

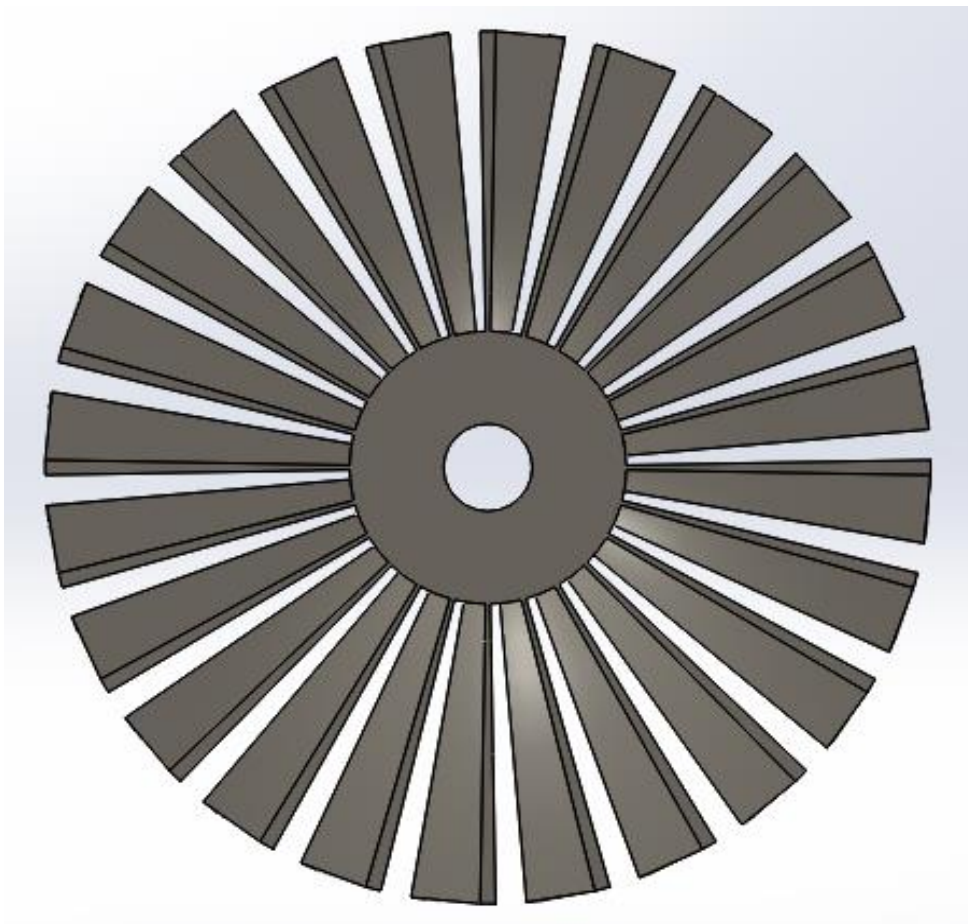
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BSc

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 stresses and strains, fatigue and creep and life cycle, this information is then used to design the Turbine in SolidWorks. Then by using FEA analysis theoretical hand calculations will either be proved correct or incorrect, any differences between the two concepts will be analysed and justified.

Blade designed in SolidWorks

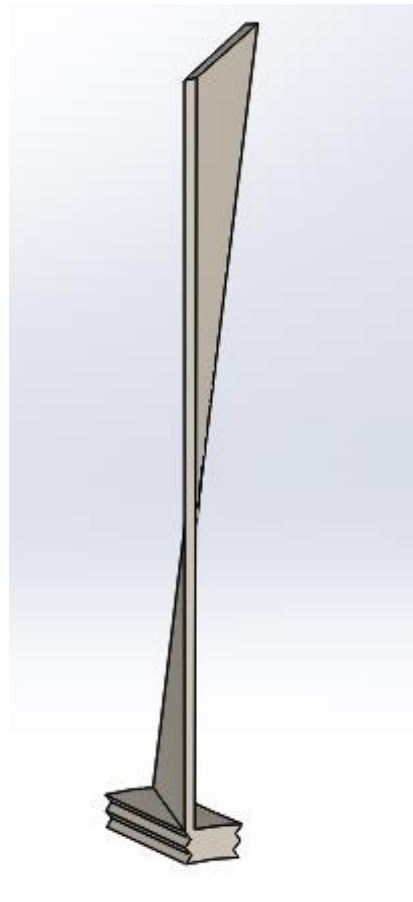


Figure 1 Blade Design SolidWorks

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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 dictates 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

Research

Turbine features

Hypothetically the steam turbine blade is going to be designed as part of my Thermodynamics project, Rankine Cycle Steam Turbine system, the CJR - 100 this sets parameters for analysis as speed of the turbine, power etc are given quantities and help towards the design of the Blade.

Design parameters

Using Rankine Cycle steam turbine model (CJR-100) 85 bar, 480° C it looks very similar to the Siemens SST-100 see macroscopic view (Siemens, 2013).

Steam Turbine Specification



SST-100

Figure 2 Siemens Steam Turbine

Table 2 Steam Turbine Specification

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

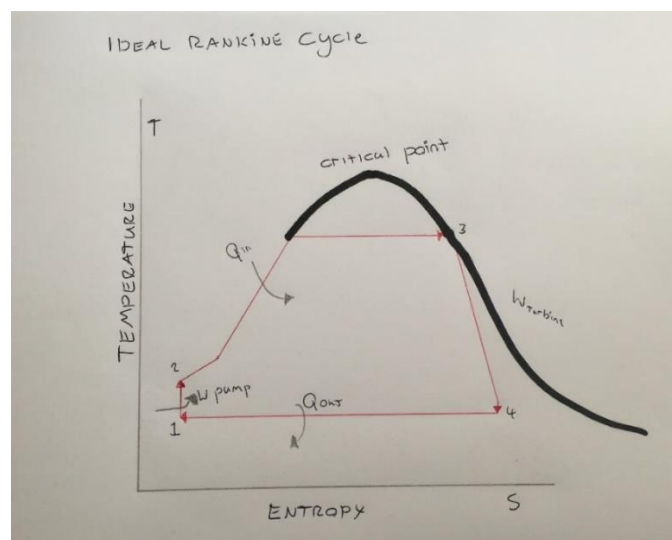


Figure 3 Ideal Rankin Cycle

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 Examples (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 (thins out reducing weight)
- Root
- 180° twist, from base to tip (no loss to strength due to curvature)

Blade Design with Root

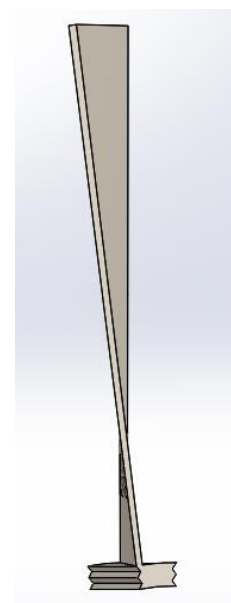
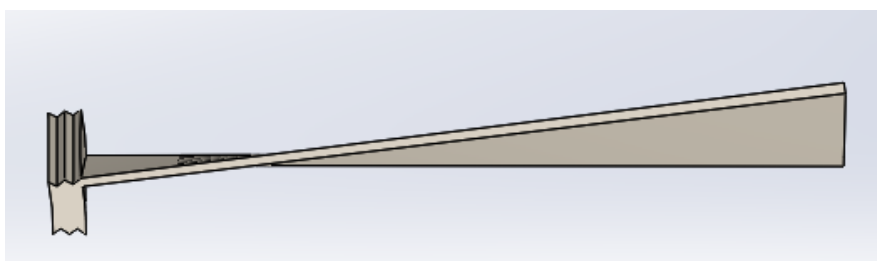


Figure 5 Blade Design with Root

Parameters

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.

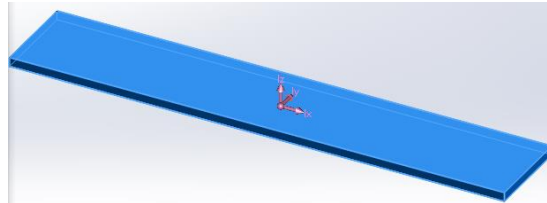


Figure 6 Rectangular Prism Shape

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

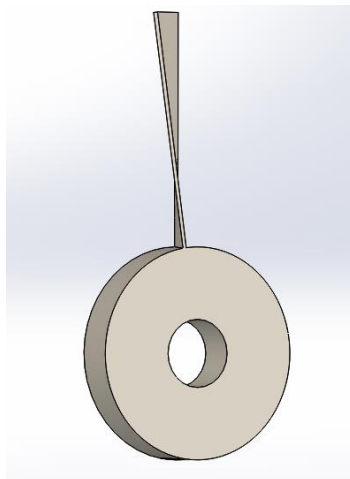


Figure 7 Blade and Rotor Disc

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 gain test results.

Before we start we 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°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

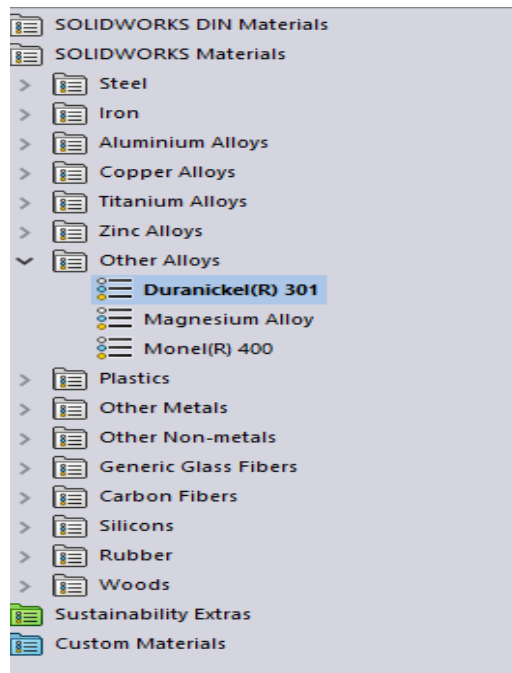


Figure 8 Material Selection

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 (at 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

