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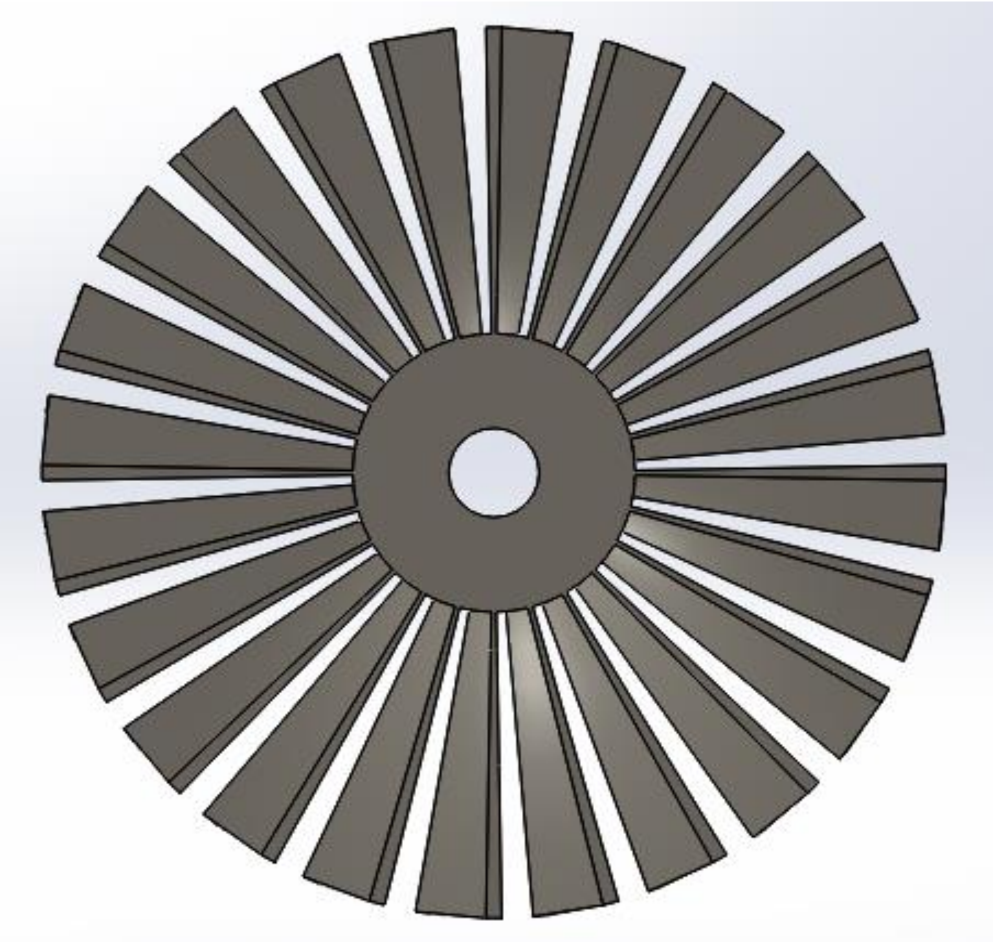
BSc

Advanced Mechanics of Solids

Turbine Blade

Word Count: 3300



Summary

This report will examine the failures of a Steam Turbine Blade using analytical methods, decision-making methodology and research. It will be used to determine the blades material selection according to requirements. My aim is to give a warranty of at least a year under constraints of stress and defects. Throughout this report a thorough test of materials will be undertaken. Testing for stress and strains, fatigue, creep and life cycle, this information is then used to design the Turbine Blade in SolidWorks. FEA Analysis and Hand Calculations will be compared and used to prove if the design is valid or not.

Basic Blade designed in SolidWorks.

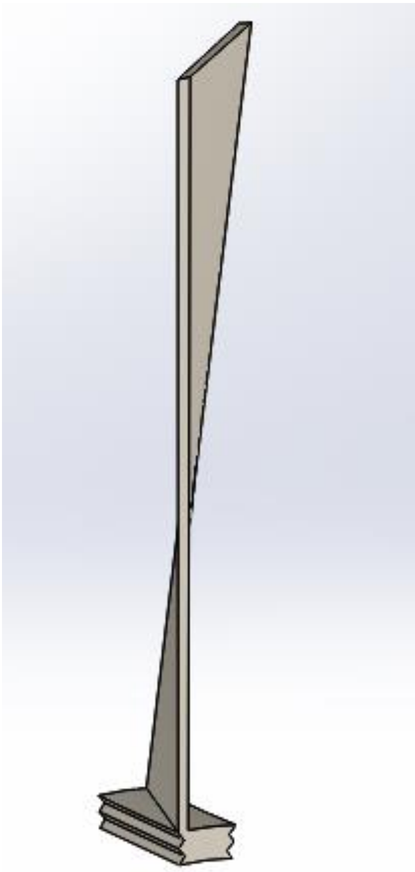


Figure 1 Blade Design SolidWorks

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Introduction

This report is based on the importance of choosing the correct material for a steam turbine blade, materials will be selected and will be subjected to breaking point conditions, this is to see where weakness lies. A turbine blade is subject to temperature, pressure and rotational forces, properties of these forces include, creep, fatigue, stress and strain, these factors dictate lifetime cycle, testing materials for these properties will help determine if the material is sufficient for its intended use.

Testing

The most common failures are listed below, these need to be calculated, and then to prove the hand calculations are correct, FEA Analysis will be used via SolidWorks, and optimisation to enhance the design finished with a conclusion including critical analysis.

Table 1 Blade Failures

Blade Failures	**%**
Unknown	26%
Stress & Corrosion Cracking	22%
High Cycle Fatigue	20%
Corrosion Fatigue Cracking	7%
Temperature Creep Rupture	6%
Low Cycle Fatigue	5%
Corrosion	4%
Other Causes	10%

Methodology

- Select a material for a turbine blade.
- Test the material.
- Compare the results with SolidWorks comparison and other techniques.
- Critically analyse the turbine blade.
- Critical conclusion.

Report

Steam Turbine Features

The steam turbine operates on the "Rankine Cycle" and is named the (CJR-100). Steam Turbine parameters such as speed, temperature and power-output are known quantities which aid the design of this Turbine Blades.

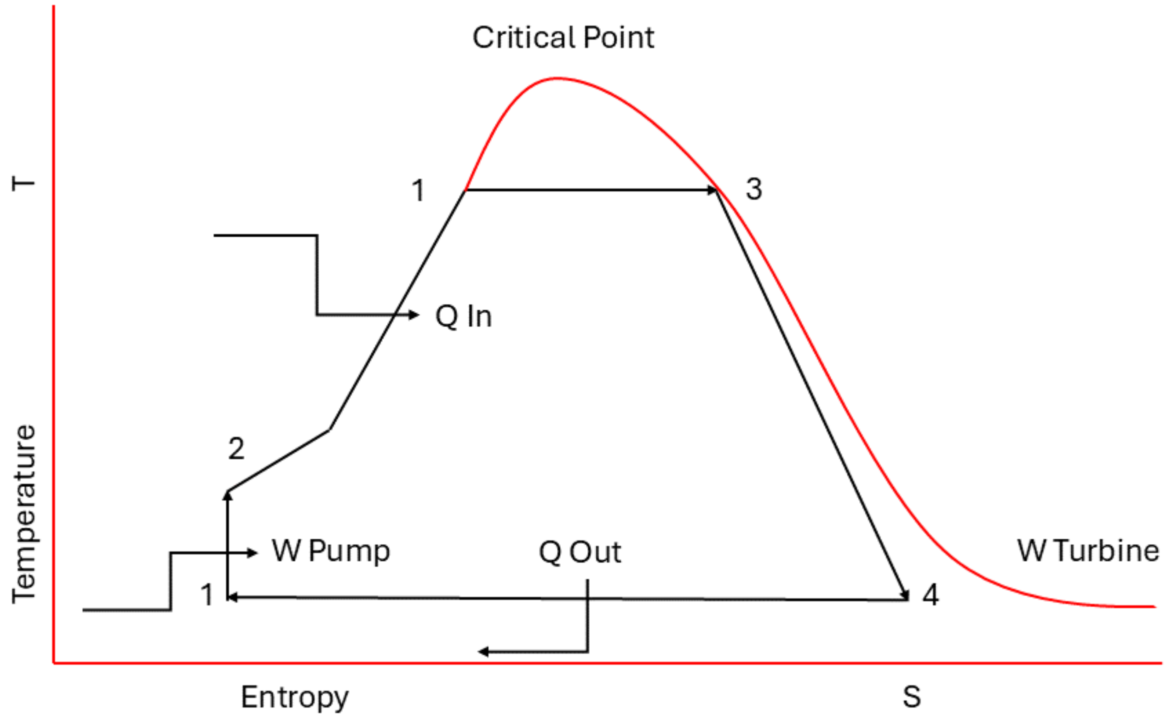

Figure 2 Ideal Rankine Cycle

Design Parameters

Using Rankine Cycle, steam turbine model (CJR-100, 85 bar, 480°C); It has a similar specification to that of a Siemens SST-100. (Siemens, 2013)

Steam Turbine Specification

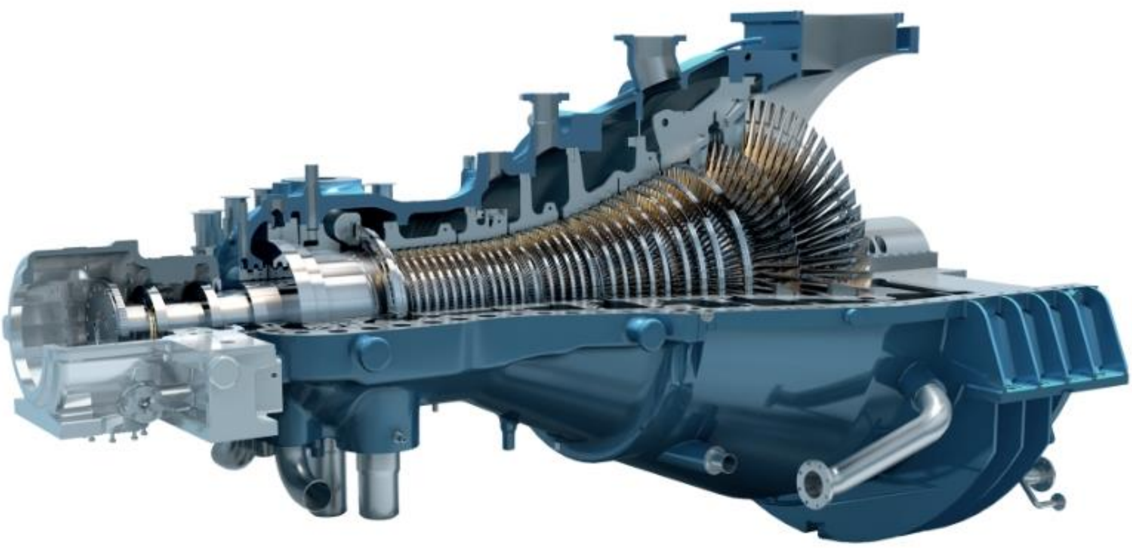

Figure 3 Steam Turbine

Table 2 Steam Turbine Specification CJR-100

Power Plant	**Power Output**	**Inlet Pressure**	**Inlet Temperature**	**Rotor Disc Rpm**
Rankine	8.5 MW	85 Bar	480°C	7.500

Power Plants

In thermal power plants, energy is extracted from steam under high pressure and at a high temperature. The steam is produced in a boiler or heat recovery steam generator and is routed to a steam turbine where it is partly expanded in a high-pressure stage, extracting energy from the steam as it passes through turbine blades. The steam is then returned to the boiler for reheating to improve efficiency, after which it is returned to the turbine to continue expanding and extracting energy in several further stages. The steam turbine shaft rotates at 125 revolutions per minute (rpm) just over (2 revolutions per second).

Turbine blades

Figure 4 Turbine Blade Example (Carter,2018)

Turbine blades come in various shapes and sizes, for this report I will design a basic turbine blade.

Blades Components

- Blade
- Blade Tip
- Root (Fir Tree)
- 180° Twist

Blade Design with Root

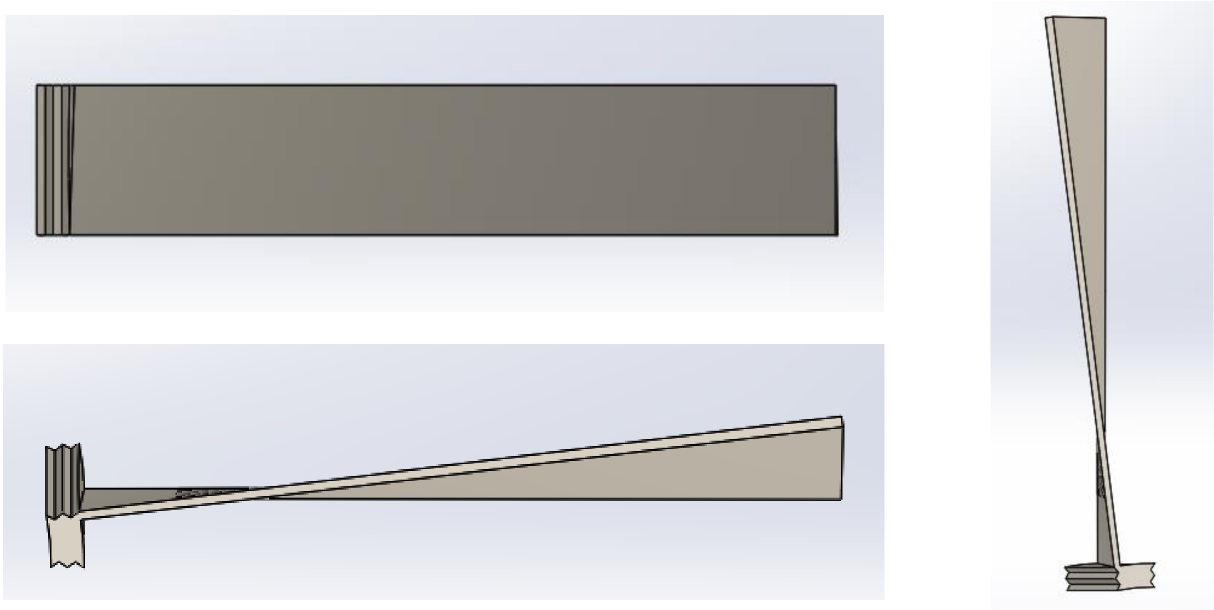

Figure 5 Blade Design with Root

Note: At certain points throughout this report**,** for testing purposes and calculations we are going to neglect the root, tip of the blade and the 180° twist which then leaves a rectangular prism shape.

