BUCKLING ANALYSIS OF CONNECTING ROD

Nagaraju K L[1], Chandan R[2]
[1]M-Tech Scholar, [2]Assistant Professor, Department of Mechanical Engineering
Dr. Ambedkar Institute of Technology Bangalore, Karnataka, India

Abstract: The connecting rod is the transitional part between the piston and the crankshaft. Its essential capacity is to transmit the push and pull from the piston pin to the crank pin, in this way changing over the reciprocating motion of the cylinder into turning movement of the crank. The capacity of connecting rod is to transmit the push of the piston to the crank shaft, and as the outcome the reciprocating motion of the piston is interpreted into rotational movement of the wrench shaft. It comprises of a pin–end, a shank section and a crank end. Pin end and crank end pin holes are machined to allow exact fitting of bearings. One end of the connecting rod is associated with the cylinder by the cylinder pin. Interfacing rods are subjected to forces generated by mass and fuel combustion. These two strengths result in axial load and bending stresses. An associating bar must be equipped for transmitting axial tension, axial compression, and bending stress brought about by the thrust and pull of the cylinder and by centrifugal force. In this thesis, a connecting rod is demonstrated utilizing catiav5, discretization utilizing HyperMesh and analysis utilizing Nastran. The outcome predicts the most extreme buckling load and basic locale on the interfacing pole. It is imperative to find the basic territory of concentrated stress for fitting adjustments. To discover the stresses created in interfacing pole under static loading with various stacking states of compression and tension at crank end and pin end of connecting rod.
Keywords: connecting rod; static analysis; buckling analysis; finite element analysis.

I. INTRODUCTION

In this paper, one segment of a motor specifically, the connecting rod, will be broke down. Being a champion amongst the most vital parts in an engine's diagram, the interfacing pole must have the ability to withstand immense loads and transmit a considerable measure of force. It is nothing sudden that a failure in an associating bar can be one of the costliest and hurting frustrations in an engine. Regardless, fundamentally saying that isn't adequate to totally fathom the elements of the situation.

The interfacing bar is the transitional part between0the cylinder and the0crankshaft. Its key limit is to0transmit the push0and draw from0the cylinder pin0to the wrench0pin, along these lines changing over the responding movement of the chamber into turning development of the wrench. The limit of associating bar is to transmit the push of the cylinder to the wrench shaft, and as the result the responding movement of the cylinder is deciphered into rotational development of the wrench shaft. It includes a pin0end, a0shank segment and a wrench end. Pin0end and wrench end0pin gaps are machined to0permit accurate0fitting of course. One end of the interfacing bar is connected with the chamber by the barrel pin. Interfacing bars are subjected to strengths produced by mass and fuel ignition. These two qualities result in pivotal load and bowing burdens. A partner bar must be prepared for transmitting pivotal pressure, hub pressure, and bowing anxiety achieved by the push and draw of the chamber and by radial power.

II. OBJECTIVES OF THIS PAPER

The objective of this study

Determine the total stress developed on the connecting rod due to the force applied on the connecting rod from the power stroke of a Spark-ignition engine. And also to study the effects of buckling load of the same magnitude as that of the power stroke applied on the connecting rod from the power stroke of a Spark-ignition engine.

Methodology

- Geometry of a connecting rod will be modeled using modeling software (CATIA v5).
- The Generated CAD Model is then discretized and a Finite Element Model will be created using a Pre-Processor (Hyper Mesh).
- The loads and boundary conditions that were calculated will be applied on the FE Model during Pre-Processing.
- The generated solver deck is fed to a solver (NASTRAN).
- Post-processing of the results obtained are performed with Post-processor (Hyper View).

Buckling analysis

There0are0two0major0categories0leading0to0the0sudden0failure0of0a0mechanicomponent:material0failure0and0structuralinstability,which0is0often0called0buckling.Formaterial0failureyou0need0to consider the0yield stress for ductile0materials and the ultimate stress for brittle0materials.Buckling is the instable phenomenon. The buckling happens only at one load, and that load is called as critical buckling load which is given by Euler formula Pcr = π2EI/ le2

Where,

E- Young’s modulus
I-Moment of Inertia
Le-Effective length of the column

When the maximum stress of the structure is less than ultimate strength and it is said to be safe, at that time critical buckling load becomes design driver i.e. if the applied load is greater than the critical buckling load, the structure will fail even though it is designed with respect to its strength.

