Fatigue and Damage Tolerance Study of Vertical Tail Attachment Bracket of a Civilian Aircraft

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ABSTRACT

Fatigue life calculations and damage tolerance studies are critical in the aircraft design. Under the influence of cyclic loading due to take off and landing, flight conditions and other aerodynamic effects, fatigue damage can occur in aircraft structures. Lug joint at vertical tail (VT) fuselage interface of civil aircraft is studied here. This is part of an effort where critical joints such as brackets, links of an aircraft are studied for fatigue using finite element (FE) based techniques. FE simulation was carried out using ABAQUS. Later, fatigue simulations were carried out using fe-safe. The effect of shrink fit of bushes, pretension on bolts, surface roughness and partial separation of bushes during loading were taken into account. 3D-crack propagation analysis is planned to be carried out using ZENCRACK, a finite element method (FEM) based linear elastic fracture mechanics (LEFM) software.

Keywords: fatigue, aircraft structure, lug joint, vertical tail, fe-safe, crack propagation, fracture, damage tolerance, cyclic loading, load spectrum

INTRODUCTION

Lug joints are commonly used for load transfer between two structural components. These are widely used in aircraft structural applications. For example, lugs are used for attaching engines to engine pylons, for connecting ailerons, flaps and spoilers to wings, to connect VT to the fuselage and also in the actuator mechanisms on undercarriages. In a lug-type joint the lug is connected to a fork by a single bolt or pin. The lug allows relative movement and permits easy installation and dismantling, also the lug can act as a pivot without local bending moments [1]. In aircraft structures careful design of a lug is of utmost importance since the consequences of its failure can be very severe.

Attachment lugs are fatigue (crack initiation and growth) critical components because of their inherently high stress concentration level near the lug holes. There is a lack of comprehensive and reliable data for lug components [2].

A VT fuselage bracket connects the vertical tail to the fuselage of the aircraft. The vertical tails are typically found on the aft end of the fuselage and are intended to control the yaw. The vertical tail generally experiences very high turbulence loads and maneuver loads which are transferred to the fuselage by the VT fuselage bracket. These loads are random and cause severe damage to the brackets. These loads will have two predominant components on the bracket; direct side load and load due to the rolling moment of the aircraft. For the current study, load experienced by these brackets were measured experimentally and were converted to constant amplitude loading. Figure 1 shows the load on the VT in the form of loading blocks. In fatigue calculations the loading sequence plays a vital role. In the current study a low-high loading sequence is taken which gives a conservative life estimate than the actual loading sequence or other loading sequences. A typical relative life predicted by different loading sequence is shown in figure 2. Fatigue life due to the original random cycle is taken as reference life '1'.

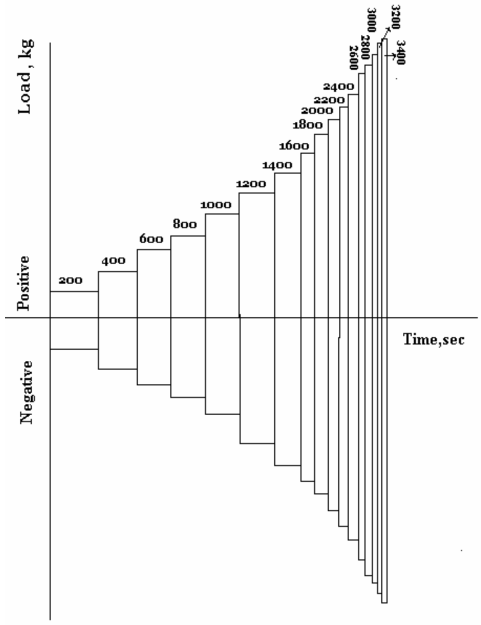


Figure 1: Load on the VT in the form of loading blocks

Figure 3 shows a typical VT fuselage bracket. The bracket has two lugs and is connected to the vertical tail by a pin and is bolted to the fuselage bulkhead. Due to fatigue loads experienced by the bracket there will be high stress concentration near the lug hole which causes premature failure of the bracket. To improve the life of the bracket, compressive residual stresses are induced in the bracket around the hole using various techniques such as cold hole expansion, providing shrink fit. In the model considered the bush is shrink fit, to develop compressive stress. The bushes are assembled by dipping the bush in liquid nitrogen, which shrinks the bush and then assembling it to the bracket. As the bushes come back to room temperature it tries to regain its actual dimension, inducing a compressive stress around the circumference of the hole.

