Redesign of a Bell Crank to Ensure Compliance with Von-mises Failure Criterion

Ikpe Aniekan[1], Owunna Ikechukwu[2], & P.O. Ebunilo[3]
Email Address: ikpeaniekan@gmail.com; ikechukwu.owunna@uniben.edu;
Patrick.ebunilo@uniben.edu
Department of Mechanical Engineering, Coventry University, West Midlands, UK

Abstract: For most engineering designs, failure is a pertinent issue that must be eliminated. To ensure compliance of engineering components to von-mises failure criterion which states that the maximum value of von-mises stress induced in a material must not exceed the yield strength of the material used for the design, attention must be given to this criterion during the course of designing a given component. In this report, a bell crank was redesigned to ensure compliance with von-mises requirements despite the strict customer requirement for weight and material. The bell crank was redesigned to remain within its elastic limit during loading. The analysis of the bell crank was done with a 12KN force applied on its long end. The bell crank had a square pivot and a clamp on its small end. The initial analysis gave a von-mises stress of 823MPa and a deflection of 1.36mm. Masses were added and removed from the design to ensure compliance with von-mises criterion and mass restriction and the new design were reanalysed. Analysis of the final design of the bell crank gave von-mises result as 234MPa with a deflection of 0.498mm, which is less than the material yield strength of 250MPa. Hence, the design was considered safe because the final weight of the new design gave a value of 1.133kg and was able to withstand a force of 12KN.

Keywords: Bell Crank, Failure, Motion, Redesign, Safety, Stress criterion, Weight.

◆

1.0 Introduction

Engineers are constantly working hard to reduce material wastage. This is often achieved either by limiting wastage during production or improving material selection with better selection criteria [8]. In cases where the material for the design cannot be changed, the weight of the design is limited to the barest minimum. To ensure that the specific strength of the design is maintained while reducing the overall weight of the component, structural optimization is often carried out. Structural Optimization has become a vital tool for design engineers. Its help in ensuring that unnecessary features and materials are removed from a component without tampering with the function of the component makes it widely accepted.

To ensure that the design meet the customer requirements, the load case scenario depicting the position of the member during operation can be modelled using finite element analysis software such as HyperWorks, CATIA or ANSYS as the case may be [3, 7, 6 5]. The best mesh size possible is one that offers high accuracy for the analysis result and minimizes run time. As shown in figure 1, a bell crank is a type of crank that enables changes in motion at angles such as 90°, 180° and 360° respectively. However, the wider the angle extended by the crank, the more non-linear the direction of motion which builds up high stresses at the angles over time. Hence, there is need to redesign the bell crank by ensuring that the stresses induced in the bell crank material does not exceed its yield strength to avoid failure. A typical device that uses bell crank is a bicycle brake where the force coming from the handlebar lever is rotated at an angle of 90^o to push the brake block against the wheel rim [2]

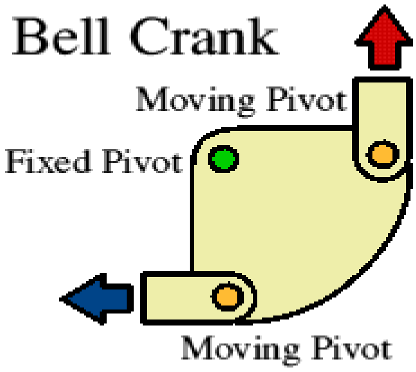


Figure 1: Typical example of a bell crank [2]

2.0 Initial Load Case

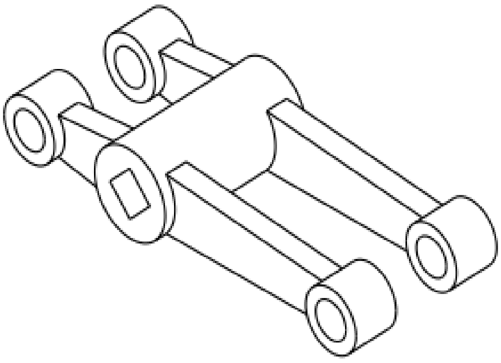


Figure 2: CAD Model of In-line Bell crank

Figure 2 shows the design of the in-line bell crank. The bell crank is expected to be made of steel which has yield strength of 250MPa and young's modulus of 200GPa. The bell crank is expected to be pivoted on square central hole, a representation of the real life scenario where the square central mount is used for the hole. The bell crank is setup to carry a 12KN force on the longer arm of the crank in the Z-axis. The loaded member is constrained at the hole on the shorter arm of the crank. The constraints on the bell crank may be inaccurate but are a good approximation of the load on the crank during operation. The mount holding the bell crank at the square pivot hole may deflect under loading and as such the result of this analysis may be rendered inaccurate. The level of error in the data would however be very minimal because minimal deflections are expected at the pivot.

The constraints on the member restrict the body in all the six degrees of freedom. The pivot fixes the square hole in the Z and Y axes only other forms of degree of freedom are permitted. The analysis of the initial model was prepared using Finite Element analysis approach. The Finite Element Analysis (FEA) was prepared from CATIA and the results for the von-mises plot and the deflections were generated. The initial model weighs 0.914kg +/- 0.05%. The maximum von-mises stress noticed from the initial design is 562MPa. The displacement in the member was a maximum of 1.02mm. Other finite element solvers gave similar results for the analysis and thus the results are taken to be as accurate as possible. There is a direct relationship between stress and displacement and thus reducing one will reduce the other. The subject of this design is to reduce the stress on the bell-crank without exceeding a specified weight limit. The test case used in this report is shown in figure 3.

