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School of Engineering

MEC 60803
Design of Engineering Components and Systems

Assignment Report

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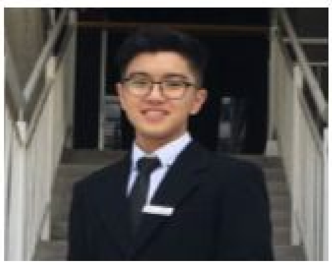

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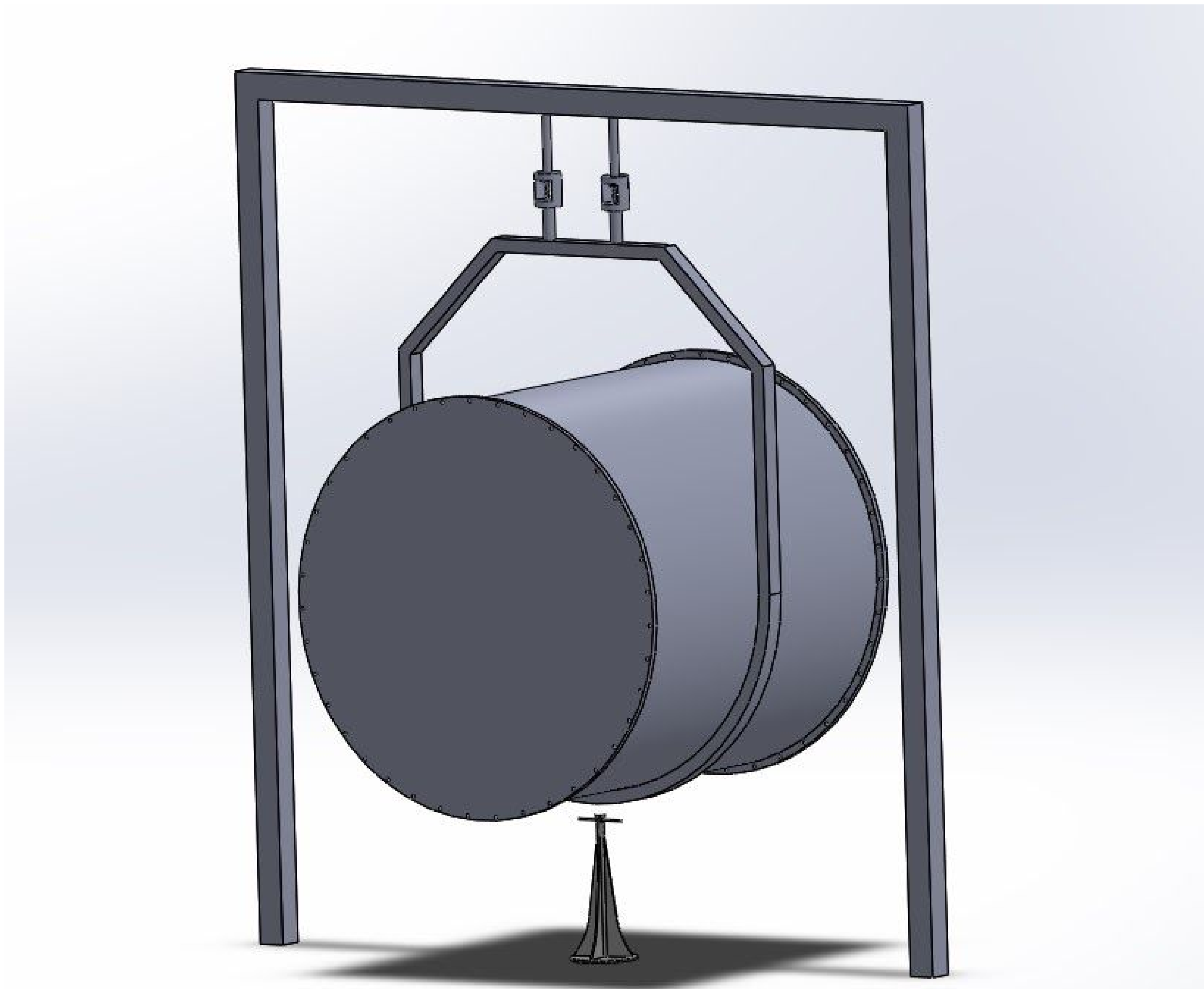

