Cylinder Liner Bore Distortion Estimation During Assembly of Diesel Engine with FEA

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***Abstract* - The trend in an IC engine industry today is toward a shorter product development cycle and faster time to market with increased emphasis on up front analysis to design, develop and optimize a reliable and durable product. The stability of components of engines is essentially affected by the existing inner distribution of stresses. Today, the calculation of stability of critical components like e.g. the cylinder head and the cylinder block is state of art. However, the decisive factor that leads to accurate simulation results is to know the boundary factors as precisely as possible for the calculation. Cylinder bore distortion in IC engines has been identified as a cause for fuel and oil consumption, thus it influences efficiency and emissions. In particular it has a significant effect on the performance of the tribological system consisting of piston, into the crankcase. It also causes an increase in hydrocarbon emissions and can deteriorate the emission control system. Due to these reasons, the evaluation of bore deformation and the optimization of the cylinder bore shape is considered significant subjects in modern engine research. For present project work, the distortion levels in the block due to clamping loads are used to measure with FEA analysis using CAE tool ANSYS. Accordingly can be checked with excremental setups.**

Key words-Bore Distortion, Liner Bore, Engine analysis.

I. INTRODUCTION

Bore Distortion means deviation from ideal circular shape (roundness), or bore out of roundness. Initially it is understood that manufacturing tolerances can cause the bore to deviate from ideal condition.

The main types of distortions that occur in an engine are

1. Cylinder liner bore distortion.
2. Main bearing bore distortion.
3. Connecting rod bore distortion.

Amongst which cylinder liner bore distortion is major as compared to the others. Even with the best of intentions and preparation, an unavoidable degree of cylinder bore distortion is likely to occur under dynamic stress (heat and pressure). The challenge is to understand how these changes take place and to establish procedures that will attempt to minimize these changes. As the block ages and is exposed to thermal changes, the casting's molecular structure will change slightly, which will affect bore geometry. In addition, distinct changes in bore shape will take place as the block is assembled. The clamping forces that result from the installation of cylinder heads will cause the bores to "morph" in most cases as the fasteners pull and squeeze metal adjacent to the bores. Sometimes these changes are insignificant, while in other situations, the changes can be so dramatic as to cause measurable ring dragging and subsequent loss of power, fuel economy and sealing due to both out-of-roundness and frictional heat. In some engines, these problems may be compounded if the specific block is severely affected by additional distortion that results from clamping forces caused by bell housing bolts, water pump bolts, motor mount bolts, etc.

A. *Literature Review*

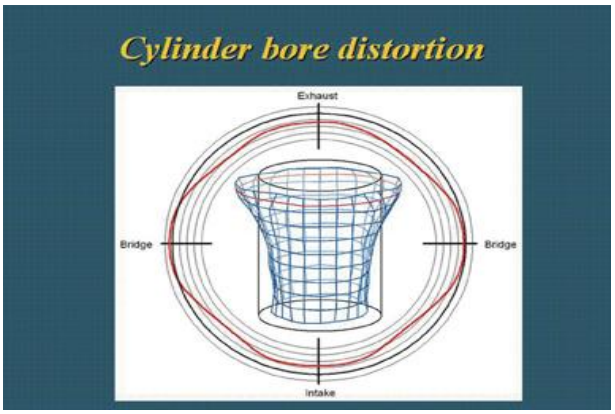
Prof. Paul Gibbs[1] has given the equations and concepts used for "Piston ring conformability". Also he has given the harmonic distortions in cylinder. A first order distortion appears as an offset of the whole section from the main axis, a second order appears as an 'ova ling' of the section, a third as a three lobed deformation, a fourth as four lobes, and so on. Theoretically an infinite number of orders combine to make up the final shape, the magnitudes and directions of these orders are calculated by using the Fourier series and applying it to measurements of the bore. He has given the information about LOTUS tolerances and how these are used for acceptability of bores.

