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Fatigue Analysis of Connecting Rod of Samand Engine by Finite Element Method

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Abstract: In this study, detailed load analysis was performed for connecting rod of samand engine after that a Finite Element routine was used to calculate stresses under the maximum compression and tension loadings in the connecting rod which were then used for critical points evaluation. Fatigue analysis and longevity assessed through using of ANSYS software. Calculations based on fatigue life and accurate loading histories permit rod to be optimized for durability without the need for expensive and time-consuming testing of series of prototypes. The results obtained from the present study can be used to bring about modifications in the process of connecting rod manufacturing.

Key words: Samand, engine, connecting rod, stress and fatigue analysis.

INTRODUCTION

Despite the fact that most engineers and designers are aware of fatigue and that a vast amount of experimental data has been generated on the fatigue properties of various metallic and non-metallic materials, fatigue failures of engineering components are still common. A number of factors influence the fatigue life of a component in service, viz., (i) complex stress cycles, (ii) engineering design, (iii) manufacturing and inspection, (iv) service conditions and environment and (v) material of construction. The use of calculations and simulations is a key feature of the modern design process. Several properties such as stress, strength, stiffness, durability, handling, ride comfort and crash resistance can today be numerically analyzed with varying levels of accuracy. Development time can be shortened by ensuring that some, or rather all, of these properties fulfill established requirements even before the first prototype is being built. Accordingly, calculations based on fatigue life and accurate loading histories permit structures and components to be optimized for durability without the need for expensive and time-consuming testing of series of prototypes. Thus designs can be obtained that are less conservative (i.e., better optimized) than those based on traditional criteria, such as maximum load or stress for a series of standard load cases (Fermer, M. and H. Svensson, 2001). The use of Finite Element Method (FEM) for calculating stress and strain is a well established procedure in analyzing fatigue and determining longevity. For example, (Del Llano-Vizcaya, *et al*., 2006) carried out stress analysis in the FE code ANSYS and then performed multiaxial fatigue study of helical compression springs using the fatigue software nCode (Del Llano-Vizcaya, L., *et al.*, 2006). (Biancolini, *et al*., 2003) designed a connecting rod based on fatigue analysis (Biancolini, M.E., *et al*., 2003). (Beretta, *et al*., 1997) presented a resistant method to failure on connecting rod design that improved the fatigue life slightly (Beretta, S., *et al.,* 1997). They found that the occurrence of fatigue phenomenon is closely related to the appearance of cycling stresses within the connecting rod body. (Lu, 1996) presented an approach to optimize the shape of a connecting rod subjected to a load cycle which consisted of the inertia load deducted from gas load as one extreme and peak inertia load exerted by the piston assembly mass as the other extreme (Lu, P.C., 1996). A FE routine was first used for calculating the displacements and stresses in the connecting rod, which were then used in another routine to calculating the total life. Fatigue life was defined as the sum of crack initiation and crack growth lives, with crack growth life obtained using fracture mechanics. (Rahman, *et al*., 2008) presented the finite element based fatigue life prediction of a new free piston linear generator engine mounting (Rahman, M.M., *et al.,* 2008). The objective was to assess the critical fatigue locations on the component due to loading conditions. They concluded that Morrow mean stress correction method gave the most conservative (less life) results for crack initiation method. (Nanaware and Pable, 2003) described a case study on the fatigue fracture of rear axle shafts of 575 DI tractors (Nanaware, G.K. and M.J. Pable, 2003). The failure of rear axle shafts was due to inadequate spline root radius, which led to crack initiation and subsequent crack growth is by fatigue under the cyclic loading conditions of field operation. In general, the shafts in power plant systems run with a steady torsion combined with cyclic bending stress due to self-weight or weights of components or possible misalignment between journal bearings (Bhaumik, S.K., 2002). A similar case study was reported in Fatigue Design Handbook AE 10 (Nanaware, G.K. and M.J. Pable, 2003). This case study was of scraper type tractor rear axle shaft failures. The rear axle shafts were failing within six months of service, even though durability tests were done in the laboratory. Fatigue was the predominate mode of failure due to reverse torque.

In this study, fatigue analysis and longevity of the Samand engine connecting rod is carried out in the FE code ANSYS. Samand is one of the numerous vehicles in Iran. Also this vehicle is national vehicle of Iran.

