Topology Optimization of an Additive Layer Manufactured (ALM) Aerospace Part

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Abstract

As part of research into the benefits of Additive Layer Manufacturing (ALM) manufacturing process, an Airbus A320 nacelle hinge bracket was optimized, incorporating a topology optimization method. The design freedom of the ALM process meant that a significant proportion of weight could be saved in the part, while also reducing maximum stress and maintaining stiffness. Optimization of small-scale parts presents a large opportunity for weight saving, and may become economically viable if tools are developed to reduce the man-hours used in the design process.

Keywords: Optimization, OptiStruct, topology,

1.0 Introduction

Metallic Additive Layer Manufacture (ALM) technology is a relatively young technology in the early stages of being implemented into the manufacture of aircraft. The main benefits of the ALM process come in design flexibility, low material waste, low CAD-to-part time and cost of producing parts from hard materials that are otherwise difficult to machine. ALM is currently a comparatively expensive process, but this expense is acceptable in high-value applications where specialised materials are used or where a customer requires a complex part.

Because of the design freedom available with ALM, it is a perfect application for topology optimization. Where usually a topology optimization has to be 'interpreted' and sacrifices in the design have to be made for manufacturability. With ALM, the principal is that the topology optimized shape can be maintained and the final weight and structural properties can be closer to that of the optimized shape.

Reducing weight also means that the part manufacture costs less. As ALM is an additive process the part cost is proportional to the volume of the part. The more material used, the more expensive the part will be. This is opposed to how many parts are currently made. Subtractive processes (e.g. milling) are often used to reduce weight, these incur a trade-off between cost and weight, this does not happen with ALM.

This project was completed as part of AVLAM: (Added Value though Layered Additive Manufacturing) a TSB-funded collaborative project involving EADS, Materialize, TWI, Bombardier Aerospace, The University of Exeter, and TISICS.

2.0 Test Case

A test case was chosen to evaluate the technical and commercial viability of producing optimized ALM parts for aerospace.

The test case chosen was an A320 nacelle hinge bracket, shown below.

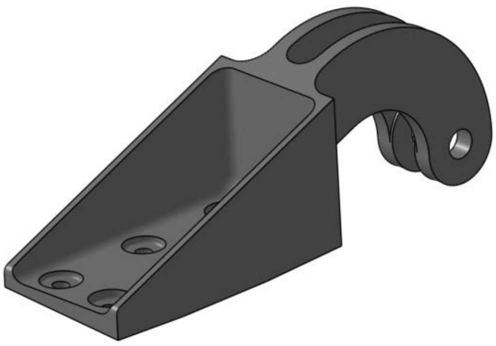

Figure 1: Original Part

Each nacelle has two large doors either side of the engine which are hinged at the top for inspection and maintenance. The hinge bracket is fixed to the nacelle door with 6 bolts and attaches to a corresponding bracket onto the main structure of the nacelle. There are 8 different hinges on each nacelle; each with slightly different geometry and load case.

The part is made out of HC101 steel, because of the proximity to the engine and strength needed.

The shape of the part is largely defined by the use of a three axis milling machine. The part is ‘near net shape’ sand-cast, and then milled to tolerance. The design is very simple and intuitive from a design perspective, but from a structural perspective it isn’t ideal and quite bulky.

2.1 Original Loading case

Figure 2 is taken from the original bracket design and validation.

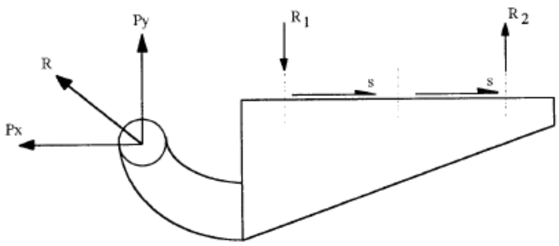

Figure 2: Loading diagram from original validation

Figure 3 and **Figure 4** show the loading direction, the arrows showing the load are proportional. The exact loads cannot be given in this paper.

Due to proximity to the engine, the temperature of the maximum and fatigue load cases are at moderately elevated temperature.

