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Analysis of Cylinder Head Bolt Cross Pattern Tightening and Its Effect on Gasket Sealing Performance and Bore Deformation

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ABSTRACT

Cylinder head gasketed joint is one of the most important sealing joints in an I.C Engine. It directly affects the engine performance. Gaskets are very crucial components in providing sealing of combustion chamber to prevent escape of combustion gases and leakage of coolant and oil. Also they prevent the cross flow of the fluids between the various passages in the engine. But to ensure this, it is necessary to have adequate pressure distribution in the various sealing regions of the gasket i.e., at the various cavities or openings in the head and block deck interface of the engine. Apart from providing sealing action, it also affects engine piston ring conformability in cylinder bore, engine blow-by and lubricant oil consumption. Thus careful design of the gasket is very crucial, in order to provide just the optimum loading at the various sealing regions. During assembly of gasketed joint, there is a relaxation of the gasket when the bolts are tightened successively, this leads to reduction of gasket pressure which causes leakages. In order to assess the variation in gasket sealing pressure, non-linear 3-D FEM technique is employed to model a contact problem to find the contact pressure distribution at the various sealing regions of the gasket. Then the results obtained will be validated against the Fuji-Film experimental Analysis.
Keywords: Fuji-Film Analysis, Gasket pressure distribution, Non-linear Analysis, bore deformation

I. INTRODUCTION

During assembly of the cylinder head gasketed joint, there is loss of bolt preload and consequently a reduction of compression force on the gasket when the bolts are tightened successively, this leads to reduction of gasket pressure which causes leakages. In order to obtain the required gasket sealing pressure at the critical sealing areas, it is necessary to combat this problem of bolt preload reduction. It has been observed that the bolt preloads applied for tightening the bolts and the numbers of passes in which these preloads are applied are critical parameters affecting the bolt preload loss, during tightening. [1] Also the sequence or pattern of tightening the bolts joining the mating surfaces, play a major role in deciding the final bolt preload retention after complete tightening. In this paper we found out effect of cross pattern tightening of bolts on gasket pressure distribution by FEA analysis using Hypermesh software and OptiStruct solver and then correlated to experimental Fuji Pressure Film results.

II. OBJECTIVE

- To apply non-linear 3-D FEM techniques to model a contact problem for finding the contact pressure distribution at interfaces of a cylinder head gasketed joint.
- To obtain uniform pressure distribution on Cylinder Head Gasket keeping number of bolts and material of bolts constant.
- To develop a bolt-up strategy including the no. of passes and the tightening sequence needed to achieve the requisite sealing pressure & obtain correlation between FEA & experimental results with tolerance up to ±15%.
- To find effect of pretension gasket pressure distribution

III. METHODOLOGY

- Structural meshed model of Cylinder head, Block, Cylinder Head Gasket, Bolts is imported in Hypermesh v13 Software

- Assigning load, boundary conditions and creating contact surfaces for Bolt. Load collector and load steps are assigned as per bolt tightening pattern.
- After completion, run is given on Optistruct Solver v13
- Post-processing in Hyper view. FEA results are found out for thickness direction pressure distribution on cylinder head gasket elements, thickness direction closure, Contact pressure on gasket elements.
- Fuji Film Analysis (Experimental) is done and Nodal Pressures on Gasket is found out.

IV. ASSIGNING LOAD, BOUNDRY CONDITIONS AND CREATING CONTACT SURFACES FOR BOLT

All the 6 degrees of freedom are constrained to zero for all the nodes on the lower face of the block using SPC load collector for Full Bank Analysis which is shown in fig 1. All the bolts are pre-tensioned using the PRETENSION MANAGER [8] [9], by creating axial force of 70kN (corresponding to 200N-m torque) for single pass at the shank section of the bolt by creating contact surface first. The Bolt Pretensions are sequenced using the CNTNLSB [8] card, to simulate the cross pattern of bolt tightening asshown in fig 2.