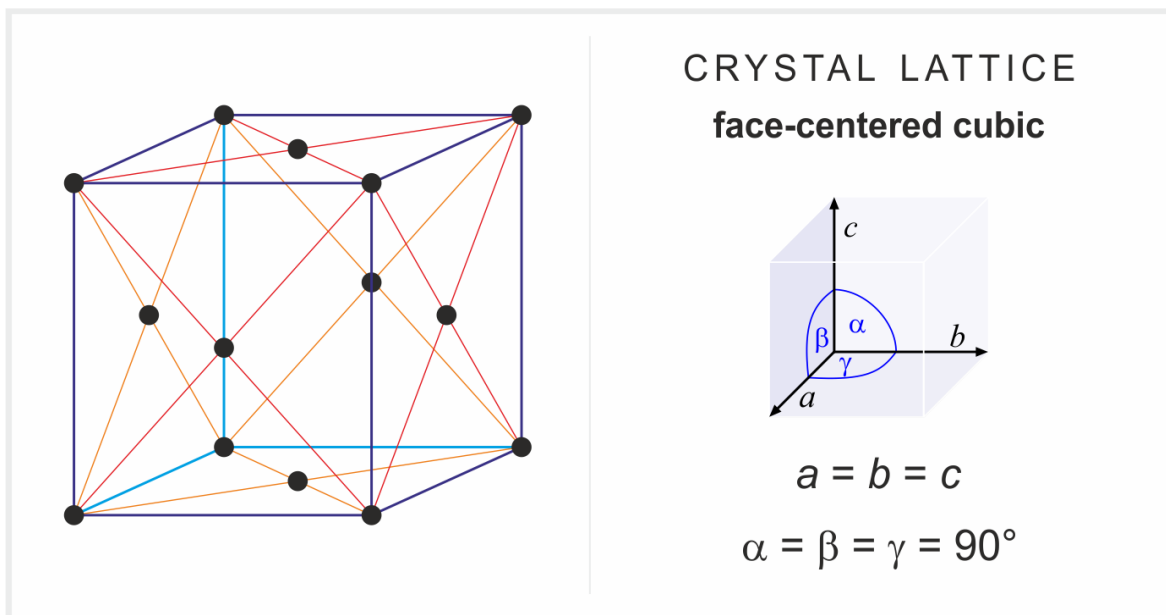


Figure 9 Nickel, Crystal Lattice Face-Centred Cubic (Chemistry Glossary, 2018)

Grain Structure of Nickel

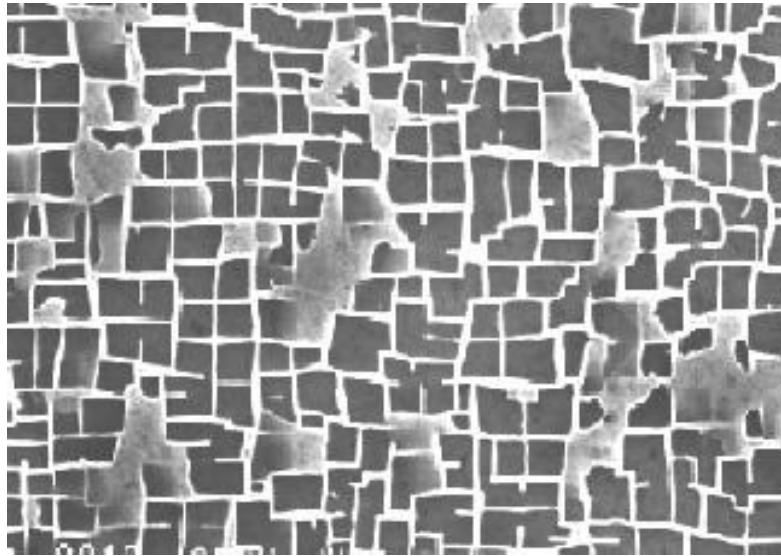


Figure 10 Grain Structure (TMS.org, 2000)

Nickel based superalloys are 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° it also 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, applied using coursework requirements. And, if this material fails or is overkill, as in it is too good to use and not cost effective.

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
Poissons 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		N/m ²

	344678000/345 Mpa	
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

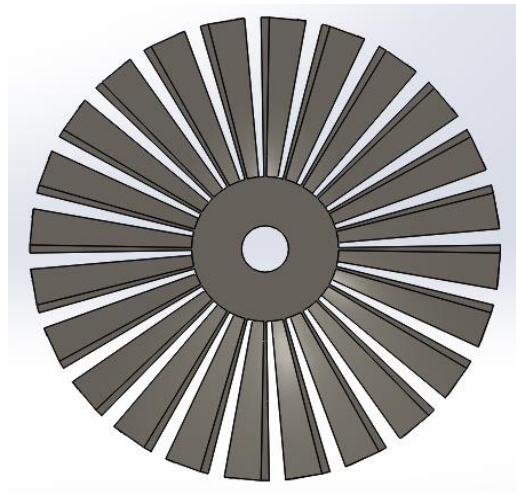


Figure 11 Turbine Blade

Single Blade Design with Rotor Disc

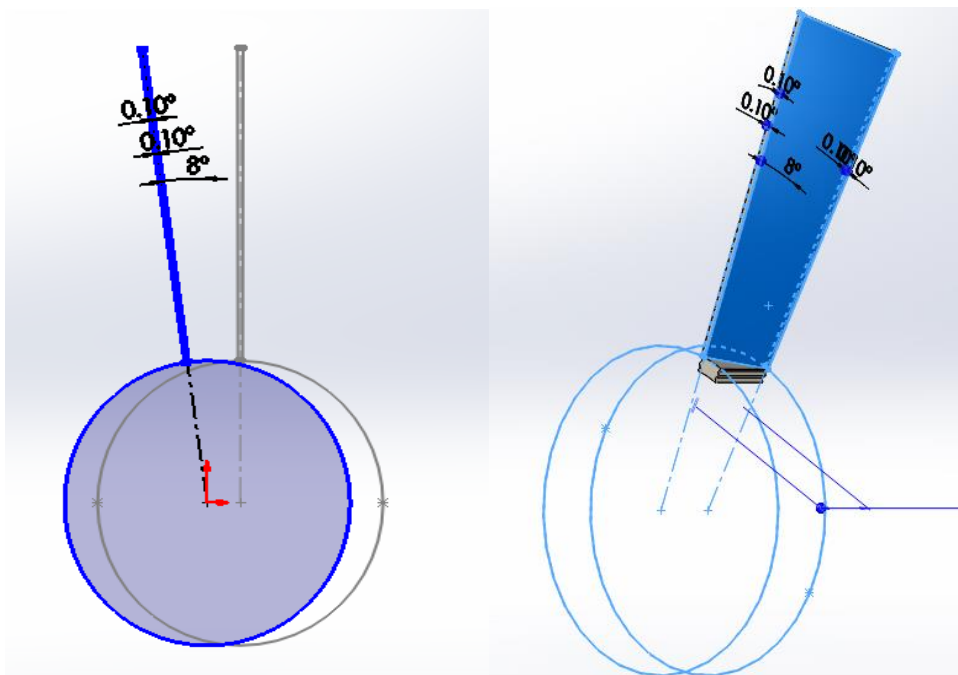


Figure 12 Single Blade Design with Rotor Disc

Blade Height = 275mm

Blade Width = 50mm

Blade Thickness = 5mm

Centre of the shaft to blade tip = 400mm

Centre of shaft to root tip = 125mm

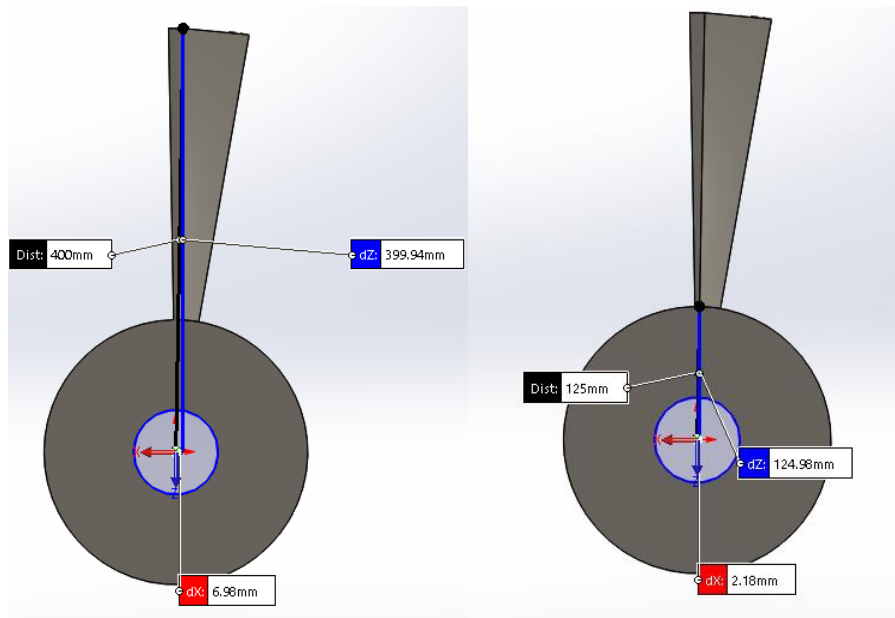


Figure 13 Single Turbine Blade

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.

