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Design, Buckling and Fatigue Failure Analysis of Connecting Rod: A Review

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***Abstract*— *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. Yield, fatigue and buckling characteristics are often used as evaluation indexes for the performance of engine connecting rods in mass reduction design to optimize vibration. Various rod cross-section like I section, + section, Rectangular section, Circular section and H section have important role in design and application. In this paper the design methodology is covered and FEA results for stresses have been presented and strain life theories studied.*

Keywords*— *Buckling, Connecting rod Shank, Design, Fatigue, Finite element method, 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 as shown in Fig. 1. Pin-end and crank-end pin holes are machined to permit accurate fitting of bearings. One end of the connecting rod is connected to the piston with the help of a piston pin. The other end revolves with the crankshaft and is split to permit it to be clamped around the crankshaft. Connecting rods are subjected to forces generated by mass and fuel combustion. These two forces results in axial and bending stresses. 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]. The connecting rods of the automobile are mostly made of cast iron through the forging or powder metallurgy. The main reason for applying these methods is to produce the components integrally and to reach high productivity with the lowest cost [2] and optimized shape [3].

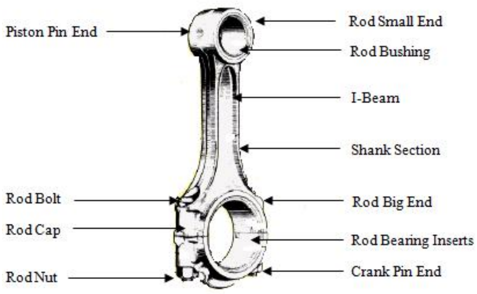

Fig.1: Schematic of a typical connecting rod

II. CONNECTING ROD MATERIALS

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. The material properties for connecting rod material are given in Table 1 [6].

Table.1: Mechanical Properties for connecting rod materials

Monotonic Properties	**Forged Steel (FS)**	**Powder Metal (PM)**	**C-70 Alloy Steel**
Young's Modulus (*E*), GPa	201	199	212
Yield Strength, MPa	700	588	574
Ultimate Tensile Strength, MPa	938	866	966
Strength Coefficient (*K*),	1400	1379	1763

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MPa			
Strain Hardening Exponent (n)	0.122	0.152	0.193
Density, kg/m^3	7.806	7.850	7.700
Poisson's Ratio	0.30	0.29	0.30
Fatigue Properties			
Fatigue Strength Coefficient (σ_f'), MPa	1188	1493	1303
Fatigue Strength Exponent (b)	-0.0711	-0.1032	-0.0928
Fatigue Ductility Coefficient (ε_f')	0.3576	0.1978	0.5646
Fatigue Ductility Exponent (c)	-0.5663	-0.5304	-0.5861
Cyclic Strength Coefficient (K'), MPa	1397	2005	1739
Cyclic Strain Hardening Exponent (n')	0.1308	0.1917	0.1919

III. FORCES ON ROD AND DESIGN

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_p),

$$F_C = \frac{F_P}{\cos \phi}$$

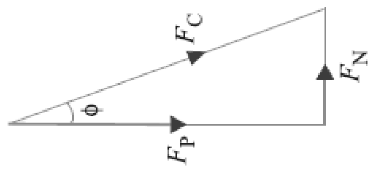

Fig.2: Force in connecting rod

In designing a connecting rod, the following dimensions are required to be determined [7]: Dimensions of cross-section of the connecting rod, Dimensions of the crankpin at the big end and the piston pin at the small end, Size of bolts for securing the big end cap, and Thickness of the big end cap. A connecting rod is which is subjected to alternating direct compressive and tensile forces. Since the compressive forces are much higher than the tensile forces, therefore, the cross-section of the connecting rod is designed as a strut. Hence the design should be according to buckling phenomenon. As shown in Fig. 3, 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.[8]

