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Effect of Pretensioning on Cylinder Head Bolt and Cylinder Head Gasket

[#1]Samir J. Khureshi, [#2]Abhijeet V. Marathe, [#3]D.H. Burande

[1]khureshisamir@gmail.com
[2] avmaratahe.edl@araiindia.com
[3] dhburande.scoe@sinhgad.edu

[#1] Mechanical Engineering, Sinhgad College of Engineering, Vadgaon (BK) Pune, India
[#2]Automotive research Association of India, Pune, India

ABSTRACT

The main function of cylinder head Gasketed joint is to avoid the leakage of combustion gases, to seal the oil and coolant and avoid to enter the outside air into combustion chamber. For sealing purpose we have to apply the preload on the cylinder head gasket.For application of the preload there are two problems like less pretension load on the bolt cause leakages problem and due to excessive preload problem of more bore deformation. Paper showed proper method for required pretension so that there is neither leakages problem nor bore deformation problem. This is necessary for high performance engines which require perfect engine head sealing as their engine heads are subject to higher cyclic loads. Finite element analysis is done on 3 cylinder diesel engine using Optistruct solver in Hypermesh software. FEA shows the results for elongation of bolt, stresses occur inside cylinder head – crankcase assembly and gasket sealing status. Optistruct solver in Hypermesh software had used for finding elongation and stress inside the bolt in given 3 cylinder engine using M12 bolt. In this FEA method analysis is done for three bolt for M12 and M10 bolt. For M12 bolt torque range is inside yield point and for M10 bolt torque range is above yield point here compare and analyze the elongation of bolt and stresses inside the bolt by using FEA method. Effect of PCD of bolted joint by applying uniform pressure distribution in this section analyze the effect of elongation in cylinder head bolt and maximum element stresses inside cylinder head bolt, cylinder head and engine block. Here four iteration had taken first decrease PCD by 8mm, second decrease PCD by 16mm, third increase PCD by 6mm and fourth iteration increase PCD by 12mm, here diameter of cylinder head bolt is constant for all iteration i.e. 12 mm.

***Keywords*— 3 cylinder diesel engine, Pretension, Elongation of bolt, Element stresses, Hypermesh software, Optistruct solver.**

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I INTRODUCTION

A. Background

There are various method of applying torque for cylinder head bolt like torque given in elastic limit and torque given beyond the elastic limit . If torque is in elastic limit then reusability of cylinder head bolt is more. The methods of tightening a screw into its plastic region require expensive equipment, which can often only be acquired by large industries or profitable small workshops, there is a risk of over-tightening the bolt or of failing to reach the yield point. In addition, the

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reusability plastic region tightening for fasteners is very poor[1].

In general, if the preload is high, cause blow by gases move inside the crankcase. Blow-by gases come mainly from gap between cylinder liner and piston rings. These gases, once in the engine crankcase, are loaded in an oil aerosol. Blow-by gases cause problems with possible risks to the engine. They involve, in particular, the pressurization of the crankcase and can contribute to oil consumption. If preload is less, there will be loosening of bolt and leakages of lubrication oil, coolant, combustion gases take place.

This Paper shows proper method for required torque so that there is neither leakages problem nor bore deformation problem. This is necessary for high performance engines which require perfect engine head sealing as their engine heads are subject to higher cyclic loads.Finite element analysis is done on 3 cylinder diesel engine using Optistruct solver in Hypermesh software. FEA shows the results for elongation of bolt, stresses occur inside cylinder head – crankcase assembly and gasket sealing status.

B. Paper Objective

- Estimation of prerequisite pretension for cylinder head bolt.
- Compare result for elongation of bolt by using analytical method, FEA method using Hypermesh software .
- Compare result of stresses for cylinder head bolt using analytical method and FEA method using Hypermesh software.

C. Paper scope

- Scope of this paper is design and analysis cylinder headed gasket assembly. To find out locations of leakage gap between cylinder head and crankcase due high pressure gases, lubrication oil, coolant etc. Design of Proper pretension estimation methodology and effect of PCD of bolted joint. Validation of FEA results is done by analytically.

D. Methodology

- Structural modeling of cylinder head, cylinder head gasket and crankcase developed by using CAD software PRO- E.
- The structural modeling then imported into the computer – aided engineering and meshing of cylinder head, cylinder head gasket and crankcase is done in Hypermesh.
- The boundary condition and loading are selected and placed at the cylinder head, cylinder head gasket and crankcase.

The finite element analysis (FEA) will be carried out at the cylinder head, cylinder head gasket and crankcase and producing results of stresses and deformation where it can be used to analyze the effect of pitch circle diameter of cylinder head bolt.

I. Different method of torquing

One of the major problems with the use of bolted joints is the precision, with regard to achieving an accurate preload. Due to inaccurate method of tightening there is chance of bolted joint failure.

Here are six main methods used to control the preload of a threaded fastener.

A. Tension indicating methods.
B. Bolt stretch method.
C. Heat tightening.
D. Torque control tightening.
E. Yield controlled tightening
F. Angle Torque Method

A. Tension Indicating Methods

This category includes the use of special load indicating bolts. This method use load indicating washers .Use of methods which determine the length change of the fastener. There are a wide number of ways bolt tension can be indirectly measured.

B. Bolt stretch /Method

A problem relating to the tightening of large bolts is that very high tightening torques are required. Use hydraulic torque wrenches Hydraulic power is use for extension of bolt. The stretch created in the bolt during tightening allows the fastener to clamp. This tension creates the clamping force, which holds our joint together. In critical joints, it is vital to ensure that the proper amount of preload is applied to the fastener and that the preload remains over the service life of the joint.

C. Heat Tightening

Heat tightening utilizes the thermal expansion characteristics of the bolt. The bolt is heated and expands: the nut is indexed (using the angle of turn method) and the system allowed to cool. As the bolt attempts to contract it is constrained longitudinally by the clamped material and a preload results.

D. Torque control bolt tightening

Torque is applied below elastic limit. Torque variation up to +/- 30 % Generate excessive scatter in preload, which is detrimental in critical joints. Bolts can be reuse in case torque control tightening . The nominal torque necessary to tighten the bolt to a given preload can be determined from table. In torque control bolt tightening method 50% torque use to overcome bolt head friction, 40% torque use to overcome thread friction, Remaining up to 10% use for bolt elongation .

