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STRESS ANALYSIS OF SINGLE POINT FIXINGS

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Abstract

The paper presents a methodology for assessing the structural integrity of Single Point Fixing (SPF's) used on a pilot control stick. The main aim is to provide a thorough process by which these SPF's and the adjoining structure can be assessed for any Active Inceptor System. In this, all the relevant processes are identified including all the possible failure mechanisms.

Keywords

Single Point Fixing, Stress, Active Inceptor System

1. Introduction

BAE SYSTEMS is an international company mainly engaged in the development, delivery, and support of advanced defence and aerospace systems in the air, on land, at sea and in space. The company designs, manufactures, and supports military aircraft, surface ships, submarines, radar, avionics, communications, electronics, and also guided weapon systems. It is a pioneer in technology with a heritage stretching from hundreds of years and is at the forefront of innovation, working to develop the next generation of intelligent defence systems.

1.1 AIS Technology

BAE Systems is engaged in the development of the first ever Active Inceptor System (AIS) for a rotary winged aircraft. The system will replace the orthodox mechanical control system. The AIS is mainly an electro-mechanical comprising of mechanisms, bearings, servo-actuators, etc. AIS provide an interface between the pilot with modernized intuitive tactile cues for easier handling and reduced workload. For rotary winged aircrafts (helicopters) the AIS allows the pilot to command longitudinal, lateral, and collective pitch inputs to the main rotor actuator. This is speciality of the AIS that this system is independent with no mechanical or hydraulic connections to the aircraft flight controls.

The AIS is a safety critical which means that any failure will occur to a critical part in the system would lead to the loss of the aircraft. Therefore, the whole system is subjected to rigorous analysis and testing to provide verification of structural integrity. This is a main reason that in aircraft safety critical systems the method of fastening utilizing screws or bolts plays an important role in the overall reliability of the equipment. These fasteners are generally responsible for a large percentage of field failures. The maximum cases of the failure are occurred due to fatigue. The reason of the fatigue is due to extreme loads that a screw thread can experience in an aircraft condition. For a typical Pilot Control Stick the loading conditions are due to pilot applied loads, vibration, aircraft acceleration and shock. This active collective and cyclic control system will replace an orthodox mechanical setup of rods and bell cranks, providing an electrical interface with the fly-by-wire system and also reducing weight. It provides dynamically variable cues which a pilot is able to feel while handling the controls. Pilot applied loads are considered as the critical design driving loads because these generally envelope all other loading conditions. Therefore, it is important that if screw threads are selected based on the pilot applied loads then other load conditions will be met. So, the selection of a screw/bolt is usually made against an appropriate balance between structural integrity and weight.

2. LITERATURE REVIEW

2.1 Aircraft Loading

The Aircraft Loading is classified into following loading conditions;

- Pilot Applied Loads
- Vibration Loads
- Acceleration Loads
- Shock Loads
- Aerodynamic Loads
- Landing Loads

2.2 Aircraft Stressing

2.2.1 Significance of stress in Aircraft

In case of mean stresses, it is assumed that the total force carried by each bar is uniformly distributed over the cross section. Stress distribution can be far from uniform, with local regions of high stress is known as stress concentrations.

Stress concentrations in engineering components are important not only because they are responsible for stresses much higher than expected from simple calculations, but also important because they tend to encourage the formation of cracks, which can cause to catastrophic failure.

The ultimate tensile strength of a material can be measured as the maximum stress in simple uniaxial tension. For simple states of normal or shear stress, an ultimate load for a component defined as the product of ultimate stress and cross-sectional area. Basically engineering structures and components must be designed and used in such a way that the actual loads they experience are substantially lower than the ultimate loads. The maximum permitted actual load is known as the allowable load and the ratio of ultimate to allowable load is known as factor of safety

Factor of Safety = Ultimate Load / allowable load

In terms of stresses, factor of safety is defined as,

Factor of Safety = Ultimate Stress / Allowable Stress

The choice of an appropriate factor of safety for the different situation is very important, which requires both experience and judgement. There are following consideration in the choice of a factor of safety are as follows:

1. Material properties
2. Loading conditions
3. Type of failure
4. Accuracy of analysis
5. Consequences of failure

2.3 Design of Single Point Fixing

2.3.1 Introduction

Single point fixing is defined as the fixing at a particular one point or single point. This following diagram shows the single point fixing.

