

Designing and Stress Analysis of the Industrial Compressor Connecting Rod Depending Upon the Design Inputs Using ANSYS Workbench

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ABSTRACT: To obtain a weight optimised connecting rod of an industrial compressor which can withstand the load acting on it due to compressed gases. The initial design of connecting rod is calculated using standard empirical formulas and designed in Ansys design modular. The connecting rod is used to translate the rotating motion of electric motor to the oscillatory motion of the piston. The rod is subjected to the high cyclic loads of compression and tension. The optimised connecting rod with reduced weight results in the reduced inertial forces in the system. Once the designing of the connecting rod is done FE analysis is done depending on the loading condition of the connecting rod. Loading of connecting rod is calculated based upon the gas pressure and compressor specification. The overall results of the analysis are also validated by the analytical method. Thus the process results in the optimised and effective connecting rod.

KEYWORDS: Ansys, Connecting rod, Designing, Finite element analysis, Industrial Compressor, Optimization.

I. INTRODUCTION

The HVAC and refrigeration are one of the most demanded sector in the industry and thus growing with pace. The high capacity compressors are the essence of these machineries and are required to compress large volume of refrigerants. The high capacities of the compressors are required to match the demands and ranges up to the capacity of 10000 cubic centimetres. The single cylinder capacity of these compressors ranges from 300-500 cubic centimetres. These kind of industrial compressors are utilised for commercial cooling, malls, cold storages, maritime application, railway air-conditioning and in transportation system of medicines etc.

These piston type compressors help in the compressions of refrigerant which finally cools the surroundings and parts. The major constituents of the compressors are piston, cylinder head & block, connecting rod, sleeves, compressor tank, air filter, electric motor, bearings etc.

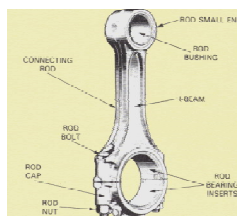


Figure 1: Details of Connecting Rod.

Figure 1 shows a generic model of the connecting rod and highlights its details. The connecting rod is mainly constitutes of I-beam, small end, big end, bolt etc. Connecting rod is one of the most important parts of the unit and affects the performance of the compressor. The connecting rod should be highly durable and efficient. It is a joining link between crank and the piston and is subjected to the high cyclic loading. They are subjected to tensile force due to

International Journal of Innovative Research in Science, Engineering and Technology

(An ISO 3297: 2007 Certified Organization)

Vol. 5, Issue 6, June 2016

the inertial motion of the parts while also subjected to compressive forces due to the pressure of the compressed gases. Hence the set of forces acting on the connecting rod is both complex and complicated. The performance of the connecting rod governs the safety and effectiveness of the compressor and requires skill and precision for its designing.

The most common class of the compressor is piston type compressor which performs the cycle of compressor with the 2 basic strokes of compression and intake. Each of these two strokes has specific role in the cycle of compression and results in different type of forces on to the connecting rod.

The most generic function of connecting rod is:-

1. Converts the rotational motion of the electric motor to the oscillatory motion of the piston.
2. Linkage between crank and piston to utilise the motor power to compress the gases.
3. Also acts a key role in cooling of the system by providing passage to the cooling oil.

The connecting rod is subjected to tensile forces due to the inertial forces and changing direction of the rotating masses. It is subjected to the compressive forces due to the compressed gases in the cylinder block. So, the loads acting on the connecting rod is both compressive and tensile. The most common failure of connecting rod is these loads are at the ends section. The transition of the shank and the ends are most crucial in terms of failure of the connecting rod. The oil hole at the crank end is also subjected to the high stresses.

The major target of this project work is to explore a standard methodology for the designing of the connecting rod depending upon the inputs of the compressor based on its usage. This paper will further analyse the connecting rod depending upon the loading condition and will also validate the results.

II. ESTABLISHMENT OF GEOMETRY SOLID MODEL OF CONNECTING ROD

For the designing of the connecting rod basic empirical formulas will be used to calculate the basic parameters for the designing of the connecting rod. The specification of the compressor and other details for designing of the connecting rod is as follows.

Inputs for the Design:

- Diameter of the piston : 100 mm
- Length of rod : 200 mm
- Reciprocating mass : 1.5 kg
- Speed : 2000-2500 rpm
- Stroke : 60 mm
- Compression ratio : 4:1
- Max. Pressure : 3 MPa

Utilizing these input parameters the base dimension of the connecting rod is calculated. The most important sections for the designing of the connecting rod are: Shank, crank end, piston end and bolt.