E. Yield controlled tightening

In this method torque is Appling inside the yield point. Elongation of bolt due to pretension is in elastic limit, plastic elongation of bolt is zero. In this method bolt can reuse.

F. Torque angle method (TTY) :

Torque is applied above yield limit. Torque variation upto +/- 10%. Torque is applied snug torque and angle of rotation. Generate less scatter, hence joint preload is precise. Bolts are not reusable. The torque required to reach the yield point of a screw is very uncertain due to the usually large variations in screw strength and friction. However, applying a specific angle after an initial torque leads to more consistent preload levels. Indirectly, the angle tightening is a form of length measurement. The elongation of the screw and the compression of the parts are thus measured concurrently. The torque and angle method gives a more uniform preload than the torque-control method, as long as the fastener is brought into its plastic region.

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II. Tightening Torque Calculation for Cylinder Head Bolts

The analytical calculations are performed to find out the cylinder head bolt Pretension by referring various books and papers.

TABLE I

Cylinder head Bolt Analysis			
Engine details			
Content	**Symbol**	**Values**	**Unit**
Bore diameter	D	80	mm
Cylinder peak pressure	Ppeak	22	N/mm²
Height of cylinder head	Hcyl	90	mm
Cylinder head gasket thickness	Tg	1.2	mm
Gap bet cylinder head and gasket before assembly	dgap	0.2	mm
Stud details			
M series diameter	D	12	mm
Pitch of bolt	P	1.5	mm
Number of studs around the cylinder	N	6	
Young modulus of elasticity of studs	Eb	206000	N/mm²

SPECIFICATION OF ENGINE AND STUD

A critical component of designing bolted joints is not only determining the number of bolts, the size of them, and the placement of them but also determining the appropriate preload for the bolt and the pretension that must be applied to achieve the desired preload. There is no one right choice for the preload or torque. Many factors need to be considered when making this determination. A basic guideline given in the Machinery's Handbook is to use 75% of the proof strength (or 75% of 85% of the material yield strength if the proof strength is not known) for removable fasteners and 90% of the proof strength for permanent fasteners. Things to consider include the tension in the bolt and therefore the clamping force, fatigue concerns (higher preload is generally preferable), how much pretension can easily be applied without risking damaging another part if the tool slips while applying the load, etc. The Machinery's Handbook and the give estimates for the accuracy of bolt preload based on application method. These uncertainties should be used for all small fasteners (defined as those less than ¾"). A general relationship between applied pretension, T, and the preload in the bolt, F, can be written in terms of the bolt diameter, D, and the "Nut Factor", K, as

$$T = \frac{KDF}{1000}$$

Where ,
T = torque applied in N-m,
D = Nominal bolt diameter in mm,
F = clamp force (preload) N
K = Nut factor

Only use nut factors when approximate preload is sufficient for the design. For cases where strain gages cannot be used, bolt extension cannot be measured, load sensing washers cannot be used, etc., there is no choice but use a nut factor. In these cases, any analysis should be done using a range of nut factors to bind the results. A low nut factor gives a higher preload and clamping force but puts the bolt closer to yield while a high nut factor gives a lower preload and clamping force but the capacity of the joint to resist external tensile loads has been reduced. Additional information on nut factors can be found in Bickford and the Machinery's Handbook.

A. Original formula

The following equation addresses friction as well as friction in bearing area of bolt and or nut. The pretension can be calculated very accurately using following formula.

$$T = F * \left(\left(\frac{d_2}{2}\right)\tan(\varphi + p') + \left(\frac{D_{km}}{2}\right)\tan(p)\right)$$

$$D_{km} = \frac{d_w + d_h}{4}$$

Nut factor K can be calculated using,

$$K = [0.16p + 0.58 d_2 \mu_k] / d$$

Effective area (A_{eq}) : Here Effective area is area under grip of bolt.

Stiffness of cylinder head (K_h):

$$K_h = \frac{E_h \cdot A_{eq}}{L}$$

To calculate stiffness of cylinder head bolt (K_b):

TABLE III
SPECIFICATION OF CYLINDER HEAD BOLT

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Sr. No		Length (L)	Diameter (D)	Area (A)
1	Head side	5	12	113.11
2	Shank	85	11	95.04
3	Exposed thread	17	10.38	84.57
4	Engaged thread	5	10.38	84.57

$$\frac{1}{K_b} = \frac{1}{K_1} + \frac{1}{K_2} + \frac{1}{K_3} + \frac{1}{K_4}$$

$$\frac{1}{K_b} = \frac{1}{E_b}\left(\frac{L1}{A1} + \frac{L2}{A2} + \frac{L3}{A3} + \frac{L4}{A4}\right)$$

Max combustion force (F_c)

$$F_c = \frac{\pi}{4} * D^2 * P_{peak}$$

III. Analysis For Elongation And Stresses Inside Cylinder Head Bolt And Cylinder Head Gasket

HyperMesh is a high-performance finite element pre-and post-processor for major finite element solvers, which allows engineers to analyze design conditions in a highly interactive and visual environment. HyperMesh's user-interface is easy to learn and supports the direct use of CAD geometry and existing finite element models, providing robust interoperability and efficiency.Pretension Bolt Analysis of an IC Engine Cylinder Head, Gasket and Engine Block Connected Using Head Bolts

This Paper consist of the procedure to perform 3D pretension bolt analysis on a section of an IC Engine. The pretension analysis is conducted to measure the response of a system consisting of the cylinder head, gasket and engine block connected by cylinder head bolts subjected to a pretension force of 80000 N each.

This analysis apply 3D bolt pretension to the cylinder head bolts and then apply a pressure to the constrained system. The applied pressure load models the pressure on the inside walls of an engine due to combustion. Pressure within the engine compartment varies with time however, we capture the response of the system at a specific instant frozen in time.

