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Fatigue Failure Analysis of Connecting Rod of Different Cross-Sections

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Abstract — *Connecting rod is the link between piston and crankshaft which is acted upon by axial forces and moments at its ends, and converts the reciprocate motion to rotary motion. Yield, fatigue and buckling phenomenon are considered for the design and performance of connecting rod. The Finite Element Analysis is capable to give the results like stress, fatigue life, strian, thermal analysis and deformations, throughout of the component subjected to complex loading. In the present work, the fatigue life analysis is carried out by ANSYS using stress and strain life theories for different cross-sections connecting rod. The equivalent Von-Mises stresses, Principal Stress and Shear Stress are calculated for (+) section, (H) section and (I) section. Finally it concluded that I section connecting rod is best rest of two in all aspects.*

Keywords*- ANSYS, Fatigue, Design, Finite element method, SWT, Von-Mises Stress*

I. INTRODUCTION

Connecting rods are widely used in variety of engines such as, oppose-piston engines, V-engines, opposed-cylinder engines, radial engines and In-line engines to transmit the thrust of the piston to the crankshaft, and results into conversion of the reciprocating motion of piston to the rotational motion of crankshaft. It consists of a pin-end, a shank section, and a crank-end. A connecting rod works in variably complicated conditions, and is subjected to not only the pressure due to the connecting rod mechanism, but also due to the inertia forces. Its behavior is affected by the fatigue phenomenon due to the reversible cyclic loadings. When the repetitive stresses are developed in the connecting rod it leads to fatigue phenomenon which can cause dangerous ruptures and damage. Bending stresses appear due to eccentricities, crankshaft, case wall deformation, and rotational mass force; therefore, a connecting rod must be capable of transmitting axial tension/compression and bending stresses caused by the thrust and pull on the piston and by the centrifugal force [1]. A primary design criterion for the connecting rod is endurance limit. The cyclic material properties are used to calculate the elastic-plastic stress-strain response and the rate at which fatigue damage accumulate due to each fatigue cycle [4]. Imahashi et al. [5] discuss the factors which affect the fatigue strength in powder forged (PF) connecting rod, i.e., hardness of the material, depth of decarburized layer, metallurgical structure, density, and surface roughness. Olaniran et al. [4] investigated a new crack able alloy of forged steel (FS) for connecting rod application. There are two practical buckling modes of connecting rod. One mode called ‘side buckling’ occurs in the direction parallel to the rotational axis of the connecting rod. The other mode called ‘front-rear buckling’ occurs in the direction perpendicular to side buckling. [7]. Webster et al., [8] discussed the loading criteria of connecting rod used in an IC engine. For tension loading the crank end and piston ends are found to have a sinusoidal distribution on the contact surface with pins and connecting rod.

II. FATIGUE FAILURE ANALYSIS

Fatigue is a phenomenon associated with variable loading or more precisely to cyclic stressing or straining of a material. Just as the human beings get fatigue when a specific task is repeatedly performed, in a similar manner metallic components subjected to variable loading get fatigue, which leads to their premature failure under specific conditions. Fatigue loading is primarily the type of loading which causes cyclic variations in the applied stress or strain on a component. Thus any variable loading is basically a fatigue loading. Variable loading is occurred when the applied load or the induced stresses in a component is not constant but changes with time, i.e., load or stress varies with time following certain pattern. The load application, particularly the distribution at the contact area, factors that decide load distribution, the calculation of the pressure constants depending on the magnitude of the resultant force and application of the restraints. The flowchart of the finite element fatigue analysis is shown in Fig.1. A study by Sugita et al. [9] used boundary element method to reduce the weight of the connecting rod. The connecting rod is designed by incorporating a thin I section column and adopting the two-rib design to the big end.

