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Design Optimization of Shear Fitting of Vertical Tail and Rear fuselage Interface of an Aircraft

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ABSTRACT - The main sections of an airplane include the fuselage, wings, cockpit, engine, propeller, tail assembly, and landing gear, etc. These parts are subjected to various loads. These loads are distributed & transferred to other load taking units of an airframe by different types of fittings namely shear fitting, bending fitting, hinge fitting etc.

This project deals with the design and analysis of a typical lug shear fitting representative of an airframe structure applications. The design will provide safety against Tension tear out failure & Shear bearing failure

In this project for preliminary sizing of the lug of a shear fitting suitable method (ESDU91008) which is used in aerospace industry & acceptable to aviation authorities like FAA, EASA, etc. is followed to design the lug of shear fitting. The shear fitting design involves performing numerical calculation, creating a solid CAD & FEM model and performing the finite element analysis. For material selection appropriate reference as per global aerospace industry standards is followed.

The 3D CAD model of fitting is realized with the help of CATIA V5 software and Finite Element Analysis is carried out with MSC PATRAN and NASTRAN software.

Keywords: Airframe, shear fitting, FEA, FEM, CATIA V5, MSC PATRAN & NASTRAN.

I.INTRODUCTION

Various parts of the aircraft consists of Different types. These fittings are attached to major components of aircraft such as wing, landing gear, horizontal tail, vertical tail etc. The basic function of an aircraft structure is to transmit and resist the applied loads, and maintaining the aerodynamic shape while protecting the passengers and cargo from the various forces which the aircraft is subjected to during flight. There are various types of aircraft constructions such as monocoque or semi-monocoque. In semi-monocoque construction the outer skin of the aircraft is normally supported by transverse frames and longitudinal stiffened members called stiffeners to protect skin panels from buckling by resisting the bending, compressive, and torsional loads. Main function of Aircraft is to provide the interface between two components and transfer the loads from larger component to smaller component and load transfer from the vertical tail to the fuselage is very good example of it. Vertical tail of aircraft is consist of horizontal and vertical stabilizer which transmits the loads to fuselage by means of shear fittings, bending fitting and hinge fitting etc. Arrangements and type of fittings are totally depending upon types of loads transfer they are responsible for. Shear Fitting transfer the shear loads, bending fitting transfer the bending loads. Lug and flange with fastener holes are main parts of any fitting. This paper is case study of Design Optimization of Shear Fitting of Vertical Tail and Rear fuselage Interface of an Aircraft. Vertical tail of aircraft consist of Spar, ribs and skin. Every part experiences a different loads and resultant of it will be transferred to the fuselage through fittings. Various types of fitting arrangements are used in aircraft construction since arrangement of fitting plays very important role in load transfer. Shear fitting we have used for study is followed by two bending fitting combination.

II.LITERATURESURVEY

In the paper Design and Analysis of Lug Joint in an Airframe Structure Using Finite Element Method author C V Rama Krishna et al. design and FEA approach for the typical lug joint of aircraft structure is carried out. Paper provides a safety against lug failure and pin failure when the lug is under axial loading condition. Two methods max peak stress and stress averaged over the contact area to calculate Margin of safety for lug joint but in the max peak stress the predicted margins were much less than those calculated from the theoretical calculations. As the name suggest shear fitting transfer the shear loads but it is critical

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in axial loading. Main part of fitting is lug and Flange. Lifting lug and clevis or pad aye are different names for lug. After applying axial loading on shear fitting lug can fail in conditions mentioned below.
1) Tension failure;
2) Shear tear-out;
3) Bearing failure.

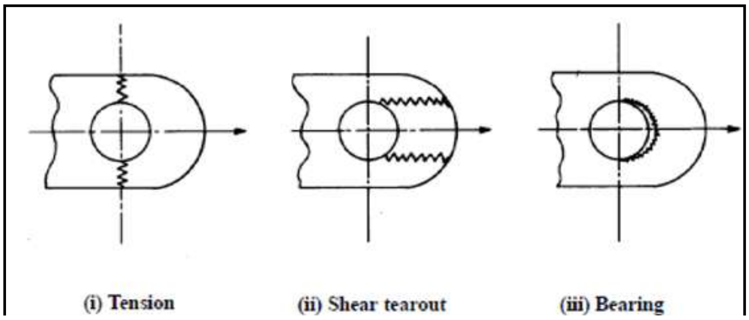

Fig.1

Fig.1, represents the lug failure mode under the action of axialtensile force represented by the arrow and the failure occursacrossthenetsectionoftheluginthemiddleregio nofthelug,Fig.1, represents the failure of the lug bysheartear-outwhenbeingloadedaxially.Sheartear-outoccurswhenshearispredominant andthefailureregionisat45degreesfromtheloadingaxi s,asshownabove.Fig.1also represents the bearing failure of the lug under axial loadingcondition.Thisfailurewhichoccursinthelugw hentheappliedconditions.in Fig1 wherein tension tear out, shear tear out and bearing tear out are shown.oblique loading conditions. Other ESDU methods are used to evaluate the stress concentration in hole analysis of the lug is shown in Paper.