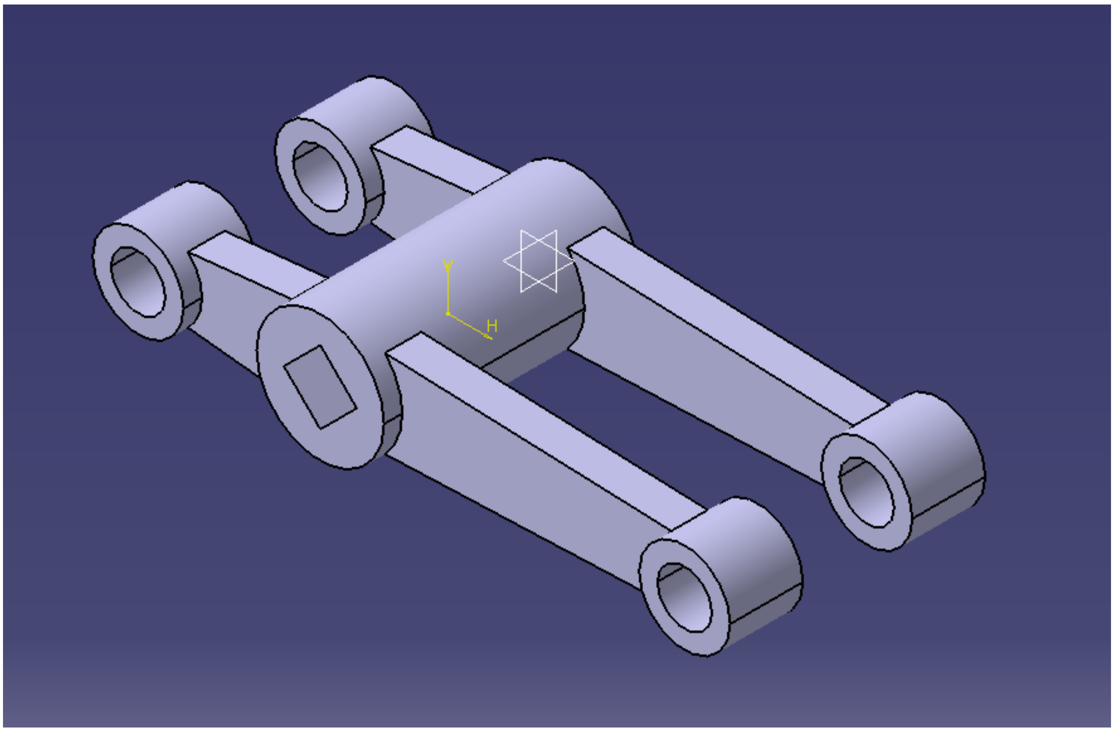


Figure 3: Bell crank initial load case

Numerical computation models are used by finite element analysis solvers to evaluate specific mechanical properties of engineering designs under a load system. Using FEA solutions ensure that physical testing is limited to the final stage of engineering projects. This can help to reduce project time and cost while ensuring standards are met adequately. Redesigning existing projects have however become easy and cost effective and sophisticated researches are possible with the use of FEA solvers. At the advent of FEA solvers, linear solvers were used. The sophistication of engineering designs and improvement in engineering materials meant that linear systems could not account for plastic deformation in materials as well as time dependent analysis [4, 8]. Non-linear system was developed to meet this need. Non-Linear systems can be used for fatigue testing and cyclic testing as well as material testing where material behaviour is required all the way to failure.

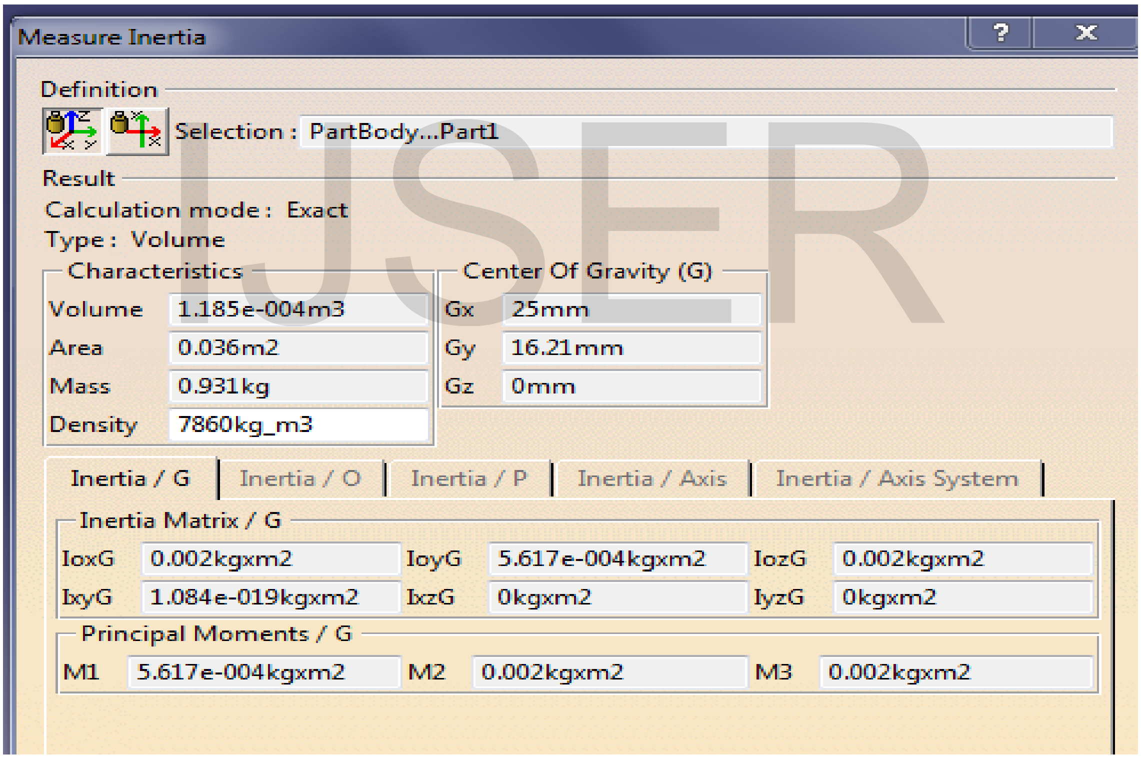
Nodes define a shape in finite element analysis. A system of nodes is referred to as mesh. Meshes are generated on a model to be analysed by FEA solvers [5, 6]. The meshes define the configuration of the model while retaining the various properties of the model that define the reaction of the model to lines of force. Two categories of meshes exist. 3D meshes are solid in nature and impact 3D properties and profiles on the model. They produce more accurate results. A major drawback of this type of meshes is the high computation time. 2D meshes are relatively less cumbersome and easy to generate when compared to 3D meshes, it applies 2D elements/profiles in creating the model and improve the computation time. Stress concentrations can be reduced by using same mesh size in a single model. Using appropriate size of mesh will ensure increased accuracy and reduces computing time. A mesh convergence study was carried out to determine the appropriate mesh size. The curve suggests a mesh size of 5mm to ensure relatively accurate results for stress and displacement.

