



Study of the effect of a chamfer on the small end of a Connecting rod and the effects of Boundary conditions in FEA

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Connecting rod investigation

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Declaration

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Abstract

The focus of this paper is the design of moving components with in an IC engine. The specific components of interest are those which are connected to either end of the connecting rod; the piston pin and the crankshaft. The author will investigate the effects of adding a chamfered edge to a connecting rods small end. The secondary part of the paper will examine static studies in an FEA package.

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4 Nomenclature

S (m)	Displacement
R (m)	Crank Throw
L (m)	Connecting rod length
W (rad/s)	Angular velocity
I (m ⁴)	Inertia
Rθ (degrees)	Rod angle

$C\theta$ (degrees)	Crank angle
I_r (m^4)	Rod inertia
F_s (N)	Side Force
F_r (N)	Force on connecting rod
F_c (N)	Force on crank
N (N/m^2)	Normal stress
S (N/m^2)	Shear stress
F (N)	Normal Force
A (m^2)	Area
σ (N/m^2)	Stress
T (N/m^2)	Shear Stress
$P_{1,2,3}$	Principal stress
X,Y,Z	Cartesian coordinates
M (kg)	Mass
Y (m)	Distance to centre axis
K (N/m)	Stiffness
E (Gpa)	Youngs modulus
X (m)	Change in length
B (m)	Breadth
h (m)	Hight

5 Aims

- The aim of this report is to examine the benefits of chamfered edge on the small end of a Ford connecting rod.
- Examine the approach used in obtain results with in a static FEA
- Examine the effect of fillets on stress reduction in a crankshafts crank pin bearing.

6 Objective

To discover the effect a chamfer may have on stress in the small end using solid works

To carry out hand calculations to validate FEA software

To use FEA software techniques to validated static studies

To investigate the effect of boundary conditions within a FEA package

7 Introduction

The purpose of a connecting rod is to turn reciprocating motion into rotational motion. It does this by connecting the piston, the reciprocating part to the crankshaft, the rotating part. When the air fuel mix in the chamber is combusted the rise in pressure forces the piston down the cylinder, this in turn rotates the crankshaft.

This report will be analysing the connecting rod, piston pin and crankshaft from the Ford eco boost. Ford had to redesign the connecting rod as the piston pins were failing. In the original design, the little end was a straight ring for the piston pin to enter but the original design was causing the piston pins to fail. The failure was propagating from the top of the piston pin near the connecting rod. The new designed connecting rod does not have this problem, the little end now has a chamfered edge and is tapered towards the top. The author will show what effect the chamfered edge was having on the stress within the piston pin using FEA methods and hand calculation.

Due to technical difficulties the paper will also study the crankshaft, using this as a base to show how important it is to ensure any FEA studies are done with the correct boundary conditions and

the effect incorrect boundary conditions can have on the results. Such conditions should represent the way in which the part is fixed in a real world situation.

7.1 Background

The crankshaft for the Ford eco boost is a four-cylinder, five bearing shaft. The crank has drilled passages that allow oil to pass from the main journals to the big end bearings.

The bearing surfaces are hardened and have the edges to the webs fillet rolled (Stone, 1999), or a counter fillet whose surface has been put into compression, adding fillets to inside corners reduces the growth of cracks and improves the fatigue life, cite here. Parts don't usually fail under compression, it is tensile forces and shear forces that cause failure. To increase the amount of tension a part can withstand it is common to put the surface into a compressed state, this can be done by rolling or shot peening.

8 Literature Survey

In order for the author to understand the subject it was important to carry out research. It became apparent that finding research on this particular subject would be very difficult despite having access to multiple online libraries. There were lots of papers which investigated a related subject to this papers, i.e. crankshaft vibration, piston pin design, stress in a connecting rod, but the author was unable to find a single paper which touched directly on the subject in some way shape or form. The particular scenario this paper is concerned with is quite a complex one. There were multiple stress theorem applicable.

9 Pressure Data

TABLE 1 NOTATIONS

Notation	Meaning
S	Displacement
R	Crank Throw
L	Con rod length
w	Angular Velocity
Rθ	Con rod Angle
Cθ	Crank angle
r _{mass}	Rotating mass
F _c	Compressive force
F _s	Side force
F _r	Rod force

To calculate the stress within the piston pin and crankshaft the forces acting on them needed to be known. The pressure data from the Ford eco boost was used to determine the net compressive force and the inertia for the piston pin and crankshaft. The forces acting on the connecting rod were already known as they had been calculated in an excel spread sheet and used in another report. The Equations (1) through to (7) (Tobin, 2017) gave the net compressive force and inertia acting on the connecting rod. To acquire the net compressive force acting on the piston pin the value for mass needed to be adjusted in the inertia calculation as only the piston and piston rings would be acting on it.

$$S = R + L - R \times \cos\theta - L \times \cos\phi \quad (1)$$

$$Velocity = Rwsin\theta + \frac{R^2}{L}w \times sin\theta \times cos\theta \quad (2)$$

$$Vertical\ acceleration = Rw^2cos\theta + \frac{R}{L} \times Rw^2cos2\theta \quad (3)$$

$$Inertia = I = Mass \times Acceleration \quad (4)$$

$$Net\ force = Gas\ force - Inertia \quad (5)$$

$$R\theta = sin^{-1} \left[\left(\frac{R}{L} \right) \times sin\theta \right] \quad (6)$$

$$Fr = \frac{f_{net}}{cos\phi} \quad (7)$$

Table (2) shows the properties of the Ford eco boost engine and the material the connecting rod is made from. The author was unsure precisely what material was used to make the connecting rod. The author assumed that it was made from a type of steel based on the figure for ultimate tensile strength. What little was known about the material used was the ultimate tensile strength and Youngs modulus, the values of which were provided by Malcom McDonald. Because the exact material used were unknown assumptions had to be made based on what little information was available. Using the solid works materials library and the understanding that a crankshaft is usually made from steel a material that best matched the UTS was chosen. The same assumption was carried out for the crankshaft also. The main focus of this study was the effect of a chamfered edge on stress and as such the material was inconsequential. The author decided it would safe to make assumptions with material choice as long as the materials chosen remained the same throughout the study.

TABLE 2 CONNECTING ROD SPECIFICATIONS

	Mm	M		Material	
Bore	79	0.079		UTS (N/mm ²)	750
stroke	80.14	0.08014		Youngs modulus (N/mm ²)	170000
throw	40.7	0.0407			
Con rod length	134	0.134			
Compression ratio	11.0:1				
Bore centre distance	87	0.087			
Component weight					
	G	kg			
Con rod	410	0.41			
Piston	268	0.268			
Piston Pin	80	0.08			
circlip	1.66	0.00166			
Total	452.16	0.45216			

The same excel spread sheet was used to calculate the forces and induced stresses on the crankshaft, however further calculations needed to be carried out. The net force acting on the crankpin journal as described in equation (5) is gas force – inertia.

The crankshaft has two inertia components, a vertical inertia and a rotational inertia. The vertical inertia is caused by the mass of the piston, piston pin and 1/3 of the connecting rod accelerating vertically within the cylinder; and the rotational inertia is caused by the mass of 2/3 of the connecting rod.

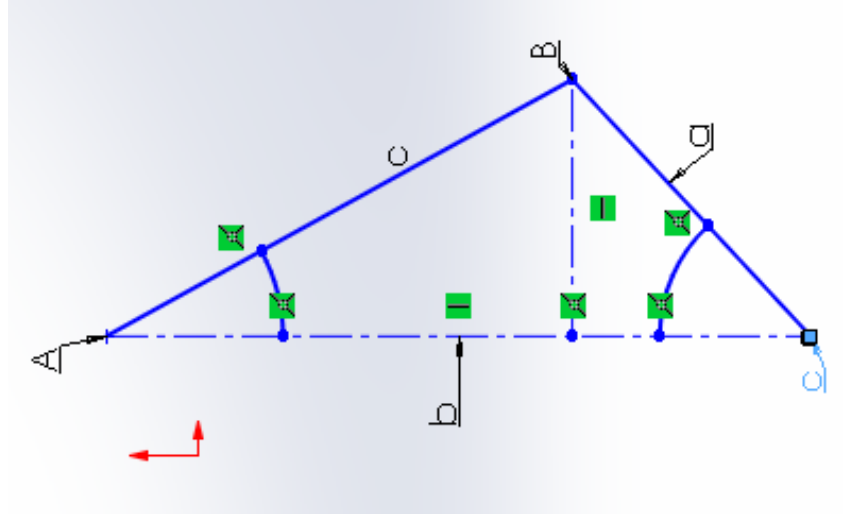


FIGURE 1 SINE RULE

Figure 1) shows a labelled non right-angled triangle, it is labelled with the notations of the sine rule. The triangle represents a connecting rod and crank shaft at any point during rotation. Equation (10) uses the sine rule to determine the angle of the connecting rod. In this case (a) is the crank throw, (c) is the connecting rod length, (b) is the displacement and the capital letters represent the angles between the sides. The sine rule requires knowing two sides and one of the angles opposite.

Law of sines (H.W.Morrow, 2004)

$$\frac{\sin A}{a} = \frac{\sin B}{b} = \frac{\sin C}{c} \quad (8)$$

When equation (8) is rearranged with known values to calculated angle A the result is equation (9)

$$A = \sin^{-1} \frac{a}{c} \times \sin C \quad (9)$$

Equation (10) is equation (9) with the substituted relevant terms.

$$R\theta = \sin^{-1}\left(\frac{R}{L}\right) \times \sin(C\theta) \quad (10)$$

Equation (11) is for the mass moment of inertia. The moment of inertia determines the amount of torque needed to rotate a ridged body at an angular acceleration about its axis. It is dependent on the mass and mass distribution. The further away the mass is from its axis the larger the moment. There are three components to equation (11), the total rotating mass, angular velocity squared, and the crank angle. The Cos (θ) resolves the crank angle

$$Ir = r_{mass} \times R w^2 \times \cos(C\theta) \quad (11)$$

Equation (12) relies on (SOH-CAH-TOA) specifically rearranging of $T\theta = O/A$ for the value of O

$$Fs = \text{Net compressive force} \times \tan(R\theta) \quad (12)$$

$$Fr = \frac{\text{Net compressive force}}{\cos(R\theta)} \quad (13)$$

$$Fc = (Fr \times \cos(R\theta)) - \text{rotating inertia} \quad (14)$$

$$\text{Horizontal force crank} = Fr \times \sin(R\theta) + \text{mass} \times R w^2 \times \sin(C\theta) \quad (15)$$

Equation (16) was derived from Pythagoras theorem. Fc and Fr were resolved to produce the net force acting on the crank journal. Equation (17) gave the angle at which the resolved force was acting.

$$F_{crank} = \sqrt{Fc^2 + Fs^2} \quad (16)$$

$$\text{Direction} = \tan^{-1}\left(\frac{Fc}{Fs}\right) \quad (17)$$

The equation used in excel to calculate the direction the resolved forces were acting in was different from what is stated in equation (35), the excel version is shown in equation (36).

$$\text{Direction} = \text{Degree}(\text{ATAN2}(X_num, Y_num)) \quad (18)$$

Excel works in radians not degrees, which was why it was necessary to convert all angles to radians before using them in any excel formula. Equation (18) converts the given radians value in to an understandable degree value, the equation gave a value between 0 and 360 degrees making it easy to see which direction the force was acting. Equation (17) would only give a positive or negative value between 0 and 90 degrees.

10 Stress

Normal stress acts perpendicular to the area on which the force is acting and a shear stress acts parallel with the area, normal stress can be either tensile or compressive (Gere, 2002). There are two types of force that act on a body, forces that act on the surface like pressure and those that act on the volume of the body like gravity or inertia. (Emilia, 2006)

Forces that act on the surface of a body are resolved into three components. These are X,Y,Z and when parallel to the coordinate axis are called normal stress.

When looking at τ_{zy} and σ_{zy} , the symbol denotes the type of stress, τ represents sheer stress and σ represents normal stress. The first letter there after denotes the face on which the stress is acting and the second is the direction.

Z is the axis upon which the stress is acting and y is the direction. (Timoshenko and Goodier, 1951)#

If we consider a straight bar in order to properly asses the stress with in it we need to look at an inclined section as opposed to a normal straight cross section.



FIGURE 2 INCLINED PLANE

The stress will be the same throughout the bar, and the stresses on a section will be evenly distributed. In a two dimensional element the force acting on the bar can be resolved into two components, normal and shear. Normal force acts perpendicular to the plane. Figure (3) shows the direction of normal force and shear force in relation to an inclined plane.

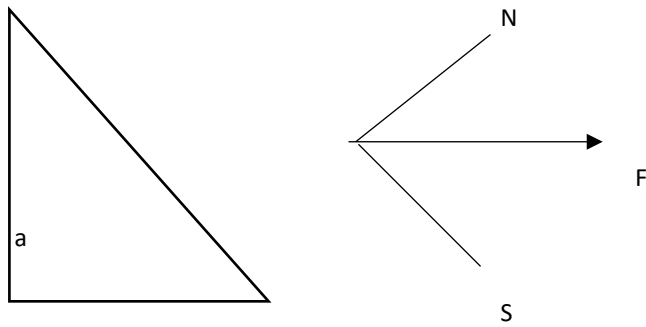


FIGURE 3 PRINCIPLE FORCES

$$N = F \cos \theta \quad (19)$$

Shear force acts tangential to the plain the equation for S:

$$S = F \sin \theta \quad (20)$$

As stress = force/area if the area of the inclined plain is known then by rearranging the formula the stress produced by the force acting on that area can be calculated. When the angle between F & N is known then by using trigonometry the area or length of the incline can be found.

$$length = A = \frac{a}{\cos \theta} \quad (21)$$

Equations (19) and (21) can be substituted into the formula for stress to give

$$\frac{F}{A} \cos 2\theta \quad (22)$$

By the use of trig identities and simplification equation (23) becomes

$$\frac{\sigma x}{2} (1 + \cos 2\theta) \quad (23)$$

The formula for the shear stress acting tangential to the incline is

$$\tau_{\theta} = \frac{-\sigma x}{2} (\sin 2\theta) \quad (24)$$

Stress elements are used to represent the stress acting on a point. The element is extremely small so the dimensions are negligible. When an element is in plane stress in the xy plane only x&y faces are subject to stress, and the shear is zero at this point. A material may yield or completely fail at an angle other than $\theta=0$ which is why it is important to consider an element at a rotated coordinates system. The shear stress maximum value occurs at $\theta=45$.

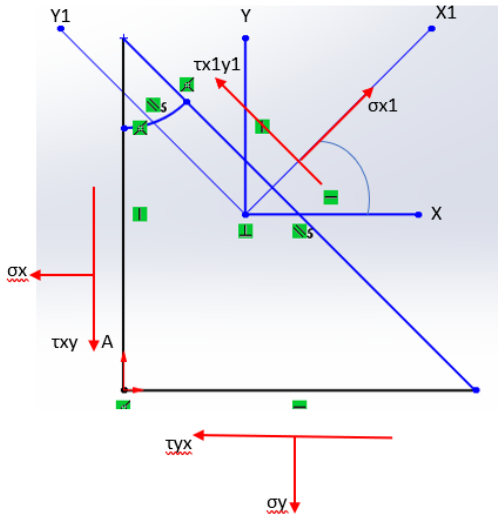


FIGURE 4 FORCES ACTING ON AN INCLINED PLANE AND DIRECTIONS

Figure (4) can be used to create transformation equations for plane stresses, σ_x , σ_y , τ_{xy} . The transformation equations can be differentiated to create equation

$$\tan 2\theta_p = \frac{2\tau_{xy}}{\sigma_x - \sigma_y} \quad (25)$$

θ_p are principle angles which are associated with the principle stress. Two values of $2\theta_p$ occur within 0-360 degrees, with θ_p differing by 90 degrees. The planes on which principal stress act are all perpendicular. Looking at equation (25) it can be seen that it's a trigonometric equation for a right-angle triangle, Using SOH-CAH-TOA to substitute values for θ_p two other equations can be made.

$$\cos 2\theta_p = \frac{\sigma_x - \sigma_y}{2R} \quad (26)$$

$$\sin 2\theta_p = \frac{\tau_{xy}}{2R} \quad (27)$$

Equations (26) and (27) can be substituted into the transformation equation. This creates the equation that calculates principal stresses. Equation (29) is the equation for principle stress (Akhtar S.Khan, 2001)

$$\sigma_1 = \frac{\sigma_x + \sigma_y}{2} + \frac{\sigma_x - \sigma_y}{2} + \frac{\sigma_x - \sigma_y}{2R} + \tau_{xy} \frac{\tau_{xy}}{2R} \quad (28)$$

This equation can be rearranged and simplified to create the equation (35)

$$\sigma_{1,2} = \frac{\sigma_x + \sigma_y}{2} \pm \sqrt{\left(\frac{\sigma_x - \sigma_y}{2}\right)^2 + \tau_{xy}^2} \quad (29)$$

Equation (35) calculates the 1st and 2nd principal stress. The principal planes upon which they act can be calculated by rearranging equation (25) for θ (Zavatsky, no date). In a 3-dimensional element there is a third principal stress to consider in the z direction. Vector is meaningless unless it is assigned a plane as there are an infinite number of planes going through any one point. (Akhtar S.Khan, 2001)

10.1 Von Mises

$$\text{Von Mises} = \sqrt{\frac{(P_1 - P_2)^2 + (P_2 - P_3)^2 + (P_3 - P_1)^2}{2}} \times \frac{1}{\sqrt{2}} \quad (30)$$

E the modulus of elasticity is the constant of proportionality. Young's modulus is the slope of the stress strain diagram in the linear part of the graph. As Young's modulus is = stress/ strain and strain is unit less Young's modulus shares the same unit as stress.

According to Von Mises theory a material will yield when the distortion energy density reaches a critical value. Since this is true for a uniaxial state as well the critical value of distortion can be estimated from uniaxial test. The Von Mises also known as the distortion energy density theory

11 Material

The material used for the main components, connecting rod, crankshaft and piston pin was 4340 annealed steel. This grade of steel is suited for shock loading or stress concentration applications such as shafts, gears, bolts, nuts, pins and couplings (AZO Materials, 2016). The material has a good ratio of strength to ductility often being used in the making of high impact parts, which are repeatedly stressed.

