Journal of Agricultural Technology 2010 Vol. 6(4): 649-662
Available online http://www.ijat-rmutto.com
ISSN 1686-9141

Optimization of connecting rod of MF-285 tractor

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Asadi, M.R., Rasekh, M., Golmohammadi, A., Jafari, A., Kheiralipour, K. and Borghei, A.M. (2010). Optimization of connecting rod of MF-285 tractor. Journal of Agricultural Technology 6(4): 649-662.

MF-285 tractor has devoted the highest production level in Iran among the other tractors. Engine internal components malfunction of this tractor is high. Above reasons necessitate proper optimization of these components. In this study, detailed load analysis under service loading conditions was performed for connecting rod of this tractor (using the Mathematica software), followed by finite element analysis (FEA) to capture stresses and fatigue cycle (using ANSYS software) and at the end proper proposals were offered for optimizing this component. The results showed that increasing the diameter of pin end, decreasing the diameter of rod, optimization of reciprocating mass and lessen friction between piston pin and connecting rod bush are proper methods to optimize the model for better resistance under hard loads.

Key words: tractor, engine, connecting rod, stress and fatigue, optimization

Introduction

In order to reach sustainable agriculture and to increase mechanization level quality and manufacturing technology of agricultural machinery and also its quantity must be reached to optimum level. One of the most important agricultural machinery is tractor that has main share in planting, retaining and harvesting operations and then in mechanization sector. Tractor MF-285 is main production of Iran Manufacturing Tractor Co. Previously researches showed that engine inner parts' faults of MF-285 are more than other ingredients of this tractor (Mahmoodi and Rezakhah, 2007). Above statements show the importance of optimization of rotating parts of tractor MF-285. In this regard, optimization in connecting rod of this tractor was studied. In 1995, optimizing of geometrical shape of connecting rods considering concentrated

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mass to properly distribute its stress was studied in Yuan-Ze Inst of Tech Research Center (Lee and Lin, 1996). Shenoy and Fatemi (2005) studied connecting rod optimization for weight and cost reduction and also they introduce various forces affecting on connecting rod (Shenoy and Fatemi, 2005). The Finite Element Analysis (FEA) is a powerful computational technique for approximate solutions to a variety of "real-world" engineering problems having complex domains subjected to general boundary conditions. FEA has become an essential step in the design or modeling of a physical phenomenon in various engineering disciplines (Madenci and Guven, 2006). One of the powerful softwares to analyze engineering problems with FEA method is ANSYS that is commercially available.

In this study, first forces affecting on connecting rod is calculated then stress and fatigue analysis is done using ANSYS (Ver.9) software and at end according to the results of stress and fatigue analysis proper proposals for optimization is offered.

Materials and methods

MF-285 engine has 4 reciprocated cylinders with linear arrangement. Engine configuration and qualifications was shown in Table 1.

Table 1. Configuration and qualifications of MF-285 engine (Anonymous, 2008).

Number of Cylinders	4
Piston Course (mm)	127
Cylinder diameter (mm)	101
Indicated Revolution (rpm)	2000
Maximum Revolution (rpm)	2200
Indicated Engine Power (Hp)	71
Maximum Torque (N/m2)	278
Revolution in Maximum Torque (rpm)	1300

Calculating forces exerted on connecting rod

In order to calculate stress in connecting rod it was analyzed for 3 separate parts, because the nature of forces exerted on difference parts of connecting rods are different.

Calculating forces exerted on pin end

The total force exerted on pin end in one cycle is state as (Jangi, 2004):

$$F = F_g + F_i = (P_g - P_0)A_P - m_p R\omega^2(\cos\alpha + \lambda\cos 2\alpha) \quad \text{……..………….(1)}$$

Where F_g is the force resulted by gas pressure in combustion chamber (N), F_i is the inertia forces (N), P_g is gas pressure (KPa), P_o is atmosphere pressure (KPa), A_p is the piston area (m^2), m_p is piston and pin mass (kg), m_{se} is the mass of above part of pin end (kg), ω is revolution speed (rpm), R is crankshaft radius (m), α is crank angle (Rad) and λ is ratio crank radius to connecting rod length.

