Stress Analysis of Shrink-fitted Cylinder Head and Valve Guide Connections via Finite Element Analysis

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ABSTRACT

In an internal combustion engine, the cylinder head rests on the top of the cylinder block. One of the parts of head assembly is the valve guide, which is used to orient the valves correctly. Typically valve guide is made up of cast iron and fitted in cylinder head with the use of interference fit. The study aims to find the stress developed due to interference fit in valve guide and cylinder head of air cooled diesel engine. This is done using 3D simulation model and FEA. The investigation aims to provide the design intent to resolve valve guide loosing issue in operating temperature ranges.

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Keyword: *- Valve Guide, Cylinder Head, Interference Fit, Contact Pressure, FEA*

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1. INTRODUCTION

In an internal combustion engine, the cylinder head sits above the cylinder block and it closes in the top of the cylinder, forming the combustion chamber. It is made up of aluminium and its alloy. One of the parts of head assembly is the valve guide, which is used to orient the valves correctly. Typically valve guide is made up of cast iron. This paper investigates the assembly of parts having a negative clearance, called interference, press or force fits. These types of fits are also commonly used for assembling bearing, attaching gear or sprockets to shaft, inserting dowel pin into hole. The Finite Element Method (FEM)-based stress analysis of interference-fitted connections is more complete and accurate than those obtained from the traditional methods. The finite element model determines for the stresses in the axial direction, which usually are not considered in the traditional design method.[1]

There are various parameters which affect on contact pressure(stress) and hoops stress like thickness of valve guide, stiffness of cylinder head (G/D ratio), material of valve guide, and material of cylinder head. This is done for different interference fit and operating temperature range. From previous study it is shown that, thickness of valve guide and stiffness of cylinder head are the main parameter which affects a lot. These parameters are changing for minimum and maximum interference fit for operating temperature range.

2. LITERATURE REVIEW

Y. Zhang et al. [1] has studied interference fit via FEM. In their studies of interference fits in ring gear-wheel connections show that the traditional design method based on thick-wall cylinder theory had some limitations. Lame's equations did not give good results for the interference stresses and deformations. This is because of the complex geometry of the problem, which involves a thin ring on a hollow, stepped shaft that protrudes unspecified,

NOMENCLATURE

r_o	: Outer radius of cylinder head (mm)	d_o	: Original diameter (mm)
R	: Inner radius of cylinder head or Outer radius of valve guide (mm)	d_f	: Final diameter (mm)
r_i	: Inner radius of valve guide, mm	T_r	: Room temperature (°C)
P_c	: Contact Pressure (MPa)	T_f	: Final temperature (°C)
E_o	: Modulus of Elasticity of cylinder head (MPa)	σ_H	: Hoop stress in the cylinder (MPa)
E_i	: Modulus of Elasticity valve guide (MPa)	σ_r	: Radial stress in the cylinder (MPa)
L	: Length of valve guide (mm)	σ_z	: Axial stress in the cylinder (MPa)
C	: Coefficient	σ_{eq}	: Equivalent stress, i.e., von- Mises stress (MPa)
F	: Push-out force, (Kg)	δ	: Interference fit, (mm)
F	: Coefficient of friction	ν_i	: Poisson's ratio of valve guide
G	: Outer diameter of cylinder head (mm)	ν_o	: Poisson's ratio of cylinder head
D	: Inner diameter of valve guide (mm)	α	: Coefficient of thermal expansion, (m/mK)
Δd	: Change in diameter (mm)		

large distances beyond the He has introduced two safety factors λs and λp which provides a new method for evaluating the quality of interference fits.

Adnan O'zel [2] analysed the stresses of shrink-fitted joints for various fit forms via FEM. He has worked on stress and deformation analysis of shrink fitted shaft (without hollow) with six different fit forms having same size fit features was performed by FEM and the effects of geometry on stress and deformation were researched. He has compared their value with maximum normal stress, maximum shear stress and von-Mises stress.

Ayub A. Miraje [6] has introduced the optimum design for minimization of thickness of three-layer shrink-fitted compound cylinder to get equal maximum hoop stresses in all the cylinders. He has applied Lame's theory for compound cylinder. His effort was made to find optimum minimum thicknesses of three cylinders so that material volume was reduced and hoop stress was equal in all the cylinders. It was clearly proved that the difference in analytical and ANSYS Software results is within acceptable limits. This difference is due to numerical techniques of FEM in ANSYS.