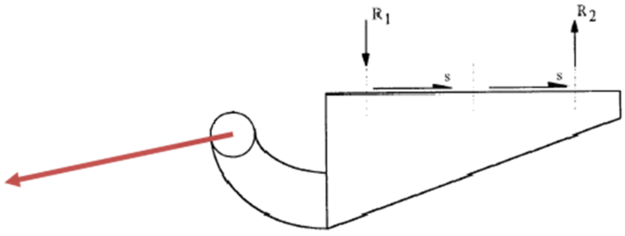

Figure 3: Loading direction – Maximum strength load case

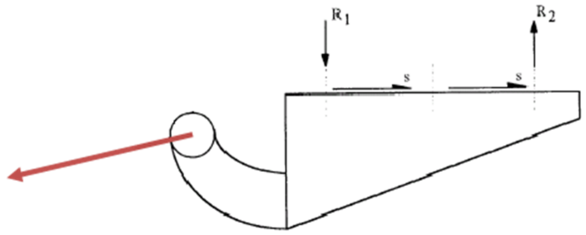

Figure 4: Loading direction – Fatigue load case

The fatigue case requires a part life of 400,000 cycles, the resultant force on the hinge from the resultant of the fatigue load case is high compared to the maximum load case: It is around 80% of the maximum strength load case, so the fatigue case appears to be the main design driver.

2.2 Material Comparison

Some tensile and fatigue tests were carried out for ALM manufactured Ti6Al4V, a summary of the results is shown in **table 1**.

Property	Minimum Test Value (9 coupons)*
Young's Modulus (Gpa)	116
Yield Strength (MPa)	1008
UTS (MPa)	1085
Elongation %	13%

Table 1: Material properties of new process – ALM / Ti6Al4V

The density of the original material was 7.7 g/cm^3, whereas the new material is 4.42 g/cm^3, so there is a weight saving to be expected through the weight of the material, although the young's modulus is less than the 193Gpa of the original steel, so more material may be required to maintain stiffness.

The tests were promising, the tensile tests revealed material properties equivalent to Ti6Al4V manufactured through traditional means. The fatigue tests were also good, although further testing is needed to confirm this to an acceptable reliability. Because of this a conservative figure of 350MPa will be used for the maximum stress in the fatigue case and once new figures are obtained a decision can be made as to whether the design is acceptable.

2.3 Analysis of original part

Maximum stress in ultimate load case – 836MPa
Maximum stress in fatigue load case – 701MPa

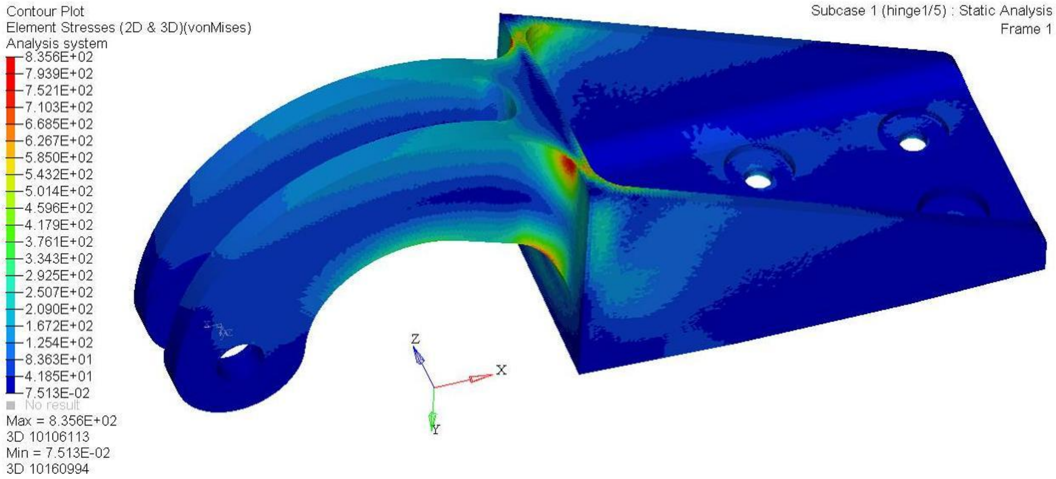

Figure 5: FEA analysis of original part

The FEA revealed large stress concentrations in the part **(Figure 5)**. The large tensile and bending stresses created by the loads at the hinge have to travel through shapes which do not carry the load effectively; in particular, there is a kink where the stress concentrations appear. The large stress concentrations and large amounts of relatively unstressed material indicate an inefficient use of material in the part. The stress concentrations limit the maximum strength of the part and also have a negative effect on its stiffness. It is also very likely that there is material that could be removed from the part with a negligible effect on the parts performance; this is effectively a waste of material.

3.0 Optimization Strategy

The objective of the optimization is to produce a viable part with as little material as possible, i.e. minimum weight.