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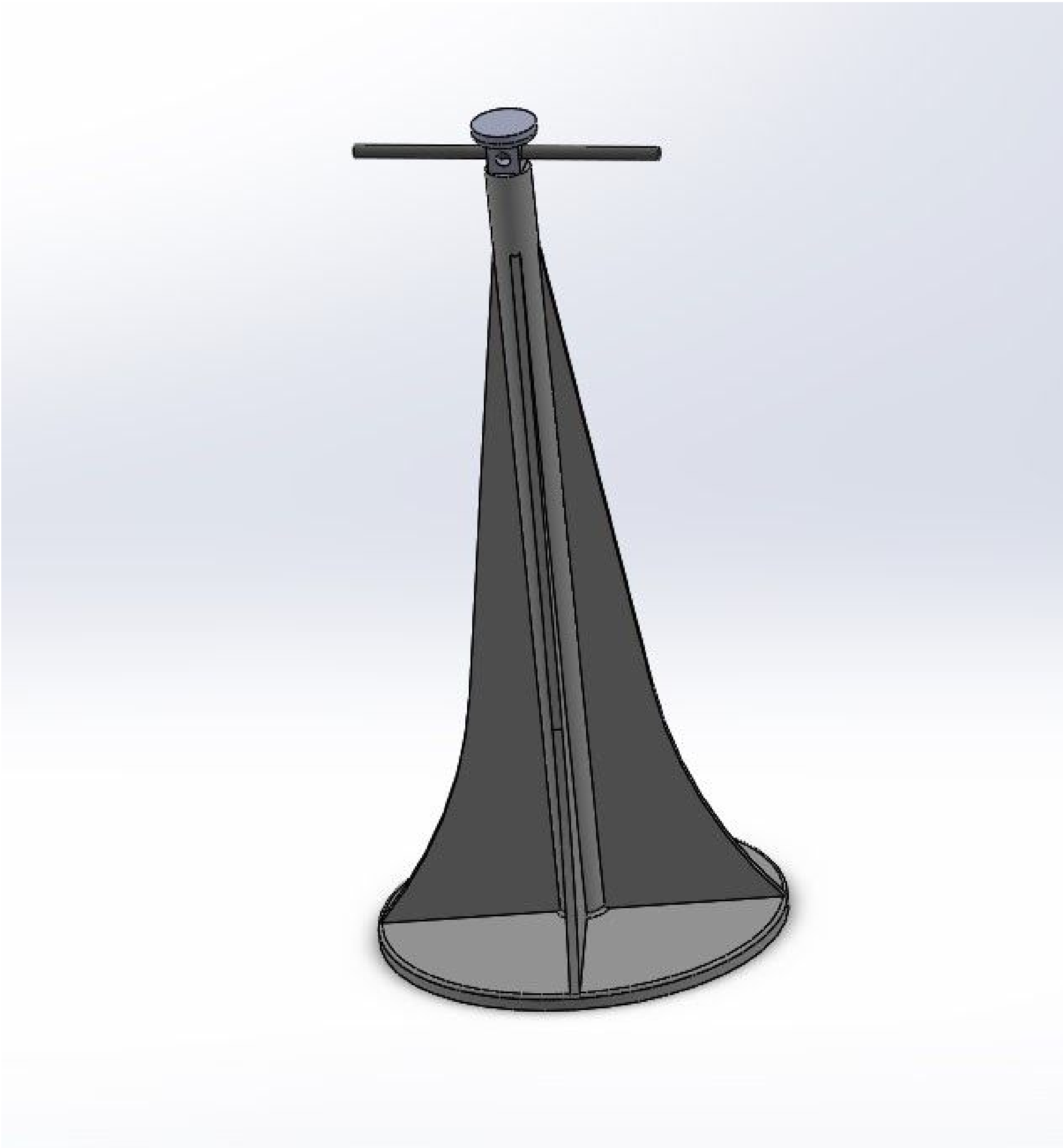

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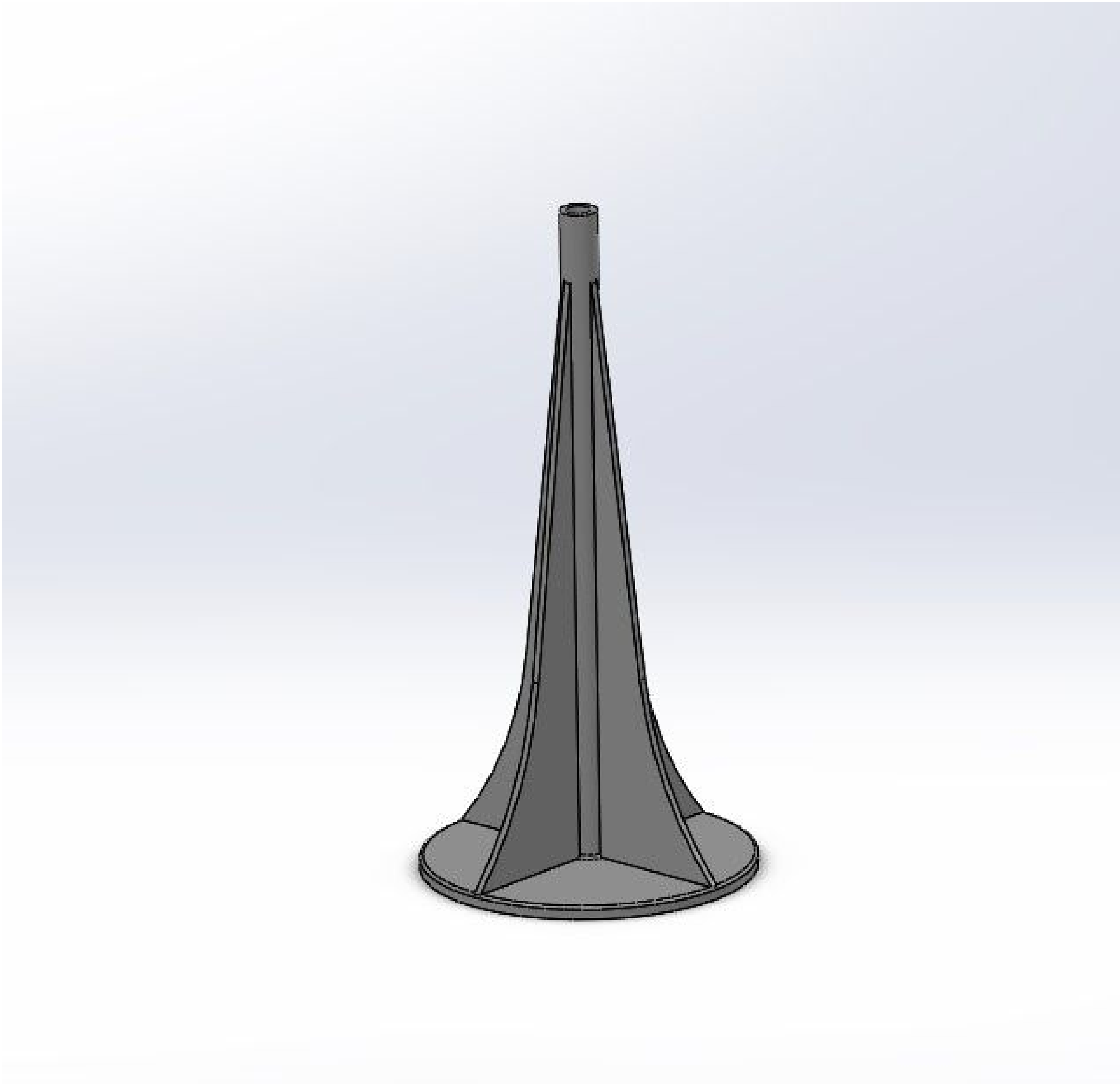

