FATIGUE ANALYSIS OF CONNECTING ROD OF INTERNAL COMBUSTION ENGINE MADE FROM ALUMINUM ALLOY – A COMPARISON WITH STEEL AND TITANIUM ALLOY MATERIALS +

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Abstract :

The internal combustion engine is consist of three main components, the Piston, the Connecting Rod, and the Crankshaft. Connecting rods can be manufactured from many different materials and with many design shapes, each type of connecting rod is designed according to the type of engine fuel (diesel or petrol). During manufacturing an engine it is important to choose the correct connecting rod. Information about fatigue analysis was needed for connecting rod to properly decide which one is optimum for there engine according to safety and life, by using numerical technique of Finite Element method (ANSYS) Package. Higher safety factor is achieved by using aluminium alloy rather than titanium alloy and steel materials with about 32.1 % and 35.6 % percentage increment respectively. Aluminium connecting rod has life reached higher than both mention materials about 99 %.

Keywords : **Connecting rod, Fatigue, Life, ANSYS**.

تحليل الكلال لذراع التوصيل في محرك الإحتراق الداخلي المصنوع من سبيكة الألمنيوم – مقارنة مع الذراع المصنوع من الفولاذ وسبيكة التيتانيوم

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المستخلص :

ان محركات الاحتراق الداخلي تتكون من ثلاث اجزاء رئيسية هي المكبس وذراع التوصيل والمحور الدوار. يمكن تصنيع ذراع التوصيل من معادن وتصاميم مختلفة , وكل نوع من اذرع التوصيل له استخدام حسب طبيعة وقود المحرك (كاز او بنزين) . من المهم عند تصميم المحرك اختيار ذراع التوصيل المناسب , في حين ان من يتخذ مثل هكذا قرار ينبغي ان يكون مسلحا بالمعلومات المطلوبة لتحليل الكلال لذراع التوصيل لغرض اخذ الاختيار الافضل بالاعتماد على الامان والعمر الاكبر لذراع التوصيل في المحرك بواسطة استخدام التحليل العددي لطريقة العناصر المحددة في برنامج ANSYS. اعلى معامل امان تم استنتاجه من البحث هو من سبيكة الالمنيوم بمعدل زيادة مئوية مقدارها % 32.1 و % 35.6 عن سبيكة التيتانيوم والفولاذ على التوالي. يزيد عمر ذراع التوصيل المصنوع من سبيكة الالمنيوم عن بقية المعادن المستخدمة بنسبة مئوية تصل الى % 99.

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Nomenclature :

A_c : Cross – sectional area of crank shaft (m^2).
A_p : Cross – sectional area of piston (m^2).
F_m : Mean force on connecting rod (N).
F_a : Alternating force on connecting rod (N).
m : Mass (kg).
t : Time (sec).
N : Safety factor.
P_m : Mean pressure on piston (Mpa).
P_a : Alternating pressure on piston (Mpa).
σ_a : Alternating stress (Mpa).
σ_e : Endurance stress (Mpa).
σ_m: Mean stress (Mpa).
σ_y : Yield stress (Mpa).
ρ : Density (kg/m^3).
ω : Angular velocity (rad/s).

1. Introduction:

Connecting rod in the automobile engines is an important component transferring linear motion of the piston resulting from combustion of fuel to the rotational motion of crankshaft. *Tae-Gyu Kim* [1] was state that, the fatigue characteristics of SMA40 materials as a potential application in connecting rod material for an automobile. Average fatigue life of 168700 cycles and fatigue limit of 437MPa was determine from SMA40 specimens. However, when tested under fretting condition, decrease in fatigue lifetime was 20 % and fatigue limit was reduced to 350 MPa.

Copper is also frequently admixed in the case of connecting rod. PF materials undergo little material deformation resulting in isotopic mechanical properties, contributing to superior fatigue resistance. Mechanical properties are further enhanced by better surface finish and finer grain size have been investigated by *James R. Dale* [2].

The implication of the data investigated in [2] is state that PF materials demonstrate improved fatigue strength on the order of 25 – 33 % over C-70 material of the same design. Connecting Rods can be made from various grades of steel, aluminum, and titanium. Steel rods are the most widely produced and used type of connecting rods. Their applications are best used for daily drivers and endurance racing due to their high strength and long fatigue life.

Columbus and Nebraska were state that connecting rod failures are usually caused by over revving the engine, which places a tremendous amount of stress on the rod bolts. The rod bolts will come apart as the piston travels up on the exhaust stroke, the force and inertia of the rotating assembly rips the rod cap off of the connecting rod [3].

A finite element analysis was performed in connection with an analytical fracture mechanics approach aiming to evaluate the relation between tightening force and fatigue crack propagation in connecting rod bolts. The engine collapse occurred due to forming laps in the grooves of the bolt shank [4].