Teimuraz Bardzimashvili,James F. Kelly and Elene Romelashvili et.al.[2] have given the details of piston rings. They have stated the causes of bore distortion such as imperfections in design and assembly, gas

pressure and thermal variations, the cylinder may distort before and during operation. Hence, the circular piston ring must conform to a slightly imperfect cylinder bore. Also they have stated the three equation of GOETZE , Dunaevsky and Tomanik. Another approach to the ring conformability problem is Finite Element Analysis (FEA). In order to test the FE data and evaluate the effectiveness of each bound, wrote a MATLAB program called RINGPACK.

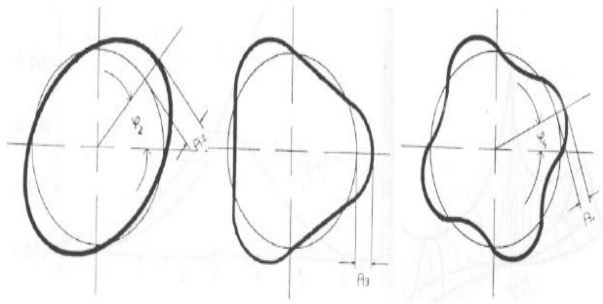
Lawrence E.Bird[4] has described common procedure used to define bore distortion. Orders of distortion are also classified as zero to Fourth order and its effect on geometry of bore liner .Assembly load causes fourth order distortion due to which four lobe deformation of liner takes place.

K.Loenne[5] has explained the causes of deformation. Deviation from the cylinder shape through the production process. The possibility of influencing this factor lies solely with engine manufacturer cylinder distortion through the assembly of the cylinder block with the cylinder head and peripheral components called static initial clamping distortion. These elastic deformations may be due to the design changes to the installed components. The gas pressure acts on the cylinder wall in restricted areas. Distortion of the cylinder due to combustion pressure is significant only in highly rated diesel engines with thin walled wet liners. The magnitude of the thermal expansion is far larger than the deformation caused by clamping bolts.



B. Harmonic Distortions:

This kind of distortion causes a regular, sinusoidal deformation in the cylinder, which can be derived mathematically from measurements of the surface of the cylinder. The 'harmonic' refers to the building up of the real shape from a combination of many sinousidal waves of different orders (see below). A first order distortion appears as an offset of the whole section from the main axis, a second order appears as an 'ovaling' of the section, a third as a three lobed deformation, a fourth as four lobes, and so on. Theoretically an infinite number of orders combine to make up the final shape, the magnitudes and directions of these orders are calculated by using the Fourier series and applying it to measurements of the bore.



(a):2nd order. (b): 3rd order (c):4th order

Causes of Distortions:

The cylinder block, as cast, has only a rough shape with tolerances of around 0.5-1mm, but machining of the surfaces aims to give as perfect a cylinder as possible (first with a rough drilling and then with a polishing/honing process to reach the final tolerances in the region of tens of microns). Even after this there will be a small, but measurable deviation from the ideal shape, but the design of the piston rings allows them to conform easily to these small distortions.

Bolt Loads:

Cylinder heads, bearing shells and bearing carriers all require bolting to the cylinder block and the bolts used will unavoidably deform the block. Heads are the primary cause of fourth order distortions, bearing carriers cause second order and the unsymmetrical supports on the outside bore of blocks cause third order distortions. As expected, the higher the bolt loads, the greater the distortion, but placement of the bolt threads, and the depths to which they sink can affect distortion to a great extent. With cylinder liners, supports, cooling jackets, head gaskets, external mountings and other considerations, the design of blocks to minimise distortion is a complex subject, and is beyond the scope of this report –which aims only to find simple method of measuring, quantifying and comparing distortions.