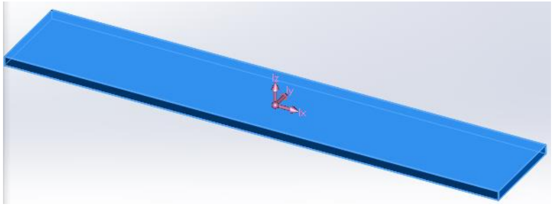

Figure 6 Rectangular Prism

However, this shape will be connected to a hypothetical rotor disc for centrifugal forces as shown below (The Royal Academy of Engineering, ND).

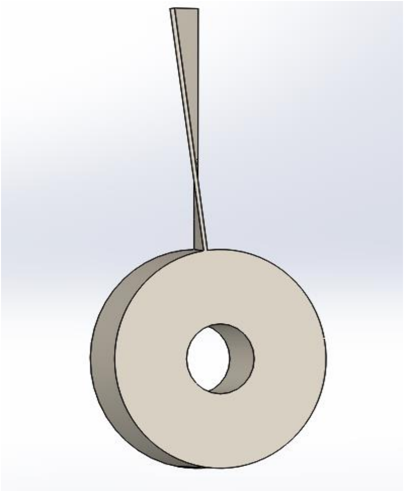

Figure 7 Blade and Turbine Disk

The root is not subjected to the intense force unlike the blade due to stature and centrifugal forces. Therefore, it is not required for testing, but does need calculating to obtain test results.

Before we start, I need to determine the turbines blade material, looking on the market for materials, knowing the Rankine cycles turbine power output at 8.5 kW, and a temperature requirement of $480^{\circ}C\ @\ 85\ bar$, the material selected must be able to withstand high temperatures, pressures and rotational forces. I am not going to select random materials run tests and choose one, I will choose a material worthy but then optimise that part if necessary.

Material Selection

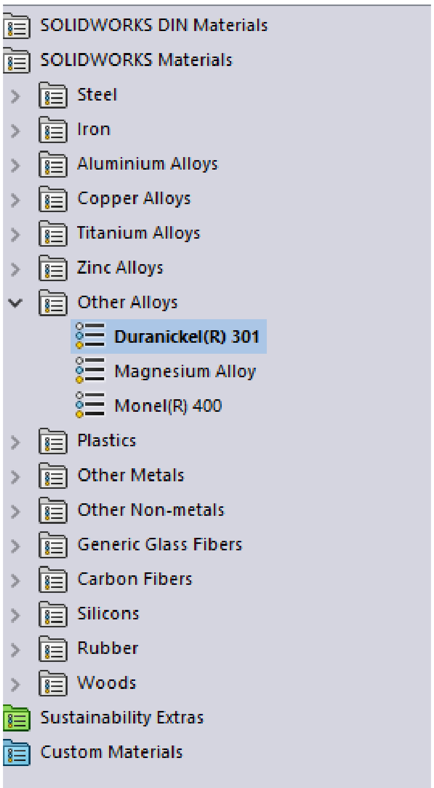

Figure 8 Material Selection

Duranickel 301 Composition

Table 3 Duranickel 301 Composition

Element	Content (%)
Nickel, Ni	≥ 93
Aluminium, Al	4.0-4.75
Silicon, Si	≤ 1
Iron, Fe	≤ 0.60
Manganese, Mn	≤ 0.50
Carbon, C	≤ 0.30
Titanium, Ti	0.25-1
Copper, Cu	≤ 0.25
Sulphur, S	≤ 0.010

(AZO Materials, 2012)

Table 4 Duranickel Facts

Elongation at break	15-35%
Hardness, Brinell (3000 kg)	185-300
Hardness, Rockwell B	≥90
Hardness, Rockwell C	≤40

(AZO Materials, 2012)

Table 5 Duranickel Properties

Sound Thin Rod	4900 m/s (at r.t)
Electrical Resistivity	69.3 nΩ·m (20 °C)
Magnetic Ordering	ferromagnetic
Bulk Modulus	180 Gpa
Mohs Hardness	4.0
Vickers Hardness	638 MPa
Brinell Hardness	667–1600 MPa
Cas Number	7440-02-0
Melting Point	1427°C

(AZO Materials, 2012)

FCC Crystalline Structure

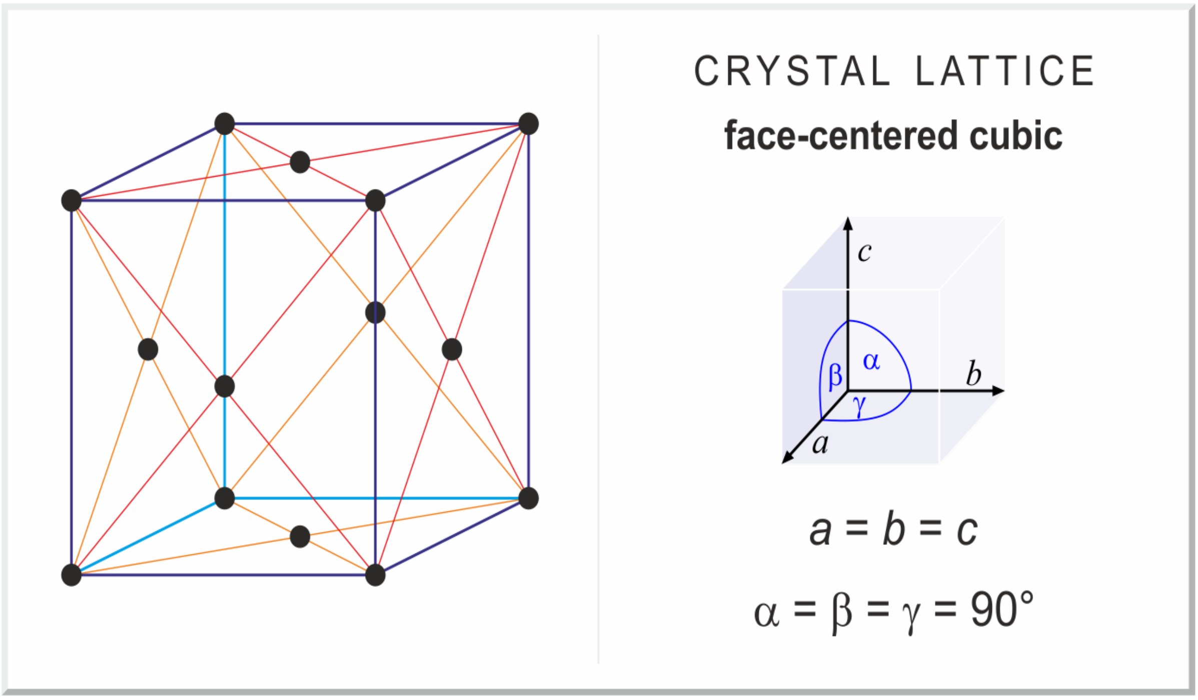

Figure 9 Nickel Crystal Lattice Face-Centred Cubic

(Chemistry Glossary, 2018)

Grain Structure of Nickel

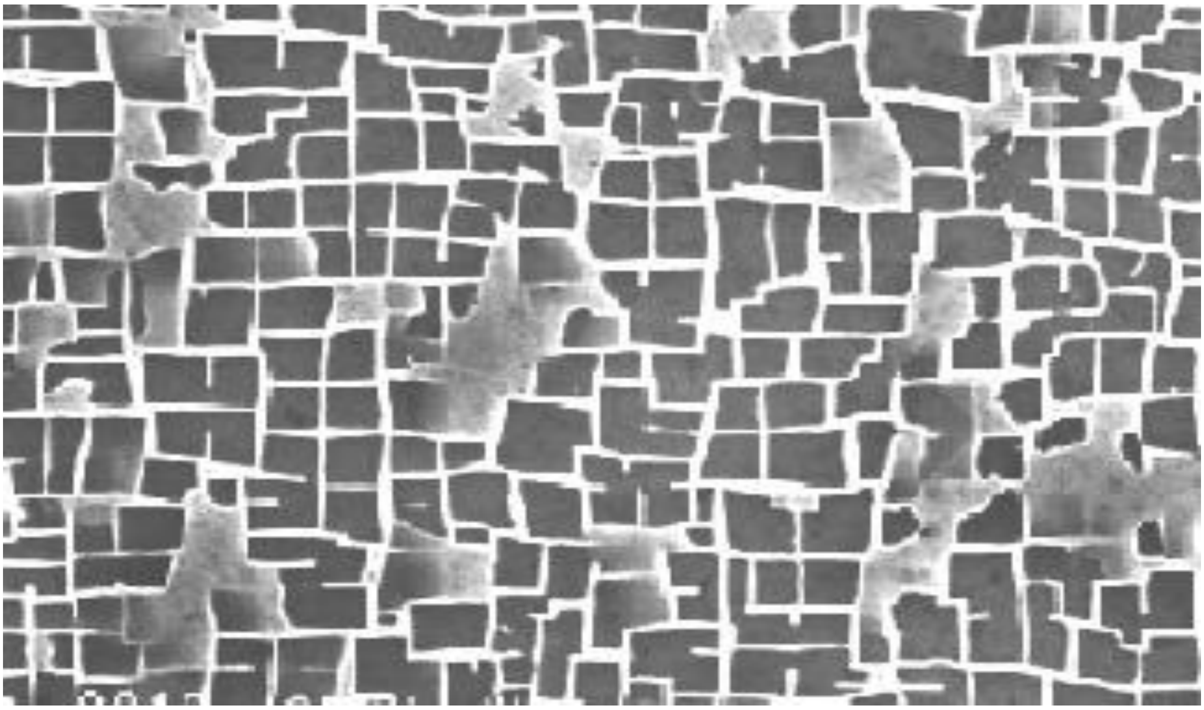

Figure 10 Grain Structure

Nickel based superalloys are normally used in load-bearing structures to the highest temperature of any common alloy system up to 90% of their own melting point. Used amongst the most demanding applications for a structural material are those in the hot sections of turbine engines (TMS.org, 2000).

Justification

I have selected Duranickel via SolidWorks, this material as research suggests can withstand high temperatures, the specific heat is 590° this gives rise to other values which we will use later to determine not only if this material can withstand heat, but other forces such as creep and fatigue.

Values obtained via Solid Works for Duranickel 301 Material

Table 6 Values for Duranickel 301 Material

Properties	**Value**	**Units**
Mass of Blade	563.7	G
Elastic Modulus	2.1e+011	N/m^2
Poisons Ratio	0.34	N/A
Shear Modulus	7.7e+010	N/m^2
Mass Density	8200	Kg/m^3
Tensile Strength	620420000/620 Mpa	N/m^2
Yield Strength	344678000/345 Mpa	N/m^2
Thermal Expansion Coefficient	1.5e-005	/k
Thermal Conductivity	24	w/(m-k)
Specific Heat	590	J/Kg/k
Thermal Expansion	1.5e - 00.5	K