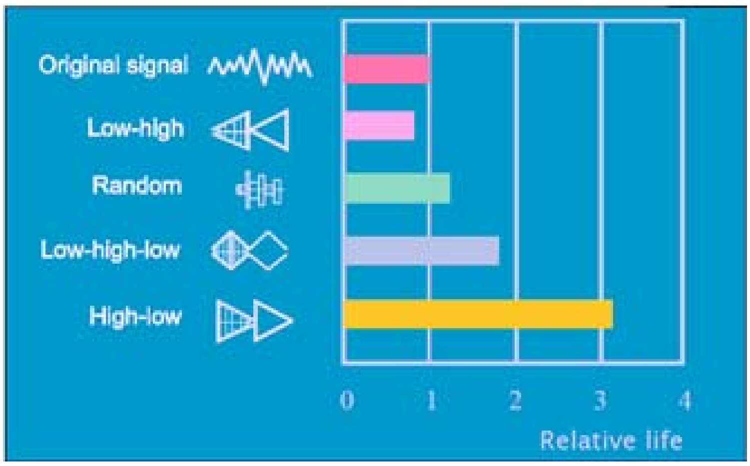


Figure 2: Sequencing effect in fatigue

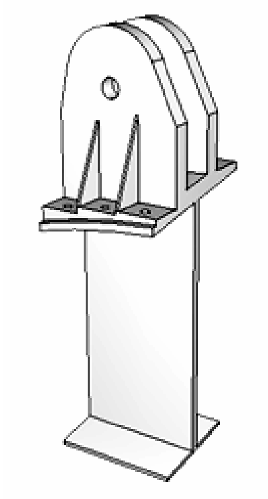
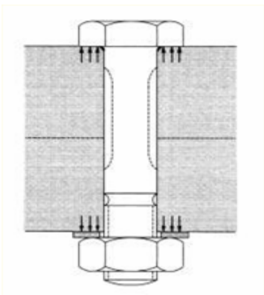


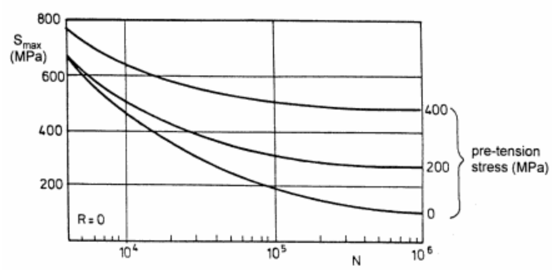
Figure 3: Typical VT fuselage bracket

The bolts connecting the bracket to the fuselage are tightened by applying a prescribed torque to the nut. This introduces a pre-tension load in the bolt as shown in figure 4(a). When a cyclic load is applied to the structure, the pre-tension increases the mean stress in the bolt, but reduces the stress amplitude. In view of the predominant effect of the stress amplitude, a significant gain of the fatigue strength can be obtained. This is illustrated by the S-N curves in figure 4(b) [3].

All these complexities make the study for fatigue and fracture life of such components using empirical relations almost impossible and will be based on assumptions. To overcome these, FE based fatigue and fracture analysis is carried out in this work.



(a)



(b)

Figure 4: (a) Tension bolt with loads on bolt head and nut. (b) Effect of pre-tension on the S-N curve of a steel bolt loaded in tension

MODEL DESCRIPTION

Figure 5 shows the bracket assembly considered for the current study showing various parts. The model consists of different parts: bracket, bushes, pin, bolts, rivets and rigid base. The bracket has two lugs with hole diameter of 2R fitted with interference bush of outer diameter 2(R+I), where I is the amount of radial interference. These two lugs are connected to the vertical tail by a pin. There are six holes on the face of the bracket of diameter 2r which are also fitted with interference bush of outer diameter 2(r+i), where i is the amount of radial interference. The bracket is further bolted to the fuselage bulkhead through these six bolts and is tightened by applying a prescribed torque to the nut.

