SCHOOL OF ENGINEERING
TAYLOR'S UNIVERSITY

School of Engineering

MEC60803

Design of Engineering Components and System

Assignment

Lecturer:

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Due Date: 30th November 2019

Date of Submission: 30th November 2019

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Introduction

In this assignment, we are required to design a high pressured cylindrical tank with various spring hangers which is used to support the piping subject to vertical vibration where constant support is not required. Power screw mechanism is used for the platform when the tank is brought up and down for maintenance and other operational tasks purpose. The group is required to design spring, rivets and bolts which is used by the overall design of the tank. All the calculations are based on minimum requirements which was set beforehand. After calculating the required specification, a 3D drawing will be drawn in Solidworks with the exact dimensions used.

Table 1: Data and requirements given.

Mass of the tank	200kg
Pressure in the tank, P	1MPa
Tank inner diameter	3m
Wall Thickness	20mm
Minimum deflection of the spring base	Less than 20 mm
Maximum permissible shear stress of spring wire	400MPa
Maximum permissible shear stress for rivets and plate	120N/ mm^2
Modulus of rigidity	80kN/ mm^2

Team Organisational Chart

Name	Role	Responsibilities
Teoh Zhi Heng	Leader	• Distributed tasks to all group members • Responsible for calculation and analysis of power screw
Alwyn Yip Winn Sheng	Analyst	• Responsible for calculation and analysis of spring • Assist in the calculations of other designs • Assists in 3D drawing
Hong Jian Hua	Designer	• Assists in the calculation for design • Create 3D drawings of design using Solidworks
Jason Chong Jia Joon	Analyst	• Responsible for calculation and analysis of rivets • Assist in the calculations of other designs • Assists in 3D drawing
Tan Jia Hao	Analyst	• Assists in 3D drawing • Responsible for calculation and analysis of bolts • Assist in the calculations of other designs

Analysis

Spring

Standard wire gauge (SWG) number and corresponding diameter of spring wire

SWG	Diameter (mm)	SWG	Diameter (mm)	SWG	Diameter (mm)	SWG	Diameter (mm)
7/0	12.70	7	4.470	20	0.914	33	0.2540
6/0	11.785	8	4.064	21	0.813	34	0.2337
5/0	10.973	9	3.658	22	0.711	35	0.2134
4/0	10.160	10	3.251	23	0.610	36	0.1930
3/0	9.490	11	2.946	24	0.559	37	0.1727
2/0	8.839	12	2.642	25	0.508	38	0.1524
0	8.229	13	2.337	26	0.457	39	0.1321
1	7.620	14	2.032	27	0.4166	40	0.1219
2	7.010	15	1.829	28	0.3759	41	0.1118
3	6.401	16	1.626	29	0.3454	42	0.1016
4	5.893	17	1.422	30	0.3150	43	0.0914
5	5.385	18	1.219	31	0.2946	44	0.0813
6	4.877	19	1.016	32	0.2743	45	0.0711

Figure 1: SWG table for spring.

Based on the SWG table above, SWG 0, 1, and 2 are assessed based on the requirements given for the spring. The requirements given are minimum deflection and also the maximum permissible shear stress for the spring wire. By comparing between 3 standard wire gauge, it is found that SWG 0 meets all the standard requirements. SWG 1 produces a deflection of less than 20mm however exceeds the shear stress exceeds the permissible value which may cause the spring to fail. As for SWG 2, the deflection and shear stress exceeds both the minimum and maximum requirements.

As for the type of spring, plain ends and ground ends are not stable and will wobble when attached to a flat surface. These ends absorbs less force and more elasticity which is not suitable to hold a high pressure tank. In addition, plain and ground ends does not contain inactive coil which is not suitable for this application. As for squared and ground ends, they are more commonly used, but it is slightly costly. This is because the ends of these springs requires grinding which adds up to the cost of the spring. The reason why we chose squared and ground is because it helps with the vibration of the high pressure tank since the spring will be standing vertically on a straight surface.

Table 2: Comparison between three SWG classes.

SWG	**0**	**1**	**2**
Diameter of spring wire, d(mm)	**8.23**	7.62	7.01
Number of turns , n	**12**	12	12
Waals factor , K	**1.45**	1.41	1.37
Mean Diameter, D(mm)	**30**	30	30
Deflection, δ (mm)	**13.86**	18.85	26.33
Max Shear Stress, τ (MPa)	**390.61**	478.19	597.15

Table 3: Comparison between the type of ends.

Type of End	**Total number of turns, n'**	**Solid Length(mm)**	**Free length(mm)**	**Pitch(mm)**
Plain Ends	12	106.99	114.67	10.42
Ground Ends	12	98.76	114.67	10.42
Squared Ends	14	123.45	131.16	10.09
Squared and ground ends	**14**	**115.22**	**131.16**	**10.09**

1) **Mean diameter of the spring**

$$\delta = \frac{8WD^3n}{Gd^4}$$
$$= \frac{8(200\times 9.81)(30)^3(12)}{(80000)(8.23)^4}$$
$$= \mathbf{13.86mm}$$

$$C = D/d$$
$$= 30/8.23$$
$$= \mathbf{3.65}$$

$$K = \frac{4C-1}{4C-4} + \frac{0.615}{C}$$
$$= \frac{4(3.65)-1}{4(3.65)-4} + \frac{0.615}{(3.65)}$$
$$= \mathbf{1.45}$$

$$\tau = \frac{8FC}{\pi d^2}K$$
$$= \frac{8(200\times 9.81)(3.65)}{\pi(8.23)^2}(1.45)$$
$$= \mathbf{390.61MPa}$$

2) **Number of turns of the coils**

The number of turns of the coil assumed is at 12 turns which produces a deflection at 13.86mm.