Sunil A. Patil [7] carried a FEM of optimized compound cylinder. Optimally designed compound cylinder had equal maximum hoops stress in both the inner and outer cylinders. He proposed many design parameters in his study on design of compound cylinder, that value of hoops stress is closer to value of yield stress. He had found out three important parameters for optimization interface diameter, interference and outside diameter kept other parameter such as material, internal diameter constant

Investigation of stress distribution on interference fits by focusing on the following objectives:

i. Calculate the contact pressure and hoops stress between valve guide and cylinder head assembly at operating temperature range.

ii. Compare the analytical results with FEA and experimentally results of contact pressure.

3. METHODOLOGY

The connection in cylinder head and valve guide occurs between inner surface of cylinder head and outer surfaces of valve guide. Stresses in contact surface of selected machine elements were investigated. These results are comparing with FEA results. Also experimentally we calculate push-out for aluminium cylinder head and cast iron valve guide.

I. Engine Details

For HA4 (H- series Air cool 4 Cylinder) diesel engine there specification are as below:

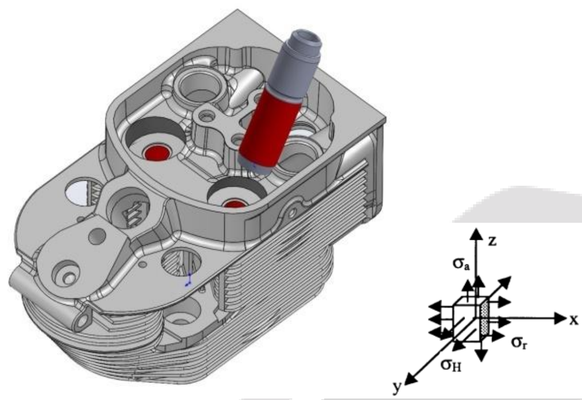


Fig. 1 a. Actual cylinder head and valve guide assembly
b. stresses acting on an element of the mating surface

Table 1 Specification of HA4 Diesel Engine

Sr.No.	Parameter	Specification
1	Engine name	HA4
2	Compression Ratio	18:1
3	Displacement Volume	3.77cc
4	Engine type	Diesel engine
5	Max. Power of engine	31.6kW at 1500 rpm
6	Valve guide material	Cast iron
7	Cylinder head material	Aluminum alloy

Fig. 1(a) shows the interference-fitted assembly of a cylinder head and valve guide, and Fig. 1(b) shows the stresses acting on an element of the mating surface.

II. Material Properties

Material property for teased cylinder head and valve guide assembly as shown in table 2 and table 3 :

Table 2. Material Properties Of Cast-Iron,

Sr.No.	Parameters	Values
1	Tensile Strength (S_{ut})	430MPa
2	Yield Strength (S_{yt})	276 MPa
3	Compressive Yield Strength (S_{yc})	1210MPa
4	Young's modulus (E_0)	110000 MPa
5	Poisson's Ratio (ν_o)	0.27
6	Coefficient of Thermal Expansion (α)	$10.8x10^{-6}$ mm/mm.K
7	Thermal Conductivity (k)	50.2 W/mK

Table 3. Properties of Aluminium Alloy,

Sr.No	Parameters	Values
1	Tensile Strength (S_{ut})	300 MPa
2	Yield Strength (S_{yt})	286 MPa
3	Compressive Yield Strength (S_{yc})	290 MPa
4	Young's modulus (E_i)	71000 MPa
5	Poisson's Ratio (ν_i)	0.33
6	Coefficient of Thermal Expansion (α)	22x10-6 mm/mm.K
7	Thermal Conductivity (k)	160 W/m.K

III. COMPUTATION OF STRESSES BASED ON LAME'S EQUATION

Interference fit Calculation

In cylinder head and valve guide assembly, the interference fit and its tolerance calculation as shown below:

For loose fit = lower limit of inner cylinder – upper limit of outer cylinder
= 0.0026

For tight fit = upper limit of inner cylinder – lower limit of outer cylinder
= 0.0037

Lames equation

The interference fit of two cylinders is usually dealt with based on Lame's equation, which is applied in the elastic range. Equations used for computing the stresses and interference with this method are listed in Table 5.