Step 1: ApplyingProperties, materials, and gasket normal direction: Optistruct solver had used for analysis In Hypermesh software. Modulus of elasticity for cylinder head, head bolt and for block is given Young's Modulus of Elasticity = 2.1 x 10^5 N/m^2 and Poisson's Ratio = 0.3. Young's Modulus of Elasticity = 2.2 x 104 N/mm2 had used for cylinder head gasket. Gasket behavior is nonlinear and it may undergo cycles of loading and unloading which lead to changes in its properties at each step. Here focuses on 3D pretensioning, the loading and unloading paths for the gasket material are pre-populated in the MGASK data entry via the TABLES1 entries referenced by corresponding load collectors. As we are using a quasi-static analysis, the initial applied pressure load is compared with corresponding values within the loading/unloading path tables and the initial material properties of the gasket is determined.

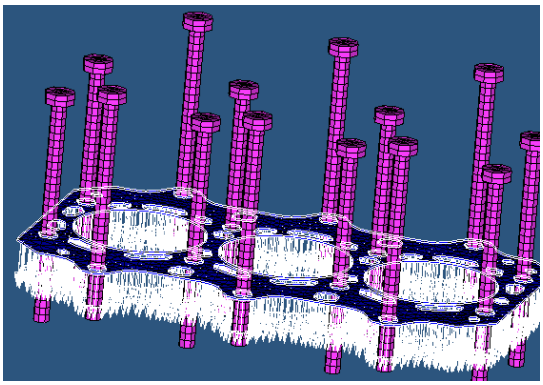

Fig. 1Displaying the gasket normals (negative Y direction).

Step 2: Generating contact surfaces using the Auto Contact Manager, Seventy-four (74) contact surfaces and 30 contact interfaces are generated by HyperMesh within the specified proximity distance.

All elements or mesh lines within the model will turn transparent and the surfaces in contact are shaded in red and blue depending on their master-slave relationship.

Auto Contact browser to allow HyperMesh to create contact elements at each of these 30 interfaces. These contact elements are created as 30 separate groups and can be reviewed under Groups in the Model browser.

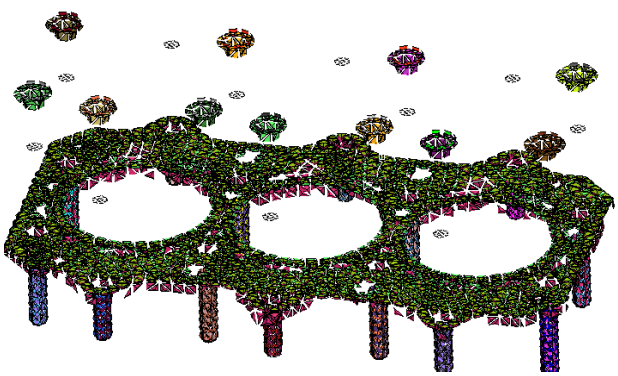

Fig. 2 Contact element generation at the interfaces.

Step 3: 3D bolt pretensioning :

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Bolt pretensioning analysis is used to determine the elongation inside cylinder head bolt, elemental stresses and gasket sealing status of a system which contains bolts holding two or more components together as a result of pretensioning.

A surface PT_Surf has been predefined to demonstrate 3D pretensioning on existing surfaces. To additionally demonstrate 3D pretensioning by creating a new surface, the bolt is left unchanged.

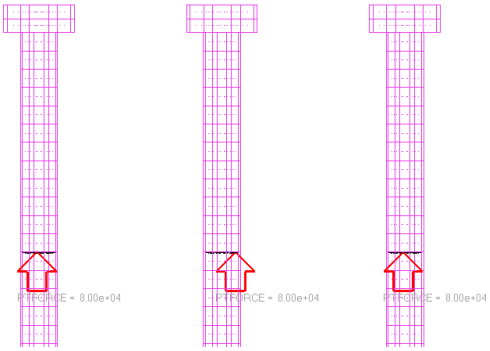

Fig. 3 Reviewing the three pretensioned bolts.

For bolt pretensioning first one surface is creating for specifying magnitude as well as direction for the force. For M12 bolt pretension is apply up to 80000N which calculate by analytical method.

Step 4: Creating a pretension loadstep and a subsequent analysis loadstep, OptiStruct nonlinear quasi-static analysis loadsteps will be created for both pretensioning and the subsequent analysis. The analysis is nonlinear due to the presence of contact elements and the gasket loading/unloading paths. The CNTNLSUB bulk data entry is used to define new subcase for pressure. Finally Launch OptiStruct for analysis.

IV. Result by FEA Method

A. Elongation in cylinder head bolt

Optistruct solver in Hypermesh software had used for finding elongation and stress inside the bolt. Given 3 cylinder engine using M12 bolt. Torque range is using for this bolt is 140Nm which is inside yield point. In this FEA method analysis is done for three bolt for M12 and M10 bolt. For M12 bolt torque range is inside yield point and for M10 bolt torque range is above yield point here compare and analyze the elongation of bolt and stresses inside the bolt by using FEA method. Hyperview software used for results.

For iteration one M12 bolt had used and pretension or final force is applied up to 80000N maximum elongation of bolt is 0.3717 mm .

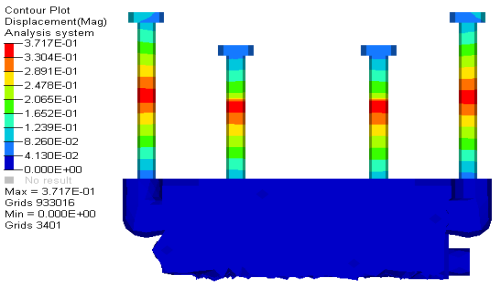

Fig. 4 Maximum Elongation in M12 cylinder head bolt

As shown in Fig. 4maximum elongation of cylinder head bolt is occur at that surface where the pretension force is acting where in hypermesh software create a surface inside bolt. After pretension surface elongation inside cylinder head bolt is decreasing not only on cylinder head bolt head side also towards the engine block side.

For iteration third M10 bolt had used and pretension or final force is applied up to 77000N maximum elongation of bolt is 0.4535 mm.