Figure 1. Boundary Conditions for Full Bank analysis

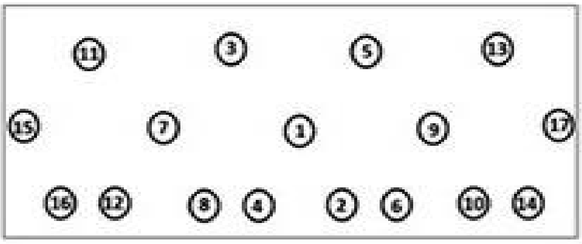


Figure 2. Cross bolt tightening pattern

V. FEA RESULTS OF CROSS SINGLE PASS

The main focus of the Assembly load analysis is on the Gasket Thickness direction Pressure of the combustion bead area of the gasket. The gasket thickness direction pressure is representative of the sealing pressure created at the various bead areas of the gasket, due to the deformation of the beads, on application of the pre-tensioning load. By use of a graphical representation of the pressures along the circumference of the cylinders at every 45°, as it is the most critical sealing area in the engine, the sealing efficiency of the gasket is estimated under different assembling strategies. Also experimental Fuji-film test results are available for the combustion bead only as the high pressure range Fuji-film was not able to capture the results of other gasket regions where the results were below the measuring range of the Fuji-film used. The 0° point corresponds to the exhaust side of the engine and 180° corresponds to the intake side.

VI. RESULTS AND DISCUSSION

The thickness direction pressure for the combustion bead area of the gasket, at the last load increment after tightening all 17 bolts for cross pattern is shown in Fig. 3.

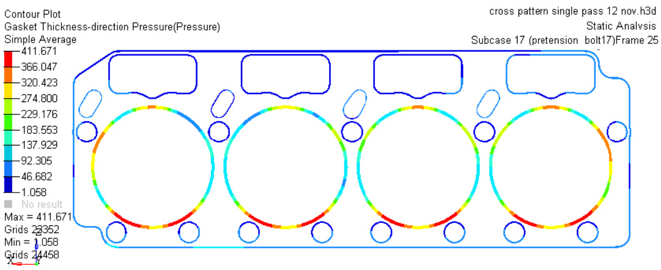


Figure 3. Gasket Thickness direction pressure (MPa) for the various gasket bead areas.

The thickness direction pressure is found maximum at the intake side close to the head bolt location. This may be because of asymmetric geometry of the head. More material volume is on intake side rather than the exhaust side. Also the placement of the cylinder head bolts around the combustion bore area is asymmetric with respect to the longitudinal axis of the cylinder head. This asymmetry in the material distribution in the head and the placement of the head bolts causes uneven distribution of compressive stresses on the gasket. This can be assumed to be a cause for the non-symmetric pressure distribution in the thickness direction of the gasket.

The variation of thickness direction pressure around the combustion bead in the gasket for cylinder bore 1, 2, 3 & 4 is plotted in the graph shown in figure 4. The pressure values are plotted for locations in the combustion bead.

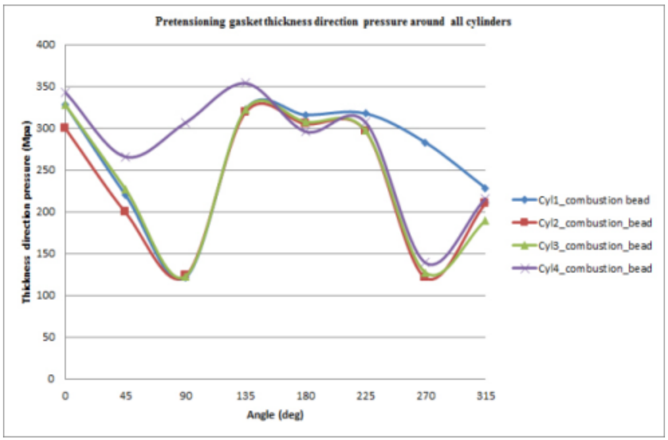


Figure 4. Comparison of gasket thickness direction pressure around Cylinders 1, 2, 3 & 4 in combustion bead area

