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Obtaining Maximum Stresses in Different Parts of Tractor (Mf-285) Connecting Rods Using Finite Element Method

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Abstract: Tractor MF-285 is a high volume production of Iran Manufacturing Tractor Co. Engine inner part faults of MF-285 are more than other engine components. So stress analysis in rotating parts of tractor MF-285 for optimizing them is important. In this study, detailed load analysis was performed for a MF-285 connecting rod, followed by finite element method. In this regard, In order to calculate stress in connecting rod, the total forces exerted connecting rod were calculated and then it was modeled, meshed and loaded in ANSYS, 9, software. The maximum stresses in different parts of MF-285 connecting rod were determined. The maximum pressure stress was between pin end and rod linkages and between bearing cup and connecting rod linkage. The maximum tensile stress was obtained in lower half of pin end and between pin end and rod linkages.

Key words: Tractor; Engine; Connecting Rod; Finite Element; Stress Analyzing.

INTRODUCTION

Tractor, as the most important agricultural machinery, has main share in planting, retaining and harvesting operations and then in mechanization sector. Hence, in order to reach sustainable agricultural and to increase mechanization level quality and manufacturing technology of this agricultural machinery and also its quantity must be reached to optimum level. Tractor MF-285 is main production of Iran Manufacturing Tractor Co. Researches show that engine inner parts' faults of MF-285 are more than other engine ingredients (Mahmoodi, A., H. Rezakhah, 2007). Above statements show the importance of stress analysis in rotating parts of tractor MF-285 for optimizing them. In this regard, dynamic stress analysis in connecting rods of this tractor was studied.

In 1995, optimizing of geometrical shape of connecting rods considering concentrated mass to properly distribute its stress was studied in Yuan-Ze Inst of Tech Research Center (Lee, H.J., M.C. Lin, 1996). In this research stresses in connecting rod were calculated using Finite Element method and proper profile for that was obtained.

In this research, authors embarked on determination of points with maximum stress in tractor MF-285 connecting rod to future optimization.

MATERIALS AND METHODS

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

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Table. 1: Configuration and qualifications of MF-285 engine.

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/m^2)	278
Revolution in Maximum Torque (rpm)	1300

Calculating Forces Exerted on Connecting Rods:

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

1. Calculating forces exerted on pin end:
The total force exerted on pin end in one cycle is state as (Jangi, N., 2004):

$$F_{com} = F_g + F_i = (P_g - P_0)A_p - (m_p + m_{se})R\omega^2(\cos\alpha + \lambda\cos 2\alpha) \quad (1)$$

Where P_0 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) and F_g is the force resulted by gas pressure in combustion chamber (N).

Gas pressure in combustion chamber (kp) in one cycle for this engine is introduced as (Asdai, M., 2008):

$$P_g = \begin{Bmatrix} 101.3 & 0 \le \varphi \le \pi \\ 7.53x^{-13} & \pi \le \varphi \le 2\pi \\ 2950 & 2\pi \le \varphi \le 13/6\pi \\ 29.8x^{-1.3} & 9/4\pi \le \varphi \le 3\pi \\ 101.3 & 3\pi \le \varphi \le 4\pi \end{Bmatrix} (KPa) \quad (2)$$

That x in above equation is:

$$x(\varphi) = R\left[1 - \cos\alpha - \frac{\lambda}{4}(1 - \cos 2\alpha)\right] + \frac{2R}{r-1} \quad (3)$$

Where α is crank angle (Rad) and 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, P.S., A. 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). Figures 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 figures 1 and 2, the maximum pressure force exerted on pin end was 19730 N and the maximum tensile force was 8950 N.

2. Calculating forces exerted on rod:
The total force exerted on rod in one cycle is state as (Jangi, N., 2004):

$$F_{com} = F_g + F_i = (P_g - P_0)A_p - (m_p + m_{crp})R\omega^2(\cos\alpha + \lambda\cos 2\alpha) \quad (4)$$

Where m_{crp} is mass of connecting rods 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. Figures 3 and 4 obtained for total force exerted on pin end in 1300 and 2200 rpm considering Eq. 4 and using MATLAB software.