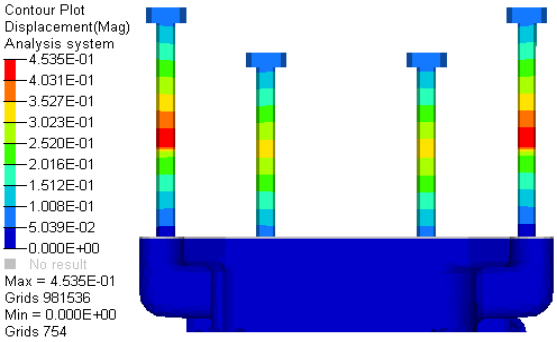

Fig. 5 Maximum Elongation in M10 cylinder head bolt

B. Comparison for M12 and M10 bolt for elongation of cylinder head bolt

Comparison of M12 and M10 cylinder head bolt for elongation of cylinder head bolt here table 1 consist of Elongation of bolt by Analytical method (mm) and Elongation of bolt by Analysis in Hypermesh (mm) values for M12 and M10 cylinder head bolt.

TABLE IIIII
COMPARISON FOR M12 AND M10 BOLT FOR ELONGATION OF BOLT

Sr. No.	Bolt size	Elongation of bolt by Analytical method (mm)	Elongation of bolt by Analysis in Hypermesh (mm)	% Error
1	M12	0.387	0.378	1.02
3	M10	0.448	0.4535	0.81

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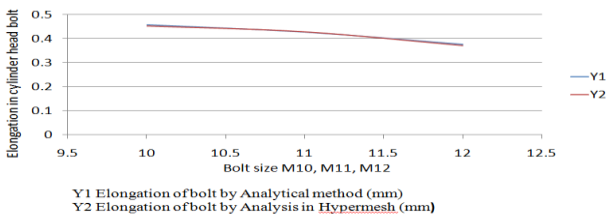

Fig. 6 Graph for Comparison in elongation for M12, and 10 bolt

From Fig. 6 we conclude that of Elongation of bolt by Analytical method (mm) and Elongation of bolt by Analysis in Hypermesh (mm) values for M12 and M10 cylinder head bolts are matching to each other. In this comparison percentage of errors are very less.

C. Element stresses inside cylinder head bolt

For iteration one M12 bolt had used and pretension or final force is applied up to 80000N maximum Element stresses inside cylinder head M12 bolt is 950N/mm².

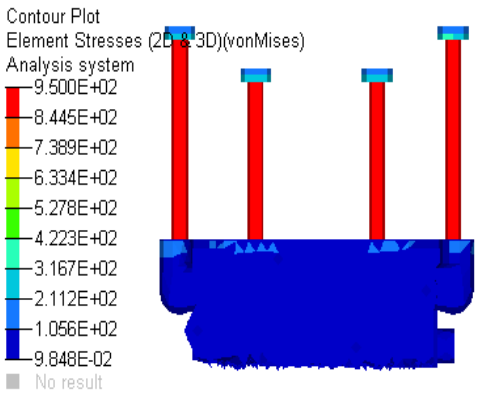

Fig. 7 MaximumElement stresses inside cylinder head M12 bolt

As shown in Fig. 7 maximum Element stresses inside cylinder head M12 bolt is 950N/mm² occurs at whole body of cylinder head M12 bolt. Maximum Element stresses inside cylinder head is 316.7 N/mm² and maximum Element stresses inside engine block is 211.2 N/mm².As shown in Fig. 8 maximum Element stresses inside cylinder head Gasket for M12 bolt is 120 N/mm². Maximum Element stresses near to hole of cylinder head bolt is 90 N/mm² to 120 N/mm². Maximum Element stresses near to oil hole is 50 N/mm² to 60 N/mm². Maximum Element stresses near to cooling hole is 40.04 N/mm² to 53.37 N/mm². Element stresses top portion to circumference of cylinder liner on the gasket is 53.37 N/mm² to 66.7 N/mm². Element stresses inside cylinder head Gasket for M12 bolt is compressible because gasket is sandwiched between cylinder head and engine block.

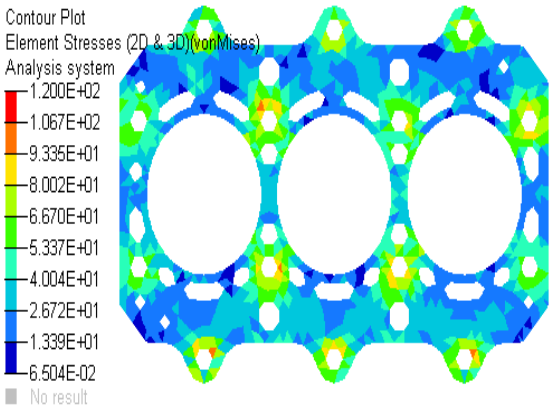

Fig. 8 MaximumElement stresses inside cylinder head Gasket for M12 bolt

For iteration three M10 bolt had used and pretension or final force is applied up to 77000N maximum Element stresses inside cylinder head M10 bolt is 962.4 N/mm².
As shown in Fig. 9 maximum Element stresses inside cylinder head M10 bolt is 962.4 N/mm² occurs at whole body of cylinder head M10 bolt. Maximum Element stresses inside cylinder head is 320.9 N/mm² and maximum Element stresses inside engine block is 214.0 N/mm². As shown in Fig. 10 maximum Element stresses inside cylinder head Gasket for M10 bolt is 120 N/mm². Maximum Element stresses near to hole of cylinder head bolt is 93.37 N/mm² to 120 N/mm². Maximum Element stresses near to oil hole is 53.42 N/mm² to 66.74 N/mm². Maximum Element stresses near to cooling hole is 40.11 N/mm² to 53.42 N/mm². Element stresses top portion to circumference of cylinder liner on the gasket is 53.42 N/mm² to 66.74 N/mm². Element stresses inside cylinder head Gasket for M10 bolt is compressible because gasket is sandwiched between cylinder head and engine block.