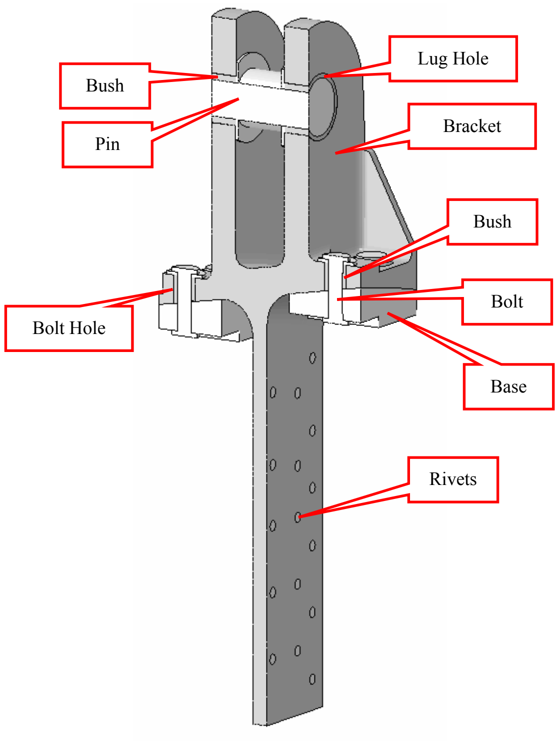


Figure 5: VT fuselage bracket assembly

The bracket is made up of Al-Cu alloy (AL 2124) and is CNC machined. This alloy is a high purity version of alloy 2024. The higher purity provides higher elongation in the short-transverse direction and improved fracture toughness over that exhibited by conventionally produced 2024 alloy [4].

METHOD OF ANALYSIS

The complete analysis can be divided into four different stages as shown in figure 6. First stage was to generate FE mesh for the geometry. In second stage FE analysis was carried out. FE analysis has 3 steps:

1. Stress analysis due to shrink fit.
2. Stress analysis due to pretension on bolts.
3. FE analysis for VT-fuselage loading.

This loading had 17 different elastic-plastic FE analysis. At all loads stress, strain, displacement, reaction force developed in the model are predicted. The results of these analysis obtained in second stage were taken to third stage for finding out the location of crack initiation. The results from FE analysis and fatigue analysis are further taken for crack growth analysis.

FE MESH GENERATION

In the model considered for the current study; bracket, bushes, pin, bolts and rivets were generated using two different linear isoparametric solid elements, such as eight noded brick (C3D8) and six noded wedge (C3D6) elements with three translation degrees of freedom (U_1, U_2 and U_3) at each node. Figure 7 shows the FE model in standard views. FE model consists of 1, 40,339 elements. The mesh was generated by considering all the quality parameters, contact definitions to be given, interference etc.

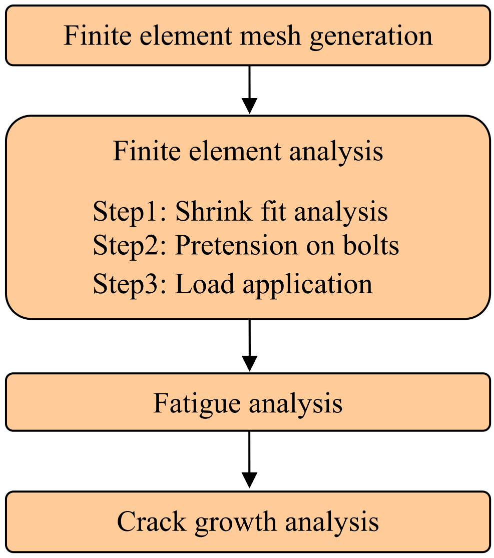


Figure 6: Methodology flow chart

FE ANALYSIS

A commercially available software ABAQUS was used for FE analysis. The complete analysis was carried out in three different load steps. In the first step shrink fit was defined using contact elements. These results were taken to the next step where pretension in the bolts were defined. In the final step the load coming on the bracket found from experiments were applied.

