Fatigue Analysis In Aircraft Landing Gear Axle Shaft To Develop The Life Cycles

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Abstract

Aircraft landing gear failure is a high concern in the aviation industries. According to the Federal Aviation Administration reports, 55% of aircraft failures occur during takeoff and landing while 45% of failures occur during flight. A mechanical system, such as aircraft landing gear, can have a large number of parts that interact in a complex nonlinear fashion. This project deals with increasing structural strength of the landing gear axle shaft such as stress and also reducing the fatigue and failure. Objective is to analyze main landing gear axle shaft to determine the fatigue stress behavior and the displacement of an aircraft landing gear axle during taxing in the ground. The modified design of the landing gear axle shaft has made by using CATIA V5 software and selection of material has been very important. Thus the result is compared with various materials in which titanium alloy has high strength and also increase in fatigue life cycles.
Keywords— *Landing gear, Axle shaft, Von-mises stress, Fatigue cycles, Safety factor*

1. Introduction

This project done for the Master of Aeronautical Engineering to the college of Anna University. The landing gear is the structure that supports an aircraft and allows it to move across the ground or water. The failure of a landing gear on a Boeing 737-400 registered “PK-GZN”, which suffered from a broken axle on the left main gear, was used in an accident analysis, leading to a modification recommendation that can be applied on Boeing 737-400 aircrafts. The purpose of this report is to be analysis of landing gear axle shaft and submit the findings of an investigation into landing gear static load analysis. The process is three phases, there are the first is static structural analysis of axle, second one is fatigue and fracture analysis and third is cost aspects. The material included within the submissions includes an overview of the landing gear axle shaft and the general equations pertinent to these designs, an in depth technical investigation of the landing gear failure Boeing PK-GZN 737-400 aircraft. Selection of materials is an important because some components have to stand high forces, the materials has to be appropriate to the operational forces. High forces will be absorbed by the landing gear axle shaft components. After the shock absorbing the deceleration starts, this is done by the use of brakes. Also some systems are related to the deceleration, like auto brakes and anti-skid. When the aircraft is decelerated enough the aircraft has to be able to taxi. During taxiing different components and systems are used to move the aircraft on the ground. In this project to analyze the fracture in the bogie because the left hand side of the left main gear bogie was broken during taxiing, the bending moment during that action will be calculated. The next product to solve the maximum tension equation is the moment of inertia. The maximum tension in the bogie during taxiing can be calculated and will be compared to the maximum tension of the bogie material In case of a failure.

2. Landing Gear Basics

According to the basics of the landing gear is study about the purposes and constructions. The main purpose of the landing gear in a modern aircraft is to absorb the shock from the landing, brake the aircraft and to manoeuvre the aircraft on the ground. At touchdown all the weight of the aircraft rests on the landing gear. The landing gear needs to resist the shock and absorb it so the passengers and structure of the aircraft are spared as much as possible. The brakes, attached to the landing gear and the rims slow down the aircraft after touchdown. When the aircraft has stopped it can taxi to the gate or platform by using the steerable nose gear. The construction of The 737-400 has a modern jetliners standard tricycle configuration. The tricycle gear is the most common landing gear used in modern aviation. It is a three point gear with a nose wheel in the front and two main landing gears positioned slightly behind the centre of gravity **(1, 3)** of the aircraft (figure 1). The centre of gravity is positioned well in front of the main gear when evenly loaded. The main gears are positioned in a way that gives the most stable ground position possible. Because the main gears are 5.23 meters separated from each other they are mounted on the wings instead of the fuselage. The two main gears and

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nose gear feature each two wheels. The two main landing gears are evenly positioned from the longitudinal and horizontal axis of the aircraft.