The fatigue stress of connecting rod under the maximum combustion pressure and inertia force condition is calculated by *Zhou Qinghui, Wang Yunying and Ji Wei* using the durability module. The stress is mainly produced on the joint of connecting rod shell and the bottom end or the top end. And the biggest stress acting on the connecting rod is just 34.0613 MPa, early smaller than its limited stress 355 MPa. The method provides the theoretical evidence for connecting rod structure improvement and optimum design [5].
A finite element analysis was performed by *S. Griza, F. Bertoni, G. Zanon, A. Reguly, etc* in connection with an analytical fracture mechanics approach aiming to evaluate the relation between tightening force and fatigue crack propagation in connecting rod bolts. The engine collapse occurred due to forming laps in the grooves of the bolt shank [6].
Fatigue analysis and longevity after a 1000000-cycle load, assessed through using of ANSYS software was performed by *M. Omid, S.S. Mohtasebi, S.A. Mireei, and E. Mahmoodi* Calculations were 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. According to this study, the critical point of the connecting rod of U650 is end of the shank and near piston pin hole [7].
The stress pins as an innovative pre-stressing element was taken into consideration by *M. Koc and M.A. Aslanm* , and presents the results of a design methodology investigation through combined finite element analysis (FEA) and design of experiment (DOE) studies for large and non – Axisymmetric precision forming tooling (i.e. as in hydro-forming and precision forging of connecting rod) where conventional shrink-fit solutions cannot be applied effectively and economically [8].

2. Mathematical Modeling :

The schematic of internal combustion engine chamber (piston, piston pin, and connecting rod) is clearly shown in figure (1), with periodic pressure loading. The head and lumen endure the alternating hydraulic pressure, as shown in figure (2), can be determine by equation [9] :

$$P = P_m + P_a \sin(k\omega t) \quad \ldots\ldots 1$$

To estimate the mean and alternating stresses, maximum and minimum forces must be firstly determined as follows :

$$F_{max} = P_{max} \cdot A_p \quad \ldots\ldots 2\text{-a}$$

$$F_{min} = P_{min} \cdot A_p \quad \ldots\ldots 2\text{-b}$$

$$F_m = \frac{F_{max} + F_{min}}{2} \quad \ldots\ldots 3\text{-a}$$

$$F_a = \frac{F_{max} - F_{min}}{2} \quad \ldots\ldots 3\text{-b}$$

Therefore, mean and alternating stresses are determined as follows :

$$\sigma_m = \frac{F_m}{A_c} \quad \text{...}4$$

$$\sigma_a = \frac{F_a}{A_c} \quad \text{...}5$$

According to Soderberg fatigue failure criteria, the safety factor can be estimated from following formula [10] :

$$\frac{\sigma_a}{\sigma_e} + \frac{\sigma_m}{\sigma_y / N} = 1 \quad \text{...}6$$

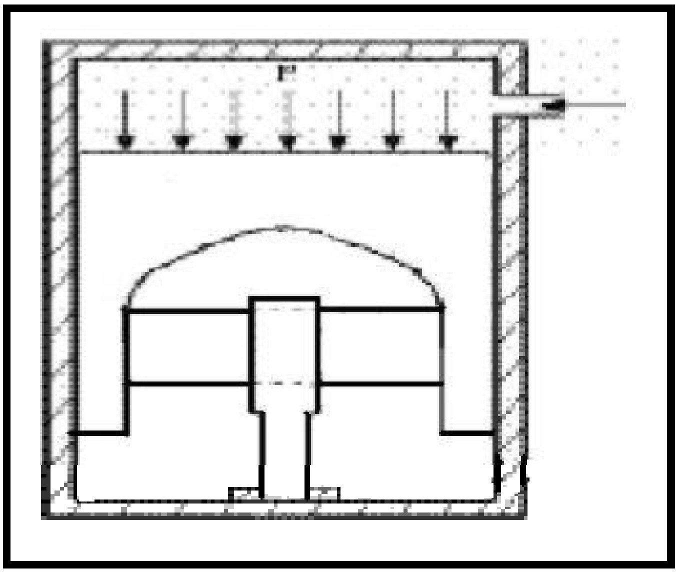


Figure (1): Schematic of pressure in chamber.

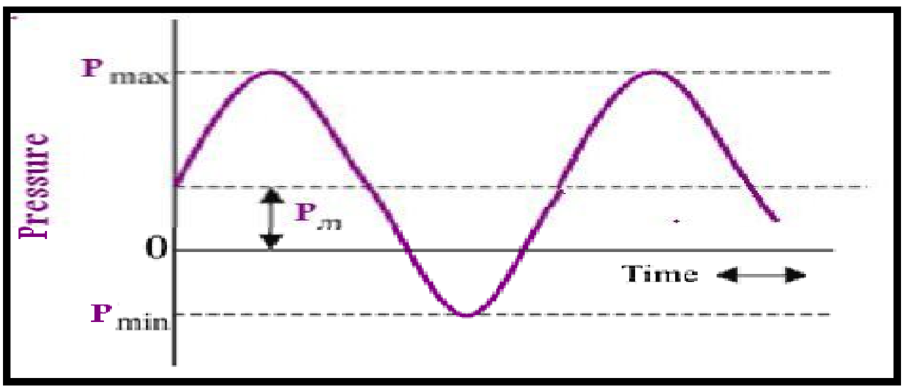


Figure (2): Layout of of pressure deviation from mean value.