3) **Free length of the spring**

$$L_F = n'd + \delta_{max} + 0.15\delta_{max}$$

$$= 14(8.23) + 13.86 + 0.15(13.86)$$

$$= \mathbf{131.16mm}$$

4) Pitch of the coil

$$p = \frac{L_F}{n' - 1}$$

$$= \frac{131.16}{14 - 1}$$

$$= \mathbf{10.09mm}$$

Details for different types of end connections for springs

Type of end	*Total number of turns (n')*	*Solid length*	*Free length*
1. Plain ends	n	$(n + 1)\,d$	$p \times n + d$
2. Ground ends	n	$n \times d$	$p \times n$
3. Squared ends	$n + 2$	$(n + 3)\,d$	$p \times n + 3d$
4. Squared and ground ends	$n + 2$	$(n + 2)\,d$	$p \times n + 2d$

Figure 2: Table for type of spring end.

The type of spring chosen would be squared and ground end because it has a flat surface at the end of spring. This will ensure when the spring is placed on a flat surface it does not wobble a lot and it helps to withstand the vibration of the high pressure tank movement. The mean diameter of the spring is 30mm and has a total number of turns on the coil of 14 including the inactive coil. The spring also has a free length of 131.6mm with the pitch of 10.09mm. The 3D model of the spring is shown in the next page.

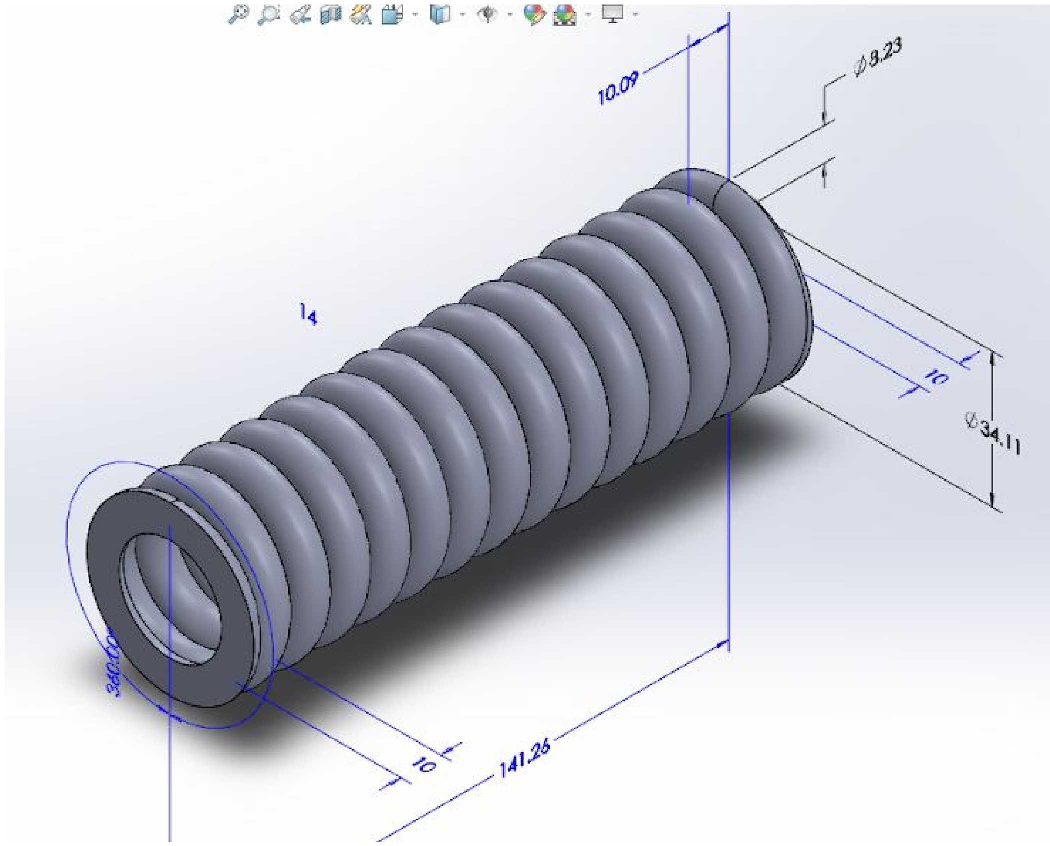

Figure 3: 3D Model of the spring with dimensions.

Rivet

Before deciding whether to use bolts or rivets to secure the flange of the pipe, an analysis was done to compare which option would be better. The criteria used for comparison were number of bolts or rivets that would be used to secure the flange and the size of the flange based on the results of the best rivet and bolt. Some information was provided before the beginning the analysis such as the diameter of the tank, the thickness and other relevant forces.