Gas pressure in combustion chamber (KPa) in one cycle for this engine is introduced as Asadi (2008), Asadi *et al*. (2008) and Asadi *et al*. (2009).

$$P_g = \begin{cases} 101.3 & 0 \le \varphi \le \pi \\ 7.53.x^{-1.21} & \pi \le \varphi \le 2\pi \\ 2950 & 2\pi \le \varphi \le 13/6\pi \\ 29.8.x^{-1.21} & 9/4\pi \le \varphi \le 3\pi \\ 101.3 & 3\pi \le \varphi \le 4\pi \end{cases} (KPa) \quad \text{………………………...(2)}$$

That x in above equation is:

$$x(\varphi) = R\left[1 - COS\varphi - \frac{\lambda}{4}(1 - COS2\varphi)\right] + \frac{2R}{r-1} \quad \text{………………………..(3)}$$

Where r is compaction ratio.

The maximum pressure force exerted on connecting rod is happened in the maximum torque but the maximum tensile force happened in the maximum revolution speed (Shenoy and Fatemi, 2005). Hence, to calculate the maximum pressure force exerted on pin end, 1300 rpm, and to calculate the maximum tensile force, 2200 rpm, were considered (as the information taken from company). Figs. 1 and 2 obtained for total force exerted on pin end in 1300 and 2200 rpm considering Eq. 1 and using MATLAB software.

As shown in Figs. 1 and 2 the maximum pressure force exerted on pin end was 19730 N and the maximum tensile force was 8950 N.

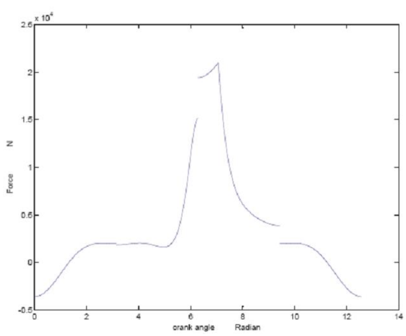

Fig. 1. Total force exerted on pin end versus crank angle diagram in 1300 rpm.

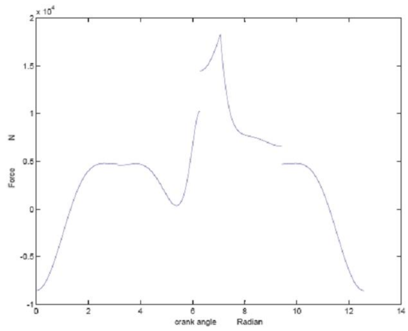

Fig. 2. Total force exerted on pin end versus crank angle diagram in 2200 rpm.

Calculating forces exerted on rod

The total force exerted on rod in one cycle is state as (Jangi, 2004):

$$F = F_g + F_i = (P_g - P_0)A_P - (m_p + m_{crp})R\omega^2(\cos\alpha + \lambda\cos 2\alpha) \qquad \text{........(4)}$$

where m_{crp} is mass of connecting rod above part from gravity center (kg). As stated above, to calculate the maximum pressure force exerted on pin end, 1300 rpm, and to calculate the maximum tensile force, 2200 rpm, were considered. Fig. 3 and 4 obtained for total force exerted on pin end in 1300 and 2200 rpm considering Eq. 4 and using MATLAB software.

The maximum pressure force exerted on rod was 18597 N and the maximum tensile force was 10365 N as shown in Figs. 3 and 4.

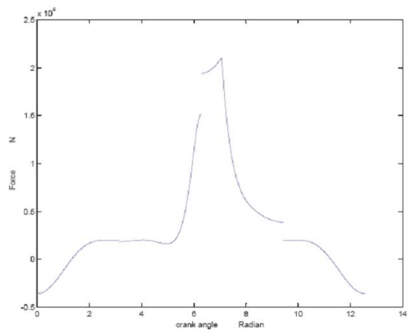

Fig. 3. Total force exerted on rod versus crank angle diagram in 1300 rpm.