Properties	Metric	Imperial
Tensile strength	745 MPa	108000 psi
Yield strength	470 MPa	68200 psi
Bulk modulus (typical for steel)	140 GPa	20300 ksi
Shear modulus (typical for steel)	80 GPa	11600 ksi
Elastic modulus	190-210 GPa	27557-30458 ksi
Poisson's ratio	0.27-0.30	0.27-0.30
Elongation at break	22%	22%
Reduction of area	50%	50%
Hardness, Brinell	217	217
Hardness, Knoop (converted from Brinell hardness)	240	240
Hardness, Rockwell B (converted from Brinell hardness)	95	95
Hardness, Rockwell C (converted from Brinell hardness. Value below normal HRC range, for comparison purposes only)	17	17
Hardness, Vickers (converted from Brinell hardness)	228	228
Machinability (annealed and cold drawn. (Based on 100 machinability for AISI 1212 steel.)	50	50

TABLE 3 MATERIAL PROPERTIES 4340 (Azom, 2013)

For the bearings in the assemblies a natural rubber was used. The material was chosen for its ductile nature which allowed a controlled movement within a static study. Its specific material properties had no relationship to a bearing or oils film in a real world engine assembly

12 Connecting rod

12.1 Description

There are two types of fixture used when joining the piston, piston pin and connecting rod, (Stone, 1999). Full floating and fixed. The full floating type is free to rotate within the connecting rod and piston bushings but is prevented from moving sideward and damaging the cylinder wall by circlips at either end, fixed within grooves of the piston. This type of assembly is used most often in modern combustion engines and is what the Ford eco-boost uses. A fixed pin is unable to rotate, its locked in place by friction referred to as an interference fit. The pin is press fit into the connecting rod bushing, it has a clearance fit within the piston but has a limited arc and contact surface.

The piston pin is designed to withstand the compressive and inertial forces put upon it during a cycle. Bending and distortion need to be kept to a minimum, a thick-walled tube reduces weight while maintaining integrity. During a cycle there is some distortion and ovaling of the piston pin. The stresses within a piston pin are cyclic, compression being the overriding force during combustion and inertia during exhaust stroke. The ever-changing direction of force means the components are susceptible to fatigue.

The function of a connecting rod assembly is to transmit the combustion forces to the crankshaft, the connecting rod is subjected to compressive forces due to the pressure rise of combustion and to inertia forces from the reciprocating mass, there are also inertia forces from the rotating mass. A connecting rods mass is divided into two parts, one third is considered as reciprocating and two thirds rotating.

The initial design of small end had a bronze phosphor bushing, the width of the eye being equal from top to bottom. The piston pin was under engineered, a straight steel tube. The press fit bronze phosphor bushing pre-stresses the small eye to a small degree. The softer material of bronze phosphor acts like a bearing trapping any potential bits of metal that may have found its way in to the oil and thus between the connecting rod and piston pin. The newer design has no bronze

phosphor bushing, the piston pins surface has been coated with a DLC diamond like coating, this hard surface smooth surface combined.

The small end of the connecting rod is distorted at the top dead centre of the intake stroke as the inertia force stretch the eye. In the expansion stroke there is virtually no distortion as the pin is pushed into the connecting rod shack thus dispersing any stresses. (Emilia, 2006)

12.2 FEA

This study was a simplification of a connecting rod and gudgeon pins interaction. The mesh details of the study carried out are shown in table (3). The conditions of this study were fixed in all cases, the only change being the size of the chamfer.

The piston pin as previously mentioned has a floating connection with the connecting rod. This small gap means the piston pin is able to bend within the little eye of the connecting rod, an exaduration of this is shown in figure (6). This gap was replicated by thin layer of natural rubber as a boundary between the two surfaces. The elasticsity of the natural rubber allows the piston pin to move freely within the eye of the connecting rod. Trying to produce the same results by using boundary conditions and fixtures instead of a rubber boundary would result in large displacements causing inaccurate results or failed studies. Adding a thin rubber layer between the surfaces of the connecting rod and piston pin also greatly reduced the size of stress singularities.

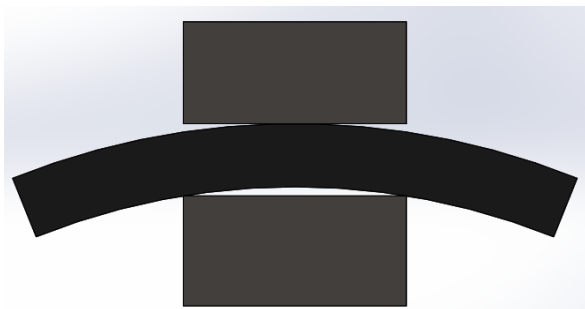


FIGURE 5 CONNECTING ROD AND PISTON PIN

The force was applied as a remote load at either end of the rod, this is not how the load is applied in an engine but it accurately simulates the effect of an applied force at the points of interest.

The force applied in this study is not representative of that which a connecting rod and piston pin are subjected to during a combustion cycle. The purpose of this study was to determine the effect a chamfered edge would have on stresses. Any effects determined would be scaleable and applicable to a connecting rod. The simple block shape is representative of a cross section of the little end, as shown in figure (6). Figure (7) shows the accurate cad version of a piston pin, connecting rod cross section.

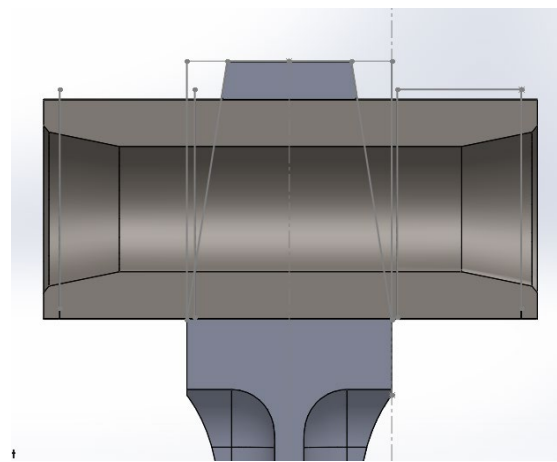


FIGURE 6 CONNECTING ROD AND PISTON PIN CAD

A force of 3000 N was applied 17mm from the connecting rod, producing a bending moment of 51N-mm. Table (3) shows the details taken from the static study, of figure (6). As it shows there was a dense mesh with a high overall quality.

TABLE 4 MESH DETAILS OF STATIC STUDY

Mesh type	Solid Mesh
Mesher Used:	Standard mesh
Jacobian points	4 Points
Element Size	1.9 mm
Tolerance	0.095 mm
Mesh Quality Plot	High
Mesh information - Details	
Total Nodes	991395
Total Elements	708002
Maximum Aspect Ratio	8.3368
% of elements with Aspect Ratio < 3	99.8
% of elements with Aspect Ratio > 10	0
% of distorted elements(Jacobian)	0
Time to complete mesh(hh:mm:ss):	00:00:34

To ensure the results acquired from each test were comparable it was important they came from the exact same coordinates. The point of interest here was at the top centre of the piston pin near the connecting rod, this point was located and saved as a sensor for all further studies. Any variations to the mesh would not effect the location the stress data was taken from. A total of three sensors were placed for the study:

Sensor 1, The top of the piston pin.

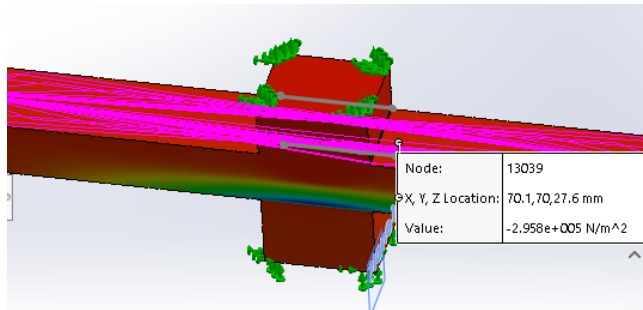


FIGURE 7 SENSOR LOCATION ONE

Sensor 2, The underside of the piston pin.

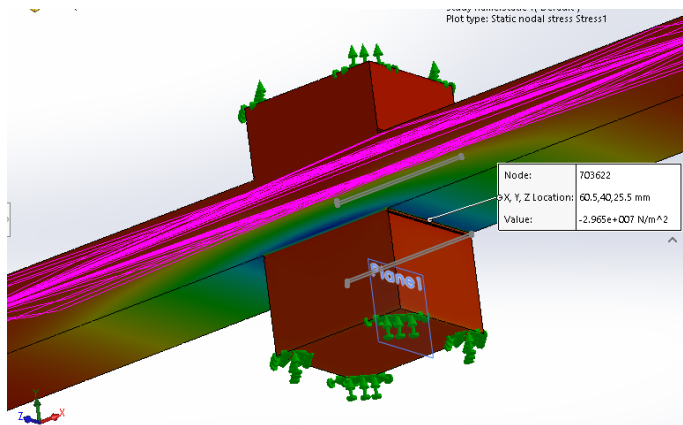


FIGURE 8 SENSOR LOCATION TWO

Sensor 3, The contact edge between the connecting rod and pin.

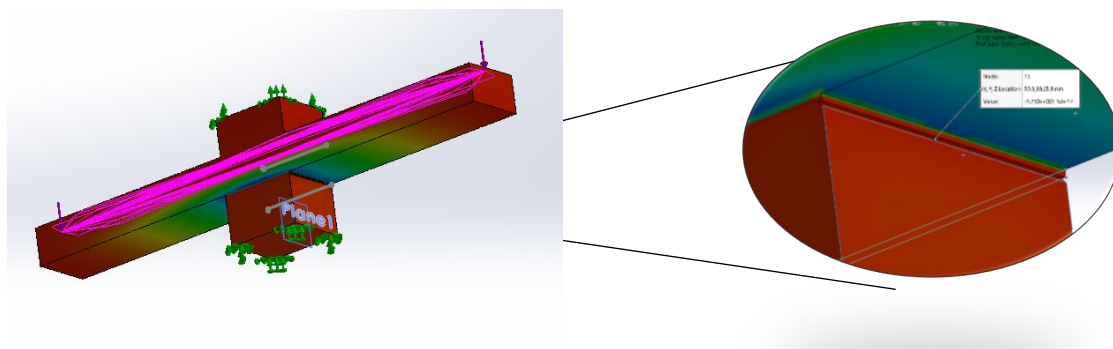


FIGURE 9 SENSOR LOCATION THREE

LOCATION 3

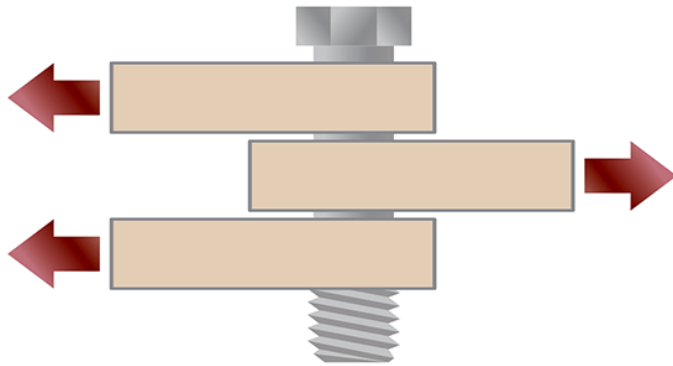


FIGURE 10 BETWEEN, D. (2016).

Figure (10) shows a bolt under double shear conditions. This is what happens to a piston pin during a combustion cycle. The directions of the force switching every 180 degrees. This causes bearing stress, to calculate bearing stress bearing force and bearing area are needed. The area is that which is perpendicular to the force, in the case of the piston pin the area would be a rectangle. The force acting on each bearing would be half the total force, with a reaction force acting in the opposite direction. For steel typical values of shear modulus of elasticity are 75 Gpa. (Gere, 2002)

12.3 Boundary Conditions of the study?

The top and bottom blocks are rigidly fixed on one face limiting the movement while still allowing the deformation of material. The natural rubber is bonded to the top and bottom blocks and the piston pin has a non penetration bond. A non penetration bond means the two surfaces cannot lift away from each other but are free to move in the x and z direction.

12.4 Results

Before the stress values could be taken from any of the sensors a stress convergence test needed to be carried out. The mesh size is gradually reduced until the stress values level out, the convergence was carried out on one of the sensors to eliminate interference of stress singularities. The stress singularities occur at the root of ridged fixtures, sharp re-entrant corners and sharp edges in contact. The stress of a singularity does not converge, as the mesh is refined the stress increases in theory it is infinite.

Stress = force/area, if the area is zero the stress = infinity

St. Venant's principle states that the effect of local disturbances to a uniform stress field remains local. Further away from the disturbance the results will not be perturbed. The stress directly under a concentrated load may be several times higher than the average (Gere, 2002). At a distance from the disturbance equal to the width of the bar the stress becomes uniform. (Gere, 2002)

For this reason the stress at the singularity can be ignored as the point of interest is far enough away that it would have no effect and the values of stress can be taken as correct.

TABLE 5 MESH CONVERGENCE RESULTS

mesh size	Node Number	Stress	Location Coordinates		
			x	y	z
1	9562	5.74E+07	70.12075	70	27.56615
1.5	4415	5.74E+07	70.12075	70	27.56615
1.9	2857	5.74E+07	70.12075	70	27.56615
2	2634	5.74E+07	70.12075	70	27.56615
4	791	5.73E+07	70.12075	70	27.56615
7.3	253	5.69E+07	70.12075	70	27.56615
10	180	5.64E+07	70.12075	70	27.56615
15	101	5.62E+07	70.12075	70	27.56615

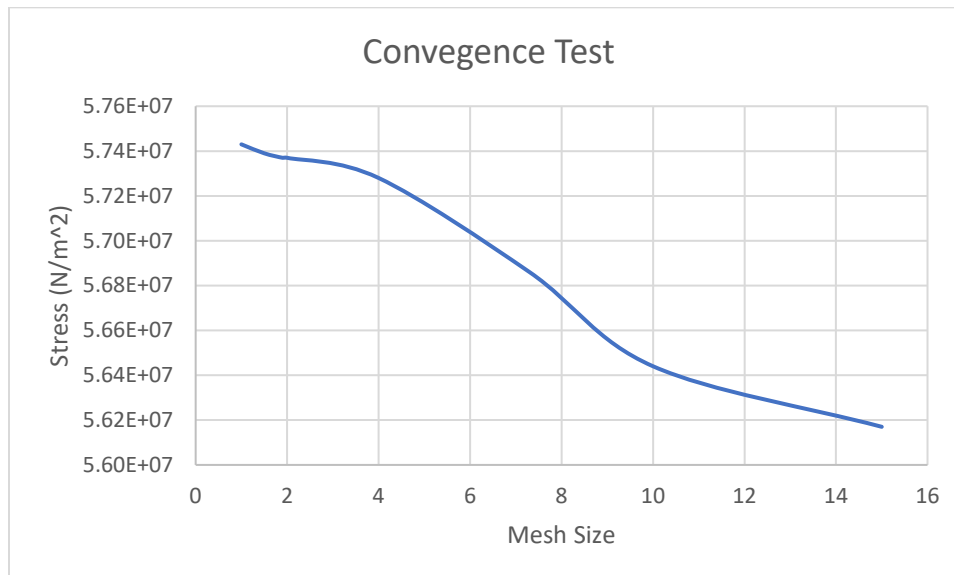


FIGURE 11 MESH CONVERGENCE GRAPH

Figure (11) shows the results of mesh refinement, the stress does not converge as it should but instead keeps increasing, there is a flat spot at 2mm mesh. The mesh used for further studies was 1.9 the point before the stresses start to increase again. This is a local convergence not a global.

TABLE 6 STATIC STUDY RESULTS

Chamfered edge + 1.9 mm mesh		
Chamfer size (mm)	node	Stress (N/m ²)
0	2857	5.74E+07
0.5	1230	5.76E+07
1	2857	5.77E+07
1.5	1183	5.78E+07

The values given in table (5) were taken from a previous study. In this study the piston pin was not free to bend within the small end. As expected by adding a chamfer the distance to the point load was increased, hence the bending moment also increased and the stress thusly. The increase in stress is linear. When bending of the piston pin is allowed within the small end the stress results do not follow the same trend, not when looking at the individual stresses. The normal stress in direction x rise and fall. This could be down to errors in the mesh as the changes in stress are very small.

TABLE 7 STATIC STUDY RESULTS

Chamfer Size (mm)	Normal stress (N/mm ²)		
	x	z	y
0	26.45	0.03636	-0.2491
0.5	26.43719	0.022882	-0.25234
0.7	26.41754	-0.29582	-0.0242
0.8	26.43556	-0.28873	-0.01967
0.9	26.39823	-0.31425	-0.04038
1	26.44228	0.00906	-0.29912
1.5	26.45509	-0.03242	-0.28223

Total Stress =(Normal stress+shear stress+principal stress) by adding the stress it can be seen that the total value of stress for the sensor is virtually unchanged, the chamfer however dose change the way the stress is distributed. The solidworks study has shown that the change in chamfer does little to reduce the stress within the piston pin.