The main role of SPF's transfers the flight control stick motions, imparted by the pilot, through a series of linkages and mechanisms to a rotary motion at the servo-actuator unit (SAU). Therefore, the SPF's transfers all pilot applied loads and it is very important component for active Inceptor Systems (AIS).

Materials for Single Point Fixing **-** Spherical Bearing

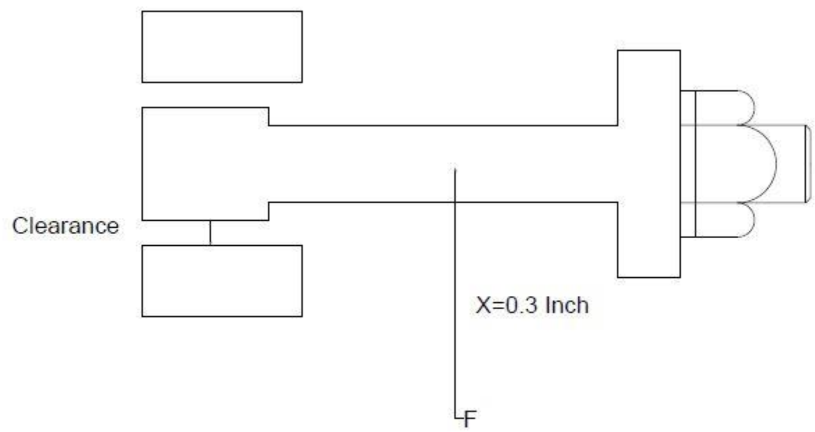

Figure 1: *Single Point Fixings*

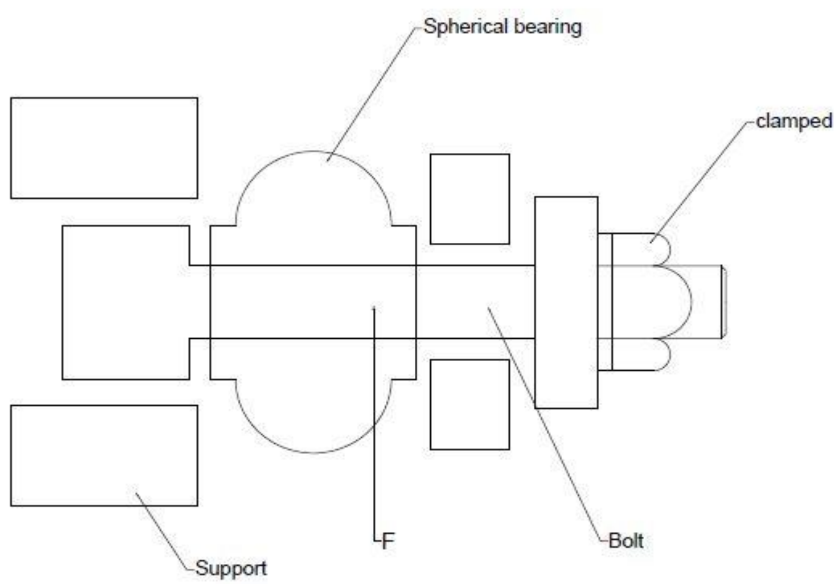

Figure 2: *Materials Used in SPF*

Generally, materials which are used for Single Point Fixing is Aluminium (used for linkage). Some other Aluminium and Titanium alloys are used in the manufacturing of this component. Al-alloy 7075, Ti 6ALV, Ti-6Al-4V, MIL-s-7742 and MIL-S-8879 are some important metal alloys which is used in Single Point Fixing

2.4. Failure Mechanism: There are primarily three different failure mechanisms.

- Bending Moment
- Shear
- Fatigue

2.5 Factors driving failures

2.5.1 Excessive loads

Limit loads are usually defined as the maximum load that could be applied under any combination of loading. Limit loads are explained for pilot control sticks for the following conditions; through mechanism and mechanical end stop. The through mechanism is a special condition where a failure has caused the system or mechanism to jam thereby preventing the stick movement. Mechanical end stop loads are defined when the stick has hit a mechanical hard stop. Therefore, under limit load conditions, all the stresses must be below the proof strength of the material.