To make sure that the part is viable and safe, the part has to be constrained. The first major constraint is the stiffness of the part. To ensure the stiffness of the part, the maximum displacement along the hinge line was constrained so that it was no greater than the equivalent displacement in the original part. In reality this displacement will be kept within +/- 10% of the original. As there are many hinges holding the door, a dramatic change in bracket stiffness would alter the load distribution across all of the hinges. In other words if the bracket is stiffer it may attract more load through it instead of through adjacent hinges. In this case this would mean that the values for the loads would be incorrect therefore making the validation incorrect also.

The maximum stress must not be above critical values. For the maximum static load case the maximum stress is 1000MPa (yield stress of the material) and for the fatigue the maximum will be 350MPa.

Non-designable areas were also used around the hinge and lug as there needs to be contact at these places, and to mitigate inaccuracies caused by using rigid elements to connect these areas to forces and constraints.

A tetrahedral mesh is preferred for the topology optimization stage for ease of use. For validation hexahedral elements are preferred for accuracy of the result.

4.0 First Design Cycle

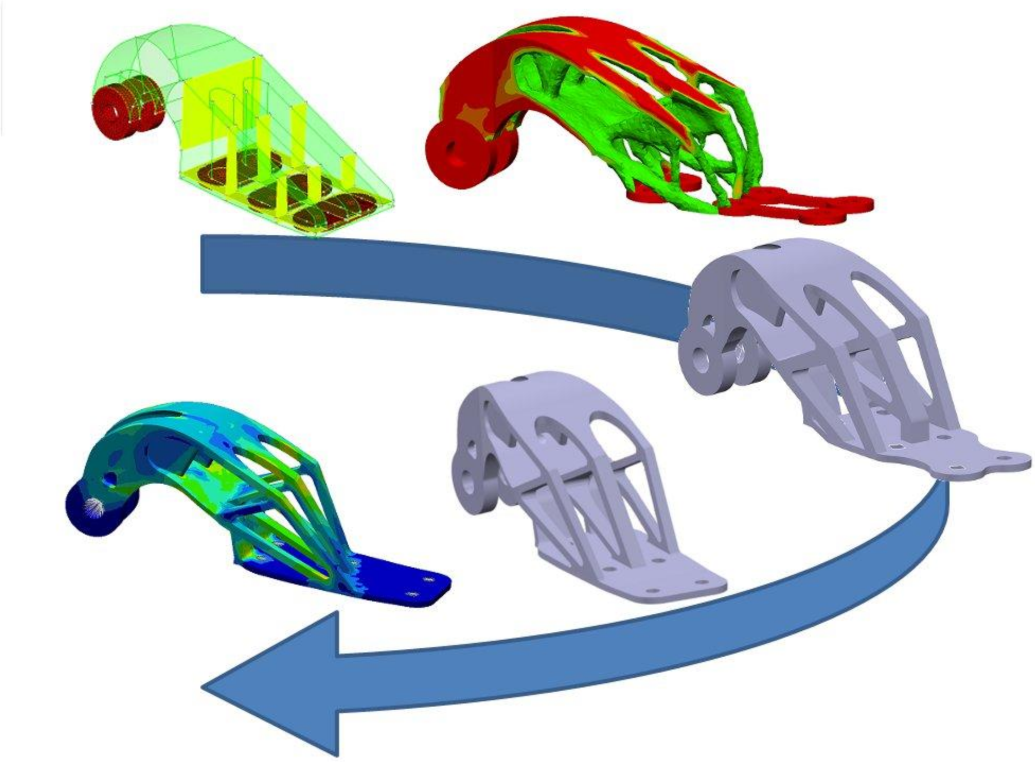

Figure 6: Optimization loop 1

Figure 6 shows the whole of the first optimization cycle. During the topology optimization there was much trial and error involving the non designable areas of the bolts and with hindsight, a more liberal application of the non-designable area at the bolted area would have made the interpretation much easier.

The stresses in the topology were very low, so a constraint on stress did not have to be applied. The stiffness of the part was the main design driver.

The design was interpreted in CATIA v5, following the topology optimization as closely as possible. Mesh morphing using the morphing tool in Hyperworks and Optistruct were used to shape and size optimize the part.

Final design weight – 310g

Maximum stress in the final part: maximum – 365MPa, fatigue – 320MPa

The design wasn't approved as the final row of bolts (rightmost pair of bolts, **Figure 6**) were not supported and there were concerns of increased forces through certain bolts. Concerns of the robustness of the design if certain areas broke were also expressed. On further inspection it was found that the loadings on the bolts nearest the hinge line were much higher than in the original design.