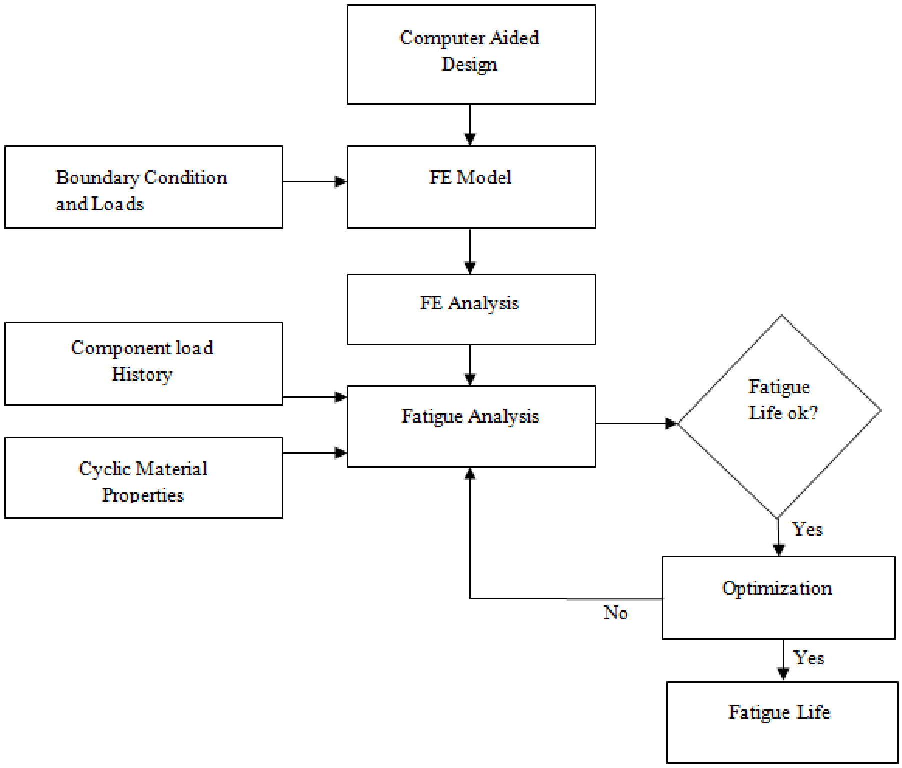


Figure 1: The flow chart of finite element based fatigue analysis [10]

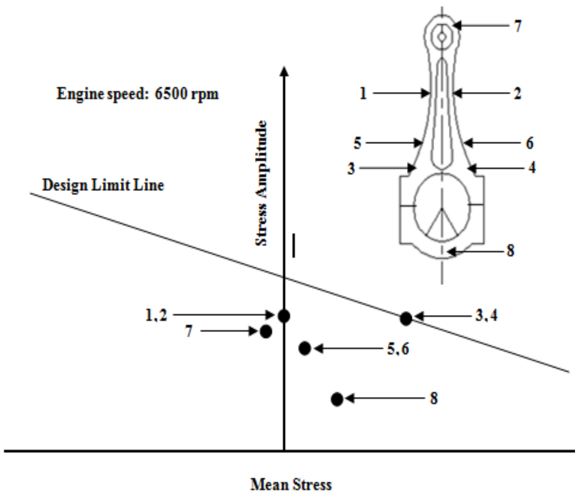


Figure 2: The comparison of FE calculated And strain gage measured stresses [9]

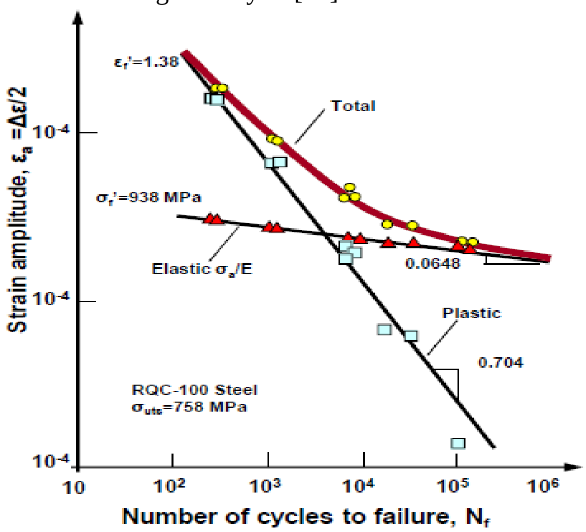


Figure 3: Strain-life curve [11]

The fatigue resistance of metals can be characterized by a strain-life curve as shown in Figure 3. Coffin [12] and Manson [13] established a mathematical relationship between the total strain amplitude, and the reversals to failure cycles as,

$$\frac{\Delta\varepsilon}{2} = \frac{\sigma_f'}{E}\left(2N_f\right)^b + \varepsilon_f'\left(2N_f\right)^c$$