TABLE 5 COMPUTATIONAL EQUATIONS BASED ON LAME'S THICK-WALL CYLINDER THEORY

Parameters and definition	Outer part	Inner part
Coefficient (C)	$C_o = \frac{r_o^2 + R^2}{r_o^2 - R^2} + \nu_0$	$C_i = \frac{R^2 + r_i^2}{R^2 - r_i^2} - \nu_i$
Contact Pressure (P_c)	$P_c = \frac{\delta_{radial}}{2b\left[\frac{1}{E_o}C_o + \frac{1}{E_i}C_i\right]}$	
Radial stress at the mating interface (σ_r)	$\sigma_{r\,(o)} = -P_c$	$\sigma_{r\,(i)} = -P_c$
Hoop stress at the mating interface (σ_H)	$\sigma_{H\,(o)} = -P_c \frac{r_o^2 + R^2}{r_o^2 - R^2}$	$\sigma_{H\,(i)} = -P_c \frac{R^2 + r_i^2}{R^2 - r_i^2}$
Push-out Force	$F=2\pi RLP_c f$	$F=2\pi RLP_c f$

Analytical Calculation

For 250°C, we use radial interference fit= 0.0152mm

1) Contact pressure

$$P_c = \frac{\delta_{radial}}{\frac{R}{E_o}\left(\frac{r_o^2+R^2}{r_o^2-R^2} + \nu_0\right) + \frac{R}{E_i}\left(\frac{R^2+r_i^2}{R^2-r_i^2} - \nu_i\right)}$$

$P_c = 41.83$MPa

2) Hoops stress at outer cylinder

$$\sigma_{H_o} = -P_c \frac{r_o^2 + R^2}{r_o^2 - R^2}$$

σ_{max} = -75.10 MPa (-ve sign indicate compressive)

3) Hoops stress at inner cylinder

$$\sigma_{H_i} = P_c \frac{R^2 + r_i^2}{R^2 - r_i^2}$$

σ_{max} = 85.90 MPa (Tensile)

4) Push-out force

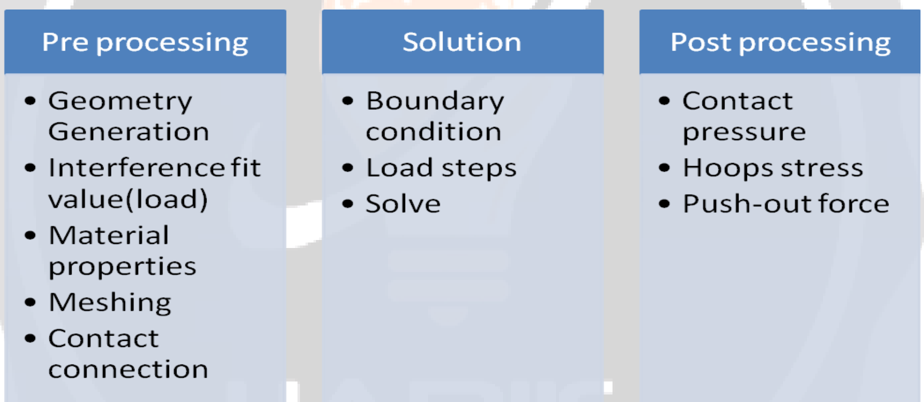
F= $2 R l P_c$ f

F = 7537.92 N

F= 768 kg

5. COMPUTATION OF STRESSES BASED ON FEM

FEA model is built up in which contact is defined between valve guide and cylinder head. The interference is relatively very small compared with the sizes of guides, it is essential to specify the interference (or overclosure) in the model numerically rather than building it in the model geometrically. This can precisely define the overclosure or interference. We take a cut section from whole cylinder head assembly for better calculation and easy to simulate in ANSYS.

