Structural and Fatigue Analysis of Diesel Engine Connecting Rod

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Abstract *—Connecting rod is the intermediate member between the piston and the crankshaft. Its primary function is to transmit the push and pull from the piston pin to the crank pin and thus convert the reciprocating motion of the piston into rotary motion of the crank.*

In our project a connecting rod used in a diesel engine using theoretical calculations will be designed for two materials Carbon Steel and Aluminum alloy A360.

The connecting rod will be modeled in 3D modeling software Creo 2.0.

To validate the strength of the connecting rod, static analysis will be done. Modal analysis to determine number of modes and Fatigue analysis to determine life, damage and safety factor will also be done. Analysis will be done using two materials Carbon Steel and Aluminum alloy A360 to verify the best material for connecting rod using Ansys 14.5.

Keywords — *Ansys, Connecting rod, CREO 2.0, FEM.*

I. Introduction

Connecting rods are widely used in variety of car engines. The function of connecting rod is to transmit the thrust of the piston to the crankshaft, and as the result the reciprocating motion of the piston is translated into rotational motion of the crankshaft. It consists of a pin-end, a shank section, and a crank end. 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 by the piston pin. The other end revolves with the crankshaft and is split to permit it to be clamped around the crankshaft. [1] The two parts are then attached by two bolts. 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. [2] Therefore, a connecting rod 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. The connecting rod of the tractors is 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. Nevertheless, connecting rod design is complicated because the engine is to work in variably complicated conditions and the load on the rod mechanism is produced not only by pressure but also inertia. When the repetitive stresses occur in connecting rod it leads to fatigue phenomenon which can cause so dangerous ruptures and damages. An example of the fatigue analysis and design was presented in 2003 by some researchers. [3] A rupture due to the fatigue and the method of correcting the connecting rod design was also reported presented a strengthening method for the connecting rod design. Finite element (FEM) method is a modern way for fatigue analysis and estimation of the component longevity which has the following advantages compared to the other methods. Through this method, we can access the stress/strain distribution throughout the whole component which enables us to find the critical points authentically. [4] This achievement seems so useful particularly when the component doesn't have a geometrical shape or the loading conditions are sophisticated. The influential component factors are able to change such as material, cross section conditions etc. Component optimization against the fatigue is performed easily and quickly. Analysis is performed in a virtual environment without any necessity for prototype construction. Totally these qualities, lead to savings in time and cost. For the reason that the connecting rod failure is usually due to the fatigue phenomenon, consequently in this research a U650 tractor connecting rod behavior, from the fatigue point of view, is investigated through the ANSYS software [5-11].

II. MATERIALS AND METHODS

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 strains.

$$\frac{\Delta\varepsilon}{2} = \frac{\sigma F}{E}(2N)^{b} + \varepsilon f\,(2N)^{c}$$

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 *F* σ and *F* ε are the coefficient of fatigue strength and elasticity respectively. The necessary parameters for determining the connecting rod material is brought in dimensions are represented in. The next stage was to mesh the model. The 10 node pyramid elements were used as shown in. The reason for choosing this element was to make the geometrical parts of a complicated mechanical component so enable us to gain more authentic results based on the high techniques of fatigue life calculation. First of all, the boundary conditions were defined, exerting a tension force. Afterwards, a compressive force, exactly with the same magnitude but in a reverse direction substituted the tension force and it was solved again. In every phase of loading by entering to the POST1 processor, the Von Misses stresses were activated and the critical points were determined. Through the tension loading while node 46 was the result of compressive loading. After determination of these critical points, they were elected as the points for fatigue investigation. Eventually a 106 force cycle was exerted to the model and partial consumption rate which indicated the number of exerted cycles to allowable ones for each node was gained.