Size of rivet diameters for rivet hole diameter (IS:1928-1961)

Basic size of rivet mm	12	14	16	18	20	22	24	27	30	33	36	39	42	48
Rivet hole diameter (min) mm	13	15	17	19	21	23	25	28.5	31.5	34.5	37.5	41	44	50

Figure 4: Rivet size and rivet hole size table.

An analysis was done for the rivet to determine the number of rivets needed to secure the flange. The permissible stress for rivets was given to be 120MPa and the thickness of the plate was 20 mm. Since the thickness is larger than 8mm, the formula $d = 6\sqrt{t}$ was used to determine the minimum requirement for the diameter of the rivet, which was 28.5mm. After the minimum diameter was determined, the next step of the analysis was to adjust the size of the diameter and determined the number of rivets needed for each selected diameter. The diameter that was analysed was 28.5mm, 31.5mm, 34.5mm and 50mm. By using the inner radius of 1.5m, thickness of 0.02m and the distance between the centre of the rivets to the body, the circumference of the flange and clearance for the rivets was determined The table below were the results of the analysis.

Table 4. Results of rivets.

Diameter of rivets, mm	Number of rivets	Clearance between rivets, mm	Size of flange, mm
28.5	93	104.62	57
31.5	76	128.27	63
34.5	64	152.61	69
50	30	328.82	100

Based on the analysis above, the clearance between rivets were sufficient hence all of the rivets analysed were accepted. Based on the results of number of rivets needed, the most number is 93 and the least is 30. The lesser the number of rivets used, the easier during the installation and maintenance process. The size of the flange is dependant on the diameter of the rivet and is twice the size of the diameter. Hence, the best rivet was with the rivet with a diameter of 50mm, with the flange size of 100mm and number of rivets of 30. This is shown in the solidworks drawing below.

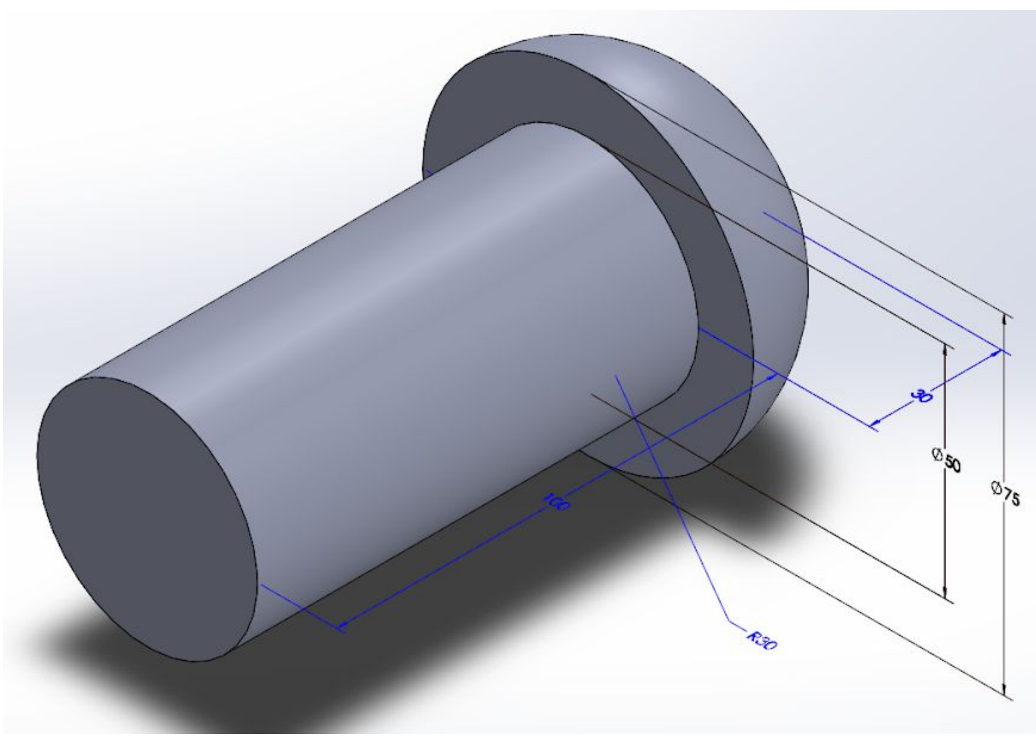

Figure 5. 3D Model of rivet with dimensions.

Bolts

After the analysis of rivet was done, the next step was to analyse the bolts that would be used to secure the flange. For the analysis of the bolt, the pressure and the diameter of the tank which were 1MPa and 3m respectively were determined based on the information provided. The area and the force of the tank were calculated to be 7.0685m^2 and 7068583.471 N. The assumption made were the SAE class for the bolt and a safety factor of 2.5 was used during the analysis. The first step of the analysis was to determine whether a coarse or fine thread bolt would be a more suitable choice. To do this, the nominal diameter of the bolt was selected to be 16mm and 18mm for comparison of coarse and fine thread bolts. The table below are some information of coarse bolt and fine bolt.