3.0 Discussion

3.1 Moment of Inertia

The resistance of any component to changes due to rotation or bending is termed moment of inertia. The axes of reference where the measurement is taken, depends on the axis from which the centroid is evaluated.

The deflection of any structure under a system of forces is dependent on the second moment of area [1]. Figure 4 shows the measurement of the moment of inertia of the initial bell crank. To ensure that the bell crank is modified to ensure it meets the stress criterion, the bending moment can be lowered or the moment of area improved.



Selection : PartBody...Part1

Calculation mode : Exact

Type : Volume

Volume 1.185e-004m3

Area 0.036m2

Mass 0.931kg

Density 7860kg_m3

Gx 25mm

Gy 16.21mm

Gz 0mm

IoxG 0.002kgxm2

IoyG 5.617e-004kgxm2

IozG 0.002kgxm2

IxyG 1.084e-019kgxm2

IxzG 0kgxm2

IyzG 0kgxm2

M1 5.617e-004kgxm2

M2 0.002kgxm2

M3 0.002kgxm2

Figure 4: Measure of Moment of Inertia

3.2 Design Requirement

The requirements for this design are that of stress and failure criterion. The stress on the members is expected to be below the yield strength of the material for the bell crank. The bell crank is to be made of steel whose yield strength is 250MPa. By ensuring that the bell crank meets the Von-mises failure and yield strength criterion, the safety of the machine during operation is ensured. In the bell crank redesign, the strength of the material and the factor of safety will be taken into consideration. The bell crank will be redesigned to support a 12KN force in the negative Z-direction. The welds for the joining of the various parts are not taken into consideration as the material is assumed to be made by casting or machining of a large chunk of material rather than adding small components.

3.4 Factor of Safety and Von-Mises Stress

For the purpose of this re-design, the Factor of Safety (FS) is given as 1.6. The factor of safety is described as the ratio of failure load to the design load. It can otherwise be defined as the ratio of the yield stress (strength) of the design material to the allowable or design stress (strength) of the body in question. The load at which a body under a system of forces fails is termed the failure load or the ultimate load. The working load for the body is termed the design load. In real life design, the factor of safety is estimated by taking the ratio of the tensile strength of the material for the design to its yield strength. In applying the factor of safety to this design:

$$\text{Factor of safety (FS)} = \frac{\text{Yield strength}}{\text{Allowable or design Stress}} \quad (1)$$

Therefore,

$$\text{Allowable or design Stress} = \frac{\text{Yield strength}}{\text{Factor of safety (FS)}} \quad (2)$$

$$= \frac{250\text{MPa}}{1.6} = 156.25\text{MPa}$$

The redesign of the bell crank will therefore be done to ensure that the stresses induced in the material are less than the material yield strength. The von-mises stress is an important criterion in design for ductile materials to analyse failure. Von-mises stress helps predict whether any point in a design is experiencing stresses that are very high. At any given point in time when the von-mises stress in a material is higher than the yield strength, failure occurs.

3.4 Elastic Region & Ultimate Tensile Strength

The stress level that indicates the advent of plastic deformation in a material is referred to as the elastic limit or the yield strength of the material. This refers to the maximum stress a material can sustain, beyond which the material cannot return to its original state. Deformation of any material is linear or elastic before the yield point [4, 8]. The elastic region is therefore defined by the stages in the deformation of a material that exist before the advent of plastic deformation. This design is expected to remain in its elastic region while withstanding the applied load.

4.0 Development Process

4.1 Previous Designs

Examples of existing bell crank are shown in figures 5 and 6. Figure 5 shows a bell crank used in the suspension systems of racing cars. Figure 6 shows an Aircraft Bell Crank. There are common issues that the design of the bell cranks in figure 5 and 6 show and they are highlighted as follows:

i. Different shapes of holes can be created along the bell crank arms to reduce weight

ii. Smooth edges rather that sharp ones can prevent stress concentration

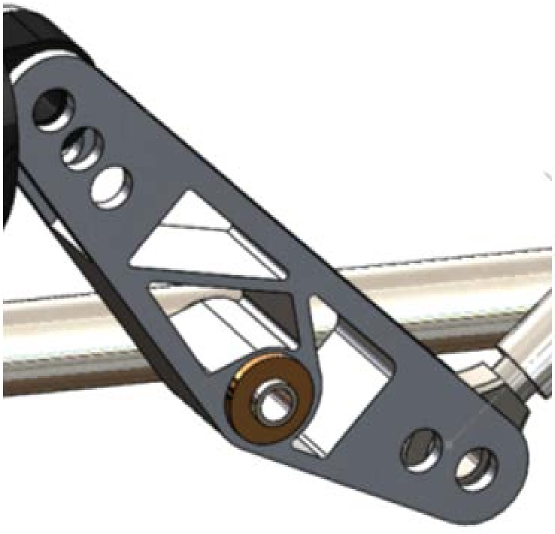


Figure 5: Real Life Example of Suspension Bell Crank [9]

Figure 6: Example of Aircraft Bell Crank Design [1]

The weight requirement and the mechanical properties of the design material will have a large effect on the redesign process. However, the cross-section of the structure has a role to play in both factors and it is most times the maximized or minimized function in redesign of the bell crank. This is done by removing mass from some area and adding mass in others.