According to ESDU 91008 strength To an interference-fit bush, and endurance of a lug under cyclic loading and the stress intensity factors (SEF) for cracks in loaded holes [3].

Again lug analysis can be done by different methods as mentioned below:

Analysis simplified method – In This method by calculating nature of failure and calculating the FOS (Factor of safety). This method is easy but it doesn't give an exact result.

Air force method - Most of the failure modes are considered in this method as mentioned above, and allowable loads are calculated by using empirical curves to determine more accurate results. This method allows for lugs under axial loading, transverse loading, and oblique loading. This method also accounts for the interaction between the lug and the pin.

In Paper **Design and Analysis of Shear Fitting For Vertical Tail to Rear Fuselage of an Aircraft**Author done Initially the model of the lug is created with 2D elements, specifically using 2D-quad elements for meshing, and considering the dimensions calculated the analysis is performed. The model is as shown below.

When the analysis is carried out, two main components are observed Von-mises stress and maximum principal stress. Von- mises stress represents shear critical regions, whereas, maximum principal stress represents bending critical regions. Results of this analysis are as

III.LITRATURE GAP

At the educational level, we did not find much work done on Attachment fittings. Many aircraft parts get qualified under the FEA approach only. There is a scope of improvement in available shear fitting literatures. Forthis Project/dissertation we have referred a research paper **Design and Finite Element Analysis of Shear Fitting for the Vertical Tail of An Aircraft** by author Vikas Sanmati.

Following Observations made after study –

Fitting Design is not actually safe as claimed in research paper because stresses are higher than material allowable (428N/mm2).

Von mises stresses = 455N/mm2

Max principle stresses = 459N/mm2

IV.METHODOLOGY

1. Study and Understand available literature papers, among them one is selected for detailed study and identified the literature gap.
2. Since Shear fitting in research paper **Design and Analysis of Shear Fitting for Vertical Tail to Rear Fuselage of an Aircraft**Fitting is not safe for given loading conditions hence, we have made the fitting safe to do comparative study.
3. Worked on problem statement as stated.

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4. Understanding design problem inputs and extraction of Critical Forces from previous research paper.
5. Sizing of Lug as per approved method ESDU 91008
6. Analyzefitting using FE method (2D iteration).
7. Iteration of design based on deformation & stress criteria.
8. Prepare Updated CAD model (Modified Design).
9. Analyzefitting using FE method (Using 3D iteration)
10. Manufacturing of part.
11. Preparation of Final Report.

V.PROBLEM STATEMENT

The interface point i.e. lug hole centre must be at 88 mm from the fitting flange back facedeformation should be less than 0.8 mm for given fail safe load case. Lug-hole diameter must be 24mm in order to accommodate 20 mm diameter pin (already frozen dimension considering assembly constraints) & 24 mm outer diameter Bush (2mm thickness) & 20 mm diameter pin. Most applications for hinge designs use symmetrical double shear lugs or multiple shear lugs which are only used for fail-safe conditions.

Lug of the fitting is designed & analyzed for the following Ultimate loads.

Table 1:

Sr.No.	F_X (N)	F_Y(N)	F_Z(N)
1	12833 N	-	137840 N

A fitting factor of $\lambda = 1.15$ should be used (both ultimate and yield strength.) In any sizing, ifboth fitting and casting factors are involved, only the larger factor shall be used. In addition to the factors mentioned above, lug sizing shall show a minimum MS of 20%