3. Case Study:

For carrying out the fatigue analysis, it is necessary to determine the maximum load. As a piston reciprocates between top dead center (TDC) and bottom dead center (BDC), the rod was applied to experiences power loads and inertia loads. Power loads result from the expansion of burning gases during combustion that push down on the head of the piston and cause the crank to turn. Thus, power loads are always compressive in nature. This compressive force is equal to the area of the bore multiplied by the chamber pressure. In this study, for calculating the maximum load, the mean effective pressure of the cylinder and mechanical yield of the engine were assumed to be 0.8 and 0.7 MPa respectively. The maximum total load was thus found to be ***9500 N*** [7], then, only half of this force is apply on one half of upper connecting rod ring (***4750 N***). The dimensions of the present connecting rod under study is shown in figure (3).

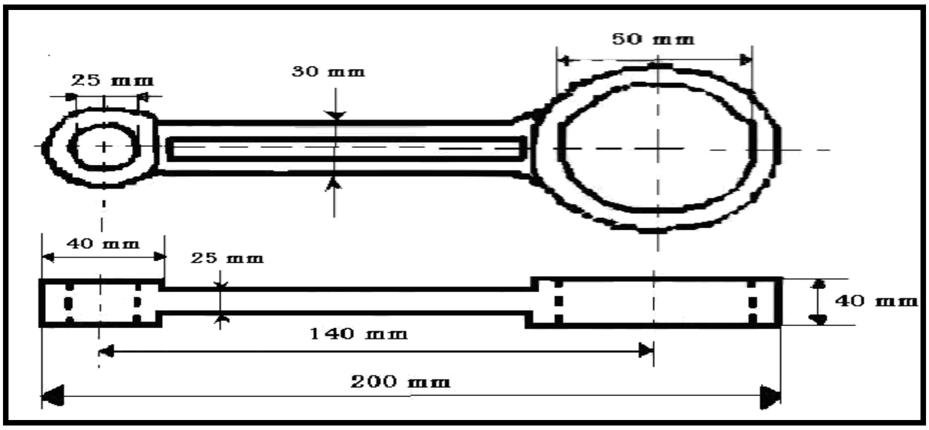


Figure (3): Dimensions of connecting rod.

The mechanical properties for Aluminum alloy, Titanium alloy and forged steel are available in ANSYS libratory and can be listed in Table (1), while physical properties, density and mass, of same dimensions of connecting rods (same volume) seen in figure (3) made from materials under study is listed in Table (2). The volume is estimated from following basic formula :

$m = \rho * V$ ……………………………………………………………………….7

Table (1): Mechanical properties for considering materials [11].

Material	Modulus of Elasticity (GPa)	Yield stress (MPa)	Ultimate stress (MPa)	Poisson's ratio
Aluminum alloy	71	280	310	0.33
Titanium alloy	96	930	1070	0.36
Forged steel	200	250	460	0.3

Table (2): Mass weight of connecting rods [11].

Material	Aluminum alloy	Titanium alloy	Forged steel
Density (kg/m^3)	2770	4620	7830
Mass (kg)	0.612	1.021	1.730

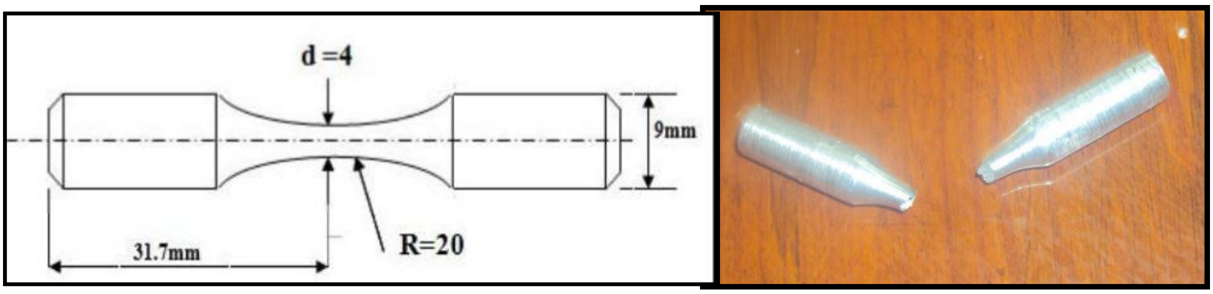
4. Experimental Part :

The experimental part in this study determines the aluminum alloy life during cycles using the specimen shown its dimensions and layout after breakage in figure (4). The fatigue test machine is seen in figure (5), where test is done in AL-Nahrain University / college of engineering – Mechanical Engineering department. The specimen in this test is rotating at $0.33 * 10^6$ cycle/hr while load applied at center. By taking d = 4 mm and L = 125.7 mm bending stress developed in material can be determined from using following formulas :

$$\sigma_b = \frac{M_b * y}{I} \quad \text{.........8}$$

$$\sigma_b = \frac{P * L * 32}{\Pi * d^3} \quad \text{.........9}$$

$$\sigma_b = 20\,P \quad (\text{Mpa}) \quad \text{.........10}$$



(a) (b)

Figure (4): Fatigue test specimen.
(a) specimen dimensions, (b) specimen after test.