Shank:

The connecting rod shank should be strong enough to sustain the tensile and compressive forces. Also it should be as light as possible to reduce the inertial forces and load onto the operating electric motor. An overview of the requirement suggests that I-section is best suited for the shank section.

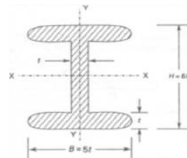


Figure 2: Details of Standard I-Section.

Figure 2 highlights the parameters of I-section proportions that are chosen, which are as follows:

Thickness of web: t, Width: 5t and Height: 6t

Maximum force acting on the connecting rod is given by,

$$\text{Force}_{\max} = \text{Pressure}_{\max} * \text{Area of cross-section}$$

International Journal of Innovative Research in Science, Engineering and Technology

(An ISO 3297: 2007 Certified Organization)

Vol. 5, Issue 6, June 2016

$$F_{\max} = P_{\max} * \pi * \left(\frac{D}{4}\right)^2$$

$$= 3 * 3.14 * (100 * 100) / 4 = 23550 \text{ N}$$

Web Thickness:

Rankine formula is utilised to calculate the crippling stress in the rod which is given by,

$$\sigma_{\text{crippling}} = \frac{F_{\text{crippling}}}{A} \left(1 + R \left(\frac{L}{k}\right)^2\right)$$

$$F_{\text{crippling}} = F_{\max} * \text{Factor of safety}$$

$$= 23550 * 2.5 = 58875 \text{ N}$$

A high factor of safety is considered to counter the fatigue loading on the part and high fluctuating nature of the forces. The factor of safety considered is 2.5

The yield strength of the material of the connecting rod i.e. high strength steel is $\sigma_{\text{crippling}} = 680 \text{ N/mm}^2$ and R, Rankine constant as $1/6250$ for connecting rod ends.

$$k = k_{xx}^2 = I_{xx} / A = 4.9t^2$$

where I_{xx} is Moment of Inertia about X-X axis and A= area of cross-section of I-section = $14t^2$

$$F_{\text{crippling}} = \frac{\sigma_{\text{crippling}} * A}{1 + R \left(\frac{L}{k}\right)^2}$$

$$58875 = \frac{680 * 14t^2}{1 + \frac{1}{6250} \left(\frac{200}{4.9t^2}\right)^2}$$

Solving this, $t = 2.495 \text{ mm} \sim 3.0 \text{ mm}$

$$\text{Thickness} = 3.0 \text{ mm}$$

So, the uniform web thickness of I-section is 3.0 mm.

$$\text{Width} = 5t = 5 * 3.0 = 15 \text{ mm}$$

$$\text{Height} = 6t = 6 * 3.0 = 18 \text{ mm}$$

Height of the rod will remain 18 mm at the centre while will range from 1.15-1.35 at crank end while 0.75-0.90 at the piston end. So, the height of I-section at crank pin will be 22 mm and at the piston pin will be 15 mm.

Piston Pin End:

Using the relation between the diameters of the piston pin as per design standards following relation is established. If D1 is the outer diameter of the piston pin end

$$\text{Outer diameter of the pin end, } D2 = (1.25 \text{ to } 1.65) * D1$$

$$\text{Inner diameter of the pin end, } D3 = (1.1 \text{ to } 1.25) * D1$$

$$\text{Length of pin end, } L1 = (1.3 \text{ to } 1.4) * D1$$

The maximum pressure which the bearing will stand is the limiting pressure of bearing and is given by,

$$P_{\text{bearing}} = \frac{F_{\text{gas}}}{L1 * D1^2}$$

Since the allowable bearing pressure for connecting rod is 35 N/mm^2 and $L1 = 1.35 * D1$

$$D1 = \sqrt{\frac{F_{\text{gas}}}{1.35 * P_{\text{bearing}}}}$$

$$D1 = \sqrt{\frac{23550}{1.35 * 35}} = 22.32 \text{ mm} = \text{say, } 22 \text{ mm}$$

$$L1 = 1.35 * 22 = 29.7 = \text{say, } 29 \text{ mm}$$

Other diameter will be,

$$D2 = 1.4 * 22 = 30.8 = \text{say, } 30 \text{ mm}$$

$$D3 = 1.1 * 22 = 24.2 = \text{say, } 24 \text{ mm}$$

Crank Pin End:

Considering the bearing failure criteria for the crank end the dimension of this end is given by,

International Journal of Innovative Research in Science, Engineering and Technology

(An ISO 3297: 2007 Certified Organization)

Vol. 5, Issue 6, June 2016

Crank pin diameter,

$$D4 = (0.6 \text{ to } 0.75) * \text{Diameter of piston}$$

$$D4 = 0.65 * 100 = 65 \text{ mm}$$

Length of the crank end,

$$L2 = (0.4 \text{ to } 1.0) * D4$$

$$L2 = 0.55 * 65 = 35.75 \text{ mm} = \text{say, } 35 \text{ mm}$$

Bolt Design:

The maximum pressure to which the bolts are subjected is due to the inertial forces at the beginning of suction at TDC

$$F_{inertia} = -mr\omega^2 (\cos \theta + (\cos 2\theta)/n)$$

Since, $\omega = \frac{2\pi\alpha}{60} = 2 * 3.14 * 2500/60 = 262 \text{ rad/s}$ # α : RPM of compressor

$$F_{inertia} = -1.5 * \frac{60}{1000} * 262 * 262 (\cos 0 + (\cos 0)/3.2) = 8108.6 \text{ N}$$

Since, there are 2 numbers of bolts, material's tensile strength 600 N/mm² and Factor of safety: 5

So, the maximum allowable pressure in bolt is 600/5 i.e. 120 N/mm²

Initial tightening force per bolt = 2.5 * (8108.6/2) = 10135.75 N

#Safety factor: 2.5

Force per bolt due to $F_{inertia} = (0.25 * 8108.6)/2 = 1013.58 \text{ N}$

#coefficient for the hard gasket: 0.25

Total force on bolt, $F = 10135.75 + 1013.58 = 11149.33 \text{ N}$

Area of cross section of bolt shank = $F/\text{Allowable Pressure}$
= $11149.33/120 = 92.91 \text{ mm}^2$

As per IS standard, M 14X1.5 mm is requirement for the design.

Output of the Design Calculation:

The summary of the finalised parameter for the designing of the connecting rod is as tabulated as follows:

Web thickness	3 mm	Diameter of crank end	65 mm
Width of flange	15 mm	Length of crank end	35 mm
Height of I section	18 mm	Number of bolts	2
Diameter of pin end	24 mm	Size of bolts	M14 X 1.5
Length of pin end	29 mm	Length of the rod	200 mm

Table1: Design parameters of Connecting Rod.

The Table 1 provides the finalised design parameter details of the connecting rod. These design parameters will now be used to prepare the CAD model of the connecting rod.

Designing of the Connecting Rod:

The design parameters shown above are used to design the CAD model of connecting rod in Ansys using design modular platform. Model is created by following standard designing procedure.



Figure 3: CAD Design of Connecting Rod.

Figure 3 shows the final CAD model of the connecting rod which is designed in the Ansys Design Modular.

International Journal of Innovative Research in Science, Engineering and Technology

(An ISO 3297: 2007 Certified Organization)

Vol. 5, Issue 6, June 2016

III. SIMULATION METHODOLOGY OF THE FE ANALYSIS OF CONNECTING ROD

Finite element analysis method is used to perform the analysis of the connecting rod. The requirement for the proper analysis is suitable model and boundary condition. In this analysis high strength steel connecting rod is analysed using Ansys. The loads and boundary condition is simulated as per the operation of the connecting rod. The static structural analysis of the part is done.

Material Property:

Properties	Values
Young's Modulus	210 GPa
Poisson's Ratio	0.3
Density	7890 kg/m ³
Yield Strength	680 MPa
Ultimate Tensile Strength	760 MPa

Table2: Material property of high strength steel

Table 2 provides the complete material property of the high strength steel which is used in the analysis of the connecting rod.

Meshing:

Meshing size is very important for the accurate results. Meshing of the part is done using 2 mm size. The tetrahedral element of second order is utilised for the modelling of the rod for better convergence of the results.

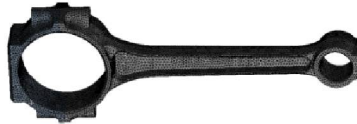


Figure 4: Meshed model of the connecting rod.