VI SAFE FITTING

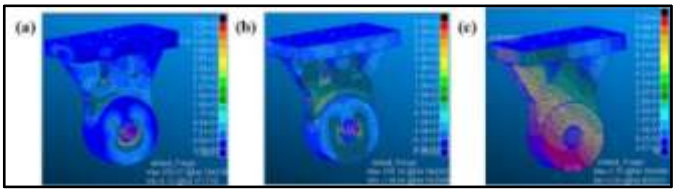

Fig. 2 (a) Von mises, (b) Max principle (c) Deformation

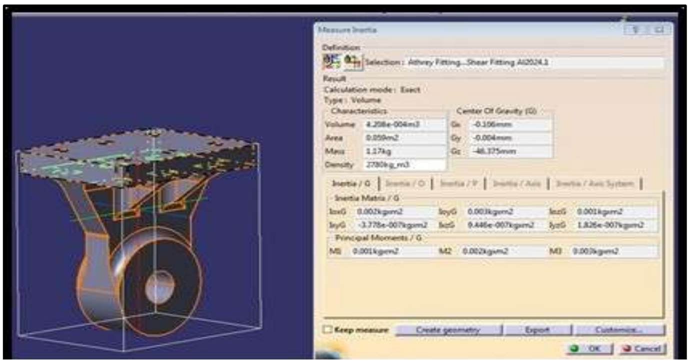

Fig 3. Weight of Current Fitting

Fitting has been made safe for given loads we can see Von Mises, Maximum Principle and Maximum deformation in fig.2 which are respecting all the design requirements as stated. Now final weight of fitting is 1.17kg in fig. 3

In this paper Optimization of fitting is done based on result we got in after making the authors fitting safe.

Table 2:

Lug material is 7010-T3 (Ref MIL-HDBK 5H, Table 3.2.3.0)			
For Thickness 40<t<= 62 and 'A' Grade			
Ultimate Tensile Strength, L, F_{tul}=	70 ksi	483 N/mm^2	Ult. Tensile strength, L
Ultimate Tensile Strength, LT, F_{tuc} =	70 ksi	483 N/mm^2	Ult. Tensile strength, LT
Yield stress/ 0.2 % of Proof Stress, Cross grain F_{tpc} =	42 ksi	290 N/mm^2	Yield Tensile strength, LT
Ultimate Bearing Strength, L, F_{bru}	94 ksi	649 N/mm^2	Bearing yield strength, e/D =1.5
Ultimate Bearing Strength, L, F_{bru}	115 ksi	794 N/mm^2	Bearing yield strength, e/D =2.0

VI. HANDCALCULATIONS

Sizing & Static Analysis –

1. Lug Analysis Geometric Data (Initial assumed lug dimensions)
2. Lug geometric data is listed.

Table1: Lug geometric data

W (mm)	d_h (mm)	a (mm)	t (mm)
50	24	36	16

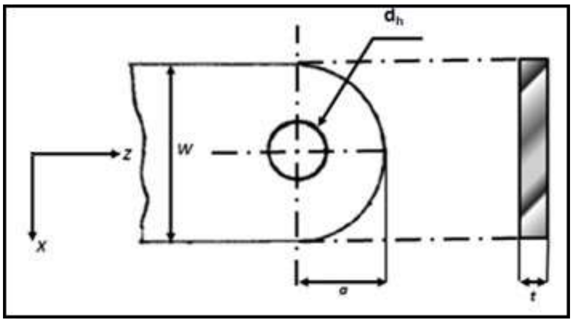

Figure 4 Lug dimension

Axial Loading

For a lug under axial load three modes of lug failure are considered:

i. Tension

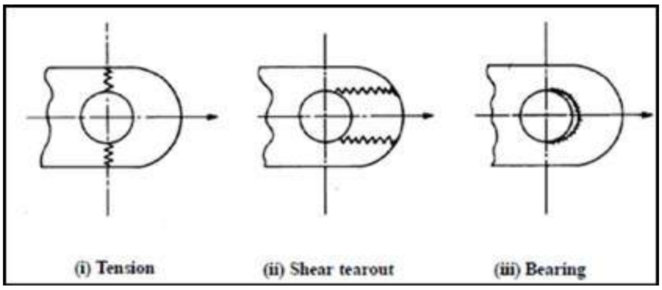

Figure 5 Failure mode by applying axial loading

Shear tear-out

Applied Load:

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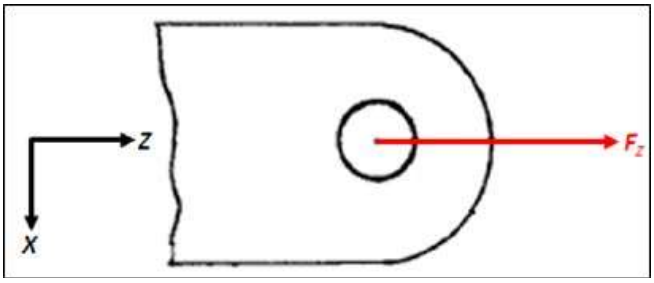

Figure 6Applied Loads

Tension Tear out

• Calculation of Ktux:

Based on (W/dh) = (50/24) =2.08 &a/W=36/50= 0.72

Lug stress concentration factor, Ktux =0.925 (from Graph)

• Lug failure due to tensile rupture

$P_{tux} = K_{tux} \times f_{tux} \times (W - d_h) \times t = 0.925 \times 483 \times (50 - 24) \times 16$ Ptux = 185858.4 N.