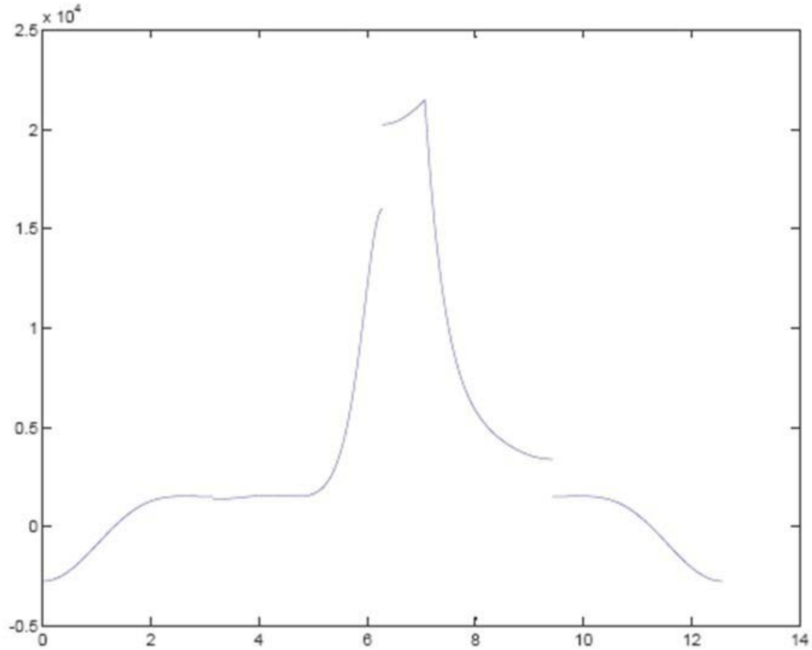

Fig. 1: Total force exerted on pin end versus crank angle diagram in 1300 rpm.

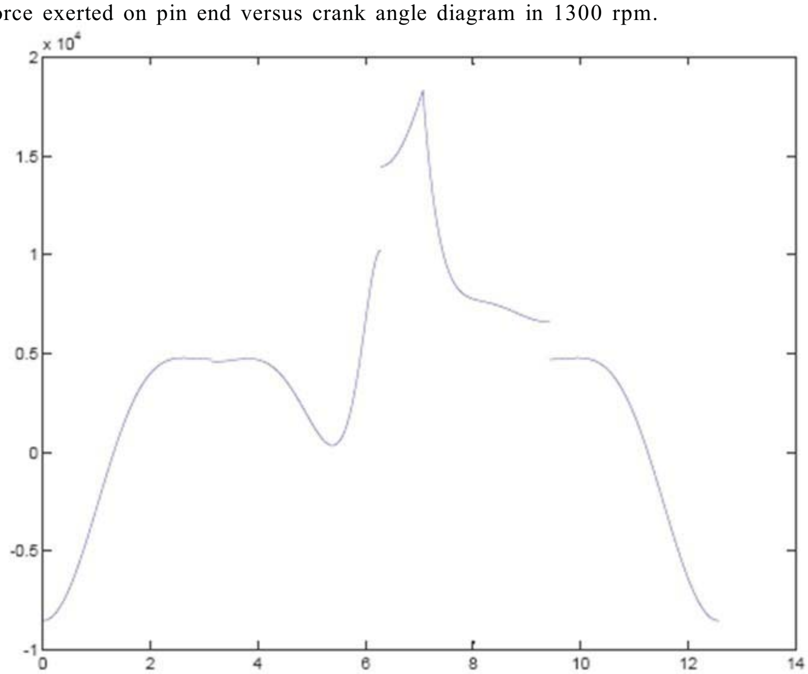

Fig. 2: Total force exerted on pin end versus crank angle diagram in 2200 rpm.

As shown in figures 3 and 4, the maximum pressure force exerted on rod was 18597 N and the maximum tensile force was 10365 N.