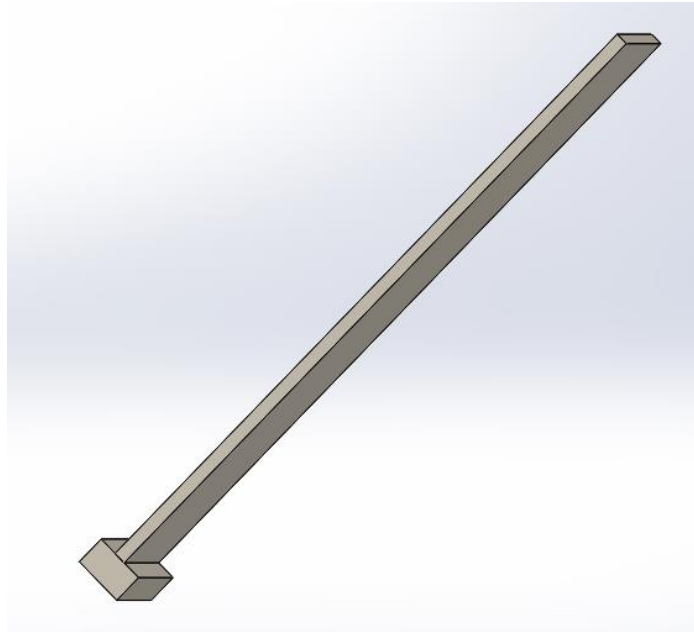


Figure 14 Blade

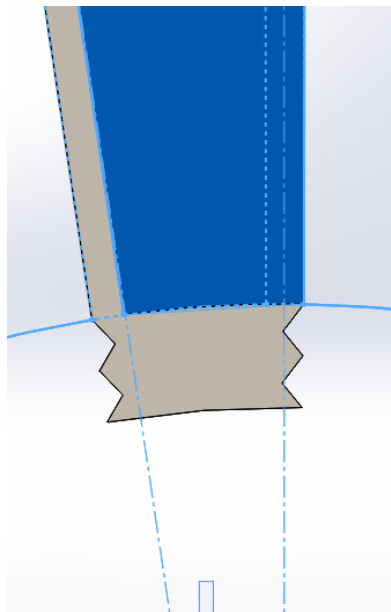


Figure 15 Blade Root

Blade Surface Area

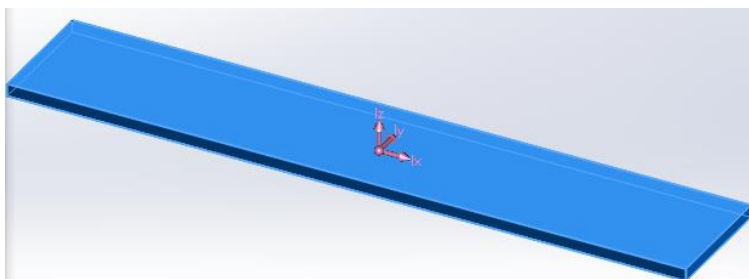


Figure 16 Blade Surface Area

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

$$5 \times 275 = 1375\text{mm}$$

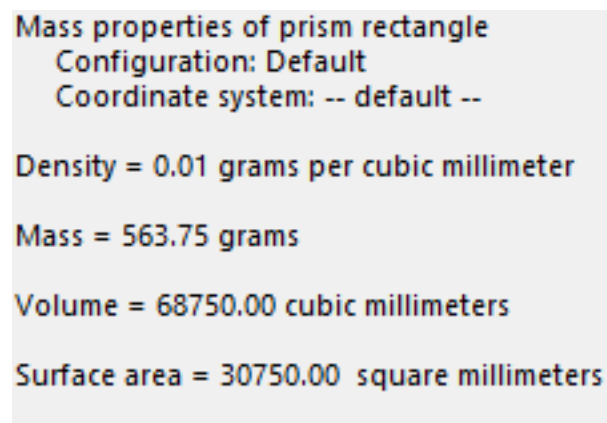
$$50 \times 5 = 250\text{mm}$$

$$50 \times 275 = 13750$$

$$13750 + 1375 + 250 = 15375\text{mm}$$

$$2 \times 15375 = 30750\text{mm}$$
 6 faces to rectangle prism

$$\text{Surface area} = 30750\text{mm}^2$$



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 17 SolidWorks Mass and Surface Area

Root Surface Area

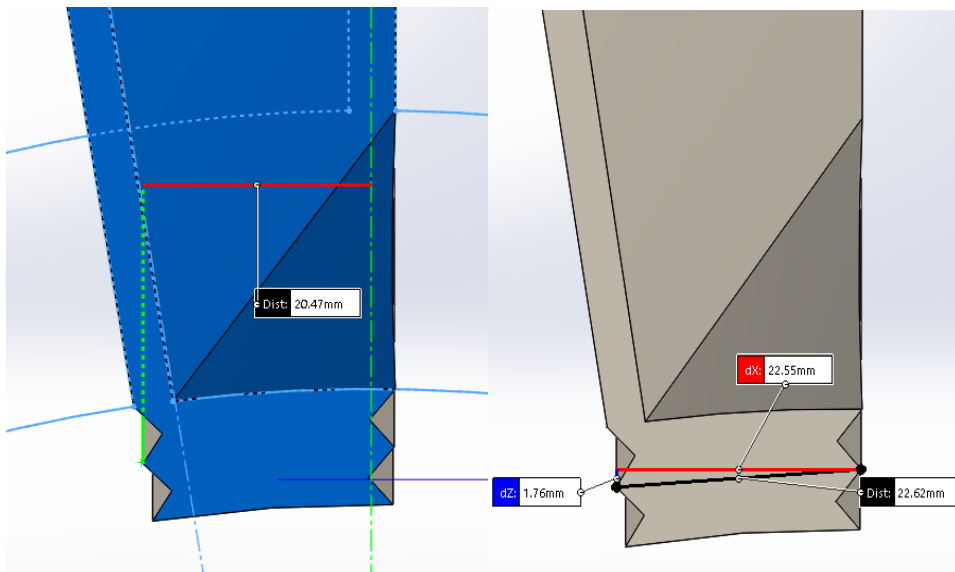


Figure 18 Root Surface Area

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

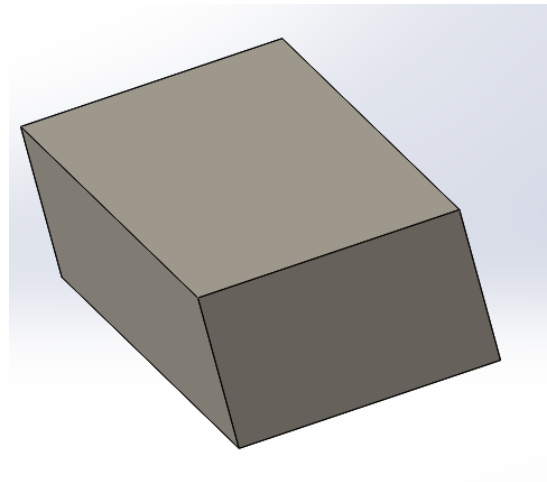


Figure 19 Rectangle Prism

Root Surface Area

Root Height = 12mm

Root Width = 20.47mm

Root Length = 50mm

$$50 \times 12 = 600mm$$

$$20.47 \times 50 = 1023.5mm$$

$$20.47 \times 12 = 245.64mm$$

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

$$2 \times 1869.14 = 3738.28mm \text{ (6 faces to a rectangle prism)}$$

$$\text{Surface area} = 3738.28mm^2$$

Mass = 100.71 grams

Volume = 12282.00 cubic millimeters

Surface area = 3738.28 square millimeters

Figure 20 SolidWorks Mass and Surface Area

Blade Including Root Properties

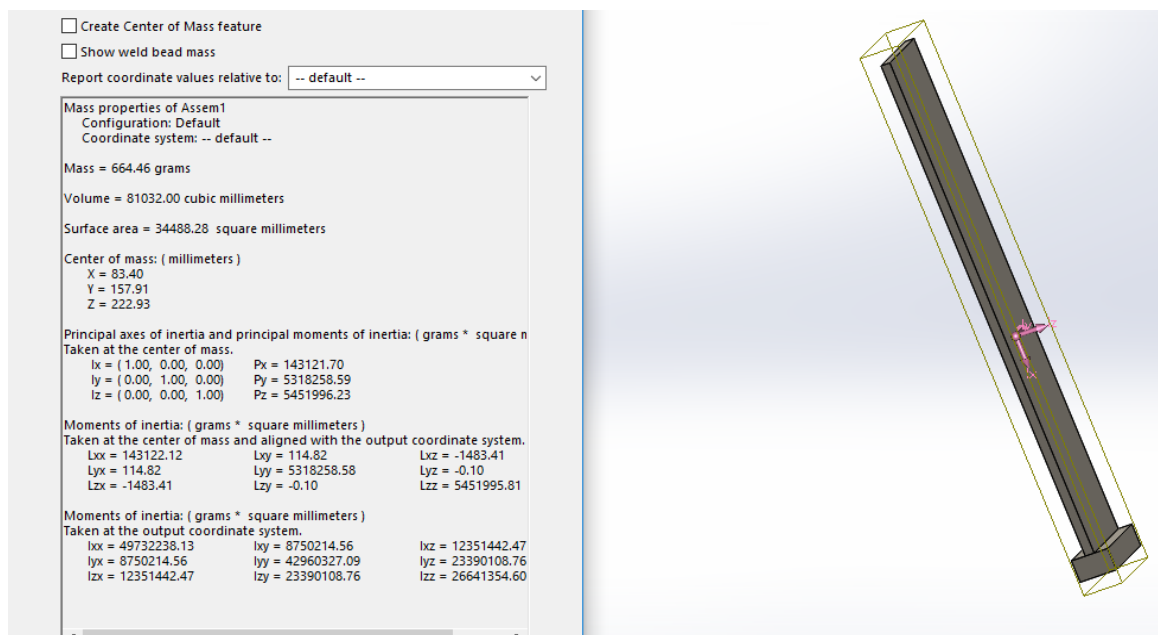


Figure 21 Blade Root Properties

SolidWorks picture

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 = 42210mm² 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.