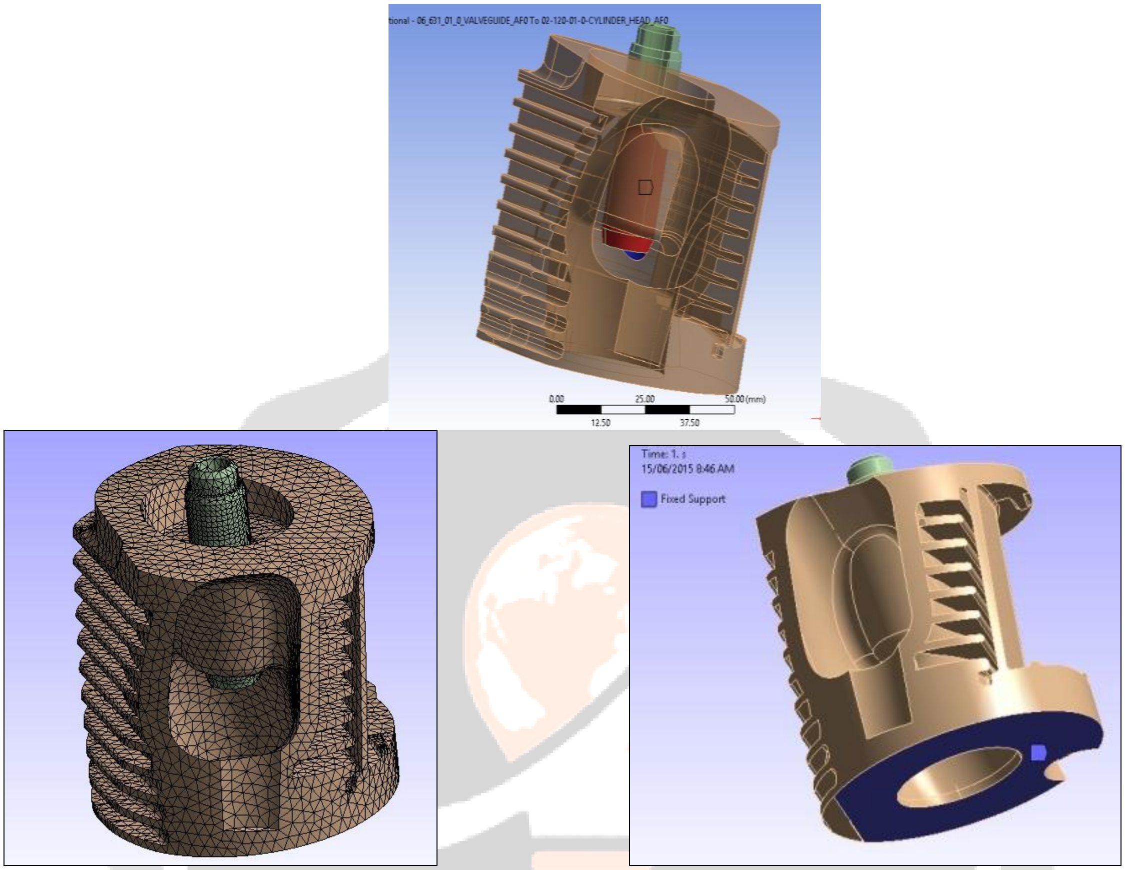


Under each "Contact Region", the Contact and Target surfaces are shown. The normal of the Contact surfaces are displayed in red while those of the Target surfaces are shown in blue. The Contact and Target surfaces designate which two pairs of surfaces can come into contact with one another. We used CONTAC174 and TARGE169 element for contact and target surface.

For solving contact problem we used *Augmented Lagrange method*. It is an iterative series of penalty methods to enforce contact compatibility. Contact tractions (pressure and friction stresses) are augmented during equilibrium iterations so that final penetration is smaller than the allowable tolerance. This offers better conditioning than the pure penalty method and is less sensitive to magnitude of contact stiffness used, but may require more iterations than the penalty method.

The *Normal Contact Stiffness* k_{normal} is the most important parameter affecting both accuracy and convergence behavior. A large value of stiffness gives better accuracy, but the problem may become more difficult to convergence. If the contact stiffness is too large, the model may oscillate, with contacting surfaces bouncing off of each other. Therefore here we used *Normal contact Stiffness* value is 0.01 for analysis. And Coefficient of friction used for frictional contact is 0.15.

Because it’s a complicated geometry we are selecting 10 node tetrahedral elements for meshing. Map meshing used for contact and target surface. We calculate the mesh convergence for this model.



