

Stress Analysis Of Rocker Arm Of Internal Combustion Engine

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ABSTRACT

The effect of total force i.e. gas pressure, inertia force and initial spring force acting on the existing rocker arm is evaluated. It is found that the maximum stress is occurred at the push rod side end. For optimization the various models of rocker arm are developed, keeping its weight constant. The effect of total force on this various models are evaluated. The model showing less stress is found suitable as compared to other models. Geometric models are developed in software Pro-E and for analysis software ANSYS 11 is used.

Keywords: Gas pressure, Inertia force, Initial spring force, Analysis.

I. INTRODUCTION

A rocker arm is a reciprocating lever used in an internal combustion engine to actuate the inlet and exhaust valves motion as directed by the cam and follower. The rocker arm of the exhaust valve is the most heavily loaded. On the other hand the force required to operate the inlet valve is comparatively less. However, it is usual practice to make rocker arms for inlet and exhaust valves identical. This result in ease of manufacturing. The main objective of the rocker arm as a 'lever' is to change the direction of force and not the multiplication of the effort.

While operating, the internal combustion engine valve produces the various forces which are acting on the rocker arm. For absorbing this forces, the rocker arm model must be properly designed. There is always possibility of optimizing the designed rocker model by means of stress analysis, so that stresses induced in the rocker should be minimized.

NOTATION

ω Angular Velocity
E Modulus of Elasticity
 σ bending stress
 P_c Cylinder pressure
 P_g Force equivalent to gas pressure
 P_a Inertia force
 P_i Spring force

II. NUMERICAL RESULTS

The various forces acting on the rocker arm of exhaust valve are :-

- (i) The gas pressure (P_g) on the valve when it opens.
- (ii) The inertia force (P_a) when the valve moves up.

(iii) The initial spring force (P_i) to hold the valve on its seat against suction or negative pressure inside the cylinder during the suction stroke.

Calculation for gas pressure, P_g

The gas load P_g is given by,

P_g = area of valve \times gas pressure when the exhaust valve opens

$$P_g = \left(\frac{\pi d_v^2}{4}\right) P_c$$

Where,

d_v = diameter of valve head (mm)

P_c = cylinder pressure or back pressure when the exhaust valve opens (MPa)

Cylinder pressure, P_c calculation

$$\text{Indicated Power (I. P.)} = \frac{\text{Horse Power (H. P.)}}{\text{Mechanical Efficiency}}$$

Mechanical Efficiency = 80% from engine specification

$$I. P. = \frac{215}{0.8} \quad \text{H.P.} = 215 \text{ (maximum horse power assumed)}$$

$$I. P. = 268.75 \text{ H.P.}$$

$$I. P. = 200487.5 \text{ watt.}$$

$$I. P. = \left(\frac{MEP \times L \times A \times N}{60}\right) \times \text{No. of cylinders}$$

Where,

MEP = Mean Effective Pressure

L = stroke = 4.02 inch = 0.12 m (assumed)

A = area = $\frac{\pi D_b^2}{4}$ (D_b = bore diameter is 0.102m assumed)

N = 2500 rpm

No. of cylinder = 6

$$IP = \left(\frac{MEP \times 0.12 \times \left(\frac{\pi(0.102)^2}{4}\right) \times \left(\frac{2500}{2}\right)}{60}\right) \times 6$$

$$200487.5 = \left(\frac{MEP \times 0.12 \times 0.0081 \times 1250}{60}\right) \times 6$$

$$200487.5 = 0.122 \times MEP$$

Cylinder pressure, $P_c = \text{MEP} = 1636537.8 \text{ N/m}^2$

Cylinder pressure, $P_c = 1.64 \text{ N/mm}^2$

Gas pressure,

$$P_g = \left(\frac{\pi \times d_v^2}{4} \right) \times P_c$$

$$P_g = \left(\frac{\pi \times 36^2}{4} \right) \times 1.64 \quad (\text{valve dia., } d_v = 36 \text{ mm assumed})$$