Parameters of the Turbine Blade

Diameter of complete blade set = 800mm

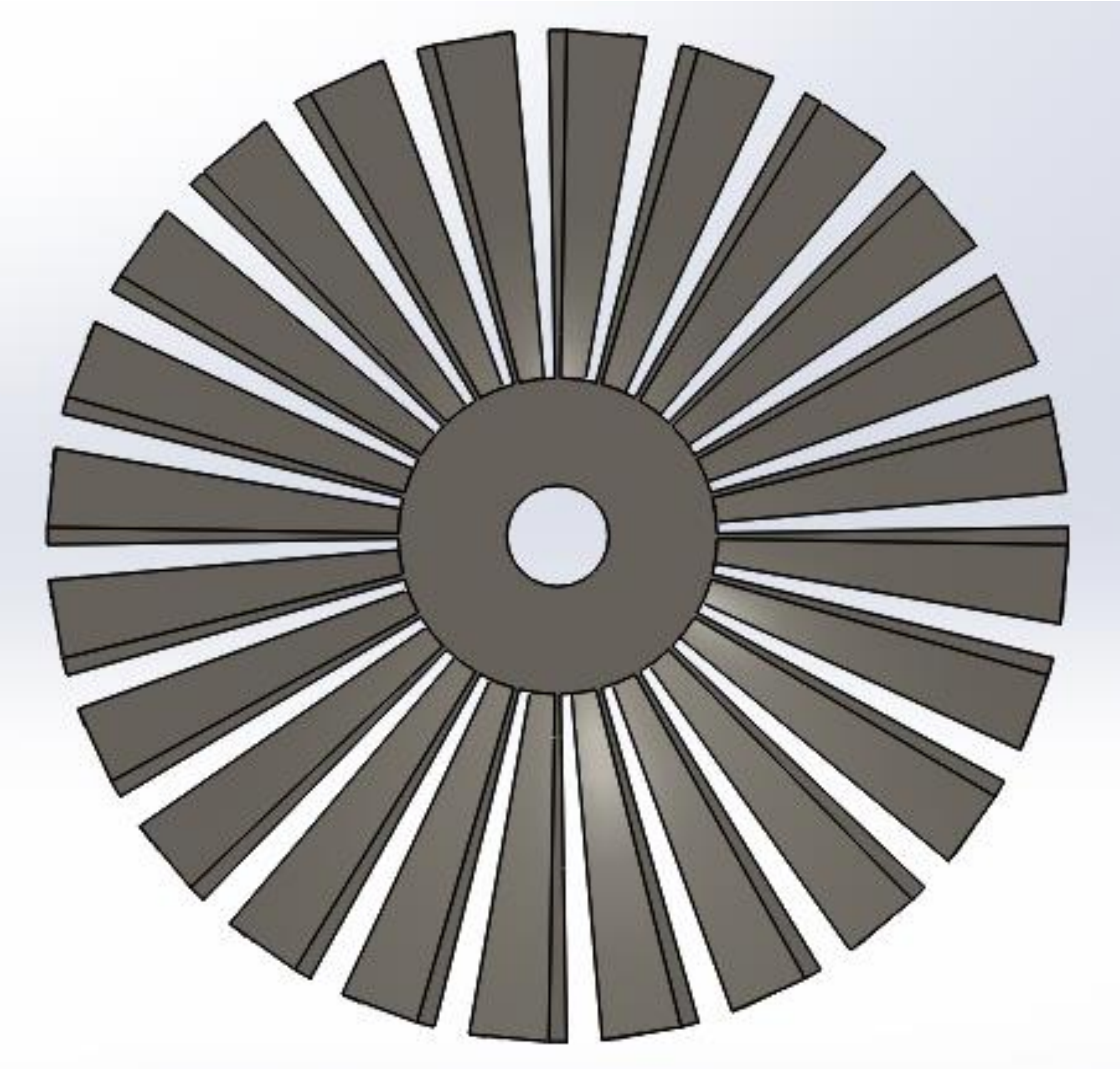

Figure 11 Turbine Blade

Single Blade Design with Rotor Disc

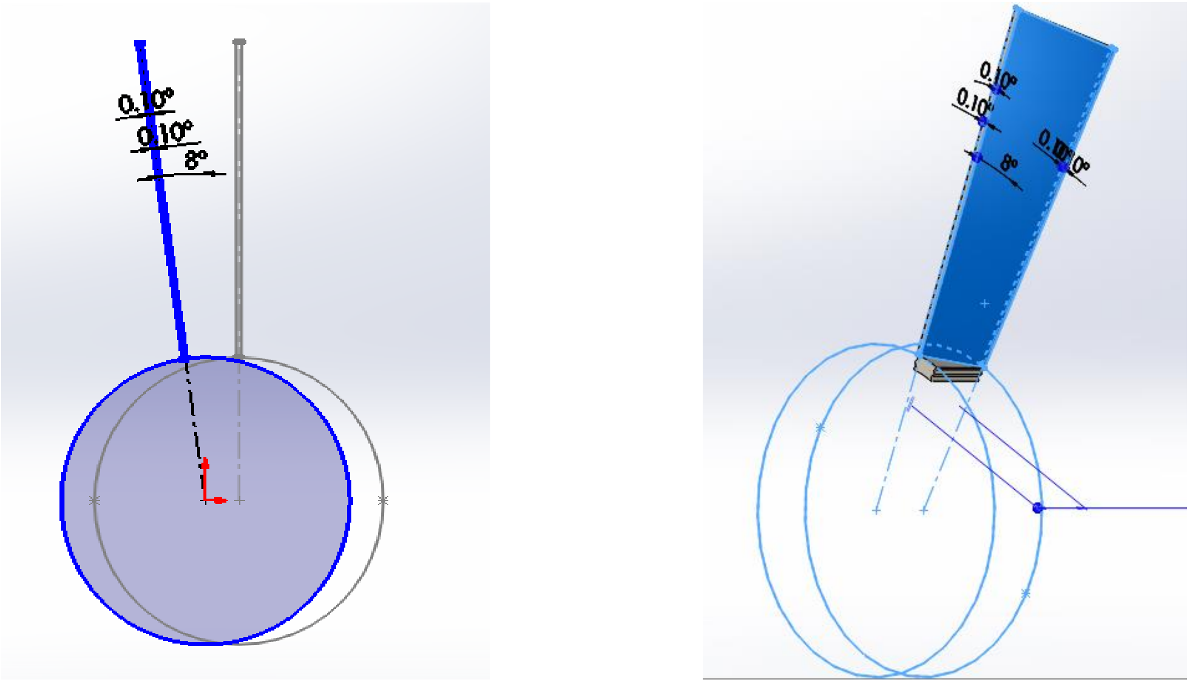

Figure 12 Single Blade Design

Blade Height = 275mm
Blade Width = 50mm
Blade Thickness = 5mm
Centre of the shaft to blade tip = 400mm
Centre of shaft to root tip = 125mm

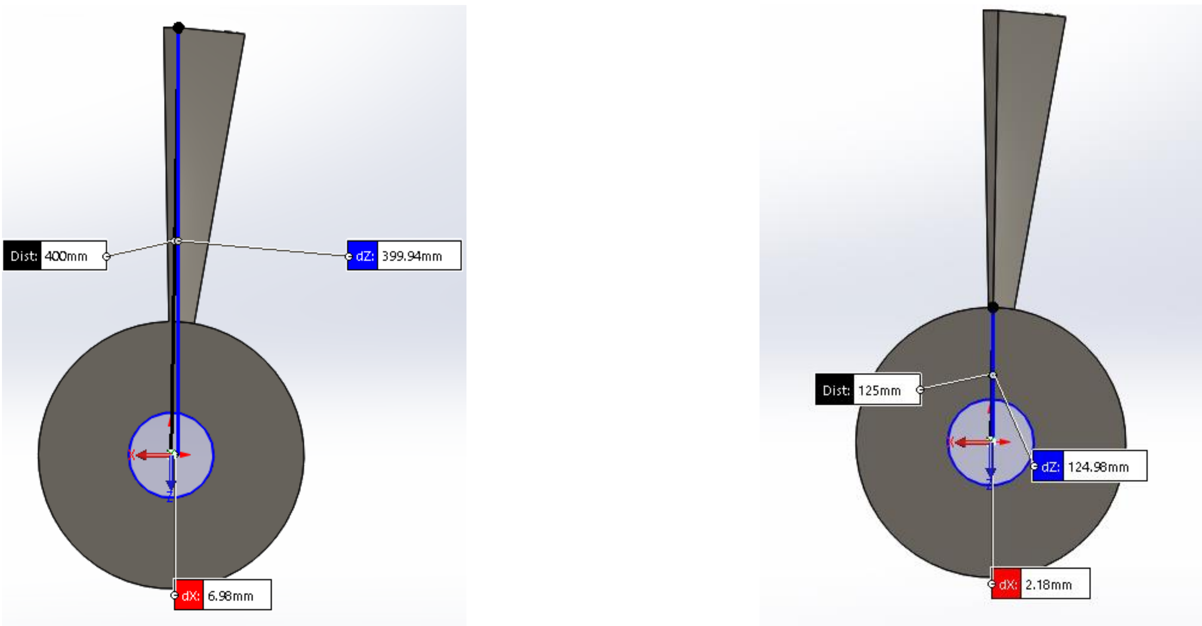

Figure 13 Single Turbine Blade Design

Calculations

First, simplify the blade design using mass calculation via solid works, split it into two parts, blade and root, and then select a suitable material in SolidWorks. By doing this we can determine the surface area and mass individually and then by adding the blade and root together for a total area of surface and mass. The reason we neglect the root is because when it is attached to the rotor Disc it becomes a part of the Disc, we are not testing the disc, only the blade is subject to forces. However, root calculations are necessary to obtain blade data.

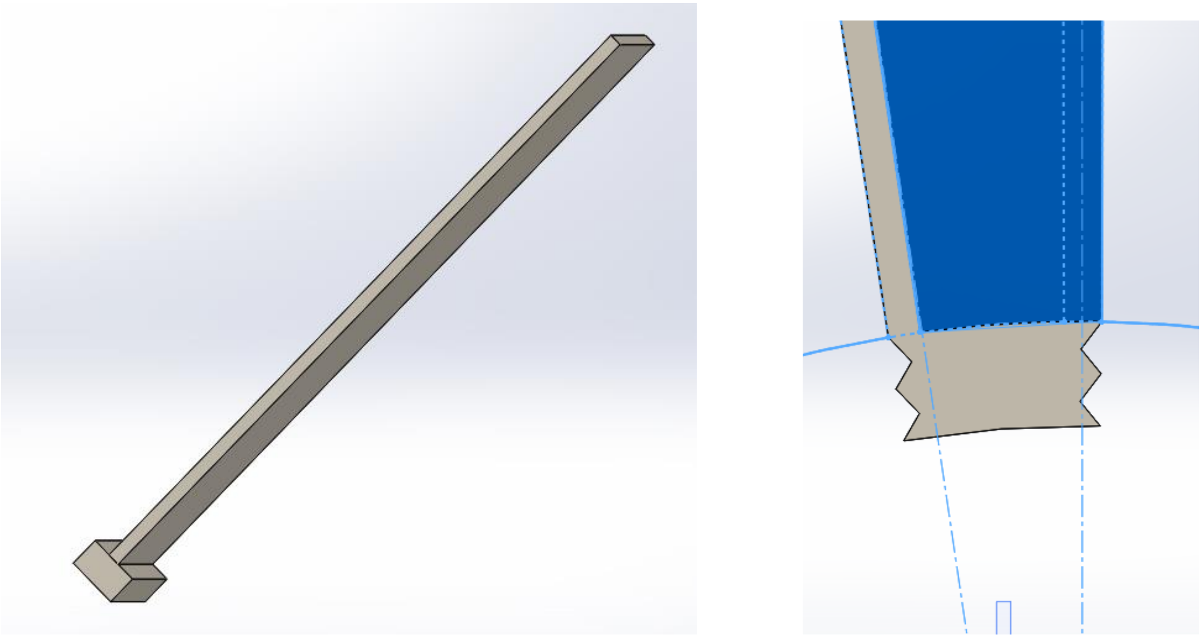

Figure 14 Simplified Root and Blade

Blade Surface Area

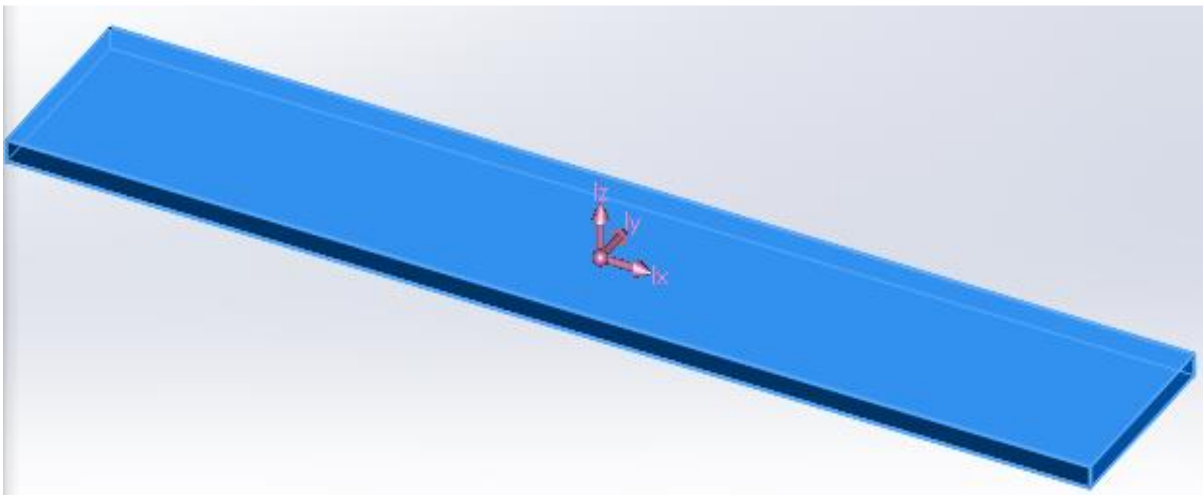

Figure 15 Blade Surface Area

Note: Assume a rectangle prism shape neglecting the root, blade tip and twist.

$5\ x\ 275 = 1375mm$

$50\ x\ 5 = 250mm$

$50\ x\ 275 = 13750$

$13750 + \ 1375\ + 250 = \ 15375mm$

$2\ x\ 15375 = 30750mm\ 6\ faces\ to\ rectangle\ prism$

$Surface\ area = 30750mm^2$

Mass properties of prism rectangle
Configuration: Default
Coordinate system: -- default --

Density = 0.01 grams per cubic millimeter

Mass = 563.75 grams

Volume = 68750.00 cubic millimeters

Surface area = 30750.00 square millimeters

Figure 16 Mass, Surface Area, and Volume

Root Surface Area

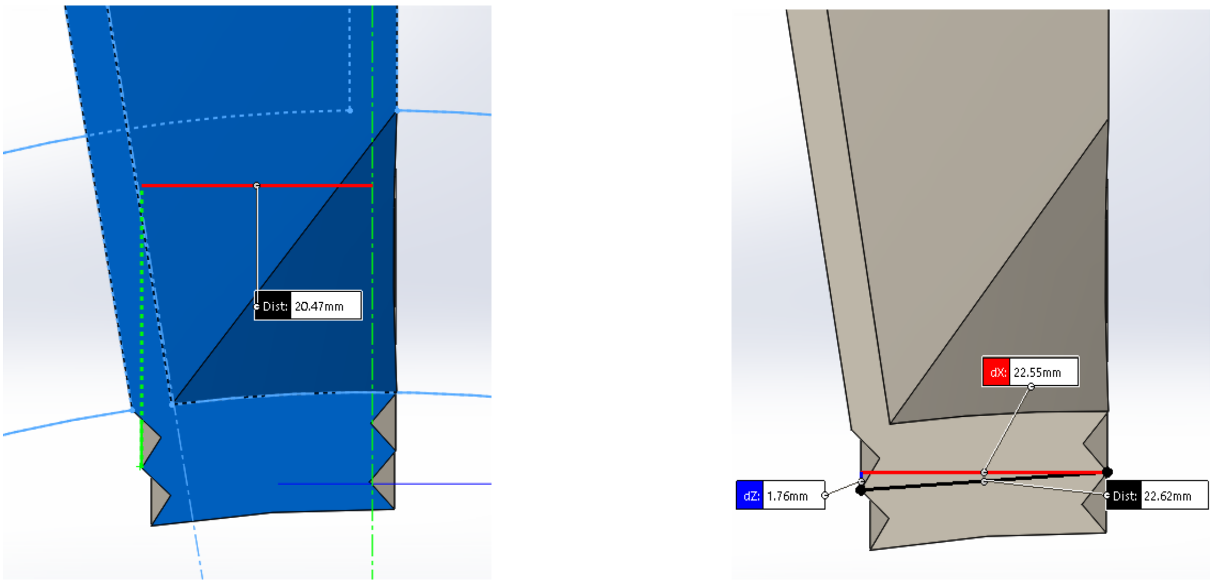

Figure 17 Root Surface Area

Simplify root design, original width = 22.62mm transform into rectangle prism so mass remains the same.