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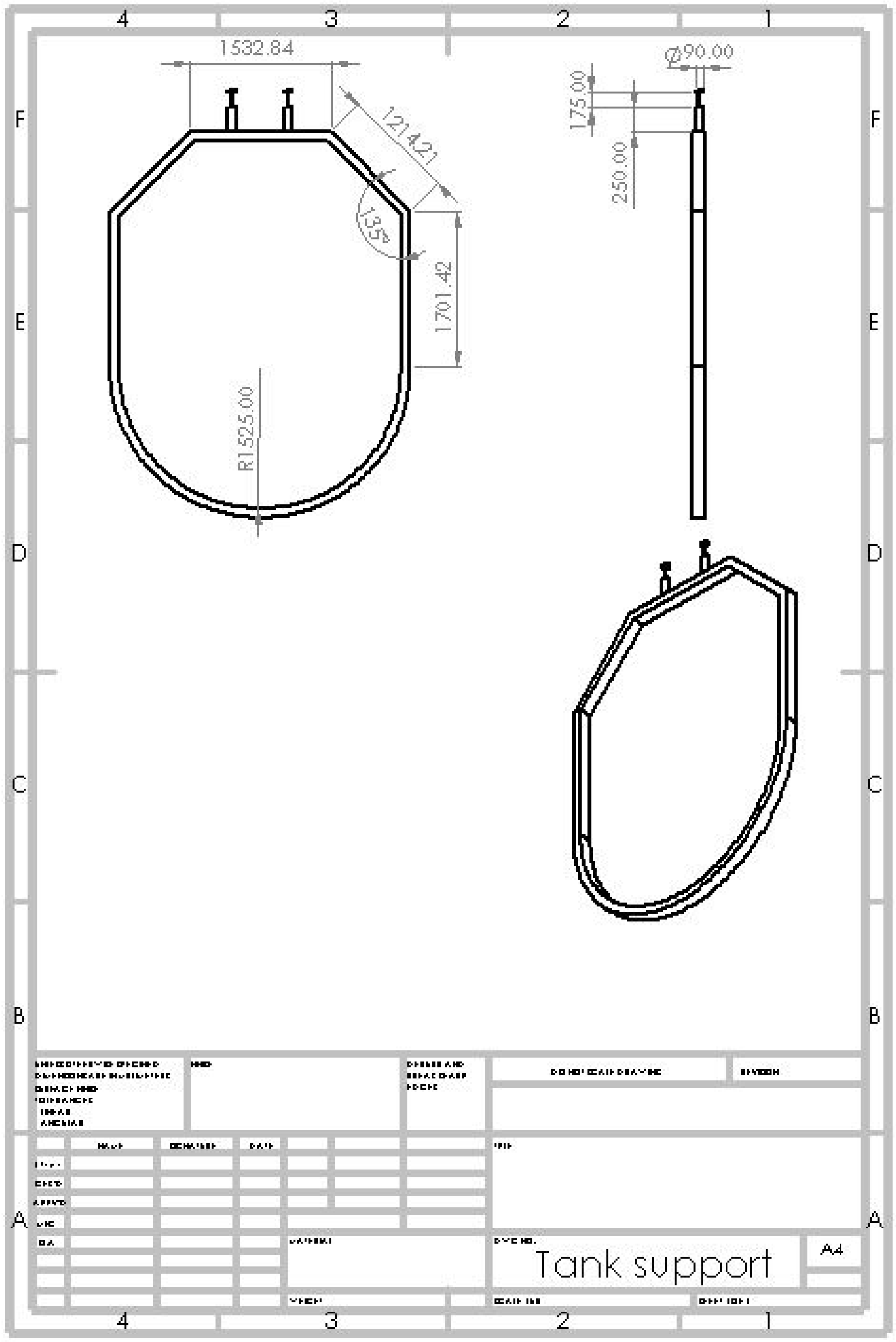