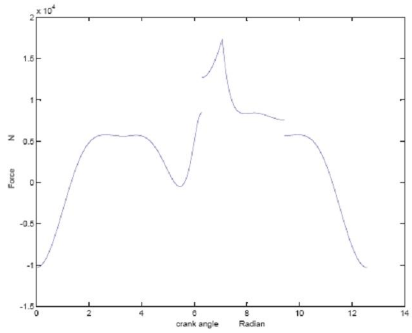

Fig. 4. Total force exerted on rod versus crank angle diagram in 2200 rpm.

Calculating forces exerted on crank end:

The combustion pressure force doesn't have effect on crank end, but it is affected by inertia force (Jangi, 2004). Also, screws in crank end are over load. Always, they preloaded 2 to 3 time related to the maximum inertia force to prevent departing of two bearing cup (Froozanpoor, 1997). Inertia force results tensile stress and preloading force results pressure stress in crank end of connecting rod. Preloading (MPa) in screws to link bearing cup and above part of crank end strongly and also to prevent screws' breaking is equal to:

$$P_{t.l} = 3P_{jr\max} / i_b \quad \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots(5)$$

Where i_b is number of screws in crank end and $P_{j\max}$ is the maximum inertia force exerted on crank end of connecting rod.

The inertia force exerted on crank end was calculated as Jangi (2004).

$$P_{jr} = -\omega^2 R\left[(m_p + m_{crp})(\cos\alpha + \lambda\cos 2\alpha) + (m_{crc} - m_c)\right] \quad \ldots\ldots\ldots\ldots\ldots\ldots(6)$$

Where m_p is Mass of the piston assembly (kg), m_{crc} is concentrated mass of connecting rod on the crank end (kg), m_{crp} is concentrated mass of connecting rod on the pin end and m_c is concentrated mass of crankshaft on crank end (kg). Fig. 5 shows the inertia force exerted on crank end versus crank angle diagram in one cycle. As seen in Fig. 5 the maximum inertia force exerted on crank end was 11170 N. Hence, the maximum pressure loud exerted on crank end from each screw was 16755 N.

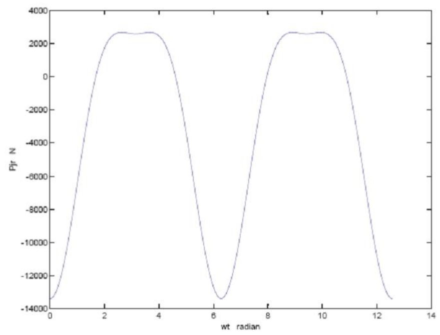

Fig. 5. Inertia force exerted on crank end versus crank angle diagram in 2200 rpm.

Modeling, meshing and loading forces on connecting rod

After calculating forces exerted on different parts of connecting rod in most critically state, it was modeled and meshed in ANSYS (Ver.9) software. Solid92 element was considered to carry analyzing. This element is three dimensional with 10 nods. Also, this element related to Solid72 is better specially, in problems with curve bounds had more accuracy, but it increases time need to solve problems. Material qualification was considered as shown in Table 2. To calculating stress in each connecting rod parts, calculated forces for each parts was exerted on corresponding parts in modeled connecting rod in ANSYS software's medium considering following notes:

1. Inertia forces were evenly exerted on pin end inner level as seen in Fig. 6 (Kolchin and Demidov, 1984). The value of these forces was calculated using following formula:

$$P_i = \frac{F_i}{2r_m l_s} \quad (N/m^2) \qquad \text{.....................................(7)}$$

Where P_i is force per unit area (N/m^2), l_s is pin end width (m), F_i is inertia force and r_m is pin end mean radius (m).

Table 2. Material properties of connecting rod (Anonymous, 2008).

Elasticity module (Pa)	200*109
Poisson ratio	0.33
Density (Kg/m3)	7800

2. As seen in Fig. 7, the force resulted from combustion pressure were sinusicaly exerted on pin end inner level (Kolchin and Demidov, 1984). The value of this force was calculated using following formula:

$$P_g = (\frac{2F_g}{\pi\ r_m l_s})\sin\theta \quad (N/m^2) \qquad \text{.......................................(8)}$$

Where P_g is force per unit area (N/m^2) and F_g is force resulted from combustion (N).