1. Centre of Gravity
2. Angle greater than wing stalling angle
3. 6°-20
4. 40+°

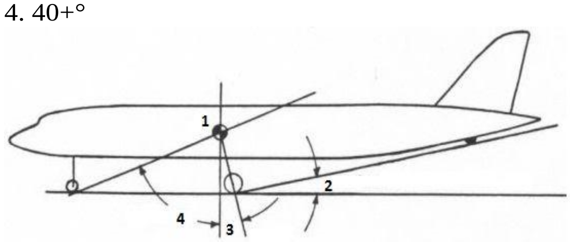

Figure 1: Tricycle gear

An angle greater than the wing stalling angle is given to the back of the aircraft to prevent a tail slide during landing or takeoff **(2)**. To keep optimum stability and manoeuvrability the angle between the nose gear and centre of gravity must be at least 40° **(4)**. The main advantage of the tricycle gear over the conventional gear, a landing gear with a tail wheel instead of a nose wheel, is that it counteracts a ground loop. Since the centre of gravity is now positioned in front of the main landing gear instead of behind, the aircraft has the natural tendency to keep straight on the runway. Furthermore the tricycle gear is easier to steer and taxi and gives the pilots a better view of the runway. The aircraft has a horizontal position giving the thrust of the engines a better angle of attack and the cabin more accessible. Like any configuration the tricycle gear comes with some disadvantages as well. The biggest drawback is the weight of the strong nose wheel needed in this configuration. Another critical factor is balancing the aircraft, loading to much cargo in the rear can cause the aircraft to tip back.

3. Calculation of Force

To calculate the forces in the left main gear axle, the forces on the landing gear during taxing. This result is used to calculate the forces in the axle during the failure. In ground phase is assumed the aircraft is taxiing with an fixed speed and no crosswind component. The entire weight of the aircraft engages in the Centre of Gravity (CG). The Boeing 737-400 has a maximum landing weight of 56245 kilograms. This weight is spread over two main gear struts and one nose gear strut. We can use an equilibrium equation the weight on each strut can be calculated. While taxiing there are no external forces on the aircraft. The wing generates no lift and the air resistance is negligible.

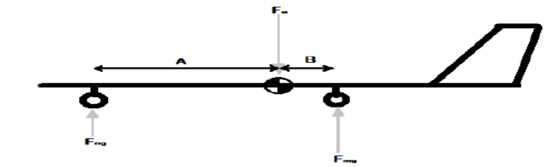

Figure 2: Distance and loads

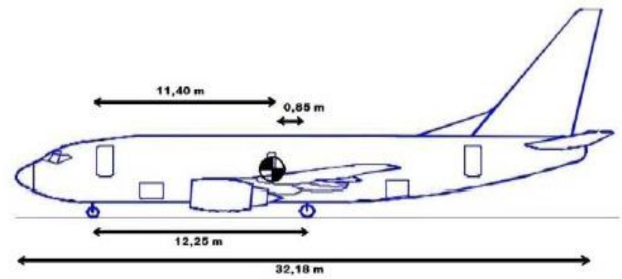

Figure 3: Distances between CG to Landing Gears

Formula 1

Fa A – (A + B) * Fmg = 0

11.40*551763.45 – (11.40 + 0.85) ∗ Fmg = 0

Fmg = 513475.02 Newton

Formula 2

Fng = Fa – Fmg

Fng = 551763.45 – 513475.02

Fng = 38288.43 N

Where,

A,B – Distance (m)

Fa – mass of the aircraft (N)

Fng – force on the nose gear (N)

Fmg – force on the main gear (N)

A main landing gear force 513475.02N as using to determining the stress behavior and fatigue life cycles of the landing gear axle. Because the main landing gear axle fracture happened before its life cycles during taxing phase. After that will discussion about the modification possibilities.