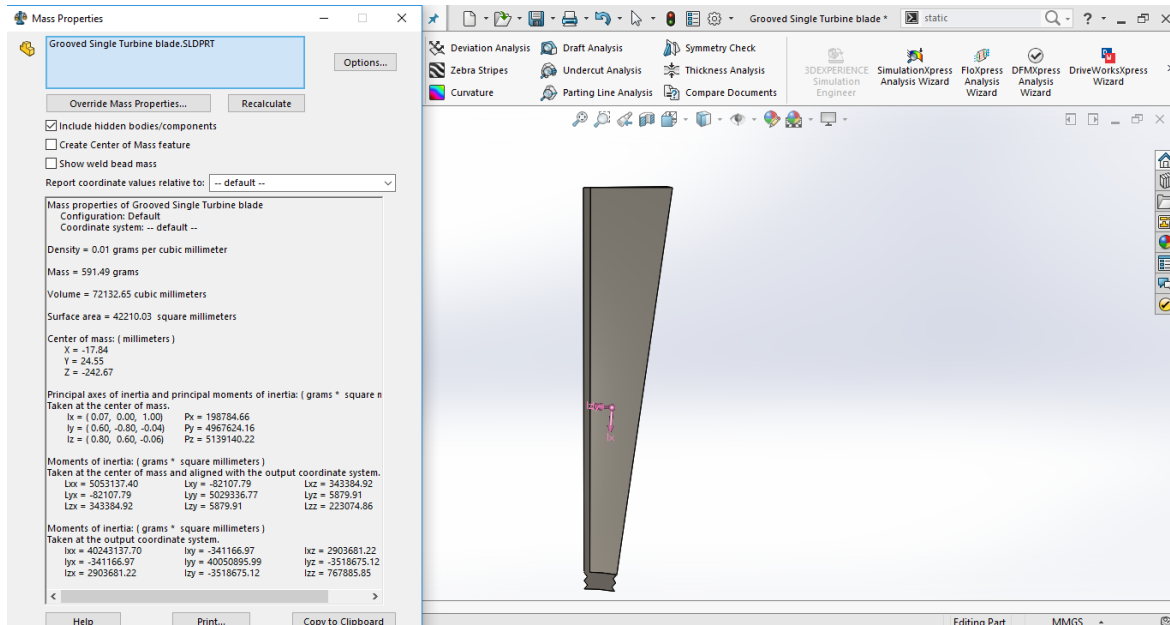


Figure 22 Actual Mass and Surface Area

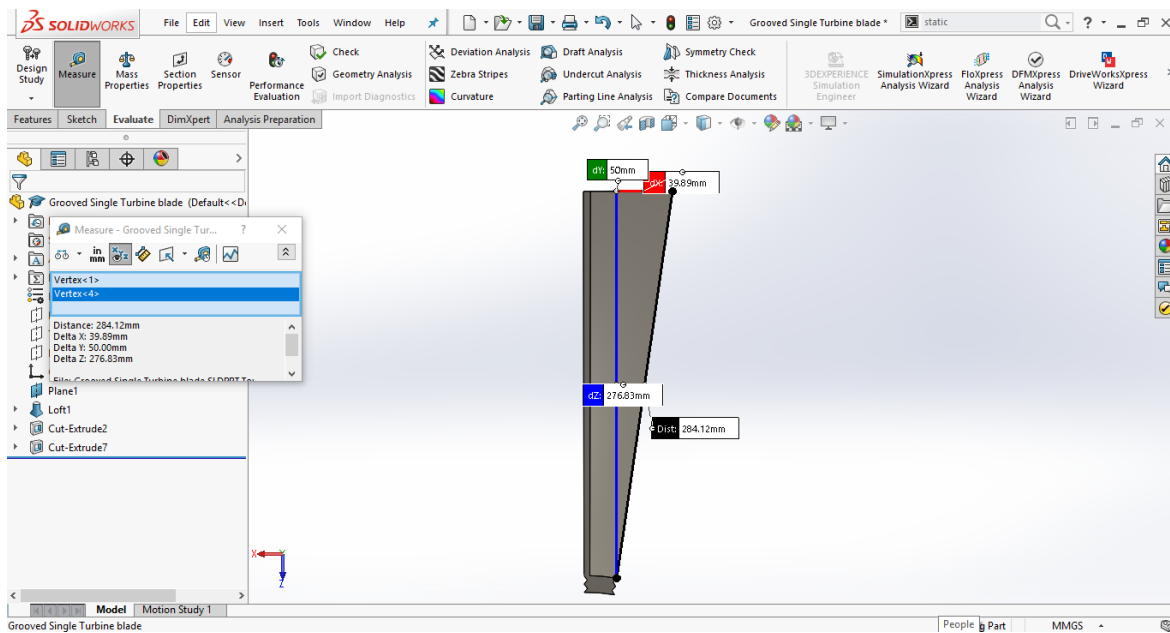


Figure 23 Angle Increasing Surface Area

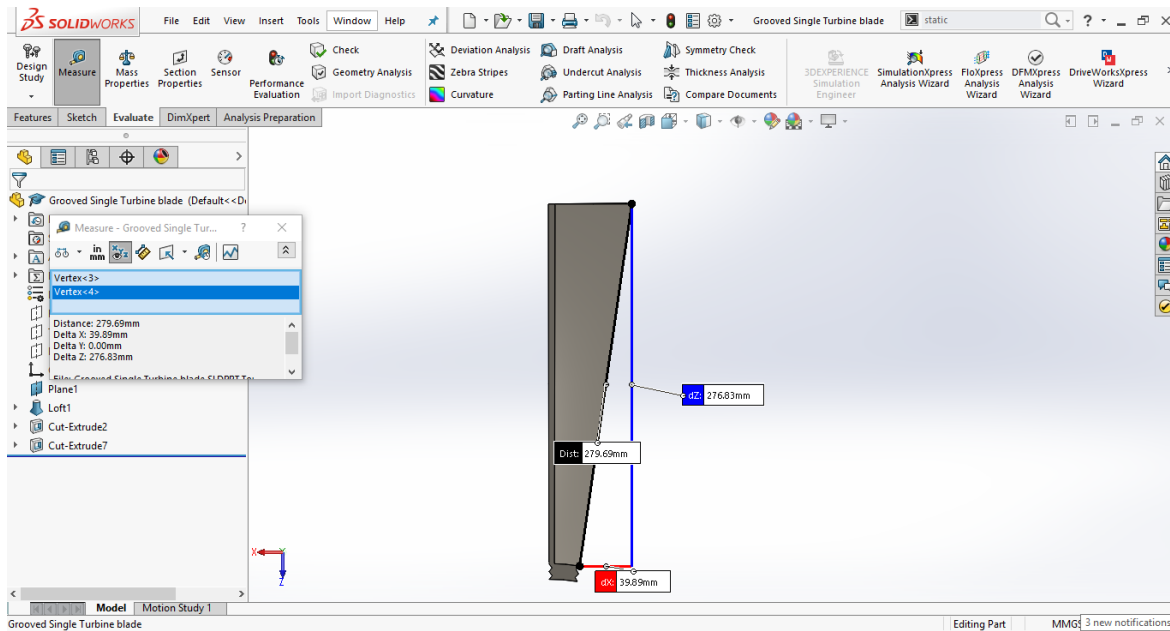


Figure 24 Angle Increasing Surface Area

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.

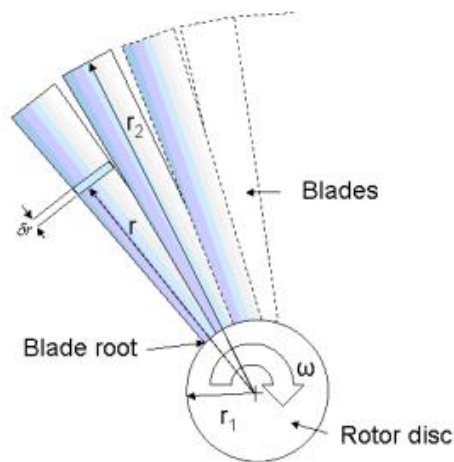


Figure 25 Centripetal Force

Blade Length

$$r_2 = 400\text{mm}$$

$$r_1 = 125\text{mm}$$

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

$$= 275\text{mm}$$

$$= 0.275\text{m}$$

Blade Root Cross Section Area

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

Angular Velocity

$$\frac{7500 \times 2\pi}{60} = 250\pi = 785.4 \text{ Radians per second}$$

$$= 125 \text{ RPS}$$

Blade Tip Radius

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

$$= 0.125m$$

$$0.4 - 0.125 = 0.275m$$

Force

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

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

$$= 52281.2kN$$

$$= 52.3 \text{ Mpa}$$

(The Royal Academy of Engineering, ND)

Conclusion

Table 8 Results

Tensile strength	620420000	N/m ²
Yield strength	344678000	N/m ²

Duranickel 301, is capable of absorbing 52281.2kN/m² of rotational forces, given results in table for tensile and yield strengths.

Stress

$$1 \text{ Pascal} = N / m^2 = kg/(m*s^2)$$

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

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

$$\sigma = 213392640 \text{ N/m}^2$$

$$\sigma = 213.4 \text{ Mpa}$$

Steam Pressure Force on Blade

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

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

$$p = \frac{8.5 \times 10^6}{0.245} = 34.6 \text{ Mpa}$$

SolidWorks Deflection Pressure over blade

Steam pressure causes deflection across a blade, mainly at the tip.