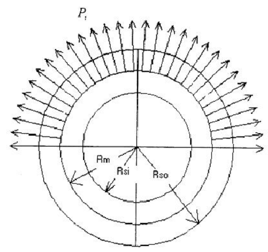

Fig. 6. Inertia force distributing on pin end.

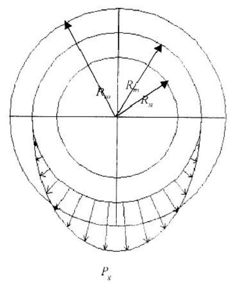

Fig. 7. Force resulted from combustion pressure distributing on pin end.

3. The force resulted from falsifying of pin end's linier and also from friction between linier and piston pin were evenly exerted on pin end inner level all situations. These forces cause pressure stress in linier and tensile stress in connecting rod. This pressure was calculated using following formula (Kolchin and Demidov, 1984):

$$P_b = \frac{\Delta_{tot}}{d_{su}\left[\frac{(d_{su}{}^2 + d_{si}{}^2)(d_{su}{}^2 - d_{si}{}^2) + U}{E_S} + \frac{(d_{su}{}^2 + d_{si}{}^2)/(d_{sa}{}^2 - d_{si}{}^2) - U}{E_b}\right]} \quad \ldots\ldots\ldots\ldots(9)$$

Where Δ_{tot} is sum of initial diameter differences and diameter differences resulted from friction (m), d_{su} is pin end's outer diameter (m), d_{si} is pin end's inner diameter (m), U is Poisson ratio and E_s, E_b is elasticity module of connecting rod and linier (Pa). The value of pressure using above formula for MF-285 was obtained as 26.4 MPa, that this pressure was evenly exerted on pin end level (Kolchin and Demidov, 1984).

4. To obtain stress resulted from preloading in crank end, the force resulted from preloading each screw must be evenly exerted on backrest level of screws (Jangi, 2004)

Results and discussion

Stress analyzing in different parts of connecting rod

Following results were obtained after exerting forces in ANSYS medium.

Pin end

The maximum pressure stress was obtained as 96 MPa in nod: 10296. This nod was located between pin end and rod linkage (Fig. 8). The maximum tensile stress was obtained in nod: 9957, located in lower half of pin end. The value of this stress was 280 MPa (Fig. 9).

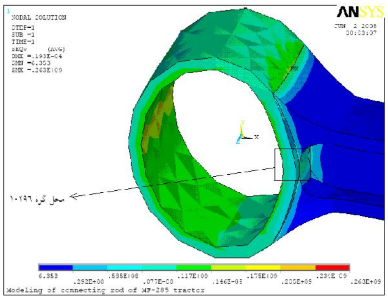

Fig. 8. Stress distribution in pin end, resulted from maximum pressure force considering Van Misses.

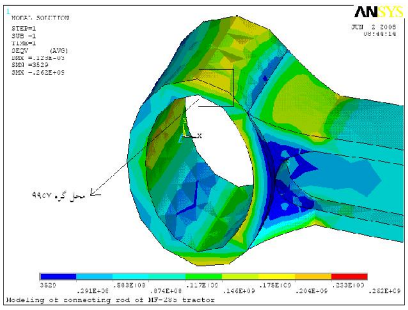

Fig. 9. Stress distribution in pin end, resulted from maximum tensile force considering Van Misses.

Rod

The maximum pressure stress was 109 MPa in nod: 13302, located between pin end and rod linkage (Fig. 10). The maximum tensile stress was 209 MPa in nod: 13439, located between pin end and rod linkage shown in Fig. 11.

Crank end

The maximum stress was obtained in nod: 10186, between bearing cup and connecting rod linkage. The value of this stress was 185 MPa (Fig. 12).

Fatigue analysis in different parts of connecting rod

For doing fatigue analysis and calculating lowest fatigue cycle, various critical nods in different parts of connecting rod (pin end, rod and crank end) were investigated for fatigue analysis (Mireei *et al.,* 2005; Afzal and Fatemi, 2004). Among critical analysed nods lowest fatigue cycle was calculated equal 10^8 cycles.