Morrow [14] established a relationship between the mean stress, and fatigue life as,

$$\varepsilon_a = \frac{\sigma_f' - \sigma_{mean}}{E}\left(2N_f\right)^b + \varepsilon_f'\left(2N_f\right)^c$$

Smith *et al.* [15] established another relationship, Smith-Watson-Topper (SWT) mean stress correction model, expressed as,

$$\sigma_{max}\varepsilon_a E = \left(\sigma_f'\right)^2\left(2N_f\right)^{2b} + \sigma_f'\varepsilon_f' E\left(2N_f\right)^{b+c}$$

III. LOADING ON CONNECTING ROD

The various forces acting on the connecting rod are as follows: Force on the piston due to gas pressure and inertia of the reciprocating parts, and Force due to inertia of the connecting rod or inertia bending forces, For all practical purposes, the force in the connecting rod F_C is taken equal to the maximum force on the piston due to pressure of gas F_G [6],

$$\boxed{F_C = (m_{piston} + m_{con-rod}).r\omega^2.(\cos\theta + \lambda\cos 2\theta) + F_G}$$

Force due to in-cylinder gas,

$$\boxed{F_G = p_{\max} \times A}$$

$\lambda = {}^{r}\!/\!{}_{L}$, Where r is crank radius and L is connecting rod length. Θ is crank angle and ω is angular velocity.

The connecting rod is subject to inertial bending forces (rod whip) as it swings through TDC. The gas force is determined by the speed of rotation, the masses of the piston, gudgeon pin and oscillating part of the connecting rod consisting of the small end and the shank. Axial loading on rod is due to gas pressure and rotational mass forces. Bending moments originate due to eccentricities, crankshaft, case wall deformation, and rotational mass force, which can be determined only by strain analyses in engine [4]. Loading on piston and then to connecting rod in an engine is the function of crank angle. It varies up to a maximum value from suction pressure and again lowers to exhaust pressure. The pressure in cylinder is the maximum at TDC. In this work, the maximum pressure $\boldsymbol{p_{max}}$ is obtained from engine configuration and is taken 7 times to mean effective pressure of the given engine [2]. The mean effective pressure is obtained by numerical analysis for given data of considered engine. The mean effective pressure is given as,

$$\boxed{Power = \frac{mep \times A \times L \times K \times N}{z \times 60000}}$$

Where, A is cross-section area, K is number of cylinders, N is rpm and $z = 2$

The mean effective pressure is calculated for the given engine configuration is 6.75 bar, thus the force on connecting rod is taken approximately as 40 kN for the analysis.

Table 1: Engine Specifications

Bore, *mm*	108
Stroke, *mm*	130
Connecting rod length *L*, *mm*	250
Swept Volume, *Litre*	4.8
No. of cylinders *K*	4
Engine speed *N*, *RPM*	1800
Compression ratio	17:1
Power, *kW*	48.47
Torque, *N-m*	289.1 @ 1250 RPM

Table 2: Design Parameters

Design Parameters	Value, mm
Piston pin end diameter	39
Crank pin end diameter	80
Effective length	250
Shank Length	155.4
Fillet radius at crank end	48.5
Total width	37.8

For the given (Table 1) engine configuration, a typical connecting rod is taken as per the drawing [3]. The rod is open at crank end and the total length from pin to pin is 250 mm as in Fig. 4. The design parameters corresponding to given drawing are tabulated in Table 2.