Temperature:

At present the most accurate measures of the shape of the bore can only be made with the engine disassembled as the harsh conditions during combustion give rise to serious (but not unsurpassable) problems when trying to measure the bore. As a result, most distortions are measured cold (25°C). Since the temperature gradient during operation can range from below zero °C outside

(at start-up) to over 2000°C inside the bore, deformation of the bore by thermal expansion is common (with the greatest expansion occurring, naturally, towards the combustion face of the block).

C. Bolting concept :

The connection between block and head can be done either conventionally or by the through-bolt technique. In case of conventional bolting, the cylinder head and grey iron bearings are bolted directly to the block resulting in high stresses in the vicinity of the thread and the bolt head. This bolting concept reaches its limits in case of high loaded direct injection diesel engines. In order to prevent high tensile stresses in the engine block, the main bearings and the cylinder head can be connected directly by long bolts which penetrate the whole block and head, thus setting them under compressive stress only (through-bolt concept). The drawback of this solution is a more complicated assembly because bearing caps and cylinder head are not any more independent of each other, i.e. the final assembly of bearings and heads has to be carried out at the same time. This problem can be solved by screwing in the through-bolts so that head and bearing caps can be mounted separately while maintaining the load-bearing benefit of the through-bolt.

D. Need Of This Project:

Air-cooled Diesel engine, for instance, is commonly used in heavy-duty transport fleets applications due to their high performance, efficiency, and low fuel consumption. But when bore deformation occurs the following problem may arises:

Blow by:

Blow by is the loss of combustion gases from the combustion chamber, past the piston and into the crankcase. This is undesirable as the gas takes energy with it, reducing the pressure on the piston face and reducing the power output of the engine.

Oil consumption:

If the piston (or piston rings) were in direct contact with the cylinder wall then the combustion chamber would be sealed off perfectly, but it would require a huge force to move the piston against the friction generated. To overcome the friction the piston rings are designed to ride on an oil film, which, although very thin, provides a small amount of friction. If the rings are designed badly, then instead of riding over this film, they scrape it away - either into the combustion chamber or out through the exhaust (giving rise to high hydrocarbon emissions and reducing the life of catalytic converters), or down into the sump where it is recycled through the engine.

Engine seize:

Due to bore distortion in the engine temperature rise occur which causes engine seize problem. so these are the some major problems occurs due to bore distortion. so to reduce these problems it is necessary to find out the bore distortion in initial stages.

And it is the basic motivation of this project.

II. FEM APPROACH

It is not always possible to obtain the exact analytical solution at any location in the body, especially for those elements having complex shapes or geometries. Always matters are the boundary conditions and material properties. In such cases, the analytical solution that satisfies the governing equation or gives extreme values for the governing functional is difficult to obtain. Hence for most of the practical problems, the engineers resort to numerical methods like the finite element method to obtain approximate but most probable solutions.

Finite element procedures are at present very widely used in engineering analysis. The procedures are employed extensively in the analysis of solids and structures and of heat transfer and fluids, and indeed, finite element methods are useful in virtually every field of engineering analysis.

Description of the method:

In any analysis we always select a mathematical model of a physical problem, and then we solve that model. Although the finite element method is employed to solve very complex mathematical models, but it is important to realize that the finite element solution can never give more information than that contained in the mathematical model.

Physical problems, mathematical models, and the finite element solution:

The physical problem typically involves an actual structure or structural component subjected to certain loads. The idealization of the physical problem to a mathematical model requires certain assumptions that together lead to differential equations governing the mathematical model. The finite element analysis solves this mathematical model. Since the finite element solution technique is a numerical procedure, it is necessary to access the solution accuracy. If the accuracy criteria are not met, the numerical solution has to be repeated with refined solution parameters (such as finer meshes) until a sufficient accuracy is reached. It is clear that the finite element solution will solve only the selected mathematical model and that all assumptions in this model will be reflected in the predicted response.