TABLE 8 STATIC STUDY RESULTS PRINCIPAL STRESS

Chamfer Size (mm)	Principal Stress (N/mm ²)			N/mm ²	N/mm ²
	p1	p2	p3	Total	Von Mises
0	26.45	0.03656	-0.2498	52.61845	26.55778
0.5	26.44	0.02242	-0.2523	53.04015	26.55601
0.7	26.42	-0.02452	-0.2958	52.9803	26.5812
0.8	26.44	-0.01983	-0.289	53.16977	26.59544
0.9	26.4	-0.04048	-0.3145	53.06598	26.57855
1	26.44	0.008353	-0.2991	53.44638	26.58671
1.5	26.46	-0.03292	-0.2822	53.90027	26.61844

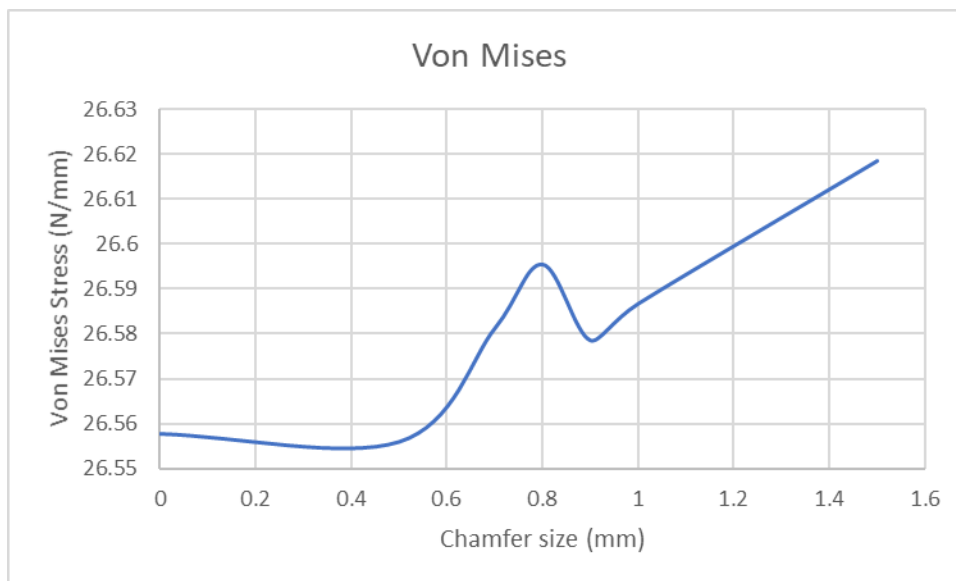


FIGURE 12 VON MISES GRAPH

Figure (12) shows a graph of the Von Mises results for the connecting rod, the stress increase with chamfer size. The stress values drop slightly at 0.9mm chamfer, but continue to rise afterwards. The drop is 0.18 N/mm^2 and could be attributed to an error in the mesh.

12.5 Conclusions

The study shows that adding the chamfer does very little to the induced stress on the piston pin. There were some slight reductions to Von Mises stress at 0.5mm chamfer but only by 0.00177 N/mm^2 as or 1770 N/m^2 . Such a small number that is insignificant when compared to the applied stress. Increasing the applied force would make no difference to that.

12.6 Boundary Conditions

Within an IC engine the connecting rod and piston pin are not fixed rigidly. The piston pin can rotate freely about the x axis, there would also be a limited movement in both the z and y directions. The piston pin is also able to move side to side. These movements are small, but they have drastic effects on stress within the components. If when setting the foundations these movements are not taken in to consideration the results obtained would be wrong. The study in which these components were being assessed was a static. Within a static study any large movements will result in a failed calculation.

12.7 Mesh

At the beginning of this study it was the belief of the author that the greater the density of a mesh, the better the results would be. This is not the case. The results from the first set of studies were wrong due to high mesh density creating, exaggerated stress concentrations. However due to the location of interest, getting correct results would prove difficult as the little end of a connecting rod has geometry that tends to produce a singularity, or a stress concentration. Careful mesh control was used to reduce the impact of stress concentrations on the overall result.

Hand Calculations

The author used stress equations to calculate the effect that adding a chamfer might have, the calculations produced the results in table (8). The results in table (8) show that by increasing the size of the chamfer, the bending stress is increased. As the chamfer increased the surface area in contact with the piston pin was reduced and therefore so was the cross-sectional area. A smaller surface area caused an increase in stress; and a reduction in the cross-sectional area caused a decrease in the stiffness. There was no point in the study where the reduction in stiffness was enough to counteract any rise in stress caused by a reduction of the area. As show in equation (34), stiffness does affect stress. Any increase in a chamfer size also increased the bending moment, as the geometry of the piston pin never changed y and I in the equation (36) remain the same, therefore as bending moment increases so does bending stress.

$$\text{Bending stress} = \frac{M \times y}{I} \quad (31)$$

TABLE 9 HAND CALCULATION RESULTS

Using Area			
chamfer (mm)	bending moment (N-mm)	stress (N/mm ²)	stiffness N/mm
0	360.00	394.74	1900.00
0.1	362.12	396.21	1892.93
0.2	364.24	397.70	1885.86
0.3	366.36	399.19	1878.79
0.4	368.49	400.70	1871.72
0.5	370.61	402.22	1864.64
0.6	372.73	403.75	1857.57
0.7	374.85	405.30	1850.50
0.8	376.97	406.85	1843.43
0.9	379.09	408.42	1836.36
1	381.21	410.00	1829.29
1.1	383.33	411.59	1822.22

1.2	385.46	413.19	1815.15
1.3	387.58	414.81	1808.08
1.4	389.70	416.43	1801.01
1.5	391.82	418.08	1793.93
1.6	393.94	419.73	1786.86
1.7	396.06	421.40	1779.79
1.8	398.18	423.08	1772.72
1.9	400.31	424.77	1765.65
2	402.43	426.48	1758.58

Figure (13) shows a linear increase in stress as the area is reduced.

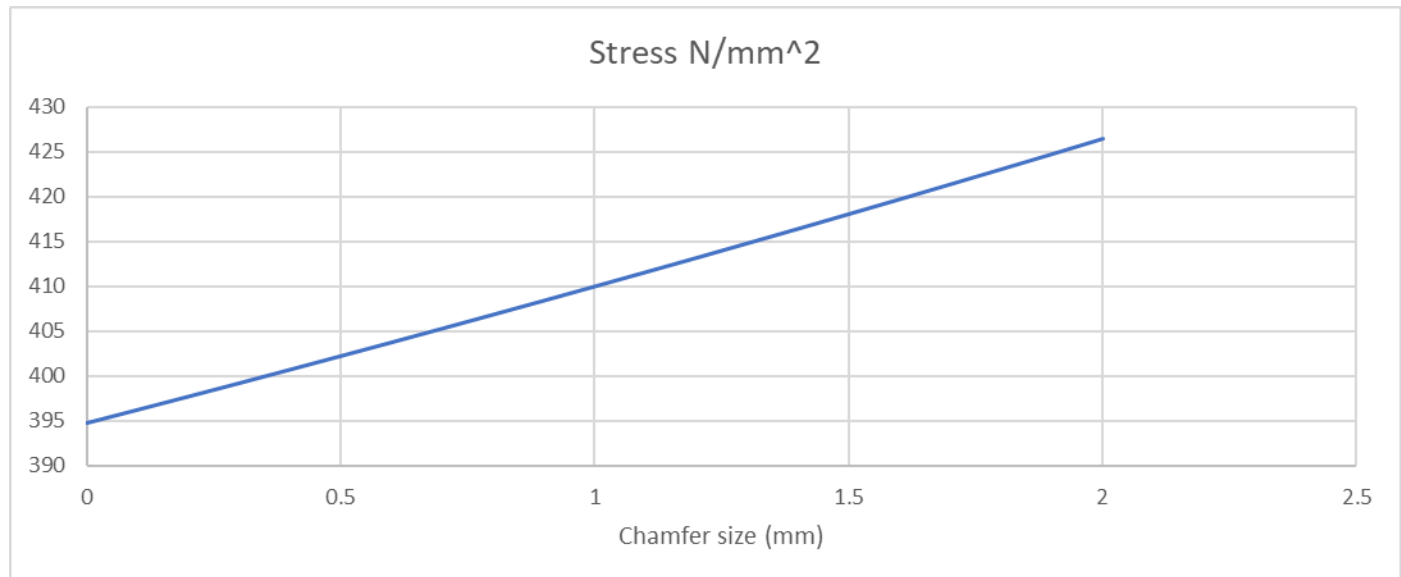


FIGURE 13 HAND CALCULATIONS STRESS GRAPH

Figure (14) shows the linear increase in bending moment as the chamfer increased and therefore the distance between the connecting rod and centre of force increased.

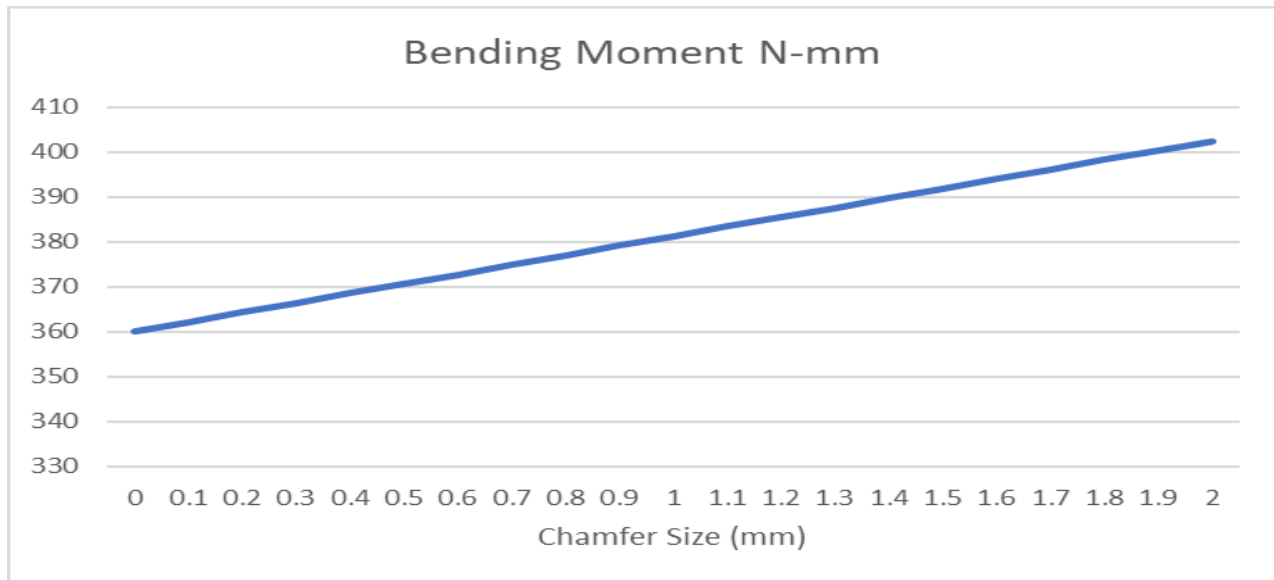


FIGURE 14 HAND CALCULATIONS BENDING MOMENT GRAPH

Figure (15) shows the linear decrease in stiffness as the cross-sectional area was reduced.

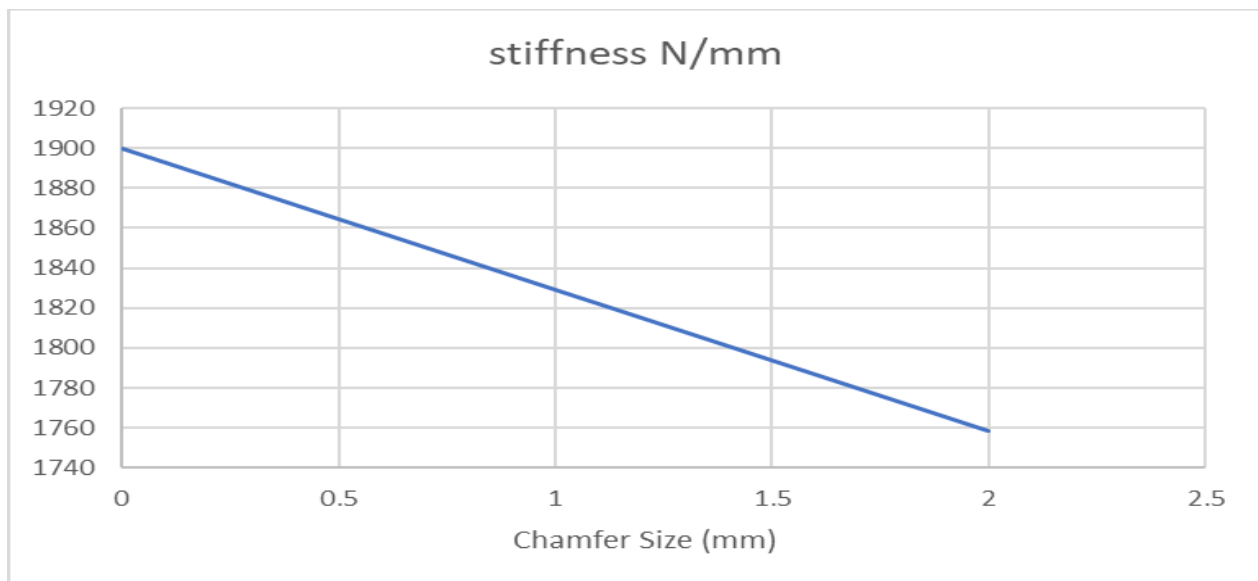


FIGURE 15 HAND CALCULATIONS STIFFNESS GRAPH

Both the FEA results and the hand calculations show that by adding or increasing the size of a chamfer by any degree the stress is increased accordingly. The results show that a chamfered edge does not help with stress.

13 Crankshaft

13.1 CAD

The cad drawings for the full crankshaft are shown in Figure 16). The model was measured as accurately as possible by the author given the method used. Ideally a detailed scan of the crankshaft would have been used as a base to apply the solid 3D geometry. The natural geometry of the spans made them very difficult to measure and was beyond the skill of the author to draw within solid works. On the Ford crankshaft the face of each span curves about the x axis and the z axis, like the surface of a computer mouse. The freeform option within solid works creates these natural geometries.

There are further differences between the cad drawing in figure (16) and the actual crankshaft. The oil wells have been neglected within the entirety of the model and the holes drilled within each crank web used to balance the crankshaft have also been left out. The holes within the crank span should not have any effect on the areas of interest which are the crank journals. The author is aware that the oil ways would affect the crank journals

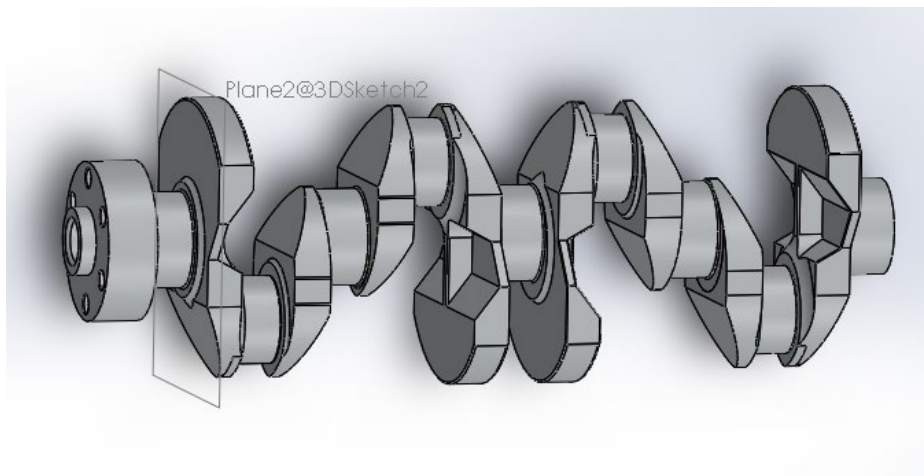


FIGURE 16 FULL CRANKSHAFT CAD

13.2 FEA

13.2.1 One Span

There was an assumption on the author's part that the crankshaft was made of a perfect homogeneous material. It would therefore be perfectly balanced, and the crank spans would be symmetrical. One crank span could be studied instead of the whole crankshaft. The results from a single span would be transferrable. Due to manufacturing margins and the need for local and global balancing there are differences from one span to the next.

The creation of the crankshaft within solid works was done utilizing symmetry to make the task easier. Certain parts of the crankshaft were altered to limit stress concentrations. Sharp edges were rounded, and the size of certain fillets increased. A designer should make a point to fillet to some degree all re-entrant corners and edges. If a corner is left without a fillet then it has; within solid works a perfect 90 degree corner to infinity. In reality there is generally a rounding of some degree to edges and re-entrant corners. Adding a small fillet of 0.01 mm would reduce stress singularities.

13.2.2 Boundary conditions

The crankshaft of the Ford eco was measured by hand using a Vernier calliper. The basic shapes that make up the geometry of the crankshaft were easy to take measurements from. For the more natural geometry like curved faces and edges a grid system was used.

The grid system of measuring involved marking points at equal spaces on four sides, the distances between two sides were measured at each point. These distances were put in solid works as horizontal and vertical lines, the ends could then be joined by a curved line creating a fairly accurate circumference of the section measured.

Setting the boundary conditions to get accurate results that best reflected real life proved to be fairly complicated. The method used in numerous papers is to fix the crankshaft at either end. If the point of interest were in the centre of a full crankshaft, away from singularities caused by the ridged

fixtures this would be acceptable. This method of fixing the crankshaft was used in (Apte, 2014). This study however focuses on a single span, using this same boundary condition lead to inaccurate stress values that far exceeded yield.

13.2.3 Rigid

In this study either end of the crankshaft was fixed rigid in all directions. This added a lot of stiffness to the crank span which would not normally be there, giving extremely high stress concentrations.

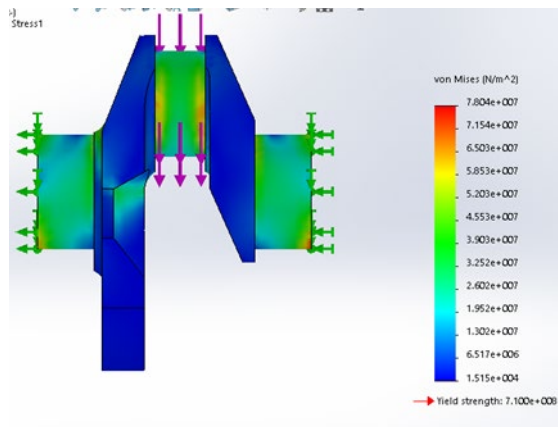


FIGURE 17 RIDGED FIXTURE RESULTS

13.2.3.1 Results

The result of this study was deemed to be incorrect, this is because of the unrealistic fixture, and the span is not allowed to move. However, ignoring the study conditions, the stress values are reasonable. The peak value of 78.04 N/mm² is not far off from calculated values shown in table (9). The stress value does not pass yield or tensile stress. The material used in this study was 4340 annealed steel, it had a tensile strength of 750 N/mm meaning there is a safety factor of 9.1.