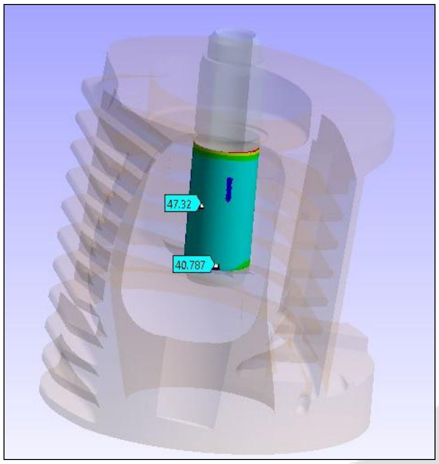
(a) (b)

Fig. 2 Cylinder head and valve guide assembly a) Contact Finite element mesh b) Fixed support

The boundary conditions used in the Finite element analysis are as follows.

The cylinder head sits on the cylinder block. So we fixed the bottom of cylinder head as shown in figure 2 (b). Also we take body temperature 250^{o}C to the assembly.

In FEA we calculate contact pressure and hoops stress is calculate using normal stress in Y-direction, in cylindrical co-ordinate system. These results are match with



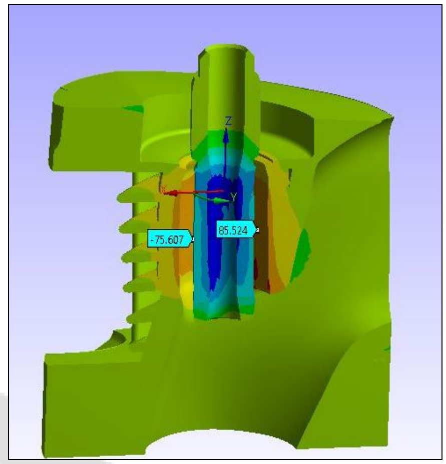


Fig. 3 Contact pressure in cylinder head and valve guide assembly

P_c=47.05MPa (Average)

Fig. 4 hoops stress at valve guide and cylinder head

For σ_{H_o}=85.52 MPa and σ_{H_o}= -75.95 MPa

I. EXPERIMENTAL SETUP

In One side of mandrel is inserted in the valve guide and on the other side a load cell is fitted with threads. A Load cell is mounted on the valve guide such that, it transfers the loadand calculate the force exerted on it. Load cell is connected to the digital controller.

When load is gradually applied on the load cell digital controller shows reading. The resolution of the digital controller is 1Kg. Range of digital controller is 1350 Kg.

Fig.5Experimental setup for calculating Push out force

The intention of the current experiment is to calculate push-out force. In this experiment, tested head is made up of aluminium alloy and valve guide is made up of cast iron. Place the whole assembly on inclined fixture to get valve guide straight position check where it is straight or not. If yes then apply the load. Press load is pushing valve guide to downward. When load apply on load cell, there is strain gauge in which deflection occur. At the same time digital controller showing result in screen.

From experiment we get push-out force, from push-out force we calculate contact pressure and hoops stress.

I. RESULTS AND OBSERVATION

The analytical, FEA and experimental investigation of valve guide contact analysis has been conducted. The results

of these investigations are represented in the graphical form to find the best possible combination of valve guide and cylinder head parameter.

Table 6 Comparison between analytical FEM and experimental data

	Result	Contact Pressure (MPa)	Hoops stress Inner (MPa)	Hoops stress outer (MPa)	Push out force (kg)
1	Analytical	41.83	-75.10	88.90	768.31
	FEA	47.05	-75.95	85.52	860.56
	Error	11%	-	-	11%
2	Analytical	41.83	-75.10	88.90	768.31
	Experimental	44.01	-79.07	58.25	809
	Error	6%	-	-	6%

Also we found out the error in between valve guide and cylinder head assembly at 250^{o}C. Result from analytically, FEA and experimentally are as follows:

At 250^{o}C operating temperature, interference fitvalue is 0.00152. Thickness of valve guide changes to 1) 4.9 2) 7 and 3) 9.1. For changing the thickness we have to calculate inner and outer radius of valve guide. But outer radius made a contact with cylinder head. Therefore we are changing the inner radius. From this calculation we get inner radius 1) 5.05mm 2) 4mm and 3) 2.95mm respectively. We changing inner radius and calculate contact pressure. Also we plotted graph in between inner radius and hoops stress with the help of trend line. This shows result correctly.