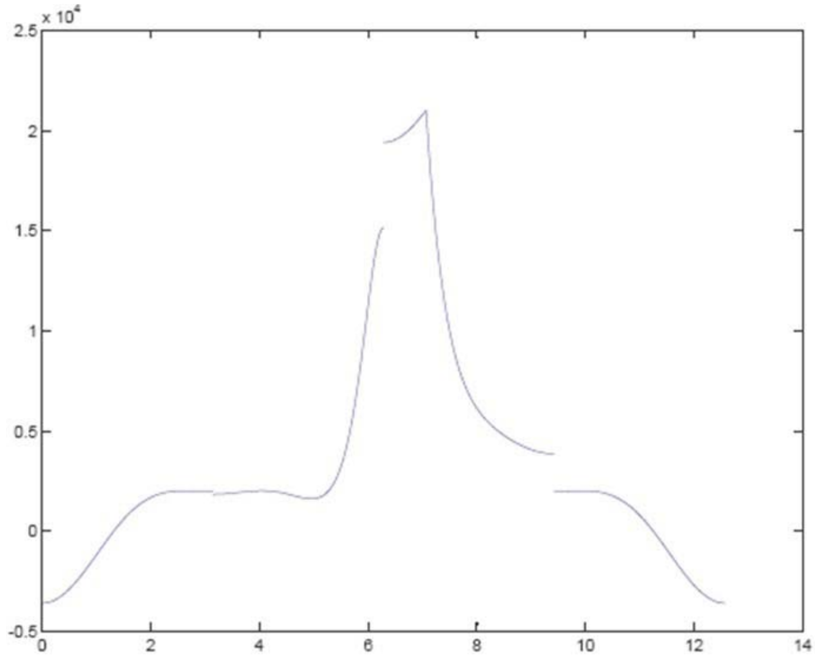

Fig. 3: Total force exerted on rod versus crank angle diagram in 1300 rpm.

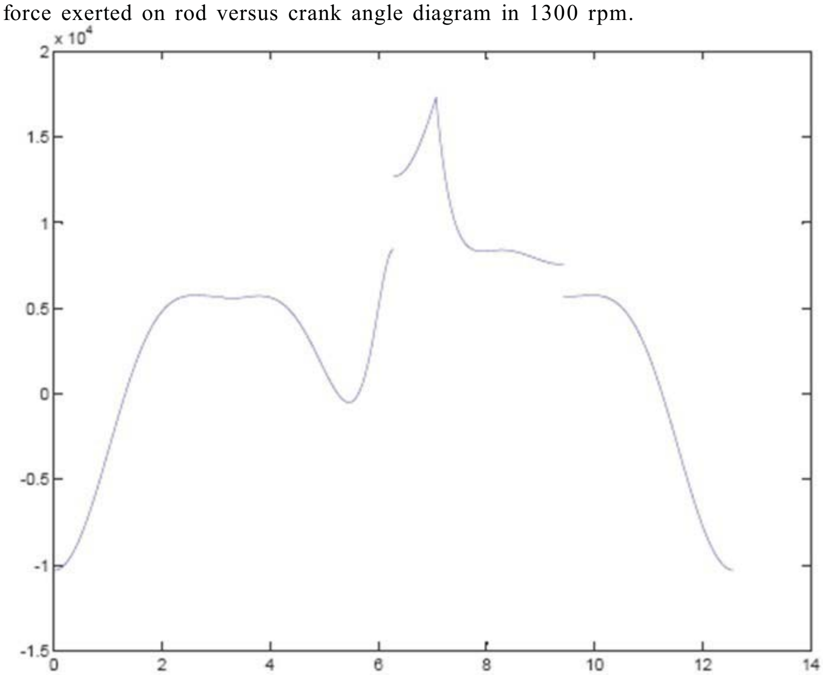

Fig. 4: Total force exerted on rod versus crank angle diagram in 2200 rpm.

3. 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, N., 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, H., 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:

$$P_{t.l} = 3P_{jr\max} / i_b \tag{5}$$

Where i_b is number of screws in crank end and P_{jrmax} is the maximum inertia force exerted on crank end of connecting rods.

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

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

Where m_p is Mass of the piston assembly (kg), m_{crc} is concentrated mass of connecting rods on the crank end, m_{crp} is concentrated mass of connecting rods on the pin end and m_c is concentrated mass of crankshaft on crank end.

Figure 6 show 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.