Shear tear out & Bearing

• Calculation of Kqux:

Based on (a/dh) = (36/24) =1.5 & dh/t =24/16 = 1.5

Lug stress concentration factor, Kqux =1.45 (from graph)

• **Lug failure due to shear-bearing rupture**

Pqux = Kqux×ftum×dh×t = 1.45× 483× 24 × 16

Pqux= 268934.4N

• **Shear/Bearing reserve factor, RFS**

RFS =Pqux / (FF×Fz)

RFS =268934.4/ (1.15 × 158516) = 1.47

Transverse Loading

Applied Load

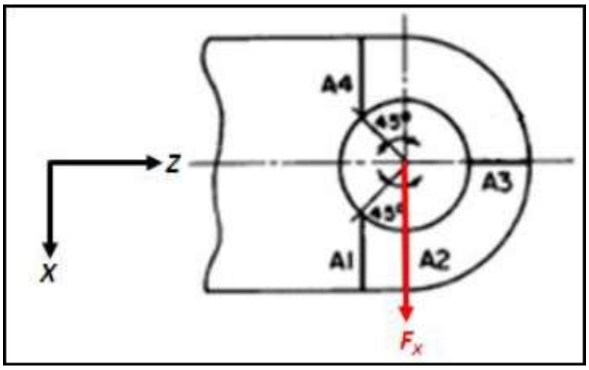

Fig 7

Applied Loads

$A_1 = A_4$ =[a – (dh× 2) × Cos 45°] t = [36 – (24 / 2) × Cos 45°] X 16 =440.23mm²

A_2=[(W/2)- (dh× 2)] × t = [(50/2)- (24 / 2)] × 16 = 208 mm²

A_3= [a - (dh× 2)] × t = [36 - (24 / 2)] × 16 = 384 mm²

Average Bearing Area

$$A_E = \frac{6}{\frac{3}{A_1}+\frac{1}{A_2}+\frac{1}{A_3}+\frac{1}{A_4}}$$

A_E=421.74 mm²

• Calculation of Kuy:

Based on - (AE) ÷ (dh X t) = (421.74) ÷ (24 X 16)= 1.09

Kuy = 1.09

Puy = Kuyxftumxxdh x t = 1.09 X 483 X 24 X 16 = 202164.48 N

RFS =Puy/ (FF*Fx) = 202164.48/1.15 X 12833 = 13.69

•**Reserve factor due to axial loading**

Reserve factor, RF A =1.51

•**Reserve factor due to transverse loading**

Reserve factor, RF T = 14.32

•**Reserve factor – Ultimate RFA**

$$RF_{ult} = \frac{1}{\left(\left(\frac{1}{RF_A}\right)^{1.6} + \left(\frac{1}{RF_T}\right)^{1.6}\right)^{0.625}}$$

RFult = 1.48

VII CONCLUSION

For current sizes of lug, it is Safe in Axial, transverse and combine loading. Since, all Reserve factors are greater than 1.

DETAIL DESIGN BY FINITE ELEMENT METHOD

Now lug is safe by analytical method (Chapter.2) so let's validate design by proceeding to FEM approach.

SIZING IERATION 1

• Input for FEM model 1st iteration

Flange dimension = 50×40 mm.

Thickness of flange = 10 mm

Number of Fastener = 4, Pitch 4D= 40mm & Edge distance=2D= 20mm

Diameter of fastener D = 10mm

Initially the model of the lug is created with 2D elements, specifically using 2D-quad elements for meshing, and considering the dimensions calculated the analysis is performed. The model is as shown below. Fig. 8 (a), represents the shear fitting with a 2D element mesh, which consists of flange as well as lug with a hole.

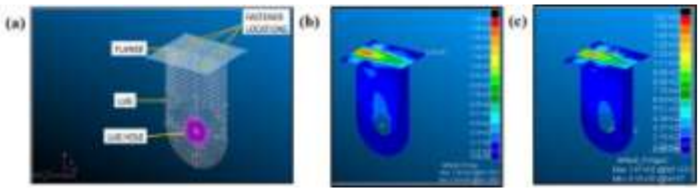

Fig 8 (a) Shear fitting with 4 fasteners positions with 2D element mesh, (b) Von Mises Stress (c) Maximum Principle Stress.

The analysis is carried out, & two main components are observed Von-mises stress and maximum principal stress. Von-mises stress represents shear critical regions, whereas, maximum principal stress represents bending critical regions. Results of this analysis are as shown in the figure Fig. 8(b) and.

Figure 8(c), which represents the Von-mises stress for the

Combined loading consists of axial load and the transverse load. The highest Von Mises stress observed in the combined loading is 1560N/mm2 near the flange and lug intersection area.