Figure (5): Fatigue test machine.

Stress – Life curves for investigated aluminum alloy and the other two materials under study are collected in figure (6).

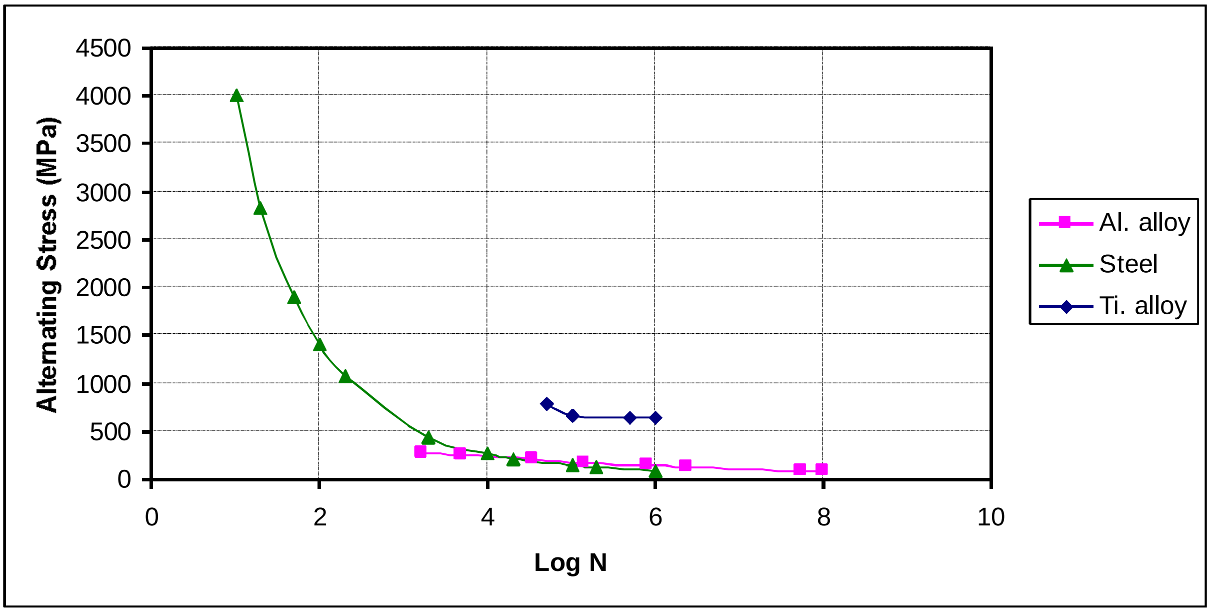


Figure (6): Stress – Life curves for considering materials.

The alternating stress – life relationships can be investigated and listed from previous chart using polynomial equations for Aluminum alloy, Titanium alloy and forged steel respectively as follows :

$$\sigma_a = 5.0373 (\text{Log } N)^2 - 96.924 (\text{Log } N) + 535.8 \quad \ldots\ldots 11$$

$$\sigma_a = 200 (\text{Log } N)^2 - 2235.4 (\text{Log } N) + 6857.5 \quad \ldots\ldots 12$$

$$\sigma_a = 262.27 (\text{Log } N)^2 - 2437.9 (\text{Log } N) + 5595 \quad \ldots\ldots 13$$

The investigated experimental lives and factors of safety are listed in table (3). The lives for three materials under study are determined directly from figure (6). Factors of safety are computed from equation (6) by substitution the endurance limit for steel of 230 Mpa and fatigue strengths of 135.6 and 654.1 Mpa for aluminum and titanium alloys respectively which already determined from figure (3) at 10^6 cycles; while other stresses in equation (6) are taken from figure (6), table (1) and case study loading.

Table (3): Experimental results for duration and safety.

Material	Aluminum alloy	Titanium alloy	Forged steel
Life (cycle)	10^8	10^6	10^6
Factor of safety	1.820	1.328	1.125

5. Numerical Investigation:

The numerical analysis of connecting rod, shown its dimensions in figure (3), by using *ANSYS – WORKBENCH PACKAGE V.11* which carried out through applying cyclic compressive fatigue load of 9500 N on lower end of rod (at crank shaft). The supported of connecting rod is in two ends, fixed upper end (at piston joint) and two cylindrical support at two screws that fastening two half rings around crank shaft on lower end as shown in figure (7).