III. FINITE ELEMENT ANALYSIS

The below figure shows the geometric model of a connecting rod used in a two-cylinder SI Engine. The modeling has been done using the modeling software, CATIA v5. It contains a piston end, shank and a crank end divided into two parts for easier installation. For model simplification, all fillets and chamfers below 3mm are neglected and the final model is represented in the above figure. The shank is an I-section of varying section with the smaller end at the piston end and the larger section at the crank end.

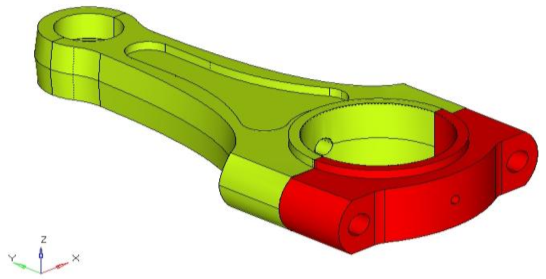

Figure 1: Geometric model of connecting rod used

The geometric model shown in the previous section is discretized using tetrahedral elements. (Figure 1) These elements are selected because of their ability to capture the geometry of any complex model. Since this model is of various shapes and sizes, a tetra element of the first order is selected. The crank was divided into two components in the geometric model and the same has been maintained in the FE model also.

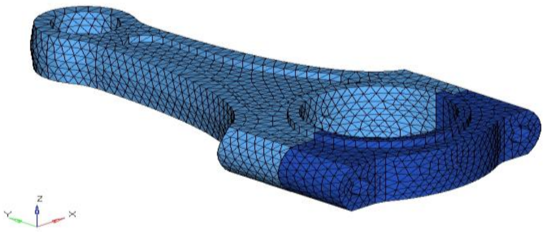

Figure 1: Finite Element model of connecting rod

Table 1 gives the material properties applied to the model. The material is forged steel and its properties are given below. The material properties required for the current analysis are Young's, Modulus, Poisson's ratio, density and ultimate strength. This is because the analysis is of structural nature and other properties are not considered in the analysis.

Table 1 – Material Properties

Material Name: Forged steel

Property	Unit	Value
Young's Modulus	GPa	206.7
Poisson's Ratio	-	0.3
Density	Kg/m3	7820
Ultimate strength	MPa	827

Table 2: Elements and nodes count

Table 2 gives the number of nodes and elements generated during the discretization. The total mass of thee model is seen to be around 2.2 kg which is close to the actual value.

Nodes	3739
Elements	12839

A. LOADS AND BOUNDARY CONDITIONS

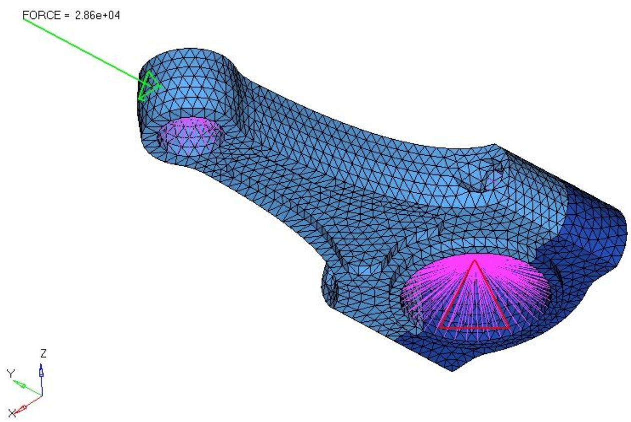

Figure 3: Finite Element model of connecting rod with Loads and Boundary conditions.

Figure 3 shows the discretized model with loads and boundary conditions applied. For this analysis, the connecting rod is placed at the top dead center during the commencement of the power stroke. Hence, the crank is constrained in all degrees of freedom and the load is applied on the piston end of the connecting rod. The calculations relating to the load applied is shown below.