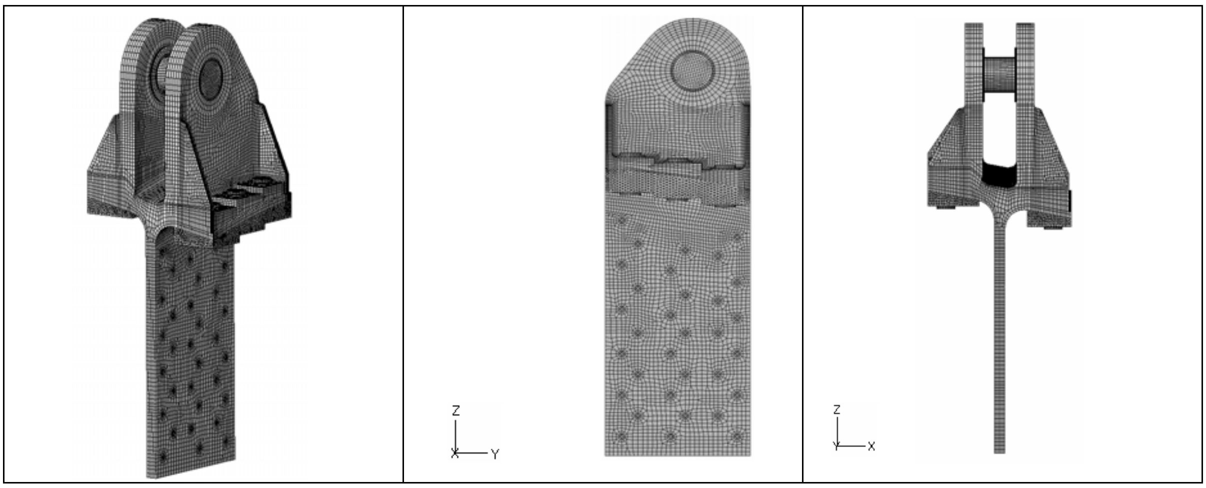


Figure 7: FE model of VT lug joint

Interference is defined between the bush outer face and the bracket hole using contact elements. This induces compressive stresses around the circumference of the hole.

In the second step pretension is applied to the bolts. The application of pretension tightens the joint and induces compressive stresses on the face of the bracket which in turn improves the fatigue life.

In the third step load is applied on the pin attaching the VT and bracket. The load is a combination of vertical and side load developed due to the various loads applied on the VT. A cyclic load was applied for the 17 load cases. The state of stress and displacement will be different in tension and compression for each load case. For lower loads on the pin the bush will be in contact throughout the circumference of the lug hole and the compressive stresses developed during shrink fit starts reducing in the region opposite to the direction of load around the lug hole. As the load is increased on the pin the compressive stresses go on reducing and at some load magnitude the radial stresses become zero around the circumference of the hole in the region opposite to the loading direction. This effect is caused due to the partial separation of the bush from the lug hole during loading.

In the FE analysis of VT lug joints non-linearity in contact, material behavior pose numerical problems. Care should be taken to achieve good convergence and accuracy [5].

FATIGUE ANALYSIS

The stress and strain obtained at various increments during loading from FE analysis is imported to “fe-safe”, a fatigue and durability analysis software. “fe-safe” has multi-axial and strain based fatigue algorithms [6]. Strain life approach was used for the current analysis which uses the following governing equation:

$$\Delta\gamma_{max}/2 + \Delta\epsilon_n/2 = C_1\sigma'_f/E\,(2N_f)^b + C_2\varepsilon'_f\,(2N_f)^c \qquad \text{... (1)}$$

where, $\Delta\gamma_{max}/2$ is the shear strain amplitude and $\Delta\epsilon_n/2$ is the normal strain amplitude. C_1=1.65 and C_2=1.75 are the constants derived based on the assumption that cracks initiate on the plane of maximum shear strain, σ'_f is the fatigue strength co-efficient, E is the Young's modulus, b is the fatigue strength exponent, $2N_f$ is the number of reversals, ε'_f is the fatigue ductility co-efficient and c is the fatigue ductility exponent [7].

