STATIC AND DYNAMIC ANALYSIS OF CONNECTING ROD OF COMPRESSOR

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ABSTRACT

The objective of study is performed on a cast iron connecting rod of a compressor for the static and dynamic analysis. Literature survey suggests cyclic loads comprised of static tensile and compressive loads are often used for design and analysis of connecting rods. The S-N approach by Modified Goodman criterion to fatigue life prediction of the connecting rods is presented. The possibility failure of the connecting rod approach by Failure Index is the ratio of Maximum Principal Stress to yield strength of material. The three-dimensional finite element model is constructed in PROE and used for Analysis. The model is meshed in ANSYS workbench and used for Analysis.

KEYWORDS: Connecting Rod, FEA Analysis, Failure Index

INTRODUCTION

The function of connecting rod is to transmit the thrust of the crankshaft to piston by translating the rotational motion of crankshaft to the reciprocating motion of piston. Connecting rods are subjected to inertia force due to reciprocating mass and gas forces due to maximum gas pressure results in axial and bending stresses. Bending stresses originate due to eccentricities, crankshaft, case wall deformation, and rotational mass force. Therefore, a connecting rod must be capable of transmitting axial tension, axial compression, and bending stresses. Failures of connecting rods are often caused by bending loads, acting perpendicular to the axes of the two bearings. Failure in the shank section as a result of these bending loads occurs in any part of the shank between piston-pin end and crank-pin end.

Pravardhan S. Shenoy and Ali Fatemi shown that dynamic analysis is the proper basis for fatigue performance calculation and explain .if the failure index is less than one then the possibility of failure is less [2]. Pravardhan S. Shenoy conducted dynamic analysis of loads and stresses in the connecting rod component and explain the tensile load was applied over 180° of crank contact surface with cosine distribution, whereas compressive load was applied as a uniformly distributed load over 120° of crank contact surface [4]. Vivek C. Pathade, Ajay N. Ingale the result of compressive, tensile and bending stresses is maximum stresses are developed at the fillet section of small end are greater than big end [5].

Table 1: Configuration of the Compressor to which the Connecting Rod Belongs

<table>
<thead>
<tr>
<th></th>
<th>Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>crankshaft radius</td>
<td>R1 40 mm</td>
</tr>
<tr>
<td>connecting rod length</td>
<td>R2 216 mm</td>
</tr>
<tr>
<td>mass of reciprocating assembly</td>
<td>Mp 34.5 kg</td>
</tr>
<tr>
<td>crankshaft angular velocity</td>
<td>N 1500 rpm</td>
</tr>
</tbody>
</table>

LOAD ANALYSIS

Figure 1 shows the pressure crank angle diagram through 360° is used to obtain gas loads and inertia loads varies with crank angle. Forces at the crank end and pin end are function of crank angle at maximum speed of 1500 rpm. Two components of forces are acting one along the length and other normal to it. Under cyclic load results consists angular velocity and angular acceleration at center of crank end. Figure-2 shows the linear acceleration acting at C.G. of connecting
rod and forces generated at ends forms external loads and inertia loads forms internal loads. Static compressive loads and dynamic tensile loads are used for analysis as under the effect of dynamic load, forces at the two ends varies at a given instant of time.

![Figure 1: Crank Angle Vs. Pressure](image1.png)  ![Figure 2: Crank Angle Vs. Linear Acceleration at C.G](image2.png)

**FEA ANALYSIS**

In this section we discuss the modeling of connecting rod, boundary conditions and finite element analysis of connecting rod using FEA. Finite Element method (FEM) simulates a physical parts behavior by dividing the geometry into a number of elements of standard shapes, applying constraints. Uses of proper boundary conditions are very important since they strongly affect the results of the finite element analysis. The connecting rod is modeled in Pro-E. The step file of model is imported in ANSYS workbench.

Definition of FEM is hidden in the world itself. Finite - any continuous object has finite degree of freedom & it’s just not possible to solve in this format. Finite Element Method reduces degree of freedom from infinite to finite with the help of discretization. Element - all the calculations are made at limited number of points known as nodes. Entity joining nodes and forming a specific shape such as quadrilateral or triangular etc. is known as Element. To get value of variable at where between the calculation points, interpolation function is used. Method - There are three methods to solve any engineering problem. Finite Element Analysis belongs to numerical method category. Finite element modelling of any solid component consists of geometry generation, applying material properties, meshing the component, defining the boundary constraints, and applying the proper load type.

**MESH GENERATION**

The geometry is meshed in mechanical model window of an ANSYS 14. The hex dominant method is applied for the geometry. This method is used for applying maximum hexahedron elements to complicated geometry. The body sizing is applied for the whole geometry and element size is given as 2 mm. The FE model of the connecting rod geometry is meshed with hexahedral elements, with the global element length of 2 mm and local element length of 0.3233 mm at the fillets where the stresses are higher due to stress concentrations. The meshed connecting rod with 218624 elements and 790833 nodes is shown in Figure 3.

![Figure 3: Meshed Geometry of Connecting Rod](image3.png)
STATIC ANALYSIS

The static analysis is carried out for tensile and compressive conditions under maximum gas load. The compressive load was applied as a uniformly distributed load over 120° of contact surface. The figure 4 (a) and (b) shows the compressive loads over 120° at crank end and pin end respectively with other ends restrained.

Figure 4: Loading Condition for Compressive Static Analysis at (a) Crank End, (b) Pin End

The tensile load was applied over 180° of contact surface with cosine distribution. The figure 5 (a) and (b) shows the tensile loads over 180° at crank end and pin end respectively with other ends restrained.