13.2.4 With end blocks

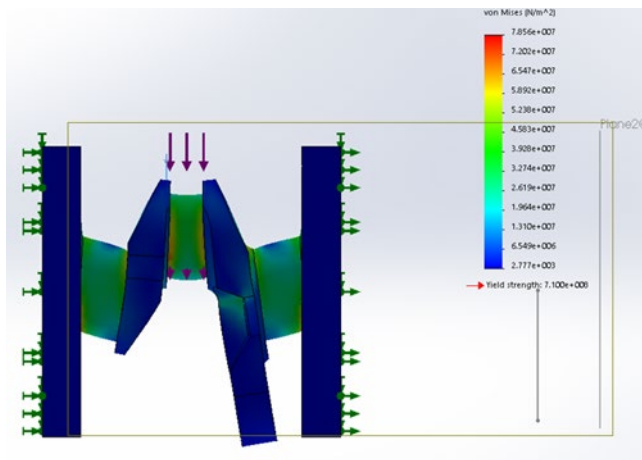


FIGURE 18 END BLOCKS

13.2.4.1 Results

Adding a plate on either end of the crankshaft served two purposes, it added distance between the singularity caused by the ridged fixture and the point of interest, the crank journal. It also meant no point of the crank span was fixed, every part that was of interest was able to move in a natural way. The plates added to the end of the span were of no interest and served only as a fixture, they could be left out of any stress plots. The manner in which the crank span was fixed in this study slightly reduced the stiffness but made a marginal difference to the stress values. The highest stress value being 78.5 N/mm, 0.1 N/mm higher than the ridged study.

13.2.5 Unrestricted Movement

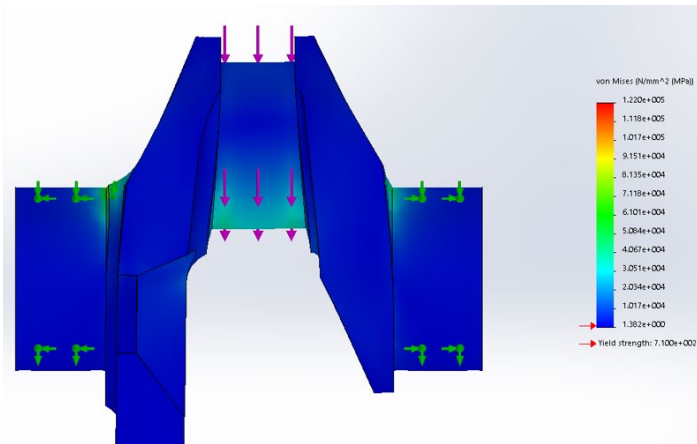


FIGURE 19 FREE MOVEMENT

13.2.5.1 Results

When the movement of the crank span was unrestricted along the x and z axis the stress values were extremely high. The peak value for stress in this study was, 122000 N/mm. This rise in stress can be explained by rearranging the formula for stiffness so that stress is the subject. Equation (36) is the formula for stiffness, substituting E for its formula and the same for strain gives equation (37), and this can be rearranged for stress as shown in equation (35). Equation (35) shows that if the displacement, X is increased the stress also increases.

$$K = \frac{EA}{L} \quad (32)$$

$$K = \frac{\left(\frac{\sigma}{\epsilon}\right) \times A}{l} \quad (33)$$

$$\sigma = \frac{k \times l \times \left(\frac{x}{l}\right)}{A} \quad (34)$$

- Crankshaft mounted on rubber bearings

Mounting the crankshaft on to rubber bearings in the same manner as the connecting rod created the best results. The stress values produced were similar to values calculated using basic stress equations. The rubber bearing allowed for movement that would occur naturally within an engine when the crank span is subjected to load. The crankshaft is not fixed rigidly there is some movement at bearings journals. Effectively the crankshaft has three degrees of movement, although the arc at the y and z axis is very small. The thickness of the rubber used wasn't a calculated value, it was an assumption. Calculating the thickness of rubber needed to realistically simulate an oil film at a specific velocity would require a deeper understanding of the Reynolds equation, something the author could investigate in further work.

13.3 Hand Calculations

The table (9) shows the stress values calculated for the crankshaft. The second moment of area was taken from the cad drawings in solid works. Solid works can be used to obtain the material properties or structures properties of the entire body or on individual faces. The second moment of area was taken from the span at each side of the crank journal, as shown in figure (20). The Equations used for calculating the stress in the crank journal were:

$$I = \frac{Bh^3}{12} = mm^4 \quad (35)$$

$$Bending = F \times D = N - MM \quad (36)$$

$$\text{Stress} = \frac{M \times Y}{I} = \frac{N}{\text{mm}^2} \quad (37)$$

The distance in the table refers to the length in mm from the point of force to the centre of the crank journal. The top set of figures with in table (9) are for the right side of the crank span and the bottom set are for the right.

TABLE 10 HAND CALCULATIONS FOR CRANKSHAFT

Second moment of area (mm ⁴) (x axis)	Compressive load (N)	Net force (N)	Force on span (N)
103598.2	48113.81178	36010.14115	18005.07058
101316.32	48113.81178	36010.14115	18005.07058
Distance	y (mm)	Bending @ A (N/mm)	Stress @ A (x axis) (N/mm ²)
25	13.51	450126.7644	58.69998308
32.44	11.375	584084.4895	65.57641521

The stress values calculated by hand have a safety factor of 12.78 and 11.44. The stress values calculated are less than what solid works predicted in all of the studies. The Maths used to predict stress values will give a reasonably accurate figure. There should not be any great disparity between hand calculations and solid works stress values.

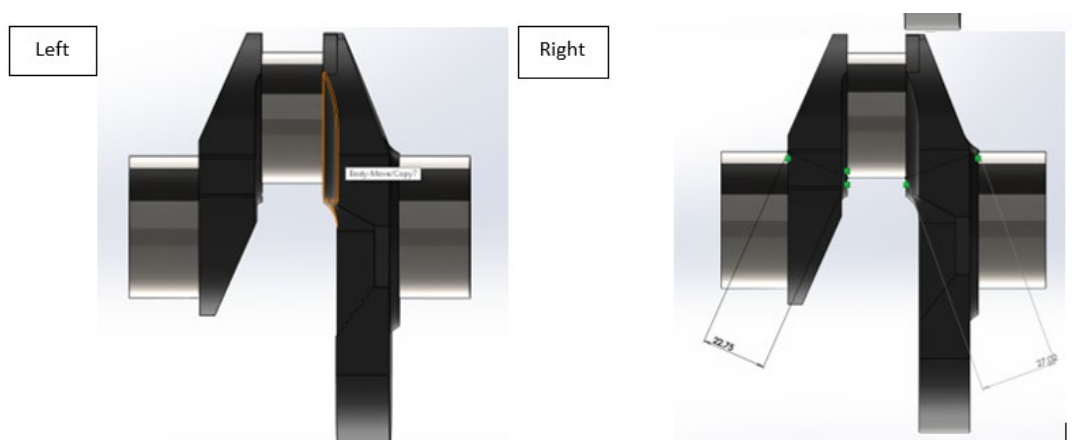


FIGURE 20 INERTIA DIAGRAM

14 FEA Software

14.1 Solid works

The purpose of finite element analysis is to solve engineering problems numerically, while simulating real life conditions in an advanced computer program. The programs used during this analysis were solid works and Ansys. It is essential that the model's geometry and the model's boundary conditions accurately reflect a real-world situation. The results of a mesh analysis are dependent on the inputs. Bad inputs create bad outputs. The finite element method is an approximation, it finds a solution by dividing the component into small sub sections. The solution for each of these small sections can be calculated a lot easier than that of the whole region. These sub sections are referred to as elements, these are formed through interconnecting points called nodes.

There are three basic types of finite elements, one dimensional, two dimensional and three dimensional. One dimensional elements are line elements, truss or boundary elements. The two dimensional are plane elements, membrane or shell elements and three-dimensional elements are for volume, and are tetrahedral, hexahedral and brick. There are many more elements specific to certain test conditions, solid works has 27 element other software packages have far more.

14.2 Analysis

Before any results could be taken the mesh needed to be checked for quality. There were a number of methods employed during this stage of analysis. The Solid works study tree present during a static test enabled mesh refinement, this included but was not limited to convergence study. Before undertaking a mesh convergence, the author used mesh control on the assembly. The mesh control was focused on areas interest which warranted a higher mesh density or areas of concern, like stress concentrations where a reduced mesh density would be needed.

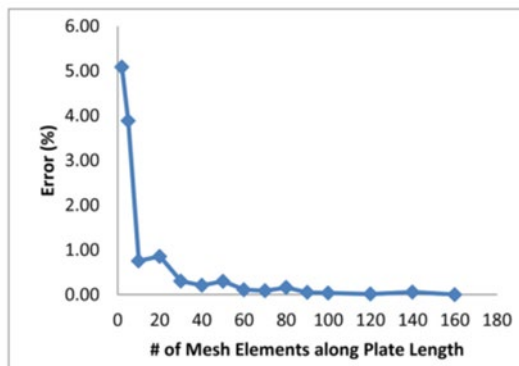


Figure 2. Element size vs accuracy for maximum von Mises stress

FIGURE 21 (Liu and Glass, 2013)

According to tests carried out as (Liu and Glass, 2013), error margin improves with a higher mesh density. This is true for sections of a part without singularities or which aren't too close to ridged fixtures.

Figure (22) shows a static study of the crank span with rubber bearings. The mesh has an element size of 10mm. It shows the completed mesh; from this a visual examination can be made to see if any areas of the mesh are too distorted. The ideal mesh here being an equilateral triangle, distorted elements produce less accurate calculations.

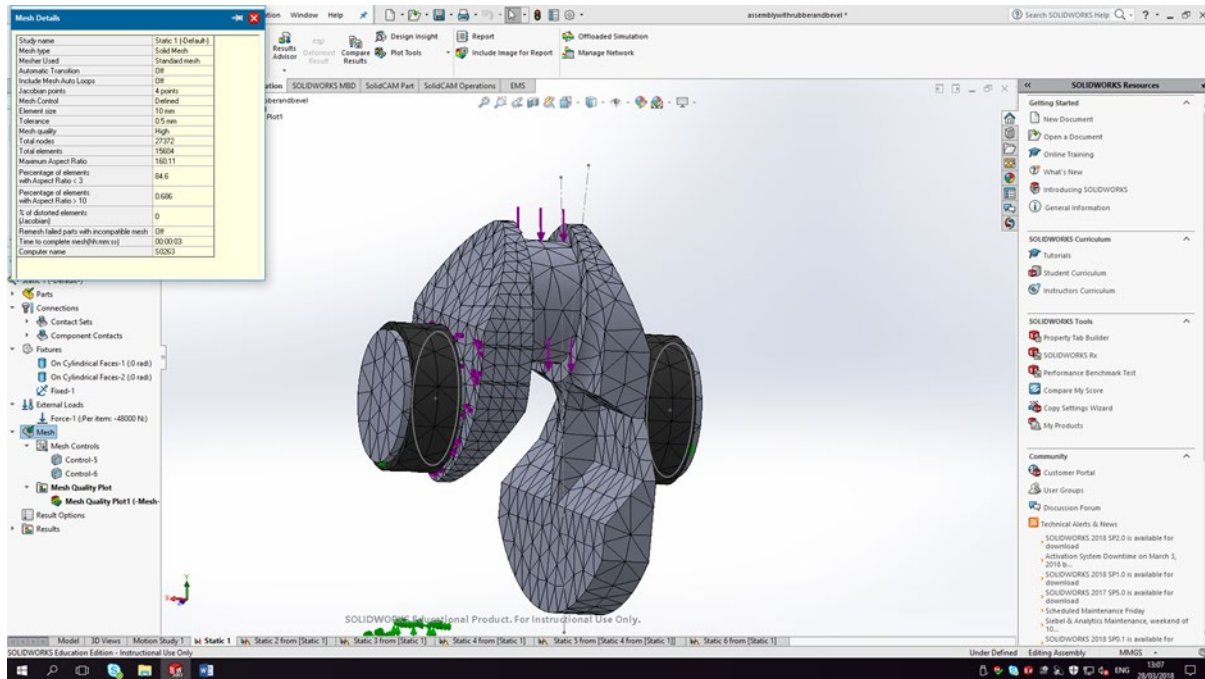


FIGURE 22 MESH

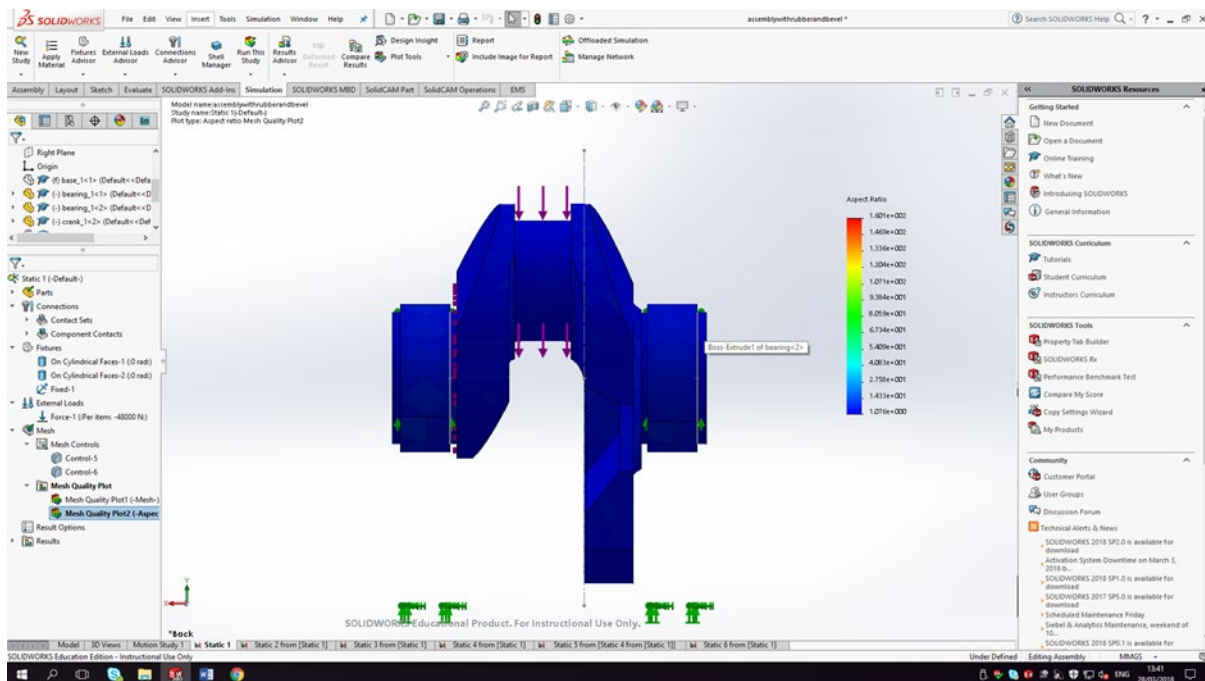


FIGURE 23 ASPECT RATIO

Figure (23) shows the results of a mesh quality plot, this shows the aspect ratio. The ideal aspect ratio for any areas of importance should be below 50 and be no more than 1000 in anywhere else. Aspect ratio assesses distortion of elements, it is a ratio of the element length, element normal length and circumscribed circles, the perfect ratio of a tetrahedral is 1. The percentage of elements with an aspect ratio of less than 3 for this study was 84%, ideally it should be above 90%.

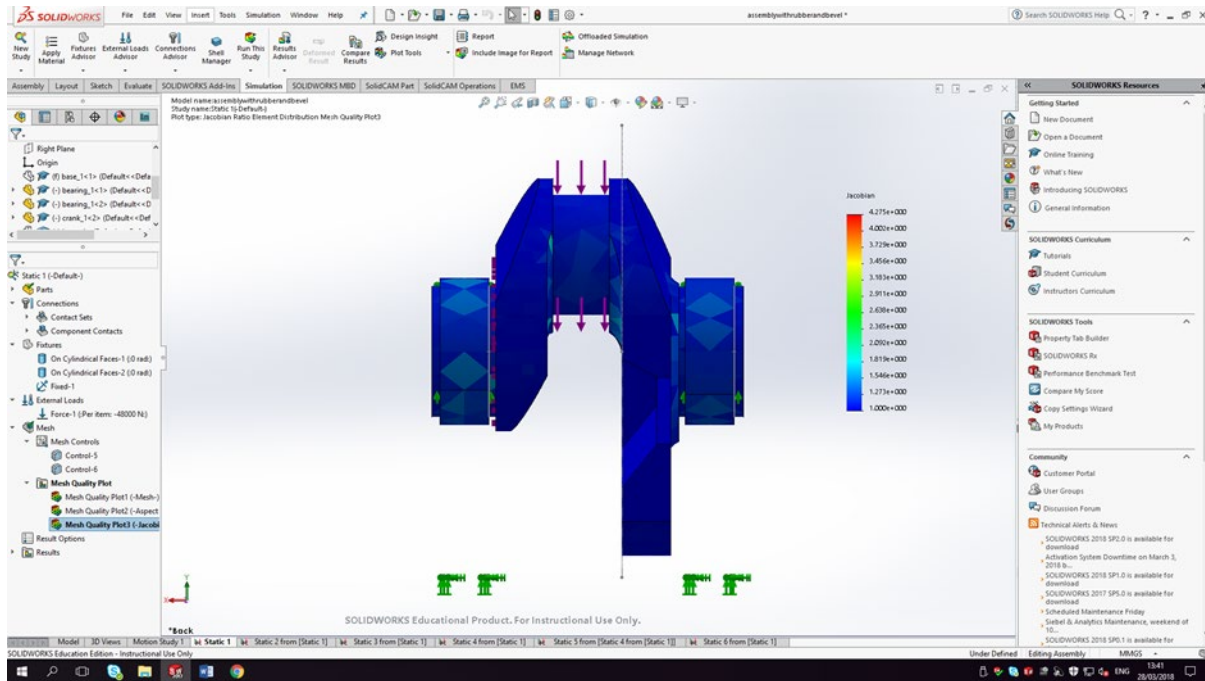


FIGURE 24 JACOBIAN RATIO

Figure (24) shows the Jacobian ratio looks for distorted elements based on the curvature of the element. When the elements of a mesh are placed over highly curved geometry, the midpoint nodes distort the element. As the intensity of the curve increases so does the distortion eventually producing a negative Jacobian value, at this point the study will fail. Figure (24) shows a Jacobian ratio between 4.2 and 1. These values combined with the values for aspect ratio prove that the mesh is of high quality, meaning calculated results will be accurate.