4.2 Design Improvements

The connecting edges of the bell crank retained their shapes and sizes all through the re-design. This is basically because they are connection points for other parts of the system in which the bell crank functions. The design material remained the same throughout the process. Since the product is expected to be either machined or cast, there were no considerations taken for joining materials.

The final design concept has a mass of 1.133kg which meets the design criteria for mass. Because of the nature of the new design, the centroid changed values on the y-axis while the x-axes centroid remained the same. The final design ensured that the bell crank remains within its elastic limit after performance. The maximum stress on the bell crank is 234MPa with a deflection of 0.498mm. Different approaches were employed in re-designing the bell crank; this led to a few failed designs. The final factor of safety of the design was 1.07.

4.3 The Final Design

To further modify the bell crank, the properties of the bell crank can be parameterised and then used in running optimization procedure to ensure a reduced material consumption. The maximum stress attained on final model of the bell crank is 234MPa with a deflection of 0.498mm.

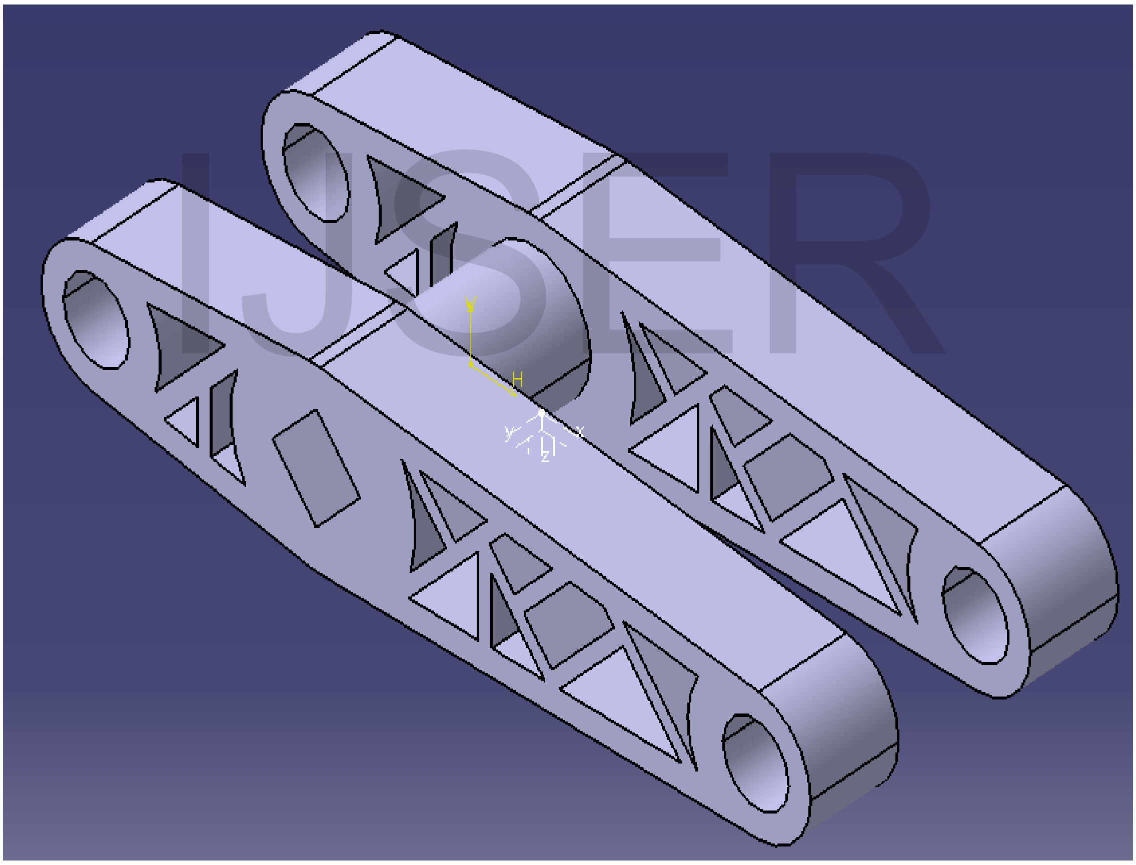


Figure 7: Final Design of the Bell Crank

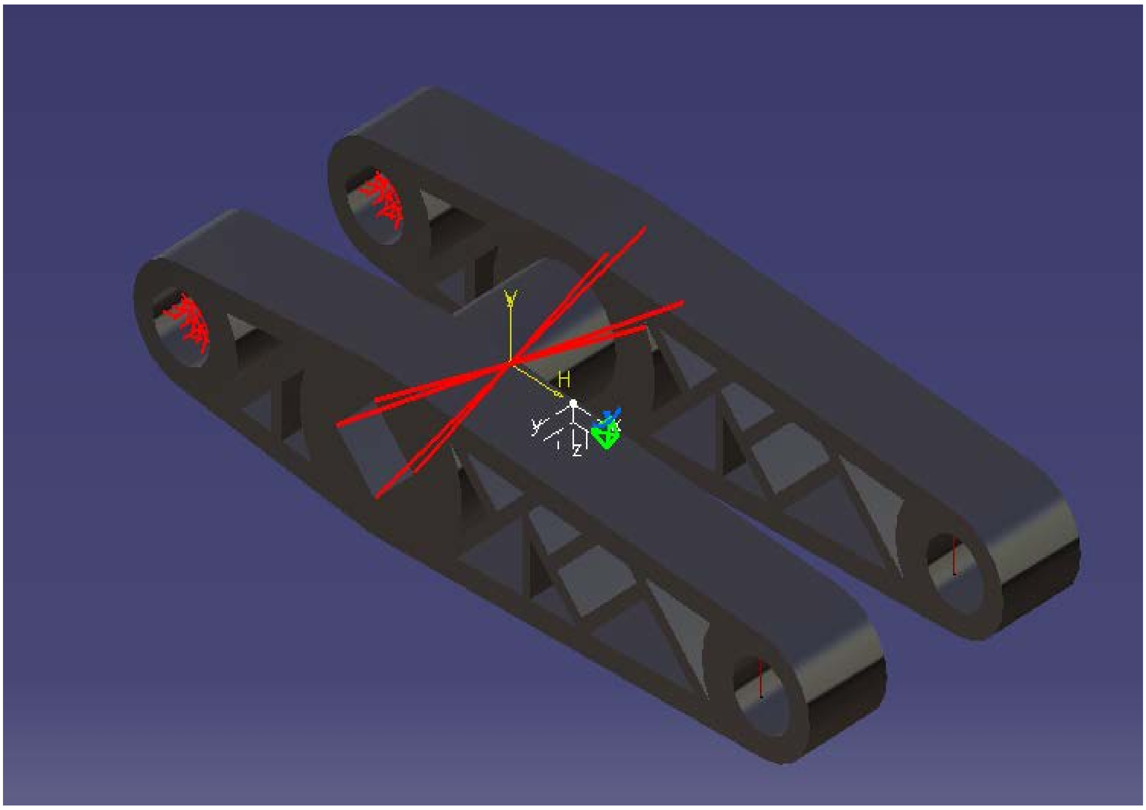


Figure 8: Load case of final design of bell-crank

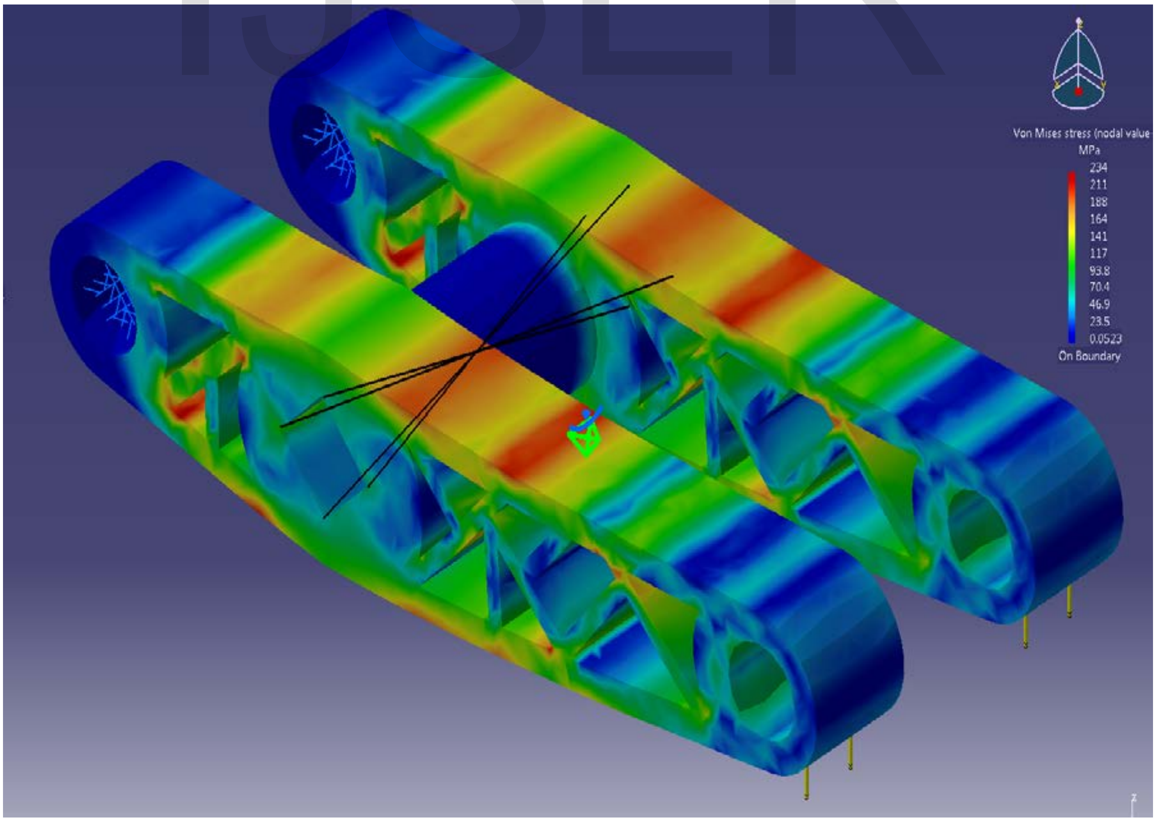


Figure 9: Von-Mises of Bell crank

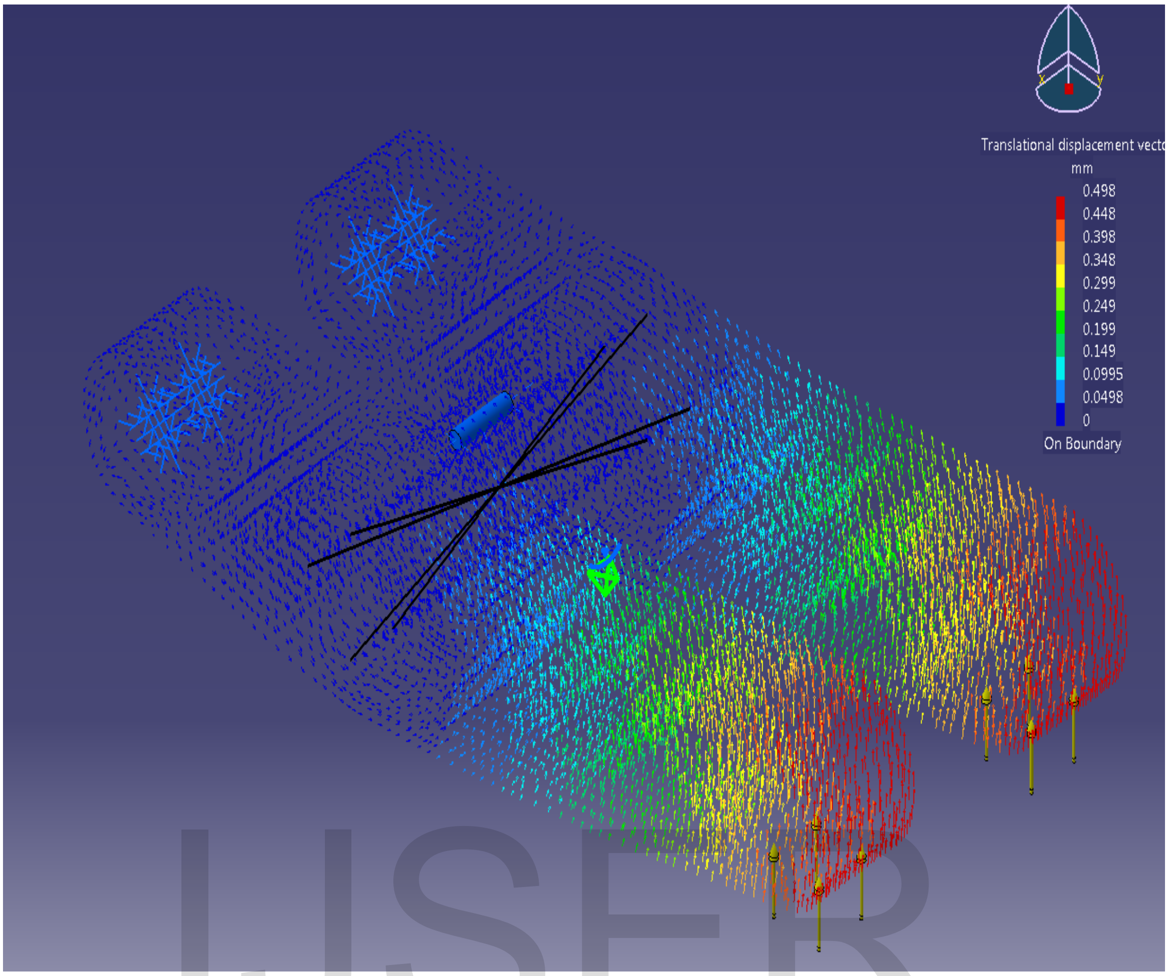
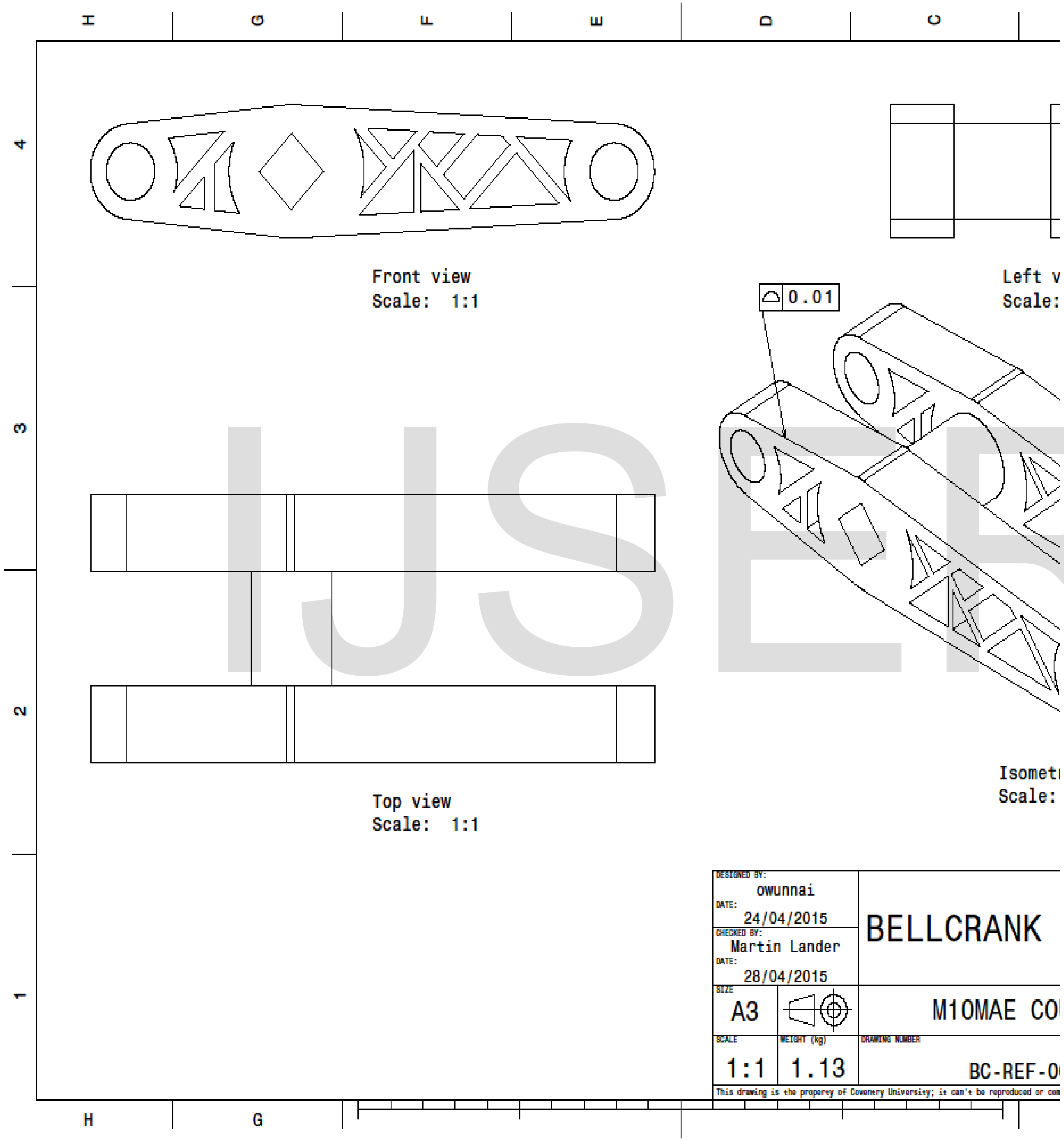
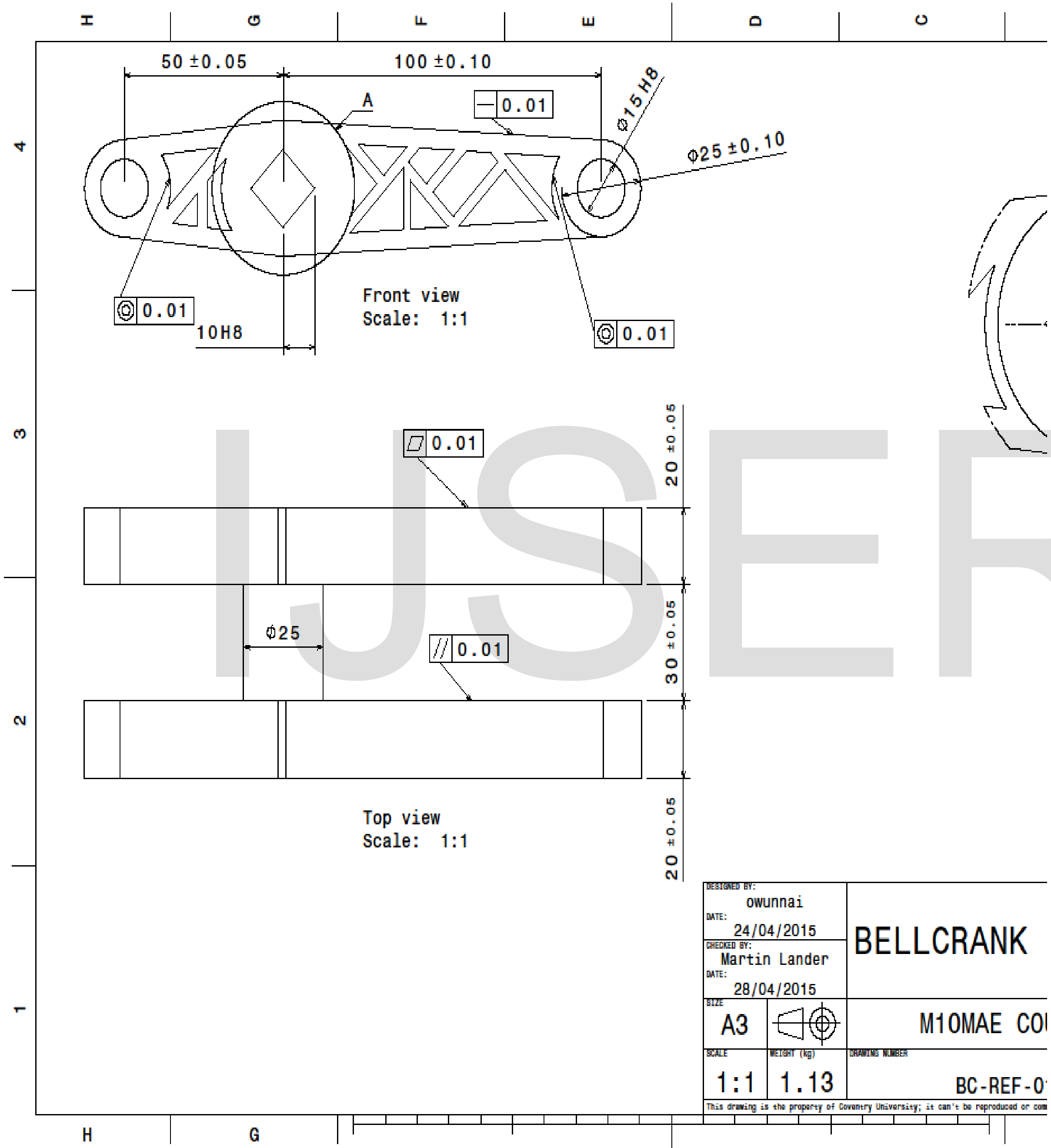
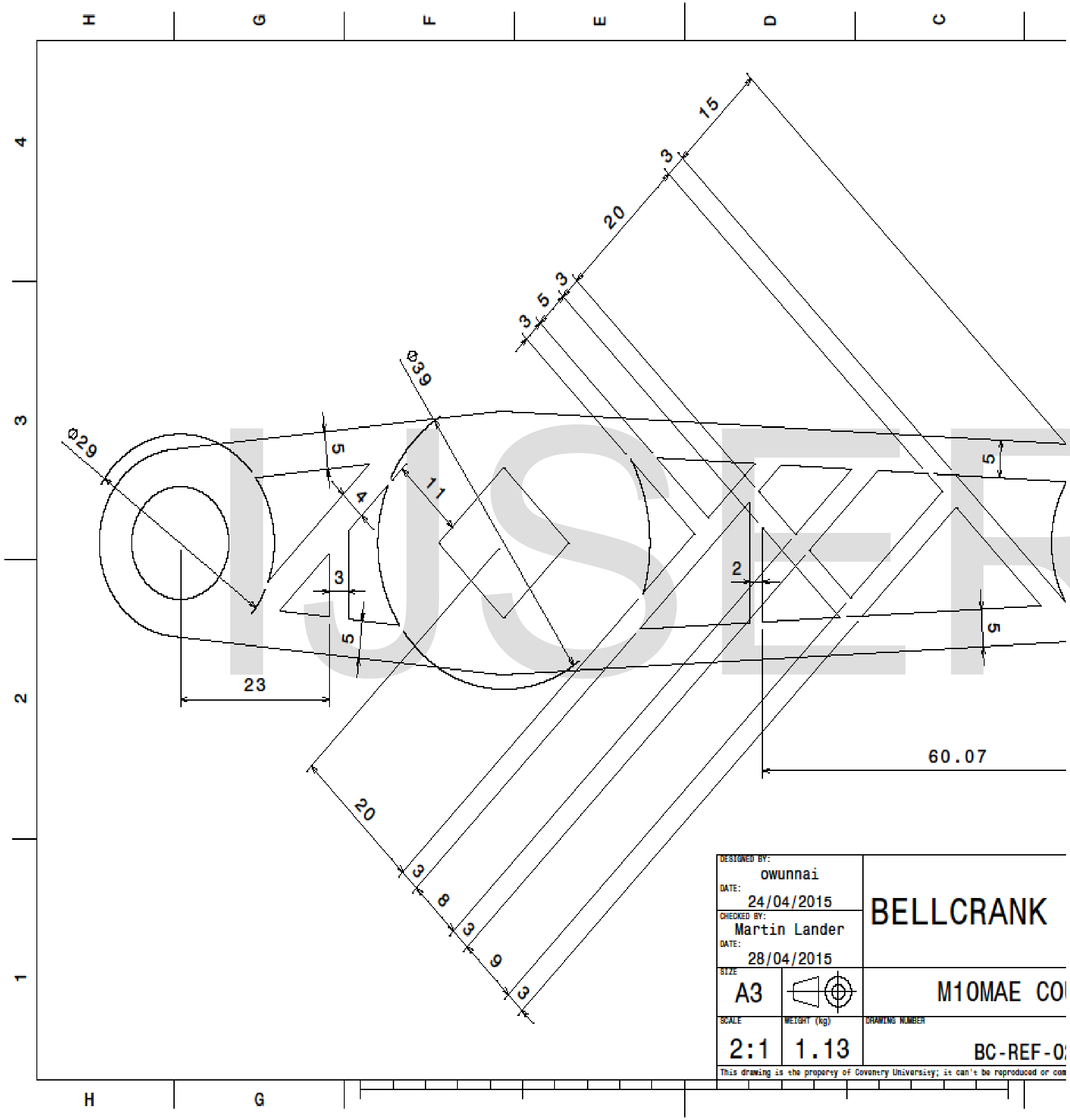