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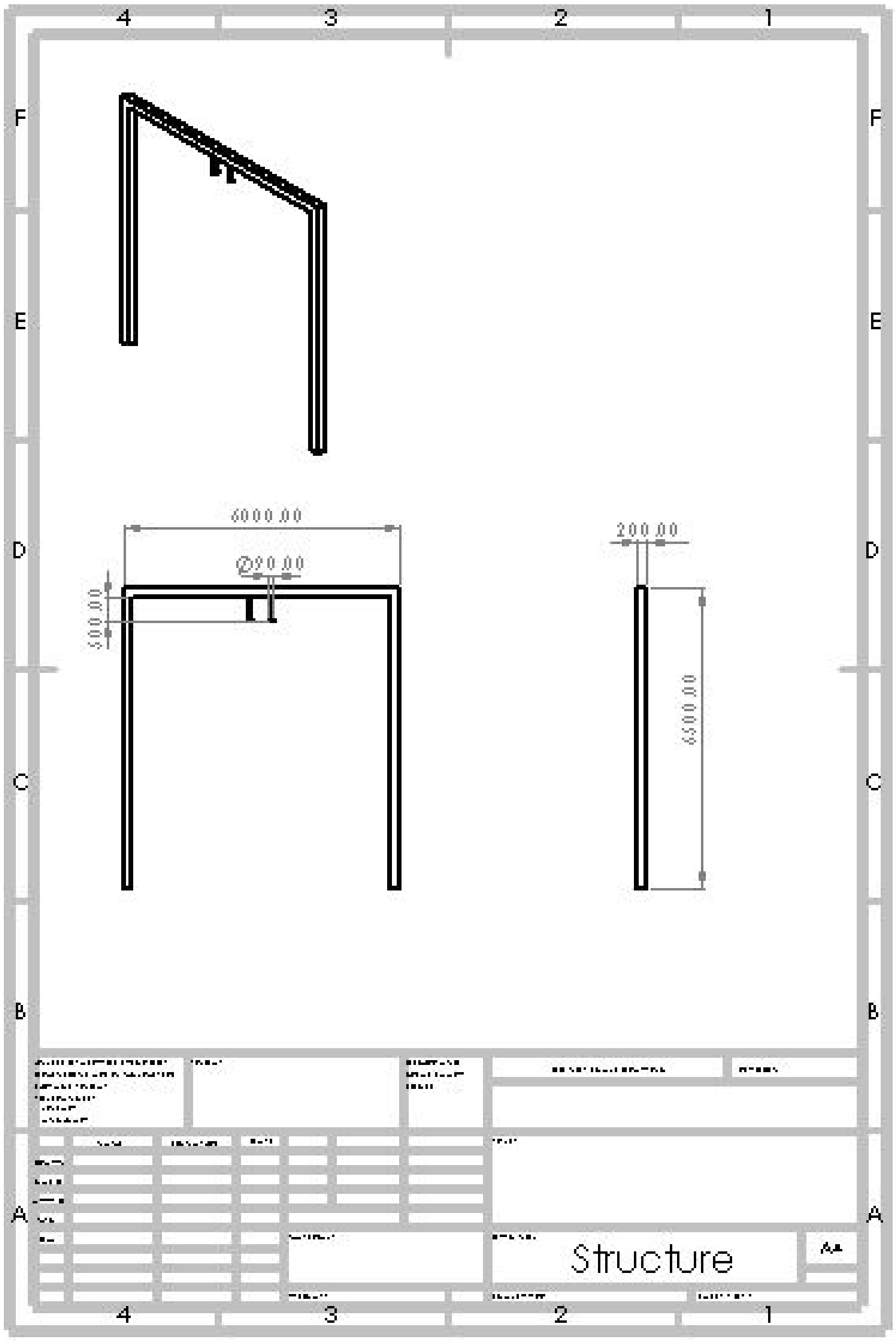

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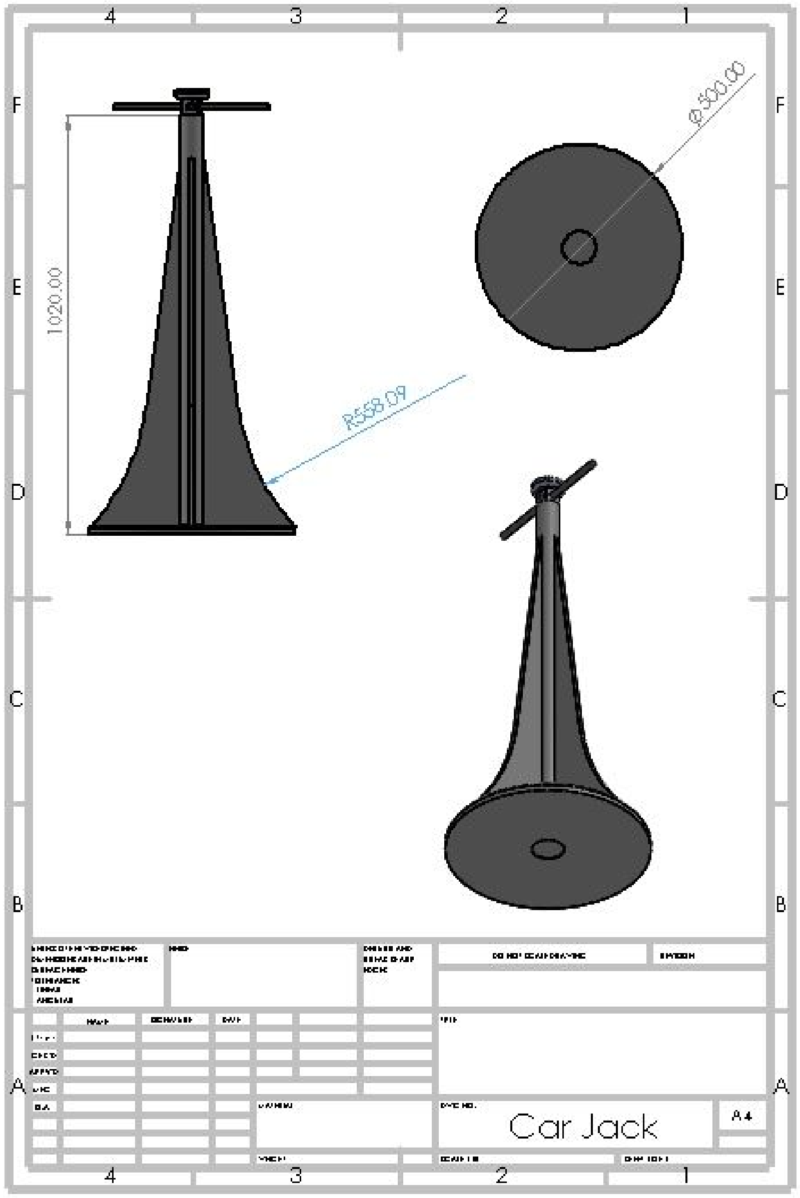