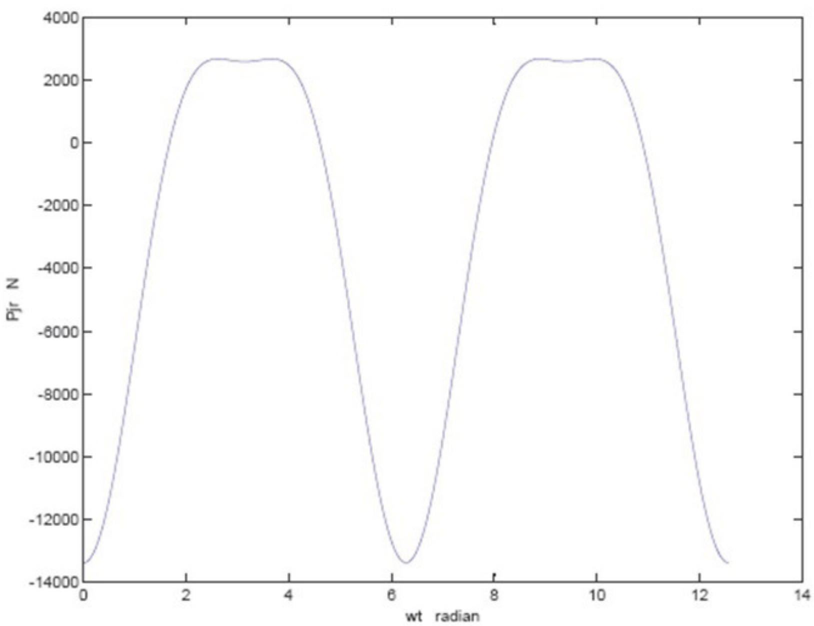

Fig. 5: Inertia force exerted on crank end versus crank angle diagram in 2200 rpm.

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, 9, software. Solid92 element was considered to carry analyzing. This element is three dimensional ي‌جهوراهچ with 10 nods. Also, this nod related to Solid72 is higher in degree and, 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.

Table 2 Material properties of connecting rod.

$200*10^9$	Elasticity module (Pa)
0.33	Poisson ratio
7800	Density (Kg/m^3)

To calculating stress in each connecting rod parts, calculated forces for each parts was exerted on corresponding pats in modeled connecting rod in ANSYS, 9 software's medium considering following notes: Inertia forces were evenly exerted on pin end inner level (Fig. 6) (Kolchin, A., V. Demidov, 1984). The value of these forces was calculated using following formula:

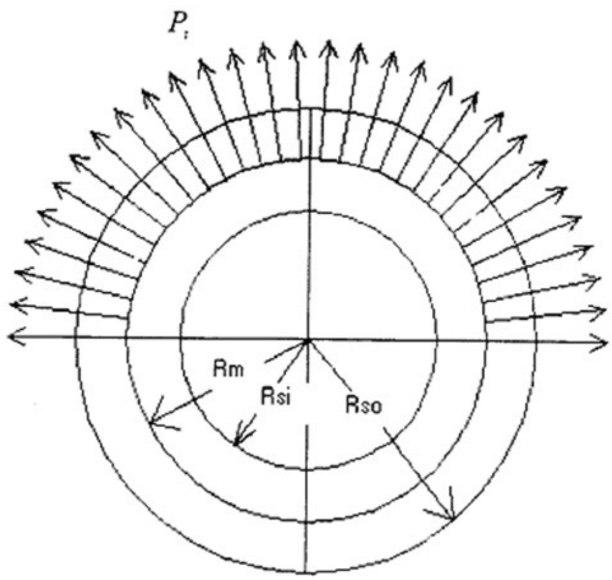

Fig. 6: Inertia force distributing on pin end.

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

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).

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

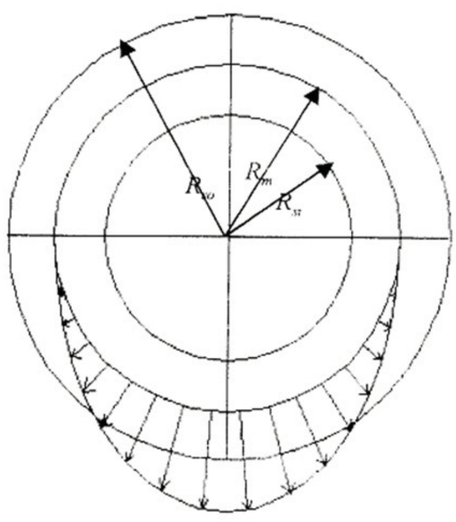

Fig. 7: Force resulted from combustion pressure distributing on pin end.

$$P_g = (\frac{2F_g}{\pi\, r_m l_s}) \sin\theta \qquad (8)$$

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

3. The force resulted from falsifying of pin end's linier and also from friction between linier and piston pin that were 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, A., V. Demidov, 1984):

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

Where Δ_{tot} is sum of initial diameter differences and diameter differences resulted from friction, d_{su} is pin end's outer diameter, d_{is} is pin end's inner diameter, U is Poisson ratio and E_0/E_b is elasticity module of connecting rod and linier. 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, A., V. Demidov, 1984).