2.1 The C-70 Story

A new steel, C-70, has been introduced from Europe as a crackable forging steel. Alloying elements in the material enable hardening of forged connecting rods when they undergo controlled cooling after forging. This material fractures in a fashion similar to powder forged materials. Recently the American Iron and Steel Institute's (AISI) Bar and Rod Market Development Group has promoted C-70 as an improved material over PF alloys on the basis of optimization work and economic analysis performed by a candidate for a Master's of Science Degree in Mechanical Engineering at the University of Toledo. The thesis advisor was Dr. Ali Fatemi. The study investigated weight and cost reduction opportunities of steel forged connecting rods. Analysis focused on comparing and then optimizing a rod design using crack able forged steel (C-70). Using finite element analysis (FEA) techniques, the author was able to reduce the weight by 10% and by using "crackable" C-70, reduce the costs by 25% (over current forged steel connecting rods) and ostensibly 15% less than a PF rod with similar or better fatigue behavior. The study identified fatigue strength as the most significant design factor in the optimization process. The AISI funded study focused on using FEA analysis to show where and how the original connecting rod design configuration could be reconfigured to reduce weight and by using the "crack able" C-70, to eliminate some cost considerations.

III.DESIGN CALCULATIONS

3.1. Carbon steel

3.1.1. Dimensions of cross section of connecting rod

T = Thickness of flange and web of the section
B = 4t = Width of the section
H = 5t = Height of the section
Area of the section
$A = 2(4t\text{x}t) + 3t\text{x}t = 11t^2$
Moment of Inertia of section about x-axis
$I_{xx} = 419/12\ t^4$
Moment of Inertia of section about Y-axis
$I_{yy} = 131/12\ t^4$
$I_{xx}/ I_{yy} = 3.2$
Stroke length l = 82 mm
Bore diameter D = 69.6 mm
No. of cylinders = 4
Length of the connecting rod = 2 times the stroke length
L = 2 l = 2x 82 =164 mm
Buckling load W_B =maximum gas force x factor of safety
Face =max gas load
$f_c = p\ \text{x}A$
$A = \pi/4\ D^2 = 3802.6656\ mm^2$
$f_c = 10.936 \text{ x } 3802.6656$
Fc = 41585.95 N
Factor of safety =5 to 6
$W_B = 41585.95 \text{ x } 6$
$= 249515.7$ N
$W_B = \sigma_c \text{ x } A/ [1+a(L/K_{xx})]$
Σc = compressive yield strength =285 N/mm^2
$K_{xx} = I_{xx}/A = \frac{[419/12\ t^2]}{[11t^2]} = 3.17\ t^2$
$K_{XX} = 1.78$ t
$a = \text{constant} = \sigma_c / \pi^2 E$
$E = 200000\ N/mm^2$
$a = 285 / \pi^2\ (200000) = 0.0001445$
$249515.7 = 285 \text{ x } 11t^2/ [1+0.0001445(164/1.78t)^2$
$t^4 - 79.59t^2 - 97.0998 = 0$
t = 8.98mm=9mm
B = 4t =4x 9 = 36 mm
H = 5t = 5x 9 =45 mm
Depth near the small end H_1 =0.75H to 0.9 H
H = 0.9 x 7.5 =40.5 mm
Depth near the big end H_2 =1.1H to 1.25H
H_2 = 56.25 mm

3.1.2. Dimensions of the crank pin at the big end

Load on the crank pin = projected area x bearing pressure
$F_L = d_C \text{ x x } P_{bc}$
$l_C = 1.25 \text{ x } d_c$ to $1.5\ d_c$
P_{bc} = Allowable bearing pressure at the crank pin

P_{bc} = 50 N/mm^2
$F_L = \pi/4\ D^2 \times P = \pi/4 \times (69.6)^2 \times 10.936$

F_L = 41585.951
41585.951 = 1.5$d_c{}^2$ x 50
$d_c{}^2$ = 41585.951/1.5x50
d_c = 23.54mm
l_C =1.5 d_c =35.32 mm