Hence, the choice of an appropriate mathematical model is crucial and completely determines the insight into the actual physical problem that we obtain by the analysis. Once the mathematical model has been solved accurately and the results have been interpreted, we may well decide to consider next a refined mathematical model in order to increase our insight into the response of the physical problem. Furthermore, a change in the physical problem may be necessary, and this in turn will also lead to additional mathematical models and finite element solutions. The key step in engineering analysis is therefore choosing appropriate mathematical models. These models will clearly be selected depending on what phenomena are to be predicted.

Advantages and limitations of FEA:

Planning the analysis is arguably the most important part of any analysis, as it helps ensure the success of the simulation. Oddly enough, it is usually the one analyst's leave out. The purpose of an FE analysis is to model the behavior of a structure under a system of loads. In order to do so, all influencing factors must be considered and determined, whether their effects are considerable or negligible on the final result. The degree of accuracy to which any system can be modeled is very much dependent on the level of planning that has been carried out.FEA is an approximate way of simulating the system behavior. But the results can be quite close to actual testing values. FEA can never replace actual physical testing all the times. This is due to the fact, the information required for FEA simulations, like material properties emanates from physical testing.

FEA results by themselves can never be taken as complete solution. Usually at least one prototype testing is necessary before the design guided/validated through FEA can be certified. But when effectively used FEA can predict the results/behavior quiet close to reality and can reduce the design lead times as well as the number of prototypes to be tested. Also there are some situations like gears in contact, which cannot be simulated exactly using FEA techniques. Under such situations some work around such as simulating the worst condition that can happen can be followed.

Mesh requirements:

The Finite Element Method (FEM) has certain requirements on a mesh:

1. The mesh must be valid, (no holes, self-intersections, or faces joined at two or more edges).
2. The mesh must conform to the boundary of the domain.
3. The density of the mesh must be controllable, to allow trade-off between accuracy and solution time.
4. The grid density will vary depending on local accuracy requirements, but any variations must be smooth to reduce or eliminate numerical diffusion/refraction effects.
5. There are some requirements on the shape of elements. In general, the elements should as equiangular as possible in equilateral triangles & regular tetrahedron.
6. Highly distorted elements (long, thin triangles, squashed tetrahedron) can lead to numerical stability problems caused by round-off errors. This requirement is modified for boundary layers, where highly stretched elements are desired and facilitated in the FEM formulation. The min-max-angle property is still required in this case.

Element mesh Parameters:

Usually there are certain parameters that determine the quality of the results. The Engineer has to ensure that these parameters are maintained to the minimum required levels in the FE model for obtaining good results. These are called mesh quality parameters which are as discussed below:

1. **Warpage**: This is applicable to only Hexahedron and shell elements only. Any three nodes on any face of the Hexahedron element define a plane. If the fourth node on the same face is away from the plane beyond a certain angle, the results of analysis obtained by using these kinds of elements shall be erroneous.
2. **Maximum Angle**: If the obtuse angle between two edges in a Quadrilateral is more than a prescribed limit, then erroneous results can be the outcome of an analysis.
3. **Minimum Angle**: If the acute angle between two edges in a Quadrilateral/Triangle is less than a prescribed limit, then erroneous results can be the outcome of an analysis.
4. **Aspect Ratio**: If the ratio of maximum to minimum length in an element is more than a prescribed limit, the results of the analysis can be erroneous.
5. **Jacobian**: The differential of the matrix obtained during the formulation of a stiffness matrix is called the Jacobian. If the element is formed with a poor shape, this differential can become negative resulting in a negative stiffness matrix. Hence care should be taken to avoid poor shaped elements to avoid Jacobian with low values.