“fe-safe” requires stress and strain datasets from an elastic-plastic FEA (or elastic FEA results can be used with Neuber’s correction). In the current study 17 FE results obtained were combined in fe-safe and 17 different loading blocks were defined. The loading defined was equivalent to 10000 flights. In the current study elastic-plastic FEA was carried out to take the stress redistribution effect into account which is not taken into consideration in elastic FEA. Separate fatigue analysis was carried out for maximum cruise speed and economic cruise speed. Material property data, surface finish factor were specified. Morrow’s mean stress correction was used for the strain life fatigue analysis. Morrow corrects the elastic term of strain life equation by subtracting the mean stress (σ_m) from the elastic term. Modified Miner’s rule was used for damage calculation and finally total damage was calculated for all loading blocks. “fe-safe” gives the number of cycles to crack initiation, location of the crack initiation and possible direction of crack propagation.

RESULTS AND DISCUSSION

FINITE ELEMENT ANALYSIS RESULTS

Shrink Fit: The compressive stress induced around the hole depends on the percentage of shrink fit. For the structures similar to the one taken for the current study, generally 0.2%-0.3% interference is defined. At the end of shrink fit radial stress of around 30MPa (compressive) was observed at the lughole and stress of around 130MPa (compressive) was observed around the bolt hole as shown in figure 8. This stress will be uniform throughout the circumference of the hole.

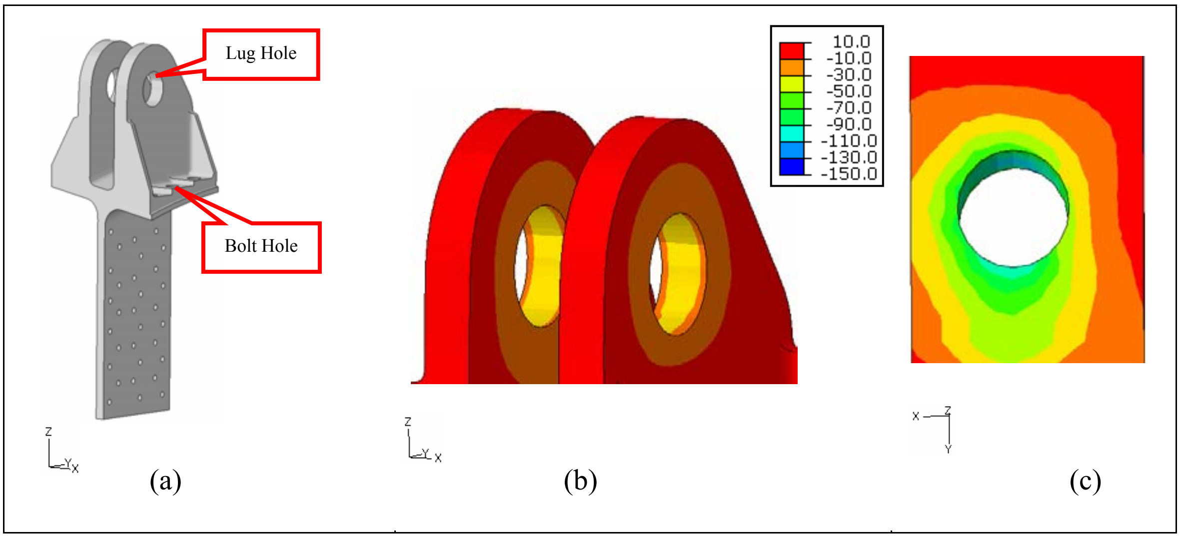


Figure 8. (a) Lug hole and bolt hole location (b) Radial stress (MPa) plot around lug hole
(c) Radial stress (MPa) plot around bolt hole

The radial stress will be compressive in nature while the tangential stresses will be tensile around the hole as shown in figure 9.