3.1.3. Size of bolts for securing the big end

Inertia force of the reciprocating parts $E_I = m_R\ x\omega^2 r\ (\cos\theta+\cos 2\theta/l/r)$
ω = Angular speed of the engine in rad/sec
ω = 4000rpm =418.87 rad/sec
m_R =mass of reciprocating parts in kg=1.747kg
Mass of piston =1.36 kg
Mass of piston pin = 35gms =0.035
Mass of connecting rod =0.352 kg
Angle of inclination of crank with the line of action
θ =0
r = radius of crank
l = length of connecting rod
l/r =4
Force on the bolts = $\pi/4(d_{cb})^2\ \sigma_t\ n_b$
σ_t = Allowable tensile stress
Bolts can be made of high carbon steel (or) nickel alloy steel
σ_t =380-620 mpa
d_{cb} = core diameter of the bolt in mm
n_b = no. of bolts = 2
$F_I = m_R\ x\omega^2 r\ (\cos\theta+\cos 2\theta/l/r)$
$F_I = m_R\ x\omega^2 r(1+r/l)$
Radius of crank = l/4 = 164/4 =41mm
$F_I = 1.747(418.87)^2 \times 0.041(1+1/4)$
F_I = 12567.1(5/4)
F_I = 15708.88
$F_I = \pi/4(d_{cb})^2\ \sigma_t\ n_b$
15708.88 = $\pi/4(d_{cb})^2$x380x 2
15708.88 = 596.6$(d_{cb})^2$
d_{cb} =5.13mm
Nominal (or) major diameter of bolt
$d_b = d_{cb}/0.84$ =5.13/0.84 =6.1mm

3.1.4. Thickness of big end cap

Maximum bending moment $M_c = F_I \times X/6$
X = distance between bolt centre
X = dia of crank pin (d_c) + 2 x Thickness of bearing +clearance
X = 23.54+2x3+3 =32.54mm
b_c = width of cap in mm =length of crank pin
b_c = 35.32mm
Section modulus for the cap
$Z_C = b_c\ (t_c)^2/6$
Bending stress $\sigma_b = M_C / Z_C = F_I \times X/ b_c\ (t_c)^2$
σ_b =230 N/mm^2
230 = 15708.88x32.54/35.32 $(t_c)^2$
$(t_c)^2$ =62.92
t_c =7.93mm

3.2. ALUMINUM

3.2.1. Dimensions of cross section of connecting rod

T = Thickness of flange and web of the section
B = 4t = Width of the section
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Area of the section
A = 2(4txt)+3txt = 11t^2
Moment of Inertia of section about x-axis
I_{xx} = 419/12 t^4
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No.of cylinders = 4
Length of the connecting rod = 2 times the stroke length
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Buckling load W_B =maximum gas force x factor of safety
f_c =max gas load
f_c = p xA
A = $\pi/4\ D^2$ = 3804.594 mm^2
f_c = 10.936 x 3804.594
f_c = 41607.044 N
Factor of safety =5 to 6
W_B = 41607.044 x 6
= 249642.264 N
$W_B = \sigma_c \times A/[1+a(L/K_{xx})]$
σ_c = compressive yield strength =172 N/mm^2
$K_{xx} = I_{xx}/A = \frac{[419/12\ t^2]}{[11t^2]}$ =3.17 t^2
K_{XX} =1.78 t
a = constant = $\sigma_c / \pi^2 E$
E = 200000 N/mm^2
a = 172 /π^2(80000) =0.0002178
249642.264 = 172 x 11t^2/[1+0.000217(164/1.78t)^2
t^4-131.946t^2-243.054=0
t = 11.56=11mm
B = 4t =4x 9 = 36 mm
H = 5t = 5x 9 =45 mm
Depth near the small end H_1 =0.75H to 0.9 H
H = 0.9 x 7.5 =40.5 mm
Depth near the big end H_2 =1.1H to 1.25H
H_2 = 56.25 mm

3.2.2. Dimensions of the crank pin at the big end

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Maximum bending moment $M_c = F_I$ x X/6
X = distance between bolt centre
X = dia of crank pin (d_c) + 2 x Thickness of bearing +clearance
X = 23.54+2x3+3 =32.54mm
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σ_b =230 N/mm^2
230 = 15708.88x32.54/35.32 $(t_c)^2$
$(t_c)^2$ =62.92
t_c =7.93mm