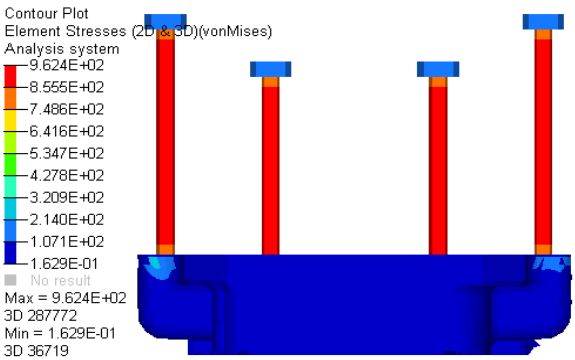

Fig. 9 Maximum Element stresses inside cylinder head M10 bolt

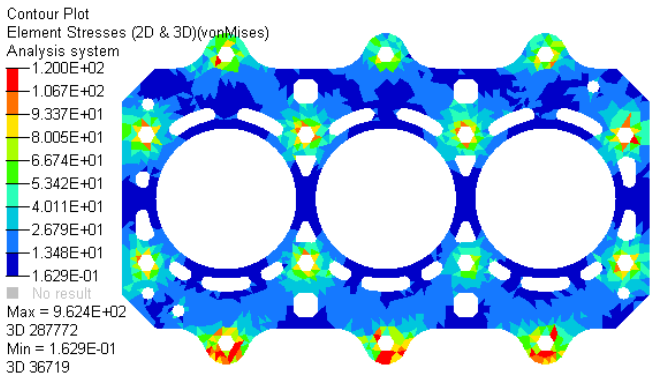

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Fig. 10 Maximum Element stresses inside cylinder head Gasket for M10 bolt

D. *Comparison for M12 and M10 bolt for Maximum Element Stresses*

Comparison of M12 and M10 cylinder head bolt for Maximum element stresses of cylinder head bolt here table 2 consist of Maximum elemental stresses by Analytical method (mm) and Maximum elemental stresses by Analysis in Hypermesh (mm)values for M12 and M10 cylinder head bolt.

TABLE IVV
COMPARISON FOR M12 AND M10 BOLT FOR MAXIMUM ELEMENTAL STRESSES

Sr. No.	Bolt size	Maximum elemental stresses by Analytical method (mm)	Maximum elemental stresses by Analysis in Hypermesh (mm)	% Error
1	M12	942.59	950	0.786
3	M10	962.41	963.23	0.086

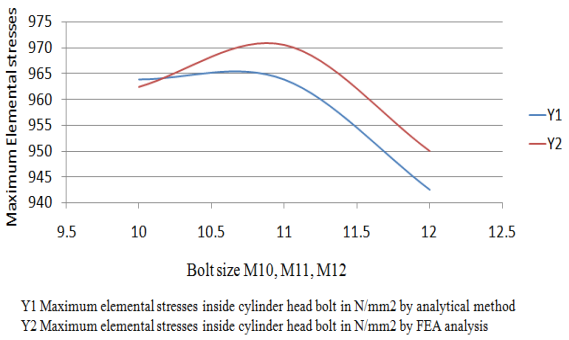

Fig. 11 Graph forComparison in elongation for M12 and M10 bolt

From Fig. 11 we conclude that of Maximum elemental stresses by Analytical method (mm) and Maximum elemental stresses by Analysis in Hypermesh (mm)values for M12 and M10 cylinder head bolts are matching to each other. In this comparison percentage of errors are very less.

E. *Gasket sealing status*

Gasket sealing status or sealing status index shows is proper sealing between cylinder head and engine block. As shown in Fig. 12 For iteration one M12 bolt had used and pretension or final force is applied up to 80000N here red color indicates there is proper sealing between cylinder head and engine block and blue color indicates there is some leakages or there is some gap between cylinder head and engine block. Here perfect sealing shows sealing status index 1 and if there is no proper sealing then it shows sealing status index 0.

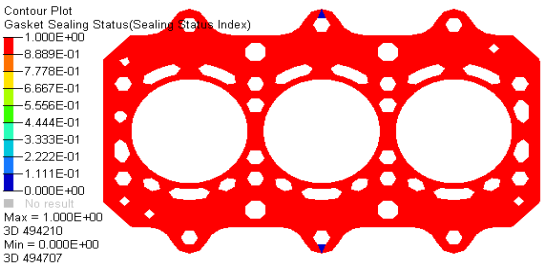

Fig. 12 Gasket sealing status for M12 bolt

As shown in Fig. 13 For iteration one M10 bolt had used and pretension or final force is applied up to 77000N here red color indicates there is proper sealing between cylinder head and engine block and blue color indicates there is some leakages or there is some gap between cylinder head and engine block. Here perfect sealing shows sealing status index 1 and if there is no proper sealing then it shows sealing status index 0.

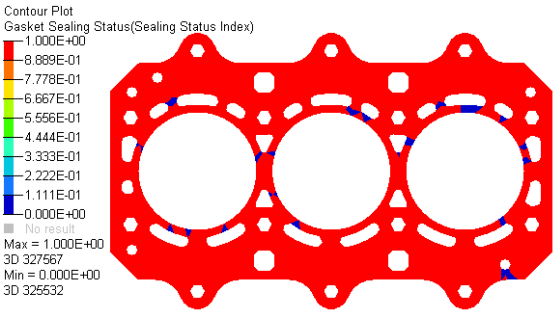

Fig. 13 Gasket sealing status for M10 bolt

A. *Effect of PCD of bolted joint by applying uniform pressure distribution :*

Effect of PCD of bolted joint by applying uniform pressure distribution in this section analyze the effect of elongation in cylinder head bolt and maximum element stresses inside cylinder head bolt , cylinder head and engine block. Here four iteration had taken first decrease PCD by 8mm, second decrease PCD by 16mm, third increase PCD by 6mm and fourth iteration increase PCD by 12mm i.e. 101.35mm, 109.35mm, 123.35mm and 129.35mm. Here diameter of cylinder head bolt is constant for all iteration i.e. 12 mm.For iteration one decreasing PCD of bolt by 8mm for M12 bolt had used and pretension or final force is applied up to 80000N maximum elongation of bolt is 0.3691 and maximum Element stresses inside cylinder head M12 bolt is 930.2N/mm².