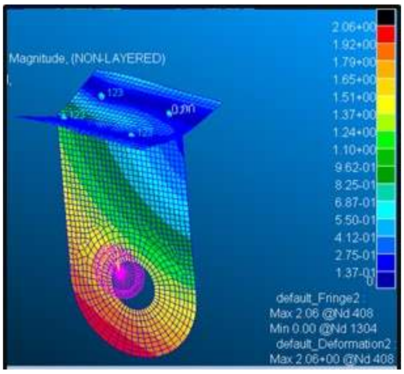

Fig 9. ITR 1 Deformation

Figure 8 (c), represents the maximum principal stress in the model under the action of combined loads. The highest maximum principal stress observed in the combined loading is 1670N/mm2 near the flange and lug intersection area.

Both von mises & principle stresses are significantly greater than material allowable property (i.e. Ultimate Tensile Strength, Ftul = 483 N/mm2).

There is a sudden change in geometry from lug to flangethickness is resulting very less material (oriented only in X direction at junction) to transfer loads. Also since load has to be transferred from one structural member (flange side) to another member through shear fitting it is necessary to transfer this stress from flange and lug intersection area to the lug hole which is achieved by adding extra stiffeners in Y direction.

SIZING ITERATION 2

Input for FEM model 2nd iteration

Flange dimension = 50×40 mm.
Thickness of flange = 10 mm
Number of Fastener = 4, Pitch 4D= 40mm & Edge distance=2D= 20mm
Diameter of fastener = 10mm
Stiffener thickness = 10mm, Stiffener height = 45 mm, stiffener base= 39 mm

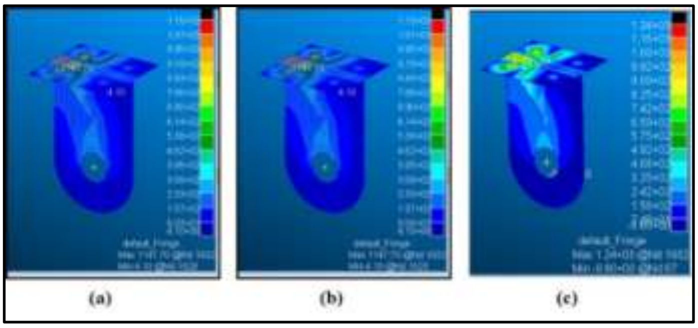

Fig. 10 (a-c) ITR 2 Von-mises & Maximum Principle stress, (c) ITR 2Deformation under the combined loads

Still both von mises & principle stresses are significantly greater than Ultimate Tensile Strength, (F_{tul} = 483 N/mm2) & are at flange are near flange lug intersection area. It is evident by the result that moment M_y created by force F_x is causing this stress. In next iteration extra stiffener will be added in X direction and the flange dimensions will be increased as well as increasing the number of fasteners in order to lower the stress in flange area & transfer the stress critical zone to lug area.

SIZING ITERATION 3

Input for FEM model 3rd iteration

Flange dimension = 120×91 mm.
Thickness of flange = 12 mm
Number of Fastener = 4, Pitch 4D= 62.5mm & Edge distance=2D= 23.5mm
Diameter of fastener = 9.5mm
Stiffeners along Y direction: thickness = 12mm, height = 45 mm, base= 39 mm.
Stiffeners along X direction: thickness = 16mm, height = 45 mm, base= 36 mm.

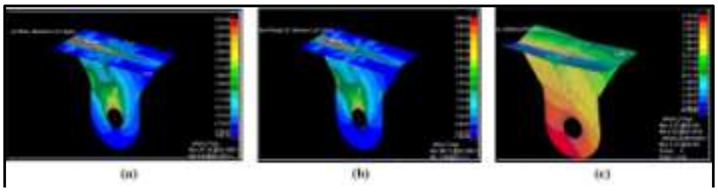

Fig.11 (a) ITR 3 Von-mises stress for the combined loading (b) ITR 3 maximum principal stress for the combined loads, (c) ITR 3 Deformation under the combined loads

Stress results are, von mises = 351.34N/mm^2& Maximum Principle stress= 384.12 N/mm^2 .Now first time in sizing iterations we have stresses less than material allowable i.e. Ultimate Tensile Strength, (Ftul = 483 N/mm^2). Maximum deformation occurring is 0.42 mm, which under 0.8 mm design acceptance criterion. So now we can proceed to 3D FEA approach to analyze stress behavior in thickness direction. Along with this let's try to optimize the design in terms of material reduction if possible.

In the Fig.11, combined loading acting on the shear fitting after adding the gusset, increasing the flange dimensions and increasing the number of fasteners, the highest value of maximum principal stress observed is 556N/mm2 and stresses are reduced in small quantity near the flange and lug intersection area. The overall stresses are increased compared to previous analysis but the main agenda is to shift the stress from flange to the lug hole and stiffen the flange. After observing the change in stress further more modifications are made and to get more accurate results design is carried out with 3D idealization and the model is meshed using 3D-tetrahedral elements.