From Fig. 4, it is observed that for angular position 00, gasket thickness direction pressure values of all cylinder combustion beads are almost same. We move on from 00 angular position to 450, there is gradual decrease of gasket pressure for cylinder combustion beads 1, 2, 3 & 4 because of bolt is away from the bead. Another reason is the bulk material volume of the block supporting this portion and the bulk material of the cylinder head compressing this section of the gasket is narrow, as it lies in vicinity of the water jacket passages. Values of gasket pressure for combustion beads corresponding to cylinder 1, 2 & 3 for region 450 to 900 are same but for cylinder 4 combustion bead it is high. This is because of gasket pressure drop at bridge gap due to cancellation effect. This portion is located at vicinity to bolt. Now as we move from 900 to 1350, gasket pressure increases steadily for all beads. For peripheral region of bead between 1350 & 2250, gasket pressure variation is very small. It has highest value for all combustion beads in this sector due to vicinity of 2 bolts in this region. We move on from 2250 to 2700, the gasket pressure for combustion bead 1 decrease slightly while reduction of gasket pressure for cylinder combustion beads 2, 3 & 4 is quite substantial. For 2700 to 3150 angular position, pressure is high for combustion bead 1 as there is no pressure drop here while for cylinder combustion beads 2, 3 & 4 there is pressure drop at bridge gap due to cancellation effect. Another reason is the section thickness of the cylinder head at the end near the cylinder 1 is higher. This adds more stiffness to the head in that region causing lesser pressure to be developed in the bead area due to less compression of the cylinder head.

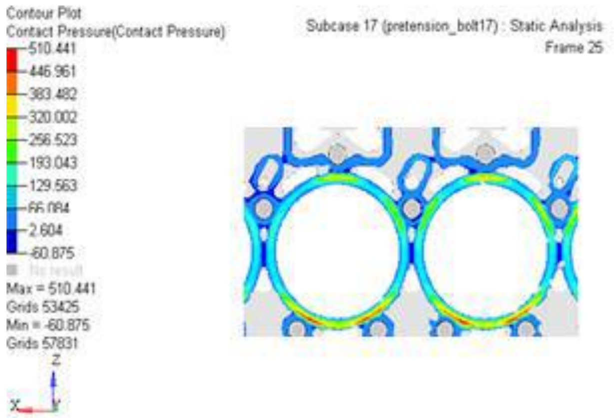


Figure 5. Contact pressure distribution (in Mpa) on top deck surface of cylinder block.

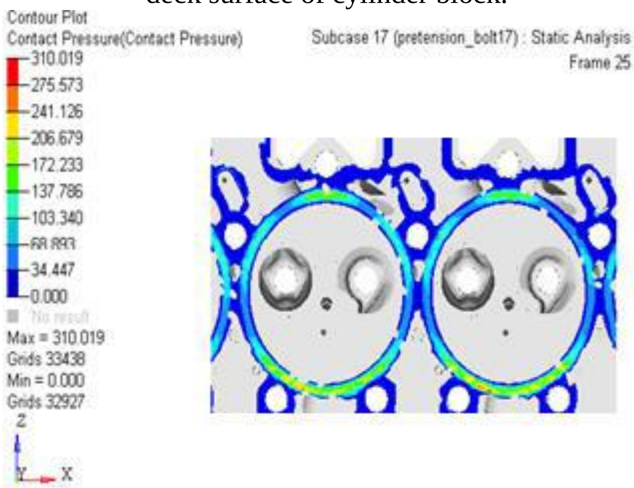


Figure 6. Contact pressure distribution (in Mpa) on bottom deck surface of cylinder head.

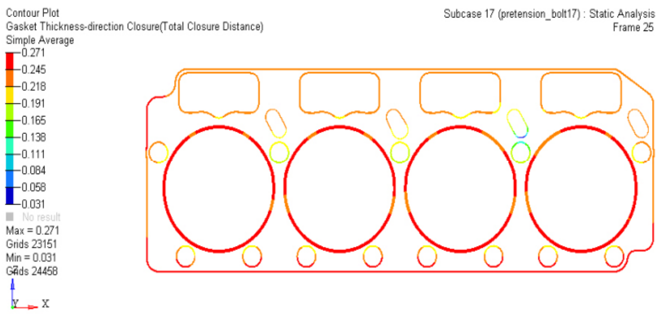


Figure 7. Gasket Thickness direction closure (in mm) for the various bead areas of the gasket.