As shown in Fig. 14 maximum elongation of cylinder head bolt is occur at that surface where the pretension force is acting where in hypermesh software create a surface inside bolt. After pretension surface elongation inside cylinder head bolt is decreasing not only on cylinder head

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bolt head side also towards the engine block side. Maximum Element stresses inside cylinder head M12 bolt by decreasing PCD of bolt by 8mm is 930.2N/mm² occurs at whole body of cylinder head M12 bolt. Maximum Element stresses inside cylinder head is 310.1 N/mm² and maximum Element stresses inside engine block is 206.8 N/mm².

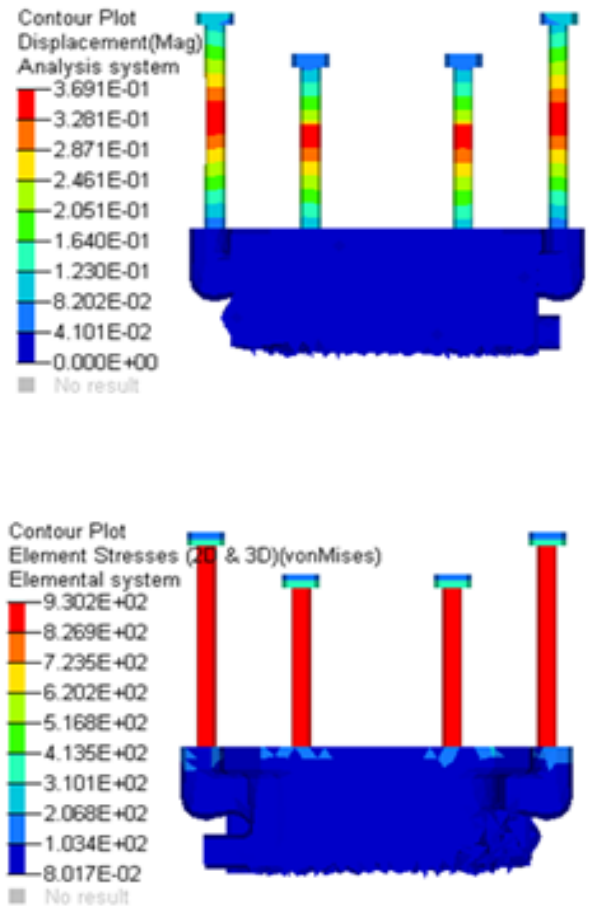


Fig. 14 Elongation of bolt and Maximum element stresses by decreasing PCD by 8mm

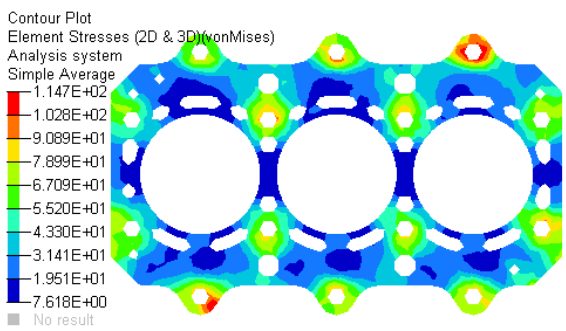


Fig. 15 Maximum Element stresses inside cylinder head Gasket for decreasing PCD by 8mm

As shown in Fig. 15 maximum Element stresses inside cylinder head Gasket for M12 bolt by decreasing PCD of bolt by 8mm is 114.7 N/mm². Maximum Element stresses near to hole of cylinder head bolt is 90.89 N/mm² to 114.7 N/mm². Maximum Element stresses near to oil hole is 43.3 N/mm² to 55.2 N/mm². Maximum Element stresses near to cooling hole is 31.41 N/mm² to 43.3 N/mm². Element stresses top portion to circumference of cylinder liner on the gasket is 55.2 N/mm² to 67.09 N/mm². Element stresses inside cylinder head Gasket is compressible because gasket is sandwiched between cylinder head and engine block.

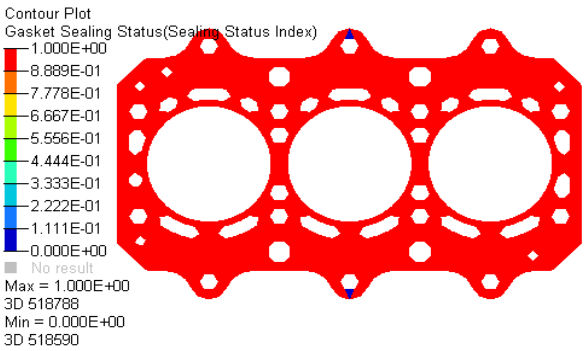
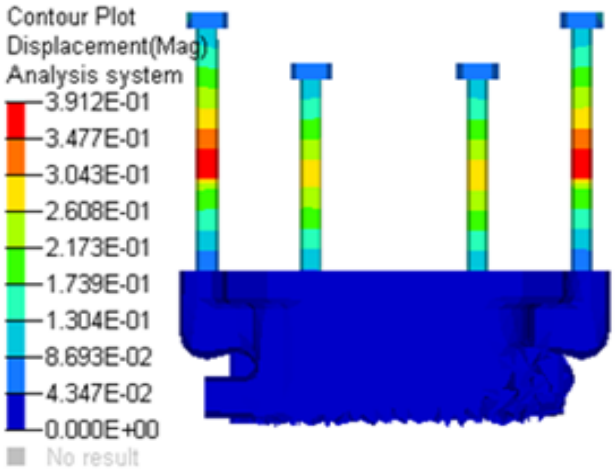


Fig. 16 Gasket sealing status for decreasing PCD by 8mm

Gasket sealing status or sealing status index shows is proper sealing between cylinder head and engine block. As shown in Fig. 16 For iteration one M12 bolt had used for decreasing PCD by 8mm and pretension or final force is applied up to 80000N here red color indicates there is proper sealing between cylinder head and engine block and blue color indicates there is some leakages or there is some gap between cylinder head and engine block. Here perfect sealing shows sealing status index 1 and if there is no proper sealing then it shows sealing status index 0. For next iteration Increasing PCD of bolt by 6mm for M12 bolt had used and pretension or final force is applied up to 80000N maximum elongation of bolt is 0.3912 and maximum Element stresses inside cylinder head M12 bolt is 965.2 N/mm².