2.5.2 Shock loads

When forces or displacement are rapidly applied to a machine member, than it is often found that the stress levels and deformations induced are very much larger than generated by the same forces or displacements applied gradually. These rapidly applied loads or displacements are usually called shock or impact loads. Shock loads are generated in a machine either by the collision of moving bodies or by the sudden application of a force or motion to the structure. Shock loads are generated in many ways like a rapidly moving load, direct impact load or suddenly applied load.

2.5.3 Vibration Loads

As compared to the pilot applied loads; vibration loads are usually much lower in the pilot control sticks. The vibration loads are more important for the electronic equipment's in aircrafts. Vibration loads are mainly produced due to the heavy engine in the aircrafts. The vibration frequency spectrum for aircrafts varies from 3 to 1000Hz with acceleration level that can range about 1 G to about 5 G peaks. The highest acceleration G level appears to occur mainly in the vertical direction in the frequency range of about 100-400 Hz. In case of helicopters, the frequency spectrum will vary from about 3 to 500 Hz and also acceleration level will range from about 0.5 to about 4 G. The highest acceleration G level appears to occur in the vertical direction nearly about 500 Hz.

2.5.4 Cyclic loads / Duty Cycle loads

The duty cycle loads are defined for a specific number of cycles for a pilot control stick. This is usually approximately 2 million cycles representing one lifetime for a pilot control stick. Also the number of cycles is further broken down into cycles for a particular specific position of the stick.

2.6. Bolt Loading

A bolt can be loaded in three ways:

- Tension
- Shear
- Combined tension and shear

A bolt is mainly designed to withstand tensile loading while clamping together. Ideally, the bolt should be loaded in tension

2.7. Importance of preloading

A superimposed mean tensile stress does not greatly alter a specific value, provided the material yield strength is not exceeded. The application of a preload to the bolt by a controlled initial tightening of the nut can allow much of an externally applied fatigue load to bypass the bolt, and be felt as a reduction in the clamping load on the joined components. For this reason, it is beneficial to apply a high preload as practicable to a bolted joint subjected to fatigue. A high preload will also help prevent the nut from loosening, by providing a large friction torque to oppose its movement relative to the bolt and the mating face of the joint.

2.8. Static Loads on Single Point Fixing

2.8.1 Preload in screw threads

This equation explain the relationship between torque and preload force [shigley (1977a)]

Torque,

$$T=k*F*d \quad (1)$$

Where, k= Torque coefficient, which approximately is 0.2

F=pre-load force developed due to tightening torque

d=diameter of screw thread

For example, there is a bolt size of **0.25-28 UNF.**

Maximum preload on bolt is 800 lbf.

For normal operating loads the preload torque is -

T=k*F*d

T=0.2*800*.25

T=40 lbf

$$A = \pi d^2/4 = .04906\ in^2 \quad (2)$$

Preload stress =F/A=16306.563 lbf/in^2=16.31 ksi.

2.8.3 Margins of Safety: The margins of safety for limit and ultimate loads are given by following equations-

$$\text{M.S. limit}=\frac{\sigma y}{\sigma cl} -1 \quad (3)$$

Where,

σy= Proof yield strength of the material

σcl=calculated stress at limit load

$$\text{M.S.ult}=\frac{\sigma uts}{\sigma cu}-1 \qquad (4)$$

Where,

σuts = Ultimate tensile strength of the material

σcu = Calculated stress at ultimate loads

2.9. Loading boundary conditions

2.9.1 Loading boundary condition

In the loading boundary condition BC1, different forces like Limit load, Normal operating load, Ultimate load and Fatigue load are applied on the bolt. As shown in fig 4.1 these forces are applied at a fixed distance x. In this case maximum chance of failure of SPFs is near to surface which is clamped by shear bolt. In this figure δc is a clearance gap between bolt and its support. This is a case of cantilever beam.