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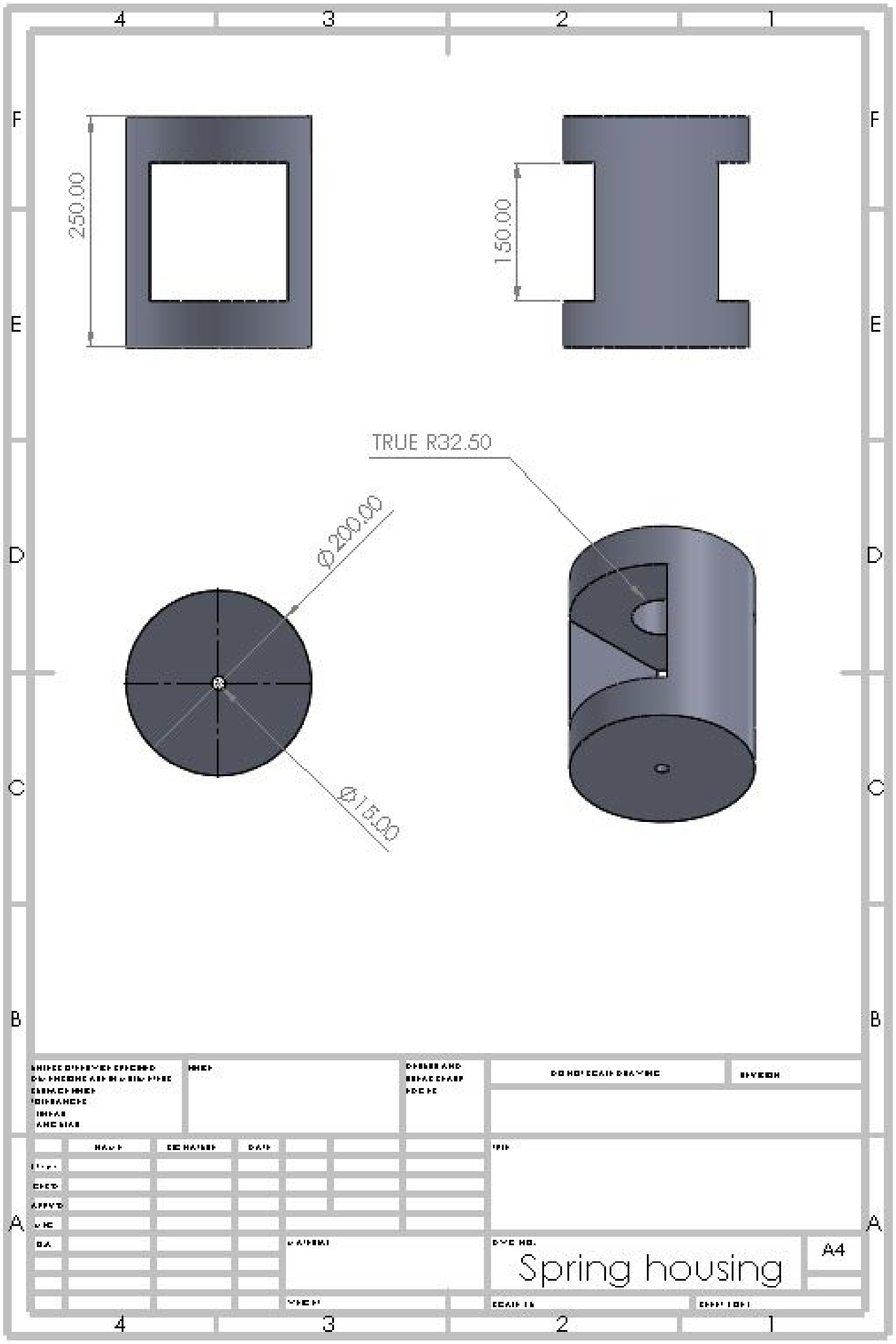