4. To obtain stress resulted from preloading in crank end, the force must be evenly exerted on both side of that. Then, average pressure was obtained from dividing force by backrest level of screws (Jangi, N., 2004).

RESULTS AND DISCUSSION

Stress analyzing in different pats 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 (9).

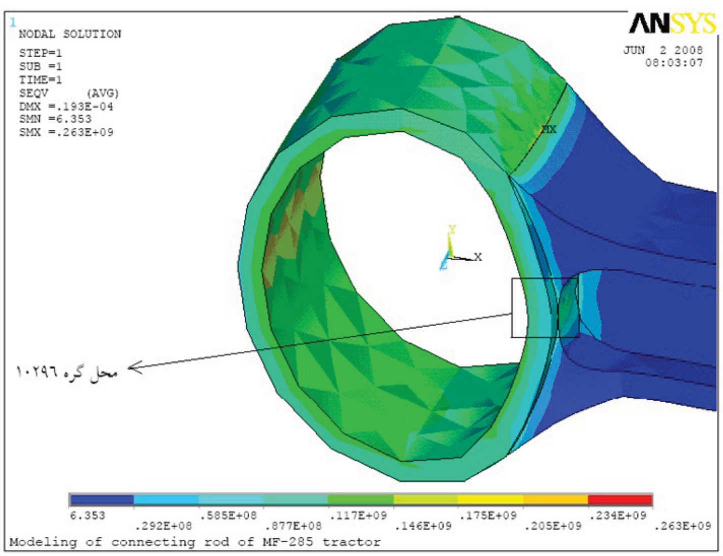

Fig. 8: Stress distribution in pin end, resulted from maximum pressure force considering Van Misses.

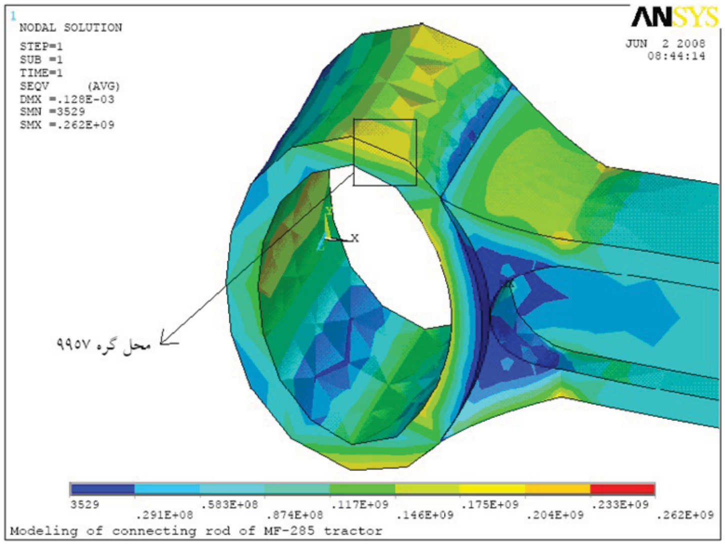

Fig. 9: Stress distribution in pin end, resulted from maximum tensile force considering Van Misses.

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 Figure 11.

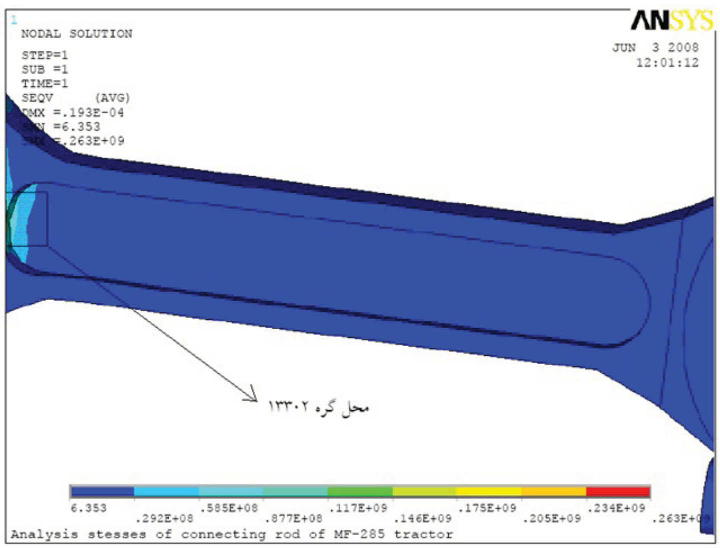

Fig. 10: Stress distribution in rod, resulted from maximum pressure force considering Van Misses.