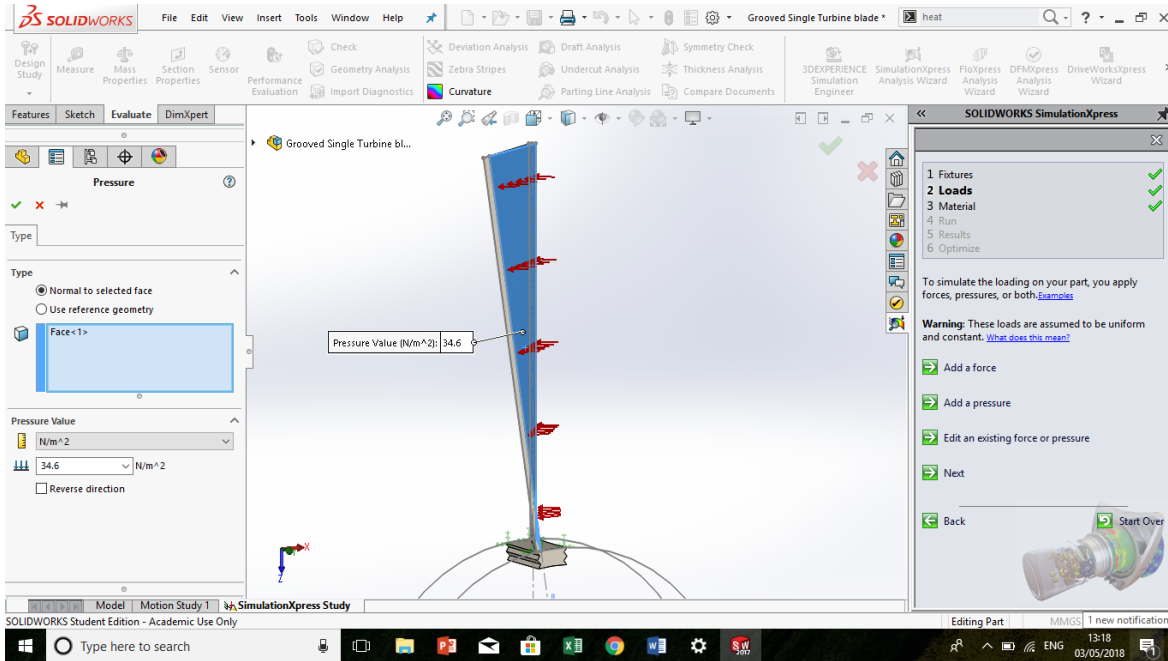


Figure 26 Apply Pressure Force

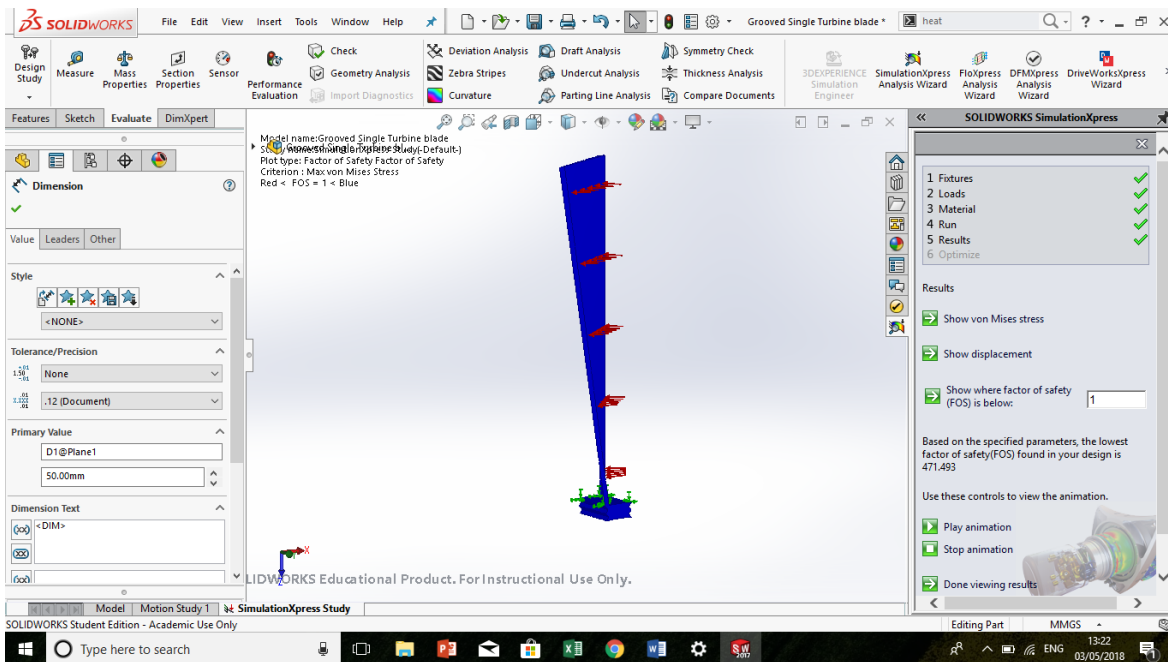


Figure 27 Safety Factor

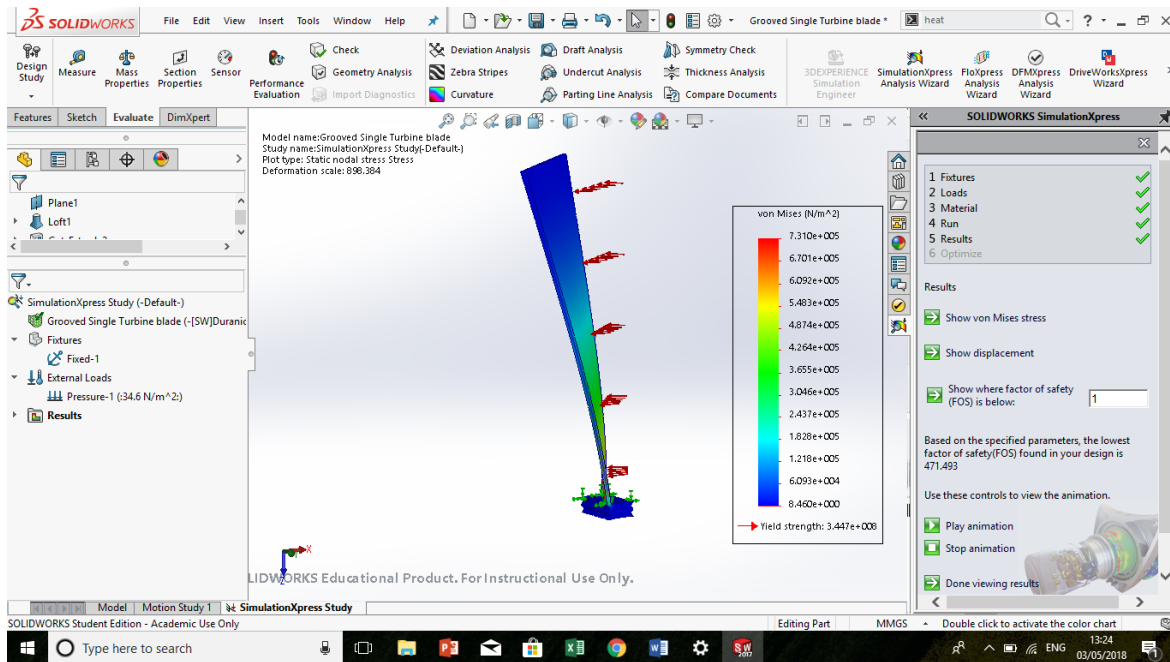


Figure 28 Von Mises = UTS

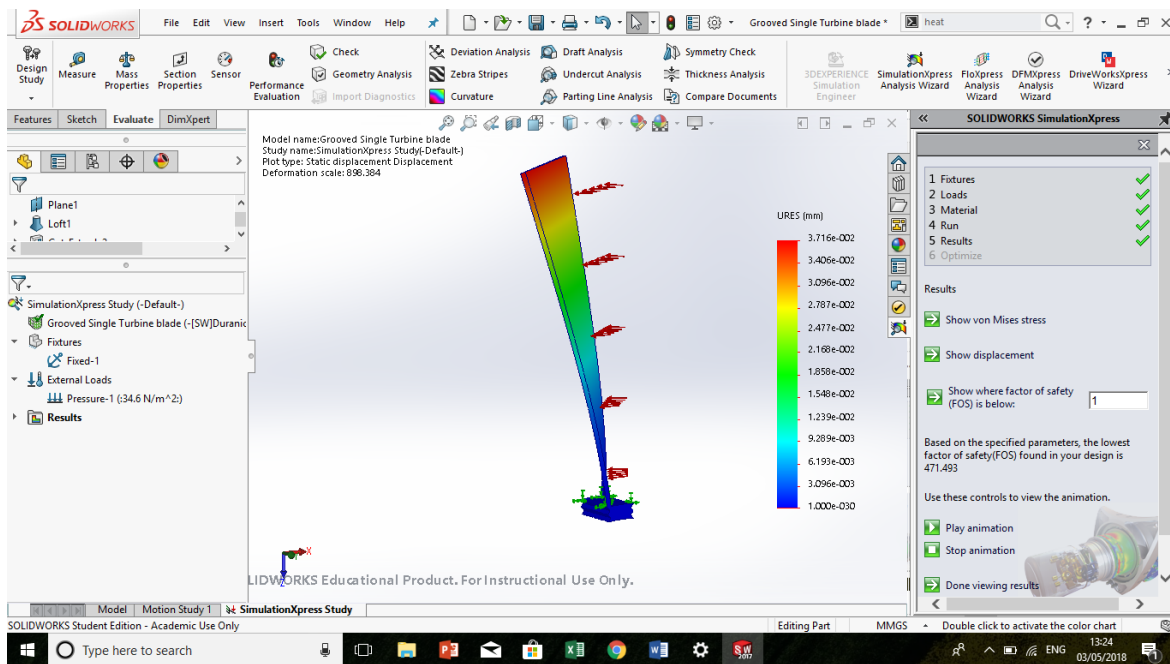


Figure 29 Displacement = Shear Force

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

Moment of Inertia

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

$$I = 564 \times \frac{1}{12} \times (50^2 + 275^2) = 22031250 = 22 \text{ Mpa}$$

(Angoni, 2011)

Deflection of the Blade Calculations

$$\frac{FL^3}{2EI} = \frac{34.6 \times 10^6}{2 \times 2.11 \times 10^{10} \times 22 \times 10^6} = 3.92 \text{ mm Displacement Value, URES. (Dunn, 2017)}$$

SolidWorks Displacement Value = 3.7 mm URES, difference due to design assumptions

Total force

$$F + p$$

$$52.3 + 34.6 = 86.9 \text{ Mpa}$$

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

Free Body Diagram Shows Pressure and Forces

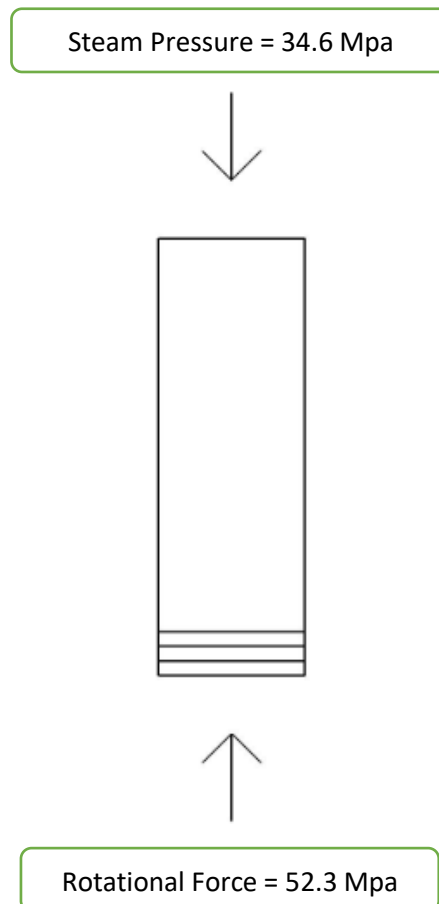


Figure 30 Free Body Force Diagram

Mpa Values According to SolidWorks

Duranickel 301

Elastic Modulus = 210000 Mpa

Shear Modulus = 77000 Mpa

Tensile Strength = 620.42 Mpa

Yield Strength = 344.678 Mpa

Values Calculated Incorporating the Blade

Strain

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

$$\varepsilon = L - L_0/L_0$$

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

$$\varepsilon = 274.55\text{mm}$$

Elastic Modulus

$$\sigma = E\varepsilon$$

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

Youngs Elastic Modulus for Duranickel Blade

$$\sigma = 2.11 \times 10^{10}$$

Yield Strain

$$w = 0.2\% \text{ of } 50\text{mm } 0.1\text{mm} = 0.001\text{m}$$

$$= 0.1 \times 10^{-3} = 1 \times 10^{-4}$$

$$L_0 = 275\text{mm} = 0.275\text{m} = 0.275 \times 10^{-4}\text{m}$$

$$y = \frac{w}{L_0} = \frac{1 \times 10^{-4}}{275 \times 10^{-4}} = 0.0036 \text{ Mpa (Yield Strain)}$$

$$t = Gy$$

$$77 \times 10^9 \times 0.0036 \times 10^{-3} = 277200 \text{ pa}$$

$$= 277.2 \text{ Kpa}$$

$$= 0.2772 \text{ Mpa (Ultimate Strain)}$$

Stress/Strain

$$\frac{\text{stress}}{\text{strain}} = \frac{213392640}{2772 \times 10^{-3}} = 76981471.86$$

$$= 76.981 \text{ Mpa (Shear Modulus)}$$

$$= 77 \text{ Gpa}$$

Table 9 Stress Over Strain Results

Properties	SolidWorks Value	Calculated Value	Units
Elastic Modulus	2.1×10^{11} Gpa	2.11×10^{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 Strain Curve SolidWorks