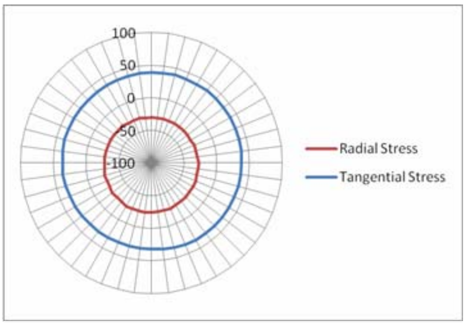


Figure 9: Radial and tangential stress (MPa) distribution (polar plots) around the bolt hole at the end of shrink fit

Pretension: During this step pretension was applied on bolts which develop compressive stress along the axis of the bolt hole. In the present study a pre-load of about 5000N was applied which induced a stress of about 50MPa (compressive) on the face of the bracket (figure 10). The effect of pretension is carried forward throughout the analysis.

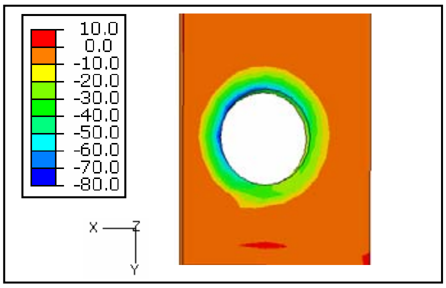
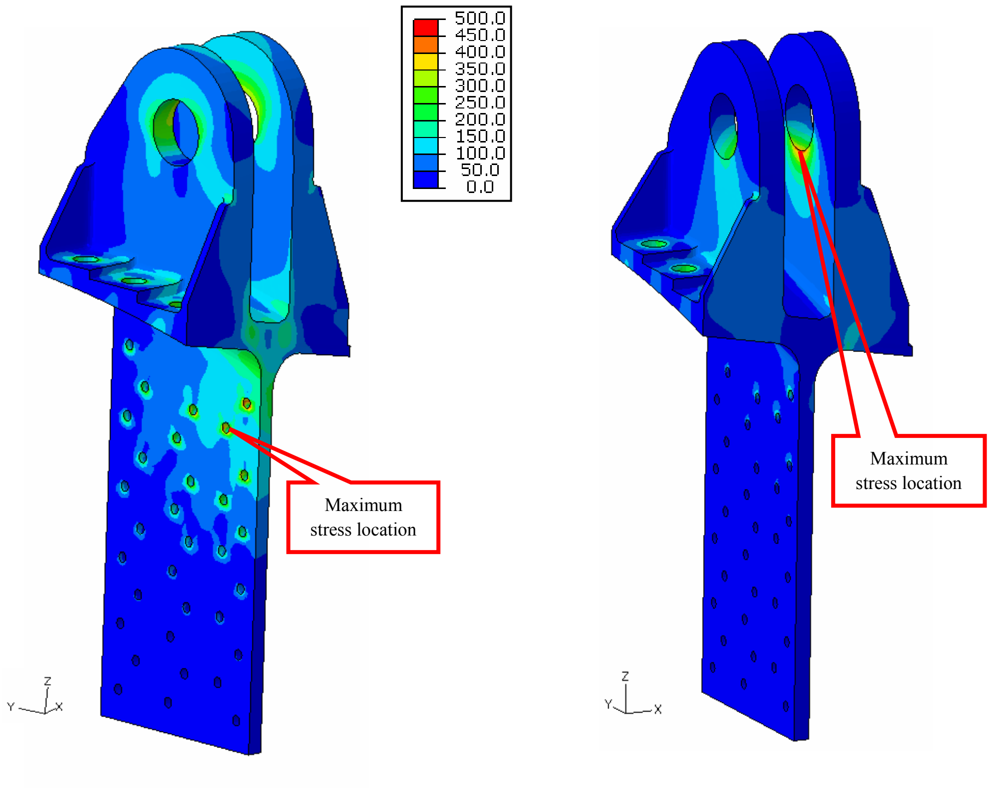


Figure 10: Compressive stress (MPa) along the axis of the bolt hole due to pretension