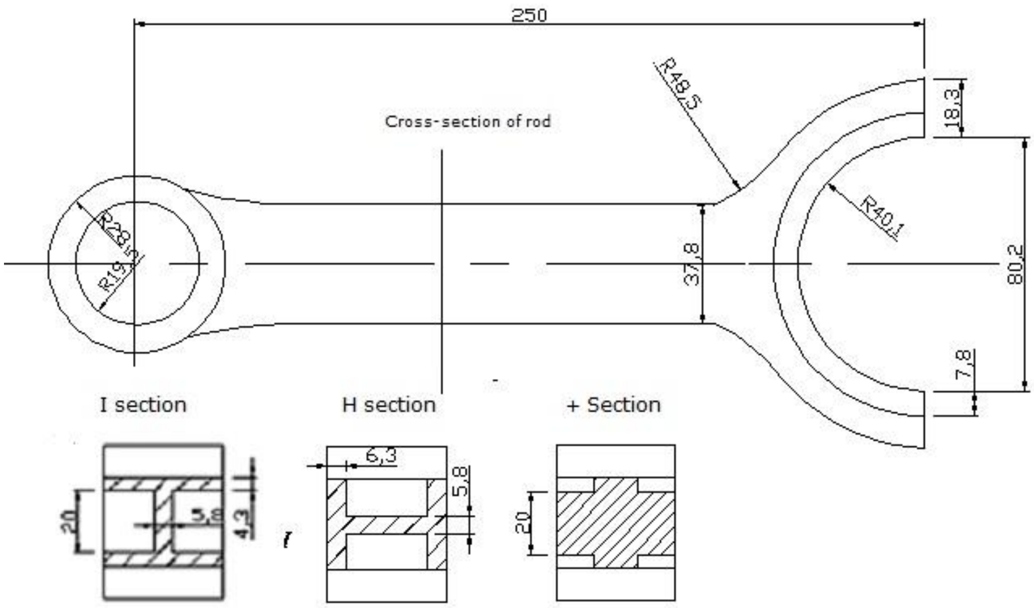


Figure 4: Drwaing of Connecting Rod

IV. FINITE ELEMENT MESH MODEL

The 3-D solid model to the particular drawing is introduced to ANSYS software. The solid modelling is done on higher end CAD software as per design parameters given in Table 2 for all section. The meshing of rod is shown in Fig. 5 through the 10–node tetragonal elements of 2 mm length. The reason for selecting elements is to make the geometrical parts of a complicated mechanical component to gain more authentic results based on the high techniques of stress calculation. The number of nodes and elements for each section is tabulated below.

Table 3: Number of Elements of each section

	Elements	Nodes	Mass of Rod, *kg*
+ Section	28827	50669	2.225
H Section	32278	57218	2.019
I Section	31490	55077	2.128