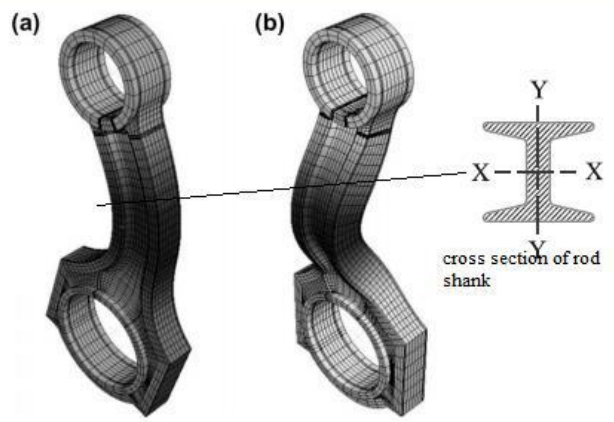

Fig.3: Buckling modes of the connecting rod: (a) side buckling and (b) front-rear buckling.

Rod may buckle with *X*-axis as neutral axis (*i.e.* in the plane of motion of the connecting rod) or *Y*-axis as neutral axis (*i.e.* in the plane perpendicular to the plane of motion). The connecting rod is considered like both ends hinged for buckling about *X*-axis and both ends fixed for buckling about *Y*-axis.

$$\sigma_{cr}^{e}\big|_x = \frac{\pi^2 E}{(K_x L/r_x)^2} \qquad \sigma_{cr}^{e}\big|_y = \frac{\pi^2 E}{(K_y L/r_y)^2}$$

Where σ_{cr}^{e} is elastic critical buckling stress (Euler formula), E is the elastic modulus, L is effective length, r is radius of gyration for each axis, K_x is 0.5 for a fixed–fixed joint and K_y is the unity for a pined–pined joint. For I section rod $I_{xx} = 4\, I_{yy}$ is quite satisfactory.

A connecting rod in a high-performance engine, compressor, or pump is a critical component: if it fails, catastrophe follows. Yet to minimize inertial forces and bearing loads it must weigh as little as possible, implying the use of light, strong materials, stressed near their limits. To design a connecting rod of minimum mass with two constraints: that it must carry a peak load F without failing either by fatigue or by buckling elastically.

The mass of rod shank

$$m = \beta A L \rho$$

Where L is the length of the con-rod, ρ the density of the material of which it is made, A the cross-section of the shaft, and β a constant multiplier to allow for the mass of the bearing housings. The con-rod, to be safe, must meet both constraints. For a given length, L, the active

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constraint is the one leading to the largest value of the mass, ***m*** out of m_1 and m_2

$$m_1 = \beta FL\left(\frac{\rho}{\sigma_e}\right) \qquad m_2 = \beta\left(\frac{12F}{\alpha\pi^2}\right)^{1/2} L^2\left(\frac{\rho}{E^{1/2}}\right)$$

where α is a dimensionless''shape-constant'' and σ_e is endurance limit.

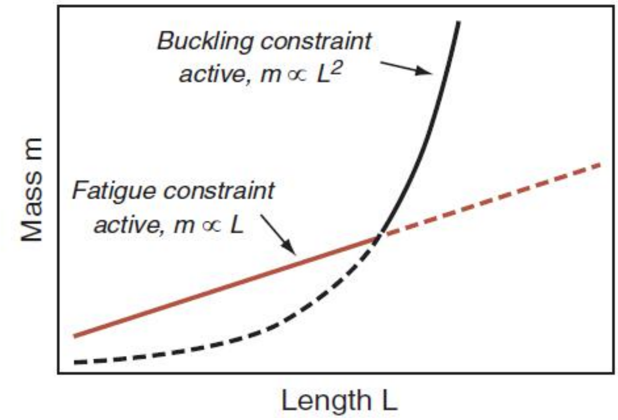

Fig.4: Mass of the rod as a function of L [9]

IV. STRESSES IN CONNECTING ROD

. The connecting rod should be designed with high reliability. It must be capable of transmitting axial tension, axial compression, and bending stresses caused by the thrust and pull on the piston, and by centrifugal force without bending or twisting. An explanation of the axial forces acting on connecting rod is provided by Tilbury [2]. The connecting rods are subjected to mass and gas forces due to the fuel combustion resulting into axial and bending stresses [3]. 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. Bending moments originate due to eccentricities, crankshaft, case wall deformation, and rotational mass force, which can be determined only by strain analyses in engine [10]. Fig. 5 shows axial loading due to gas pressure and rotational mass forces.