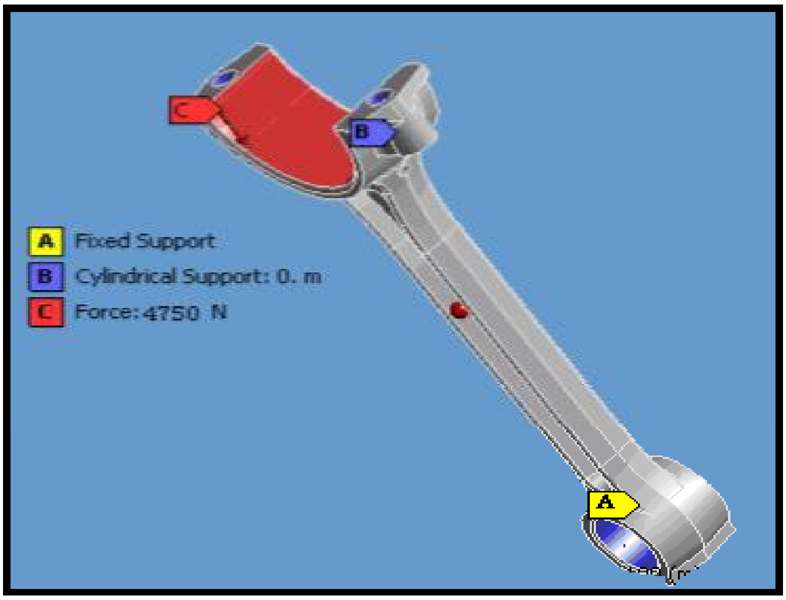


Figure (7): Loading and supports of connecting rod.

The Von – Misses stress, deformation, life and safety factor for aluminum alloy are shown in figures (8), (9), (10), and (11) respectively. The factors of safety for Titanium alloy and forged steel are seen in figures (12) and (13) respectively.

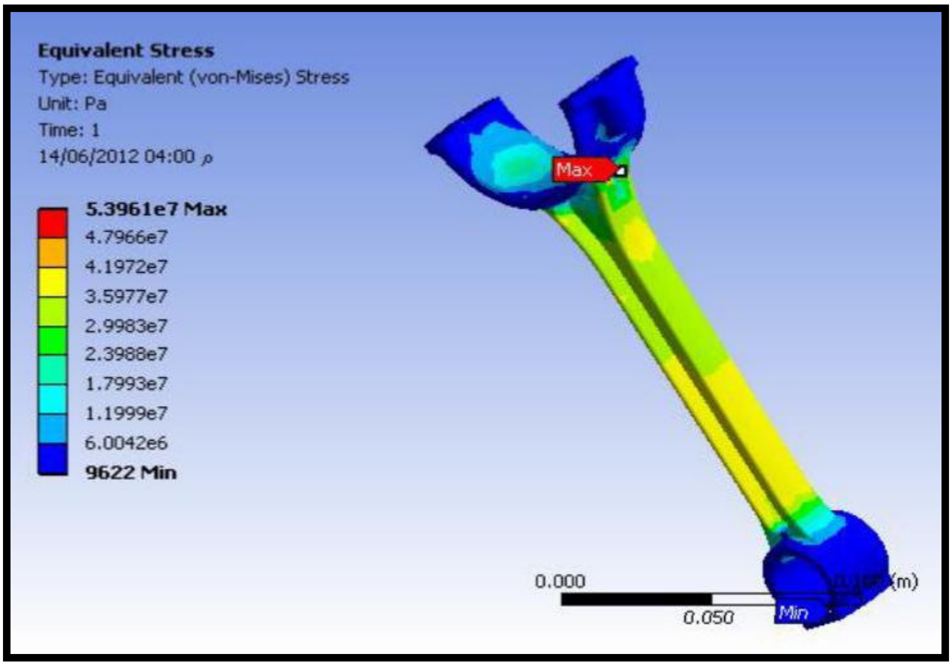


Figure (8): Von – Misses stress for Aluminum Alloy in (Pa).