B. ENGINE SPECIFICATION AND CALCULATIONS

Engine Capacity : 2967 cc
Power developed : 232 hp (173 kW)
Torque developed : 300 Nm at 4500 rpm
Stroke : 79.5 mm
Bore : 89 mm
Compression ratio : 10:1

The Forces acting on the piston end of the connecting rod is given by the equation,

Force = Gas force – Inertial force ……..………………… (1)

Where,

Gas Force = Pressure X Cylinder bore area …………...... (2)

Inertial Force = m_p X a_p……………….…………………...(3)

$$a_p = r\omega^2 \left[\cos\theta + \cos\frac{2\theta}{n}\right] \quad (4)$$

Where,

r = Crank radius
ω = angular velocity
θ = Crank angle

(For maximum power, it can be assumed that the crank angle will be 0° or 360°)

The mean effective pressure inside the cylinder can be calculated by using the below formula.

$$\text{Power (P)} = \frac{P_m LAN}{60} \quad (5)$$

Where,

P_m = Mean effective pressure (MPa)
L = Stroke length (m)
A = Area of the cylinder (m^2)
N = Speed of the engine (RPM)

$$\text{Angular velocity} = \frac{\pi DN}{60}$$

Where,

D = Diameter of the crankshaft (m)

N = Speed of the engine (RPM)

Or

$$P_m = \frac{173\ X10^3 X\ 60}{0.0795\ X\ 0.006221\ X\ 4500}$$

= 4663988.916 N/m^2

= 4.6 MPa = 46 bar

$$\text{Gas Force} = \frac{4.6\ X\pi X\ (89X10^{-3})^2 X10^6}{4}$$

= 28.621kN

$$a_p = 39.75X10^{-3}X18.7317^2$$

= 13.95 m/s^2

Inertia force = 2.507 X 13.95

= 35N

Forces acting on the connecting rod small end = Gas force – Inertial force

= 28621 - 35

= 28586N

IV. RESULTS AND DISCUSSION

The analysis was run for the above mentioned FE Model with the loads and boundary conditions attached to it. The following figures give the results obtained from both the static analysis (Displacement and Stresses) and buckling analysis (Buckling mode).

A. STATIC AND BUCKLING ANALYSIS RESULTS

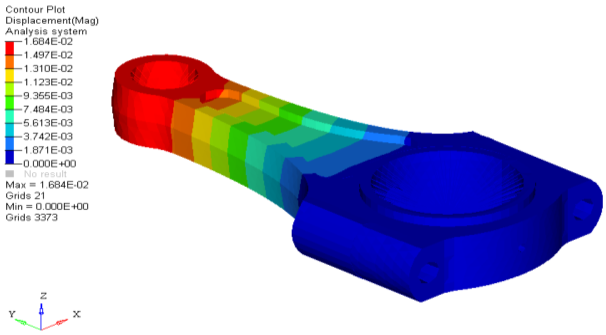

Figure 4: Displacement plot

Figure 4 above shows the displacement for the selected connecting rod under the loads applied. It is clear from the above figure that the maximum displacement occurs on the piston end at a value of 0.176 mm and minimum displacement occurs on the crank end at a value of 0 mm.

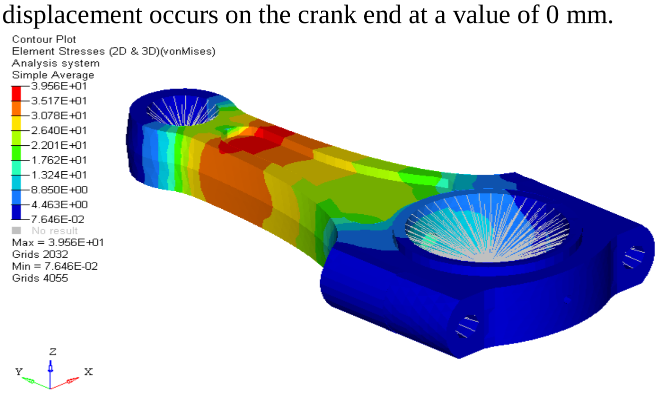

Figure 5: Elemental stress plot

Figure 5 above shows the Von Mises stress plot for the selected connecting rod under the loads applied. It is clear from the above figure that the shank is subjected to a maximum stress near the piston end at a value of 414.34 MPa. This value is very much less than the yield strength of the material and hence it can0be said that the0connecting rod is safe under the applied0load because the value of the stress developed is well within the maximum allowable stress level of 468.85 MPa. This value is obtained by considering a factor of safety of 2 for forged steel which has the ultimate strength of 937.7 MPa.