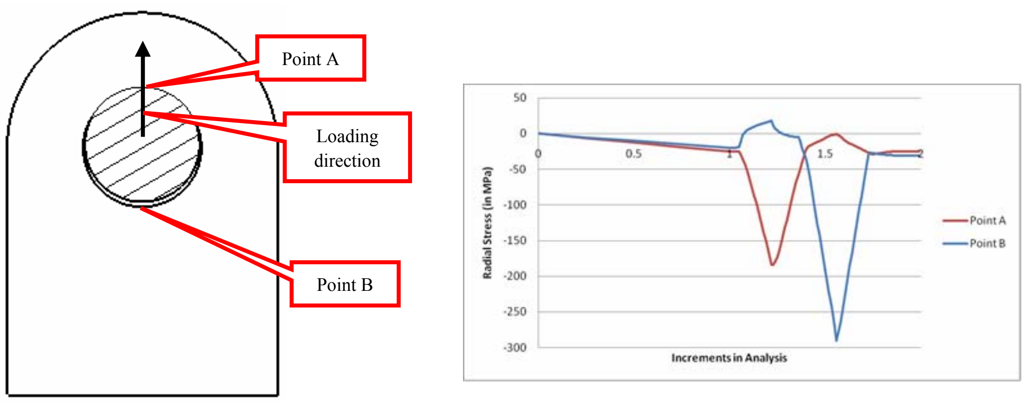
Loading: In this step, horizontal and vertical loads were applied on the pin. Only typical stress results are given here in the paper for the sake of brevity. Von Mises stress plot for tensile and compressive load for the maximum load case (3400kg on the VT) are shown in figure 11(a) and 11(b) respectively.

It is observed from the stress contours that during tensile loading high stress is observed near the rivet location, but during the compressive loading the maximum stress is observed near the lug hole.



(a) (b)

Figure 11: (a) Von Mises stress (MPa) contour due to tensile loading
(b) Von Mises stress contour (MPa) due to compressive loading



(a) (b)

Figure 12: (a) A schematic diagram showing separation effect and the locations of point A&B
(b) Radial stress (MPa) variation at point A and point B for complete loading cycle

As the load applied on the pin increases, radial stresses developed due to shrink fit reduces in the area opposite to the loading direction and becomes zero at some particular load. This is effect is shown in figures 12(a) and 12(b) where separation takes place around point B for a tensile load applied. The same effect can be observed around point A for a compressive load. Even after complete removal of load shrink fit stresses will remain in the bracket as the bush is interference fit with the bracket.

FATIGUE ANALYSIS RESULTS

Fatigue analysis is done only on the bracket excluding pin, bush, bolts and rivets. "fe-safe" gives unfactored fatigue life to crack initiation.

Economic Cruise Speed: Life to crack initiation was observed to be 1,64,816 flights ($10^{5.217}$ flights). The crack initiation location is shown in figure 13.

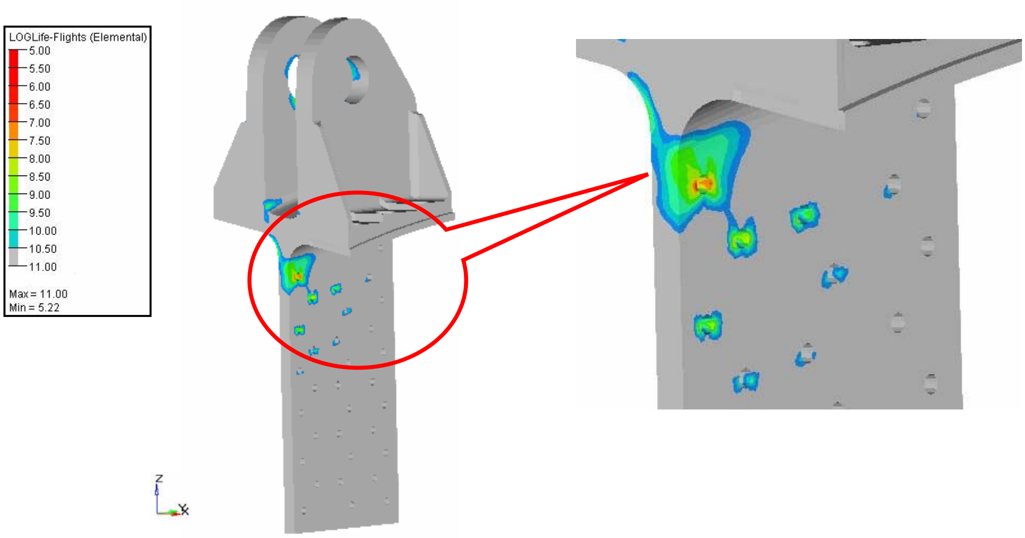


Figure 13: Fatigue life contours in log scale for economic cruise speed

Maximum Cruise Speed: Fatigue analysis was carried out for maximum cruise speed by using the spectrum specified. The crack initiation location was observed to be same as that in economic cruise speed. Life to crack initiation was observed to be 58,884 flights ($10^{4.77}$ flights). A comparison is shown in table 1