4. Dimension of The Axle Shaft

As this part of the axle is also seen on the photographs of the broken axle, it can be used as a reference for the scale. This minimal diameter of 3.370 inch, is used in the calculations as the broken axle and its sleeves have suffered at least some wearing during their period of use. By using a measuring tool on the photograph of the broken axle in software called "the GIMP", the on-screen diameter of the axle sleeve is translated to millimetres. It shows that the on-screen diameter of the axle sleeve is 12.6 millimetre. Finally calculated dimensions are following,

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Scaled dimensions of the axle shaft:

Base of the T-joint	:	173 mm
T-joint to brake flange	:	160 mm
Axle length	:	493 mm
Inner dia meter	:	58.34 mm
Outer dia meter	:	96.14 mm

4.1 Force Distribution

The force on the main gear strut that was present during then breaking of the axle in the PK-GZN has been calculated above tasks. But can be further distributed so the forces on the axle itself are shown. These forces can be drawn in a Free Body Diagram (figure 4). The aircraft was taxiing when the failure occurred, so it can be assumed that the shock strut and the forces acting on the bogie were in balance and their total addition meets zero (formula 3)

Formula 3

$$\sum Fy = 0 = -513500 + 2 \times Fnt$$
$$Fnt = 256750$$

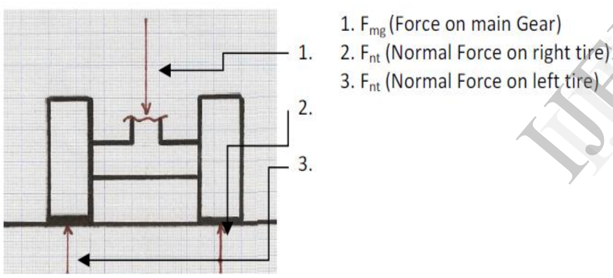

Figure 4: Vertical Forces Acting on The Landing Gear

4.2 Bending Moment

To calculate the bending moment, information on the left main landing gear is required. The bending moment is needed to solve the 'tension equation'. A Cutline in the FBD is indicated by the name A-A, the cutline is required to apply the 'method of sections'. The cutline is exactly at the middle between the left and middle force, at that point the bending moment has an average value.

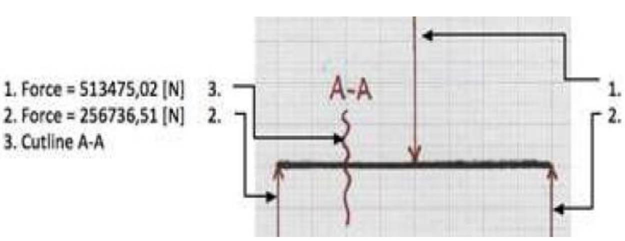

Figure 5: Divid the section

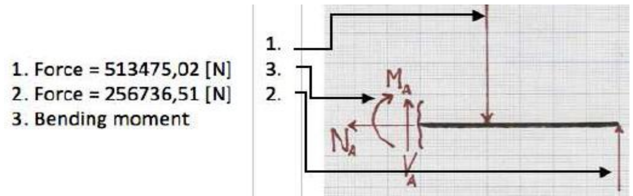

Figure 6: Bending Moment

Fomula 4

$$\sum Ma = 0$$
$$Ma + 513475.02 * 0.123 - 256736.51 * 0.3698 = 0$$
$$Ma = -63311.47 + 94941.16$$
$$Ma = 31630.06 \text{ N-m}$$

4.2 Moment of Inertia

The moment of inertia required to sole the maximum bending equation. The cross section of the circular shaft is given to take the dimension and solve the maximum tension of the axle shaft.

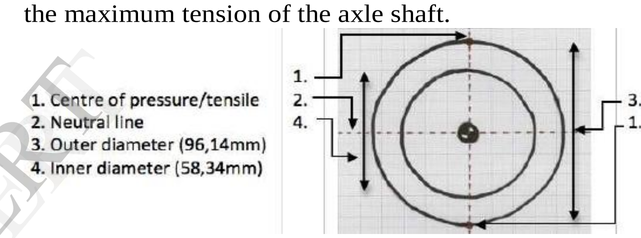

Figure 7: Cross Section of The Cut A-A

Formula 5

$$I = \frac{\pi}{64}(D^4 - d^4)$$
$$= \frac{\pi}{64}(0.09614^4 - 0.05834^4)$$
$$= 3.624956 * 10^{-06}\ m^4$$

4.3 Maximum Tension

The maximum tension determined by using the following bending equation relationship. The above criteria's are using to solve this equation and the distance calculated from the neutral axis is 0.04807 m.