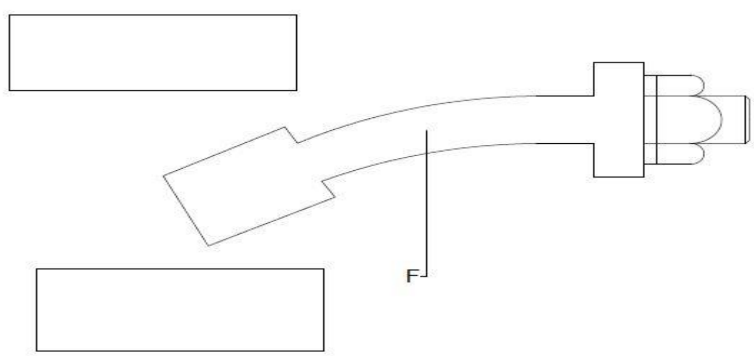

Figure 3: *Bolt under Static Loading Conditions*

2.9.2 Loading boundary condition

In this case of boundary condition BC 2,the forces are applied on bolt due to this bending force occurred in SPFs and due to this end part of the SPFs is being touch by the surface of the support bearing. Consequently, the clearance gap δc is negligible in this condition. In this particular situation as compare to BC1 the condition is changed into simply supported fixed beam.

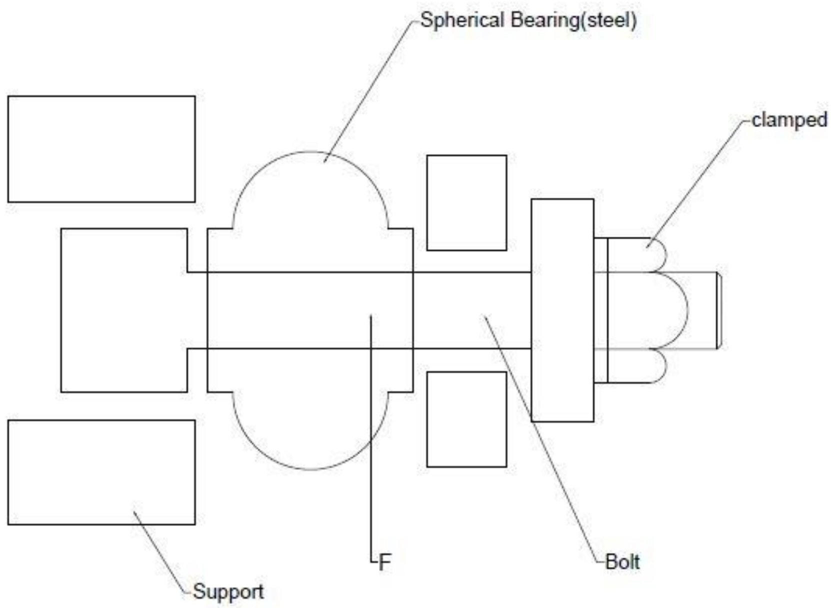

Figure 4: *Bolt under Bending Conditions*

2.10. Shear bolt stress and deflection equation

For boundary condition 1- the deflection occurred on the single point fixing is following-

$$\delta = \frac{FL^3}{3EI} \tag{5}$$

If $\delta < \delta c$ than bolt is in cantilevered condition,

Bending stress, $\delta > \delta c$, in this case, find the bending force on this component,

b=MY/I (Y=.175 inch) (6)

Shear stress,

$$\tau = F/A \tag{7}$$

For boundary condition 2,

$$\delta c = \frac{FL^3}{3EI}$$

M=F*x (8)