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).

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 design of connecting rods. Figure shows these sections.
comparing results in I-I section:

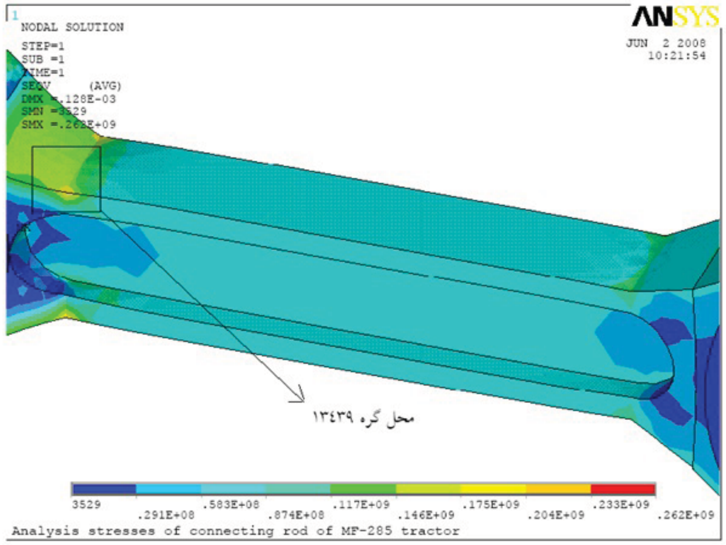

Fig. 11: Stress distribution in rod, resulted from maximum tensile force considering Van Misses.

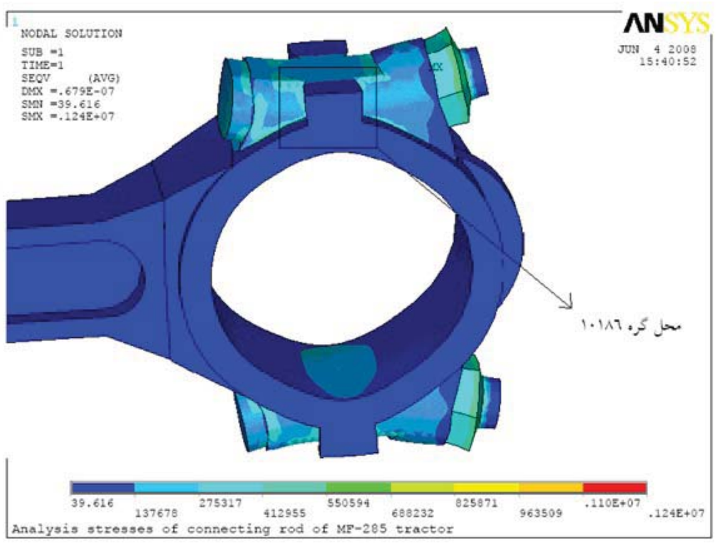

Fig. 12: Stress distribution in crank end, resulted from preloading force considering Van Misses.

Affected forces in this section are only inertia forces and pressure load don't affect this section(Kolchin, A., V. Demidov, 1984). Table shows stresses resulted stresses from tow methods in this section. And difference between tow methods equals:

$$\%Error = \frac{Analytical.Value - FEM.Value}{Analytical.Value} \times 100$$

$$= \frac{125.44 - 110.48}{125.44} \times 100 = \%11.92$$

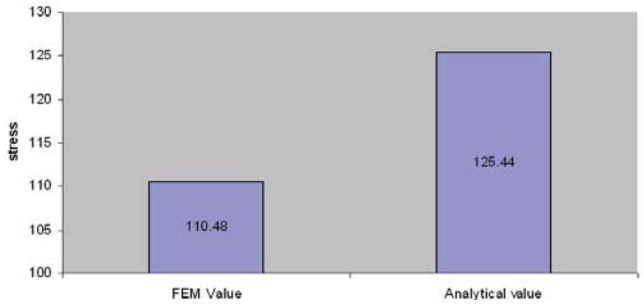

Fig. 13: Comparing calculated stresses by FEM and experimental equations methods in I-I section