Formula 6

$$\frac{M}{I} = \frac{\sigma}{y}$$

Where,

M – Bending moment (Nm)

I - Moment of inertia (m^4)

σ – Bending stress (N/m^2)

y - Distance from the neutral axis (m)

$$\frac{31630.06}{3.624956 * 10^{-06}} = \frac{\sigma}{0.04807}$$
$$= 419.44 * 10^{06} \text{ Pa}$$
$$\sigma = 419.44\text{MPa}$$

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This is the maximum tension of the axle bogie. The axle fracture happened due the fatigue. So the determined maximum stress compare to the yield strength of the material. The current axle material AISI 4340 steel has a710Mpa. Generally choosing the material should be having the yield strength above the working stress. Then only the component should able to withstand the given load. Therefore here the axle material has the yield strength above the working stress. So this is did not cause for the fatigue. Another reason other than just force for the fracture could be fatigue in the material, corrosion in relation with fatigue is also a possible reason. Therefore investigation into better resistant corrosion and fatigue properties of the material is recommended.

5 Concept

To prevent an accident like with the PK-GZN, a modification to the used material is possible. An option is to use a stronger, non- ductile and more fracture resistant material. And additionally to change the inner design, it means applying the composite beam methods reference from 'Strength of Materials'. This method is defined as a beam made up of two or more different materials assumed to be rigidly connected together and behaving like a single piece is known as a composite beam or a flitched beam. Dimensions of the modified design

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5.1 Modified Dimensions

The flitched beam consists of titanium alloy and steel alloy. The steel covered by the titanium bottom top curved surfaces. The area of the hollow shaft is equal to the actual dimensions of the axle. That means the modified dimensions matches exist area of the axle. The figure 8 shows the composite structure of axle shaft and it contains 30% of titanium alloy and 70% of the steel alloy. Finally calculated modified dimensions are following,

Base of the T-joint	:	173 mm
T-joint to brake flange	:	160 mm
Axle length	:	493 mm
Titanium alloy		
Inner ring thickness	:	3 mm
Outer ring thickness	:	3 mm
Steel alloy		
Inner dia meter	:	64.34 mm
Outer dia meter	:	90.14 mm

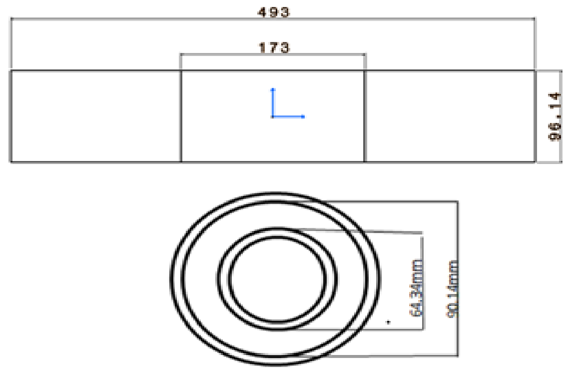


Figure 8: Preliminary Design of Modified Axle

Properties	AISI 4340		Titanium alloy	
Stress in yield	710	MPa	880	MPa
Young`s modulus	205	GPa	113	GPa
Strain	0.54	%	1.13	%
Poisson ratio	0.29	-	0.342	-
Density	7860	Kg/m^3	4650	Kg/m^3
Melting temperature	1400	°C	1649	°C