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Method:

Fatigue phenomenon is a complicated subject which seems to be not known a lot. The best theory for the explanation of fatigue phenomenon proposal, is the strain-life theory which is used for the fatigue strength estimation. But for the application of this theory there must be some assumptions made for the ideal state, so it results in some uncertainties.

Rupture due to the fatigue is usually occurred in discontinuities or where we have the stress concentration. When in these places the existing stress, exceeds the allowable one it gives rise to the plastic strain. For the ruptures resulted from the fatigue, there must be some plastic cyclic strains. So, It was needed to seek for the component behavior during the cyclic deformations. Monsoonkoffin suggested the Equ.1 to present the relationship between fatigue life and the total strain (Shigley, J.E. and Mischke, C.R., 2001).

$$\frac{\Delta\varepsilon}{2} = \frac{\sigma_F}{E}(2N)^b + \varepsilon_F (2N)^C \tag{1}$$

Where Δε is the total stress, N is the fatigue longevity, E is the Young's modulus, b and c are the exponents of fatigue strength and fatigue elasticity, and finally σ_F and ε_F are the coefficient of fatigue strength and elasticity respectively.

For fatigue analysis by ANSYS software first different forces exerted on connecting rod should be calculated after that stress analysis should be done and critical nod for tensile and pressure stress should be known finally fatigue analysis should be done for critical nodes.

In this project MSC.ADAMS/Engine software was used to load analysis of slider-crank mechanism. For obtaining to this purpose crank mechanism was simulated in ADAMS/Engine software. Figure 1 shows dynamic model of Samand engine in ADAMS/Engine software.

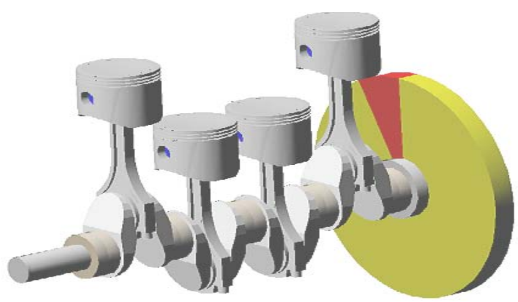

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Fig. 1: Dynamic Model of Engine in ADAMS/Engine.

Combustion chamber pressure curve was measured in Iran Khodro's power test lab. These experimental data have been shown in Figure 2. Data of these curves were exerted on piston in modeled mechanism.

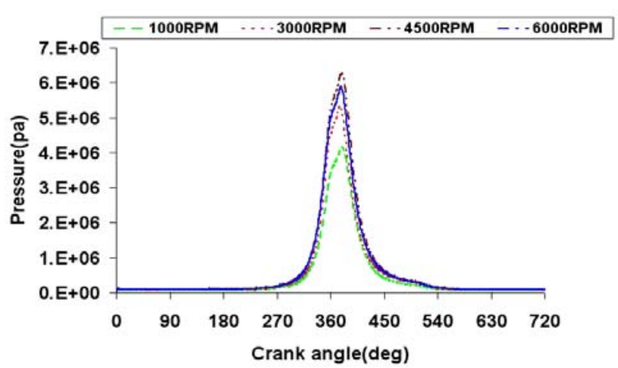

Fig. 2: Combustion Pressure in Different RPM and Load.

For stress analysis of connecting rod it was modeled and meshed in ANSYS (Ver.9) software. Solid92 element was considered to carry analyzing. Figures 3 shows modeled and meshed connecting rod in ANSYS software. Material qualification of C70S6 steel (used for this connecting rod) has been shown in table 1 (Anonymous, 2008).

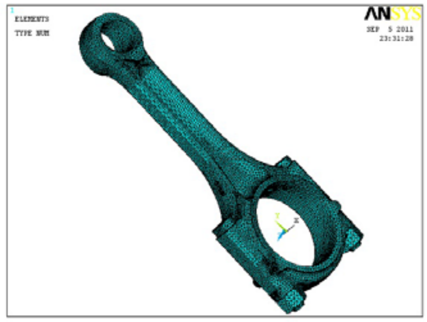

Fig. 3: Modeled and Meshed connecting rod in ANSYS ver9 software.

Table 1: Material qualification of C70S6 steel .