ISO Metric Screw Thread

	Coarse Threads			Fine Threads		
Nominal Diameter d (mm)	Pitch p (mm)	Minor Diameter d_r (mm)	Stress Area A_t (mm^2)	Pitch p (mm)	Minor Diameter d_r (mm)	Stress Area A_t (mm^2)
3	0.5	2.39	5.03			
3.5	0.6	2.76	6.78			
4	0.7	3.14	8.78			
5	0.8	4.02	14.2			
6	1	4.77	20.1			
7	1	5.77	28.9			
8	1.25	6.47	36.6	1	6.77	39.2
10	1.5	8.16	58.0	1.25	8.47	61.2
12	1.75	9.85	84.3	1.25	10.5	92.1
14	2	11.6	115	1.5	12.2	125
16	2	13.6	157	1.5	14.2	167
18	2.5	14.9	192	1.5	16.2	216
20	2.5	16.9	245	1.5	18.2	272
22	2.5	18.9	303	1.5	20.2	333
24	3	20.3	353	2	21.6	384
27	3	23.3	459	2	24.6	496
30	3.5	25.7	561	2	27.6	621
33	3.5	28.7	694	2	30.6	761
36	4	31.1	817	3	32.3	865
39	4	34.1	976	3	35.3	1030

Note: Metric threads are identified by diameter and pitch as "M8 × 1.25."

Specification for steel used in milimeter series screws and bolts

SAE Class	Diameter d (mm)	Proof Load (Strength)[a] S_p (MPa)	Yield Strength[b] S_y (MPa)	Tensile Strength S_u (MPa)	Elongation, Minimum (%)	Reduction of Area, Minimum (%)	Core Hardness, Rockwell Min	Core Hardness, Rockwell Max
4.6	5 thru 36	225	240	400	22	35	B67	B87
4.8	1.6 thru 16	310	—	420	—	—	B71	B87
5.8	5 thru 24	380	—	520	—	—	B82	B95
8.8	17 thru 36	600	660	830	12	35	C23	C34
9.8	1.6 thru 16	650	—	900	—	—	C27	C36
10.9	6 thru 36	830	940	1040	9	35	C33	C39
12.9	1.6 thru 36	970	1100	1220	8	35	C38	C44

[a] Proof load (strength) corresponds to the axially applied load that the screw or bolt must withstand without permanent set.
[b] Yield strength corresponds to 0.2 percent offset measured on machine test specimens.
Source: Society of Automotive Engineers standard J1199 (1979).

Figure 6: SAE class and nominal diameter size table of bolts.

Table 5: Coarse thread bolt.

Nominal diameter, mm	Stress Area, mm^{2}.	Proof strength, MPa	Ultimate force, N
16	157	600	94200
18	192	600	115200

Table 6: Fine thread bolt.

Nominal diameter, mm	Stress Area, mm^{2}.	Proof strength, MPa	Ultimate force, N
16	167	600	100200
18	216	600	129600

Based on the information above, an analysis was done and the actual force of the bolt and the number of bolts needed were compared to see which thread bolt would be better.

Table 7: Analysis of coarse thread bolt.

Nominal diameter, mm	Actual force, N	Number of bolts	Size of flange, mm
16	62800	113	32
18	76800	93	36

Table 8: Analysis of fine thread bolt.

Nominal diameter, mm	Actual force, N	Number of bolts	Size of flange, mm
16	66800	106	32
18	86400	82	36

Based on the results of the analysis, the actual force with a safety factor of 2.5 of fine thread bolt was higher than coarse thread and the number of bolts needed was lesser. Hence, a fine thread bolt was used. The next step was to analyse the fine thread bolt at different diameter for comparison with rivet.

Table 9: Results for fine thread bolt.

Nominal diameter, mm	Stress Area, mm^2	Ultimate force, N	Actual force, N	Number of bolts	Size of flange
16	167	100200	66800	106	32
18	216	129600	86400	82	36
20	272	163200	108800	65	40
22	333	199800	133200	54	44
24	384	230400	153600	47	48
27	496	29760	198400	36	54
30	621	372600	248400	29	60

Based on the results above, as the diameter of the bolt increased, the number of bolts needed to secure decreases. As mentioned earlier in the analysis on rivet, the lesser the number of rivets, the easier during the installation and maintenance process. The same principle is applied to the analysis of bolt. The size of the flange is dependant on the diameter of the bolt used and is twice the size of the diameter. Hence, the best option for bolt was the bolt with a diameter of 30mm, the flange size of 60 and the number of bolts to be 29. This will be shown below, with a 3D Model drawing with dimensions in mm.