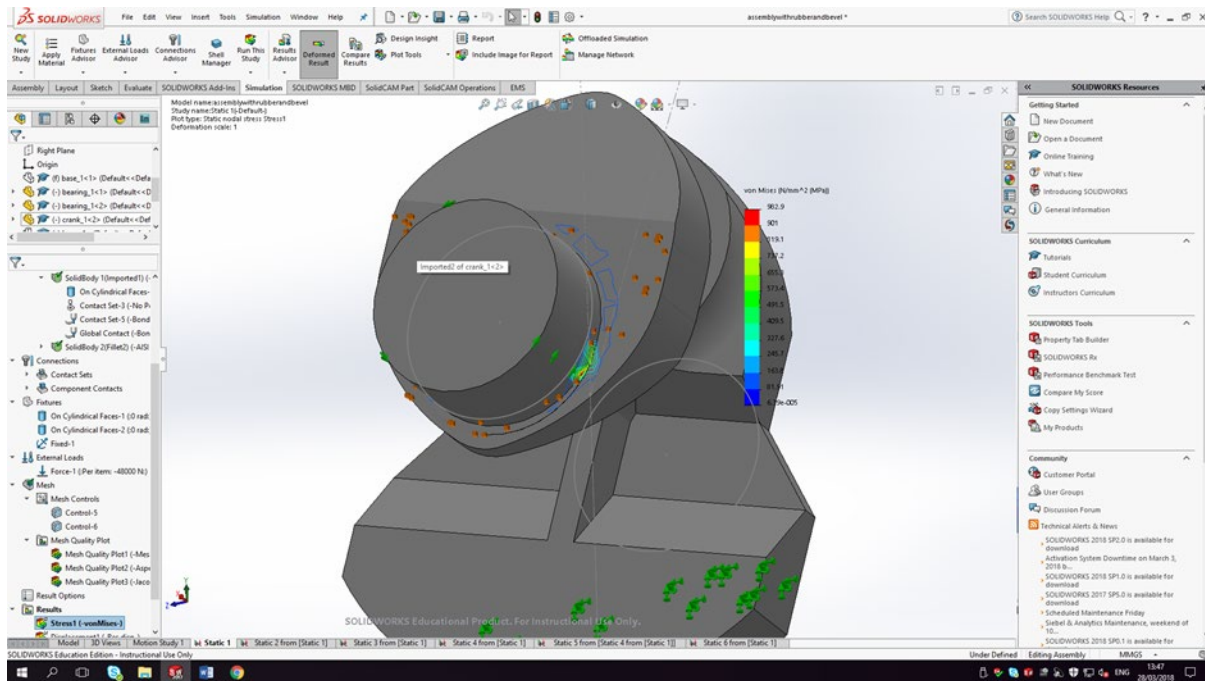


FIGURE 25 VON MISES

Figure (25) shows an alternative method of viewing the Von Mises stress. This method makes the detection of stress concentrations or singularities easier to detect. Generally, these can be removed or minimised by altering the geometry.

15 Conclusions

In this study both the piston pin and crankshaft have been discussed. Firstly, the effects of a chamfered edge on stress in a piston pin was investigated followed by an investigation of FEA boundary conditions and mesh analysis of a crank shaft. The findings of these two investigations have been summarised.

1. The first part of the study has shown that adding a chamfer has less than desirable effect on stress values. The stress and bending moments were increased. The belief that a reduced stiffness could counter any stress increase caused by the change in chamfer was wrong.
2. Although the study of chamfer effect did produce results that disprove the benefit of a chamfered edge, the study is not conclusive, physical tests would need to be carried out to support any findings. There is also the possibility that aspects not considered in this investigation have an effect.
3. The methods used to constrain a part within solid works has a direct impact on the results obtained.
4. The geometry of a part will impact the quality of a mesh and therefore the results
5. Without thoroughly testing and checking the mesh of a study and ensuring a quality mesh is high in all areas of interest the results are meaningless. They cannot be guaranteed to be within a reasonable margin of error.
6. While singularities are not ideal sometimes they are unavoidable, as long as the author is aware of a singularity and its impact it is ok to include them in a study so long as the effects are taken in to consideration.
7. Whenever testing of a part is carried out any points of interest should be a reasonable distance from any fixtures which would negatively impact the results.

16 Further work

The thickness of rubber used could be refined to more accurately represent an oil film between surfaces. The stiffness of oil between the surfaces of the crankshaft and the bearing is dynamic. The oil's stiffness does not remain a constant value so one calculated thickness of rubber would not work in a dynamic situation. The factors affecting the stiffness of an oil film are, pressure, viscosity and density, the rotational velocity of the crank and the film thickness.

Reynolds number is used to calculate the stiffness of an oil film (F.T.Barwell, 1979)

$$\text{Reynolds Equation} = \frac{d}{dx} \left[\frac{h^3}{\eta} \cdot \frac{dp}{dx} \right] + \frac{d}{dz} \left[\frac{h^3}{\eta} \cdot \frac{dp}{dz} \right] = 6 \left[(u_1 + u_2) \cdot \frac{dh}{dx} + 2v \right]$$

In this equation x is measured in the direction of motion and z is measured tangential. The lubrication of a crankshaft bearing is known as Elasto-hydrodynamic-lubrication.

Any inertia effects can be ignored when considering an oil film between two solid surfaces, this is because the oil film is so thin.

A further investigation into the effect of bearing stress and shear stress may produce a more conclusive answer in regards to the chamfers effect on stress.

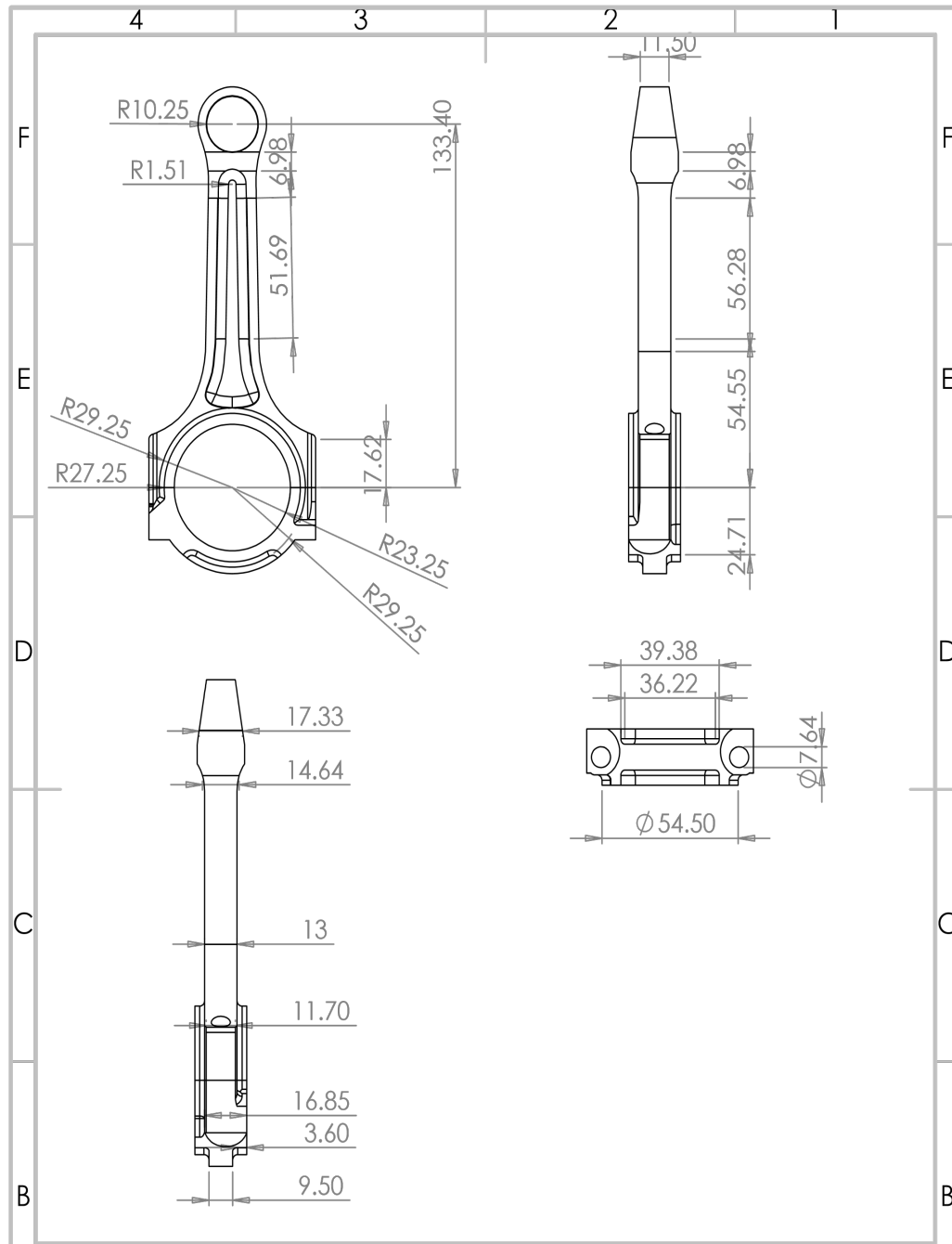
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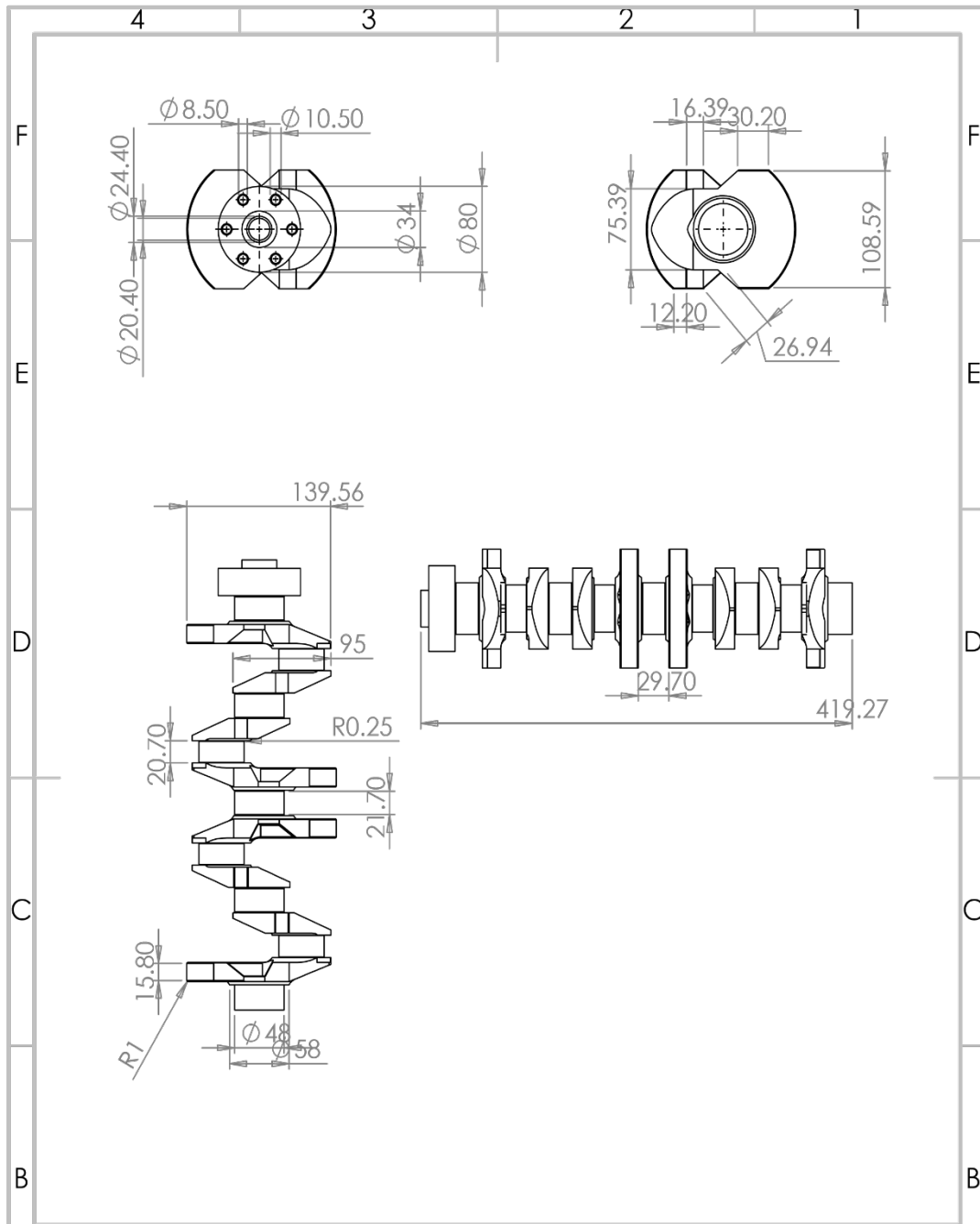
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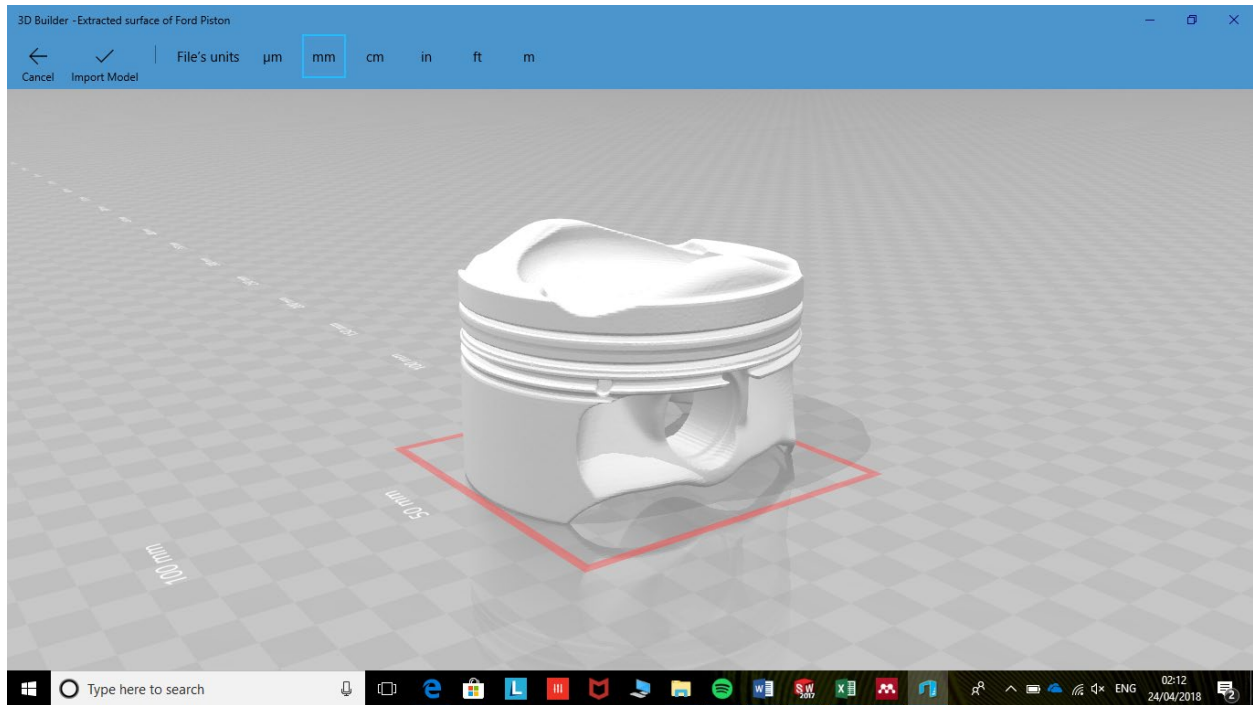
19 Appendix



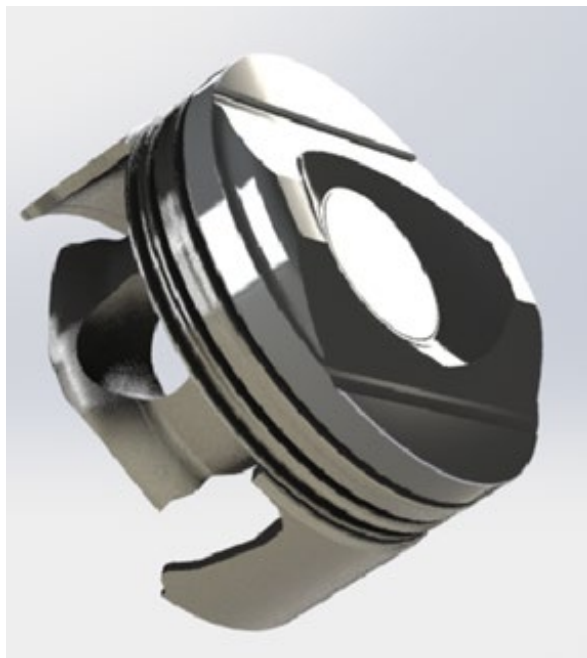
APPENDIX 1 CONNECTING ROD TECHNICAL DRAWING