Modelling of the panel and bolts, and either proving the design or re-optimizing it could have been an option but it was decided that this would be not only unreliable, depending on how the panel was modelled and constrained, but it also would be very time consuming.

The solution was to measure the forces in each of the bolts in the original and constrain the same forces in the optimization so the optimization produced a satisfactory loading distribution across the bolts. To do this beams with a very small length were added in-between the constraints in the centre of the bolt holes and the rigid 'spider' elements connected to the perimeter of the bolt holes. These beams were small enough not to influence the model and were purely to measure the forces at the bolts. In each beam three responses were measured: axial force, and two values of shear force in different planes. An equation was created in HyperMesh to calculate the resultant shear force.

The axial and shear forces were recorded from the model of the original hinge. For the original part, most of the load was transferred into the two bolts nearest the hingeline. For the new optimization it was decided that the forces at these two bolts would be constrained to be no more than the corresponding axial or shear force in the model of the original hinge.

5.0 Second Design Cycle

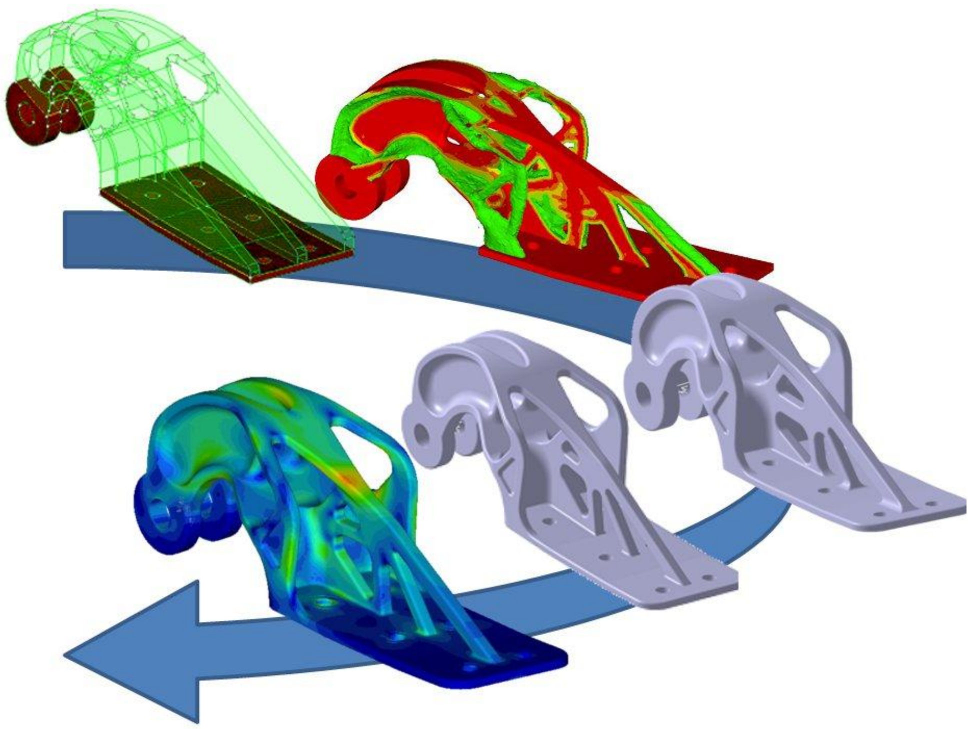

Figure 7: Optimization loop 2

The optimization was repeated with the new constraints mentioned in **section 4,** and several other improvements in the optimization approach, correcting mistakes in the first. The whole bolting flange was added to the non-designable region of the topology optimization and the design volume was more restrictive, making the interpretation easier.

The part now distributes the loads across the bolts better than the original and only uses 16g more material than the first design.

6.0 Final Design

Validation confirms that the design is viable; the main concern was stress in the fatigue case. In the final design the final stress was 310MPa. Further fatigue testing should confirm whether this will be acceptable for 400,000 cycles. Stress in the maximum load case is acceptable. Maximum loads on the bolts are less than the original design.