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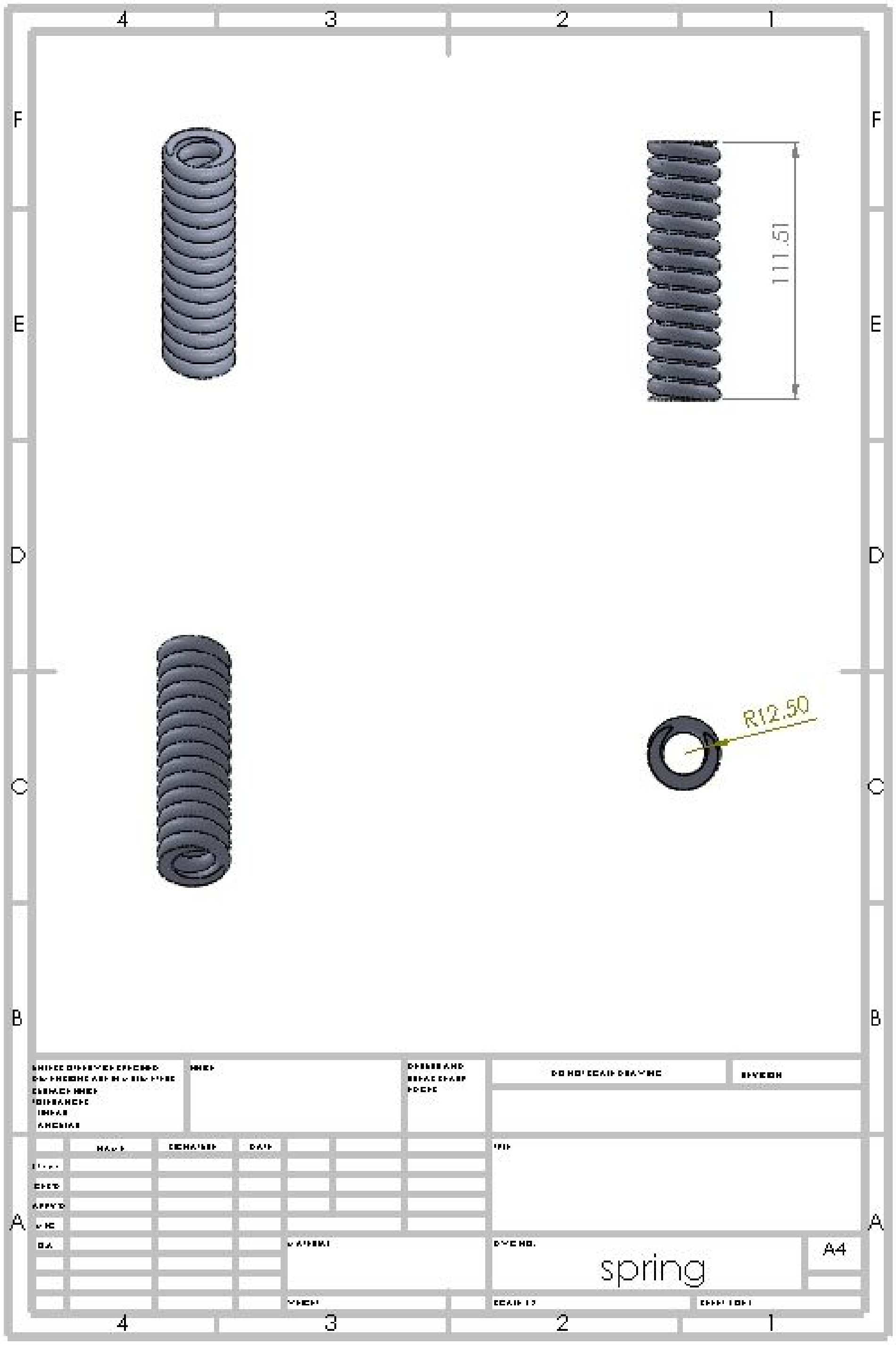

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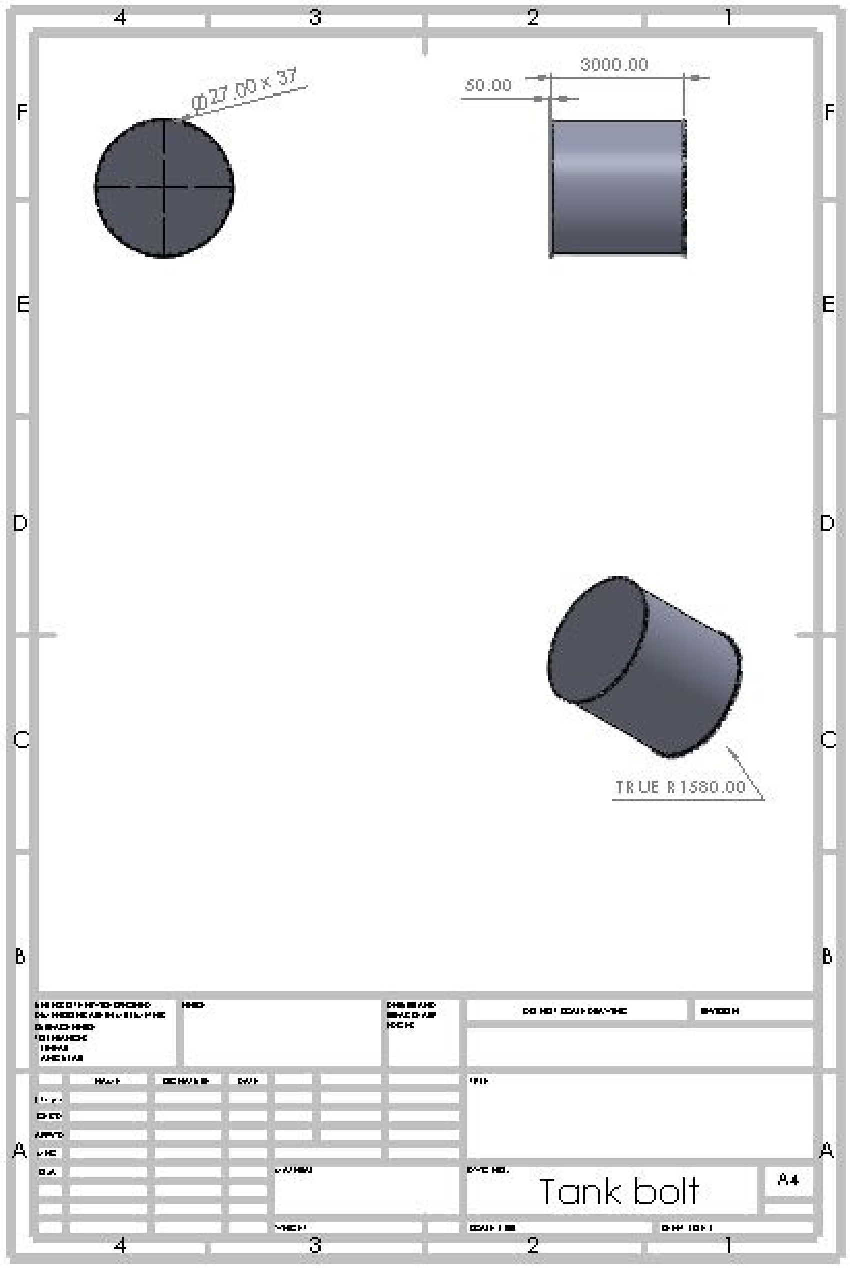