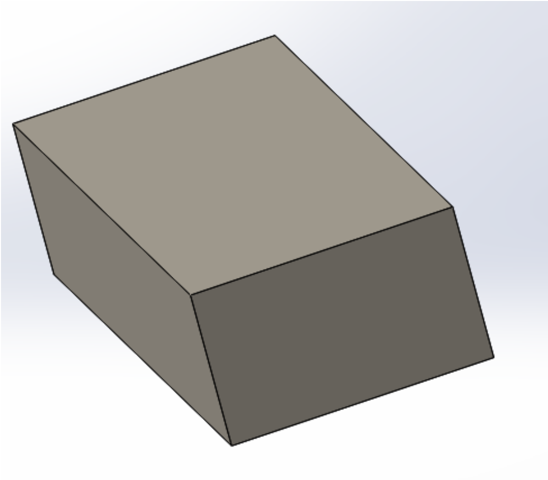

Figure 18 Rectangular Prism

Root Surface Area

Root Height = 12mm
Root Width = 20.47mm
Root Length = 50mm

$50\ x\ 12 = 600mm$

$20.47\ x\ 50 = 1023.5mm$

$20.47\ x\ 12 = 245.64mm$

$600 + \ 1023.5\ + 245.6 = \ 1869.14mm$

$2\ x\ 1869.14 = 3738.28mm\ (6\ faces\ to\ a\ rectangle\ prism)$

$Surface\ area = 3738.28mm^2$

Mass = 100.71 grams

Volume = 12282.00 cubic millimeters

Surface area = 3738.28 square millimeters

Figure 19 SolidWorks Mass, volume, and Surface Area Calculations

Blade Including Root Properties

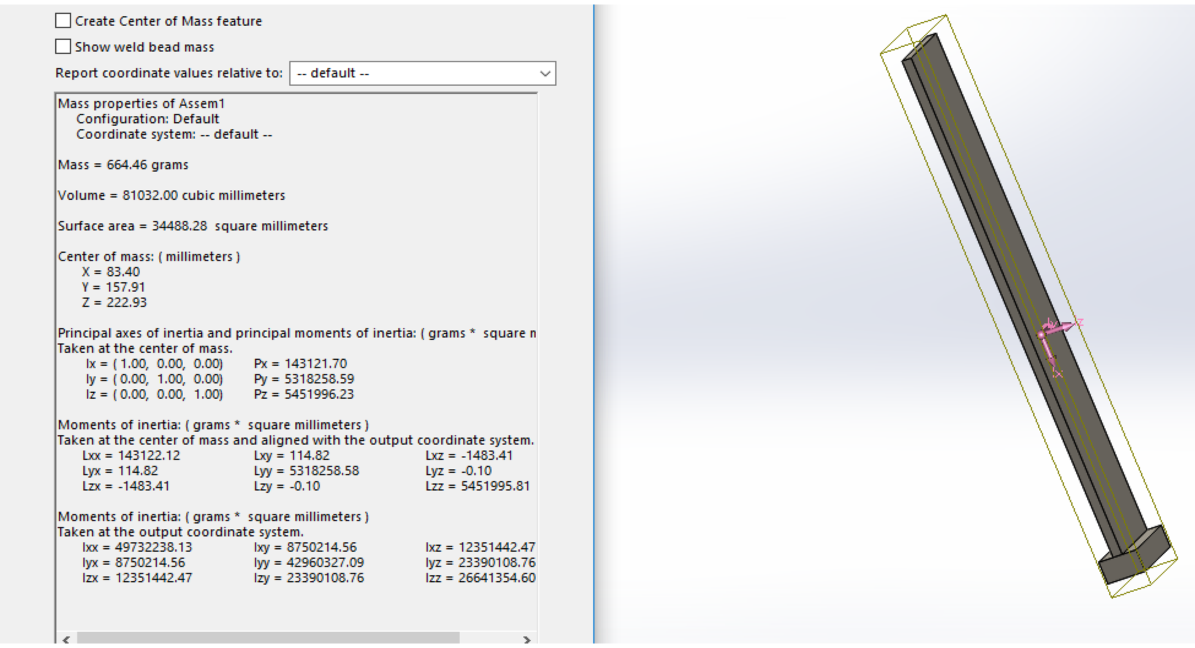

Figure 20 Blade and Root Properties

Table 7 Blade Root Properties

Mass	664 g
Surface area	34488.28mm^2

Actual weight, due to thinning of the blade, curvature of the root base = 591 Grams, Surface area = 42210mm^2 value is higher due to angle of the blade, but we will use the simplified SolidWorks Calculations from the table above which will give higher values, assume a small percentage decrease in overall values to obtain accuracy of the original blade design as regards to dimensions and mass.

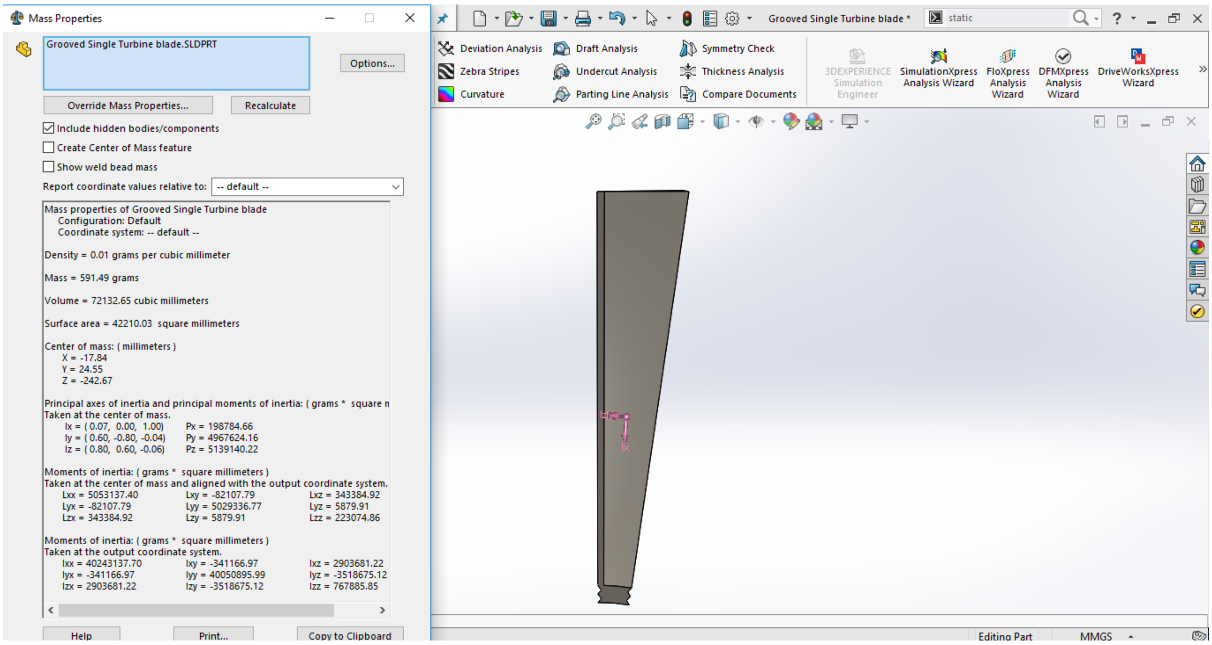

Figure 21 Actual Mass Properties

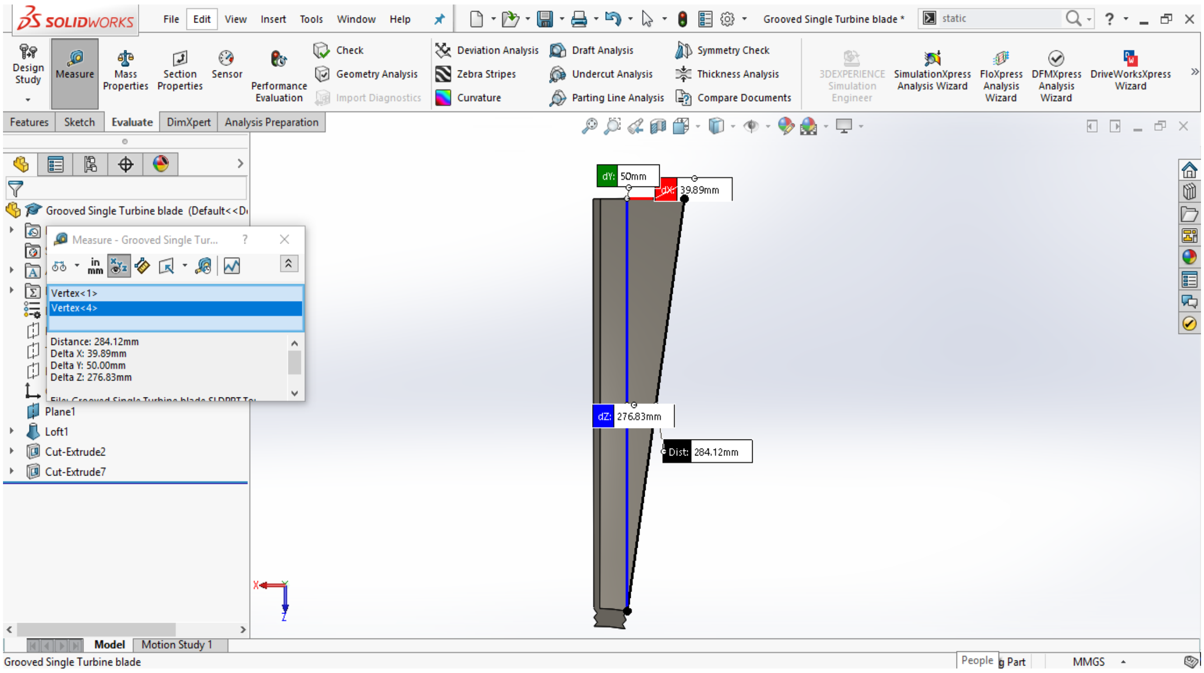

Figure 22 Angle Increasing Surface Area 1

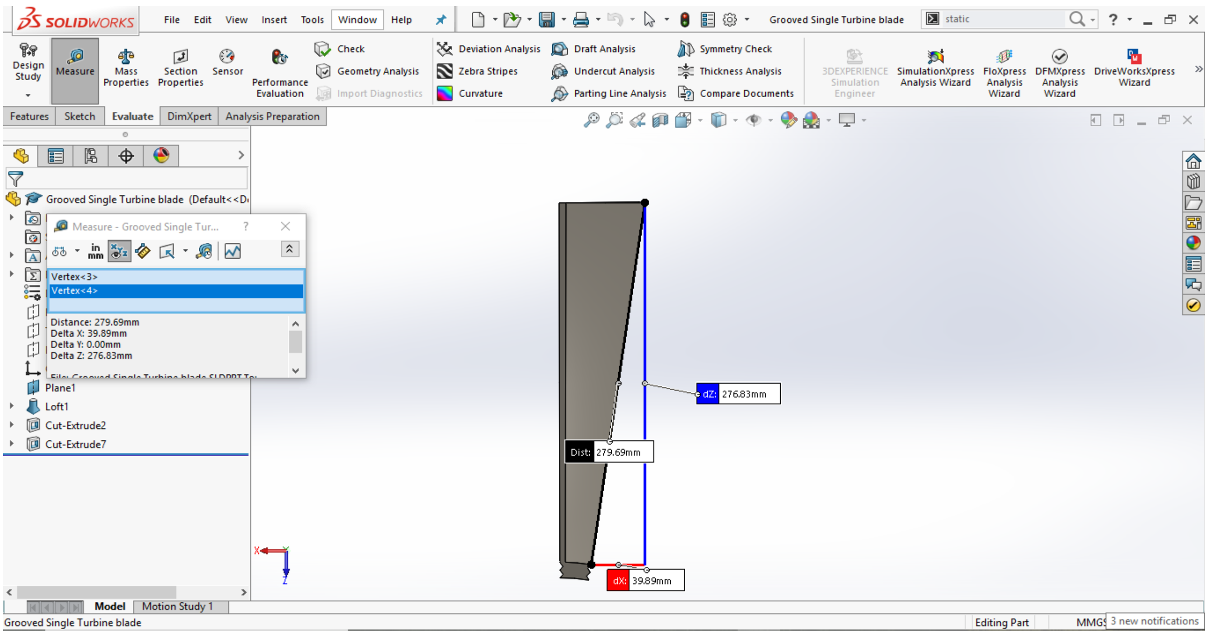

Figure 23 Angle Increasing Surface Area 2

Centripetal Force, Turbine Blade

The centripetal force is the external force required to make a body follow a curved path. Any force (gravitational, electromagnetic, etc.) or combination of forces can act to provide a centripetal force. This force is directed inwards, towards the centre of curvature of the path.

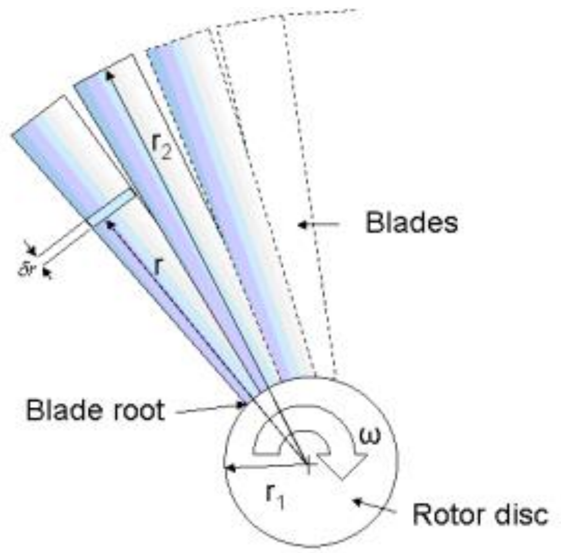

Figure 24 Centripetal Force

Blade Length

$r_2 = 400mm$

$r_1 = 125mm$

$(r_2 - r_1) = (400 - 125)$

$= 275mm$

$= 0.275m$

Blade Root Cross Section Area

$12\ x\ 50 = 0.245m^2\ (A_{root})$

Angular Velocity

$\frac{7500x2\pi}{60} = 250\pi = 785.4\ Radians\ per\ second$

$= 125\ RPS$

Blade Tip Radius

$r_1 = r_2 - (r_2 - r_1) = 125mm$

$= 0.125m$

$0.4 - 0.125 = 0.275mm$

Force

$$F = pA\omega^2[\frac{(r_2^2 - r_1^2)}{2}]$$

$$F = 8200\ x\ 0.245\ x\ (785.4)^2 x[\frac{(0.4^2 - 0.275^2)}{2}] = 52281196.81\ N$$

$$= 52281.2kN$$

$$= 52.3\ Mpa$$

(The Royal Academy of Engineering, ND)

Conclusion

Table 8 Results

Tensile strength	620420000	N/m²
Yield strength	344678000	N/m²

Duranickel 301, is capable of absorbing 52281.2kN/m^2 of rotational forces, given results in table for tensile and yield strengths.