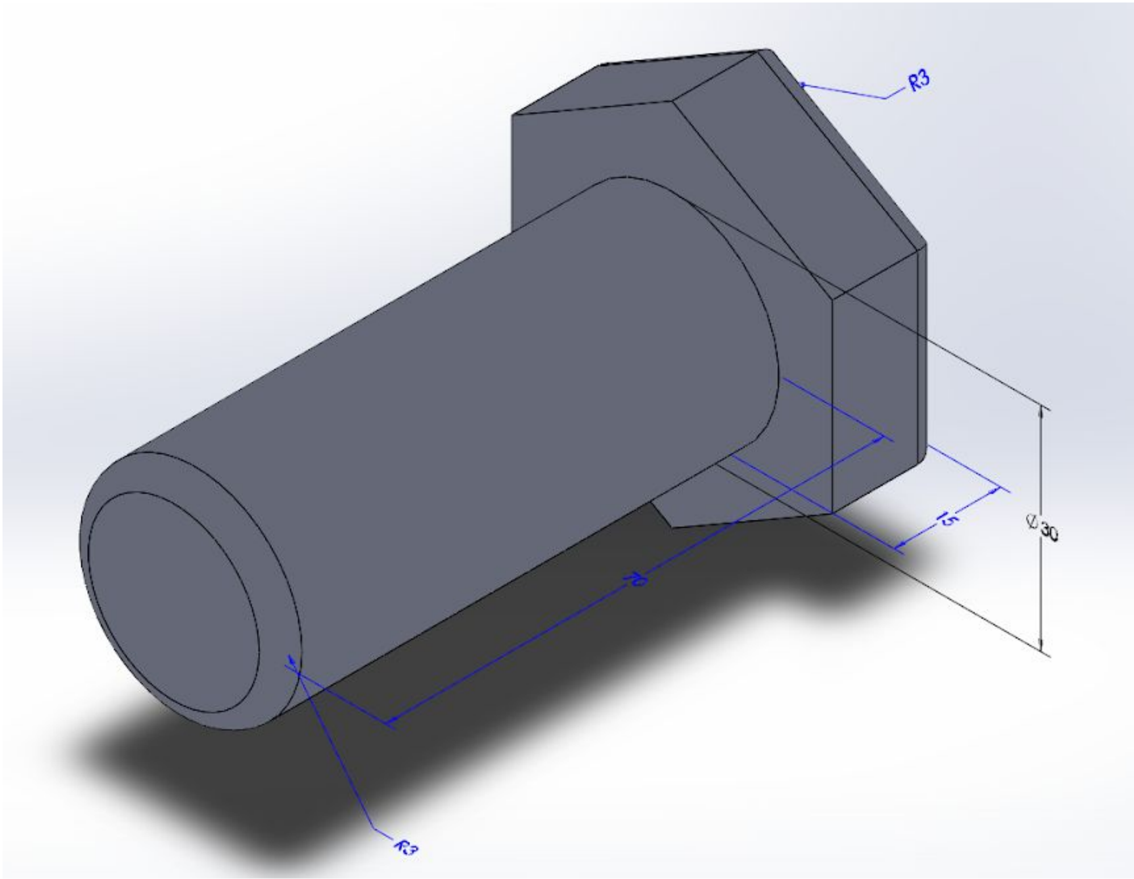

Figure 7: Bolt 3D Model with dimensions in mm.

After the analysis for both rivets and bolts were done, the final step would be to compare the results and determine which option would be more suited for the applications. The table below was the results of the analysis for the best rivet and the best bolt.

Table 10: Results of rivet and bolt.

Type of fastener	Diameter, mm	Number required	Size of flange, mm
Rivet	50	30	100
Bolt	30	29	60

Based on the results obtained, bolt would be the most suitable fastener for the applications as the number required was lesser and the size of the flange was smaller and the overall cost would be reduced. As mentioned earlier, the criteria that would be used to determine which fastener is more suitable is the number of rivets or bolts used and the size of the flange. These two criteria was chosen as the focus of the analysis as it is the main consideration for this application. The number of bolts or rivets used would affect the cost, efficiency and the installation and maintenance of the flange. The size of the flange is dependant on the diameter of the type of fastener used. The bigger the diameter of the fastener the bigger the flange would be. Another advantage of choosing bolts over rivets is due to the method of fastening. Using a bolt allows for future disassemble for replacement of rusted bolt or for maintenance purposes while rivet is permanently fastened to the flange.

Power Screw

Threads are used for fastening, adjusting and transmitting power. We will be focussing on power screws in this section. Power screw is an object that is designed for smooth transmission of power, with no emission of noise. Power screw is to convert the rotary motion to slow linear and it has an ability to withstand and carry large loads. Besides, it can produce a uniform motion. Screw jack is the best example to prove that power screw can move heavy loads with minimal effort. Tensile test machine is the best example to prove that power screw can generate strong forces. Camera Calibration Rig is the best example to prove that power screw can allow object to be positioned precisely along the axial movement. In this assignment, the forms of power screw were recommended to be in square form, therefore, the sample calculations below are calculated based on the square form power screw. Below shows the formula that will be used to calculate the results. The results would be compared, and recommendations would be discussed below.