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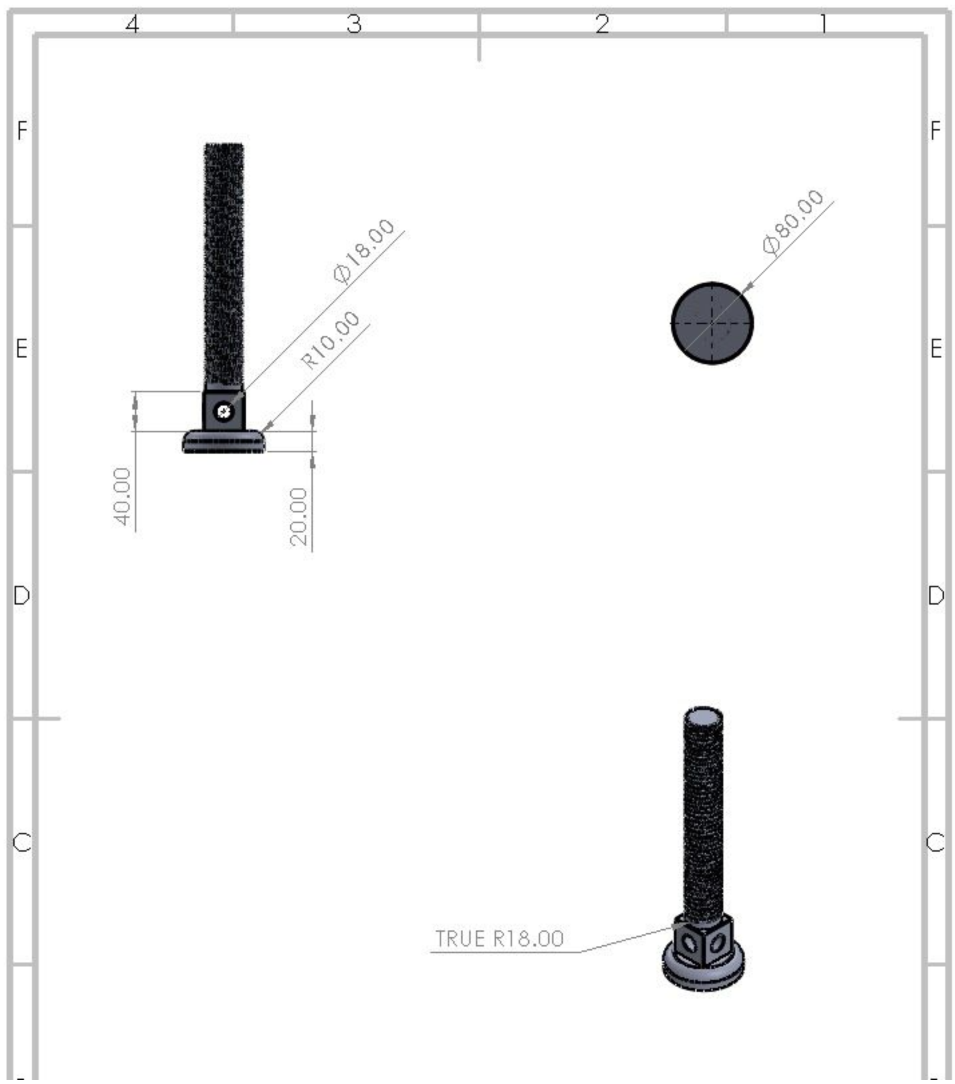

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3.0 SPRING ANALYSIS

i. Mean diameter of the spring
D = 25 mm

ii. Number of turns of the coils
n = 15

iii. Free length of the spring
L_F = 111.51 mm

iv. Pitch of the coil
7.97 mm

m = 200 kg, G = 80 kN/mm^2

Sample calculation
Assuming **D = 25 mm**, d = 6.401 mm @ SWG 3, **n = 15**,

$$W \ or \ F = \frac{mg}{2}$$
$$= \frac{200(9.81)}{2}$$
$$= 981 \text{ N}$$

$$C = \frac{D}{d}$$
$$= \frac{25}{6.401}$$
$$= 3.906$$

$$K = \frac{4C-1}{4C-4} + \frac{0.615}{C}$$
$$= \frac{4(3.906)-1}{4(3.906)-4} + \frac{0.615}{3.906}$$
$$= 1.416$$

$$= (\frac{8FD}{\pi d^3})K$$
$$= (\frac{8mgD}{2\pi d^3})K$$
$$= (\frac{8(981)(25)}{\pi(6.401)^3})(1.416)$$
$$= 337.186 \text{ MPa}$$

The assumption is valid since the permissible shear stress does not exceed the recommended maximum permissible shear stress of spring wire.
337.186 MPa < 400 Mpa

$$\delta = \frac{8WD^3n}{Gd^4}$$

$$= \frac{8(981)(25)^3(15)}{(80000)(6.401)^4}$$

$$= 13.7 \text{ mm}$$

The deflection also does not exceed the maximum deflection of the spring base of 20 mm.