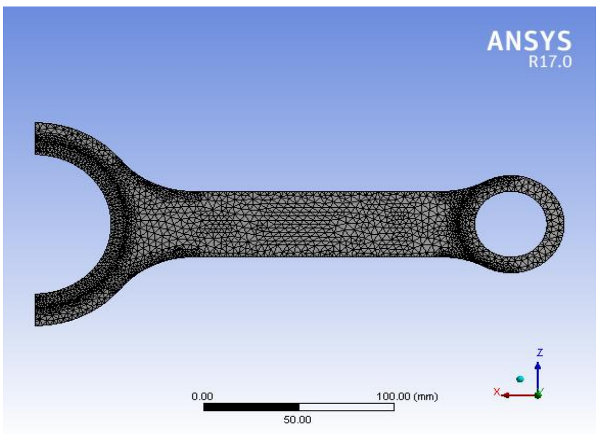


Figure 5: Mesh Model

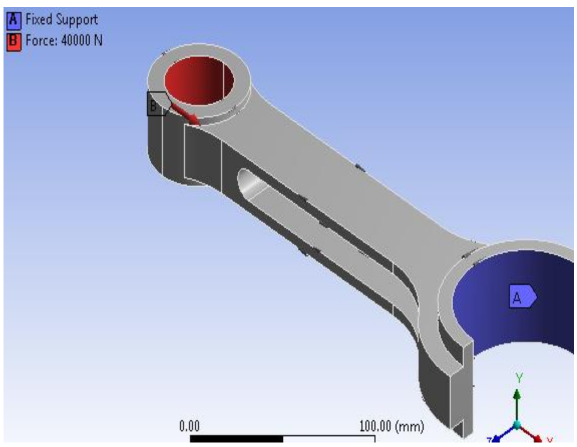


Figure 6: End Constraints

Boundary Condition and Loading

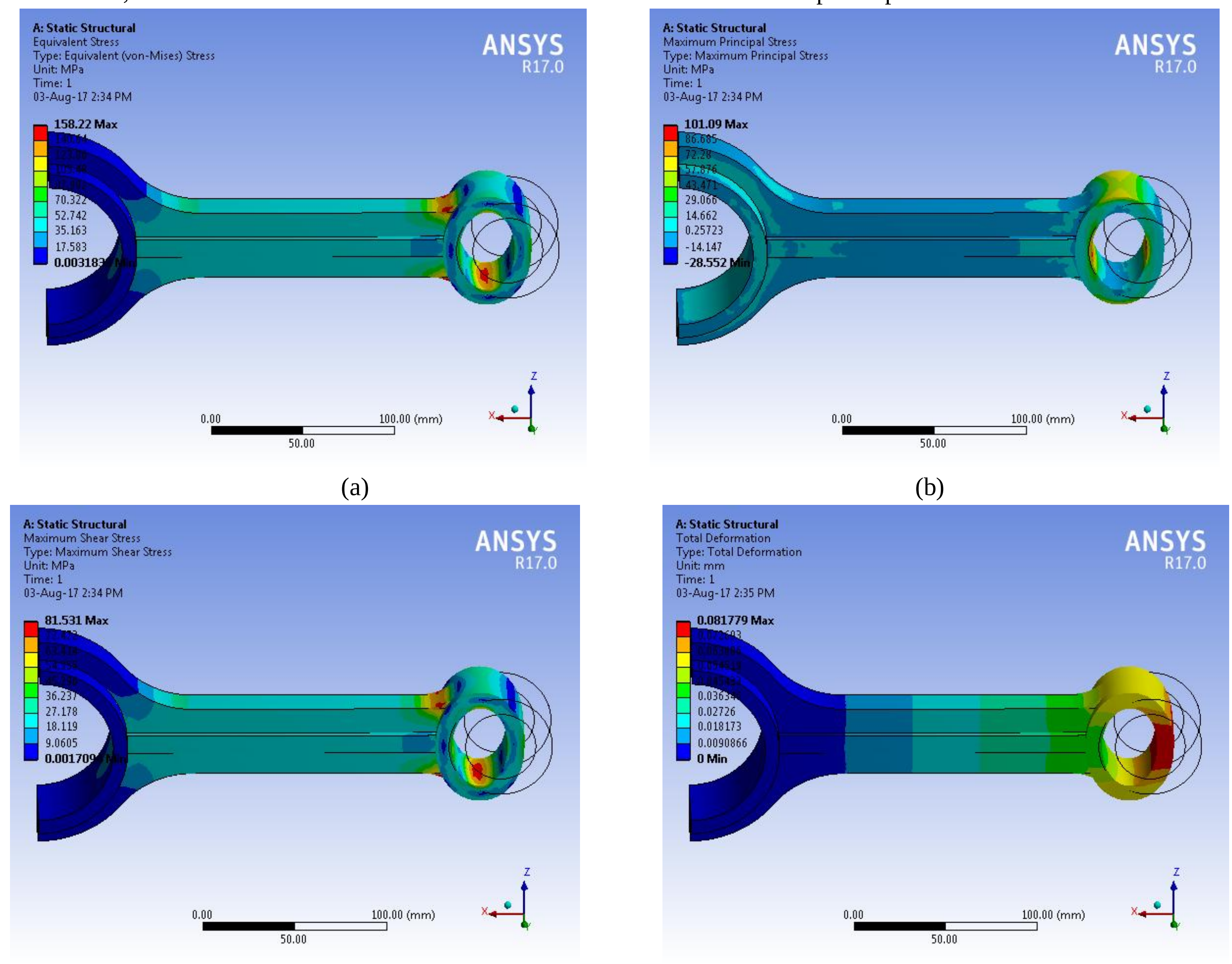
The stress and strain is the most serious when the explosive pressure of the fuel gas achieves maximum when the piston is at TDC. The maximum pressure is calculate using numerical calculation for the given engine parameters. The force is max when piston is at TDC and same force is transferred to the connecting rod. The displacement restrictions for the rod are restrained to let the rod in a static condition. The crank end face is kept fixed for the analysis. The crank pin end is kept fixed and load is applied on piston pin end towards stroke line of the connecting rod axis as shown in Fig. 6

Table 4: Mechanical properties for connecting rod materials [11]