Gas pressure, $P_g = 1669.32 \text{ N}$

Calculation for Inertia force, P_a

As the valve moves down, the inertia force acts opposite to the direction of motion. The upward inertia force P_a is given by,

$$P_a = m\alpha$$

Where, m = mass of valve (kg)

α = acceleration of valve (m/s^2)

$m = 0.121 \text{ kg}$ (assumed)

Calculation for acceleration, α

Engine speed is 2500rpm, this gives a cam shaft rotation of 1250 rpm (in a four stroke engine the cam turns at half the cam shaft speed). So the time taken for one revolution of the cam shaft is 0.048 seconds.

Total crankshaft angle when the valve is open, $58+180+50 = 288^\circ$ (assumed)

Total angle of cam action for four stroke engine,

Angle of camshaft = $\frac{1}{2} \times$ angle of camshaft.

Angle of camshaft = $\frac{1}{2} \times 288^\circ = 144^\circ$

Cam is open and close the valve for 144° of its rotation. Hence the complete valve cycle is $\frac{2}{5}$ of camshaft rotation or 0.019 seconds.

The equation describing simple harmonic motion is,

$$x = a \times \cos(\omega t)$$

Where, x =displacement

a = amplitude

t = time

ω = angular velocity in radian per second

$$= 2 \times \pi = 2 \times 3.142 \quad (\text{time taken for one cycle})$$

So the ω value here is,

$$\frac{2 \times 3.142}{0.019} = 330.737 \text{ rad/sec.}$$

Differentiating the equation for expression for displacement gives ,

$$v = -\omega \times \text{amplitude} \times \sin(\omega t), \text{ where } v = \text{velocity}$$

Then differentiating again,

$$a = -\omega^2 \times a \cos(\omega t)$$

The maximum acceleration occurs when the term $\cos(\omega t)$ has the maximum value of 1, this occurs at the extreme of the motion.

Cam has the lift of 11 mm. (measured value of the cam shaft of 5.9 liter Cummins B-series engine)

The amplitude of the motion is $5.5 \text{ mm} = 0.0055 \text{ m}$.

So the maximum acceleration is given by,

$$\text{acceleration}_{\max}, a = (330.737)^2 \times (0.0055) = 601.63 \text{ m/s}^2$$

Inertia force, $P_a = m\alpha$

$$= 0.121 \times 601.23$$

$$= 72.596 \text{ N}$$

Calculation for initial spring force, P_i

Following is the valve spring data assumed:-

Unloaded valve spring length = 53.6mm

Valve spring length installed condition (valve fully closed) = 42.5 mm

Axial spring force at the axial deflection of the spring ($53.6 - 42.5 = 11.1 \text{ mm}$) = 22.6 kg

$$= 22.6 \times 9.8$$

$$= 221.48 \text{ N}$$

So initial spring force $P_i = 221.48 \text{ N}$

Total Force on the Rocker Arm :-

Total force (P_e) on the rocker arm of the exhaust valve is given by,

$$P_e = P_g + P_a + P_i$$

$$= 1669.32 + 72.796 + 221.48$$

$$P_e = 1963.593 \text{ N}$$

By taking the moment the force on the push rod end is calculated i.e. of 3057.45N

Above calculated force of 1963.593N is applied to the valve end and the force of 3057.456N is applied to pushrod end. From the ANSYS result obtained it is found that expected maximum stresses near the bearing bore as shown in fig.1 where the moments are highest. Though the moments on either side of the boring bore are same, the bending stresses on the push rod side are higher because of the curvatures and stress concentration. The deflections and stresses are in close agreement with the theoretical calculation.

Since the bending stresses and equivalent stresses obtained by ANSYS are much lesser than the permissible stress values. There is a scope for certain optimization in shape which would equally distributed stresses as against the current results which show stress concentration in a particular zone. Hence it was decided to optimize the shape without altering the mass. Following section described the alternate model of rocker arm.