$F = \delta c * \frac{3EI}{L^3}$ (Force required closing the gapδc)

c=MY/I

2.11. Static Loads

The main formula to calculate deflection in a cantilever beam under normal operating, limit and ultimate loading conditions is following-

$$\delta = \frac{FL^3}{3EI}$$

The different parameters are-

F=Normal operating force=175 lbf, Limit load =600 lbf, ultimate load=900 lbf

E=Young's modulus of component material (titanium)=16.6*10^6psi

L=Length of bolt=.86 inch

I= $\pi d^2/64$=Second moment of area=3.06*10^{-3}

δ=.00073 inch, δ =.00254, $\delta = .0037$ (for normal, limit and ultimate load respectively)

δc=.376-.373=.003 inch

Clearly, $\delta < \delta c$(for normal operating load)

In this condition the deflection δoccurred in the bolt is less than the clearance gapδc.This is safe condition for bolt because it will not lead to failure of SPFs. In this case calculate bending stress for boundary condition 1 and also will find out maximum and minimum principal stress value

$F = \delta c * \frac{3EI}{L^3}$ =718.6 lbf

M=F*x=215.6 lbf-inch (x=.3 inch)

b1=MY/I=12330.61 lbf/in^2=12.3 ksi

The value of b1 remains same in all 3 loading condition as bending stress in all 3 conditions is calculated and found to be same. While b2 is calculated only for ultimate load as it is the worst condition for bolt and will lead to failure of SPFs.

2.12. Boundary condition for ultimate load

In this case force required to close this gap is less than normal operating loads, so the remaining force will be 900-718.6=181 4 lbf

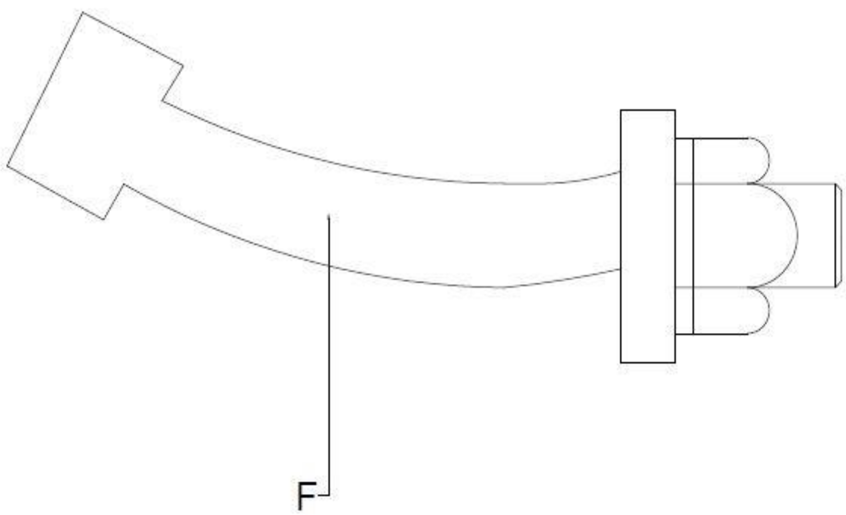

Figure 5: *Single Point Fixing under Bending due to Reaction Force*

M=F*x=54.42

b2= MY/I= 3.1 ksi

Shear stress on the single point fixing –

Normal operating condition-

τ =F/A = 3567.061 lbf/in^2=3.5 ksi

Limit load-

τ =F/A = 12229.93 lbf/in^2 =12.3 ksi

Ultimate load-

τ =F/A = 18344.88 lbf/in^2=18.3 ksi

Now applying the formula of maximum and minimum principal stress,

Principal stress=$\frac{\sigma}{2} \pm \sqrt{(\frac{\sigma^2}{4} + 4\tau^2)}$ (9)

Putting the values of bending stress and shear stress for all 3 conditions we get Maximum and Minimum principal stress as shown in table.