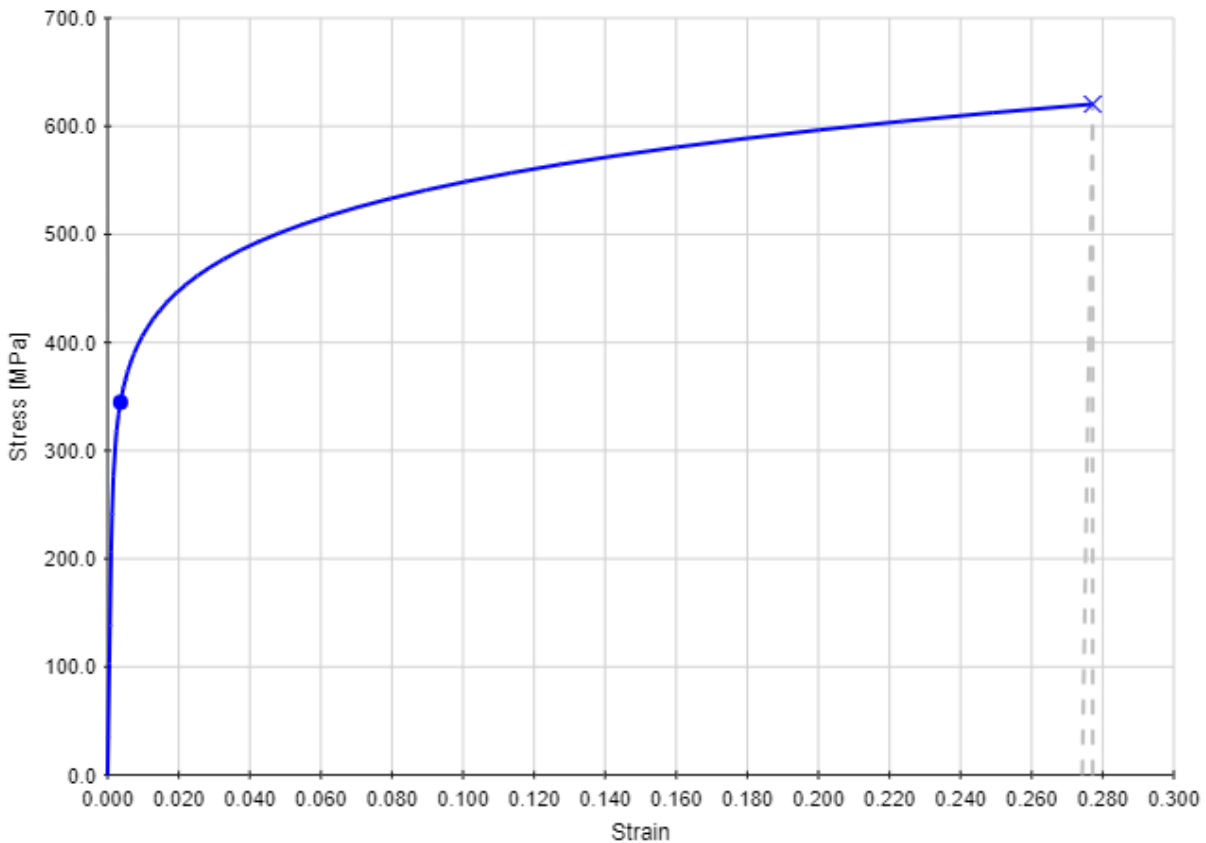


Figure 31 Stress Strain Curve (MechaniCalc, 2018)

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 Point/ Ultimate Point

Yield Point		Ultimate Point	
Yield Strength:	344.68 Mpa	Tensile Strength:	620.42 Mpa
Yield Strain:	0.003634 Mpa	Tensile Strain:	0.2772 Mpa

(MechaniCalc, 2018)

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.

Shear Force

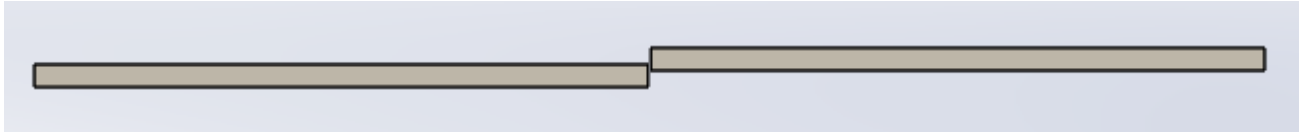


Figure 32 Shear Force Anti Plane Shear

Mode 3 (Anti Plane Shear)

Blade Mass = 564 g

$$\sigma = 213.4 \text{ Mpa} = 213.4 \text{ MN/m}^2$$

$$\text{Sum of intervals} = 0.275/4 = 0.06875 \text{ m}$$



Figure 33 Shear Force Turbine Blade

$$213.4 + (564 \times 0.275) = 368.5 \text{ MN/m}^2$$

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

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

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

$$213.4 + (564 \times (0.275 - 0.275)) = 0$$

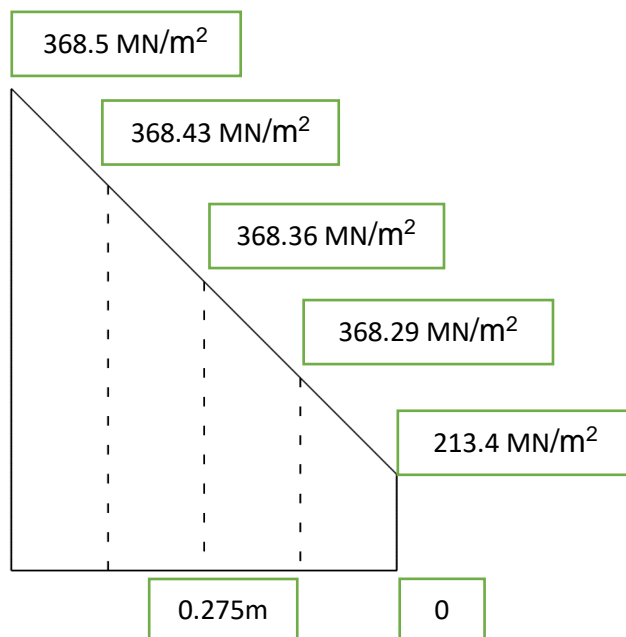


Figure 34 Shear Force Diagram

Bending Moment

$$(-213.4 \times 0.275) - \left(\frac{564 \times 0.275^2}{2} \right) = -11.969 \text{ MN/m}^2$$

$$(-213.4 \times (0.275 - 0.20625)) - \left(\frac{564 \times (0.275 - 0.20625)^2}{2} \right) = -16.00 \text{ MN/m}^2$$

$$(-213.4 \times (0.275 - 0.1375)) - \left(\frac{564 \times (0.275 - 0.1375)^2}{2} \right) = -34.6700 \text{ MN/m}^2$$

$$(-213.4 \times (0.275 - 0.06875)) - \left(\frac{564 \times (0.275 - 0.06875)^2}{2} \right) = -56.0000 \text{ MN/m}^2$$

$$(-213.4 \times (0.275 - 0.275)) - \left(\frac{564 \times (0.275 - 0.275)^2}{2} \right) = 0 \text{ MN/m}^2$$

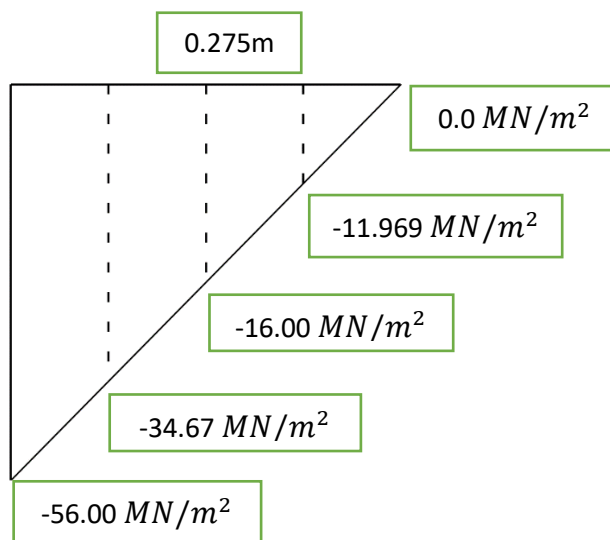


Figure 35 Bending Moment Diagram

Conclusion

Shear force = 368 MN/m²/Mpa

Bending Moment = 56 MN/m²/Mpa

The shear force is slightly more than the yield as less would not shear the material, the bending moment happens just after allowable stress.

Factor of Safety

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

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

And

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

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

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

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

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.

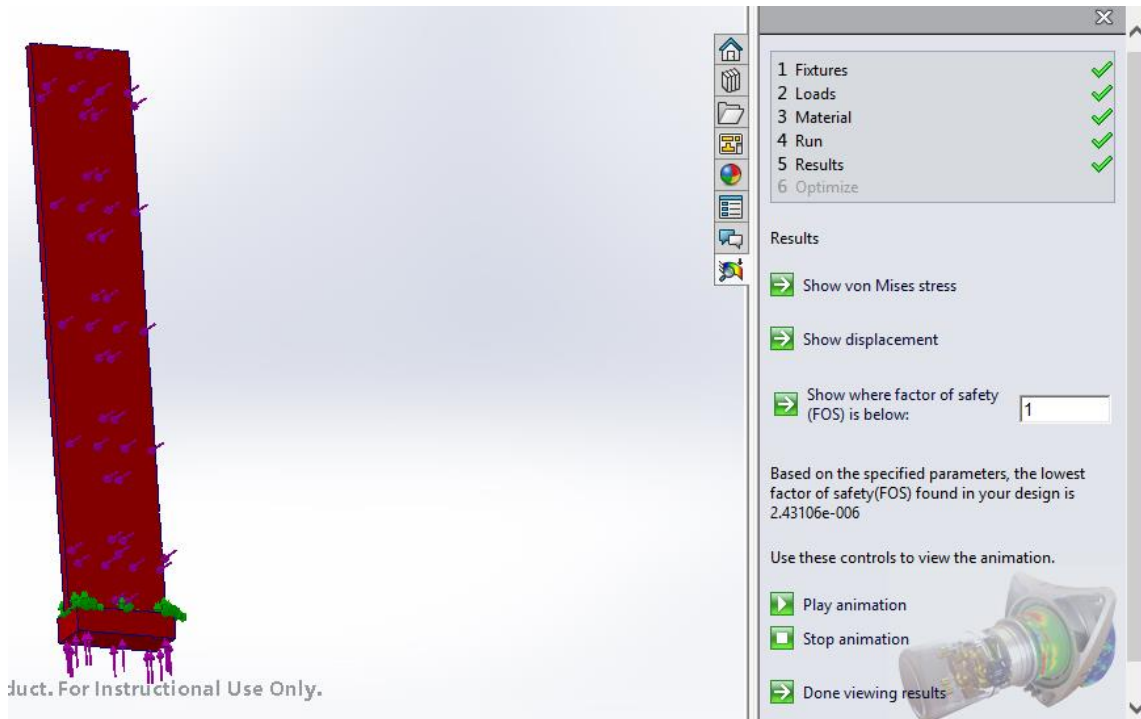


Figure 36 SolidWorks Factor of Safety

(The Royal Academy of Engineering, ND)

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	2.11x10 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

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

The Blade has a life estimation of 9.44 years (10⁷). Life Cycle per year is 65.7 million Cycles (10⁶), 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 (10⁷).