Stress

1 Pascal = N / m^2 = kg/(ms^2)*

$$\sigma = \frac{f}{A_{root}}$$

$$\sigma = \frac{52281196.81}{0.245} = 213392640$$

$$\sigma = 213392640\ N/m^2$$

$$\sigma = 213.4\ Mpa$$

Steam Pressure Force on Blade

$$p = \frac{F}{A}$$

$$p = \frac{85\ bar}{A_{root}}$$

$$p = \frac{8.5x10^6}{0.245} = 34.6\ Mpa$$

SolidWorks Deflection Pressure over blade

Steam pressure causes deflection across a blade.

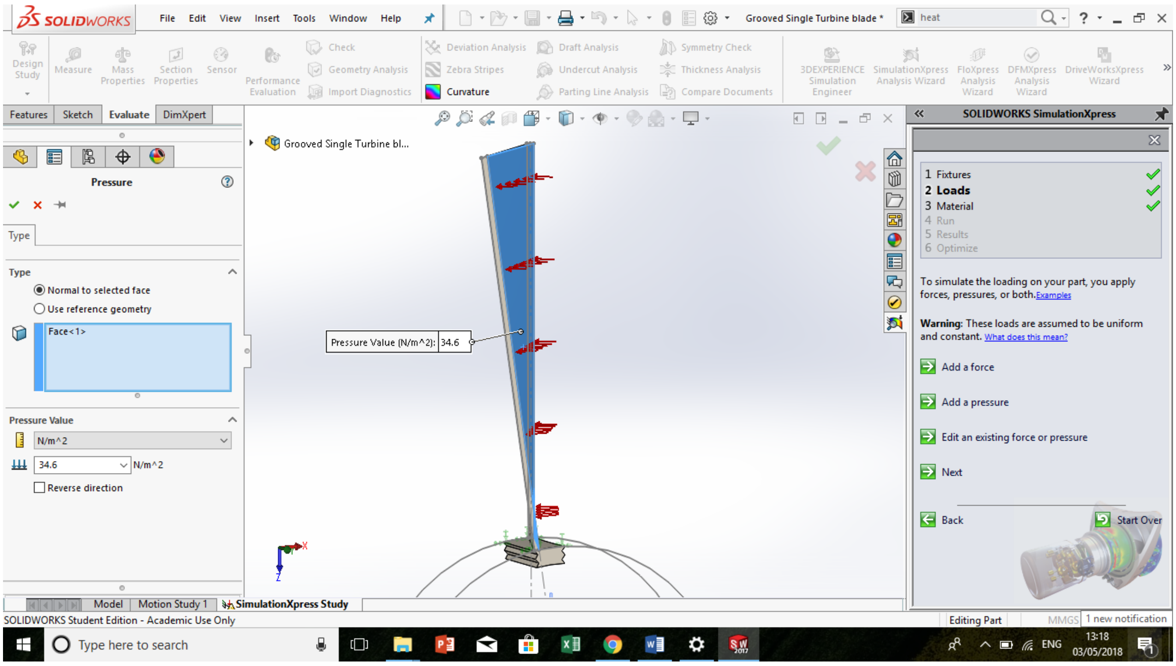

Figure 25 Applied Pressure Force

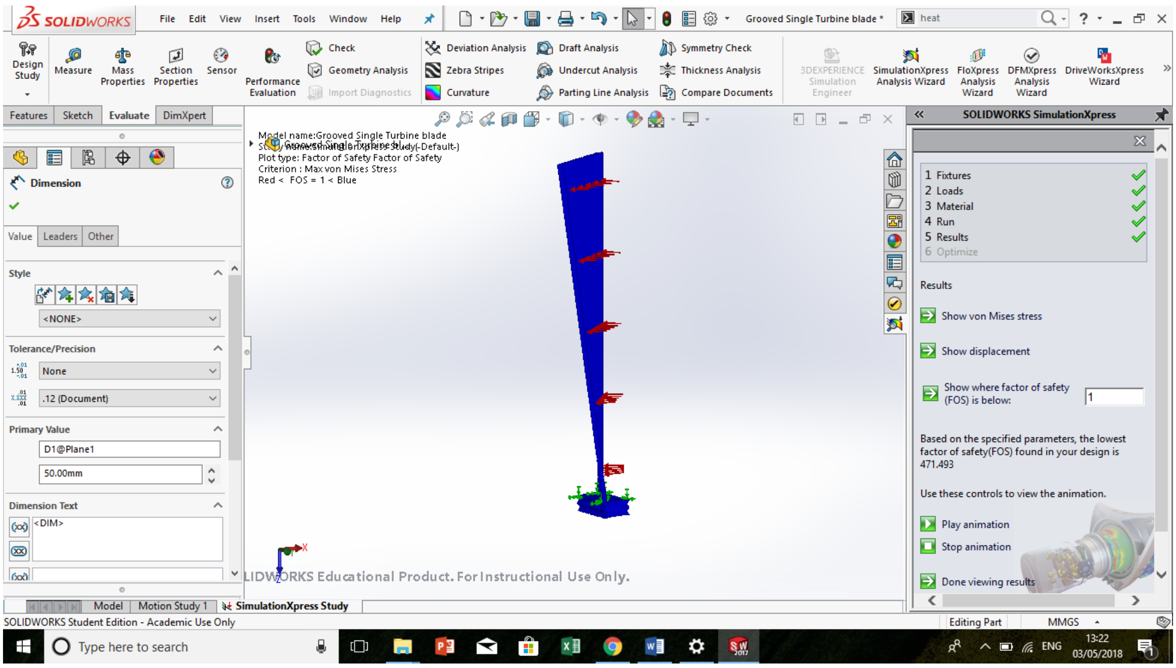

Figure 26 Safety Factor

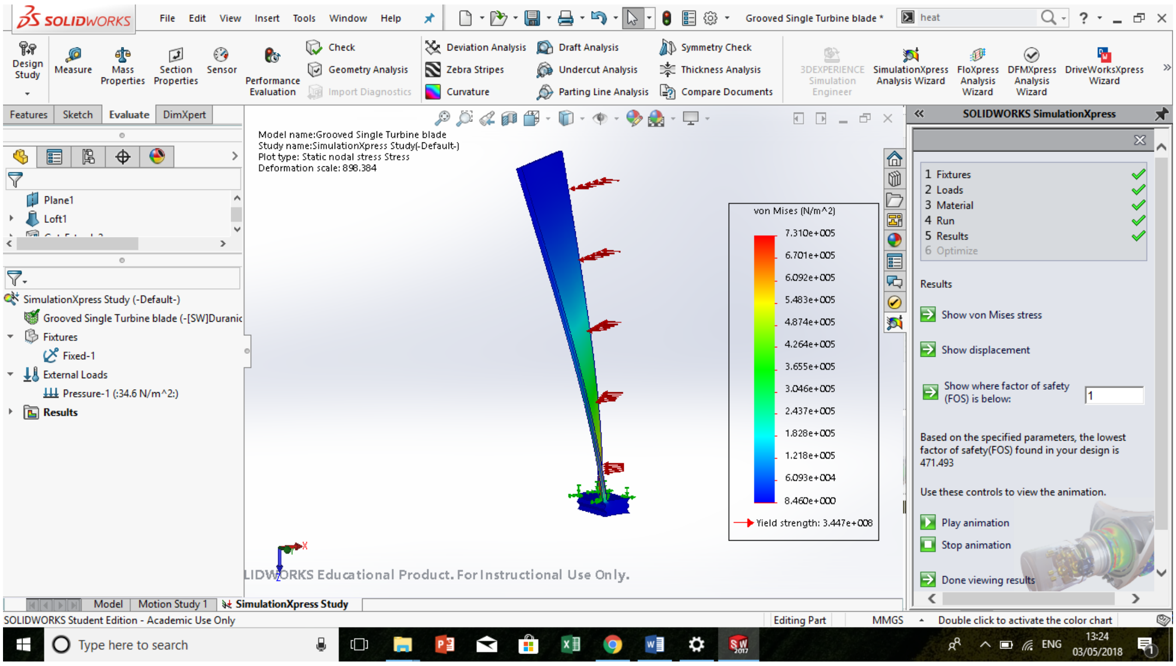

Figure 27 Von Mises (UTS)

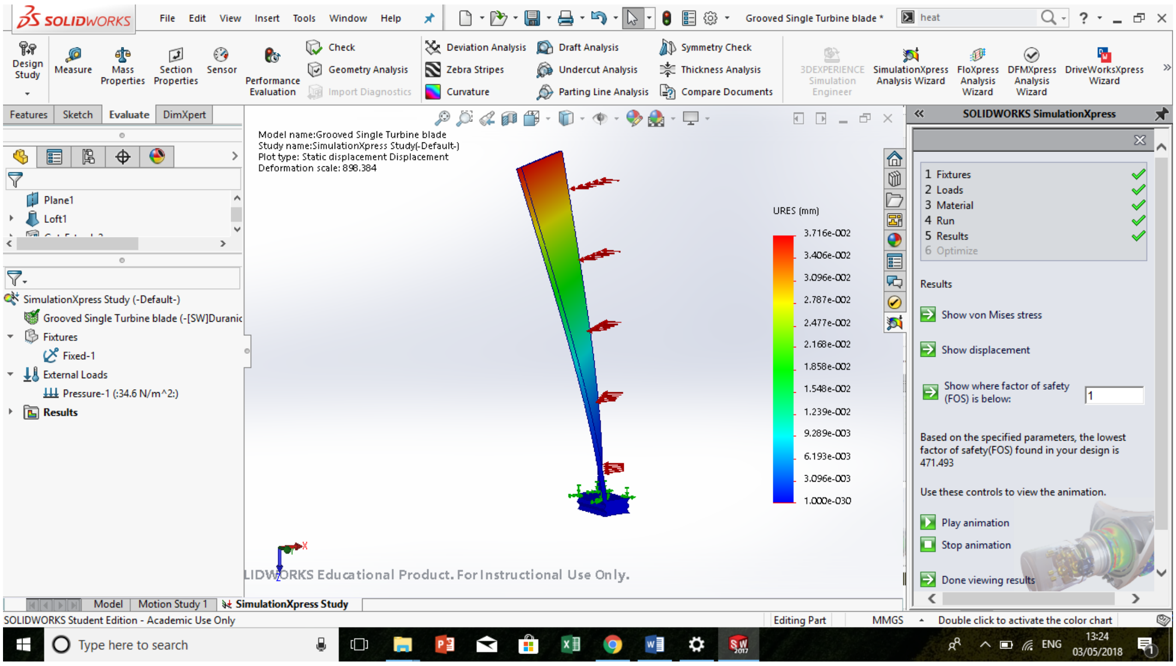

Figure 28 Displacement (Shear Force)

Factor of safety = 471.493, when a steam pressure of 34.6 Mpa is applied.

Displacement Shear Force = 3.7mm

Moment of Inertia

$$I = M\ \frac{1}{12}\ (a^2 + b^2)$$

$I = 564\ x\frac{1}{12}\ x\ (50^2 + 275^2) = 22031250 = 22\ Mpa$ (Angoni, 2011)

Deflection of the Blade Calculations

$\frac{FL^3}{2EI}$ = $\frac{34.6x10^6}{2x2.11x10^{10}x22x10^6} = 3.92mm\ Discplacement\ Value, URES.$ (Dunn, 2017)

$SolidWorks\ Discplacement\ Value\ 3.7mm\ URES, difference\ due\ to\ design\ assumptions.$

Total force

$F + p$

$52.3 + 34.6 = 86.9Mpa$

Combination of rotational force and steam pressure on the turbine blade.

Free Body Diagram Shows Pressure and Forces

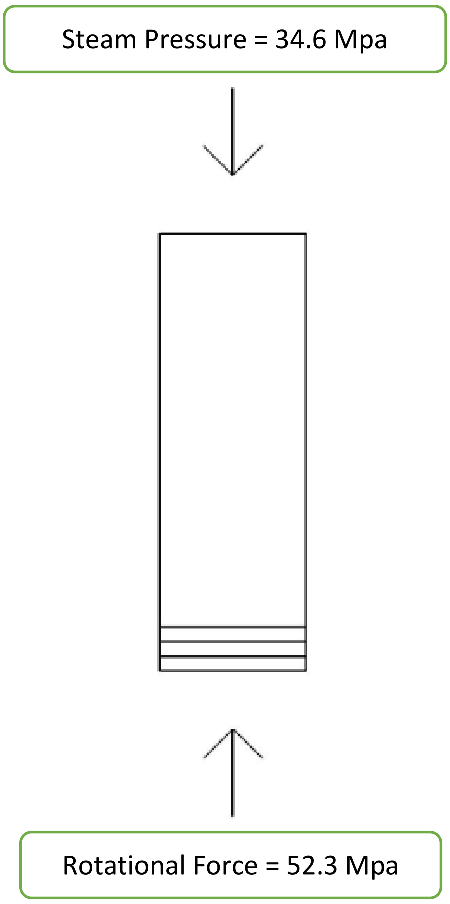

Figure 29 Free Body Force Diagram

Mpa Values According to SolidWorks

$Duranickel\ 301$

$Elastic\ Modulas = \ 210000\ Mpa$

$Shear\ Modulas = 77000\ Mpa$

$Tensile\ Strength = 620.42\ Mpa$

$Yield\ \ Strength = 344.678\ Mpa$

Strain

$Yield\ Strain\ for\ Duranickel\ = \ 0.2\%\ Engineers\ rule\ original\ length$ (Sole 2018)

$\varepsilon = L - L_O/L_O$

$\varepsilon = 275 + 0.55 - 275/275$

$\varepsilon = 274.55mm$

Elastic Modulus

$\sigma = E\varepsilon$

$\sigma = 77x10^9\ x\ 274.55x10^{-3} = 2.11x10^{10}$

$Youngs\ Elastic\ Modulus\ for\ Duranickel\ Blade$

$\sigma = 2.11x10^{10}$

Yield Strain

$w = \ 0.2\%\ 0f\ 50mm\ \ 0.1mm = 0.001m$

$= 0.1x10^{-3} = 1x10^{-4}$

$L_0 = \ 275mm \ = \ 0.2750m = \ 0.2750x10^{-4}m$

$$y = \frac{w}{L_0} = \frac{1x10^{-4}}{2750x10^{-4}} = 0.0036\ Mpa\ (Yield\ Strain)$$

$\boldsymbol{t = Gy}$

$77x10^9x0.0036x10^{-3} = 277200\ pa$

$= 277.2\ Kpa$

$= \ 0.2772\ Mpa\ (Ultimate\ Strain)$

Stress/Strain

$$\frac{stress}{strain} = \frac{213392640}{2772x10^{-3}} = 76981471.86$$

$$= 76.981\ Mpa\ (Shear\ Modulus)$$

$$= 77\ Gpa$$

Table 9 Stress Over Strain Results

Properties	SolidWorks Value	Calculated Value	Units
Elastic Modulus	$2.1x10^{11}$ Gpa	$2.11x10^{10}$ Gpa	pa = 1 N/m^2
Shear Modulus	77 Gpa	76.981 Gpa	pa = 1 N/m^2

Conclusion

Results for stress/strain are almost identical to values given by SolidWorks, however, there are variations as calculations incorporate the Blade and rounding of numbers.