2.1 ALTERATION OF ROCKER ARM

For decreasing the stresses, the model of rocker arm is modified. The sharp edges of outer circle is removed by increasing the radius of edge. Also the material in the outer side of the push rod circle is increased. While doing this emphasis been given to the fact that the weight of the rocker arm should not been increased.

Changes in the existing model is given below:

Models-I: In this model the edge of the rib is changed from radius of 0.2mm to 0.4mm.to remove the sharp contour

Modal-II: In this model the outer radius of push rod end outer circle is taken as 0.5mm. while the radius of the outer surface with the rib is taken as 4.5 mm.

Model-III: In this model the outer edge of push rod end outer surface is taken as 1mm. while the outer surface radius with the rib is taken to be 4.5 mm.

Model-IV : In this model the outer edge radius of push rod end outer circle is taken as 1.5 mm. while the radius of outer surface to the rib is taken to be 5 mm. All the alternate models were further analyzed.

The result obtained are discussed in following chapter.

III. RESULTS IN ANSYS

“Fig. 1” shows the stress distribution of rocker arm in which the maximum stress of 560MPa is observed near the push rod end.

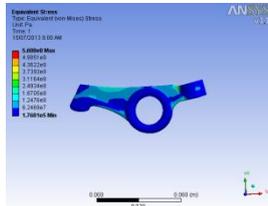


Fig. 1: The equivalent stress distribution of rocker.

“Fig. 2” shows the ANSYS result of Model-I, in which the maximum equivalent stress of existing rocker arm value is reduced to 380MPa.

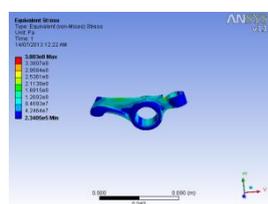


Fig. 2: Model-I stress result.

Model-II result shows that there is further reduction in maximum equivalent stress of Model-I value of 380MPa to 325MPa. The stress analysis result of Model-II is shown in the “Fig. 3”

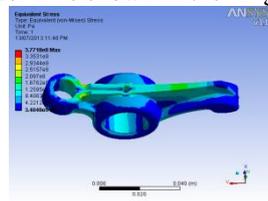


Fig. 3: Model-II stress result

The “Fig. 4” result shows the maximum equivalent stress of Model-III which is lesser than the maximum equivalent stress value of Model-II.

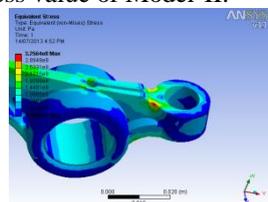


Fig. 4: Model-III stress result.

The result of Model-IV shows the maximum equivalent stress obtained is of 302MPa which is less than the maximum equivalent stress of Model-III. The model-IV result is shown in “Fig. 5”.

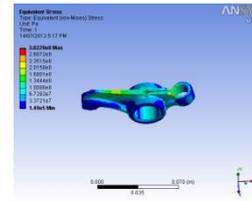


Fig. 5: Model-IV stress result.

IV. CONCLUSION

The particular section near the top face of bearing bore towards push rod side in the current rocker arm shows high stress concentration zone. In respect of current design of rocker arm it may be concluded that the stresses in the rocker arm are not uniformly distributed and there is a scope for modifying the contour on the said face so as to reduce the stress concentration.

The curvature in the original design push rod hole with the main body of rocker arm is too sharp its radius was varied from 0.5mm to 1.5mm for a four different values at the same time the region beneath this curve is filled with material to strengthen the region. This resulted in strengthening as analysis showed significant decrease in equivalent stresses in subsequent alternate models. The best result was obtained for Model-IV having the maximum stress of 302MPa and the wt. of body is 5.3262×10^{-002} kg is find suitable as compared to previous three models.

Further it may also be concluded that though the current rocker arm is in having the maximum equivalent stress much less than the ultimate values and the rocker arm is serving satisfactorily, there is scope for modification in respect of shape The modification proposed are a few and there could be number of ways, the variation could be made. However it does not mean that there is an immediate need to have any alternate design.

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