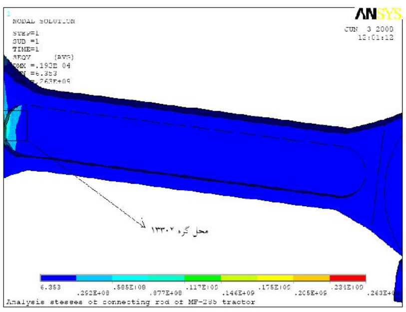

Fig. 10. Stress distribution in rod, resulted from maximum pressure force considering Van Misses.

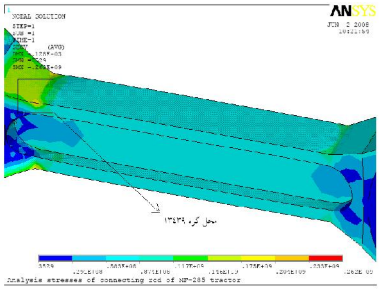

Fig. 11. Stress distribution in rod, resulted from maximum tensile force considering Van Misses.

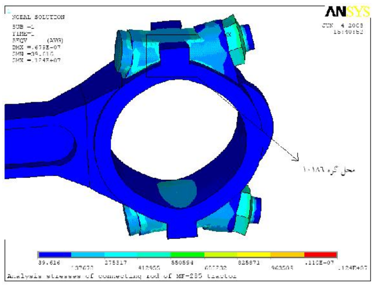

Fig. 12. Stress distribution in crank end, resulted from preloading force considering Van Misses.

Comparing results

For investigating results of FEM accuracy, results of this method were compared with results of experimental equations. In this regard results were compared in three cross sections that are important for designing connecting rods. Fig. 13 shows these sections.

Comparing results in I-I section

Affecting forces on this section are only inertia forces, the force resulted from falsifying of pin end's linier and also from friction between linier and piston pin. Pressure loads don't affect on this section (Kolchin and Demidov, 1984). Fig. 14 shows stresses resulted stresses from tow methods in this section. And percent of difference between tow methods equals:

$$\%Error = \frac{Analytical.Value - FEM.Value}{Analytical.Value} \times 100 = \frac{125.44 - 110.48}{125.44} \times 100 = \%11.92 \quad \ldots\ldots(10)$$

Comparing results in B-B section

This section is in gravity center of connecting rod and should be investigate for pressure and tensile forces also stresses in y-y and z-z direction in this section are important for designing connecting rods (Fig. 15). The pressure stresses resulted from tow method in z-z and y-y direction as shown in Fig. 16 and 17. Percent of difference equals

$$\%Error_{(z-z)} = \frac{50-55}{50} \times 100 = \%10 \qquad \%Error_{(y-y)} = \frac{52-54}{52} \times 100 = \%3.8$$

The tensile stress in B-B section was shown in Fig. 18. Percent of difference equals:

$$\%Error = \frac{26.65-27.12}{26.65} \times 100 = \%1.76$$

Comparing results in II-II section

Stresses resulted from inertia loads by tow methods were compared in this section. (Fig. 19) percent of difference equals:

$$\%Error = \frac{19.63-17.08}{19.63} \times 100 = \%12.9$$

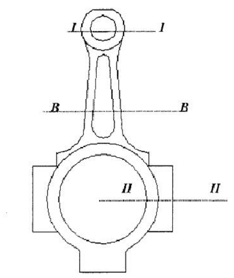

Fig. 13. Important cross sections for designing connecting rods.

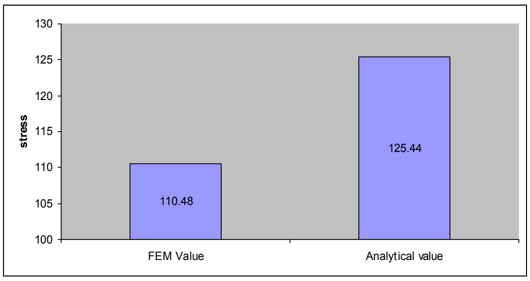

Fig. 14. Comparing calculated stresses by FEM and experimental equations methods in I-I section.

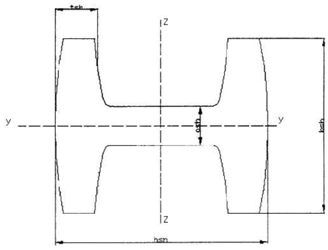

Fig. 15. Diagram of B-B cross section.