Gasket Thickness direction closure follows same trend as the gasket thickness direction pressure values. Refer Fig. 7. The maximum closure value is found to be 0.271 mm, which occurs at the combustion bead area and peripheral bead area of the gasket at the intake side of the cylinder head. Asymmetric geometry and placement of cylinder head bolts around the cylinder bore apertures.

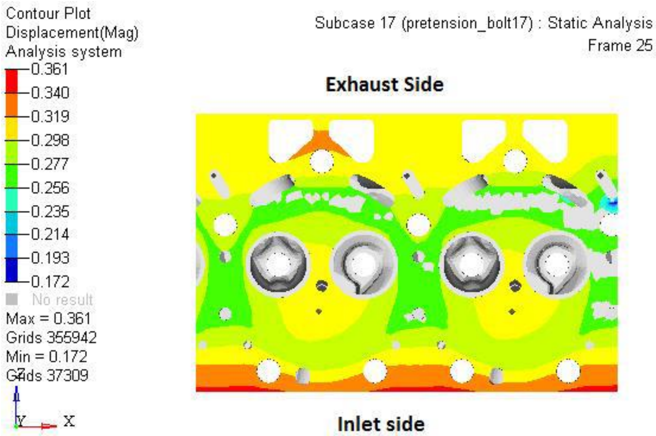


Figure 8. Head Deck Surface deformation

Deformation more at the intake side due to placement of 2 cylinder head bolts at intake side and the closer spacing between them. Cylinder head is bending towards the intake side as compared to the exhaust side. Refer Fig. No 8.

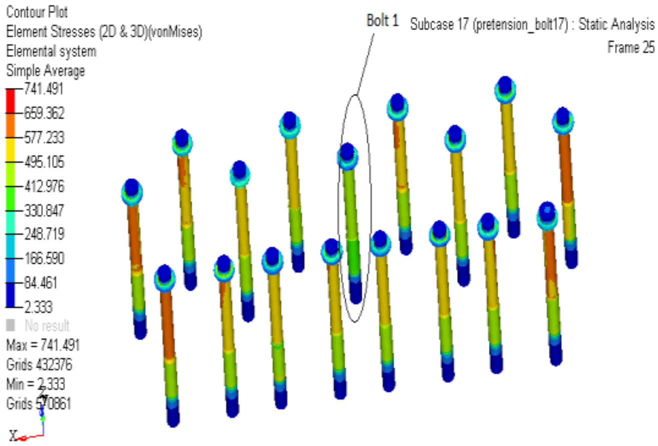


Figure 9. Bolt 3D stresses (von-mises) in Mpa

Fig. 9 gives stress distribution in bolts after pre-tensioning of all 17 bolts. The stress in the shank section of all bolts is approximately between 500-577 MPa. The maximum stress in each bolt of about 600-700 MPa, is found to occur at the section of the shank just below the bolt head bearing area (because of sudden change in cross section, stress is higher here). All the bolts show this trend except the bolt which is tightened first in the sequence, this bolt undergoes bolt preload loss as the stress in the shank is 412-495 MPa, at the end of the entire tightening sequence.

VII. FUJI FILM ANALYIS

Experimental validation of FEA results is carried out with the help of Fuji Film. The Fuji-film is a special type of pressure sensitive film, which develops a red color marking on the Fuji paper after application of pressure.The Fuji paper results for the contact interface at the head deck face supplied by the gasket manufacturer are shown in Fig. 11 having a maximum pressure value of 137.8 MPa and a minimum value of 115.1 MPa, at the combustion bead area of the gasket. As the combustion bead area is the most critical part for providing sealing to the combustion gases, this particular area will be studied more closely and the results of FEA will be validated for this region. Fuji-pressure film used for this study is a High Pressure type with the contact pressure measuring range of 50-130 MPa. Hence the impression obtained is only for this bead region, as the pressure developed at other regions is less than the lower limit of this film. The contact pressure on the gasket from FEA results will be queried for nodes corresponding to the positions at which the discrete pressure values have been specified by the manufacturer. These nodal locations corresponding to the discrete positions given in the Fuji-film results are shown in Fig. 10, along with the contour plot for the contact pressure distribution. The maximum pressure values of the FEA Analysis were approximately 300 MPa at the combustion bead area. Both analyses indicate the body of the gasket generally does not contain any areas of pressure. Fig. 12 shows the comparison of the pressure values obtained from FEA with a high pressure Fuji Film.