Stress vs Strain Curve SolidWorks

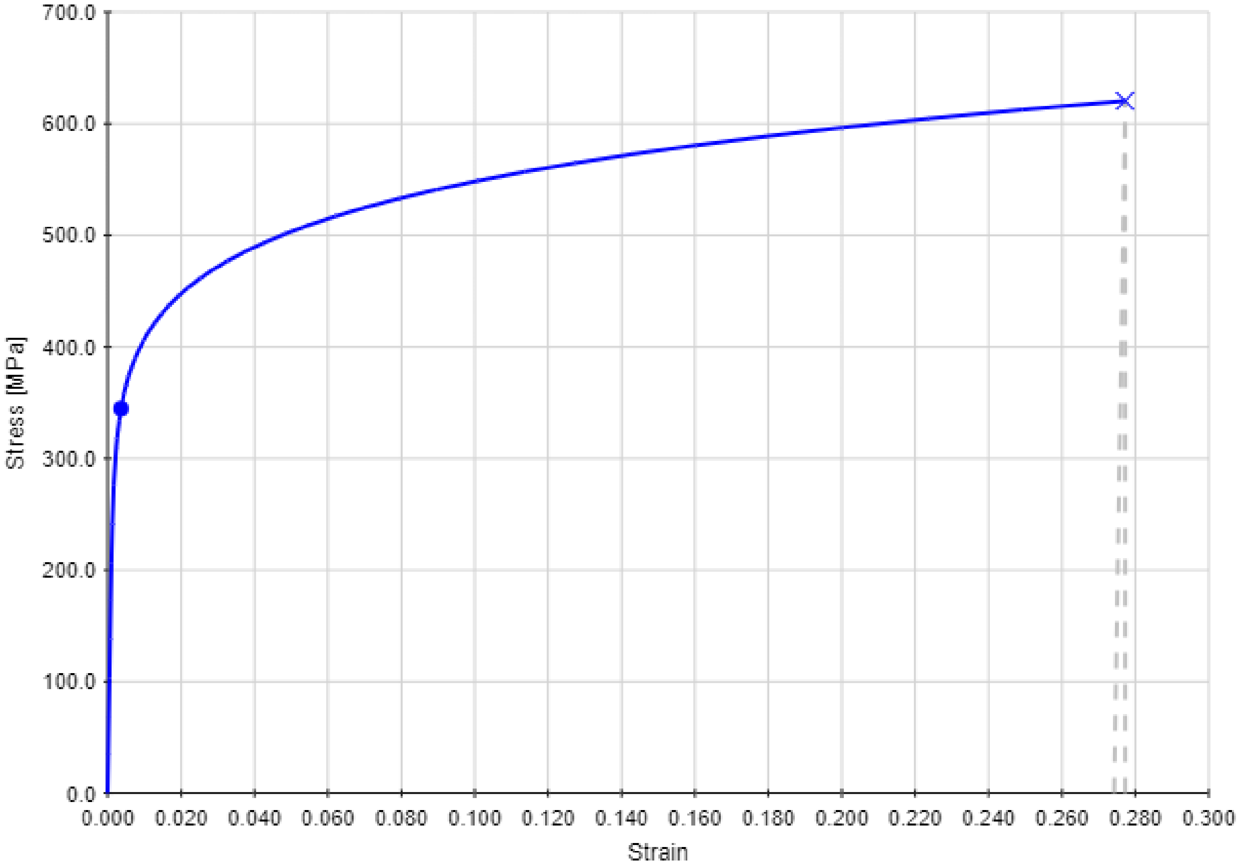

Figure 30 Stress vs Strain Curve

Stress/strain curve does not need to Show 0.2% proof stress linear line as the yield point is easily found at 344.68 Mpa.

Table 10 Yield and Ultimate Point

Yield Point	
Yield Strength:	344.68 Mpa
Yield Strain:	0.003634 Mpa

Ultimate Point	
Tensile Strength:	620.42 Mpa
Tensile Strain:	0.2772 Mpa

Using online calculator to prove yield strain and ultimate strain calculations are correct, the yielding point is 344.68Mpa and the ultimate strength is at 620.42Mpa, it elongates 27.426% before rupture occurs 0.2772Mpa, this is within parameters of Duranickel which is rated between 15-35% elongation before rupture.

Results slightly differ from previous shear force and bending moment calculations as we neglect the root and SolidWorks considers material selection, there may be a variance rounding numbers also. (MechaniCalc, 2018)

Shear Force

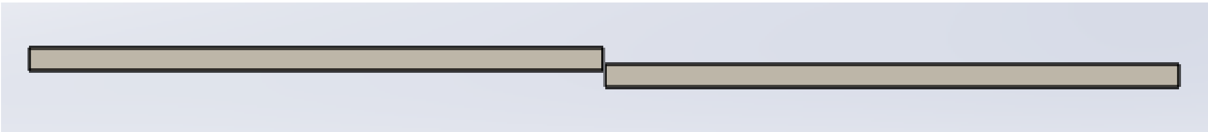

Figure 31 Shear Force Anti Plane Shear

Mode 3 (Anti Plane Shear)

Blade Mass = 564 g

σ **=** 213.4Mpa = 213.4 MN/m^2

Sum of intervals = 0.275/4 = 0.06875 m

Figure 32 Shear Force Turbine Blade

Shear Force Calculation

$213.4 + (564x0.275) = 368.5\ MN/\mathrm{m}^2$

$213.4 + (564x(0.275 - 0.20625) = 368.29\ MN/\mathrm{m}^2$

$213.4 + (564x(0.275 - 0.1375) = 368.36\ MN/\mathrm{m}^2$

$213.4 + (564x(0.275 - 0.06875) = 368.43\ MN/\mathrm{m}^2$

$213.4 + (564x(0.275 - 0.275) = 0$

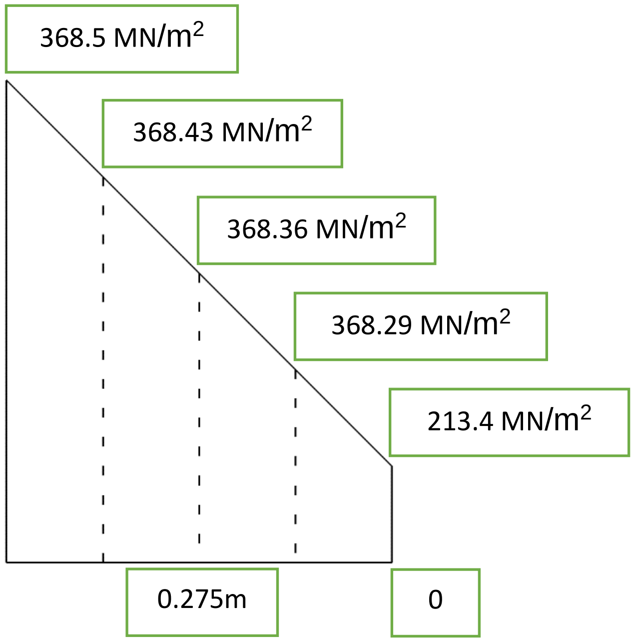

Figure 33 Shear Force Diagram

Bending Moment Calculation

$$(-213.4x0.275) - (\frac{(564\text{x}0.275)^2}{2}) = 12.086\ MN/m^2$$

$$(-213.4x(0.275 - 0.20625)) - (\frac{564x(0.275 - 0.20625)^2}{2}) = -16.\ MN/m^2$$

$$(-213.4x(0.275 - 0.1375)) - (\frac{564x(0.275 - 0.1375)^2}{2}) = -34.67\ MN/m^2$$

$$(-213.4x(0.275 - 0.06875)) - (\frac{564x(0.275 - 0.06875)^2}{2}) = -56\ MN/m^2$$

$$(-213.4x(0.275 - 0.275)) - (\frac{564x(0.275 - 0.275)^2}{2}) = 0\ MN/m^2$$

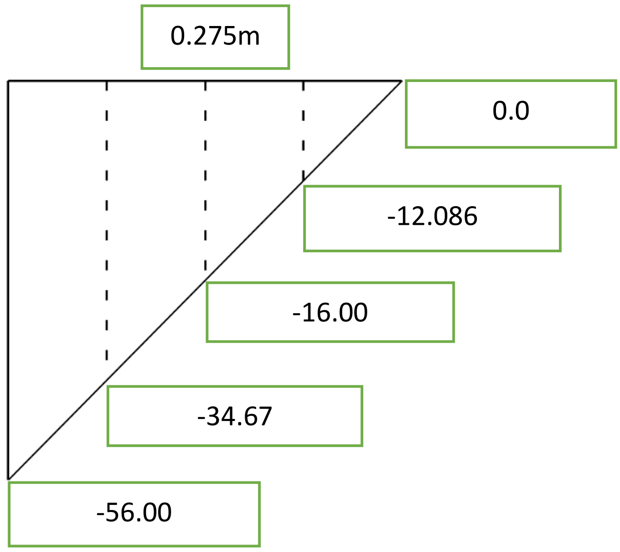

Figure 34 Bending Moment Diagram

Shear force = 368 MN/m^2/Mpa, and the Bending Moment = 56 MN/m^2/Mpa. The shear force is slightly more than the yield as less would not shear the material.

Factor of Safety

$$FS = \frac{F_{fail}}{F_{allow}}$$

$$\frac{345_{Mpa}}{52.3_{Mpa}} = 6.59 \text{ Mpa}$$

$$\sigma = \frac{\sigma_{fail}}{FS}$$

$$\sigma = \frac{213.4}{6.59} = 32.4\ Mpa\ (lower\ stress\ limit)$$

$$\sigma = \frac{\sigma_{fail}}{\sigma_{allowed}}$$

$$\sigma = \frac{213.4}{32.4} = 6.59 \text{ Mpa}$$

Duranickel can withstand forces up to 6.6 times stronger than the force applied through calculations.

Safety Factor SolidWorks

Does not comply to SF value calculated, it shows when a safety factor is below 1, which is the value of its own strength, we aim to look at strengths 2.5 times its own strength, hence 6.6 calculated is very good in terms of strength. it also states “based on specified parameters, the lowest safety factor (FOS) found in your design is 2.4106e-006. (The Royal Academy of Engineering, ND)

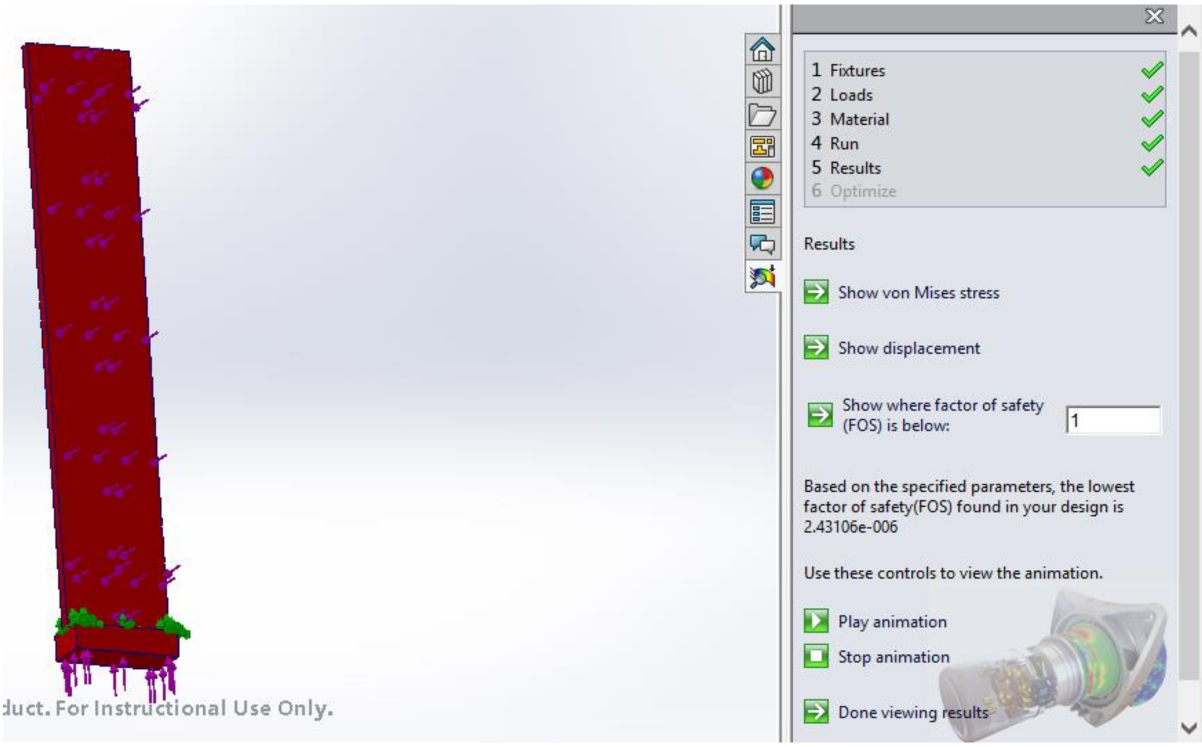

Figure 35 SolidWorks Safety Factor

Facts and Figures

Table 11 Facts and Figures

Facts	Value		
Blade Cross Section	0.245 mm		
Angular Velocity	785.4 Radians		
Blade Tip Radius	0.275 mm		
Centrifugal Force	52.3 MN/Mpa		
(σ) Minimum Stress Blade	32.4 Mpa		
(σ) Pressure Force Over Blade	34.6 Mpa		
(σ) Maximum Stress Blade	213.4 Mpa		
(ε) Strain Elastic Modulus Blade	2.11×10^{11} Gpa		
Stress/strain Blade	76.9 Gpa		
(ε) Ultimate Strain	0.2772 Mpa		
Ultimate Yield	0.0036 Mpa		
Total Mass (M)	664 g	564g Blade	100g Root
Duranickel Mass Density (rho)	8200 Kg/M^3		
Safety Factor	6.6		
Duranickel Tensile Strength	620 Mpa		
Duranickel Yield Strength	345 Mpa		

Duranickel Temperature	590 J/Kg/K
Total Surface Area	34488.28 mm^2
Blade Surface Area	30750 mm^2
Root Surface Area	3738.28 mm^2
Poisons Ratio	0.34 N/A
RT = Gas Constant	8.31 mol/k
Duranickel Thermal Expansion	1.5e - 00.5 k
Warranty	1 Year

Fatigue Cycle Test

Life Expectation (**No stress)**

Turbine blade run time = 1 year

UTS = Endurance limit = 620.420000 Mpa

125 RPM x 60 = 7500 RPH

7500 x 24 = 180000 RPD

180000 x 365 = 65.700000 RPY = N^f (number of cycles)

Warranty = UTS/N^f

$$\frac{620420000}{65700000} = 9.44\ years$$

The Blade has a life estimation of 9.44 years (10^7). Life Cycle per year is 65.7 million Cycles (10^6), with no faults or stresses. High Cycle Testing is required which is above 10 thousand cycles, the endurance limit is the Ultimate tensile strength = 620.420000 N^f (10^7). Testing two cycles, first cycle at maximum stress to the lowest stress, second cycle shows maximum stress to the highest allowable stress.