Shaft

5.2 Material Selection

The 4340 steel is a heat treatable, low alloy steel containing nickel, chromium and molybdenum. This steel is known for its toughness and capability of sustaining high strength in the heat treated condition while retaining good fatigue strength. And the typical applications are, power transmission gears, shafts and aircraft landing gears. Then titanium alloy has a lot of advantages: it is strong, light and corrosion prove. Titanium alloys are often used in aircrafts because the material is resistant against high temperature and titanium is as strong as steel. Titanium is often used in combination with aluminium, vanadium, molybdenum or chrome. Properties of the alloys are follows,

Table 1: Properties of Materials

6. Calculation of Composite Design

In this composite structure consist of two materials, they are titanium and steel.

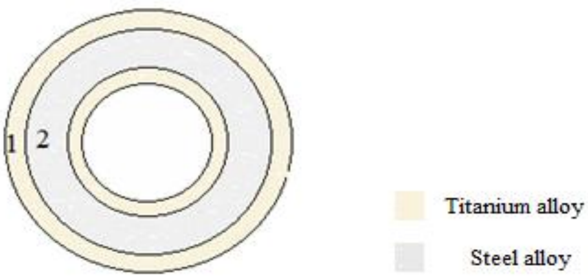


Figure 9: Flitches Attached Inner And Outer Of The Axle Shaft

Let suffix 1 reprsents the titanium alloy and suffix 2 represents steel alloy.

Max. distance from N-A Ti y_1 = 48.07 mm
Max. distance from N-A steel y_2 = 45.07 mm
Steel alloy
Inner dia meter (d_2) = 64.34 mm
Outer dia meter(D_2) = 90.14 mm

Number of titanium rings = 2

Stresses of the titanium and steel alloys are σ_1 and σ_2.

In previous chapters already determined the stress and moment of steel alloy. So in this way using to find the stresses in composite structure.

Moment of inertia of the titanium alloy I_2

$$= [\frac{\pi}{64}(D_2^4 - d_2^4)]$$

Here, D_2 = External diameter (m)
d_2 = Internal diameter (m)

I_2 = 2.34 $*10^{-06} m^4$

And we are already determined above the bending moment of steel alloy, M_2 = 31630.06 N-m

Then using the bending equation to find the tension in steel alloy,

$$\frac{M_2}{I_2} = \frac{\sigma_2}{y_2}$$

$$\frac{31630.06}{2.34 * 10^{-06}} = \frac{\sigma_2}{0.04507}$$

$$= 609*10^{06} \text{ N}/m^2$$

$$\sigma_2 = 609 \text{ MPa}$$

Now, using the strain relation to determine tension of the titanium alloy. The strain at a common distance of 45.07 mm from neutral axis is steel and titanium would be same. Hence using equation get,

$$\frac{\sigma_1}{E_1} = \frac{\sigma_2}{E_2}$$

$$= \frac{113}{205} * 609$$

$$\sigma_2 = 335 \text{ MPa}$$

But σ_2 is the tension in titanium alloy at a distance 45.07mm from neutral axis. Maximum tension σ^* calculated in 48.07 from the neutral axis. As tensions are proportional to the distance from the N.A.

$$\frac{\sigma_1}{0.04507} = \frac{\sigma^*}{0.04807}$$

$$\sigma^* = 357 \text{ MPa}$$

Finally calculated the tension in the flitched structure is the 357 MPa. In this using to calculate the fatigue life cycles and compare to the currently used material of AISI 4340 steel.

7. Fatigue Analysis

Fatigue is the structural damage occurs when material subjected to cyclic loading. An above the yield strength, a microscopic cracks will begin to form at the stress concentration as surfaces. There are two type of the fatigue is high fatigue and lower fatigue.

High fatigue is the lower stress acting in a longer time. Fatigue strength about 10^{03} to 10^{07} cycles. And lower fatigue is the higher load acting in a short time limit. Cycles about less than the 10^{03} The stress life method is the classical method for fatigue analysis of metals and has its origins in the work of Wohler in 1850. Stresses in the structure or component are compared to the fatigue limit of the material. The basis of the method is the materials S-N curve which is obtained by testing small laboratory specimens until failure.