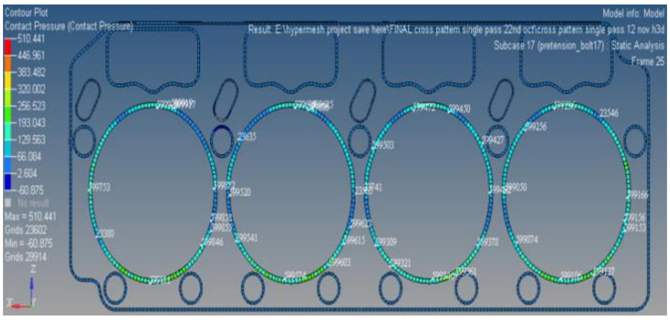


Figure 10. FEA results of Contact Pressure distribution on gasket elements

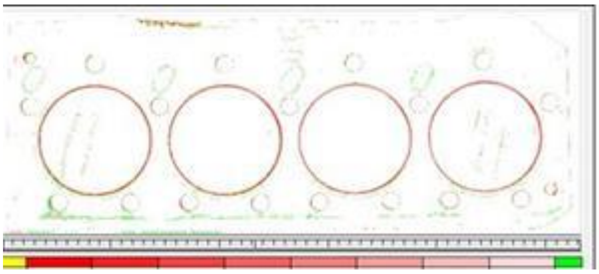
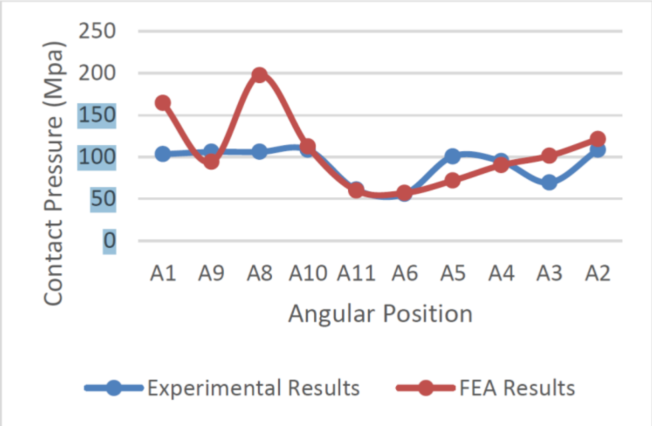
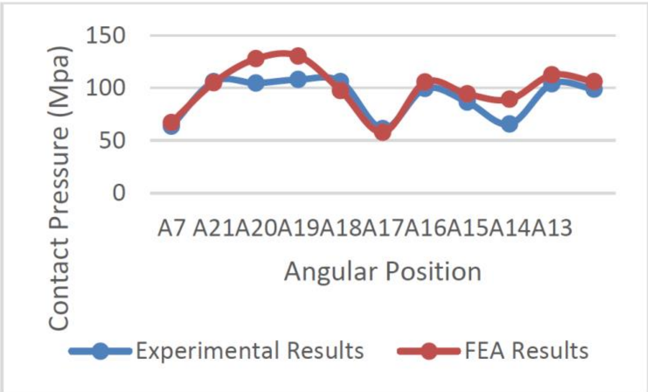


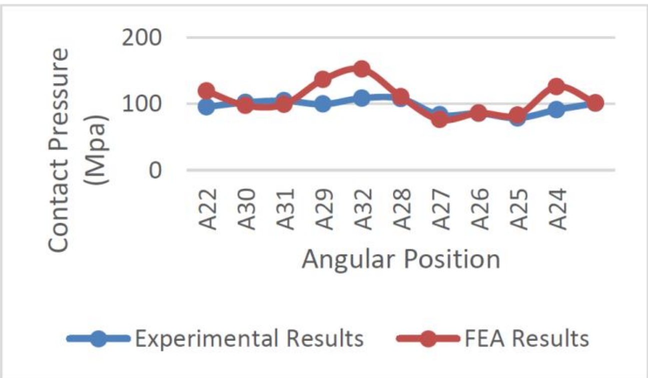
Figure 11. 11 High Pressure Fuji-film results on the gasket