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SIZING ITERATION 4

Input for FEM model 4rd iteration

Flange dimension = 120×91 mm. (in order to maintain Pitch & edge distance)
Thickness of flange = 12 mm
Number of Fastener = 4, Pitch 4D= 62.5mm & Edge distance=2D= 23.5mm
Diameter of fastener = 9.5mm
Stiffeners along Y direction: thickness = 18mm, height = 41mm, base= 34 mm.
Stiffeners along X direction: thickness = 18mm, height = 37 mm, base= 34 mm Remaining dimensions are same as previous iteration.
3D CAD model is created using CATIA V5. All the appropriate edge fillets value are assumed for this iteration are as follows.

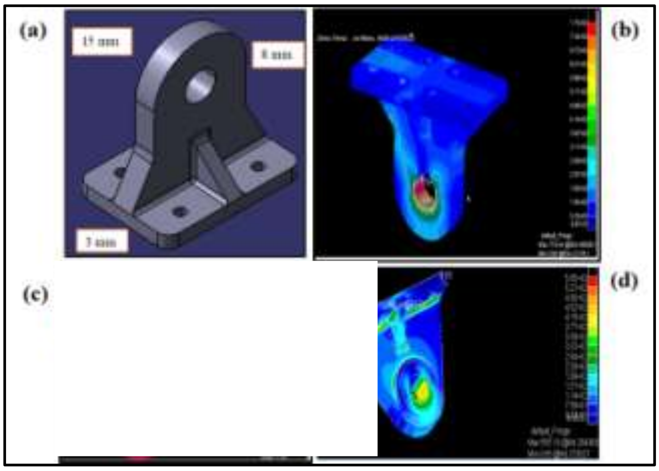

Fig 12 (a) Iteration 4- 3D CAD model with edge fillet radii, (b) ITR 4 Von-mises stress for the combined loading
(c) ITR 4 Deformation under the combined loads, (d) Fig 19.

After performing analysis stress results is von mises = 775.44 N/mm2. Maximum deformation is 0.87 mm these result values are higher than previous 2D iteration since here in 3D iteration loads are applied as a pressure on surface, in 2D iteration load were applied as a point load on a RBE element. So in next iteration let'smake lug face tapered.
Let'sanalyze other critical areas in the model. Let's hide lug hole area as shown in image below to see next stress critical area.

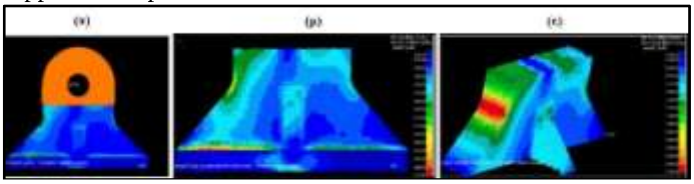

Fig 12(a)ITR 4 Second critical stress area after lug hole,
(b)ITR 4 Second critical stress area after lug hole,(c) ITR 4 Critical stress area after lug flange junction

Criticality shifted to lug flange junction with von mises stress value 522.32 N/mm2 (which is greater than Ultimate Tensile Strength, Ftul = 483 N/mm2). This means edge fillet = 3 mm in this area need to be modified. Similarly let's check next criticality in lug & stiffener in x direction edge fillet area. For this let's erase element at flange area as shown in image below.

Here von mises stress value 416.38 N/mm2. Which is slightly lesser than Material ultimate strength. But to ensure next modification does make this area stress critical & unsafe, this region i.e. edge fillet will be modified in next iteration. Let's eliminate this fillet by creating complete taper edge tangent to lug outer circular edge.

SIZING ITERATION 5

Input for FEM model 5th iteration

Flange dimension = 120×91 mm.
Thickness of flange = 15 mm
Number of Fastener = 4, Pitch 4D= 62.5mm & Edge distance=2D= 23.5mm
Diameter of fastener = 9.5mm
Stiffener base dimensions are adapted as per change in flange dimensions. Remaining dimensions are same as previous iteration.