Total Torque to LIFT the tank

$$T = \frac{W d_m}{2} \frac{f \pi d_m + L cos \alpha_n}{\pi d_m cos \alpha_n - f L} + \frac{W f_c d_c}{2}$$

Total Torque to LOWER the tank

$$T = \frac{W d_m}{2} \frac{f \pi d_m - L cos \alpha_n}{\pi d_m cos \alpha_n + f L} + \frac{W f_c d_c}{2}$$

$T = Torque$

$W = Load$

$f = Coefficient\ of\ friction\ of\ thread$

$f_c = Coefficient\ of\ friction\ of\ trust\ collar$

$d_m = Mean\ diameter\ of\ thread\ contact$

$L = Lead$

$d_c = Collar\ diameter\ (thrust\ bearing\ diameter)$

Starting friction is about 1/3 higher than running friction

To find α_n **,** $tan\alpha_n = tan\alpha\ cos\lambda$

To find λ **,** $tan\lambda = \frac{L}{\pi d_m}$

Efficiency, $e = \frac{WL}{2\pi T} \times 100\%$

Given data

$pitch\ ,\ p = 0.006m$

$mass\ of\ tank\ ,\ m = 200kg$

$major\ diameter\ ,\ d_{major} = 0.036m$

$running\ coefficient\ (screw)\ , f = 0.15$

$running\ coefficient\ (collar)\ ,\ f_c = 0.12$

Sample Calculation (Double Thread Power Screw)

Determine the thread depth and helix angle

For single thread power screw, the thread depth is equal to the pitch. For double thread power screw, the thread depth is double the pitch. As the sample calculation is based on double thread power depth, therefore $L = 2p$. After obtaining the thread depth and mean diameter, helix angle can be determined with formula.

$thread\ depth\ ,\ L = 2p$

$thread\ depth\ ,\ L = 2 \times 0.006$

$thread\ depth\ ,\ L = 0.012m$

$mean\ diameter\ ,\ d_m = d_{major} - \frac{p}{2}$

$mean\ diameter\ ,\ d_m = 0.036 - \frac{0.006}{2}$

$mean\ diameter\ ,\ d_m = 0.033m$

$tan\lambda = \frac{L}{\pi d_m}$

$helix\ angle\ ,\ \lambda = tan^{-1}(\frac{L}{\pi d_m})$

$helix\ angle\ ,\ \lambda = tan^{-1}\left(\frac{0.012}{\pi\times0.033}\right)$

$helix\ angle\ ,\ \lambda = 6.6°$

Total Starting Torque to LIFT the tank

Starting friction is about 1/3 higher than the running coefficients, therefore, the running coefficients shown above must be multiplied by 4/3 to obtain the starting friction. After that, total starting torque to lift the tank will be calculated with the starting friction.

$Starting\ running\ coefficient\ (screw), f = 0.15\times\frac{4}{3}$

$Starting\ running\ coefficient\ (screw), f = 0.2$

$Starting\ running\ coefficient\ (collar), f_c = 0.12\times\frac{4}{3}$

$Starting\ running\ coefficient\ (collar), f_c = 0.16$

$Total\ starting\ Torque\ ,\ T_{Lifting} = \frac{Wd_m}{2}\frac{f\pi d_m + Lcos\alpha_n}{\pi d_m cos\alpha_n - fL} + \frac{Wf_c d_c}{2}$

$Total\ starting\ Torque\ ,\ T_{Lifting} = \frac{(1962)(0.033)}{2}\frac{(0.2\times\pi\times0.033)+(0.012\times cos0°)}{(\pi\times0.033\times cos0°)-(0.2\times0.012)} + \frac{(1962\times0.16\times0.080)}{2}$

$Total\ starting\ Torque\ ,\ T_{Lifting} = 23Nm$

$Total\ starting\ Torque\ ,\ T_{Lowering} = \frac{Wd_m}{2}\frac{f\pi d_m - Lcos\alpha_n}{\pi d_m cos\alpha_n + fL} + \frac{Wf_c d_c}{2}$

$Total\ starting\ Torque\ ,\ T_{Lowering} = \frac{(1962)(0.033)}{2}\frac{(0.2\times\pi\times0.033)-(0.012\times cos0°)}{(\pi\times0.033\times cos0°)+(0.2\times0.012)} + \frac{(1962\times0.16\times0.080)}{2}$

$Total\ starting\ Torque\ ,\ T_{Lowering} = 15.2Nm$

Efficiency of the jack when raising the tank

$e = \frac{WL}{2\pi T}\times100\%$

$e = \frac{(1962\times0.012)}{(2\times\pi\times23)}\times100\%$

e = 16.3%

Tabulated Results

Power Screw		Load , W (N)	pitch , p (m)	Thread Depth , L (m)	Major Diameter , d_{major} (m)	Mean Diameter , d_m (m)	Helix Angle	Torque , T (Nm)	Effficiency (%)
Single Thread	Lifting	1962	0.006	0.006	0.036	0.033	3.3°	21	8.9
	Lowering	1962	0.006	0.006	0.036	0.033	3.3°	17.1	11
Double Thread	Lifting	1962	0.006	0.012	0.036	0.033	6.6°	23	16.3
	Lowering	1962	0.006	0.012	0.036	0.033	6.6°	15.2	24.6

Figure 8: Results for power screw.

Efficiency was the key factor in choosing the best thread in the comparison. According to the table above, the lifting and lowering efficiency of the single thread power screw are 8.9% and 11% respectively. The lifting and lowering efficiency of the double thread power screw are 16.3% and 24.6% respectively. It was clearly shown that the double thread power screw has higher efficiency compared to the single thread power screw. Therefore, double thread power screw was recommended, with the dimensions of 6mm pitch, 12mm thread depth, major diameter of 36mm. The 3D model is shown below, with its labeled dimensions.