Table 1: Life comparison table

Results	Economic Cruise Speed	Maximum Cruise Speed
Life to crack initiation	1,64,816 flights	58,884 flights

The results above are unfactored fatigue lives which are calculated for zero mean stress. To account for the loading and material property variations, a factor 3 will be used for calculating the fatigue life. Thus, the factored life for crack initiation can be taken as 58,884/3 which is equal to 19,628 flights for the maximum cruise speed.

SCOPE OF FUTURE WORK

A damage tolerance analysis is being carried out using the crack location predicted by fe-safe. This is being done using a commercially available fracture mechanics software ZENCRACK. It takes an uncracked 3D mesh supplied by the user and inserts one or more crack blocks at the specified locations in the mesh based on user input. The cracked mesh is then submitted for FE analysis. Results of the FE analysis are extracted and processed automatically. For each crack block movement stress intensity factor (K) is calculated at the crack tip. Using this, energy release rate (G) is calculated and based on this the crack movement and direction is decided. G is calculated using the following formula

$$G = \frac{K^2}{E'} \quad \text{with} \quad E' = \begin{cases} E & \text{(plane stress)} \\ \dfrac{E}{(1-\upsilon^2)} & \text{(Plane strain)} \end{cases} \qquad \text{... (2)}$$

The crack front in the mesh is advanced and a further FE analysis is carried out. This process repeats until the analysis reaches a termination point [8]. Authors propose to use CTOD and J-integral approach for crack growth estimation. da/dN and K/N characteristics are determined and cycles to failure (fracture) are estimated.

CONCLUSION

In the light of the results and discussions presented above, the following conclusions can be drawn.

Fatigue analysis results shows that life to crack initiation location need not be the maximum stress location. Fatigue life of the lug was found to be 19,628 flights (at maximum load case) for the load spectrum defined. Maintenance schedule can be planned based on the obtained results. Using commercially available software packages a methodology is formed for analysis of aerospace lug joint and this methodology can be adopted for the analysis of similar fittings.

ACKNOWLDGEMENT

Authors wish to thank authority at NAL for giving an opportunity to publish this work. Guidance and assistance of Shri. Chidananda, Dr. T. S. Prahalad, Dr. K. Yagnanarayan and Dr. K. N. Raju of NAL are gratefully acknowledged. Authors also wish to thank all colleagues at ProSIM for their support during the course of the work.

REFERENCES

1. Jong-Ho Kim, Soon-Bok Lee, Seong-Gu Hong, “Fatigue crack growth behavior of Al7050-R7451 attachment lugs under flight spectrum variation”**,** Theoretical and Applied Fracture Mechanics 40 (2003) 135–144, Elsevier Publications.
2. J. Vogwell, J. M. Minguez, “Failure in lug joints and plates with holes”, Engineering Failure Analysis, Vol 2, pp. 129-135, 1995.
3. Jaap Schijve, “Fatigue of structures and materials”, Edition: 2, illustrated, published by springer, pp. 423-424, 2001.
4. Military Handbook, Department of Defence, United States of America, MIL-HDBK-5H, pp. 3-94, 3-174, 1 December 1998.
5. Thierry Stehlin, “Fatigue analysis of riveted or bolted connections using finite element method”, Poster presentation at ICAF 2003.
6. fe-safe user’s manual, Safe technology limited, U.K, volume1, pp.15-1, www.safetechnology.com.
7. John Draper, Safe technology limited, U.K, “Modern metal fatigue”, volume1, 2008.
8. ZENCRACK user’s manual, Zentech International, U.K, pp.115, www.zentech.co.uk.