IV. MODELLING OF CONNECTING ROD

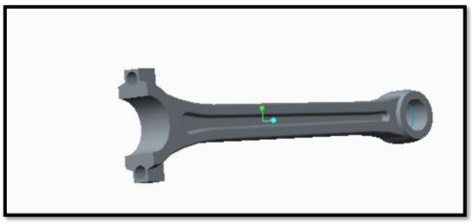

Fig.1. Connecting rod part 1


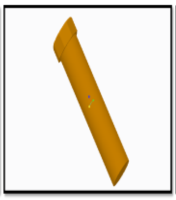
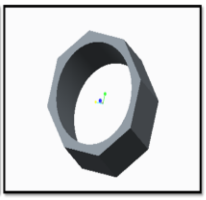

Fig.2. Connecting rod part 2 Fig.3. Bolt Fig.4. Nut

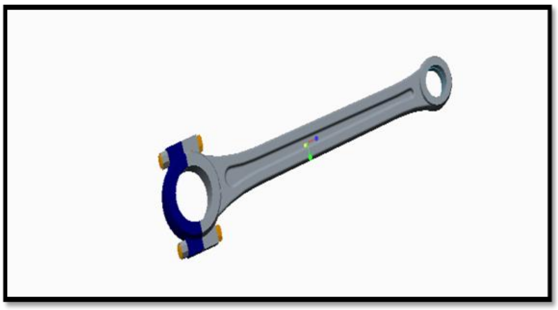

Fig.5. Assembly of Connecting Rod

V. ANALYSIS OF CONNECTING ROD

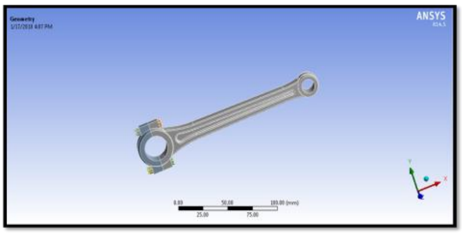

Fig.6. Imported model

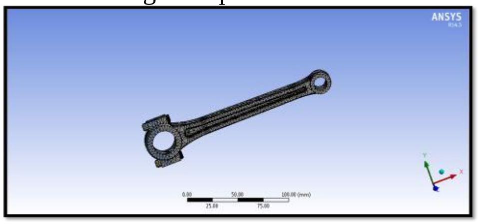

Fig.7. Meshed model

VI. RESULTS AND DISCUSSION

6.1 Static analysis

6.1.1. Carbon steel

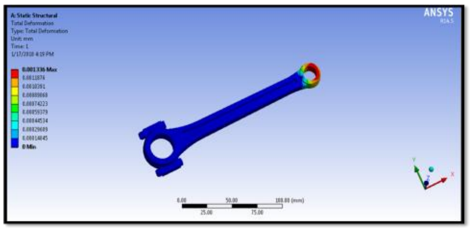

Fig.8. total deformation

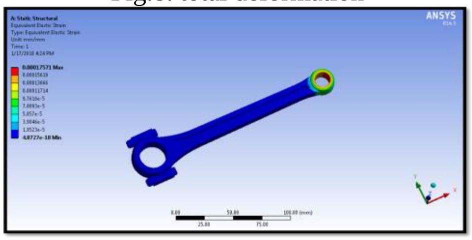

Fig.9. strain

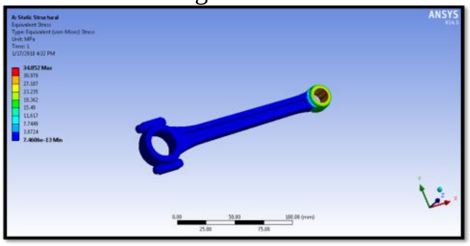

Fig.10.stress

6.1.2. Aluminium alloy A360

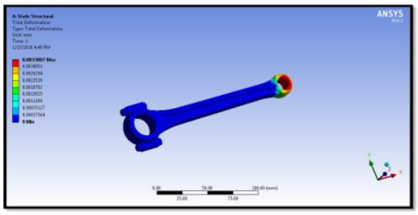