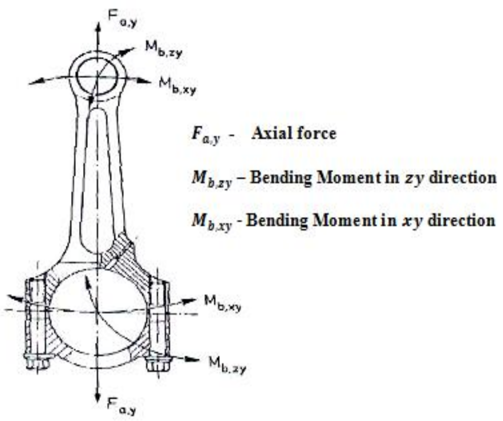

Fig.5: The origin of stresses on a connecting rod [10]

Sugita et al. [11], discussed the static analysis, quasi-dynamic analysis and design of a lightweight CR. Fig. 6 shows the boundary conditions used for static finite element analysis under tensile load. Fig. 7 shows compressions of the maximum principal stress values obtained at the critical locations based on FEA and strain gauge measurements under static loading.

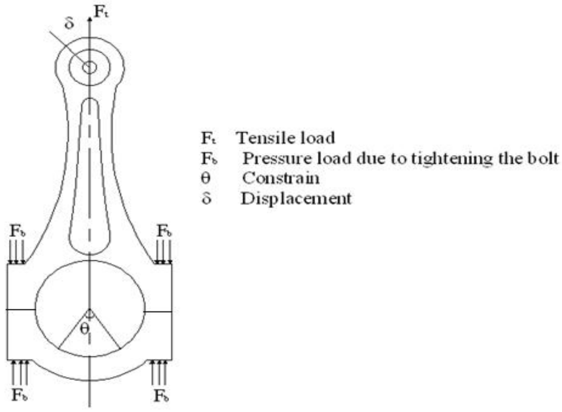

Fig.6: Boundary conditions for static FE analysis [11]

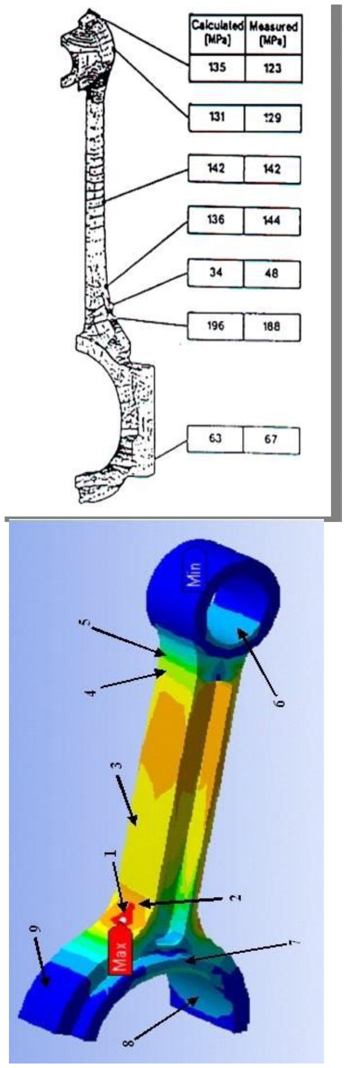

Fig.7: Comparisons between FEA and strain gage measured values

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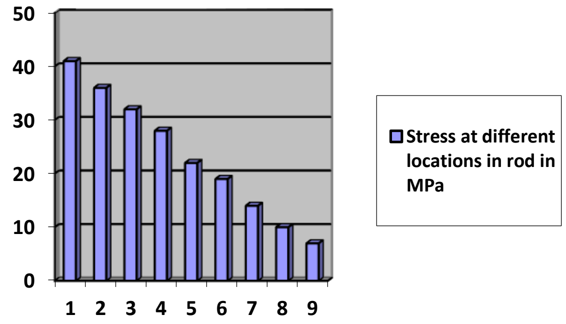


Webster et al., [12] discuss the loading criteria of connecting rod used in an IC engine. For tension loading the crank end and piston ends are found to have a sinusoidal distribution on the contact surface with pins and connecting rod. Fig. 8 shows the load distribution in tension and compression.