APPENDIX 2 CRANK SHAFT TECHNICAL DRAWING



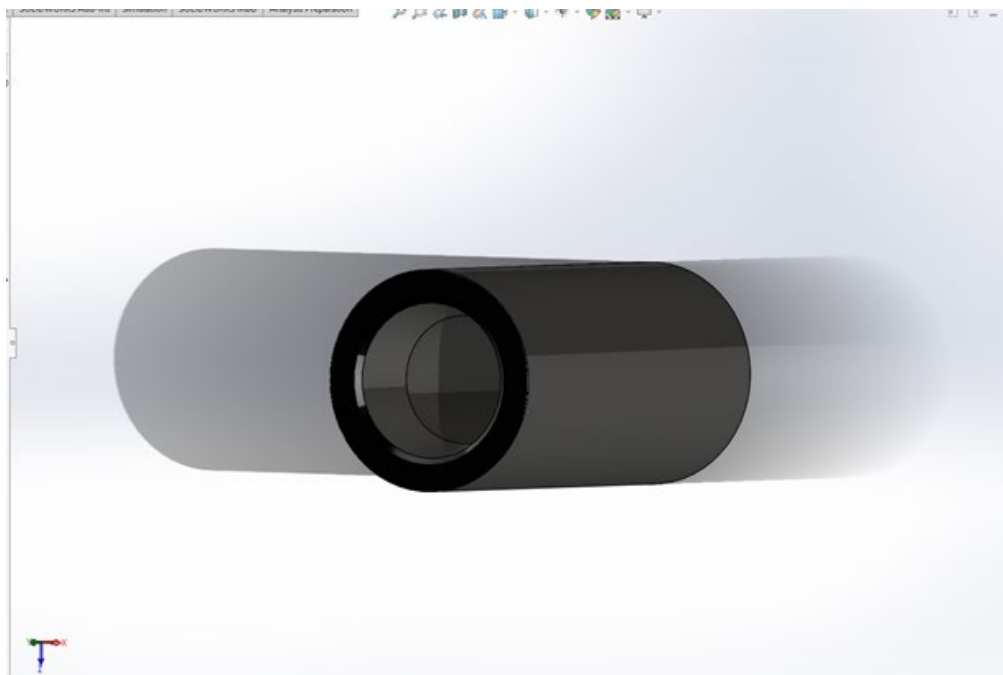
APPENDIX 3 PISTON 3D SCAN



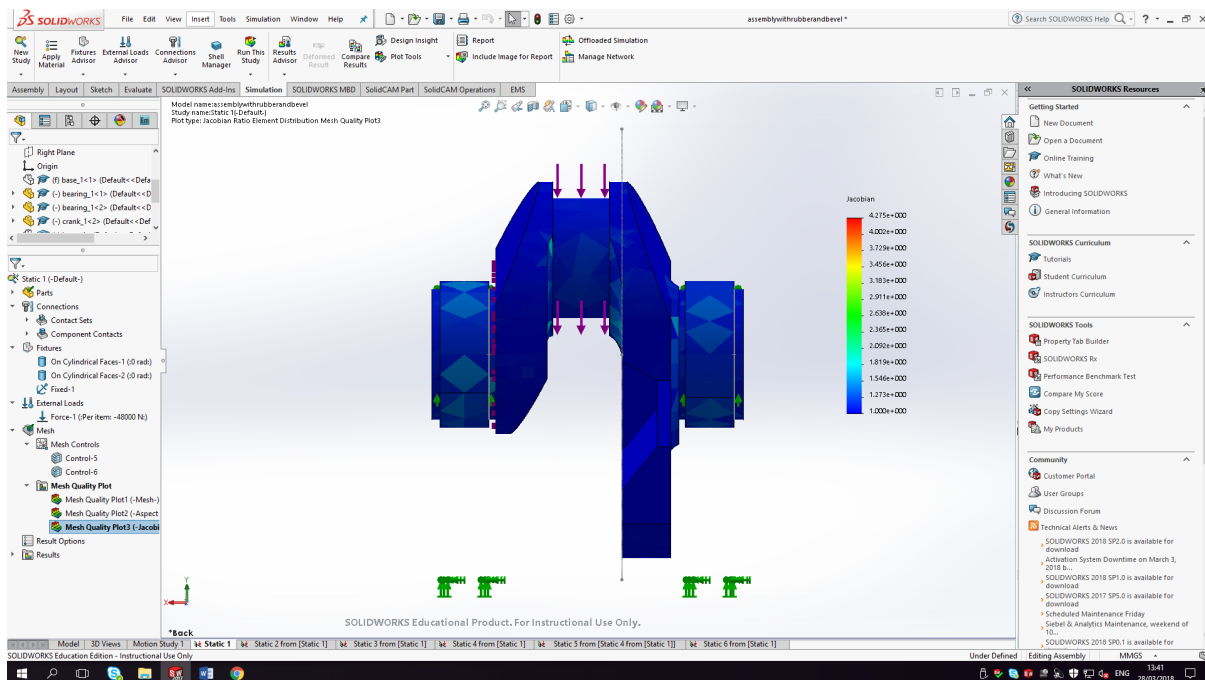
APPENDIX 4 PISTON CAD



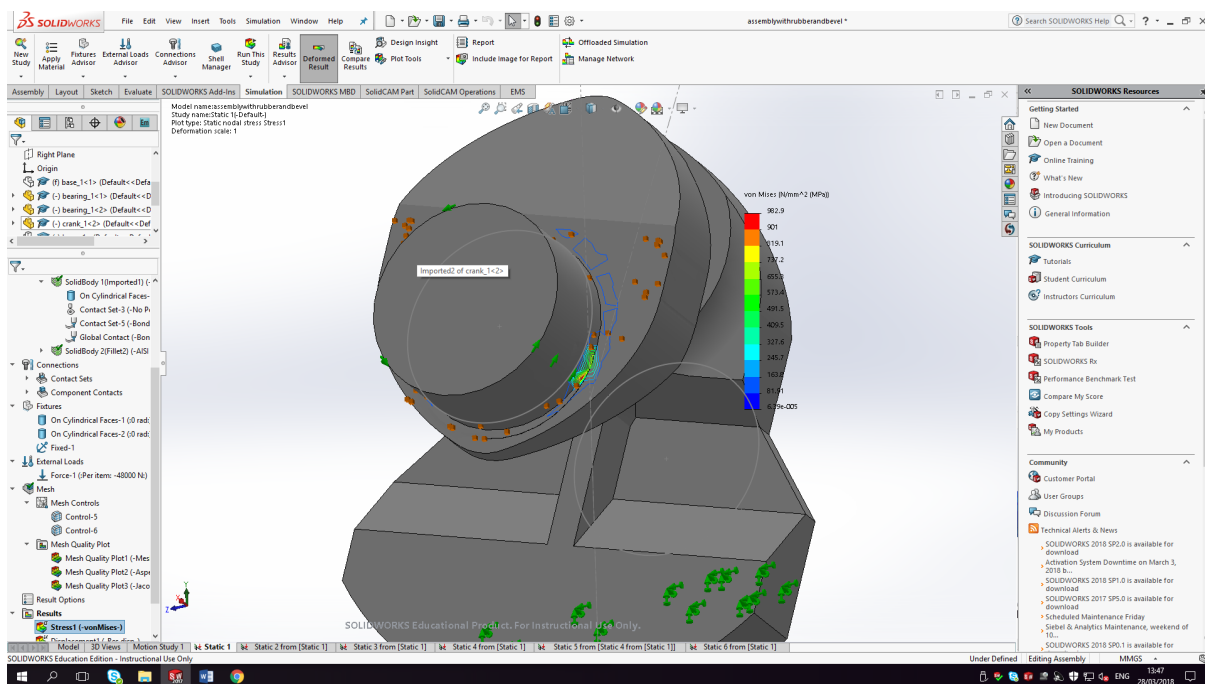
APPENDIX 5 FORD CONNECTING ROD



APPENDIX 6 FORD PISTON PIN

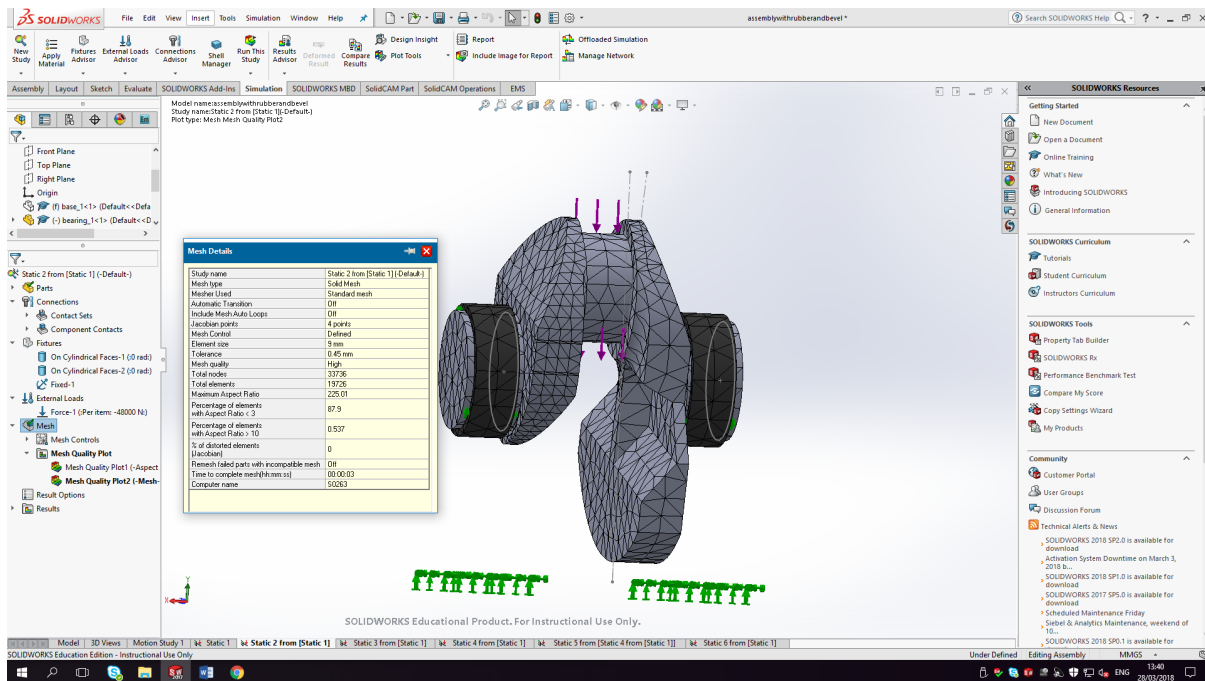


APPENDIX 9 10MM MESH JACOBIAN

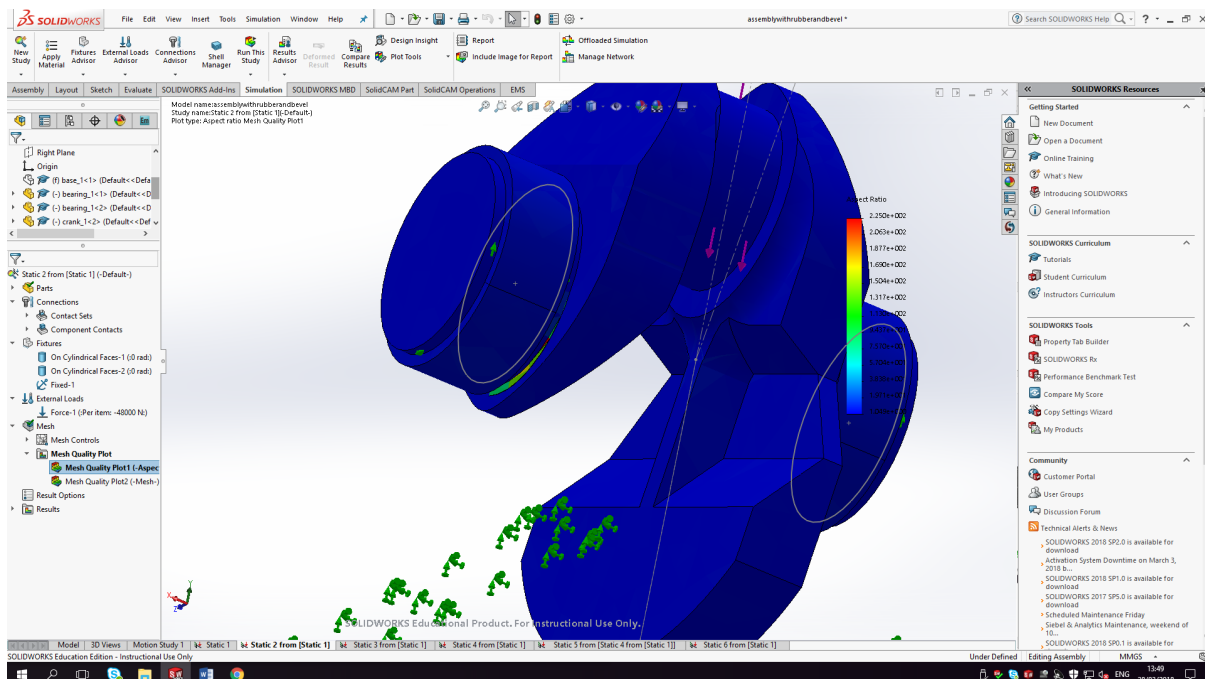


APPENDIX 10 10MM MESH VON MISES

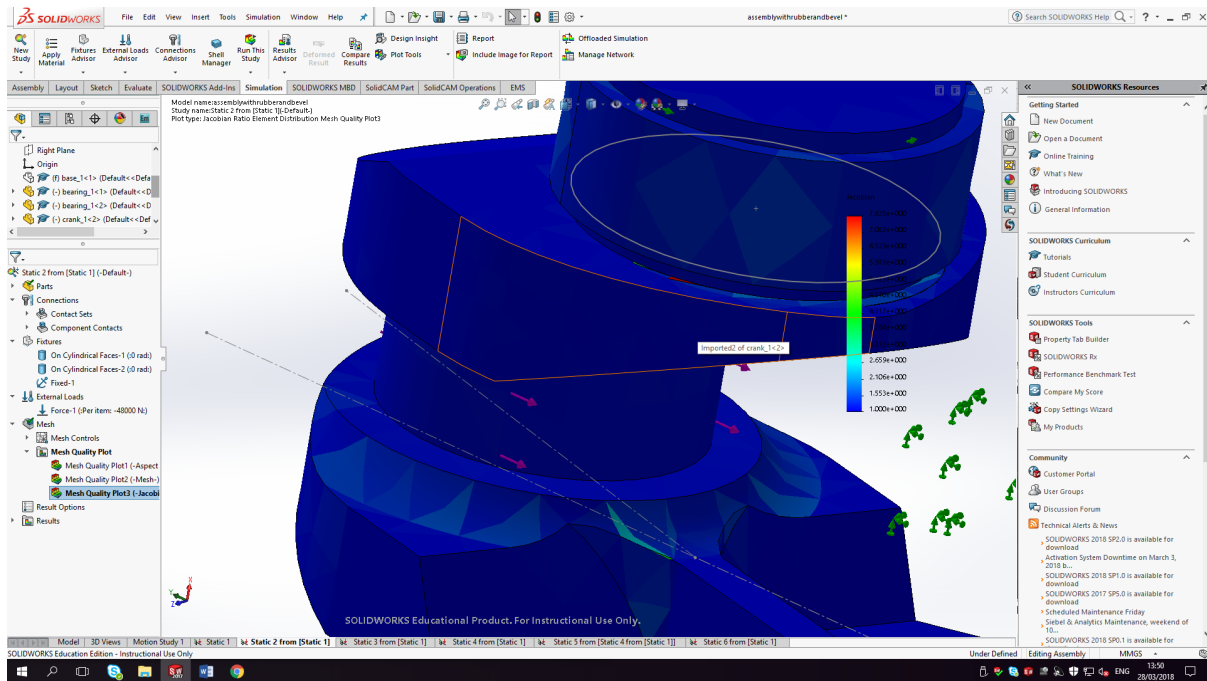
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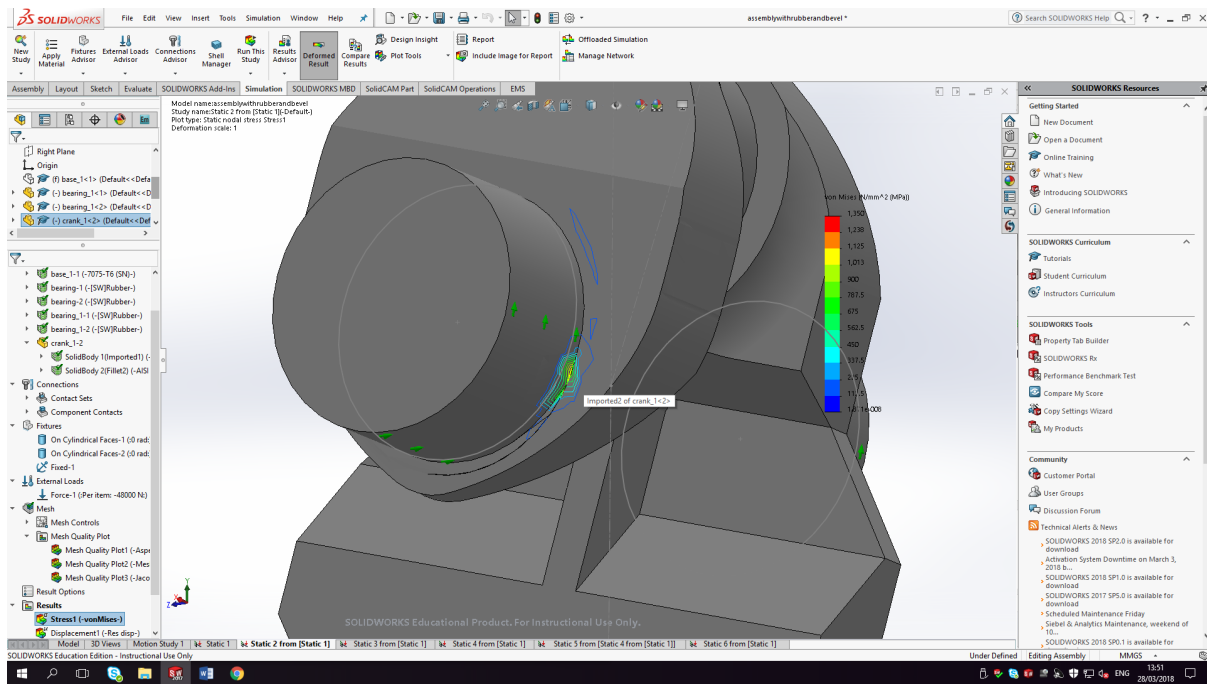
APPENDIX 11 9MM MESH