Figure 5: Loading Condition for Tensile Static Analysis at (a) Crank End, (b) Pin End

Figure 6: Analysis Results of Compressive Stresses at (a) Crank End, (b) Pin End

Figure 7: Analysis Results of Tensile Stresses at (a) Crank End, (b) Pin End
The figure 6 (a) shows the maximum compressive stress at fillet portion of crank end and (b) shows at pin end. The figure 7 (a) shows the maximum tensile stress at fillet portion of crank end and (b) shows at pin end. From the stress analysis, the stresses at shank region are seen as lower. Table 2 shows the combined compressive and tensile stresses analysis data. It can be seen that the stresses and deflection during the tensile analysis are larger than the compressive analysis.

### Table 2: Boundary Condition and Result Data of Static Analysis

<table>
<thead>
<tr>
<th>Load Type</th>
<th>Compressive Load</th>
<th>Tensile Load</th>
</tr>
</thead>
<tbody>
<tr>
<td>Load Distribution</td>
<td>UDL over 120°</td>
<td>Cosine Load over 180°</td>
</tr>
<tr>
<td>Load at Ends</td>
<td>Pin End</td>
<td>Crank End</td>
</tr>
<tr>
<td>Maximum load (N)</td>
<td>40000</td>
<td>40000</td>
</tr>
<tr>
<td>Maximum stress (MPa)</td>
<td>19.74</td>
<td>11.495</td>
</tr>
<tr>
<td>Deflection (mm)</td>
<td>0.01773</td>
<td>0.01766</td>
</tr>
</tbody>
</table>

**DYNAMIC FEA**

While performing dynamic FEA of the connecting rod, the external loads consisting of reactions or the loads computed at the connecting rod ends at the crank angle were applied to the crank and the piston pin ends of the connecting rod. The angular velocity, angular acceleration, and linear acceleration at the same crank angle were specified in both magnitude and direction for the connecting rod. The inertia and dynamic loads were calculated and applied internally based on these inputs. Figure 8 shows Locations on the connecting rod where stress variation was traced over one complete cycle. The locations at the shank region and fillets at both the ends are selected for maximum compressive and bending stresses. The external ends of both ends for maximum tensile stress.

**Figure 8: Locations on the Connecting Rod where Stress Variation was Traced over One Complete Cycle**

Figure 8 shows the axial and normal forces acting at Pin end and crank end is restrained for dynamic loading. The graph in figure 9 shows that maximum axial force acting at 40° is 32919.5 N and minimum -21240 N at 220°. The maximum normal force at 60° is 9991 N and minimum – 3077 N at 330°. The maximum principal stress at fillet of crank end is 74.417 N at 40° in figure 10.

**Figure 9: Crank Angle Vs Axial, Normal and Resultant Forces at Pin End**
Figure 10: Dynamic Analysis Results when Forces at Pin End and Crank End Restrained

The force is function of crank angle varies with instance of time in dynamic loading. The stress-time history for twelve various locations shown in figure 8. The maximum stress occurs at 40° at fillet of crank end which is 74.717 MPa and the stresses in the shank region are lower. The distances of the locations from the center of crank end is measured and stresses at respective locations are plotted as shown in figure 11 the stresses are increased from crank end to pin end. The variable cross section area causes for decrease in stresses from pin end to crank end.

Figure 11: Max Principal Stresses along Length Due to Forces at Pin End

Figure 8 show the axial and normal forces acting at Crank end and Pin end is restrained for dynamic loading. The graph in figure 12 shows that maximum axial force acting at 220° is 28309.01 N and minimum -37659 N at 30°. The maximum normal force at 310° is 9439.84 N and minimum -18301.80 N at 70°. The maximum principal stress at fillet of pin end is 177.67 N at 40° in figure 13. This variation in results is influenced by the bending stresses and higher stresses occur at transition between shank and pin end.

Figure 12: Crank Angle Vs Axial, Normal and Resultant Force at Crank End
Figure 13: Dynamic Analysis Results when Forces at Crank End and Pin End Restrained

The force is function of crank angle varies with instance of time in dynamic loading. The graph in figure 14 shows the stress-time history for twelve various locations shown in figure 8. The maximum stress occur at 40° at fillet of crank end is 177.67 MPa and the stresses in the shank region are lower. The distances of the locations from the center of crank end are measured and stresses at respective locations are plotted as shown in figure 15. The stresses are increased from crank end to pin end. The variable cross section area causes for decrease in stresses from pin end to crank end.

Figure 14: Time Vs Maximum Principal Stress at Various Locations

For cyclic loading the equivalent mean stress and equivalent stress amplitude is calculated by Modified Goodman criterion. The S-N approach is used to fatigue life prediction of the connecting rods and life range is $10^8$ to $10^9$.

Failure index (FI) is the inverse of the safety factor, and can be defined as the ratio of equivalent stress amplitude
at R= -1 to endurance limit of the material. FI is the criterion of failure nearer to one, the higher the possibility of failure. FI with maximum value 0.83 in transition between shank region and pin end is less than one; hence the possibility of failure is less.

![Failure Index Distribution along Length](image)

**Figure 16: Failure Index Distribution along Length**

**CONCLUSIONS**

In this paper the connecting rod is analyzed for its static & dynamic properties. The analysis shows that the tensile stresses are more than compressive stresses. The stress goes decreasing from pin end to crank end. Also the maximum stresses occurred at transition between shank and both ends, also at the fillet region of both pin end and crank end. There are lower stresses occurred at the shank region of connecting rod, hence there is scope for future development. The failure index of connecting rod is less than 1; the possibility of failure is less.

**REFERENCES**