III. WORK DONE -FORCE CALCULATION:

Bolt Pretension Calculation:

The Basic equation considered for bolt pretension calculation is,

$$P_i = T/ (K D)$$

Where, P_i = Bolt preload
T = Bolt installation torque
K = Torque coefficient
D = Bolt nominal size

Torque coefficient K,

$$K= \{[0.5\, p/\pi] + [0.5\, \mu t\, (D-0.75\, p\, \sin\alpha)/\sin\alpha] + [0.625\, \mu_c\, D]\}/D$$

D = Bolt nominal size =16 mm
p = thread pitch=1.25 mm
α = thread profile angle = 60° (for M, MJ, UN, UNR, and UNJ thread profiles)
β = thread profile half angle = 60°/2 = 30°
μ_t = thread coefficient of friction = 0.17
μ_c = collar coefficient of friction = 0.12

For our case K= 0.2233

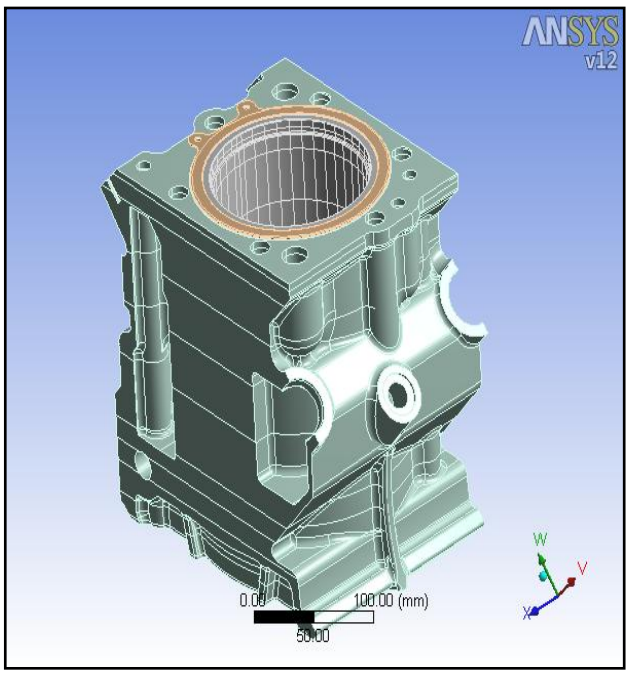
So, Pi = 255060/ (0.2233 x 16)