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

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

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

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

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

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

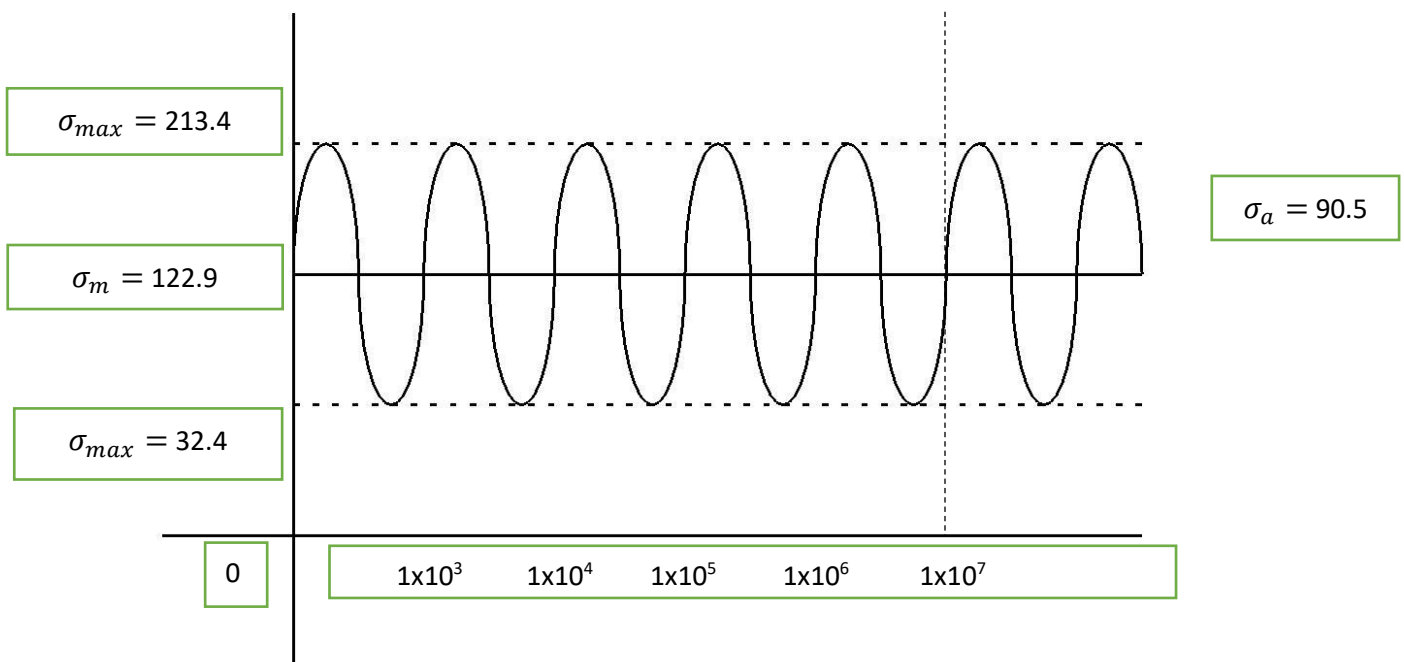


Figure 37 Fatigue Cycle Chart 1

Stress Range Values

Table 14 Stress Range Values

$(\sigma)_{max}$	213.4 Mpa
$(\sigma)_{min\ allowed}$	52.3 Mpa

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

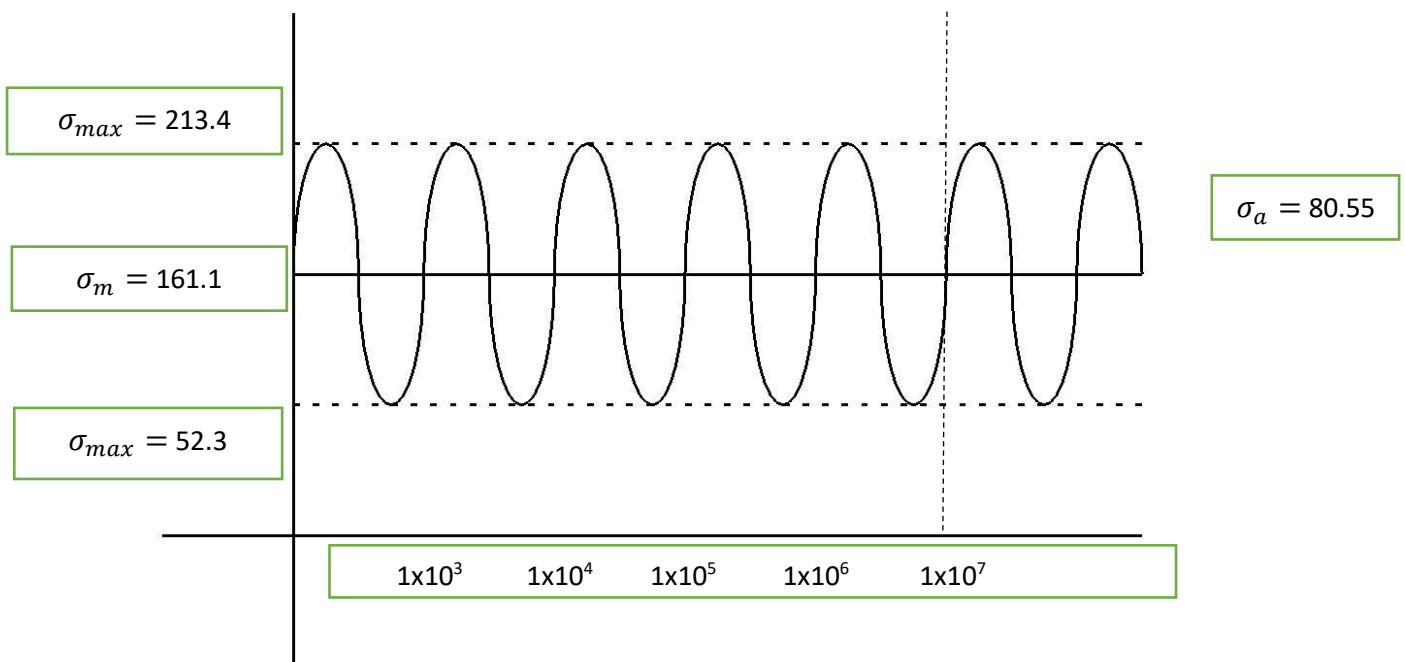


Figure 38 Fatigue Cycle Chart 2

Results test 1

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

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

Results Test 2

$$213.4 - 52.3 = 161.1 \times 620.42^{0.1} = 303 \text{ Mpa} = 4.6 \text{ yr}$$

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

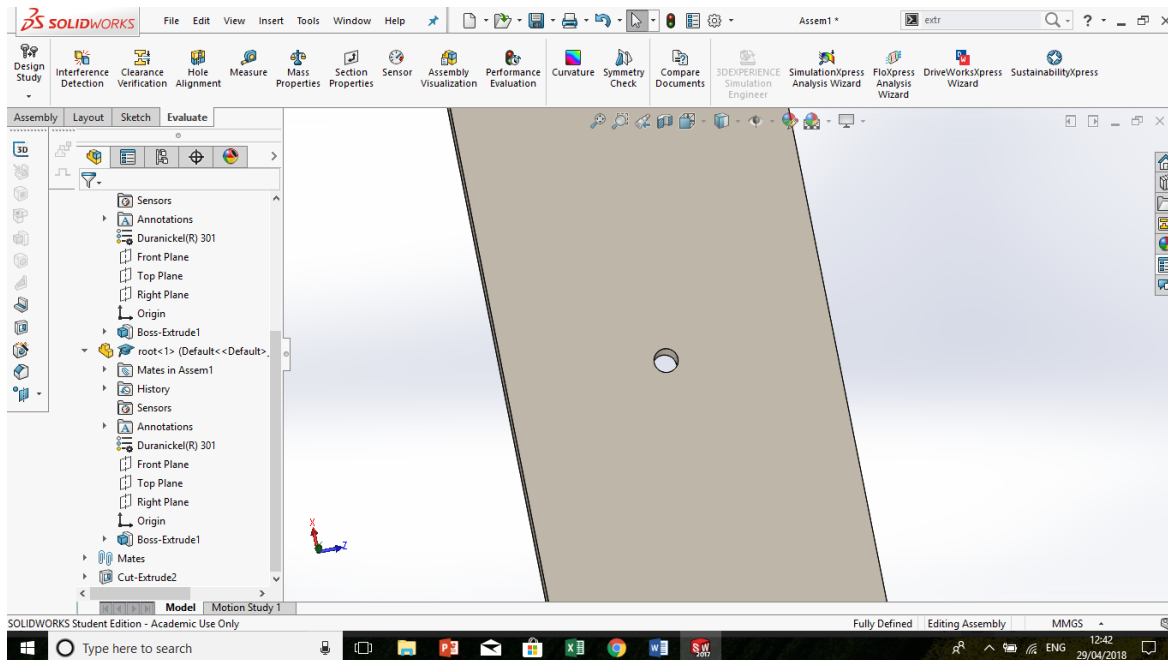


Figure 39 Fracture Toughness

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

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

$$213.4\sqrt{\pi \frac{5 \times 10^{-3}}{2}} = 18.9 \text{ Mpa}$$

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

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

$$= 16.9 \text{ Gpa}$$

52.3 Mpa

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

$$52.3\sqrt{\pi} \frac{5 \times 10^{-3}}{2} = 4.6 \text{ Mpa}$$

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

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

$$= 1 \text{ Gpa}$$

32.4 Mpa

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

$$32.4\sqrt{\pi} \frac{5 \times 10^{-3}}{2} = 2.87 \text{ Mpa}$$

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

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

$$= 0.390 \text{ Gpa}$$

(Loughborough University, 2010)

Results

Using UTS as Endurance Limit

$$213.4 \text{ Mpa} = \frac{620420000}{16929.38 \times 10^6} = 0.036 \text{ yrs}$$

$$52.3 \text{ Mpa} = \frac{620420000}{1002.8 \times 10^6} = 0.61 \text{ yrs}$$

$$32.4 \text{ Mpa} = \frac{620420000}{390 \times 10^6} = 1.5 \text{ yrs}$$

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.

$$\text{Average stress} = 52.3 - 32.4 = 19.9/2$$

$$\sigma_a = 9.95$$

$$= 32.4 + 9.95$$

$$= 42.35 \text{ Mpa}$$

Life Expectancy

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

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

<i>Average Stress (σ_1) = 42.35 Mpa</i>
<i>Lower Stress (σ_0) = 32.4 Mpa</i>
<i>Temperature Material = 590 J/Kg/K</i>
<i>Temperature Turbine = 480 J/Kg/K</i>
<i>Blade Length = 275 mm</i>
<i>Blade Casing = 1 mm Gap between case and blades</i>
<i>n = stress exponent for metal</i>
<i>R = Gas Constant = 8.314 k/mol</i>
<i>T = 273K</i>
<i>Activation Creep = $Q_c = 240 \text{ kJ/mol}$ (half of 480°C) (Lu 2018)</i>
<i>ϵ = strain</i>
<i>$\dot{\epsilon}_{ss}$ = Steady State Strain rate</i>
<i>$\frac{1}{s}$ = 1 pascal to one second</i>

Exponent $n = 5$

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

(Deepak, 2016)