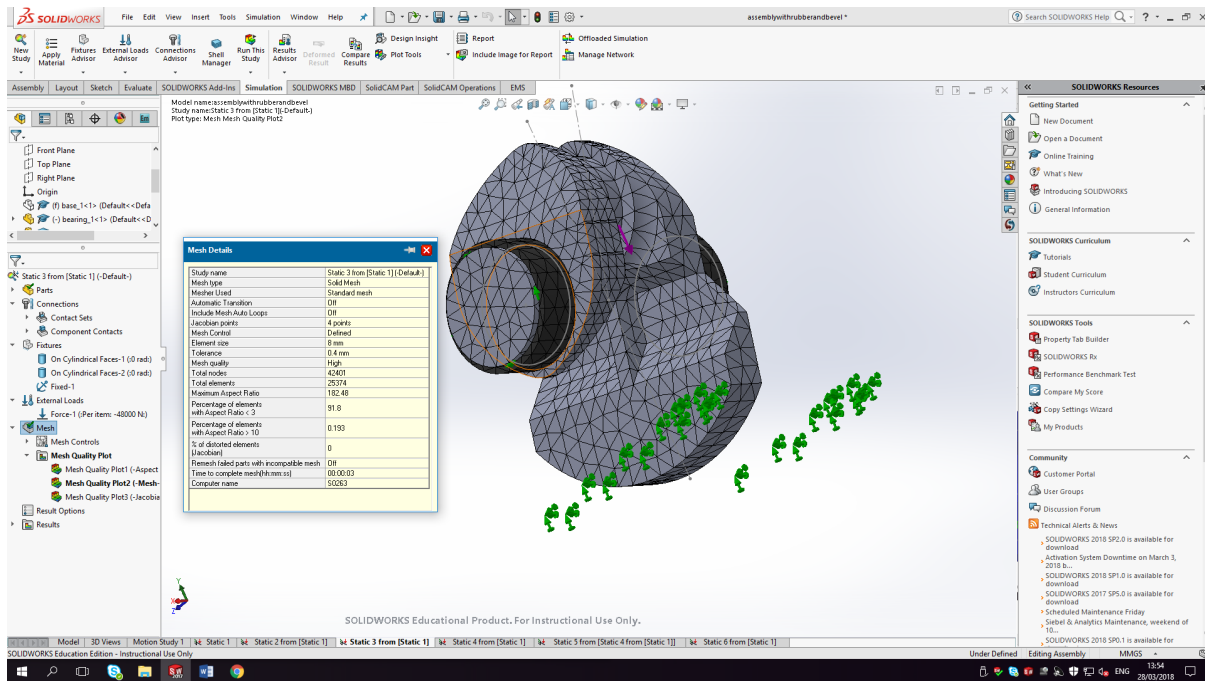
APPENDIX 12 9MM MESH ASPECT RATIO



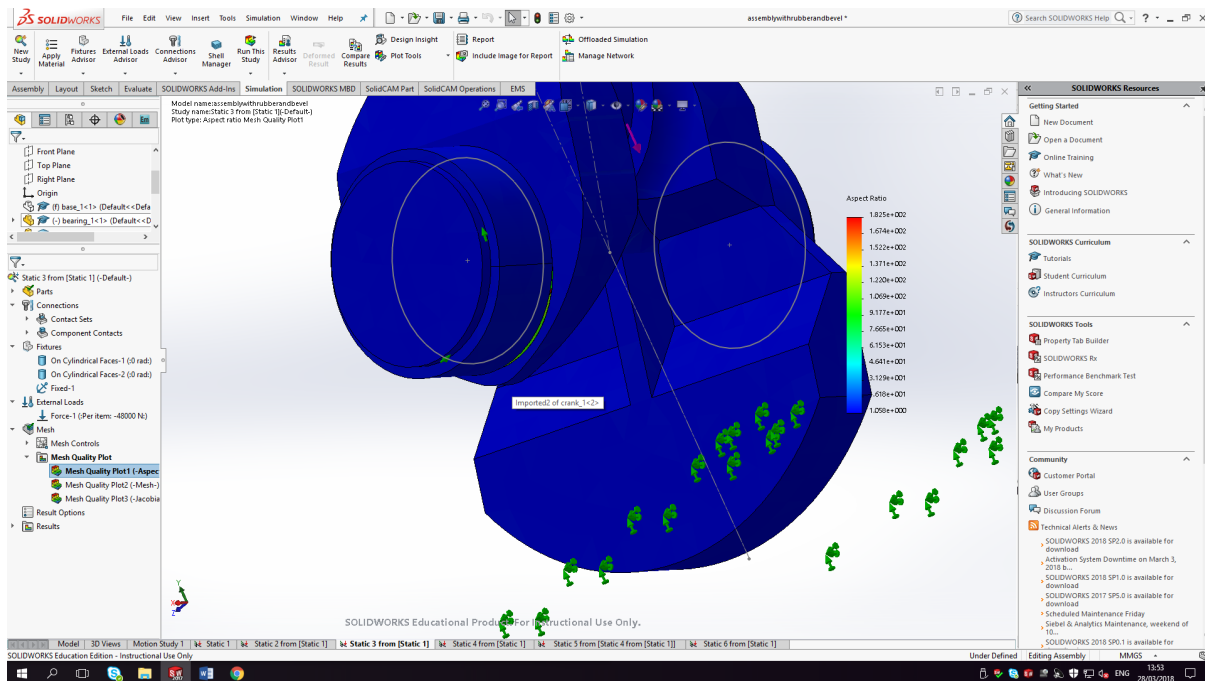
APPENDIX 13 9MM MESH JACOBIAN



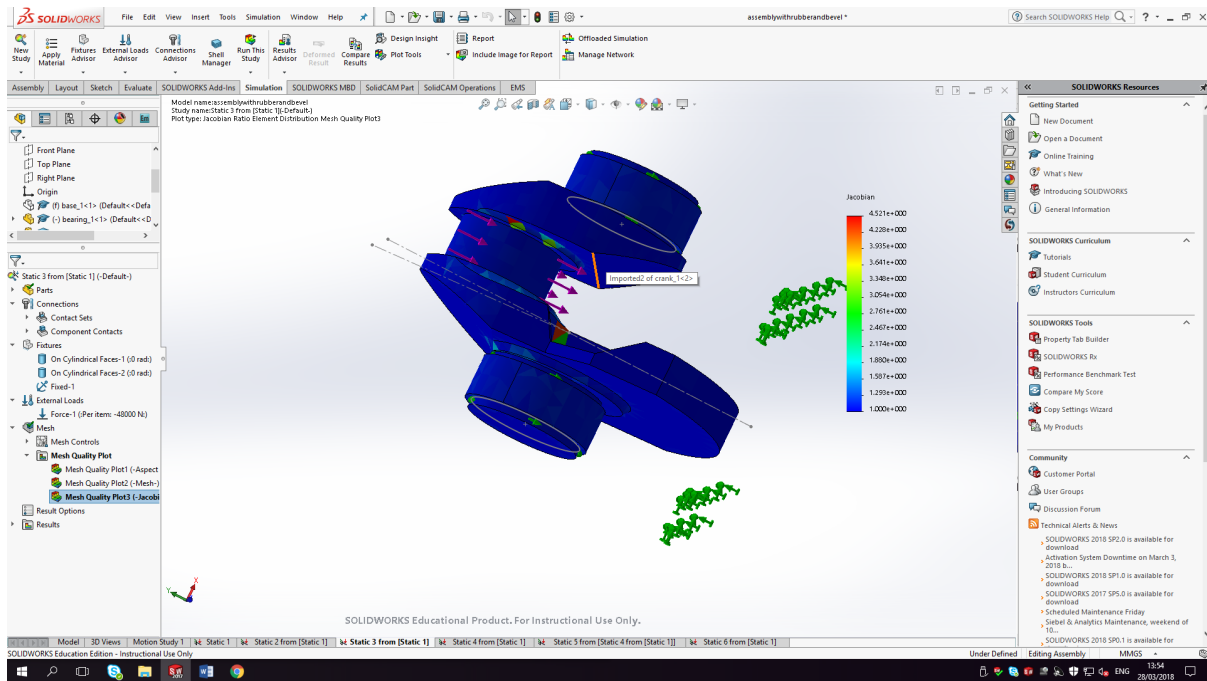
APPENDIX 14 9MM MESH VON MISES



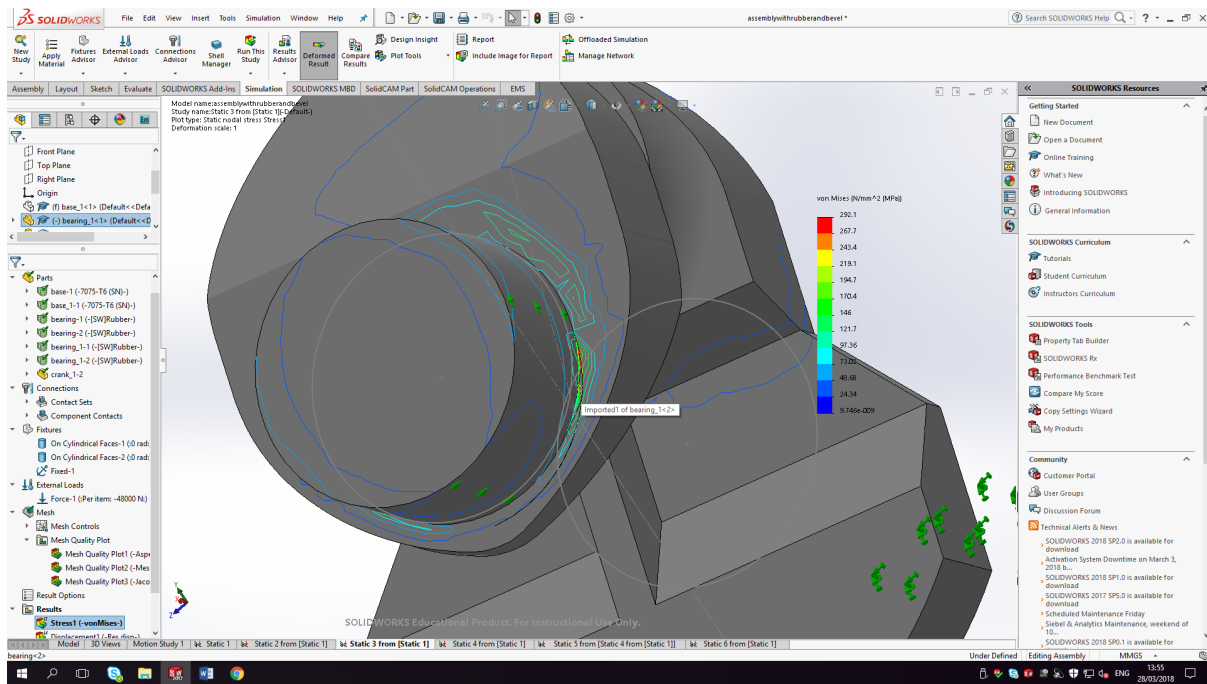
APPENDIX 15 8MM MESH



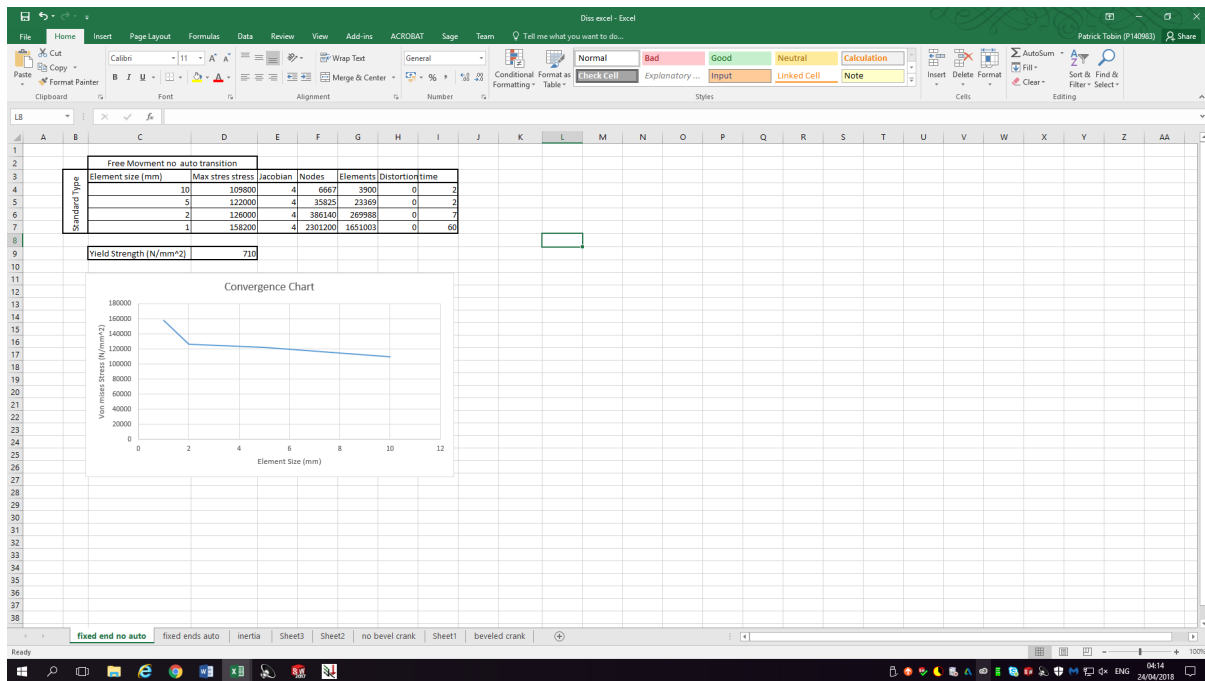
APPENDIX 16 8MM MESH ASPECT RATIO



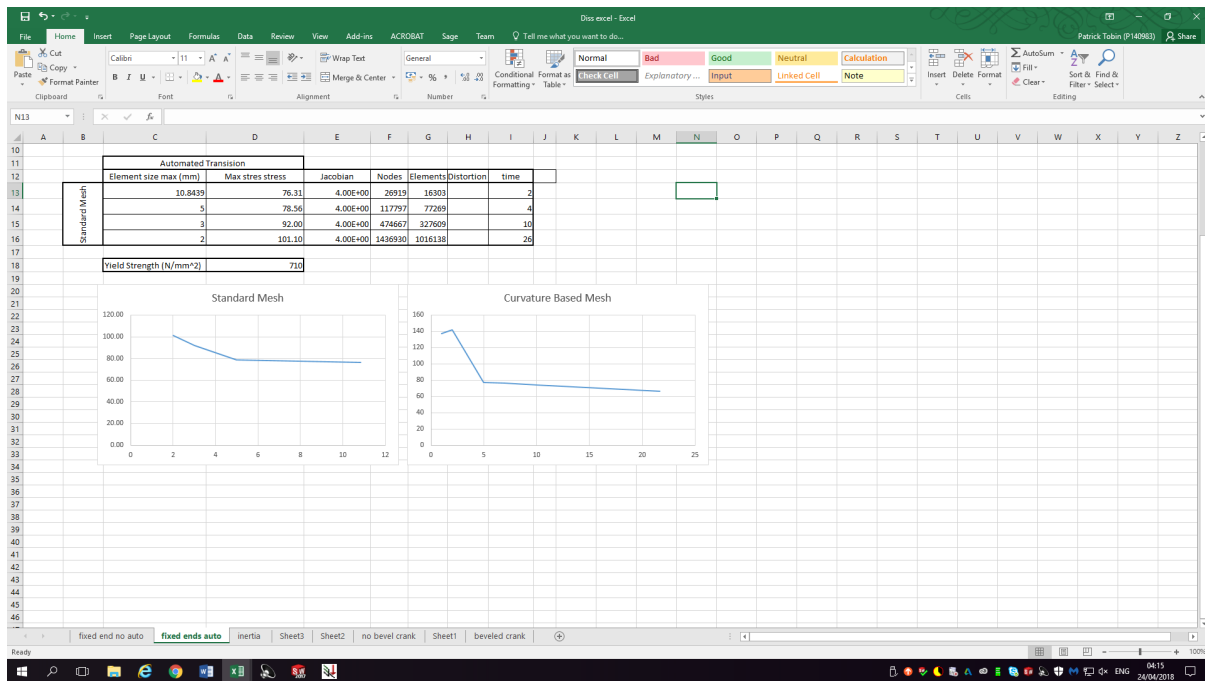
APPENDIX 17 8MM MESH JACOBIAN



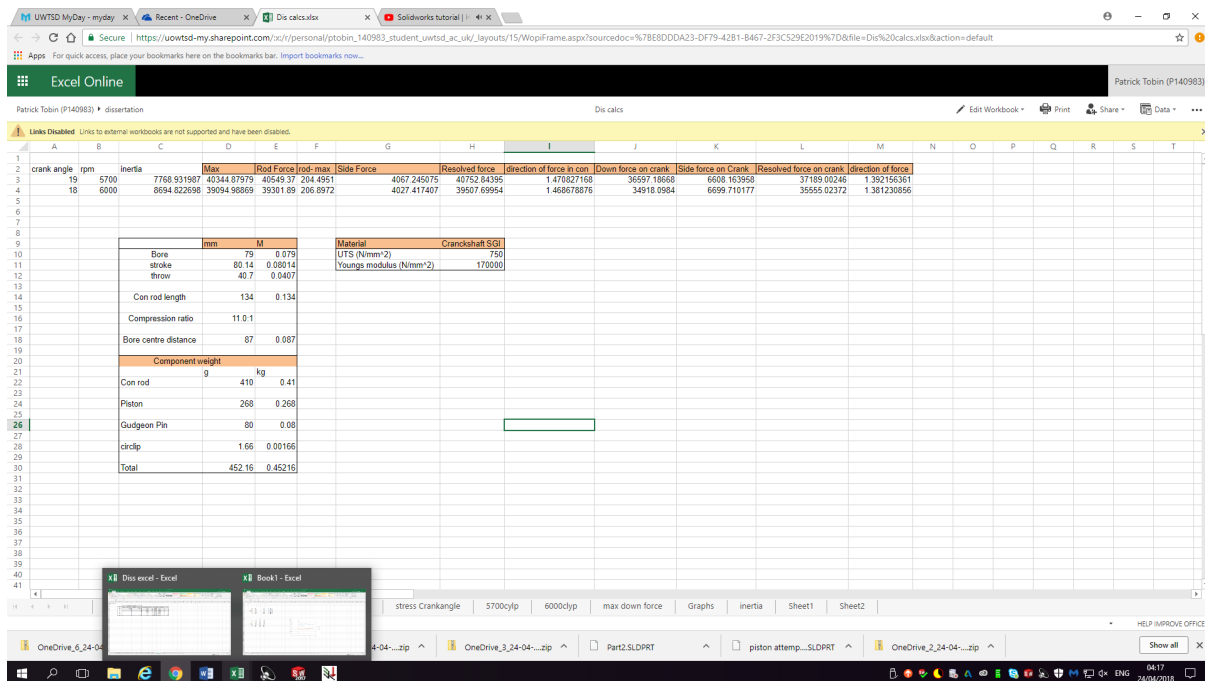
APPENDIX 18 8MM MESH VON MISES



APPENDIX 19 CRANK SPAN CONVERGENCE



APPENDIX 20 FIXED END CRANK SPAN CONVERGENCE



APPENDIX 21 RESOLVED FORCES EXCEL





APPENDIX 25 INERTIA EXCEL FORMULA

Discalcs - Excel												
File Home Insert Page Layout Formulas Data Review View Add-ins ACROBAT Sage Team Tell me what you want to do...												
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Function Library												
Formula Auditing												
Calculation Options												
Calculation												
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APPENDIX 26 ACCELERATION EXCEL FORMULA