In these tests have been conducted in rotating bending. Today, it is often common to find test data for axial loading as well.

$$S_a = \frac{\Delta S}{2} = 1.62\, S_u (N_f)^{-0.085}$$

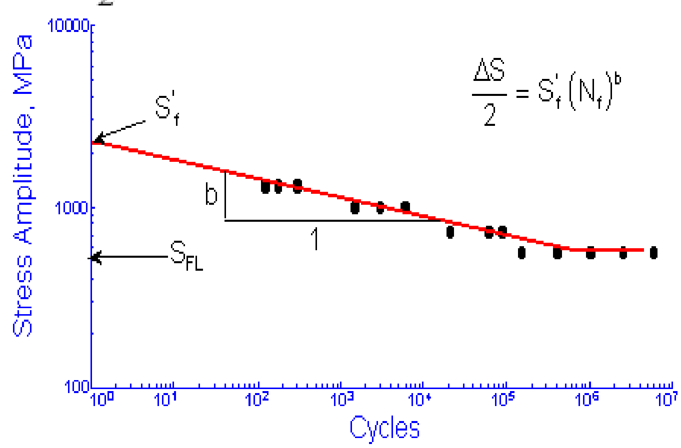


Figure 10: S-N Curve for Steel alloy

The fatigue limit, S_{FL}, is the stress below which failures do not occur in the materials. Wöhler called this a safe stress level for design. Today we know that failures will occur below the safe stress level but it will take a very large number of cycles, longer than the 10^6 or 10^7 cycles used in normal fatigue testing. The finite life portion of the curve is fit to a power function relating the stress amplitude, $\Delta S/2$, and fatigue life in cycles, N_f.

The fatigue limit is approximated as one half of the tensile strength. $S_{FL} = 0.5 S_u$. It has been observed that the fatigue strength at 1000 cycles is approximately 0.9 S_u. This gives two points on the SN curve, both in terms of hardness that can be used to estimate the entire SN curve.

7.1 Fatigue Life Cycles Current Axle Shaft

In current fractured PK-GZN axle shaft as made by the AISI 4340 steel. To determine the alternating stress of the material by using the following equation,

$$S_a = 1.62\, S_u (N_f)^{-0.085}$$

Where,
S_a – Alternating stress

S_u – Ultimate stress

N_f – Number of cycles

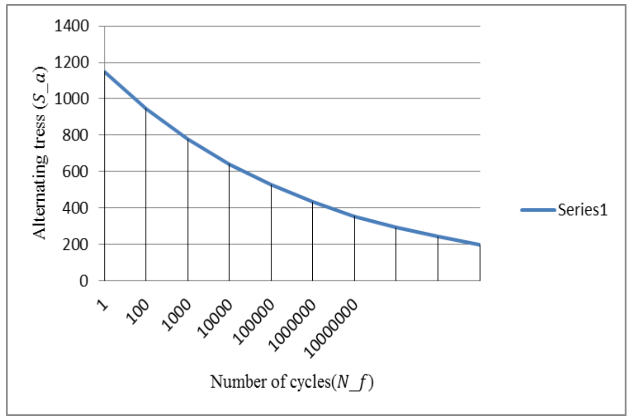
The ultimate stress take to the yields strength of the steel alloy, S_u = 710 MPa. And the working stress is σ = 419 MPa

Ultimate stress S_u = 710 MPa
N = 1 S_1 = $1.62 * 710\,(1)^{-0.085}$
= 1150 MPa

Similarly,

S.NO	Number of Cycles (N_f)	Alternating stress (S_a) MPa
1	1	1150
2	10	946
3	100	778
4	1000	640
5	1E+04	526
6	1E+05	432
7	1E+06	355
8	1E+07	292
9	1E+08	240
10	1E+09	197

Table 2: Alternating Stress For AISI 4340 Steel



Graph 1: S-N Curve for Steel Alloy

The above graph shows that the fatigue life cycles between 10^{05} to 10^{06}. In this life cycles evaluated to ten years. But the PK-GZN axle shaft failure happened before these cycles due to the corrosion. Therefore the composite structure used to avoid the unwanted fatigue failure of the axle shaft.