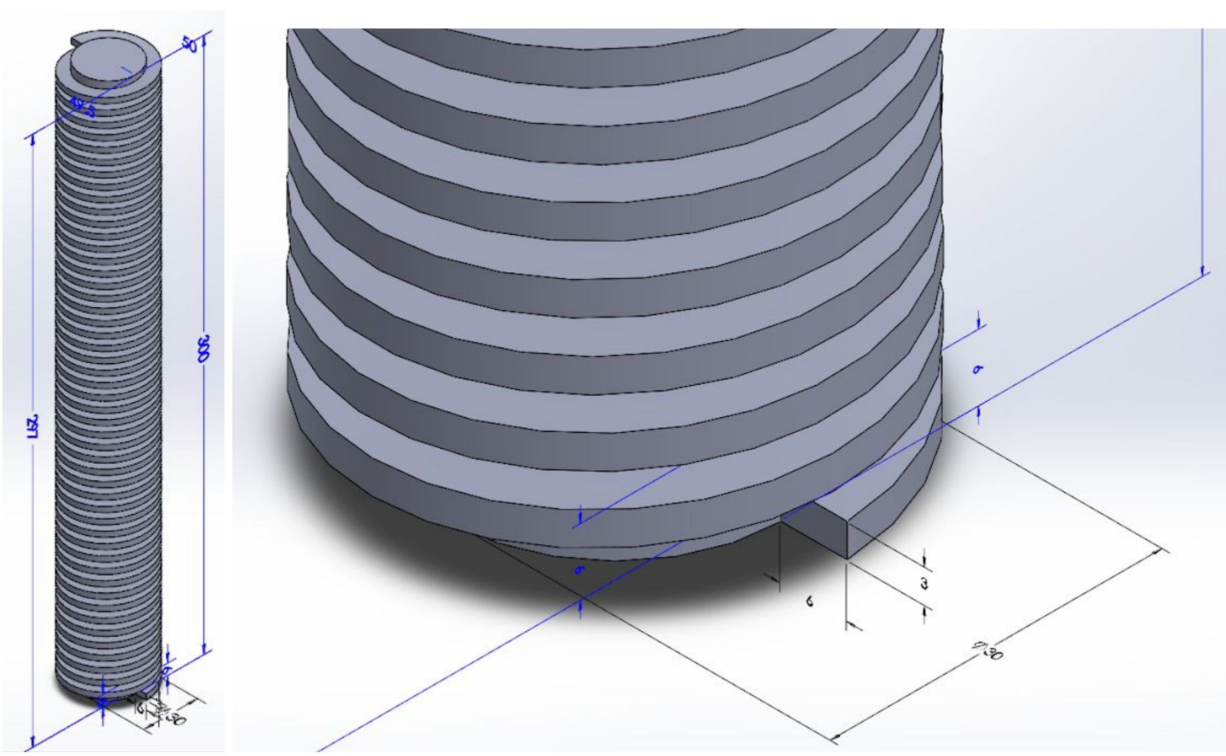

Figure 9: 3D Model of power screw with dimensions in mm.

Tank and Flange

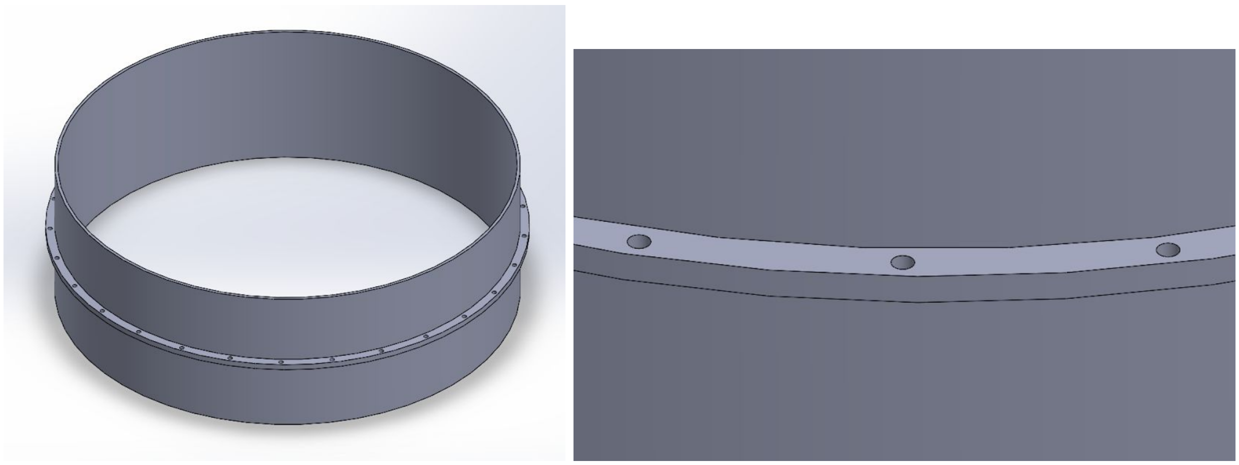

Figure 10: 3D Model of the tank.

As for the tank and flange, the values used were given as shown in Table 1, and diameter the small holes matches diameter of the bolts recommended. Dimensions are shown below along with the 3D Model.