Figure 4 shows the final meshed model of the connecting rod. The number of elements and nodes are 100674 and 161263 respectively.

IV. LOADS AND BOUNDARY CONDITIONS

The maximum compressive and tensile force acting on connecting rod is done using force analysis. The net force acting on the rod is difference of forces due to gas and inertial forces.

As calculated earlier,

Maximum force on rod, $F_{gas} = 23550 \text{ N}$

Inertial force acting on the connecting rod, $F_{inertia} = 8108.6 \text{ N}$

$$\begin{aligned} \text{Net force acting on the connecting rod} &= (F_{gas} - F_{inertia}) \\ &= 23550 - 8108.6 \text{ N} = 15441.4 \text{ N} \end{aligned}$$

Considering the factor of safety,

$$\text{Axial force acting on the connecting rod} = 1.5 * 15441.4 = 23162.1 \text{ N}$$

So, for worst load case the maximum tensile and compressive forces on connecting rod is 23162 N.

For including all the cases and situation same axial force will be applied to simulate compressive and tensile condition. The 2 cases for each condition are done to evaluate all possible kind of loading situation. Hence a total of 4 conditions are dealt in the analysis.

International Journal of Innovative Research in Science, Engineering and Technology

(An ISO 3297: 2007 Certified Organization)

Vol. 5, Issue 6, June 2016

For Compressive Loading:

At both ends of the connecting rod a uniformly distributed load is applied at the contact area of 120 degrees in 2 cases for compressive loading.

The normal pressure which will be acting on this area is given by,

$$P_c = P_o$$

Where P_o is the normal pressure acting on the surface of the crank end and piston end.

Since the complete load is acting on a span of 120 degrees. Hence the effective compressive pressure acting will be,

$$P_c = \int_{-\frac{\pi}{3}}^{\frac{\pi}{3}} P_o * \cos\theta r l d\theta$$

$$P_c = \sqrt{3} r l P_o$$

So, the normal force that is acting in the ends is given by,

$$P_o = \frac{P_c}{\sqrt{3} r l}$$

Where l is the length and r is the radius of the corresponding end of the rod.

For piston end:

$$P_o = \frac{23162}{\sqrt{3} * 12 * 29} = 38.4 \text{ MPa}$$

For crank end:

$$P_o = \frac{23162}{\sqrt{3} * 32.5 * 32} = 11.76 \text{ MPa}$$

For Tensile Loading:

At both ends of the connecting rod a uniformly distributed load is applied at the contact area of 180 degrees in 2 cases for tensile loading.

The normal pressure which will be acting the area is given by,

$$P_t = P_o * \cos \theta$$

Where P_o is the normal pressure acting on the surface of the crank end and piston end.

Since the complete load is distributed to the area of 180 degree. Hence the effective tensile pressure acting will be,

$$P_t = \int_{-\frac{\pi}{2}}^{\frac{\pi}{2}} P_o * \cos^2\theta r l d\theta$$

$$P_t = \frac{\pi}{2} r l P_o$$

So, the normal force that is acting on the ends is given by,

$$P_o = \frac{P_t}{(\frac{\pi}{2} r l)}$$

Where l is the length and r is the radius of the corresponding end of the rod.

For piston end:

$$P_o = \frac{23162}{\frac{\pi}{2} * 12 * 29} = 42.39 \text{ MPa}$$

For crank end:

$$P_o = \frac{23162}{\frac{\pi}{2} * 32.5 * 32} = 12.97 \text{ MPa}$$

Summarising all the 4 load cases in tabular form:

Compressive Loading

Crank End	Piston End
Constrained	38.4 MPa
11.76 MPa	Constrained

(a)

Tensile Loading

Crank End	Piston End
Constrained	42.39 MPa
12.97 MPa	Constrained

(b)

Table 3: Load and boundary condition in (a) Compressive Loading and (b) Tensile Loading

Table 3 shows the summary of the loading and boundary condition of the connecting rod in tabular form. Table 3(a) shows the compressive loading condition while Table 3(b) shows the tensile loading condition.