Stress Range Values

Table 12 Stress Range Values

(f) Max	213.4 MN
(σ) Min	32.4 MN

$$Stress\ Range\ \Delta\sigma = \sigma_{max} - \sigma_{min}$$

$$Stress\ Amplitude\ \ \sigma_a = \frac{1}{2}(\sigma_{max} - \sigma_{min})$$

$$Mean\ Stress\ \ \sigma_m = \frac{1}{2}(\sigma_{max} + \sigma_{min})$$

$$Stress\ Ratio = \frac{\sigma_{min}}{\sigma_{max}}$$

Test 1 Results

Table 13 Test 1 Results

$\Delta\sigma$	181 Mpa
$(\sigma)_a$	90.5 Mpa
$(\sigma)_m$	122.9 Mpa
R	0.15 Mpa

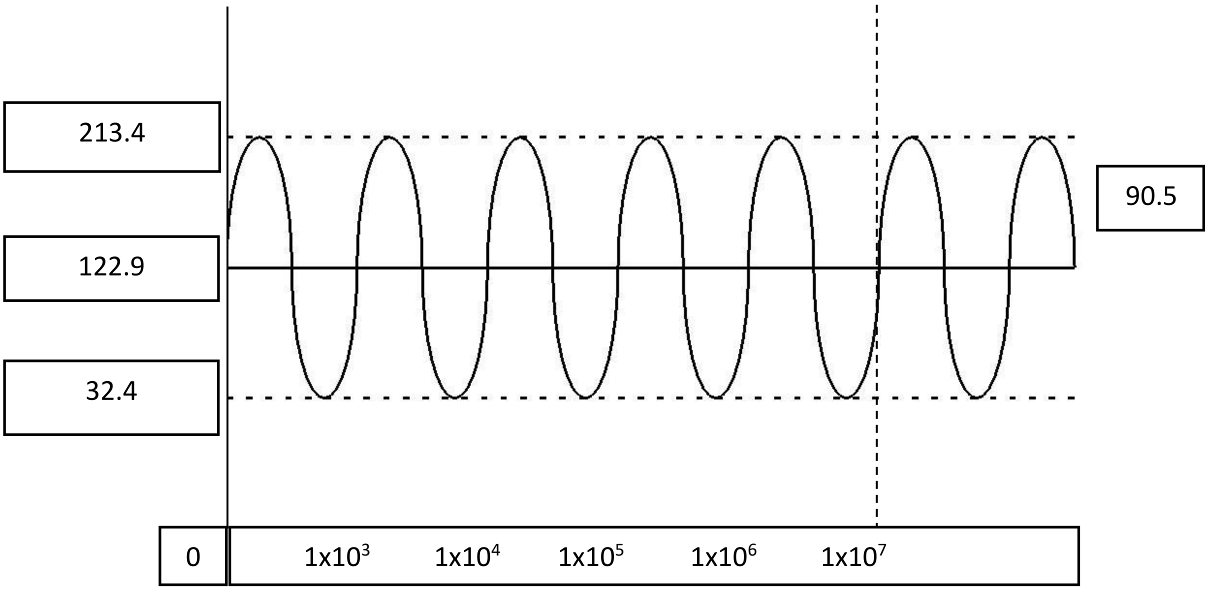

Figure 36 Fatigue Cycle Chart 1

Stress Range Values

Table 14 Stress Range Values

(σ) Max	$213.4\ Mpa$
(σ) Min allowed	52.3Mpa

Test 2 Results

Table 15 Test 2 Results

$\Delta\sigma$	161.1 Mpa
$(\sigma)_a$	80.55 Mpa
$(\sigma)_m$	132.85 Mpa
R	0.24 Mpa

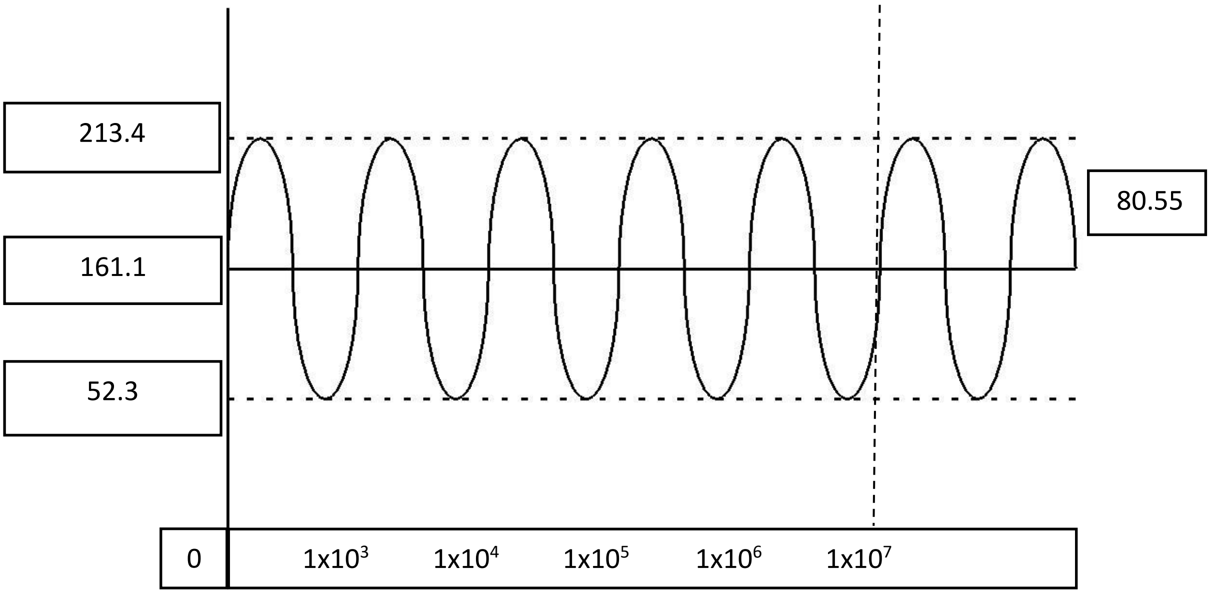

Figure 37 Fatigue Cycle Chart 2

Test Results 1

$\Delta\sigma\ x\ N_f^{0.1}$ (Basquins Law) (Lu 2018)

213 - 32.4 = 181 x 620.42$^{0.1}$ = 344.3 Mpa = 5.2 yr.

Test Results 2

213.4 – 52.3 = 161.1 x 620.42$^{0.1}$ = 303 Mpa = 4.6 yr

Conclusion

Stress Range Chart helps visualise the fatigue of a component under given restraints, over a period, two tests were carried out, highest allowable stress and lowest stress. The average of those values was used to determine fatigue which starts at 32.4 Mpa, from 0 to 32.4Mpa the blade would last 9.44 yr, operating at higher stress levels reduces life expectancy as proven.

Amplitude affects machinery as high amplitudes give rise to vibration and noise, keeping the amplitude short is the best method, using safety limits from stress and strain results set the parameters. If we introduce a crack or hole, life expectancy will shorten further.

For accuracy, results from tests will be taken and a hole will be added to the part.

Fracture Toughness

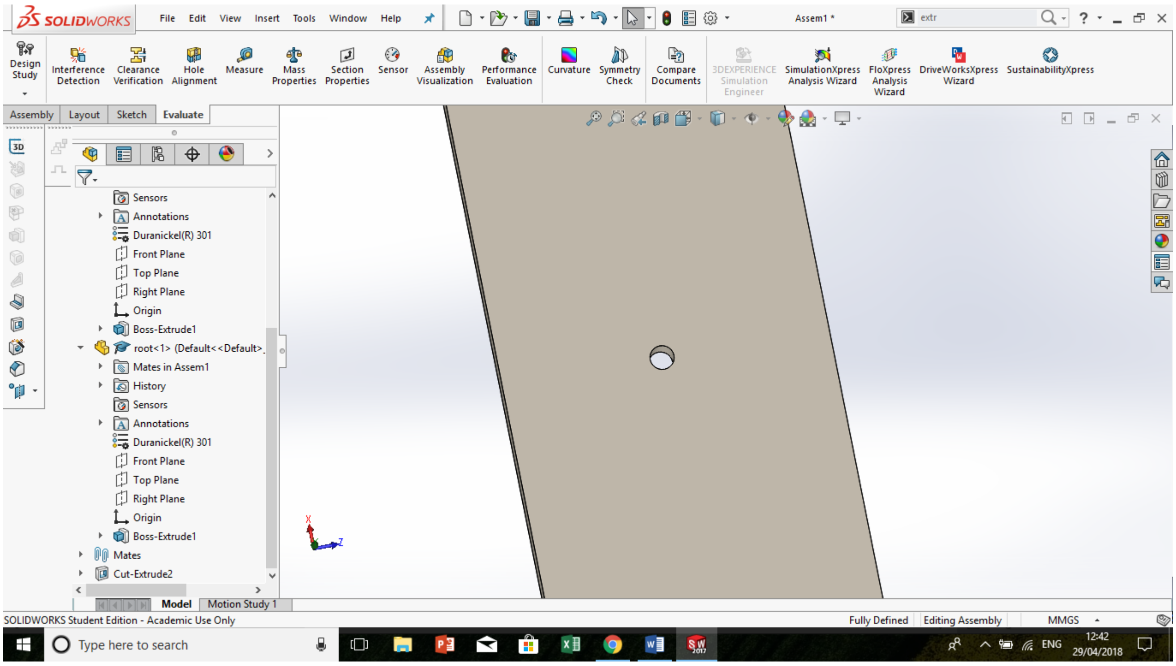

Figure 38 Fracture Toughness

Adding a hole central under stress will decrease strength and life cycle, using maximum stress values 213.4, 52.3 and 32.4 Mpa, and a hole of 5mm diameter central to the blade. My aim is to allow for a maximum hole size that still maintains a warranty of a year.

213.4 Mpa

$$k = \sigma\sqrt{\pi a}$$

$$213.4\sqrt{\pi \frac{5x10^{-3}}{2}} = 18.9\ Mpa$$

$$k = k_c \ and \ k = \sqrt{G_c E}$$

$$G_c = \frac{(k_c)^2}{E} = \frac{(18.9x10^6)^2}{2.11x10^{10}} = 16929.38\ Mpa/m$$

$$= 16.9\ Gpa$$

52.3 Mpa

$k = \sigma\sqrt{\pi a}$

$52.3\sqrt{\pi \frac{5x10^{-3}}{2}} = 4.6\ Mpa$

$k = k_c\ and\ k = \sqrt{G_c E}$

$$G_c = \frac{(k_c)^2}{E} = \frac{(4.6x10^6)^2}{2.11x10^{10}} = 1002.8\ Mpa/m$$

$= 1\ Gpa$

32.4 Mpa

$k = \sigma\sqrt{\pi a}$

$32.4\sqrt{\pi \frac{5x10^{-3}}{2}} = 2.87\ Mpa$

$k = k_c\ and\ k = \sqrt{G_c E}$

$G_c = \frac{(k_c)^2}{E} = \frac{(2.87x10^6)^2}{2.11x10^{10}} = 390\ Mpa/m$

$= 0.390\ Gpa$

(Loughborough University, 2010)

Results

Using UTS as Endurance Limit

$$213.4\ Mpa = \frac{620420000}{16929.38x10^6} = 0.036\ yrs$$

$$52.3\ Mpa = \frac{620420000}{1002.8x10^6} = 0.61\ yrs$$

$$32.4\ Mpa = \frac{620420000}{390x10^6} = 1.5yrs$$

Conclusion

With a hole central, 5mm deep and a diameter of 5mm, calculations show that at 32.4 Mpa the blades life expectancy increases as it is under less stress. Reality is the stress is between 32.4 and 52.3 Mpa, life expectancy can be easily calculated.

$$Average\ stress\ = 52.3 - 32.4 = 19.9/2$$

$$\sigma_a = 9.95$$

$$= 32.4 + 9.95$$

$$= 42.35\ Mpa$$

Life Expectancy

$$1.5 - 0.61 = \frac{0.89}{2} = 0.445 + 0.61 = 1\ year\ (Target)$$

Blade with defects would last a year with a hole central at an average stress of 42.35 Mpa, without stress or defects the blade would have a life expectancy of 9.44 yrs.