$$P_i = 255.060/3.57$$

P_i = 71445.37 N – Bolt Pretension Load/bolt

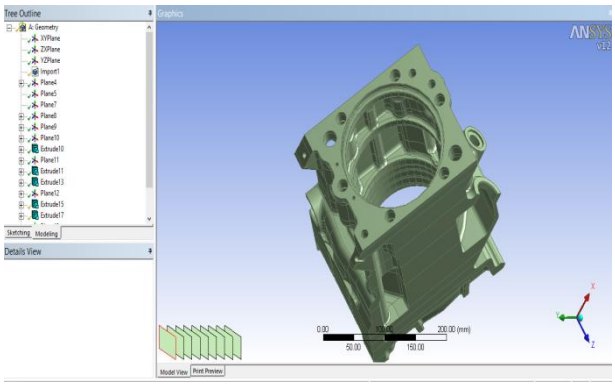
No of Bolts=6

6 x 71445.37=428672 N

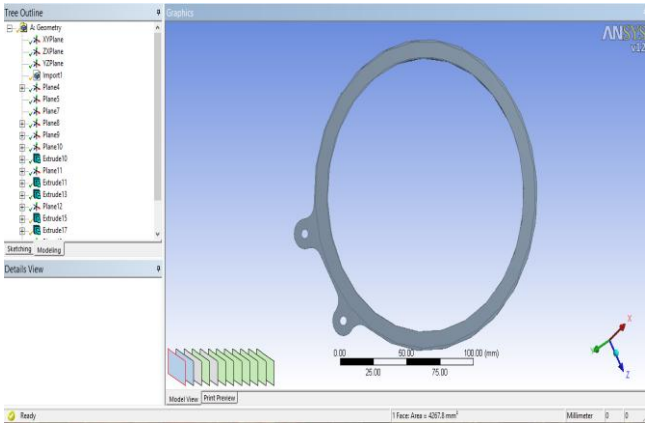


IV. COMPANY MODEL

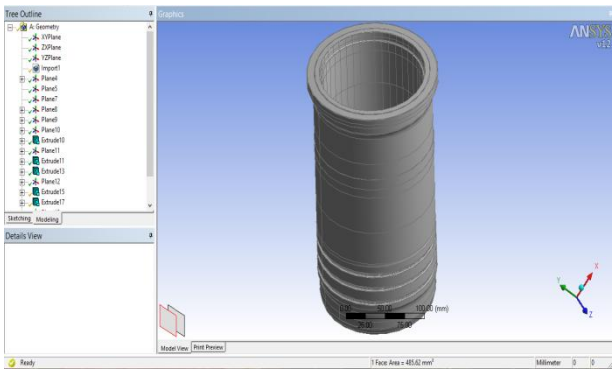
Crank Case:



Gasket:



Liner:



Nonlinear analysis:

There are three types of non-linearity.

1. Material non-linearity
2. Geometric non-linearity
3. Contact non-linearity

Non-linear Structural Analysis:

When structure response (deformation, stress & strain) is linearly proportional to magnitude of load then the analysis of such a structure is known as linear analysis. When the load to response relationship is not linearly proportional, then the analysis falls under non linear analysis. For example when a compact structure made of stiff metal is subjected to a load with relatively lower in magnitude as compared to strength of material, the deformation in structure will be linearly proportional to the load and the structure is known to have subjected to linear static deformation. But most of the time either material behavior will not linear in the operating condition or geometry of the structure itself keep it from responding linearly. Due to cost or weight advantage of non metal (polymers, woods, composites etc.) over metals, nonmetals are replacing metals for variety of applications, which have nonlinear load to response characteristics, even under mild loading conditions.:

- Material non linearity (Strains beyond the elastic limit (plasticity))
- Geometric non linearity(Large deflections, such as with a loaded fishing rod)
- Boundary non linearity Contact between two bodies.

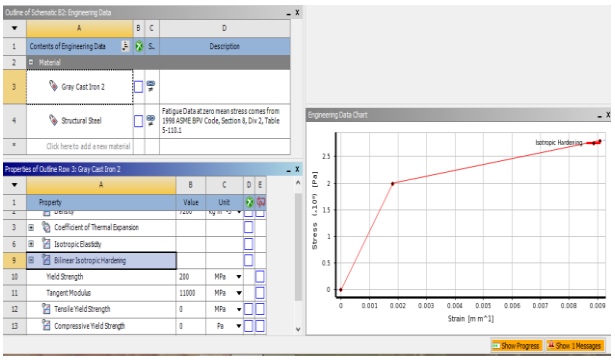
V. MATERIAL PROPERTIES:

Material Property:

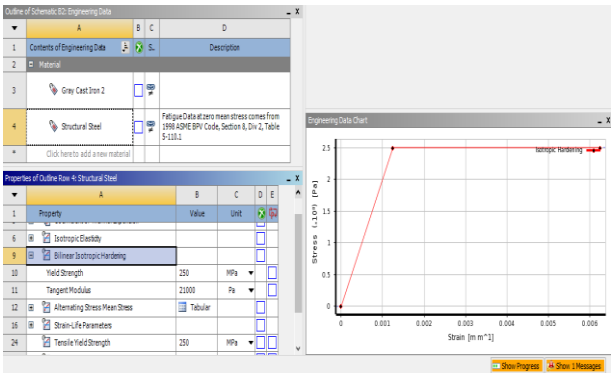
Material property must frequently be approximated in FEM. Indeed of course there is an accepted degree of variation in the young's modulus of even the most standard of engineering material.