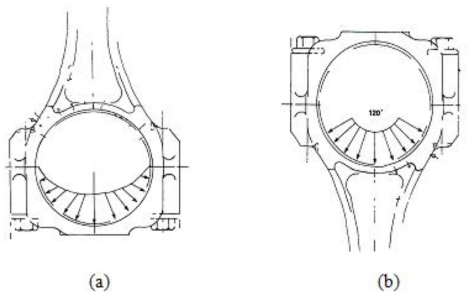


Fig.8: (a) Distribution of tension loading and (b) distribution of compressive loading in the connecting rod [12]

A. Tevatia**,** et al. discussed the maximum stress calculations in different cross sections connecting rod by FEM method for different materials, stresses were lower in I section rod and for powder metal Fig 9 [13].

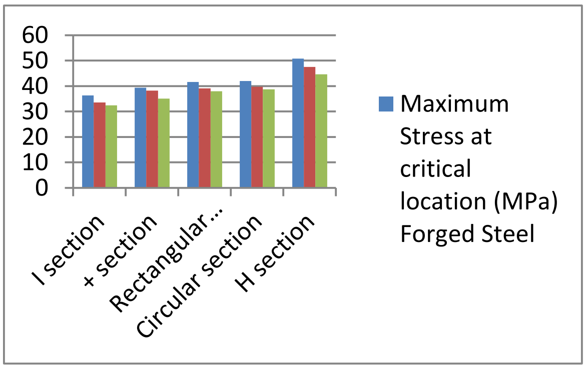


Fig.9: Comparision of max stress for different cross-section connecting rod

V. FATIGUE FAILURE IN CONNECTING ROD

Fatigue is the behaviour of materials under fluctuating and reversing loads. The various FE tools are used for analyzing the fatigue behaviour of connecting rod by the various researchers. Beretta et al. [14] investigated fatigue performance of the connecting rods made of either cast iron or hot forging carbon steel. They state that if a CR working in a car engine is subjected to bench test loading conditions, the different areas of the CR are subjected to peculiar load spectra with different stress ratios. A study by Sugita et al. [11] 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.

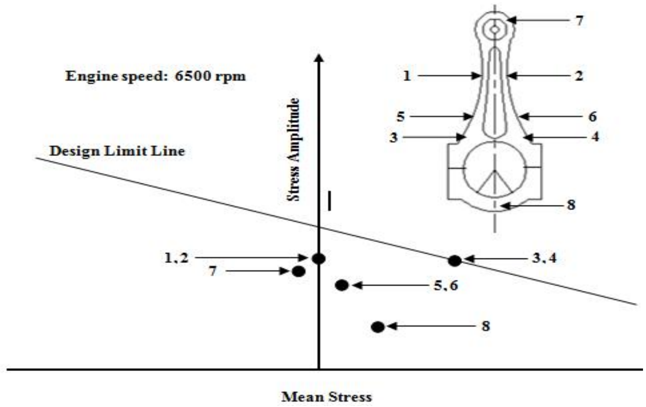
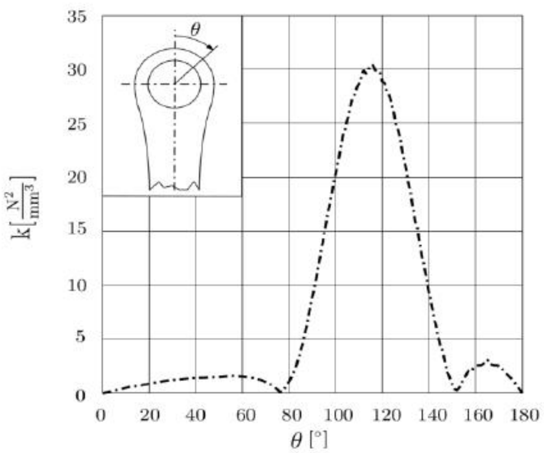


Fig.10: The comparison of FE calculated and strain gage measured stresses [11]