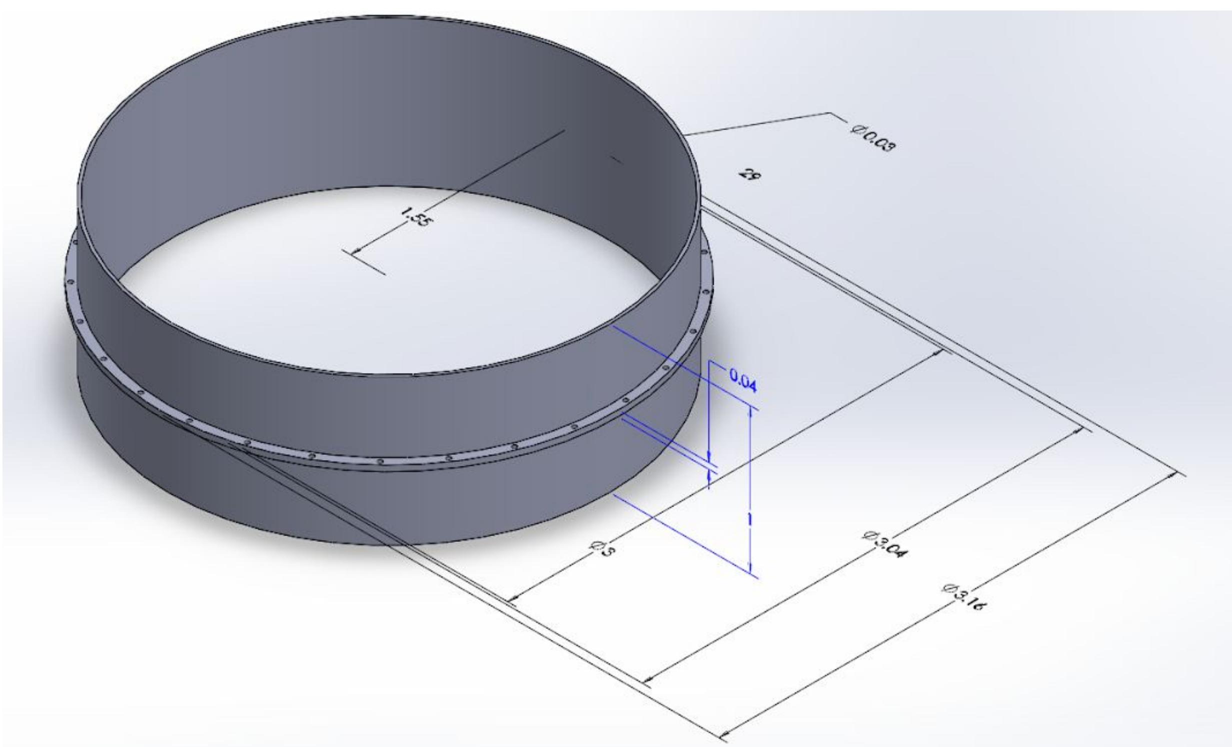

Figure 11: Dimensions of the tank.

Conclusion and Recommendation

To conclude this everything, we managed to design a special high pressure cylindrical tank with the spring hanger that is used to support the whole piping which is subject to vertical movement where there is no need of a fixed support. For the spring, we chose SWG 0 because the deflection and shear stress produced does not exceed the requirement of a maximum 20 mm deflection and 400 MPa. The type of spring used in this design is a squared and ground end because it has a flat surface at the end of spring. This will ensure when the spring is placed on a flat surface it does not wobble a lot and it helps to withstand the vibration of the high pressure tank movement. The mean diameter of the spring is 30mm and has a total number of turns on the coil of 14 including the inactive coil. The spring also has a free length of 131.6mm with the pitch of 10.09mm. For better improvement, it is recommended to use more number of spring to share the load therefore each spring will experience less shear stress and the deflection will be minimized.

To design the high pressure cylindrical tank we need to design the joint between two pipes on the flange by using either rivets or bolts. To choose which size of rivet and bolt, we decided to focus on using the less number of rivets or bolts needed to secure the pipe on the flange to save time and cost during the maintenance of the piping. The final result between the rivets and bolts is also determined by the number of fasteners needed and the ease of maintenance. Overall we decided to choose bolts instead of rivets because bolts requires less number of fasteners compared to the rivets and the size of the flanges is smaller than the rivets. Moreover bolts has an advantage that ahead of the rivet is that bolts can be disassembly during maintenance whereas the rivets is permanently fixed to the flange. For improvement of these fasteners would be to use a bigger size as it will reduce the number needed to secure the pipe together.

During the maintenance, a platform is needed to use so it could bring the tank up and down. This platform is design by using power screw. For power screw, the type of screw recommended is the square-thread with a major diameter of 36mm and has a pitch of 6mm by assuming the running coefficient of 0.15 for the screw and 0.12 for the collar. However we still need to determine whether to use single or double threaded by calculation. After calculating, we managed to find out that the double thread power screw is better than the single thread power screw because of the efficiency of the lifting and lowering the platform. Double thread has a higher efficiency compared to the single thread in both for lifting and lowering the platform. For improvement, it is better to use a multi thread power screw because it maintains a shallow thread depth relative to their longer lead distance. Moreover, the multi thread power screw has a better contact surface when is engaged in a single thread rotation.