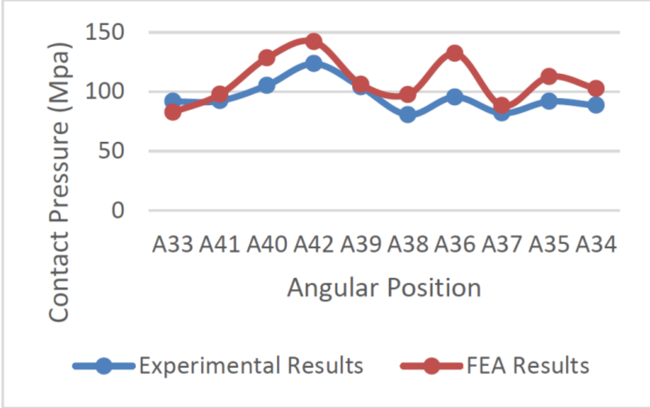
Ccomparison of contact pressure at cylinder 1 combustion bead



Comparison of contact pressure at cylinder 2 combustion bead



Comparison of contact pressure at cylinder 3 combustion bead



Comparison of contact pressure at cylinder 4 combustion bead

Figure 12. Comparison of FEA results with Fuji-film test at Combustion Bead.

In Fig. 12 at the position A8 of cylinder 1 and A32 of cylinder 3, the error percentage is more than 15%, this is mainly caused due to local distortion of the mesh adjacent to this position, where a node of one of the tetra elements has left the plane of the block deck face, which has happened while correcting the mesh quality of these elements. This is causing the mesh to pierce into the head deck surface and thus giving erroneous results. Cylinder 2 combustion bead are well correlated with the experimental results within the error tolerance limit of 15%. Only for the position A19 and A20 the error is about 19%, which is caused due to mismatching precision of FEA & Fuji-film results, wherein we are not able to get results at exactly the same position as that of Fuji-film results, due to lack of calculation interpolation points. We obtained correction of FEA and Fuji-film results more than 85 % at combustion bead and percentage variation pressure is less than 15 %, considering full assembly of engine crankcase, head, gasket it is considered as very good correlation.

8. Bore Deformation

With the application of the preload on cylinder head bolt in cross pattern bore get deformed. Minimum bore deformation is main requirement in case of the engine design for getting less lubricating oil consumption and engine blow by. Radial plots obtained from cross pattern are plotted for liner height 10 mm to 80 mm as shown in fig .13. Deformation we are getting with the cross pattern is within acceptable range. For cross pattern variation of bore deformation is generally on higher side as compare to the spiral pattern.

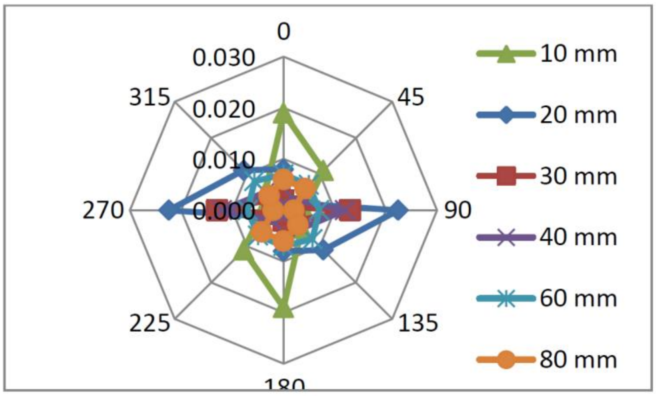


Figure 13. Bore Deformation because of cross pattern at different liner height

VIII. CONCLUSIONS

- Fujifilm impression is obtained for the gasket and FEA pressure distribution results for cross pattern are matched well within 15 %.
- FEA Analysis parameters like number of iterations for each subcase, load step increment are optimized in such a way that solution is converged and matched with experimental results.
- Gasket pressure distribution study has been carried out at design stage of engine design. Methodology has been developed for gasket pressure distribution analysis can be implemented at design stage for new engine design and development programs. This will save lot of time and cost.
- Bore deformation obtained because of preload is within acceptable range.

IX. ACKNOWLEDGEMENTS

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