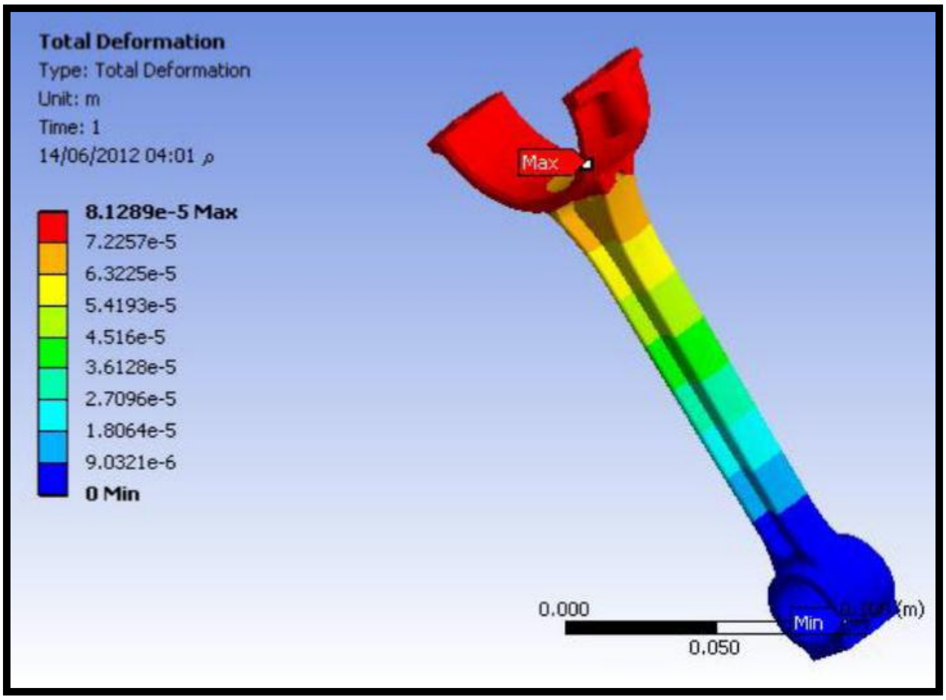


Figure (9): Deformation for Aluminum Alloy in (m).

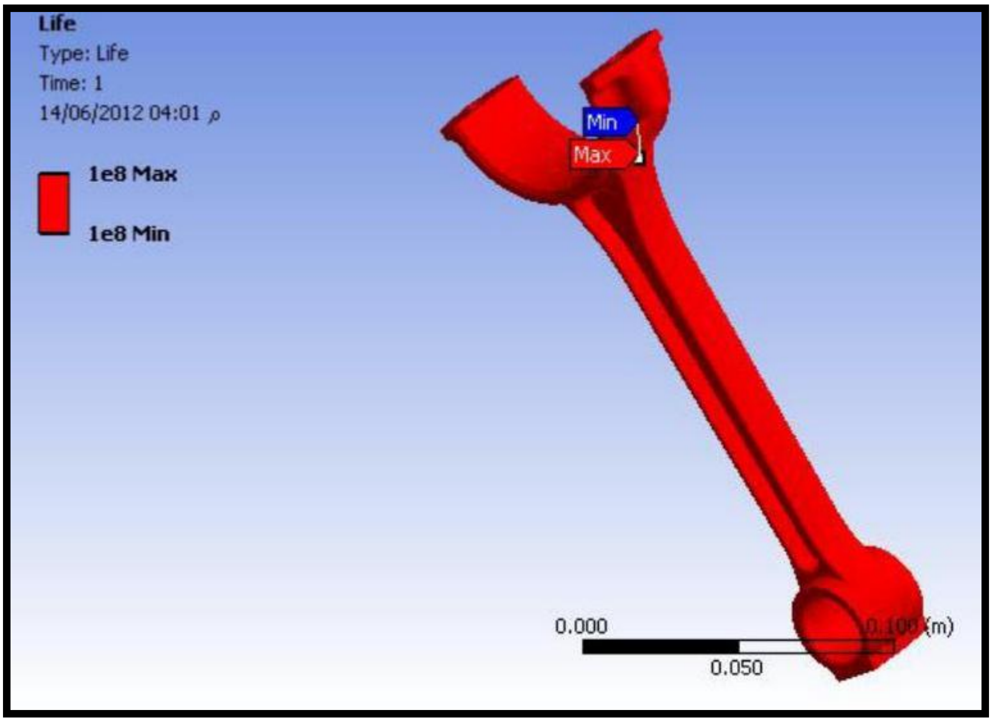


Figure (10): Life for Aluminum Alloy in (cycles).