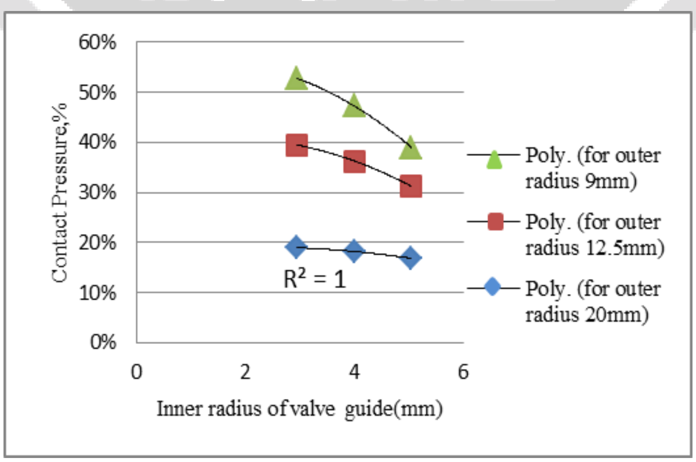


Fig. 6(a) Effect of inner radius 1)2.65 2) 4 3) 5.05mm on contact pressure when outer radius of cylinder head is 9mm, 12.5mm and 20mm respectively

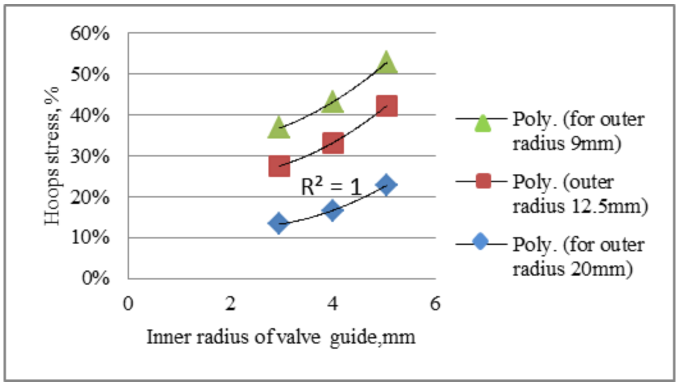


Fig. 6(b) Effect of inner radius 1)2.65 2) 4 3) 5.05mm on hoops stress when outer radius of cylinder head is 9mm, 12.5mm and 20mm respectively

At 250^{o}C operating temperature, interference fitvalue is 0.00152. Stiffness of cylinder head (G/D ratio) changes to 1) 2.25 2) 3.125 and 3) 5. For changing the stiffness we have to calculate inner and outer radius of valve guide. But inner radius made a contact with cylinder head. Therefore we are changing the outer radius. From this calculation we get innerradius 1) 9mm 2) 12.5mm and 3) 20mm respectively.

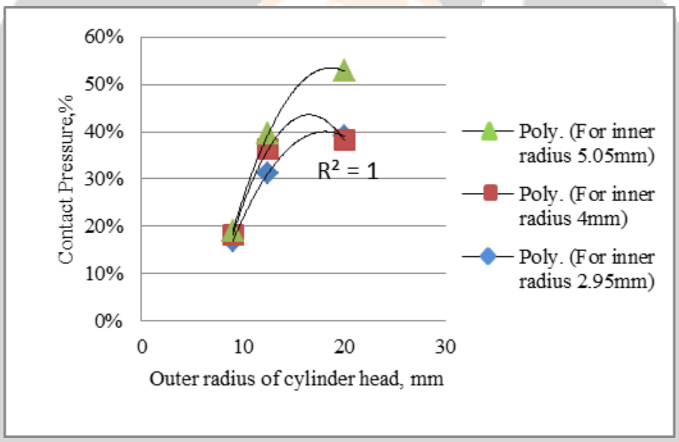


Fig. 8(a) Effect of outer radius 1)9 2) 12.5 3) 20mm on contact pressure when inner radius of cylinder head is 5.05mm, 4mm and 2.95mm respectively