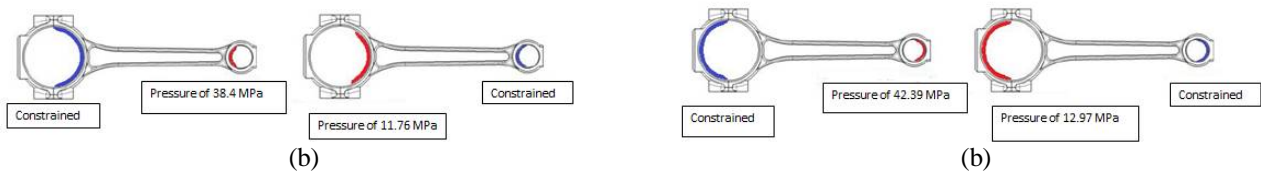


Figure 5: Load and boundary condition in (a) Compressive Loading and (b) Tensile Loading.

Figure 5 shows the summary of the loading and boundary condition of the connecting rod. Figure 5(a) shows the compressive loading condition while Figure 5(b) shows the tensile loading condition.

V. RESULTS: STRESS & DEFLECTION OBSERVATION

The static analysis of the connecting rod is done with above mentioned condition. The von-Mises stresses and deformation are taken as output in results. Since the material of connecting rod is ductile. Hence, the von-Mises stresses define the strength of component as per fracture criteria.

A. Compressive Loading:

In this analysis compressive loads are checked as per the table at both the ends of connecting rod. The maximum stresses are found at the interfacing location of shank and ends in both the cases.

Pressure at piston end:The maximum stresses and deflection observed in this case is 190 MPa and 0.12 mm.

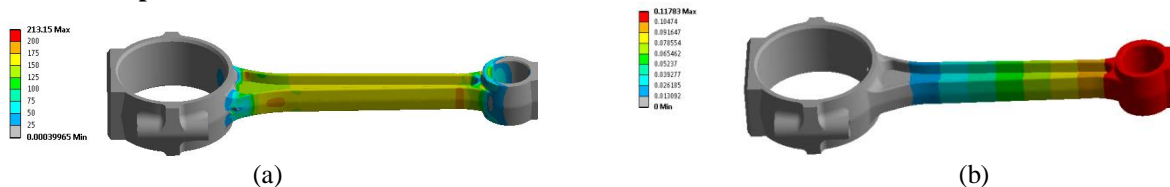


Figure 6:(a) Stress plot &(b) Deformation plot in Compressive loading at Piston end

Figure 6 shows the analysis results for compressive loading at the piston end. Figure 6(a) shows the stress plot while Figure 6(b) shows the deflection plot of the loading condition.

Pressure at crank end:The maximum stresses and deflection observed in this case is 380 MPa and 0.25 mm.

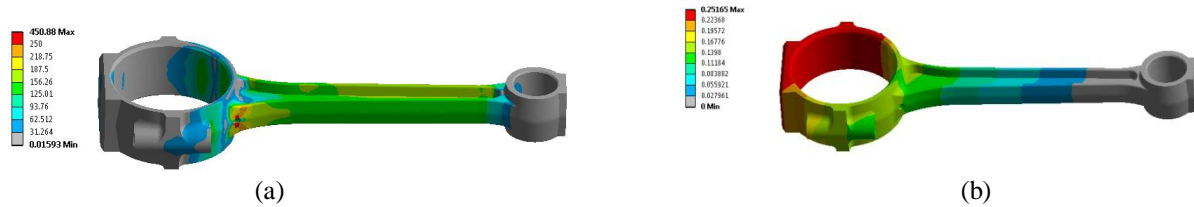


Figure 7: (a) Stress plot & (b) Deformation plot in Compressive loading at Crank end

Figure 7 shows the analysis results for compressive loading at the crank end. Figure 7(a) shows the stress plot while Figure 7(b) shows the deflection plot of the loading condition.

B. Tensile Loading:

In this analysis tensile loads are checked as per the table at both the ends of connecting rod. The maximum stresses are found at the interfacing location of shank and ends in both the cases.

Pressure at piston end: The maximum stresses and deflection observed in this case is 380 MPa and 0.24 mm.

