

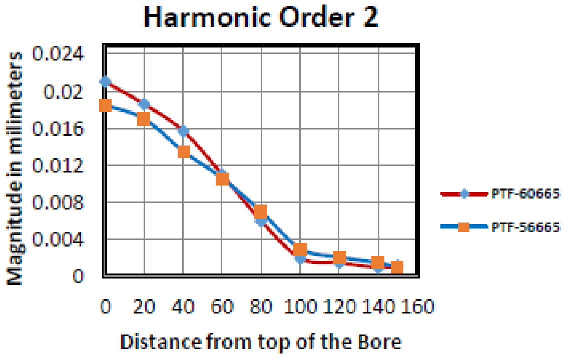

Fig.13. Effect of increase in pretension force on deformation order 2.

In Fig. 14. the effect of increase pretension force on 4th deformation order is shown. As discussed above, the cylinders bores are usually surrounded by four symmetrically placed bolts parallel to the bore axis. So these bolts tends to produce 4th order deformation. Increase in pretension force is more dominant in 4th order deformation. The same pattern of the deformation is found in various literature and fair agreement have been observed.

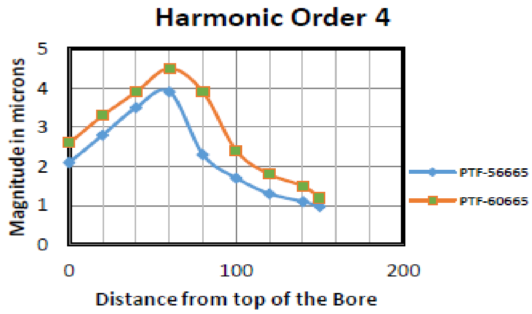

Figure 14. Effect of increase in pretension force on deformation order 4.

VI. CONCLUSION

A methodology for prediction of cylinder bore deformation has been developed using Finite Element Analysis software MSC NASTRAN. The deformation due to cylinder head assembly loads are calculated from the available data.

Also, the effect of increasing pretension on cylinder bore deformation are studied. With increasing pretension, it has been observed that amount of deformation is increasing and the fourth order deformation is more sensitive compared to second order deformation.

The parametric study can be evaluated more effectively from the following table. The table shows the change in value of the second and fourth order deformation for increase in pretension force by 4000 N, and percentage change in deformation with increase in pretension force.

Table II. Comparison of Results

Parameter	2nd Harmonic order (average) in mm	4th Harmonic order (average) in microns
Pretension force-56665N	0.008244	2.1866
Pretension force-60665N	0.008667	2.5812
% change in deformation	4.88%	15.29%

The deformation due to thermal loading cannot be avoided, but it can be minimized by proper cooling. Preventive measures should be taken to prevent the deformation due to assembly loading at minimum possible level. The deformation due to thermal loading is more significant in case of multi-cylinder engine due to constraints faced by intermediate liners for thermal expansion along the plane of crankshaft axis.

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