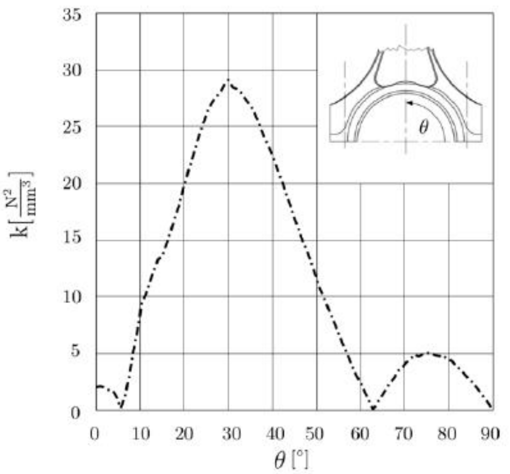
Antonio Strozzi et al. discussed about fretting fatigue in con-rod small end and big end with reference to the titanium con-rod by Rutz parameter *k* [15]

$$k = \sigma_c \cdot \Delta \cdot p \cdot f$$

where σ_c is circumferential stress, Δ is the relative tangential displacement amplitude displacement, *p* is the pressure distribution between the con-rod small end and the bush and, *f* is the friction coefficient assumed to be equal to 0.1



(a)



(b)

Fig.11: Fretting fatigue distribution in (a) small end , (b) big end [16]

The fatigue resistance of metals can be characterized by a strain-life curve as shown in Figure 12. Coffin [17] and Manson [18] established a mathematical relationship between the total strain amplitude, and the reversals to failure cycles as,

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

Morrow [19] established a relationship between the mean stress, and fatigue life as,

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

Smith *et al.* [20] established another relationship, Smith-Watson-Topper (SWT) mean stress correction model, expressed as,

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

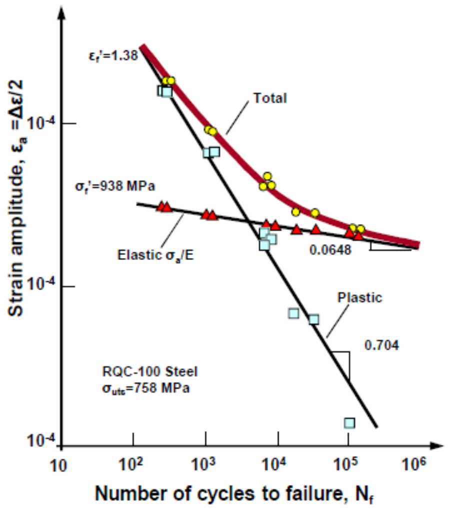


Fig.12: Strain-life curve [21]

Figure 15 shows Fatigue life at critical location for different materials and cross-sections of Connecting rod using strain life theories [22]

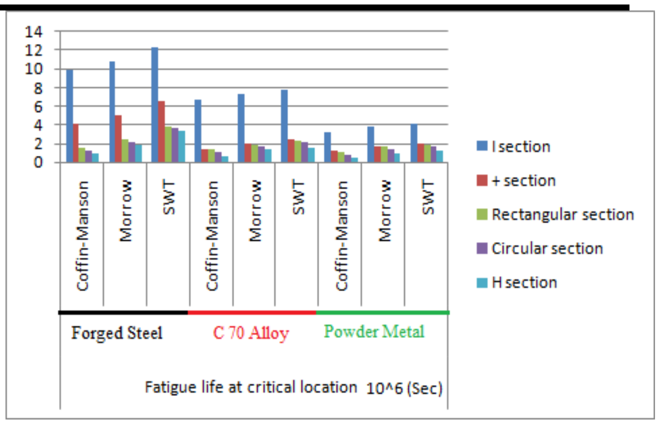


Fig.15: Comparison of fatigue life

VI. CONCLUSION

In the paper, the literature background regarding connecting rod material and their application in different kind of engines has been taken to consideration. The material properties play a vital role to design the rod for a particular application and its durability is studied. The forces in the crankshaft during a cycle and maximum load in a cycle/stroke defines the design of rod based on buckling criteria and optimum mass of rod to avoid the natural vibrations to a lower level are discussed. The stress calculation using FEM is studied for different kind of rod cross-section is viewed and concluded that the Von Mises stress is minimum for a I cross-section of rod. The fatigue failure of piston pin end and crank pin end has been studied under the pressure variation at locations of ends. It is presented that fatigue life is more in I section rod for forged steel rod under various strain life theories.