Fig.11.Total deformation

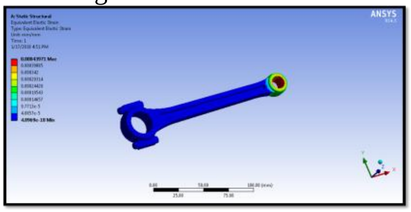

Fig.12.Strain

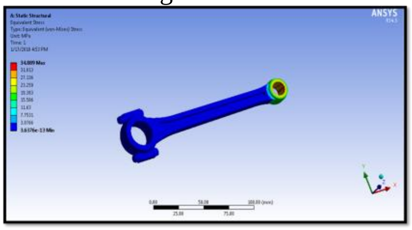

Fig.13.Stress

6.2 Modal analysis

6.2.1. Carbon steel

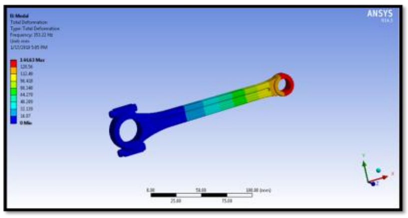

Fig.14.Mode shape 1

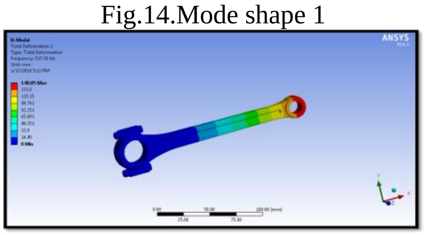

Fig.15.Mode shape 2

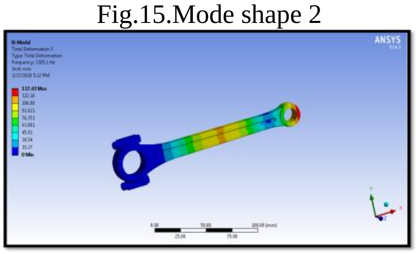

Fig.16.Mode shape 3

6.2.2. Aluminium alloy A360

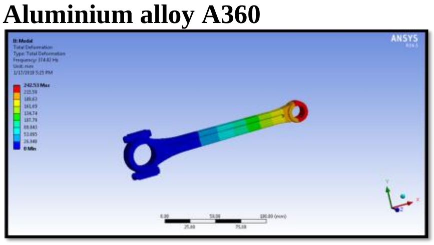

Fig.17. Mode shape 1

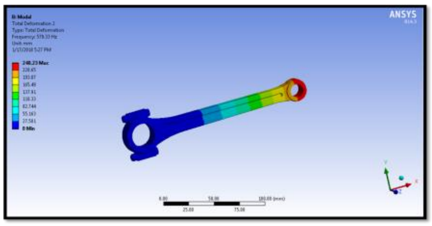

Fig.18. Mode shape 2

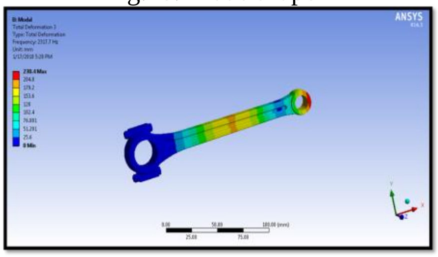

Fig.19. Mode shape 3

6.3. Fatigue analysis

6.3.1. Carbon steel

	A
1	Me...
2	5.16E+09
*	

	B	C
1	Cycles	Alternating Stress (MPa)
2	10	5160
3	100	4200
4	1000	3600
5	10000	2300
6	1E+05	1100
7	1E+06	263
*		