C (wt%)	Si (wt%)	Mn (wt%)	P (wt%)	S (wt%)	0.2%PS (N/mm^2)	UTS (N/mm^2)	EI %	R/A %
0.68/0.75	0.15/0.35	0.50/0.60	0.045	0.060/0.070	580	1000	12	15

Results:

Load analysis was done for different rotational speeds. These speeds were 1000 RPM (slow rotational speed), 3000 RPM (middle rotational speed), 4500 RPM (maximum torque rotational speed) and 6000 RPM (maximum power rotational speed of engine). Figure 4 shows exerted forces on connecting rod in mentioned speeds.

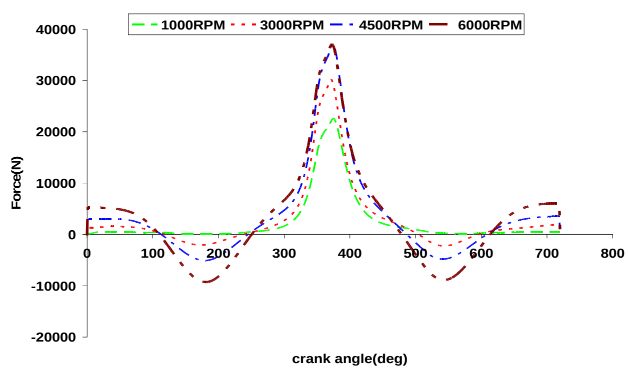

Fig. 4: Exerted forces on connecting rod in different speeds.

Calculated forces in different speeds were exerted on connecting rod and critical stresses were known. The maximum pressure stress was obtained between pin end and connecting rod linkage. The value of this stress was 297.361 MPa (Fig. 5). The maximum tensile stress was obtained in crank end. The value of this stress was 202.927 MPa (Fig. 6).

According to table 1, ultimate strength (σ_u) of C70S6 steel (used for this connecting rod) is 1000 MPa. So factor of safety (F.S.) will be:

$$F.S._{pressure\ stress} = \frac{\sigma_u}{\sigma_{all}} = \frac{1000}{297.361} = 3.363 \tag{2}$$

$$F.S._{tensile\ stress} = \frac{\sigma_u}{\sigma_{all}} = \frac{1000}{202.927} = 4.927 \tag{3}$$

Fair factor of safety for mechanical tools is about 2 to 3 (Kolchin, A. and V. Demidov, 1984), so calculated factor of safety is fair for connecting rod under pressure and tensile loads.

For doing fatigue analysis and calculating lowest fatigue cycle, various critical nods in different speed and different parts of connecting rod were investigated for fatigue analysis. Among critical analysed nods lowest fatigue cycle was calculated equal 10^9 cycles. Nodes whit this fatigue cycle were between pin end and connecting rod linkage. Figure 7 parts A and B show tow sample of analysed nods for fatigue cycle (these nods located in rod of connecting rod and between pin end and connecting rod linkage).

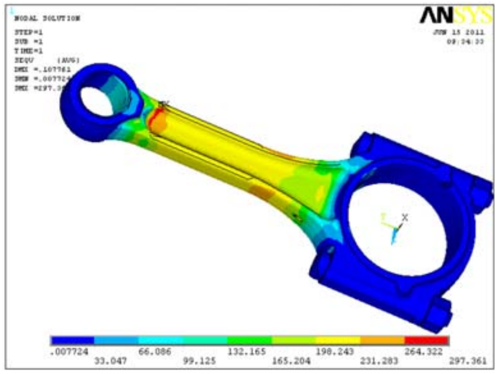

Fig. 5: Stress distribution in connecting rod, resulted from maximum pressure force considering Van Misses.

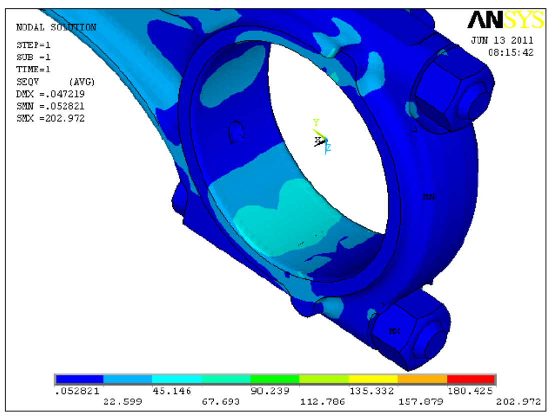

Fig. 6: Stress distribution in connecting rod, resulted from maximum tensile force considering Van Misses.