REFERENCES

[1] Schreier, L., 1999, "Tension and Compression in Connecting Rods", http://em-ntserver.unl.edu, date: 12-02-2011.

[2] Tilbury, R.J., 1982, "The Prediction and Measurement of Axial Forces, Bending Moments and Accelerations in an Engine Connecting Rod", Journal of Strain, Vol. 18, No. 2, pp. 55- 59.

[3] Nortan, R., 1994, "Design of Machinery," McGraw Hill, pp. 583

[4] Olaniran, M.A. and Stickels, C.A., 1993, "Seperation of Forged Steel Connecting Rods and Caps by Fracture Splitting", SAE Technical Paper 930033, pp. 1-6.

[5] Imahashi, K., Tsumuki, C., and Nagare, I., 1984, "Development of Powder-Forged Connecting Rods", SAE Technical Paper 841221, pp. 1-7.

[6] Afzal, A., and Fatemi, A., 2003, "A Comparative Study of Fatigue Behavior and Life Predictions of Forged Steel and PM Connecting Rods", SAE Int.

[7] Khurmi, R. S. and Gupta, J. K., 2005, "A Text Book of Machine Design", S. Chand & Company Ltd.

[8] Moon Kyu Lee et al, "Buckling sensitivity of a connecting rod to the shank sectional area reduction", doi:10.1016/j.matdes.2010.01.010

[9] Michael F. Ashby, "Materials Selection in Mechanical Design" 3rd edition, ISBN 075066168 2

[10] Sonsino, C.M., 1990, "Fatigue Design of Sintered Connecting Rods", Metal Powder Report, Vol. 45, No. 6, pp. 408-412.

[11] Sugita, J., Itoh, T., and Abe, T., 1990, "Engine Component Design System Using Boundary Element Method," SAE Technical Paper 905206

[12] Webster, W.D., Coffell, R., and Alfaro, D., 1983, "A Three Dimensional Finite Element Analysis of A High Speed Diesel Engine Connecting Rod," SAE Technical Paper 831322, pp. 83-96.

[13] Finite Element Fatigue Analysis of Connecting Rods of Different Cross-Sections, A. Tevatia, S.B. Lal and S.K. Srivastava, International Journal of Mechanics and Solids, ISSN: 0973-1881, pp. 45-53, 2011

[14] B eretta, S., Blarasin, A., Endo, M., Giunti. T., and Murakami, Y., 1997, "Defect Tolerant Design of Automotive Components", International Journal of Fatigue, Vol. 19, No. 4, pp. 319-333.

[15] Merritt D, Zhu G. "The prediction of connecting rod fretting and fretting initiated fatigue fracture" No. 2004-01-3015. SAE Technical Paper 2004

[16] Antonio Strozzi et al., "A repertoire of failures in connecting rods for internal combustion engines, and indications on traditional and advanced design", Article in Engineering Failure Analysis • November 2015 DOI: 10.1016/j.engfailanal.2015.11.034

[17] Coffin, L.F., 1954, "A Study Of The Effects of Cyclic Thermal Stresses on A Ductile Metal", Transactions of American Society for Testing and Materials, 76, pp. 931-950

[18] Manson, S.S., 1953, "Behavior of Materials Under Conditions of Thermal Stress", Heat Transfer Symposium, pp. 9-75

[19] Morrow, J., 1968, "Fatigue Design Handbook-Advances in Engineering" Warendale, PA, SAE, pp. 21-29K. Elissa, "Title of paper if known," unpublished.

[20] Smith, K.N., Watson, P. and Topper, T.H., 1970, "A Stress-Strain Functions for the Fatigue on Materials", J. of Materials. 5(4), pp. 767-78

[21] Bishop, N., and Sherratt, F., 2000, "Finite Element Based Fatigue Calculations", The Int. Association for the Engg. Analysis Community Netherlands, NAFEMS Ltd.

[22] Fatigue Life Prediction of Connecting Rod using Strain Life Theories, A. Tevatia and S.K. Srivastava, Global Journal of Engineering.