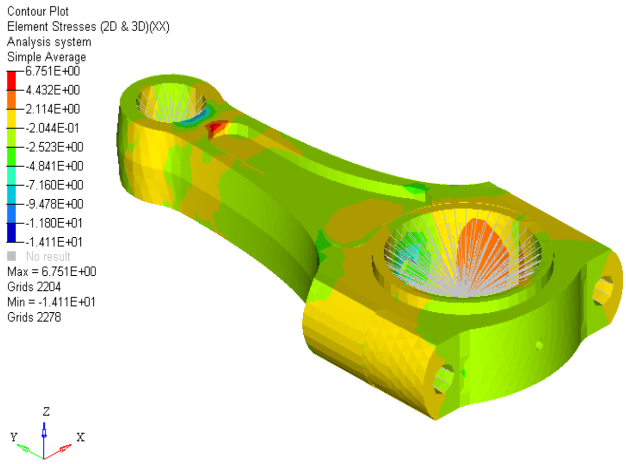

Figure 6: Elemental stress plot in X-direction

Figure 6 shows a stress plot in the X-direction of the connecting rod for the applied loads. It can be seen that a very small region on the connecting rod shows the development of stress until a value of 70.71 MPa and the rest of the connecting rod falls under the range of -2.14 MPa and 22.14 MPa

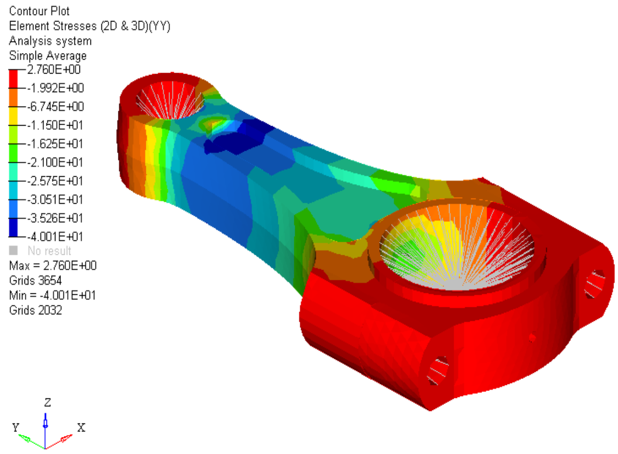

Fig 7: Elemental stress plot in Y-direction

Figure 7 shows a stress plot in the Y-direction of the connecting rod for the applied loads. It can be seen that piston end and the crank end of the connecting rod shows the development of stress to a maximum value of 28.91 MPa and the rest of the connecting rod falls under the range of -70.65 MPa and -419.1 MPa. The negative sign indicates the developed of stress in the negative Y-direction.

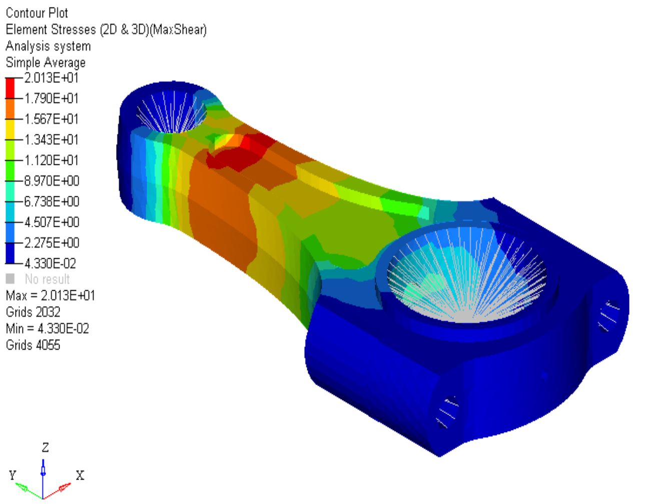

Figure 8: Maximum Shear stress plot

Figure 8 shows the shear stress plot for the connecting rod under the applied loads. It is seen that a maximum stress value of 210.8 MPa is obtained on the shank of the connecting rod since it experiences two opposite forces.