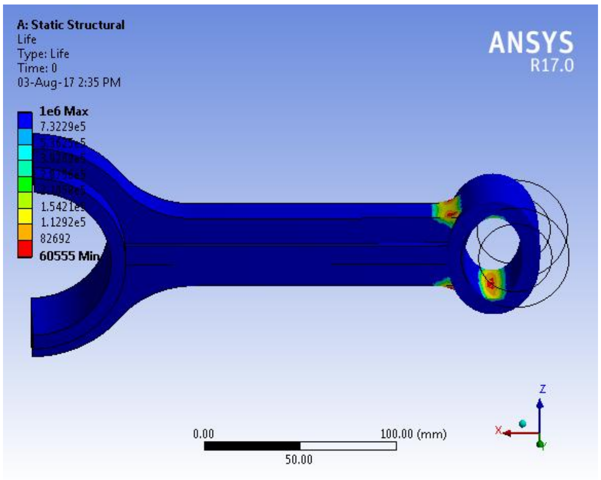
Monotonic Properties	Forged Steel (FS)	Fatigue Properties	Forged Steel (FS)
Young's Modulus, GPa	201	Fatigue Strength Coefficient σ_f', MPa	1188
Yield Strength, MPa	700	Fatigue Strength Exponent b	-0.0711
Ultimate Tensile Strength, MPa	938	Fatigue Ductility Coefficient ε_f'	0.3576
Poisson's Ratio	0.30	Fatigue Ductility Exponent c	-0.5663
Strain Hardening Exponent	0.122	Cyclic Strength Coefficient K', MPa	1397
Density, kg/m^3	7806	Cyclic Strain Hardening Exponent n'	0.1308

V. RESULTS AND DISCUSSION

Finite element meshes are generated using ten-node tetrahedral elements with various elemental lengths from 5 mm to 2 mm. The material properties are given in Table 4 for which the analysis is carried out in ANSYS. The Von-Mises stresses, when stresses generated exceed the allowable limit, are checked for the convergence at critical locations within rod. The results show that the convergence has been achieved for the entire range of elemental length and constant values of stresses generated if the size of elements 2 mm is considered. The FEA results are shown in given Fig. 7 for Von-Mises stress, strain and deformations. The maximum stress location is found on piston pin end side.



(a) (b) (c) (d)



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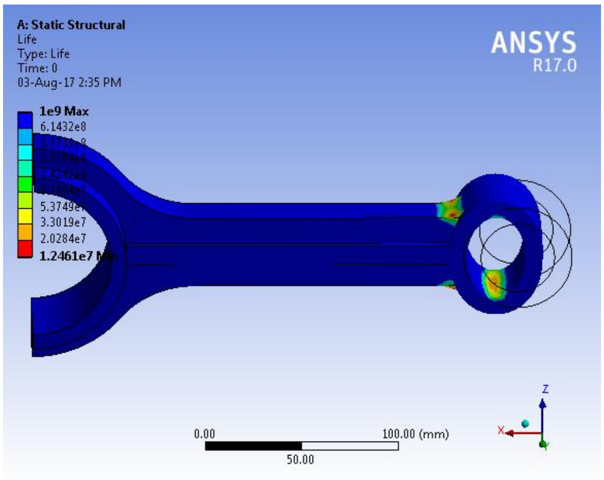
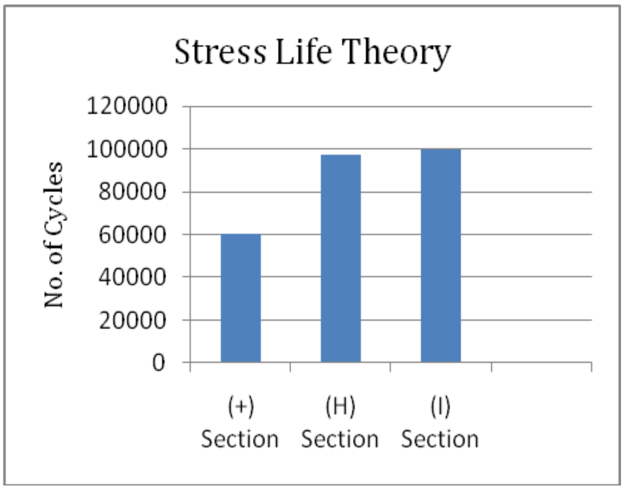