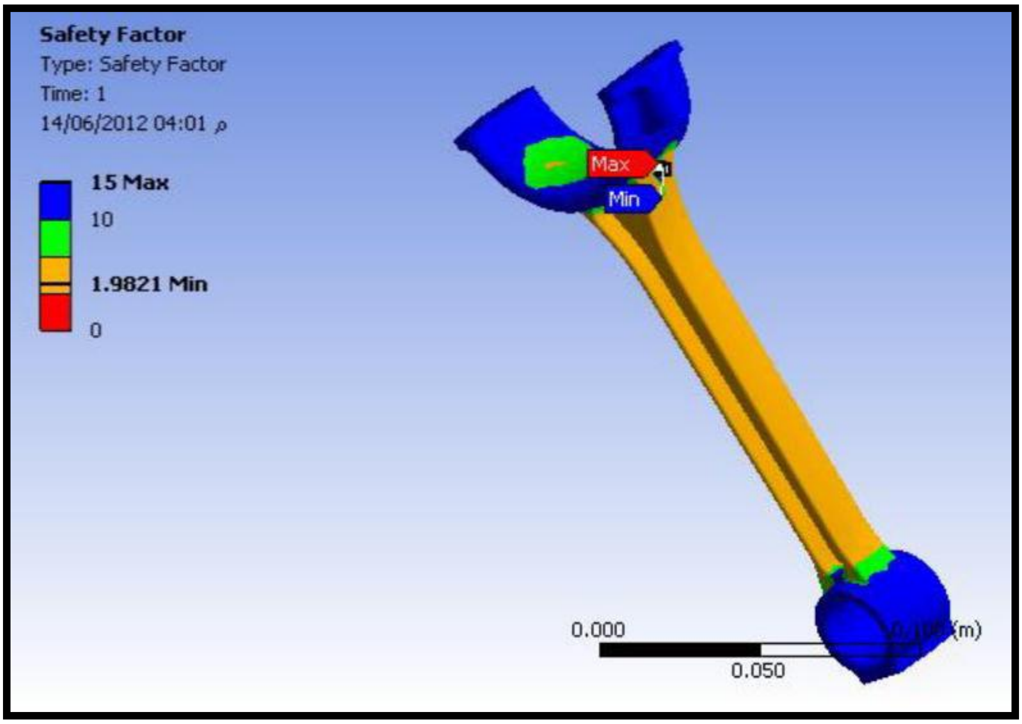


Figure (11): Factor of safety for Aluminum Alloy.

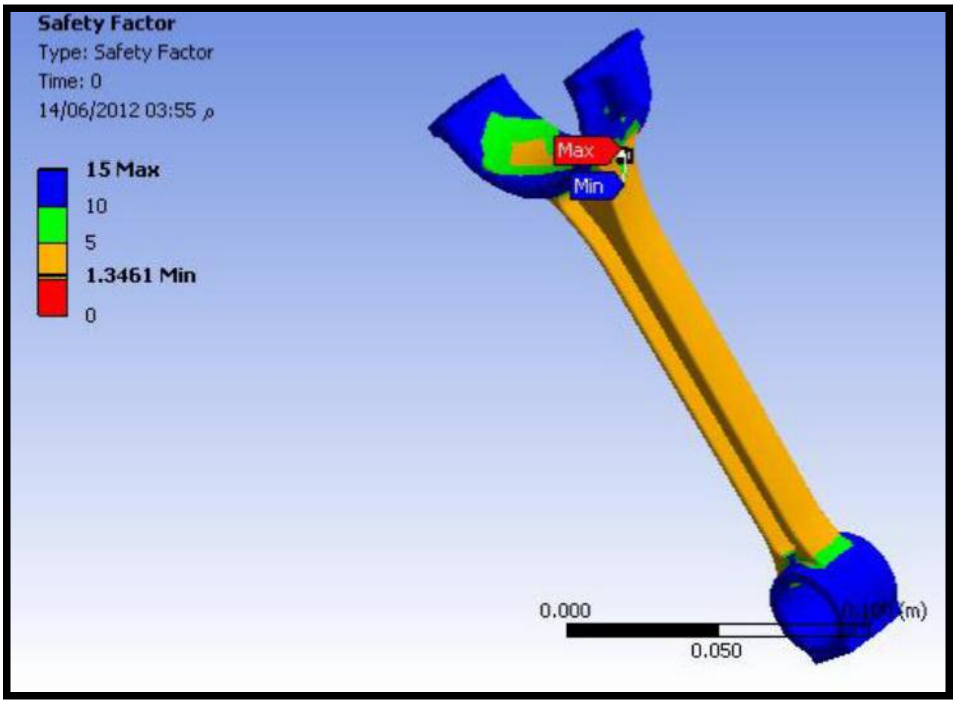


Figure (12): Factor of safety for Titanium Alloy.