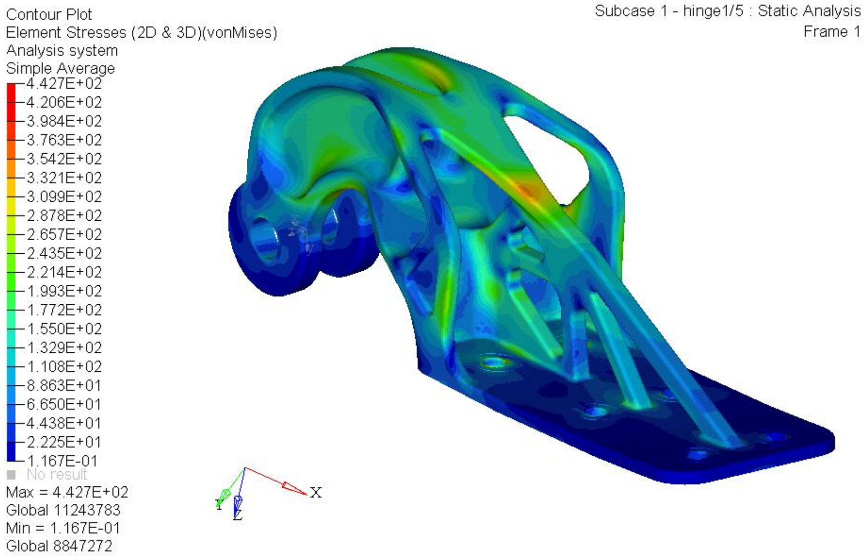

Figure 8: FEA of final hinge design

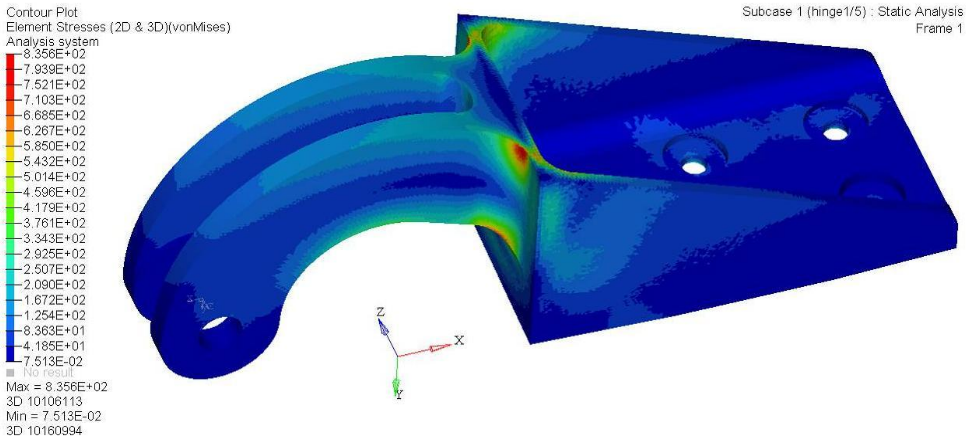

Figure 9: FEA of original hinge for comparison for comparison

Stress in the final part is much more homogenous **(Figure 8, Figure 9)**, indicating that there is a much more efficient use of material in the optimized part.

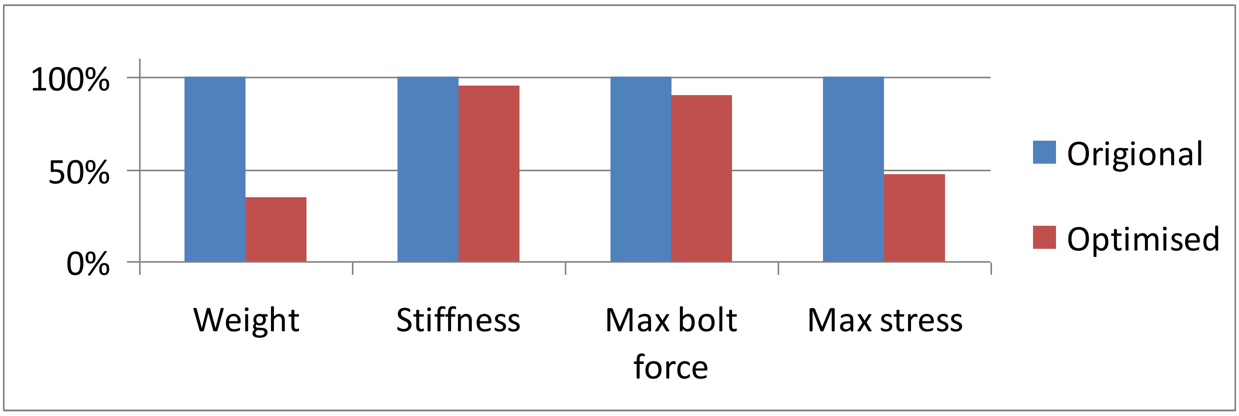

Figure 10: Performance comparison of original and new design

The optimized design weighs only 326g, compared to 918g in the original. This is a reduction of 64%, although the change in material accounts for roughly half of this change in mass.