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7.2 Fatigue life cycles for flitched structure

Composite structure contains the titanium and steel alloy. Titanium alloy good fatigue resistance compare to the steel alloy. In steel alloy covered by the titanium alloy so the ultimate strength taken to the yield strength of the titanium alloy, S_u = 880 MPa. And the working stress is σ = 357 MPa.

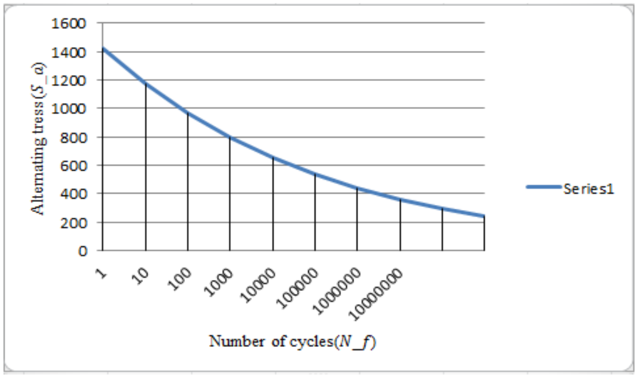
Ultimate stress S_u = 880 MPa
N = 1 S_1 = $1.62 * 880\,(1)^{-0.085}$
= 1426 MPa

Similarly,

S.NO	Number of Cycles (N_f)	Alternating stress (S_a) MPa
1	1	1426
2	10	1173
3	100	964
4	1000	793
5	1E+04	652
6	1E+05	536
7	1E+06	441
8	1E+07	362

9	1E+08	298
10	1E+09	245

Table 3: Alternating Stress For ASTM 5Grade Titanium



Graph 2: S-N Curve for Composite Structure

In flitched structure has a fatigue life cycles between 10^{06} .to 10^{07} .it is much greater than compare to the actual structure.

8. Safety factor

The safety factor is how much you want to underestimate the maximum strength of the materials in order to ensure a safe design. A single safety factor is applied to both the stress amplitude and the mean stress.

$$\frac{S_a}{S_{FL}} + \frac{S_{mean}}{S_u} = \frac{1}{n}$$

This can be graphically shown in the Goodman Diagram.

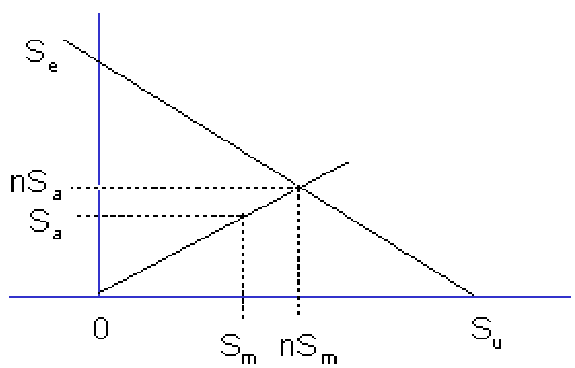


Figure 12: Goodman Diagram

The alternating stress is plotted on one axis and the mean stress on the other. The allowable alternating stress with no mean stress is the fatigue limit. The maximum mean stress, with zero alternating stress, is the ultimate strength. A straight line is then drawn between the two points. Any combination of mean and alternating stress on this line will have the same fatigue life.