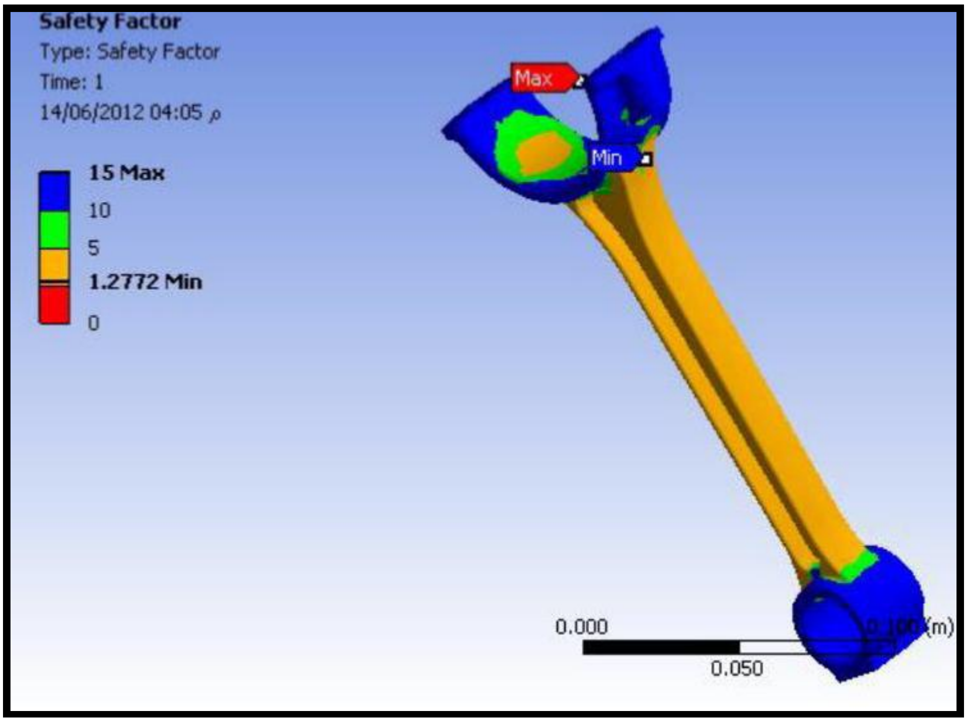


Figure (13): Factor of safety for forged steel.

All investigated fatigue items for three materials under consideration in current study for numerical results can be collected in Table (4).

Table (4): Fatigue items for Aluminum alloy, Titanium alloy, and Forged steel

Material	Von – Misses stress (MPa)		Deformation (mm)		Life (cycles)	Safety factor	
	Max.	Min.	Max.	Min.		Max.	Min.
Aluminum Alloy	53.96	0.0096	0.0813	0	10^8	15	1.9821
Titanium Alloy	51.23	0.0078	0.0270	0	10^6	15	1.3461
Forged Steel	53.99	0.0081	0.0289	0	10^6	15	1.2772

6. Discussion :

For automotive internal combustion engines, the connecting rods are most usually made of steel for engines production, but can be made of aluminium (for lightness and the ability to absorb high impact at the expense of durability) or, titanium (for a combination of strength and lightness at the expense of affordability) for high performance engines or, Steel (for daily drivers and endurance racing due to their high strength and long fatigue life) for most widely produced and used type of connecting rods.

In present work, the connecting rod is made from previous mention three materials and loaded under fatigue loading. The connecting rod's fatigue stress, deformation, safety factor ….etc for three materials under study are investigated, and calculated in the base of its fatigue performance and then pointed the best material for manufacturing the connecting rod in diesel engines.

Figure (7) is shows the regions of supporting of connected rod , in addition to loading caused by compression force developed by piston 9500 N and transition to crank shaft and divided to two halves of ring of 4750 N.

The figures (8) to (13) are shows that critical region in the connecting rod under fatigue loading is located at the lower end of shank near half ring around crank shaft. The main reason for developing higher stresses near position of crank shaft can be simply assumed that connecting rod is cantilever beam under compression load at its free end (piston in present model) and then failure was occurred first at its fixed support (half ring in present model) which represent the weakest section.

Table (4) is illustrated the all fatigue items for three materials under study. The maximum Von – Misses stress for Aluminium alloy and Forged steel is about **54 MPa** reduced about ***5.11 %*** to become **51.23 MPa** for Titanium alloy. The maximum deformation is located at lower end of shank and equal to **0.0289 mm** for forged steel reduced slightly about **6.57 %** for Titanium alloy while major increment is detected for Aluminium alloy to about ***281 %*** more than steel to become **0.0813 mm**. The fatigue life of steel and Titanium alloy is $\mathbf{10^6}$ cycles, while high increase in duration life of connected rod is investigated from using Aluminium alloy to be $\mathbf{10^8}$. The experimental result of life for Aluminium alloy is also $\mathbf{10^8}$ (as

clearly shown in figure (6)) , therefore in present work the numerical and experimental results for life were identical.
Minimum safety factor is investigated to be maximize for Aluminium alloy among other materials to **1.9821** near and close to crank half ring, while this value reduce to **1.3461** and **1.2772** for Titanium alloy and steel materials respectively. The experiential results of factors of safety are **1.82**, **1.328**, and **1.185** for Aluminum alloy, Titanium alloy and Forged steel respectively as listed in table (3), with percentage difference than against values calculated from numerical investigation **8.18 %**, **1.34 %**, and **7.22 %** respectively.
In general, experimental factors of safety are lower than against numerical values for all materials under study, because numerical investigation don't taken into consideration defects may be exist in surface and micro structure of connecting rod, therefore lower safety is obtained from experimental results.

7. Conclusions:

The major points can be concluded from present work are being listed in following two points :

1. Experimental and numerical results for aluminium alloy life was $\mathbf{10^8}$.
2. The Aluminium alloy is the best material among other materials which used to manufacture connecting rod in internal combustion engines, due to several points can be listed as follows :
   a) Higher life with **99 %** percentage increment than both Titanium and materials.
   b) b) Higher safety factor with **32.1 %** and **35.6 %** percentage increment than Titanium alloy and steel materials respectively based on numerical investigation, while these increment become **27 %** and **38.2 %** for against experimental investigation.
   c) Identical magnitude for Von – Misses stress with steel while slightly increase of only **5.11 %** in comparison with Titanium alloy.

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