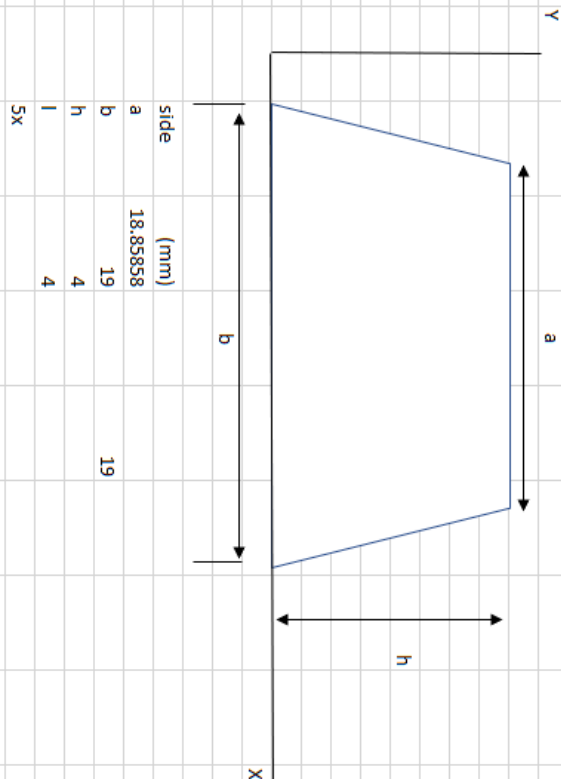
Discalcs - Excel												
Patrick Tobin (P140603) Share												
Function Library												
Formula Auditing												
Calculation Options												
Calculation												
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APPENDIX 27 ROTATIONAL INERTIA FORMULA EXCEL

Discalcs - Excel									
File Home Insert Page Layout Formulas Data Review View Add-ins ACROBAT Sage Team Tell me what you want to do...									
Insert Function AutoSum Recently Financial Logical Text Date & Lookup Math More Functions Use in Formula Trace Dependents Error Checking Evaluate Formula Watch Window Calculation Options Calculate Now Calculate Sheet									
Function Library Defined Names Formula Auditing									
H4	AN	AQ	AP	AQ	AR	AS	AT	AU	
1									
2									
3									
4	Ind. max	Side Force	Resolved force	direction of force in con	Down force on crank	Side force on Crank	Resolved force on crank	direction of force	
5	=AM5.AL5	=HAL5TANF5)	=SQRT((AM92)+(AO92))	=DEGREES(ATAN2(AM5, AO5))	=AM5/COS(F5))AF5	=AM5*SINF5)+(SDS17/3)/2)/SD58*VS102	=SQRT((AR92)+(AS92))	=DEGREES(ATAN2(AR5, AS5))	
6	=AM6.AL6	=HAL6TANF6)	=SQRT((AM92)+(AO92))	=DEGREES(ATAN2(AM6, AO6))	=AM6/COS(F6))AF6	=AM6*SINF6)+(SDS17/3)/2)/SD58*VS102	=SQRT((AR92)+(AS92))	=DEGREES(ATAN2(AR6, AS6))	
7	=AM7.AL7	=HAL7TANF7)	=SQRT((AM92)+(AO92))	=DEGREES(ATAN2(AM7, AO7))	=AM7/COS(F7))AF7	=AM7*SINF7)+(SDS17/3)/2)/SD58*VS102	=SQRT((AR92)+(AS92))	=DEGREES(ATAN2(AR7, AS7))	
8	=AM8.AL8	=HAL8TANF8)	=SQRT((AM92)+(AO92))	=DEGREES(ATAN2(AM8, AO8))	=AM8/COS(F8))AF8	=AM8*SINF8)+(SDS17/3)/2)/SD58*VS102	=SQRT((AR92)+(AS92))	=DEGREES(ATAN2(AR8, AS8))	
9	=AM9.AL9	=HAL9TANF9)	=SQRT((AM92)+(AO92))	=DEGREES(ATAN2(AM9, AO9))	=AM9/COS(F9))AF9	=AM9*SINF9)+(SDS17/3)/2)/SD58*VS102	=SQRT((AR92)+(AS92))	=DEGREES(ATAN2(AR9, AS9))	
10	=AM10.AL10	=HAL10TANF10)	=SQRT((AM102)+(AO102))	=DEGREES(ATAN2(AM10, AO10))	=AM10/COS(F10))AF10	=AM10*SINF10)+(SDS17/3)/2)/SD58*VS102	=SQRT((AR102)+(AS102))	=DEGREES(ATAN2(AR10, AS10))	
11	=AM11.AL11	=HAL11TANF11)	=SQRT((AM112)+(AO112))	=DEGREES(ATAN2(AM11, AO11))	=AM11/COS(F11))AF11	=AM11*SINF11)+(SDS17/3)/2)/SD58*VS102	=SQRT((AR112)+(AS112))	=DEGREES(ATAN2(AR11, AS11))	
12	=AM12.AL12	=HAL12TANF12)	=SQRT((AM122)+(AO122))	=DEGREES(ATAN2(AM12, AO12))	=AM12/COS(F12))AF12	=AM12*SINF12)+(SDS17/3)/2)/SD58*VS102	=SQRT((AR122)+(AS122))	=DEGREES(ATAN2(AR12, AS12))	
13	=AM13.AL13	=HAL13TANF13)	=SQRT((AM132)+(AO132))	=DEGREES(ATAN2(AM13, AO13))	=AM13/COS(F13))AF13	=AM13*SINF13)+(SDS17/3)/2)/SD58*VS102	=SQRT((AR132)+(AS132))	=DEGREES(ATAN2(AR13, AS13))	
14	=AM14.AL14	=HAL14TANF14)	=SQRT((AM142)+(AO142))	=DEGREES(ATAN2(AM14, AO14))	=AM14/COS(F14))AF14	=AM14*SINF14)+(SDS17/3)/2)/SD58*VS102	=SQRT((AR142)+(AS142))	=DEGREES(ATAN2(AR14, AS14))	
15	=AM15.AL15	=HAL15TANF15)	=SQRT((AM152)+(AO152))	=DEGREES(ATAN2(AM15, AO15))	=AM15/COS(F15))AF15	=AM15*SINF15)+(SDS17/3)/2)/SD58*VS102	=SQRT((AR152)+(AS152))	=DEGREES(ATAN2(AR15, AS15))	
16	=AM16.AL16	=HAL16TANF16)	=SQRT((AM162)+(AO162))	=DEGREES(ATAN2(AM16, AO16))	=AM16/COS(F16))AF16	=AM16*SINF16)+(SDS17/3)/2)/SD58*VS102	=SQRT((AR162)+(AS162))	=DEGREES(ATAN2(AR16, AS16))	
17	=AM17.AL17	=HAL17TANF17)	=SQRT((AM172)+(AO172))	=DEGREES(ATAN2(AM17, AO17))	=AM17/COS(F17))AF17	=AM17*SINF17)+(SDS17/3)/2)/SD58*VS102	=SQRT((AR172)+(AS172))	=DEGREES(ATAN2(AR17, AS17))	
18	=AM18.AL18	=HAL18TANF18)	=SQRT((AM182)+(AO182))	=DEGREES(ATAN2(AM18, AO18))	=AM18/COS(F18))AF18	=AM18*SINF18)+(SDS17/3)/2)/SD58*VS102	=SQRT((AR182)+(AS182))	=DEGREES(ATAN2(AR18, AS18))	
19	=AM19.AL19	=HAL19TANF19)	=SQRT((AM192)+(AO192))	=DEGREES(ATAN2(AM19, AO19))	=AM19/COS(F19))AF19	=AM19*SINF19)+(SDS17/3)/2)/SD58*VS102	=SQRT((AR192)+(AS192))	=DEGREES(ATAN2(AR19, AS19))	
20	=AM20.AL20	=HAL20TANF20)	=SQRT((AM202)+(AO202))	=DEGREES(ATAN2(AM20, AO20))	=AM20/COS(F20))AF20	=AM20*SINF20)+(SDS17/3)/2)/SD58*VS102	=SQRT((AR202)+(AS202))	=DEGREES(ATAN2(AR20, AS20))	
21	=AM21.AL21	=HAL21TANF21)	=SQRT((AM212)+(AO212))	=DEGREES(ATAN2(AM21, AO21))	=AM21/COS(F21))AF21	=AM21*SINF21)+(SDS17/3)/2)/SD58*VS102	=SQRT((AR212)+(AS212))	=DEGREES(ATAN2(AR21, AS21))	
22	=AM22.AL22	=HAL22TANF22)	=SQRT((AM222)+(AO222))	=DEGREES(ATAN2(AM22, AO22))	=AM22/COS(F22))AF22	=AM22*SINF22)+(SDS17/3)/2)/SD58*VS102	=SQRT((AR222)+(AS222))	=DEGREES(ATAN2(AR22, AS22))	
23	=AM23.AL23	=HAL23TANF23)	=SQRT((AM232)+(AO232))	=DEGREES(ATAN2(AM23, AO23))	=AM23/COS(F23))AF23	=AM23*SINF23)+(SDS17/3)/2)/SD58*VS102	=SQRT((AR232)+(AS232))	=DEGREES(ATAN2(AR23, AS23))	
24	=AM24.AL24	=HAL24TANF24)	=SQRT((AM242)+(AO242))	=DEGREES(ATAN2(AM24, AO24))	=AM24/COS(F24))AF24	=AM24*SINF24)+(SDS17/3)/2)/SD58*VS102	=SQRT((AR242)+(AS242))	=DEGREES(ATAN2(AR24, AS24))	
25	=AM25.AL25	=HAL25TANF25)	=SQRT((AM252)+(AO252))	=DEGREES(ATAN2(AM25, AO25))	=AM25/COS(F25))AF25	=AM25*SINF25)+(SDS17/3)/2)/SD58*VS102	=SQRT((AR252)+(AS252))	=DEGREES(ATAN2(AR25, AS25))	
26	=AM26.AL26	=HAL26TANF26)	=SQRT((AM262)+(AO262))	=DEGREES(ATAN2(AM26, AO26))	=AM26/COS(F26))AF26	=AM26*SINF26)+(SDS17/3)/2)/SD58*VS102	=SQRT((AR262)+(AS262))	=DEGREES(ATAN2(AR26, AS26))	
27	=AM27.AL27	=HAL27TANF27)	=SQRT((AM272)+(AO272))	=DEGREES(ATAN2(AM27, AO27))	=AM27/COS(F27))AF27	=AM27*SINF27)+(SDS17/3)/2)/SD58*VS102	=SQRT((AR272)+(AS272))	=DEGREES(ATAN2(AR27, AS27))	
28	=AM28.AL28	=HAL28TANF28)	=SQRT((AM282)+(AO282))	=DEGREES(ATAN2(AM28, AO28))	=AM28/COS(F28))AF28	=AM28*SINF28)+(SDS17/3)/2)/SD58*VS102	=SQRT((AR282)+(AS282))	=DEGREES(ATAN2(AR28, AS28))	
29	=AM29.AL29	=HAL29TANF29)	=SQRT((AM292)+(AO292))	=DEGREES(ATAN2(AM29, AO29))	=AM29/COS(F29))AF29	=AM29*SINF29)+(SDS17/3)/2)/SD58*VS102	=SQRT((AR292)+(AS292))	=DEGREES(ATAN2(AR29, AS29))	
30	=AM30.AL30	=HAL30TANF30)	=SQRT((AM302)+(AO302))	=DEGREES(ATAN2(AM30, AO30))	=AM30/COS(F30))AF30	=AM30*SINF30)+(SDS17/3)/2)/SD58*VS102	=SQRT((AR302)+(AS302))	=DEGREES(ATAN2(AR30, AS30))	
31	=AM31.AL31	=HAL31TANF31)	=SQRT((AM312)+(AO312))	=DEGREES(ATAN2(AM31, AO31))	=AM31/COS(F31))AF31	=AM31*SINF31)+(SDS17/3)/2)/SD58*VS102	=SQRT((AR312)+(AS312))	=DEGREES(ATAN2(AR31, AS31))	
32	=AM32.AL32	=HAL32TANF32)	=SQRT((AM322)+(AO322))	=DEGREES(ATAN2(AM32, AO32))	=AM32/COS(F32))AF32	=AM32*SINF32)+(SDS17/3)/2)/SD58*VS102	=SQRT((AR322)+(AS322))	=DEGREES(ATAN2(AR32, AS32))	
33	=AM33.AL33	=HAL33TANF33)	=SQRT((AM332)+(AO332))	=DEGREES(ATAN2(AM33, AO33))	=AM33/COS(F33))AF33	=AM33*SINF33)+(SDS17/3)/2)/SD58*VS102	=SQRT((AR332)+(AS332))	=DEGREES(ATAN2(AR33, AS33))	
34	=AM34.AL34	=HAL34TANF34)	=SQRT((AM342)+(AO342))	=DEGREES(ATAN2(AM34, AO34))	=AM34/COS(F34))AF34	=AM34*SINF34)+(SDS17/3)/2)/SD58*VS102	=SQRT((AR342)+(AS342))	=DEGREES(ATAN2(AR34, AS34))	
35	=AM35.AL35	=HAL35TANF35)	=SQRT((AM352)+(AO352))	=DEGREES(ATAN2(AM35, AO35))	=AM35/COS(F35))AF35	=AM35*SINF35)+(SDS17/3)/2)/SD58*VS102	=SQRT((AR352)+(AS352))	=DEGREES(ATAN2(AR35, AS35))	
36	=AM36.AL36	=HAL36TANF36)	=SQRT((AM362)+(AO362))	=DEGREES(ATAN2(AM36, AO36))	=AM36/COS(F36))AF36	=AM36*SINF36)+(SDS17/3)/2)/SD58*VS102	=SQRT((AR362)+(AS362))	=DEGREES(ATAN2(AR36, AS36))	
37	=AM37.AL37	=HAL37TANF37)	=SQRT((AM372)+(AO372))	=DEGREES(ATAN2(AM37, AO37))	=AM37/COS(F37))AF37	=AM37*SINF37)+(SDS17/3)/2)/SD58*VS102	=SQRT((AR372)+(AS372))	=DEGREES(ATAN2(AR37, AS37))	
38	=AM38.AL38	=HAL38TANF38)	=SQRT((AM382)+(AO382))	=DEGREES(ATAN2(AM38, AO38))	=AM38/COS(F38))AF38	=AM38*SINF38)+(SDS17/3)/2)/SD58*VS102	=SQRT((AR382)+(AS382))	=DEGREES(ATAN2(AR38, AS38))	
39	=AM39.AL39	=HAL39TANF39)	=SQRT((AM392)+(AO392))	=DEGREES(ATAN2(AM39, AO39))	=AM39/COS(F39))AF39	=AM39*SINF39)+(SDS17/3)/2)/SD58*VS102	=SQRT((AR392)+(AS392))	=DEGREES(ATAN2(AR39, AS39))	
40	=AM40.AL40	=HAL40TANF40)	=SQRT((AM402)+(AO402))	=DEGREES(ATAN2(AM40, AO40))	=AM40/COS(F40))AF40	=AM40*SINF40)+(SDS17/3)/2)/SD58*VS102	=SQRT((AR402)+(AS402))	=DEGREES(ATAN2(AR40, AS40))	
41	=AM41.AL41	=HAL41TANF41)	=SQRT((AM412)+(AO412))	=DEGREES(ATAN2(AM41, AO41))	=AM41/COS(F41))AF41	=AM41*SINF41)+(SDS17/3)/2)/SD58*VS102	=SQRT((AR412)+(AS412))	=DEGREES(ATAN2(AR41, AS41))	
42	=AM42.AL42	=HAL42TANF42)	=SQRT((AM422)+(AO422))	=DEGREES(ATAN2(AM42, AO42))	=AM42/COS(F42))AF42	=AM42*SINF42)+(SDS17/3)/2)/SD58*VS102	=SQRT((AR422)+(AS422))	=DEGREES(ATAN2(AR42, AS42))	
43	=AM43.AL43	=HAL43TANF43)	=SQRT((AM432)+(AO432))	=DEGREES(ATAN2(AM43, AO43))	=AM43/COS(F43))AF43	=AM43*SINF43)+(SDS17/3)/2)/SD58*VS102	=SQRT((AR432)+(AS432))	=DEGREES(ATAN2(AR43, AS43))	
44	=AM44.AL44	=HAL44TANF44)	=SQRT((AM442)+(AO442))	=DEGREES(ATAN2(AM44, AO44))	=AM44/COS(F44))AF44	=AM44*SINF44)+(SDS17/3)/2)/SD58*VS102	=SQRT((AR442)+(AS442))	=DEGREES(ATAN2(AR44, AS44))	
45	=AM45.AL45	=HAL45TANF45)	=SQRT((AM452)+(AO452))	=DEGREES(ATAN2(AM45, AO45))	=AM45/COS(F45))AF45	=AM45*SINF45)+(SDS17/3)/2)/SD58*VS102	=SQRT((AR452)+(AS452))	=DEGREES(ATAN2(AR45, AS45))	
46	=AM46.AL46	=HAL46TANF46)	=SQRT((AM462)+(AO462))	=DEGREES(ATAN2(AM46, AO46))	=AM46/COS(F46))AF46	=AM46*SINF46)+(SDS17/3)/2)/SD58*VS102	=SQRT((AR462)+(AS462))	=DEGREES(ATAN2(AR46, AS46))	

APPENDIX 28 CRANK FORCES EXCEL FORMULA

B	C	D	E	F	G	H	I	J	K	L	M	N	C
Stress	force (N) area (mm ²)	15000 37.85857864		396.2113882									
stiffness	youngs (Gpa) area (mm ²) length (mm)	200 3.78586E-05 0.004											
		1892.928932											
bend m	force (N) distance (mm)	30000 12.07071068											
bend stre	Bend m (N) y		362.1213203										
chamfer size (m)	0.1	0.070710678											
distance to load (m)	12												
lxx (m ⁴)	201.9124194												



APPENDIX 29 INERTIA CALCULATIONS