Safety factor $= \frac{S_a}{S_{FL}} + \frac{S_{mean}}{S_u}$

Where,

S_a - Alternating stress

S_{FL} - Fatigue limit

S_{mean} - Mean stress

S_u - Ultimate stress

$$S_a = \frac{S_{max} - S_{min}}{2} = \frac{1426-362}{2}$$

= 532 MPa

$$S_{mean} = \frac{S_{max} + S_{min}}{2}$$

= 894 MPa

S_{FL} $= 0.9 S_u$

= 792 MPa

Safety factor $= \frac{532}{792} + \frac{894}{880}$

= 1.69

Similarly, fractured axle safety factor calculated to 1.65.

Remember, as the factor of safety increases, the cost of the product also increases.

9. Result and discussion

The failure of the left main landing gear of a Boeing 737-400 has been analyzed in this report. All the functions, systems, subsystems and requirements of a 737-400 main landing gear were investigated. Using this knowledge, the forces and tensions on the landing gear and axle were calculated. Eventually a modification recommendation for the existing axle was made. the regular axle of the Boeing 737-400 is sensitive for metal fatigue, which was the reason for the failure on the PK-GZN. The best replacement for this material is titanium alloy ASTM Grade 5 which is stronger but also more expensive. But the fatigue resistance two times compared to the steel alloy. The comparison of results should be followed,

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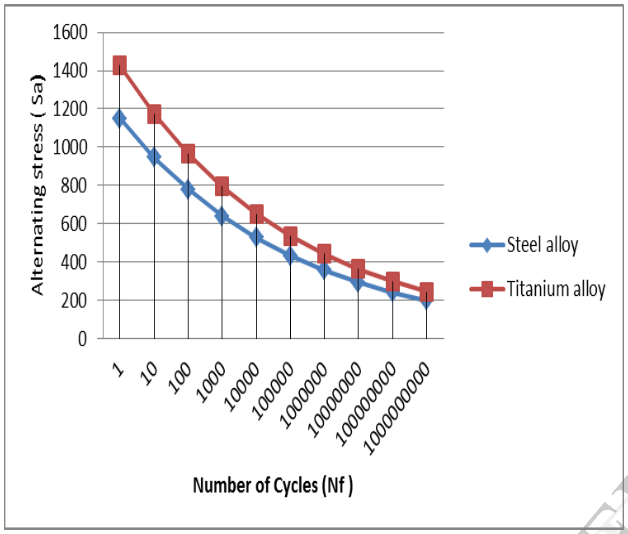
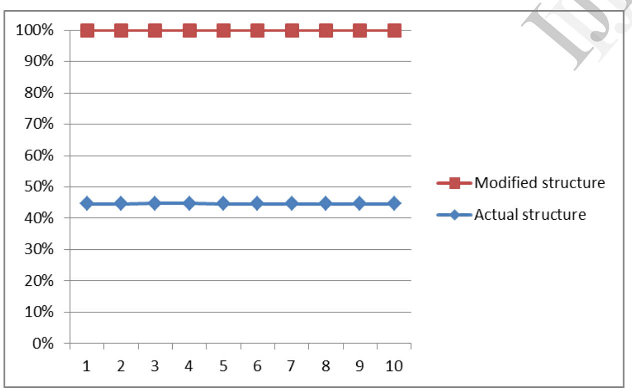
Axle shaft	Fatigue life cycles(N_f)	Safety factor(n)
Actual structure	$1.42*10^{05}$	1.65
Flitched structure	$1.2*10^{07}$	1.69

Table 4: Fatigue cycles and safety factor



Graph 3: Compaision of S-N curve



Graph 4: Comparison of percentage of life

Therefore, the above results show that the modified design very much better than the regular axle shaft. The fatigue cycles are increased to two times greater than the current axle. Then the costs are calculated for the design, certification and the production of the new axle. The development costs and modifying the axle material are estimated at €35 000. But the safety factor is also greater than current alloy therefore cost of the new model compromised by its greater life cycles. So safety is thereby improved.

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