TABLE 1: Total stress in single point fixing

Load	**$\Delta > \delta c$**	**B.C 1 (b1)**	**B.C 2 (b2)**	**Shear stress(τ)**	**Preload stress(p)**	**Total stress=b1 +b2+p**
Normal operating load	NO	12.3 ksi		3.5 ksi	16.3 ksi	28.6 ksi
Limit load	YES	12.3 ksi		12.3 ksi	16.3 ksi	28.6 ksi
Ultimate load	YES	12.3 ksi	3.1 ksi	18.3 ksi	16.3 ksi	31.6 ksi

TABLE 2: Maximum and Minimum Principal Stress for different loading

Load	Maximum Principal Stress	Minimum Principal Stress
Normal operating load	30.22 ksi	-1.62 ksi
Limit load	42.76 ksi	-14.14 ksi
Ultimate load	55.74 ksi	-24.02 ksi

For the grade specifications for bolt material Ti-Al-4V the yield strength and ultimate strength are following:

σy=120 ksi

Uts=130 ksi

M.S. limit= $\boldsymbol{\frac{\sigma y}{\sigma cl}}$ **-1**

M.S. limit=3.87

M.S.ult= $\boldsymbol{\frac{\sigma uts}{\sigma cu}}$**-1**

M.S.ult=2.85

2.13. Fatigue loading

2.13.1 S-N Curve

In the S-N diagram, a constant cycle stress amplitude S is applied to a specimen and the number of loading cycles N till the specimen fail is determined. In this case, millions of cycles might be required to cause failure at lower loading levels. In some materials, ferrous alloys, the S-N curve flattens out eventually, so that below a certain endurance limit failure does not occur no matter how long the loads are applied or cycled. The designer will size the structure to keep the stresses below endurance limit by a suitable safety factor if cyclic loads are to be withstood. In case of other materials such as aluminum, no endurance limit exists and the designer must arrange for the planned lifetime of the structure to be less than the failure point on the S-N curve.

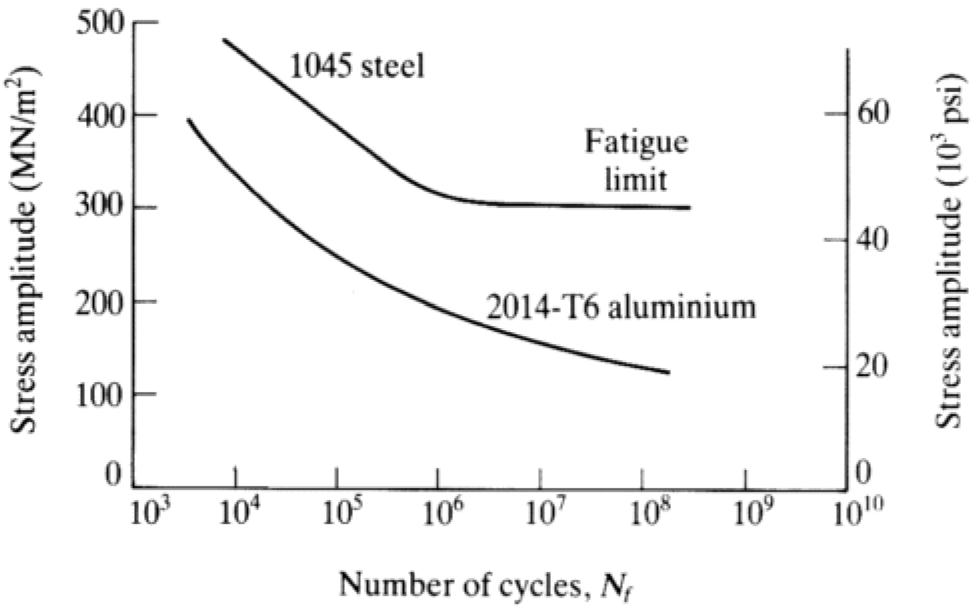

Figure 6: *S-N Curve for Aluminum and Low Carbon Steel*

This random fatigue limit is explicitly included in the S-N model of fatigue analysis. Maximum methods are then used to estimate the parameters of the S-N equation as well as the parameters of the fatigue limit distribution. The percentiles of the fatigue limit distributions calculated from the estimated parameters. The random fatigue limit model produces the proper shape of the median S-N function and the type of scatter easily seen in fatigue tests at HCF stress levels, as illustrated in fig, for Ti-6Al-4V.