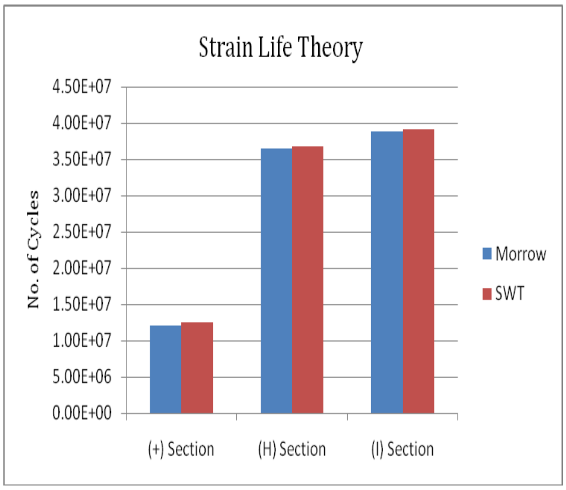
(f)

Figure 7: (a) Von-Mises Stress, (b) Max Principal Stress, (c) Max Shear Stress, (d) Deformation, (e) Goodman Criteria Life and (f) strain Life by SWT

The stress life cycles and strain life cycles are given in figure for each section for comparison. The stress life is more for H section and strain life is more for I section in SWT criteria.



(a)



(b)

Figure 8: (a) stress life (b) Strain Life

Table 5: Strain Life Cycles

Strain Life Theory, Min Cycle to Fail the Component in Fully Reverse Loading		
	Morrow	SWT
(+) Section	1.20E+07	1.25E+07
(H) Section	3.64E+07	3.68E+07
(I) Section	3.88E+07	3.91E+07

The I section show the maximum life to be fail in fully reverse loading in Morrow and SWT criteria. The minimum life in number of cycles is given in Table 5.

Table 6: Max Stresses for Different Section Connecting Rod

	Von-Mises Stress, MPa	Max Principal Stress, MPa	Max Shear Stress, MPa	Total Deformation, mm
(+) Section	158.22	101.09	81.53	0.08177
(H) Section	139.07	95	75.13	0.09425
(I) Section	138.08	94.73	74.982	0.08801

The stress value is least in I section rod, the deformation is least in + section rod. The values of stresses are given in Table 6 for each section and a comparison is shown in Fig 9. Stress generated by the load is minimum in I section 138 MPa, 139 MPa for H section and 158 MPa for + section while deformation is least in + section 0.08177 mm.

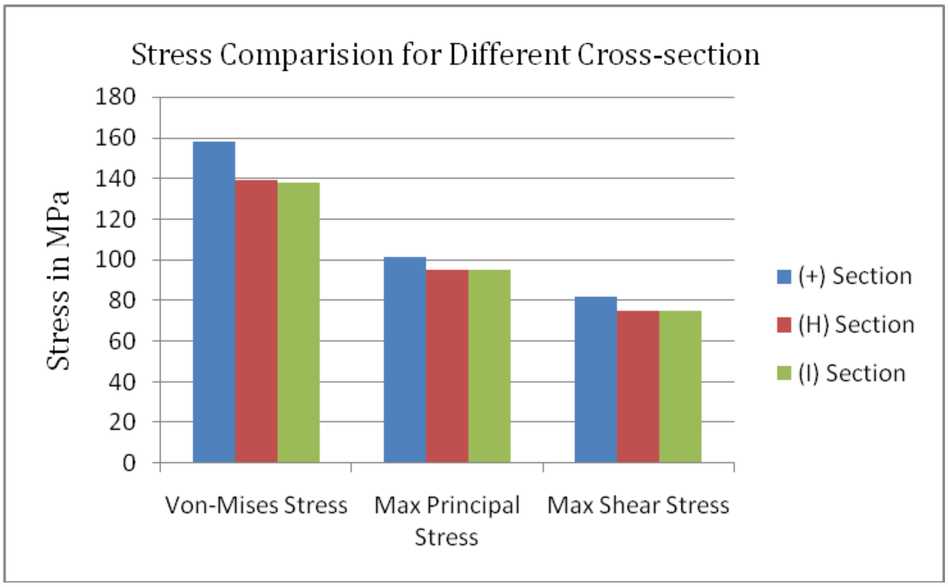


Figure 9: Max Stress Comparision

CONCLUSION

In this paper, the Fatigue failure analysis is studied, and found that the strain life theories play a role in designing the component reasonably. The life of I section connecting rod is more than + section and H section rod. The Equivalent Von-Mises, Principal Stress, Shear Stress are least for I section rod. Hence I section connecting rod is good for the design and give the satisfactory performance in operation. The critical location of stress is near the piston pin end side, and then goes to decreasing towards the big end side. For optimization, the fillet radius and diameter of small and width of rod play a role in design.

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