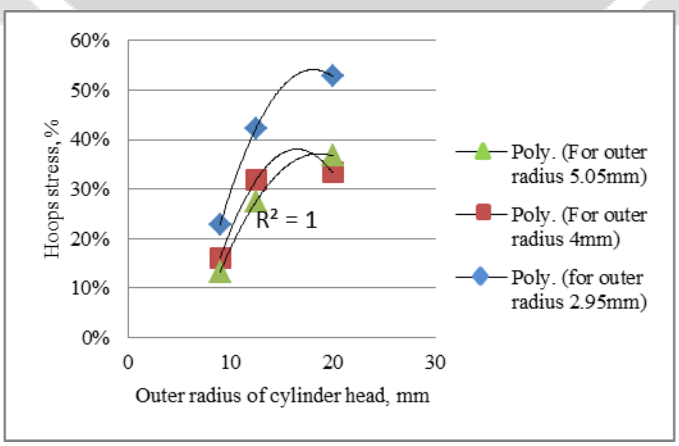


Fig. 8(b) Effect of outer radius 1)9 2) 12.5 3) 20mm on hoops stress when inner radius of cylinder head is 5.05mm, 4mm and 2.95mm respectively

At 250^{o}C operating temperature, interference fitvalue is 0.003735 Thickness of valve guide changes to 1) 4.9 2) 7 and 3) 9.1. For changing the thickness we have to calculate inner and outer radius of valve guide. But outer radius made an contact with cylinder head. Therefore we are changing the inner radius. From this calculation we get inner radius 1) 5.05mm 2) 4mm and 3) 2.95mm respectively. We changing inner radius and calculate

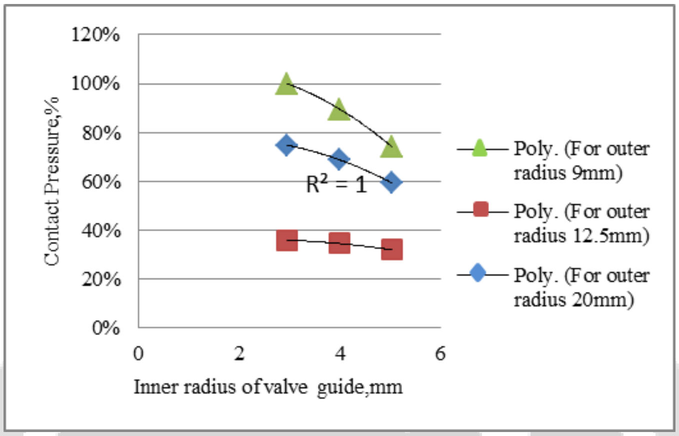


Fig. 9(a) Effect of inner radius 1)2.65 2) 4 3) 5.05mm on contact pressure when outer radius of cylinder head is 9mm, 12.5mm and 20mm respectively

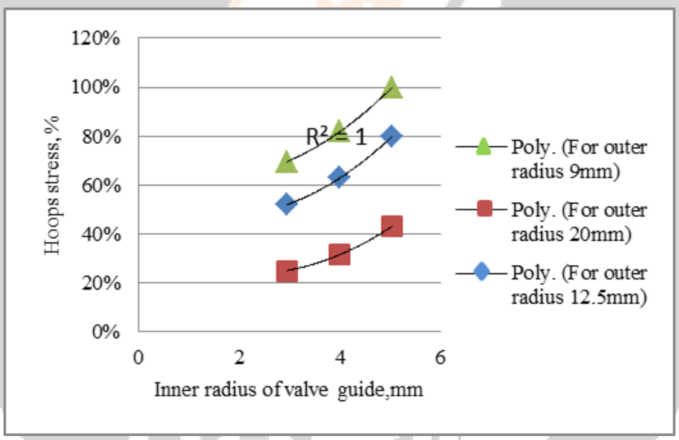


Fig. 9(b) Effect of inner radius 1)2.65 2) 4 3) 5.05mm on hoops stress when outer radius of cylinder head is 9mm, 20mm and 12.5mm respectively

At 250^{o}C operating temperature, interference fitvalue is 0.00373. Stiffness of cylinder head (G/D ratio) changes to 1) 2.25 2) 3.125 and 3) 5. For changing the stiffness we have to calculate inner and outer radius of valve guide. But inner radius made an contact with cylinder head. Therefore we are changing the outer radius. From this calculation we get inner radius 1) 9mm 2) 12.5mm and 3) 20mm respectively. We changing inner radius and calculate contact pressure. Also we plotting graph in between outer radius and hoops stress.