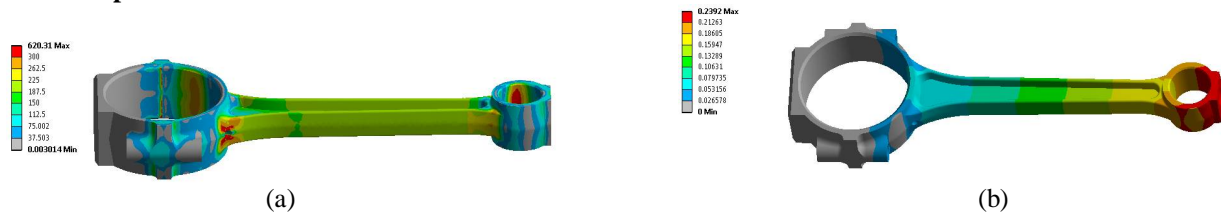


Figure 8: (a) Stress plot & (b) Deformation plot in Tensile loading at Piston end

Figure 8 shows the analysis results for tensile loading at the piston end. Figure 8(a) shows the stress plot while Figure 8(b) shows the deflection plot of the loading condition.

Pressure at crank end: The maximum stresses and deflection observed in this case is 620 MPa and 0.39 mm.

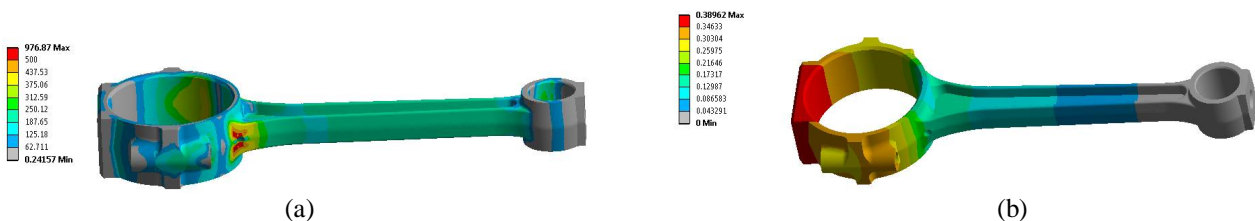


Figure 9: (a) Stress plot & (b) Deformation plot in Tensile loading at Crank end

Figure 9 shows the analysis results for tensile loading at the crank end. Figure 9(a) shows the stress plot while Figure 9(b) shows the deflection plot of the loading condition.

Observation: From the analysis results of the connecting rod it is concluded that the stresses in all the above cases are below the Yield strength of the connecting rod material. Thus the connecting rod is expected to withstand the loading during its operational cycle.

VI. VALIDATION OF THE FEA RESULTS

For the purpose of validation of the results obtained by analysis strain values at the mid connecting rod is compared by analysis results and that of strain obtained by analytical method using standard formulas.

International Journal of Innovative Research in Science, Engineering and Technology

(An ISO 3297: 2007 Certified Organization)

Vol. 5, Issue 6, June 2016

If the two results range up to the limit of 5% the analysis results will be considered acceptable and the above stated methodology for designing of the connecting rod will be acceptable.

Analytical Method:

Strain by theoretical method is given by:

$$\epsilon_{xx} = \left(\frac{F}{EA} \right)$$

where, F = Axial load on connecting rod

A = Area of cross-section

E = Young's modulus = 210 GPa

The axial force acting on rod is 23162 N and the area of cross section at middle of connecting rod shank is 133 mm².

Strain calculated by theoretical method is,

$$\epsilon_{xx} = \left(\frac{23162}{210000 * 133} \right) = 8.29 \times 10^{-4}$$

The equivalent strain obtained from analysis results is 8.48 X 10⁻⁴.

From the strain found by the two methods the variation between results is 2.3% and thus is considered acceptable.

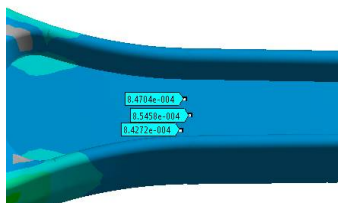


Figure 10: Strain values at the mid of connecting rod.

Figure 10 shows the equivalent strain plot of the connecting rod. The average strain obtained is 8.48 X 10⁻⁴. It is found that the variations between the two results are small and the value is below acceptable limit. This is good co-relation between the FE results and theoretical calculation. This validates our model and convergence of the solution.

VII. CONCLUSION

The results predict that the strain values are in agreement with the theoretical results which validate the methodology of the analysis and verifies model. The analysis results predict that the stresses and deflection are within acceptable limit. Thus the complete guideline for the designing of the connecting rod depending upon the compressor requirement is found to be satisfactory. The designed connecting is expected to sustain the loads to which it is subjected during its lifetime. Thus improves the performance of the compressor by resulting in the optimised connecting with lighter weight.

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