7.0 Discussion / Conclusions

The weight saving is small in proportion to the amount of work spent on the optimization and design process. Optimization in Airbus so far has focused on large structural members (e.g. wing box ribs **[1]**) because taking small percentages of weight from large parts delivers large weight savings. This project highlights the potential there is for optimization of smaller parts across the whole aircraft, although these potential weight savings are spread across thousands of parts.

Ideas for exploiting these potential weight savings include: Creating families of similar parts, all using variations of the same topology, thus spreading the cost of optimization between parts. Designs could be created procedurally for a certain type of part. Or the time involved in optimization could be reduced by developing tools to automate or assist in time-intensive parts of the optimization, improved tools to assist in interpretation and sizing optimization.

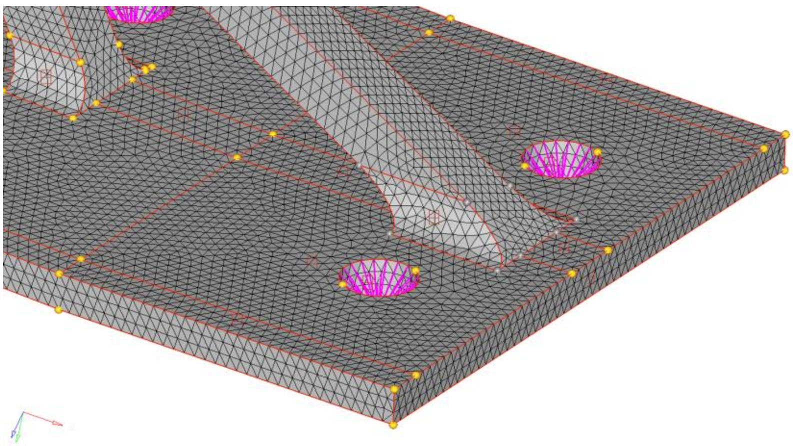

Figure 11: Mesh morphing

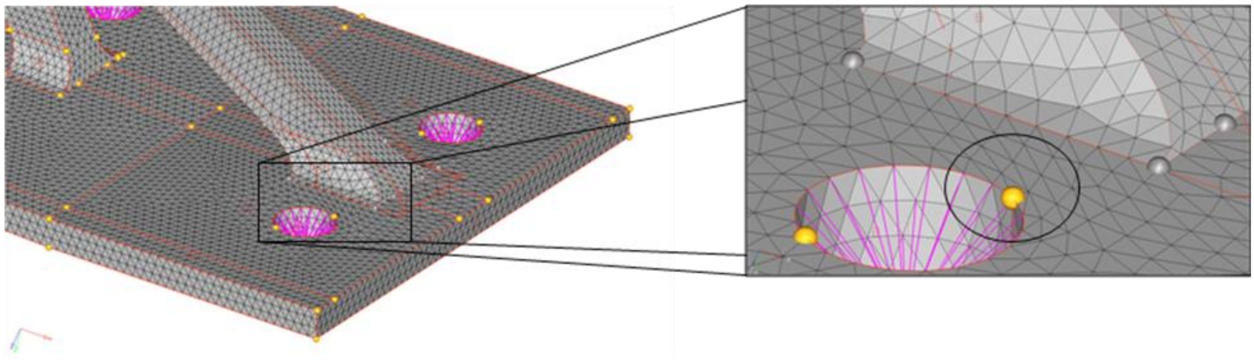

Figure 12: Problems with mesh morphing

The main problem in this project was in shape and sizing optimization. As can be seen in **Figure 11** and **Figure 12**, mesh morphing isn't ideal for large deformations of structures with complex topology, and cells can be distorted so that remeshing is necessary.

Constraints in topology optimization are difficult to design. In this project the bolts were constrained in translation in all directions, i.e. infinitely stiff. This, while being a very common approximation can cause problems in topology optimization. If stiffness is a driving force in the optimization, material will gravitate to constraints because of this artificial stiffness. In this case, the original hinge was taken as a baseline and the loadings on the constraints were distributed at least as evenly as the baseline which was considered to be a reasonable approach on the basis of equivalence.

8.0 References

[1] **'Topology Optimization of Aircraft Wing Box Ribs'**, L Krogg and M Kemp, The Altair Technology Conference 2004 (UK), 5, 2004.

[1] **'Altair HyperWorks v10'**, Altair Engineering, 2010.