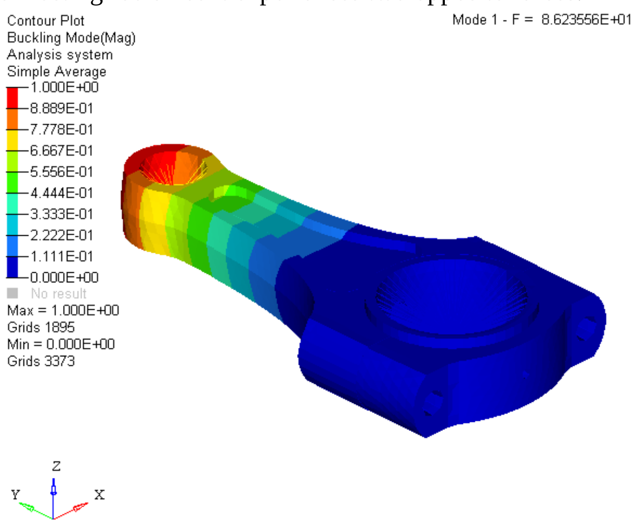

Figure 9: Buckling Mode

Figure 9 shows the buckling mode plot of the connecting rod for the applied loads. It is observed that the buckling factor is 86.23 which means that the connecting rod designed can withstand loads as high as 86 times the load applied.

V. CONCLUSION

The model of a connecting rod was generated, discretizes and analyzed. The results are tabulated and presented in the previous chapters.

From the results it can be concluded that,

- The displacement plot shows a very small value which does not affect the performance of the connecting0rod.
- The0linear static analysis of the0connecting rod0shows that the stress generated in the model is within the acceptable limits or maximum allowable stress.
- The stress plots in X, Y and Z directions show that the maximum stress is developed along the axis of the0connecting rod which is due to0the load being0applied along the axis.
- The shear stress plot shows that the maximum shear stress is developed0in the I-section of the connecting rod which is also within the acceptable limit.
- The buckling mode analysis gives the buckling factor greater than 1 and hence it0can be0concluded that the0connecting rod can withstand the load applied.

REFERENCES

[1] Webster W, D. Coffell R. and Alfaro D "A Three Dimensional Finite Element Analysis of a High Speed Diesel Engine Connecting Rod".

[2] Sarihan V and Song J "Optimization of the Wrist Pin End of an Automobile Engine Connecting Rod with an Interference Fit", Journal of Mechanical Design, Transactions of the ASME.

[3] Ali Fatemi and Mehrdad Zoroufi "Fatigue Performance Evaluation of Forged Versus Competing Process Technologies: A Comparative Study".

[4] M. SC. Anna Ulatowska "Shape Optimization of the Connecting Rod".

[5] Shenoy P.S and Fatemi A Connecting rod optimization for weight and cost reduction.

[6] Yang R.J Dewhirst D.L Allison, J.E and Lee A Shape optimization of connecting rod pin end using a generic model. Finite Elements in Analysis and Design.

[7] Pai C. L The shape optimization of a connecting rod withfatigue life constraint. Int. J of Materials and Product Technology.

[8] Athavale, S. and Sajanpawar, P. R "Studies on Some Modelling Aspects in the Finite Element Analysis of Small Gasoline Engine Components" .

[9] Rahman M.M Ariffin, A.K, Abdullah S, Noor, M.M and Bakar R.A Assessment of cylinder block for two stroke free piston linear engine using random loading.

[10] Sonsino C. M. and Esper F. J "Fatigue Design for PM Components," European Powder Metallurgy Association (EPMA).