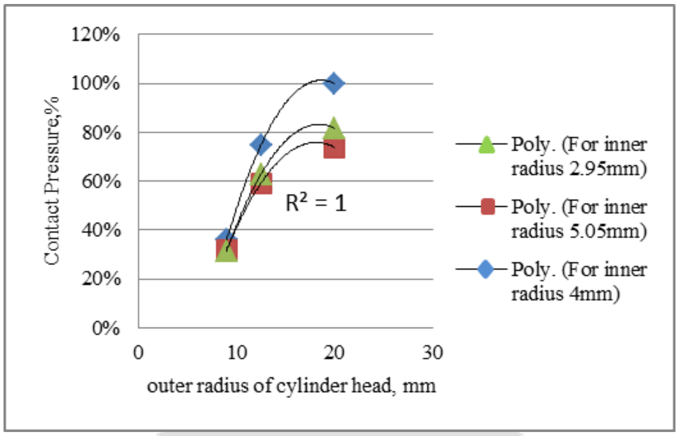


Fig. 10(a) Effect of outer radius 1)9 2) 12.5 3) 20mm on contact pressure when inner radius of cylinder head is 2.95mm, 5.05mm and 4mm respectively

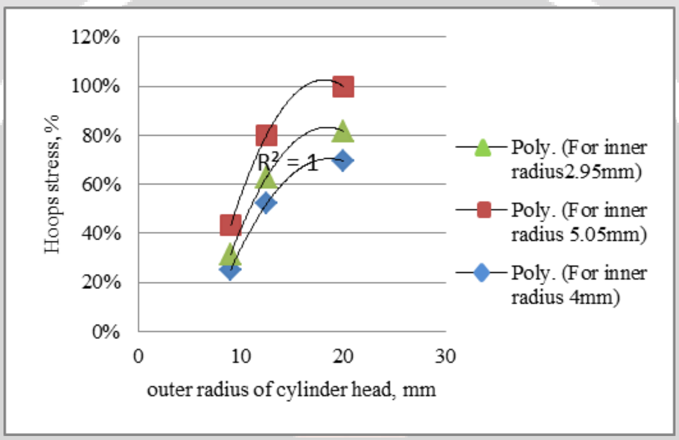


Fig. 10(b) Effect of outer radius 1)9 2) 12.5 3) 20mm on hoops stress when inner radius of cylinder head is 2.95mm, 5.05mm and 4mm respectively

II. OBSERVATION

- From the fig. 7, fig. 8, fig 9 and fig 10 we can say that, by changing the thickness of valve Guide and cylinder head stiffness contact pressure is directly changing. For maximum contact pressure, maximum push-out force required.

- In Fig. 7 (a) for actual model, Inner radius is 4mm and outer radius is 12.5mm, we get contact pressure 36%. For better combination of inner radius of valve guide and cylinder head is more than 36%. And also it is below allowable stress limit.

- From the fig. 7, fig. 8, fig. 9 and fig. 10 we conclude that, minimum inner radiusof valve guide is 2.95mm and maximum outer radius of cylinder head is 20mm is best combination.

III. CONCLUSION

- Because of the complex geometry of the problem the analytical results are not perfectly match with FEA and Experimental results. Contact pressure results are match within 11% and 6% error.

- The effect of stress distribution at the interface between valve guide and cylinder head of diesel engine is studied. By changing the thickness of valve guide and stiffness of material from this investigation, we got combinations which overcome the valve guide loosening issue and that combination is below allowable stress at operating temperature range.

- The effect of various parameters on contact pressure is studied. It is observed that when stiffness of cylinder head increases and thickness of valve guide decreases, the contact pressure on valve guide and cylinder head increases by 17% for loose interference fit and 31% for tight interference fit. This is best case for valve guide as it stays fit in cylinder head at operating temperature range.

IV. ACKNOWLEDGMENT

Thanks to Mr. Kedar Kanase, General Manager in Kirloskar Oil Engine Ltd. for giving the opportunity to work on this challenging topic. And thanks to Prof. P.B. Joshi, HOD of MIT,Pune for support.

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