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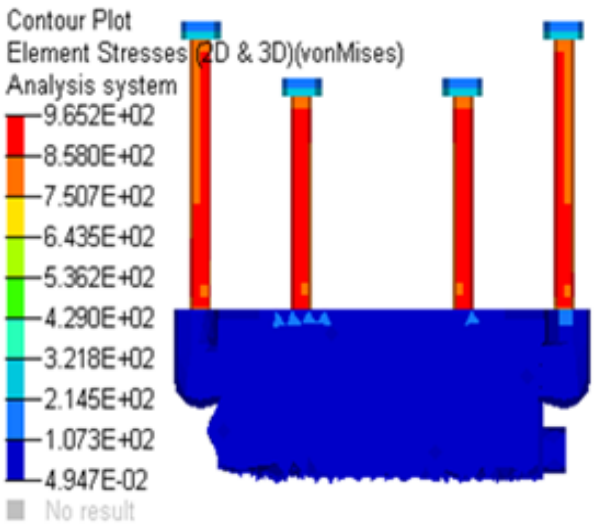

Fig. 17 Elongation of bolt and Maximum element stresses by increasing PCD by 6mm

As shown in Fig. 17 maximum elongation of cylinder head bolt is occur at that surface where the pretension force is acting where in hypermesh software create a surface inside bolt. After pretension surface elongation inside cylinder head bolt is decreasing not only on cylinder head bolt head side also towards the engine block side. Maximum Element stresses inside cylinder head M12 bolt by increasing PCD of bolt by 6mm is 965.2 N/mm² occurs at whole body of cylinder head M12 bolt. Maximum Element stresses inside cylinder head is 429.0 N/mm² and maximum Element stresses inside engine block is 321.8 N/mm².

As shown in Fig. 18 maximum Element stresses inside cylinder head Gasket for M12 bolt by increasing PCD of bolt by 6mm is 82.97 N/mm². Maximum Element stresses near to hole of cylinder head bolt is 74.4 N/mm² to 82.97 N/mm². Maximum Element stresses near to oil hole is 40.30 N/mm² to 48.84 N/mm². Maximum Element stresses near to cooling hole is 31.77 N/mm² to 40.30 N/mm². Element stresses top portion to circumference of cylinder liner on the gasket is 40.30 N/mm² to 48.84 N/mm². Element stresses inside cylinder head Gasket is compressible because gasket is sandwiched between cylinder head and engine block.

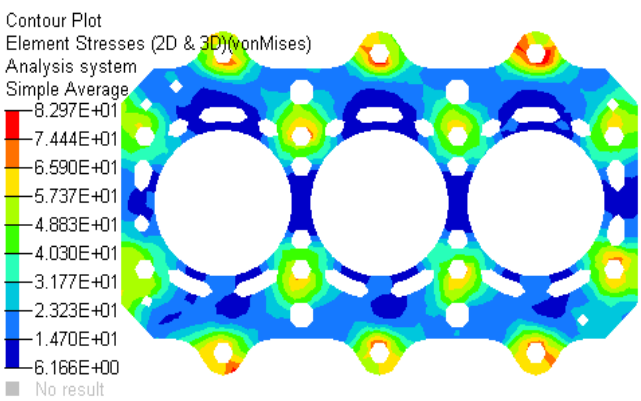

Fig. 18 Maximum Element stresses inside cylinder head Gasket for increasing PCD by 6mm

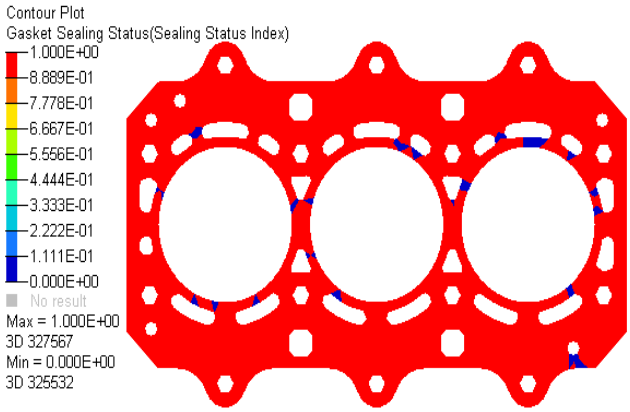

Fig. 19 Gasket sealing status for increasing PCD by 6mm

As shown in Fig. 19 For iteration one M12 bolt had used for increasing PCD by 6mm and pretension or final force is applied up to 80000N here red color indicates there is proper sealing between cylinder head and engine block and blue color indicates there is some leakages or there is some gap between cylinder head and engine block. Here perfect sealing shows sealing status index 1 and if there is no proper sealing then it shows sealing status index 0.

F. Comparison for effect of PCD of bolted joint for elongation of bolt

Comparison for effect of PCD of bolted joint for elongation of cylinder head bolt here table 4 consist of Elongation of bolt by Analytical method (mm) and Elongation of bolt by Analysis in Hypermesh (mm) values. Here four iteration had taken first decrease PCD by 8mm, second decrease PCD by 16mm, third increase PCD by 6mm and fourth iteration increase PCD by 12mm i.e. 101.35mm, 109.35mm, 123.35mm and 129.35mm. Here diameter of cylinder head bolt is constant for all iteration i.e. 12 mm.

From Fig. 20 we conclude that of Elongation of bolt by Analytical method (mm) and Elongation of bolt by Analysis in Hypermesh (mm) values for four iteration had taken first decrease PCD by 8mm, second decrease PCD by 16mm, third increase PCD by 6mm and fourth iteration increase PCD by 12mm are matching to each other. In this comparison percentage of errors are very less.