Calculation

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

For $\sigma = 42.35 \text{ Mpa}$

$$\dot{\varepsilon}_{ss1} = \dot{\varepsilon}_0 \left(\frac{\sigma_1}{\sigma_0} \right)^n \text{Exp}\left(\left(-\frac{Q_c}{RT_1}\right)\right)$$

$$10^6 \times \left(\frac{42.35 \times 10^6}{32.4 \times 10^6} \right)^5 = 3815417.104 \text{ pa}$$

$$\text{Exp}\left(-\frac{240 \times 10^3}{8.314 \times (480 + 273)}\right) = e^{38.335} = 4.45$$

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

1mm = Blade Tip Clearance

275 mm = Blade Length

Ductility

$$\varepsilon = \frac{\Delta L}{L_0} = \frac{1}{275} = 0.0036$$

$$t = \frac{\varepsilon}{\dot{\varepsilon}_{ss}} = \frac{0.0036}{857397.102} = \frac{\frac{4.2 \times 10^{-9} (\text{Sec})}{60 (\text{Min})}}{24 (\text{Day})} = \frac{4.859669641 (\text{Days})}{365 (1 \text{ Year})}$$

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

Using Formula

$$\dot{\epsilon}_{ss} = \dot{\epsilon}_0 \left(\frac{\sigma}{\sigma_0}\right)^n \exp\left(-\frac{Q_c}{RT}\right)$$

Increase blade life by running at half speed = half stress

$$\sigma_2 = 21.175 = (\text{Half the Mean Stress})$$

$$T_2 = 480^\circ\text{C}$$

$$\frac{\dot{\epsilon}_{ss2} = \dot{\epsilon}_0 \left(\frac{\sigma_2}{\sigma_0}\right)^n \text{Exp}\left(-\frac{Q_c}{RT_2}\right)}{\dot{\epsilon}_{ss1} = \dot{\epsilon}_0 \left(\frac{\sigma_1}{\sigma_0}\right)^n \text{Exp}\left(-\frac{Q_c}{RT_1}\right)} = \left(\frac{\sigma_2}{\sigma_1}\right)^n = \left(\frac{21.175}{42.35}\right)^5 = 0.03125$$

Monkman – Grant Law

$$t_f \cdot \dot{\epsilon}_{ss} = \text{constant}$$

$$\frac{t_{f2}}{t_{f1}} = \frac{\dot{\epsilon}_{ss1}}{\dot{\epsilon}_{ss2}}$$

$$\frac{\dot{\epsilon}_{ss1}}{\dot{\epsilon}_{ss2}} = \left(\frac{42.35 \times 10^6}{21.175 \times 10^6}\right)^5 = 32$$

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

Life extended by a factor of 31 (Lu 2018)

Using Formula

$$\dot{\epsilon}_{ss} = \dot{\epsilon}_0 \left(\frac{\sigma}{\sigma_0}\right)^n \exp\left(-\frac{Q_c}{RT}\right)$$

Max material Temperature = 590°C

$$\sigma_3 = 42.35 \text{ Average stress}$$

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

Thus

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

$$\dot{\varepsilon}_{ss1} = \dot{\varepsilon}_0 \left(\frac{\sigma_1}{\sigma_0}\right)^n \text{Exp}\left(-\frac{Q_c}{RT_1}\right)$$

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

$$= \text{Exp}\left(-\frac{240 \times 10^3}{8.31 \times (590 + 273)} + \frac{240 \times 10^3}{8.31 \times (480 + 273)}\right) = \text{Exp}(4.886) = e^{132.4}$$

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

1mm = Blade Tip Clearance

275 mm = Blade Length

$$\varepsilon = \frac{\Delta L}{L_0} = \frac{1}{275} = 0.0036$$

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

At an average stress and maximum temperature, the material Duranickel would last just 3 seconds under heat supplied at 590°C applied. At maximum temperature stated in SolidWorks. Metal deforms very easily especially when subjected to heat and rotational force hence creep.

Conclusion

Duranickel can withstand the high temperature and stresses when operating in a steam turbine system at 480°C, 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 are close to a blades 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 blades 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, but other factors such as environment play a role.



Figure 40 Turbine Blade Cracking

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 41 Turbine Blade Corrosion

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



Figure 42 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).

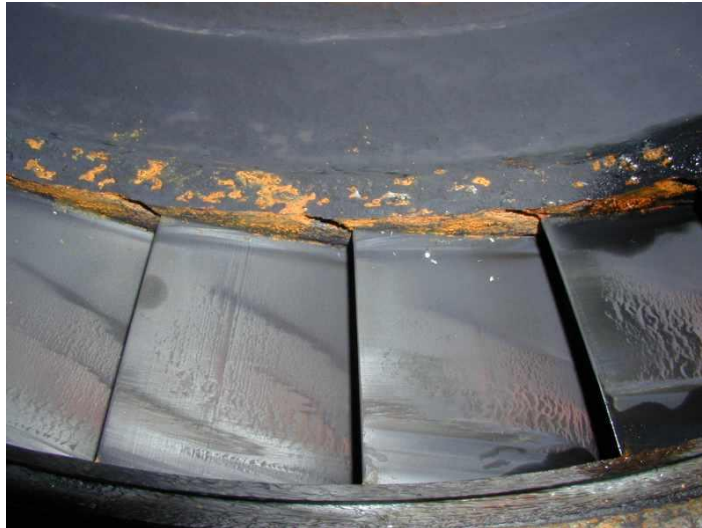


Figure 43 Turbine Blade Erosion and Corrosion (Power Corrosion Ltd, 2017)

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 error message, I tried three different computers, I removed the Add-ins it asked for and still it would not let me access the features.

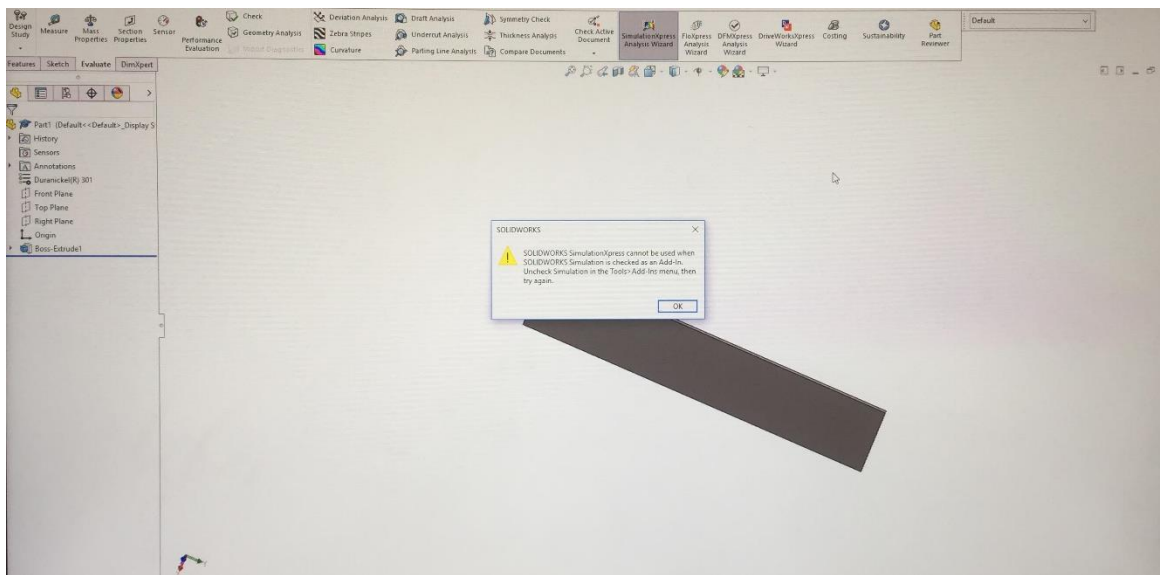


Figure 44 Denial to Access

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°C 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°C, 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°C, in a hostile environment, whilst under rotational forces rated at 125 cycles a second.

In short

Good choice of material selection, Excellent mechanical strength, supplied with a year's warranty.

Blades Positioned

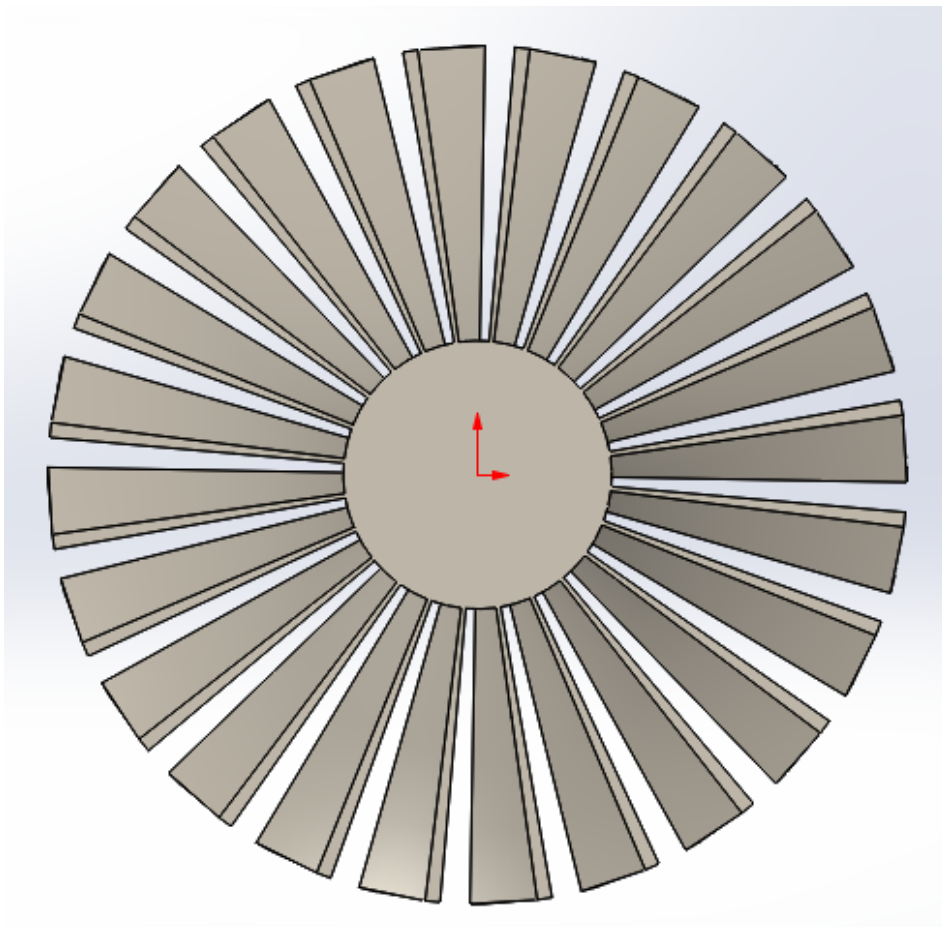


Figure 45 Blades Positioned

Research

(Siemens, 2013)

(WorldStainless.Org, N.D.)

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Appendix