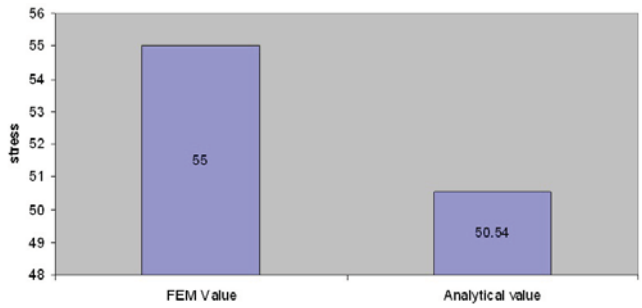

Fig. 14: Comparing calculated stresses by FEM and experimental equations methods in I-I section

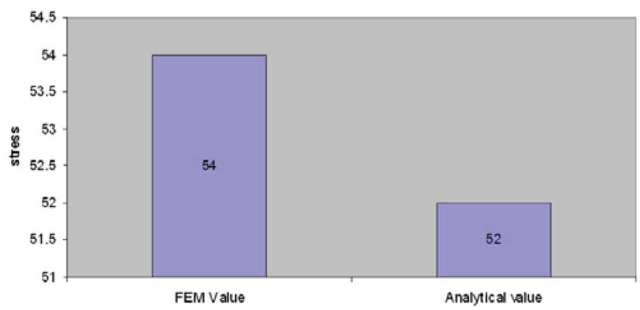

Fig. 15: Comparing calculated stresses by FEM and experimental equations methods in I-I section

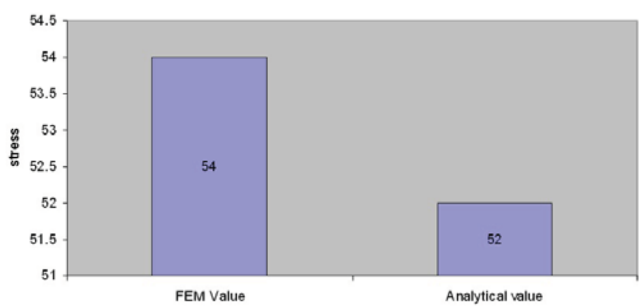

Fig. 16: Comparing calculated stresses by FEM and experimental equations methods in I-I section

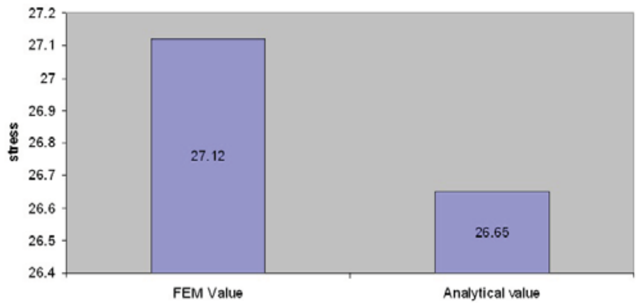

Fig. 17: Comparing calculated stresses by FEM and experimental equations methods in I-I section

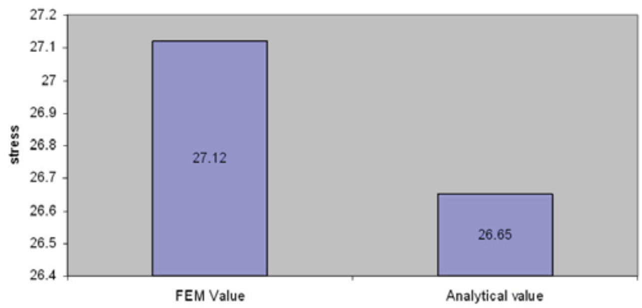

Fig. 18: Comparing calculated stresses by FEM and experimental equations methods in I-I section

b. 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.
Table and show pressure stresses resulted from tow method in z-z and y-y direction. Percent of difference equals:

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

Table shows tensile stress in B-B section. Percent of difference equals:

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

C. comparing results in II-II section:

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

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

Conclusion:

The following conclusions can be drawn from this study:

1. The maximum pressure stress was between pin end and rod linkage, between pin end and rod linkage and between bearing cup and connecting rod linkage.
2. The maximum tensile stress was obtained in lower half of pin end and between pin end and rod linkage.
3 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.

4. Common stresses in carbon steel connecting rods like this connecting rod is between 160 to 250 MPa. It can be extract that cause of high fail of this component is over stresses of common range.

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