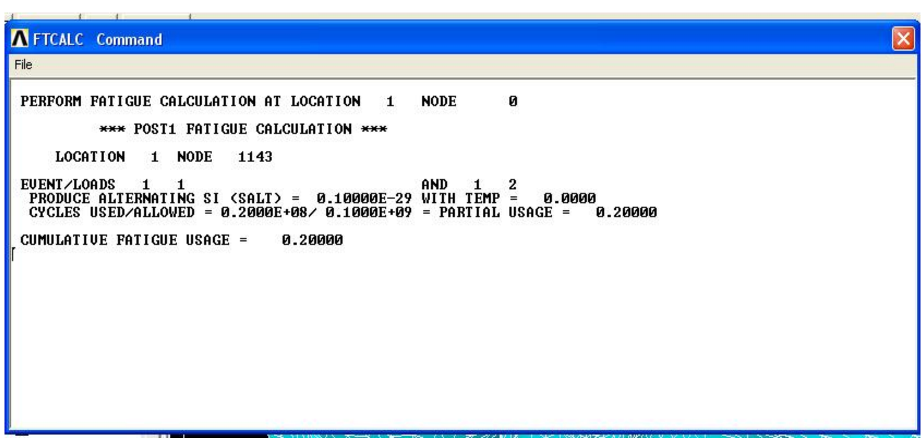

Fig. 7-A: Obtained results for fatigue calculation for a sample node in rod of connecting rod.

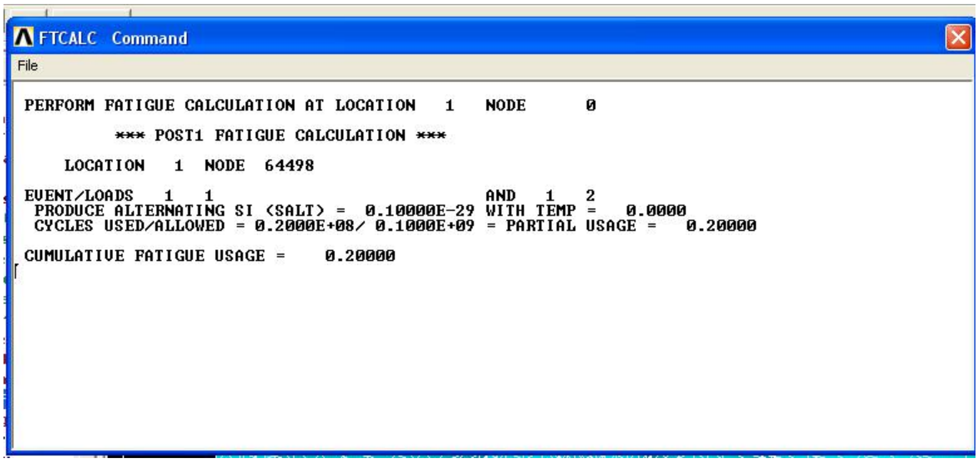

Fig. 7-B: Obtained results for fatigue calculation for a sample node between pin end and connecting rod linkage.

Conclusions:

By the finite element analysis method and the assistance of ANSIS software, It is able to analyze the different car components from varied aspects such as fatigue and consequently save the time and the cost. The way that defined loadings was effective on the results achieved. So, they should fit as much as possible the real conditions. In this research we tried to simulate real condition by notice to all of effective forces on connecting rod. Following conclusions can be drawn as the results of this study:

1. The maximum pressure stress was obtained between pin end and rod linkage
2- The maximum tensile stress was obtained in lower half of pin end.
3- The factor of safety for pressure stress was obtained 3.363 and for tensile stress was obtained 4.927.
4. Least fatigue cycle was obtained equal 10^9 cycle.

Common range of fatigue cycle for connecting rods is between 10^8 to 10^9 cycley. (Shenoy, P.S., Fatemi, A., 2005) according to this point obtained fatigue cycle for Samand engine connecting rod is fair. Also if we assume that

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