Figure 10: Displacement of bell crank

5.0 Final Design Drafts


H
G
F
E
D
C
4
3
2
1
Front view
Scale: 1:1
Left v
Scale:
0.01
Top view
Scale: 1:1
Isometr
Scale:
DESIGNED BY:
owunnai
DATE:
24/04/2015
CHECKED BY:
Martin Lander
DATE:
28/04/2015
BELLCRANK
SIZE
A3
M10MAE CO
SCALE
1:1
WEIGHT (kg)
1.13
DRAWING NUMBER
BC-REF-0
This drawing is the property of Coventry University; it can't be reproduced or com
H
G


H
G
F
E
D
C
4
3
2
1
50 ±0.05
100 ±0.10
A
0.01
Ø15 H8
Ø25 ±0.10
0.01
10H8
0.01
Front view
Scale: 1:1
0.01
20 ±0.05
Ø25
0.01
30 ±0.05
20 ±0.05
Top view
Scale: 1:1
DESIGNED BY:
owunnai
DATE:
24/04/2015
CHECKED BY:
Martin Lander
DATE:
28/04/2015
BELLCRANK
SIZE
A3
M10MAE CO
SCALE
1:1
WEIGHT (kg)
1.13
DRAWING NUMBER
BC-REF-0
This drawing is the property of Coventry University; it can't be reproduced or com
H
G


H
G
F
E
D
C
4
3
2
1
Ø29
Ø39
15
3
20
3
5
3
11
5
4
3
5
23
2
5
5
60.07
20
3
8
3
9
3
DESIGNED BY:
owunnai
DATE:
24/04/2015
CHECKED BY:
Martin Lander
DATE:
28/04/2015
BELLCRANK
SIZE
A3
M10MAE CO
SCALE
2:1
WEIGHT (kg)
1.13
DRAWING NUMBER
BC-REF-0
This drawing is the property of Coventry University; it can't be reproduced or com

6.0 Conclusion

To further develop the Bell crank, optimization techniques may be employed rather than taking manual iterations. Optimization software such as HYPERMESH, ANSYS or other finite element analysis solvers can be used in further processing the re-design of the bell crank. The optimization process can help to ensure that reinforcements are made where necessary and that material are removed to ensure weight compliance. Load paths can be generated from the optimization process and ensure that mass is placed of the path where the force travels through. The maximum value of stress on the bell crank after redesign was an estimated 234MPa. This ensured that the design had a factor of safety of 1.06. A mesh of desirable size was employed to help maximize accuracy and computer processing time. Error is possible in the design as an accumulation of approximations during the computation

7.0 References

[1] BreSeight Group (n.d) Aircraft Bell Crank [online] available from <http:www.advancedmanufacturing.com.au/news/Articles/nov2013.html> [22 March 2015]

[2] D&T (1997) Mechanisms: Bell Crank. [online] available from <http://www.dtonline.org/apps/infopage/app?1&6&1&20&0&0> [22 March 2015]

[3] Bastien, C. (2013) FEA in Continuum. University of Coventry: Lecture Note.

[4] Bucciarelli, L. L. (2009) In Engineering Mechanics for Structures. Dover Publications

[5] De Borst, R., Crisfield, M. A., Remmers, J. J., & Verhoosel, C. V. (2012). In Non-Linear Finite Element Analysis of Solids and Structures.

[6] Wiley-Blackwell. Department of Aerospace Engineering. (2013). Introduction to Finite Element Methods. [online] available from <http://www.colorado.edu/engineering/cas/courses.d/IFEM.d/> [2 January 2015]

[7] Finney, R. H. (2011). Finite Element Analysis. In A. N. Gent, Engineering With Rubber: How to Design Rubber Components (pp. 8-9). Hanser Publishers, Frederick

[8] A. L., & Dominic, J. D. (2009) Strength & Stiffness of Engineering Systems. Springer.

[9] Lancaster Racing (2014) Suspension Update 1. [Online] available from <http://www.engineering.lancs.ac.uk/lancasterracing/2014/02/17/suspension-update-1/> [22 March 2015]

D&T Mechanisms (1997)