Material properties in Ansys:

GREY CAST IRON:



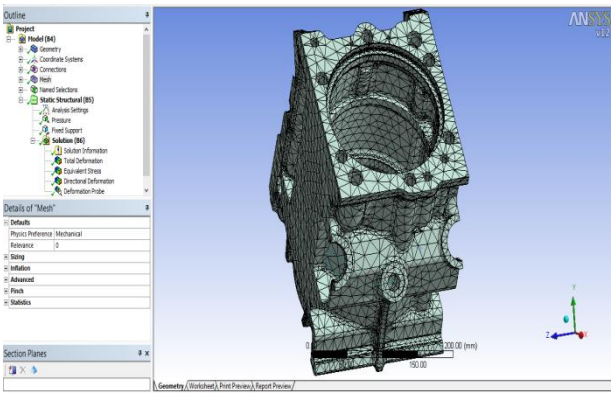
STRUCTURAL STEEL:



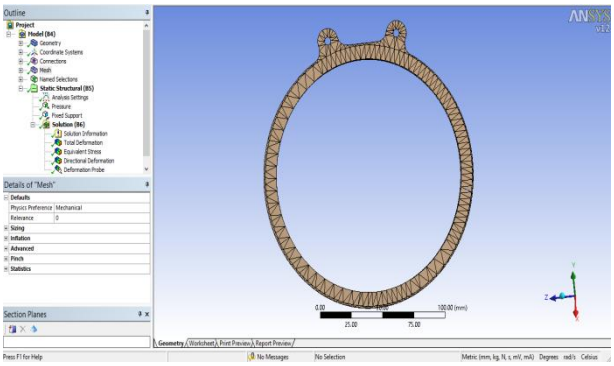
In above problem there are two non-linearity's takes places, material and contact. In material non linearity's there two types again. 1) Plastic non-linearity 2) Intrinsic non-linearity.

MESH MODELS:

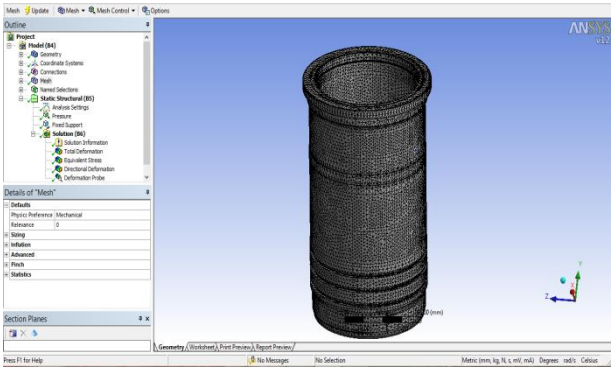
Crankcase:



Gasket:



Liner:



VI. CONTACTS:

Contact needs to be defined wherever necessary. It allows proper transmission of forces. Contact occurs when the element surface penetrates one of the target segment elements on a specified target surface. If more than one target surface will make contact with the same boundary of solid elements, you must define several contact elements that share the same geometry but relate to separate targets (targets with different real constant numbers), or you must combine the two target surfaces into one (both having the same real constant number).

Types of contacts:

1. Rigid-flex:

Bodies of vastly different stiffness

Steel against rubber seals

2. Flex-flex:

Bodies of comparable stiffness

Metal contacting metal

3. Self contact:

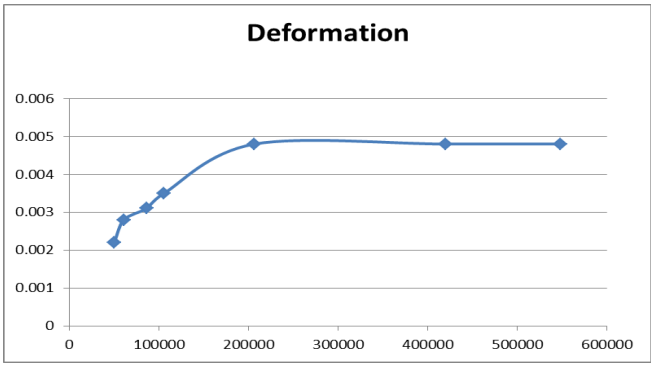
Body folds over itself

Column buckling

Large sliding with friction for all:

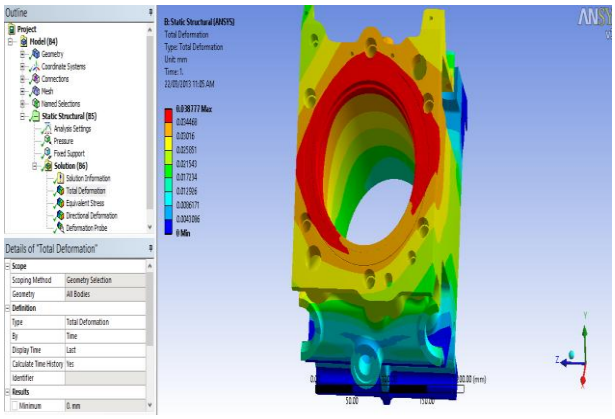
We have four Flex-Flex type contact in the case of cylinder bore distortion analysis. In Flex-Flex type of contacts, both surfaces are flexible in nature i.e. both bodies are having comparable stiffness.

Convergence Study:

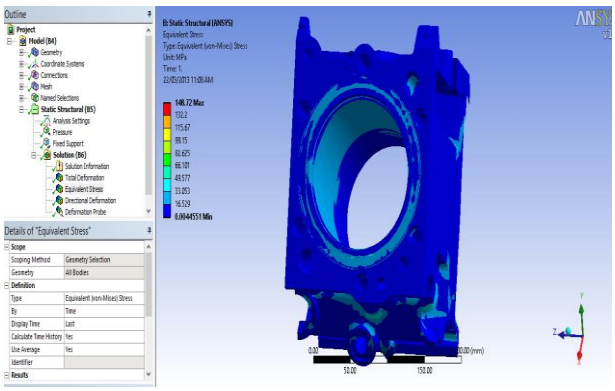


VII. RESULTS AND DISCUSSION:

Overall Deformation:

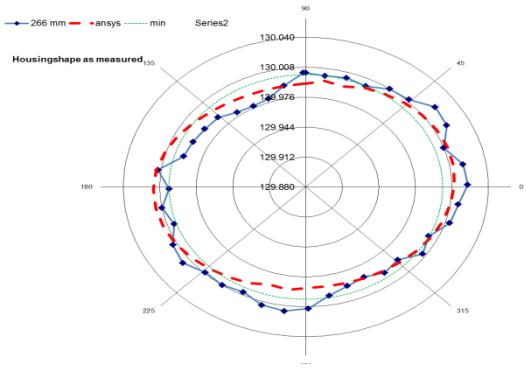


Von mises stress:

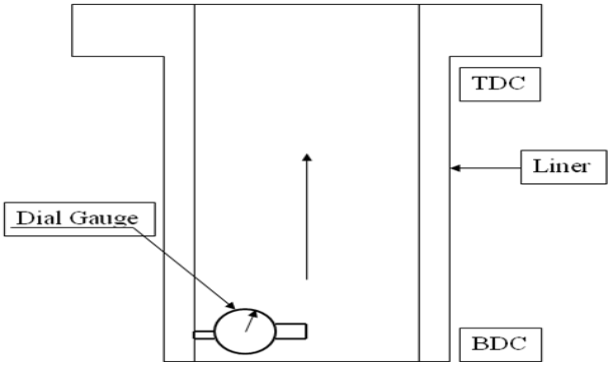


Experimental validation and conclusion:

Above all results were plotted in the excel sheet. In this the comparison of perfect bore with the actually distorted bore and FEM results were takes place. The green bore shows the perfect bore, red shows the experimental measured bore and blue shows the Fem results. As discussed in earlier chapter distortion were measured at 10 planes. The first 5 planes show the much accurate bore shape but as moving towards bottom the shape was not matching as upper portion.



The experimental results are provided by company. Readings are measured at different levels. Maximum Distortion is 15% to 20 % more than Ansys results in Liner Bore.



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