Creep Calculation

Table 16 Creep Facts

$Average\ Stress\ (\sigma_1) = 42.35\ Mpa$
$Lower\ Stress\ (\sigma_0) = 32.4\ Mpa$
$Temperature\ Material = 590\ J/Kg/K$
$Temperature\ Turbine = 480\ J/Kg/K$
$Blade\ Length = 275\ mm$
$Blade\ Casing = 1\ mm\ Gap\ between\ case\ and\ blades$
$n = stress\ exponent\ for\ metal$
$R = Gas\ Constant = 8.314\ k/mol$
$T = 273K$
$Activation\ Creep = Q_c = 240\ kJ/mol(half\ of\ 480℃)$ (Lu 2018)
$\varepsilon = strain$
$\dot{\varepsilon_{ss}} = Steady\ State\ Strain\ rate$
$\frac{1}{s} =\ 1\ pascal\ to\ one\ second$

$Exponent\ n =$ **5**

$For\ creep\ controlled\ by\ high\ temperature\ dislocation\ climb\ (lattice$ diffusion$)$

(Deepak, 2016)

Creep Calculation

$$\dot{\varepsilon_{ss}} = \dot{\varepsilon_0}(\frac{\sigma}{\sigma_0})^n\ exp\ exp\ (-\frac{Q_c}{RT})$$

$For\ \sigma = 42.35\ Mpa$

$$\dot{\varepsilon_{ss1}} = \dot{\varepsilon_0}(\frac{\sigma_1}{\sigma_0})^n(\left(-\frac{Q_c}{RT_1}\right))$$

$$10^6\ x\ (\frac{42.35x10^6}{32.4x10^6})^5 = 3815417.104\ pa$$

$$Exp\ (-\frac{240x10^3}{8.314x(480+273)}) = e^{38.335}\ = 4.45$$

$$\frac{\sigma_1}{exp} = \frac{3815417.104}{4.45} = 857397.102\ \frac{1}{s}$$

$1mm = Blade\ Tip\ Clearance$

$275\ mm = Blade\ Length$

Ductility

$$\varepsilon = \frac{\Delta L}{L_O} = \frac{1}{275} = 0.0036$$

$$t = \frac{\varepsilon}{\dot{\varepsilon_{ss}}} = \frac{0.0036}{857397.102} = \frac{\frac{\frac{4.2x10^{-9}(Sec)}{60\ (Min)}}{60\ (Hour)}}{24\ (Day)} = \frac{4.859669641(Days)}{365(1\ Year)}$$

$= 1.33\ yrs.(Creep\ Proof\ Target, 1\ year\ at\ an\ average\ stress\ range)$ (Lu 2018)

Using Formula

$$\dot{\varepsilon_{ss}} = \dot{\varepsilon_0}(\frac{\sigma}{\sigma_0})^n\ exp\ exp\ (-\frac{Q_c}{RT})$$

$Increase\ blade\ life\ by\ running\ at\ half\ speed\ =\ half\ stress$

$\sigma_2 = 21.175 = (Half\ the\ Mean\ Stress)$

$T_2 = 480℃$

$$\frac{\dot{\varepsilon_{ss2}} = \dot{\varepsilon_0}\ (\frac{\sigma_2}{\sigma_0})^n((-\frac{Q_C}{RT_2}))}{\dot{\varepsilon_{ss1}} = \dot{\varepsilon_0}\ (\frac{\sigma_1}{\sigma_0})^n((-\frac{Q_C}{RT_1}))} = (\frac{\sigma_2}{\sigma_1})^n = (\frac{21.175}{42.35})^5 = 0.03125$$

$\boldsymbol{Monkman - Grant\ Law}$

$$tf.\dot{\varepsilon_{SS}} = constant$$

$$\frac{t_{f2}}{t_{f1}} = \frac{\dot{\varepsilon_{ss}1}}{\dot{\varepsilon_{ss}2}}$$

$$\frac{\dot{\varepsilon_{ss1}}}{\dot{\varepsilon_{ss2}}} = (\frac{42.35x10^6}{21.175x10^6})^5 = 32$$

$Creep\ life\ extended\ 32 - 1\ (Monkman\ Grant\ Law - 1) = 31$

$Life\ extended\ by\ a\ factor\ of\ 31$ *(*Lu *2018*

$\boldsymbol{Using\ Formula}$

$$\dot{\varepsilon_{ss}} = \dot{\varepsilon}_0(\frac{\sigma}{\sigma_0})^n\ exp\ exp\ (-\frac{Q_c}{RT})$$

Max material Temperature = 590℃

$\sigma_3 = 42.35\ Average\ stress$

$$\dot{\varepsilon_{ss3}} = \dot{\varepsilon}_0(\frac{\sigma_3}{\sigma_0})^n(\left(-\frac{Q_c}{RT_3}\right))$$

$\boldsymbol{Thus}$

$$\frac{\dot{\varepsilon_{ss3}} = \dot{\varepsilon}_0\ (\frac{\sigma_3}{\sigma_0})^n(\left(-\frac{Q_c}{RT_3}\right))}{\dot{\varepsilon_{ss1}} = \dot{\varepsilon}_0\ (\frac{\sigma_1}{\sigma_0})^n(\left(-\frac{Q_c}{RT_1}\right))}$$

$$= Exp\left(-\frac{Q_c}{RT_3} + \frac{Q_c}{RT_1}\right) = Exp$$

$$= Exp\left(-\frac{240x10^3}{8.31x(590+273)} + \frac{240x10^3}{8.31x(480+273}\right) = Exp(4.886) = e^{132.4}$$

$$\frac{\sigma_1}{exp} = \frac{3815417.104}{e^{132.4}} = 1.204904478\ \frac{1}{s}$$

$$1mm = Blade\ Tip\ Clearance$$

$$275\ mm = Blade\ Length$$

$$\varepsilon = \frac{\Delta L}{L_O} = \frac{1}{275} = 0.0036$$

$$t = \frac{\varepsilon}{\dot{\varepsilon_{ss}}} = \frac{0.0036}{1.204904478} = 2.98\ seconds\ (complete\ failure)$$

Conclusion

At an average stress and maximum temperature, the material Duranickel would last just 3 seconds under heat supplied at 590℃ applied. At maximum temperature stated in SolidWorks. Metal deforms very easily especially when subjected to heat and rotational force hence creep. Duranickel can withstand the high temperature and stresses when operating in a steam turbine system at 480℃, according to values obtained through high cycle fatigue tests, cracks and creep values calculated the blade can last over a year, therefore a warranty of 1 year will be supplied to potential customers.

Flutter

Flutter in a blade is due to self-excited vibration of a blade, this happens because of the dynamic forces and structural design of a component. It is a major problem encountered by manufacturers of large industrial scale turbomachinery, for example steam and gas turbines, and aircraft engines. Flutter can reduce the life expectancy of a blade dramatically so constant maintenance is a must despite calculations.

Flutter occurs when blades vibrate over different areas of the same component at different frequencies. If the blades are identical and harmonised, then aeroelastic modes (patterns of blade vibration) have a constant phase angle between adjacent blades. Every aeroelastic mode has different inter blade phase angles. The inter phase angle affects the phase difference between local blade motion and local unsteady flow, which in turn influences the unsteady aerodynamic work done by the blades. Opposing phase angles may lead to positive work being done on the blades, this then results in flutter.

Flutter in a blade can be described as a cause for blade vibration. When investigating the frequency of vibration, if the frequency of the aeroelastic modes is close to a blade's mode frequency, and not near for example an engines order frequency, then the cause of vibration is most likely to do with flutter. In short turbomachinery flutter occurs when frequencies are very close to a blade's mode frequency because the structural forces dominate, and aerodynamic forces cannot significantly affect the vibration frequency. (RPM Turbo, 2016)

Defects

Steam Pressure and temperature on a turbine blade is detrimental to its life cycle, the blade must be able to withstand these forces for a period, ideally a year. That's why material selection is critical when designing a blade, however, other factors such as environment play a role.

Figure 39 Turbine Blade Pitting

Turbine blade cracking is known as pitting, this is caused by a leaking isolation valve, work needs to be done through calculations and changes to condenser and pipework are to be considered.

Figure 40 Turbine Blade Erosion

Turbine Disc Pitting caused by incorrect calculations the fault occurrence is due to acidic condensate.

Figure 41 Turbine Blade Fouling

Turbine Blade Fouling caused by chemical deposits on the blades and discs (Silica), induced by the boiler carryover (plant environment is responsible for these type defects.

Figure 42 Turbine Blade Erosion and Corrosion

Erosion and Corrosion on a turbine blade is due to low alloy steel components, the effects include rusting and scalloping caused by high velocity wet steam. To solve this problem changes to the water chemistry in the boiler must be undertaken if not then upgrade the blades to suit the conditions (Power Corrosion Ltd, 2017).

Coursework criteria

- Static
- Buckling
- Thermal

Unachievable in the sense that solid works at Loughborough College would not let me access criteria, although I did manage to use the basic student editions limited features as best I could, it's not that I am not capable, I was just unable and unaided. The picture below is the era message, I tried three different computers, I removed the Add-ins it asked for and still it would not let me access the features.

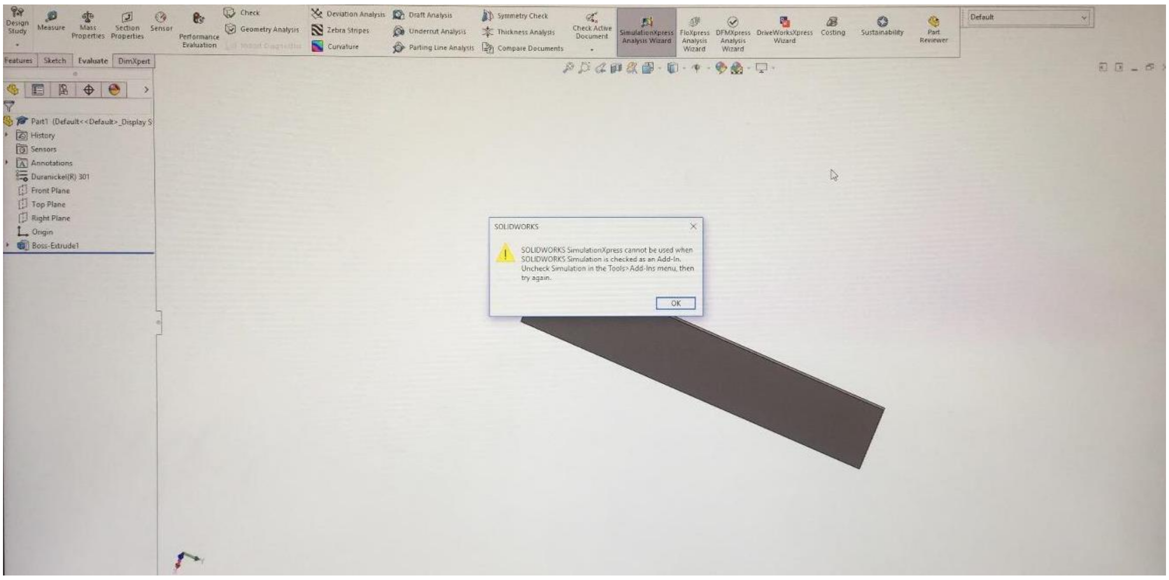

Figure 43 Denial to Access

Critical Conclusion

I chose Duranickel to begin with as research suggests it is an ideal material for a steam turbine blade, it was selected via SolidWorks Material Section as I needed a material capable of performing above 480℃ for at least a year under stressful conditions such as fatigue cracks/holes and creep tests which would allow me to supply a minimum warranty of a year to potential customers. Duranickel is known as a Nickel based Super Alloy rated at 590℃, it has high mechanical strength properties due to its alloy composition and FCC structure, after researching into properties of nickel-based alloys and calculations prove that the blade is perfect for the application of a steam turbine under constant stressful conditions over a period of a year. No optimisation required due to its density and structural soundness and intended purpose, creep is a big issue in turbine blades, when it over-heats life expectancy will reduce dramatically. For a blade to operate in a housing of a 1mm gap, the length of time it takes the blades to elongate under rotational and thermal forces has been calculated, with a value of over a year obtained against restraints. The blade having set parameters which I designed following videos on YouTube and SolidWorks tutorials enabled me to design the following, 800mm diameter blade set which consists of a 275mm blade and 250mm rotor disc, which is set to operate at 85 bar of steam pressure, with a temperature of 480℃, in a hostile environment, whilst under rotational forces rated at 125 cycles a second

Blades Positioned

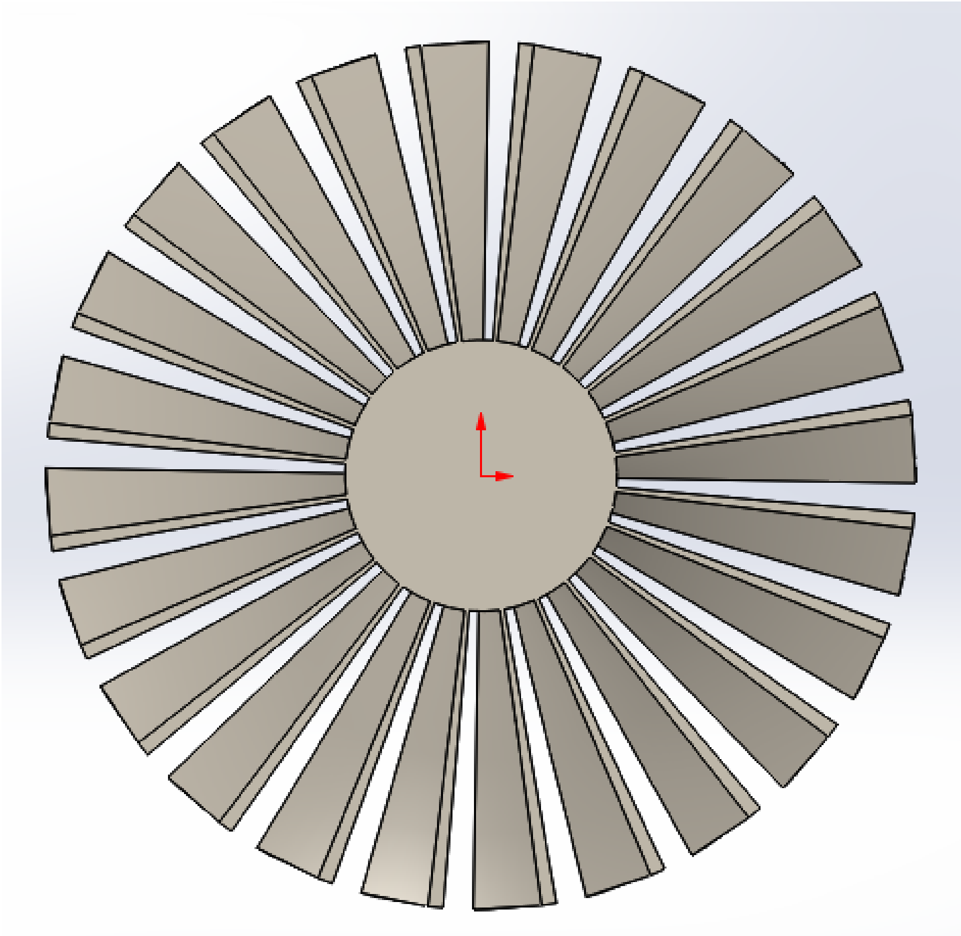

Figure 44 Blades Positioned

Research

(Siemens, 2013)

(WorldStainless.Org, N.D.)

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Appendix