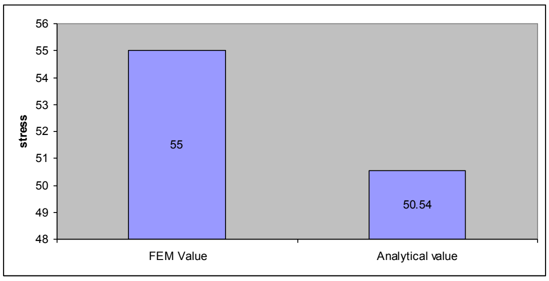

Fig. 16. Comparing calculated pressure stresses resulted from FEM and experimental equations methods in B-B section in z-z direction.

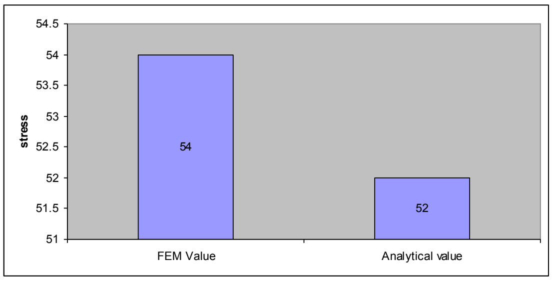

Fig. 17. Comparing calculated pressure stresses resulted from FEM and experimental equations methods in B-B section in y-y direction.

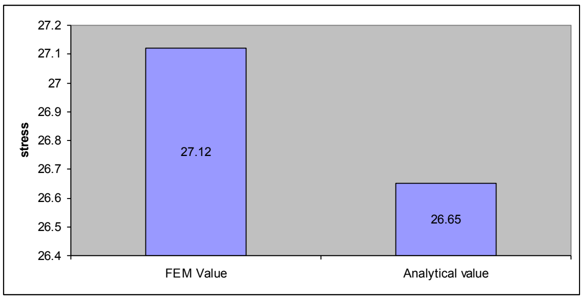

Fig. 18. Comparing calculated tensile stresses by FEM and experimental equations methods in B-B section.

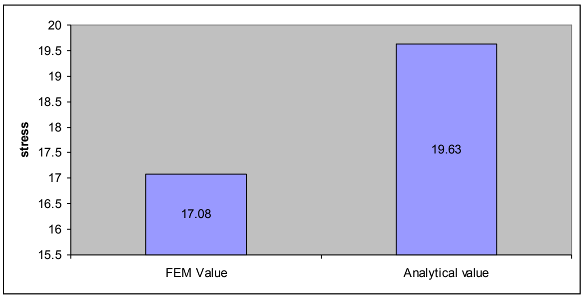

Fig. 19. Comparison calculated stresses from inertia loads by FEM and experimental equations methods in II-II section.

The conclusions can be drawn from this study as follows:- The maximum pressure stress was obtained between pin end and rod linkage and the maximum tensile stress was obtained in lower half of pin end. Least fatigue cycle was obtained equal 10^8 cycles. Results of FEM method and results of experimental equations were similar (Maximum difference was only 13%) this shows accuracy of our modeling, meshing and loading. Common stresses in carbon steel connecting rods like this connecting rod is between 160 to 250 MPa And also common range of fatigue cycle for connecting rods is between 10^8 to 10^9 cycles (Shenoy and Fatemi, 2005). It can be extract that cause of least fatigue cycle and high fail of this component is over stresses of common range.

According to above results following proposals can be offered for optimization and better resistance under hard loads. Increasing the diameter of pin end (Stresses in this part of connecting rod were very high). Decreasing the diameter of rod (Stresses in this part of connecting rod were lower of common range and also fatigue cycle for this part was higher than common range). Optimization of reciprocating masses (critical stresses were caused by inertia loads, that whit optimization of reciprocating masses, inertia loads and so critical stresses will decrease). Lessen friction between piston pin and connecting rod bush. The force caused by friction between piston pin and connecting rod bush was high somedeal. And so finding a method for decreasing this force can decrease critical stresses.

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(Receive 5 August 2009; accepted 20 August 2010)