Fig.20. Alternating stress

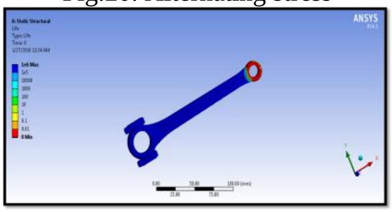

Fig.21. Life

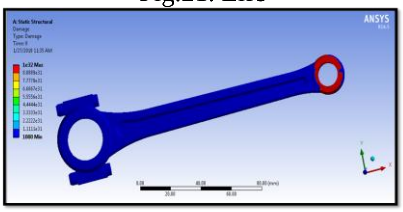

Fig.22.Damage

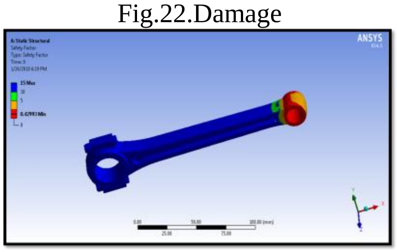

Fig.23.Safety factor

6.3.2. Aluminium alloy A360

	A
1	Mean Stress (Pa)
2	0
*	

	B	C
1	Cycles	Alternating Stress (MPa)
2	10	1500
3	100	1280
4	1000	1130
5	10000	1050
6	1E+05	850
7	1E+06	600
8	1E+07	320
9	1E+08	140
10	1E+09	75
*		

Fig.24.Alternating stress

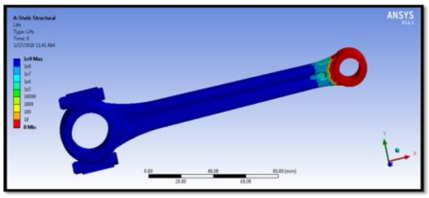

Fig.25.Life

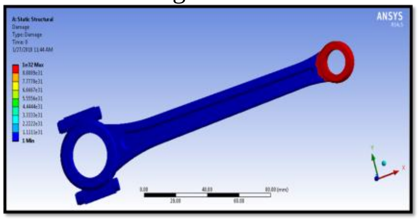

Fig.26.Damage

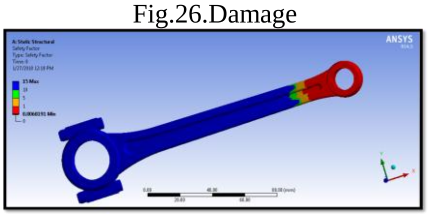

Fig.26.Safety factor

Table.1.Static Analysis

	Total deformation(mm)	Strain	Stress (MPa)
Carbon steel	0.001336	0.00017571	34.852
Aluminum alloy A360	0.0033807	0.00043971	34.889

Table.2.Fatigue Analysis

	Life	Damage	Safety factor
Carbon steel	1e6	1e32	15
Aluminum alloy A360	1e9	1e32	15

Table.3.Modal analysis

	Frequency (Hz)		
	Mode1	Mode2	Mode3
Carbon steel	144.63	148.05	137.43
Aluminum alloy A360	242.53	248.23	230.4

VII. CONCLUSION

In our project we have designed a connecting rod using Carbon Steel and Aluminum alloy A360. The models are created using 3D modeling software Pro/Engineer.

Present used material for connecting rod is Carbon Steel. We are replacing with Aluminum alloy. The density of Aluminum alloy is less than that of Carbon Steel. So weight of the connecting rod reduces by using Aluminum alloy. By using carbon steel, the weight of the connecting rod is 273.912gms and that by using Aluminum alloy is 95.277gms.

We have done static structural and modal analysis on the connecting rod using materials Carbon Steel and Aluminum alloy A360. By observing the analysis results, the stress values are less than their respective yield stress values. So using Aluminum alloy A360 is safe for connecting rod.

By observing fatigue analysis results, the life and safety factor are less at the small end of connecting rod. Life is more when Aluminum alloy A360 is used.

By observing modal analysis results, the frequencies are more when Aluminum alloy A360 is used due to which vibrations will increase.

So we can conclude that using Aluminum alloy A360 is better for connecting rod.

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