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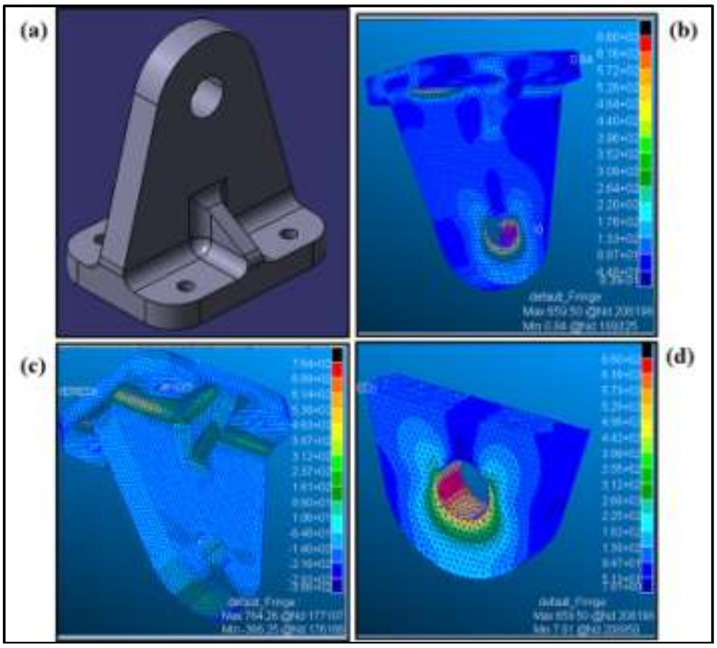

Fig. 13 (a) Iteration 5- 3D CAD model,
(b)ITR 5 Von-mises stress for the combined loading
(c) ITR 5 maximum principal stress for the combined loads, (d) ITR 5 Deformation

•At lug flange junction fillet:
1. Von mises stress = 659.50 N/mm2.
2. Maximum principle stress=764.28 N/mm2

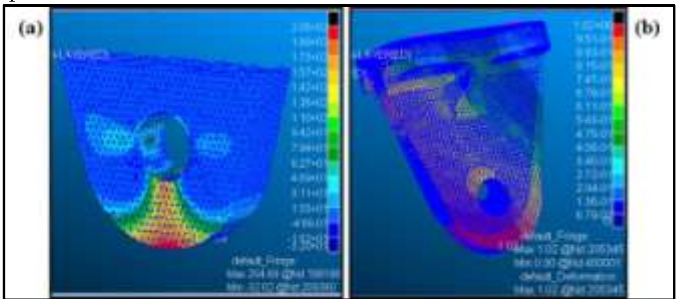

Fig. 14 (a) ITR 5 maximum principal stress at lug hole,
(b) ITR 5 deformation

At lug hole:
1. Von mises stress = 414.35 N/mm2.
2. Maximum principle stress= 398.76 N/mm2.
3. Overall maximum deformation of 0.71 mm is also observed in lug hole area.

So, by comparing above results it is evident that fitting is failing at lug flange junction by both Von mises & maximum principle criterion but is safe in lug hole area. So the fillet radius in lug flange junction need to be modified.

SIZING ITERATION 6

Input for FEM model 6th iteration
Flange dimension = 120×91 mm.
Local thickness near lug hole= 25 mm.
All the stiffeners now does not land on flange directly as before, now for more smooth landing step is created as shown in image below & sharp edges are filleted by radius 10 mm.

Also Stiffener base dimensions are adapted as per change in

Flange dimensions. Remaining dimensions are same as previous iteration.

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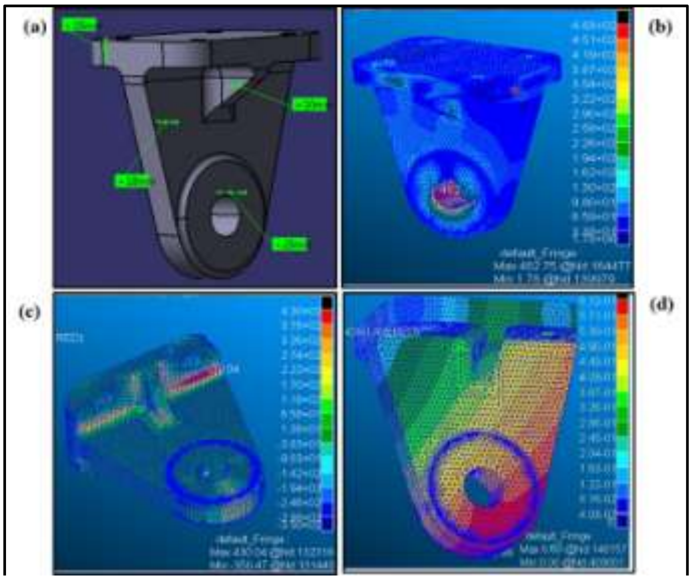

Fig.15 (a) stiffener landing step fillet radius= 10 mm,
(b)ITR 6 Von-mises stress for the combined loading,
(c)ITR 6 maximum principal stress for the combined loads, (d) ITR 6 Deformation

• Results critical area, at lug flange junction fillet:
1. Von mises stress = 482 N/mm2.
2. Maximum principle stress=430 N/mm2.
3. Maximum deformation= 0.6 mm.