TABLE III: Loading Data for Duty Cycle Loads on single point fixing

Cycles	Force (lbf)
40,000	175
60,000	125
200,000	100
300,000	65
400,000	30
500,000	15
500,000	8

In Fatigue analysis also calculate deflection for different forces for different cycles as explained in Table 5.1. Also find out total stress value during fatigue analysis and compare with static loading conditions.

The main formula to calculate deflection in a cantilever beam under fatigue loading is-

$$\delta = \frac{FL^3}{3EI}$$

The different parameters remain same as in case of normal operating load condition and value of maximum and minimum principal stress comes out to be the same as in normal operating load i.e.

Maximum principal stress=30.22 ksi

Minimum principal stress=-1.62 ksi

For the force 175 lbf the value of $\delta < \delta c$ due to this it is being easily analysed that other forces for different cycles are less than 175 lbf. As a result of this the value of δ is always less than the clearance gap δc in the further calculation of other forces in fatigue analysis consequently there is no need to calculate bending stress for every force.

These following formulas are used to calculate the value of Nf and according to that further calculation of total damage index.

Log Nf= 24.6-9.35log σ (10)

Therefore, Nf= $10^{24.6-9.35log\sigma}$

3. RESULT

Total Damage Index (CDI) <0.5. It means the chances of failure or damage of single point fixing is negligible or impossible.

Cycles [C]	Maximum principal stress σ	Nf	Damage Index(CDI) C/Nf
40,000	30.22	10^{10}	$4*10^{-6}$
60,000	29.47	10^{10}	$6*10^{-6}$
200,000	29.16	10^{10}	$2*10^{-5}$
300,000	28.8	10^{10}	$3*10^{-5}$
400,000	28.66	10^{10}	$4*10^{-5}$
500,000	28.64	10^{10}	$5*10^{-5}$
500,000	28.61	10^{10}	$5*10^{-5}$
			Total CDI=$2*10^{-4}$

4. DISCUSSION

The paper discussed stress analysis in Single Point Fixings used in pilot control sticks in military aircraft. Single point fixings usually consist screw threads joined with spherical bearings. During aircraft motion, there are different forces and loads exerted on pilot control sticks. There are chances of failure or loosening of screw threads under these loading conditions. The paper represented stress analysis under static and fatigue loading conditions.

Values of margin of safety for limit and ultimate loading was acceptable. This shows Single Point Fixing will not fail up to 5 times more than the value of margin of safety. The fear of failure of Single Point Fixing is due to fatigue loading because failure under fatigue loading occurred suddenly without giving any warning. It is important to find out damage index of material under different cycles of fatigue loading. The final value of damage index (CDI) < 1, which shows Design of Single Point Fixing is safe under fatigue loading conditions up to 5 million cycles.

This paper provides the basis for stress analysis of Single Point Fixings on which this work can be extended further.

5. CONCLUSION

The paper has presented a methodology for assessing the structural integrity of Single Point Fixings used in an AIS technology which is used on a pilot control stick. The methodology however, can equally be applied to any joint interface utilising screw thread fasteners in machines. In order to demonstrate the methodology, a typical pilot control stick is considered for the following load conditions:

- Pilot Applied Normal/Limit/Ultimate Loads
- Duty cycle loads

For the above loading conditions, structural verification has been provided, together with Margin of Safety and Damage Index for the Single Point Fixings.

For the critical interfaces analysed all Margins of Safety are shown to be within acceptable limits for each of the above failure criteria in Single Point Fixings. The final value of Damage Index (CDI) for Single Point Fixings is less than unity under Duty Cycle Loads. This shows safe conditions for Single Point Fixings.

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