TABLE V
COMPARISON FOR EFFECT OF PCD OF BOLTED JOINT FOR ELONGATION OF BOLT

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Sr. No.	PCD of bolt	Maximum elemental stresses by Analytical method (mm)	Maximum elemental stresses by Analysis in Hypermesh (mm)	% Error
1	101.35	882.75	897.34	1.65
2	109.35	928.65	930.24	0.17
3	117.35	933.28	950	1.79
4	123.35	953.32	965.21	1.24
5	129.35	961.25	970.38	0.95

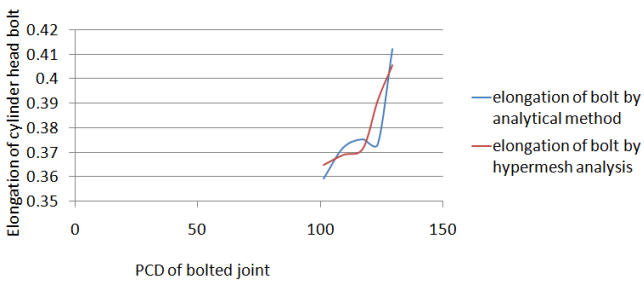

Fig. 20 Graph for Comparison for effect of PCD of bolted joint for elongation of bolt

A. *Comparison for effect of PCD of bolted joint for Maximum Element Stresses*

Comparison for effect of PCD of bolted joint for Maximum element stresses of cylinder head bolt here table 5 consist of Maximum elemental stresses by Analytical method (mm) and Maximum elemental stresses by Analysis in Hypermesh (mm)values. Here four iteration had taken first decrease PCD by 8mm, second decrease PCD by 16mm, third increase PCD by 6mm and fourth iteration increase PCD by 12mm. Here diameter of cylinder head bolt is constant for all iteration i.e. 12 mm.

Sr. No.	PCD of bolt	Elongation of bolt by Analytical method (mm)	Elongation of bolt by Analysis in Hypermesh (mm)	% Error
1	101.35	0.3594	0.365	1.53
2	109.35	0.3721	0.3691	-0.81
3	117.35	0.3755	0.3717	-1.02
4	123.35	0.3734	0.3912	4.55
5	129.35	0.4124	0.4056	-1.67

TABLE VI
COMPARISON FOR EFFECT OF PCD OF BOLTED JOINT FOR MAXIMUM ELEMENTAL STRESSES

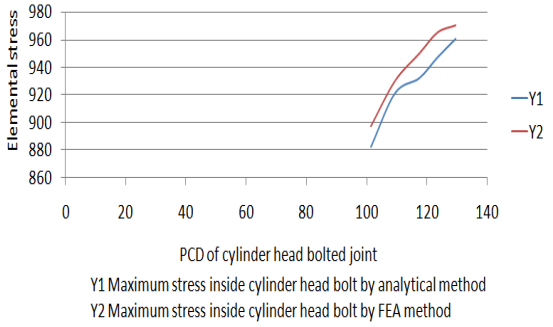

Fig. 21 Graph for Comparison for effect of PCD of bolted joint for Maximum elemental stresses

From Fig. 21 we conclude that Maximum elemental stresses by Analytical method (mm) and Maximum elemental stresses by Analysis in Hypermesh (mm) values. for four iteration had taken first decrease PCD by 8mm, second decrease PCD by 16mm, third increase PCD by 6mm and fourth iteration increase PCD by 12mm are matching to each other. In this comparison percentage of errors are very less.

V. CONCLUSIONS

Here paper shown proper range for pretension due to which there is neither leakages problem nor bore deformation problem. Torque range from analytical method apply for M12*1.5 and M10*1 Cylinder head bolt. Same torque for four iteration of M12*1.5 Cylinder head bolt while increase

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and decrease the PCD of bolted joint, from analysis by using Optistruct solver in Hypermesh software we conclude that,

- Elongation in cylinder head bolt by applying same torque 140Nm for M12*1.5 is 0.37 mm, for M10*1.5 is 0.45 mm which is less than the limiting value of 5% of length of unengaged thread that is 0.81mm, also stresses for M12*1.5 is 950 N/mm², for M10*1.5 is 965 N/mm² which is inside yield strength of bolt.
- A standard cylinder head bolt M12*1.5 of property 12.9 class, tightened below yield point can be replaced by a M10*1.5 as all bolt have applied same torque range 140Nm.
- Gasket sealing status is perfect for M12*1.5 cylinder head bolt showing value of gasket sealing status is 1, there is some point opening for M10*1 cylinder head bolt showing value of gasket sealing status is less than1.
- Maximum stresses occurs near to bolt area on the gasket which is in the range of 120 N/mm²further stresses occurs near oil hole up to 60 N/mm²and stresses occurs near cooling hole up to 50 N/mm²for all bolts.
- Effect of PCD of bolted joint by applying uniform pressure distribution in this section analyze the effect of elongation in cylinder head bolt and maximum element stresses inside cylinder head bolt. Here four iteration had taken first decrease PCD by 8mm, second decrease PCD by 16mm, third increase PCD by 6mm and fourth iteration increase PCD by 12mm i.e. 101.35mm, 109.35mm, 123.35mm and 129.35mm. Here elongation of bolt and elemental stresses less for lower PCD and stresses and elongation of bolt is increasing as increase in PCD of bolted joint.
- As we decrease PCD of bolted joint the gasket sealing status is perfect showing value of gasket sealing status is 1, while we increase PCD of bolted joint the gasket sealing status is perfect showing value of gasket sealing status is less than 1.
- PCD of bolted joint close to cylinder bore in sealing point of view.

For 101.35mm, 109.35mm, 123.35mm and 129.35mm PCD of bolted joint, Maximum stresses occurs near to bolt area on the gasket which is in the range of 90 to 120 N/mm²further stresses occurs near oil hole up to 42 to 60 N/mm²and stresses occurs near cooling hole up to 30 to 50 N/mm²for all bolts.

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NOMENCLATURE

T = torque applied in N-m,
P = friction angle under head
p' = thread friction angle
D_{km} =effective diameter of bearing area in m
d_2= pitch diameter in m
d_w=head diameter with cylinder head
d_h = clearance hole diameter
P = pitch in m
μ_k= co-efficient of friction in bearing area of bolt or nut=0.14
Eg = Young modulus of elasticity of gasket
Eb = Young modulus of elasticity of cylinder head bolt
Dp = Pitch diameter of cylinder head bolt
Dm = Mean diameter of stud head

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