Comparing these result with iteration 5 result it is evident that significant decrease in von mises and maximum principle stress which is less than material allowable.
And deformation is 0.6mm is also less than 0.8mm which was needed as per design considerations.

Results:
• Von mises stress critical at lug hole = 482 N/mm2.
• Maximum principle stress critical at lug flange junction fillet =430 N/mm2.
• Maximum deformation= 0.68 mm (<0.80mm)
Stresses are less than material allowable i.e. Ultimate Tensile Strength, (F_{tul} = 483 N/mm2). Maximum deformation is also less than prescribed design limit. Hence safe design is achieved in this final iteration. Following are the final designed dimension of the safe design.

VIII ACTUAL PHOTOS

Fig. 16 machined Fitting

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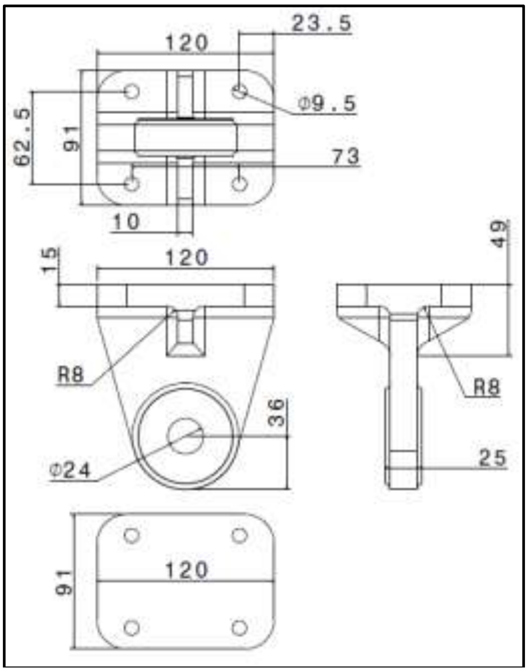

Fig. 17Orthographic drawing of the shear fitting with all the designed dimensions

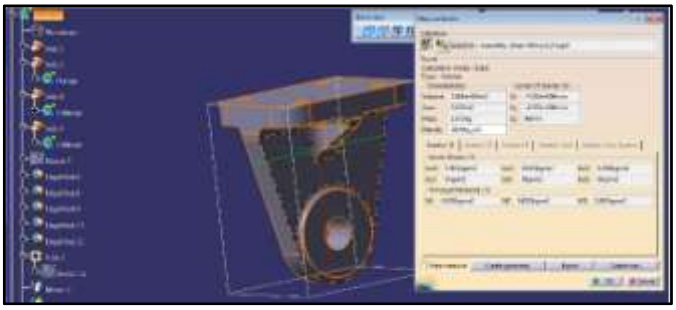

Fig. 18 Weight of optimized fitting

IX. OBSERVATIONS& CONCLUSION

Observations

It is verified by above results of iteration 6 that,

1. For Out of plane loading condition i.e. part of the component subjected to bending or critical to bending (flange in this case), maximum principle stress is critical (higher than Von mises stress).
2. And for In-plane loading condition i.e. part of component subjected to tension or critical in tension (lug in this case), von mises stress is critical (higher than maximum principle stress).

Conclusion

Depending on the configuration used in the aircraft structures, a shear fitting primarily transfers the shear loads, but its criticality depends on the axial load acting on it, and the effect of the transverse load is not very dominant, which is very evident from the analysis performed in this project. With load inputs F_x&F_z as a reference for the initial lug dimensions, calculations were performed, for different loading conditions, Failure criteria comprising tension, shear, bearing under transverse, axial and combined loading was were taken into account. This showed that the fitting was critical due to axial loading condition. The prescribed geometric constrain of interface point should be at 88 mm from back surface of flange & lug hole diameter = 24, were taken into account for preliminary sizing.

Considering this constraints, preliminary lug dimensions were lug width 50 mm & thickness 18 mm, Flange dimension = 50×78 mm, Thickness of flange = 10 mm,

Number of Fastener = 4, Diameter of fastener D = 9.5mm.

The FEM modelling was done with two-dimensional idealization. The analysis results

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showed Von-mises stresses maximum principal stress were significantly higher than material Ultimate Tensile Strength, F_{tul} = 483 N/mm2). After several iterations, dimensions of the shear fitting were completely modified and finalized, arriving at a dimension as per Figure 17.

Design has been optimized by changing the material of fitting from Al2024 to Al7010 hence material chosen for the design & analysis is Al7010-T3. And significant weight saving of 9% has been achieved by changing the fitting material since the higher stresses of the improved design of shear fitting are within the material allowable limit, dimensions obtained for the last design iteration can be considered as the final dimensions.

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