Assuming plain end type,

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

$$=$$

$$= \mathbf{111.51\ mm}$$

$$Pitch,\ p = \frac{L_F}{'-1}$$

$$= \frac{111.51}{15-1}$$

$$= \mathbf{7.97\ mm}$$

Table 1 Comparison between different types of end

Type of End	n’	L_S	L_F	p
Plain	15	102.416	111.51	7.97
Ground	15	96.015	111.51	7.97
Squared	17	115.218	124.31	7.77
Squared and ground	17	108.817	124.31	7.77

Explanation

To support and lift the load of the high-pressure cylindrical tank of weight 200kg, our group decided to go with 2 spring hangers, the 1st reason is so that the load applied on the springs will be divided into 2 where the load is 981N per spring. This will significantly reduce the stress on each spring and help the spring not to exceed the maximum permissible shear stress of spring wire which is given as 400 MPa.

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 12 Choosing the SWG for spring wire and the corresponding wire diameter

After many trials, we decided to pick spring wire with SWG 3 where the wire diameter is d = 6.401mm and decided to have n = 15 number of turns. This is to adhere to the formula to calculate deflection in spring where a higher spring wire diameter, d and a relatively lower number of turns, n will give us a lower deflection in spring, x or δ . Referring to the formula, F = k x, a lower deflection will allow the spring stiffness to be higher or stiffer. This will allow the spring to withstand a greater amount of force acting on it on top of the initial mass of 200kg.

As a common engineering practice, we initially set the mean diameter as D = 25mm. While trying to calculate D more than 25mm, we found out that this will cause an increase in deflection of spring and hence reduce spring constant, k and the overall capability of spring to withstand the load where F will decrease. The spring type that we recommend is the ground ends spring where the solid length is calculated to be 96.015mm and the free length is calculated to be 111.51mm. A smaller solid length indicates that the spring is able to compress more, enabling the spring to compress to a minimum of 96.015mm. On the other hand, a shorter free length would prevent the tank from touching the ground. The pitch is then calculated from the free length obtained and we calculated the pitch as 7.97mm. We also think that the ground end spring is ideal because the spring will be more stable as it has flat ends compared to the plain ends which may cause imbalance due to its protruding ends.

In conclusion, with spring wire **SWG 3** where the wire diameter is **d = 6.401mm**, the mean diameter, **D = 25mm**, number of turns of the coils, **n = 15**, free length of the spring is $\mathbf{L_f}$ **= 111.51mm**, pitch, **p = 7.97mm**.

4.0 RIVET ANALYSIS

i. Number of rivets required to secure the tank. n = 93 **ii. Minimum size of flange, 2A (related to size of the rivet chosen)** 2A = 60 mm

τ = 120 N/mm^2
P = 1 MPa

Calculation

$$n = \left(\frac{D}{d}\right)^2 \frac{P}{}$$

$$= \left(\frac{3000}{28.5}\right)^2 \left(\frac{1}{120}\right)$$

= 92.34

≈ **93 rivets**

Assuming flange **2A = 60 mm** and A = 30 mm,
Circumference = 2 π r
= 2 (π) (1500 + 20 + [30])
= 9738.94mm

Clearance = $\frac{Circumference}{n}$

$$= \left(\frac{9738.94}{93}\right)$$

= 104.720mm

Explanation

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 13 Choosing the rivet hole diameter and the corresponding rivet size

To determine the number of rivets that is needed to hold the cap in place, we need to set the diameter of the rivet, D and the rivet hole, d. Since the thickness of the cap which is about the thickness of the tank, t = 20mm, we calculated the appropriate diameter of rivet hole using the formula which amounts to about 26.8mm. Taking the next best diameter rivet hole based on the formula sheet is d = 28.5mm where the corresponding rivet size is 27mm. With the diameter of rivet hole set, the number of rivets calculated amounts up to 92.34 or n = 93 rivets.

The next step is to calculate the clearance of the gap between rivets. In order to do that, the radius is calculated by adding the radius of the tank (1500mm) , the thickness of the tank (20mm) and the value of A which is the distance from the outer wall of the tank to the middle of the flange. Since A is not given, we had to run some trials to get an appropriate size. After setting the A as 30mm, the total radius is summed up to 1550mm and the circumference calculated is 9738.94mm. The clearance amounts up to 104.720mm gap between rivets after dividing the circumference (9738.94mm) with the number of bolts required, n = 93. If there is an increase in tank pressure, more number of rivets can be added without hesitation since the clearance gap between rivets is a lot.

With that in mind, since the clearance gap is definitely bigger than the size of rivet hole, therefore the value of d = 28.5mm is valid. Besides that, the size of flange 2A is also bigger than the size of the rivet hole, resulting in a more secure design. In conclusion, with the appropriate diameter of rivet 27mm with the corresponding rivet hole size, **d = 28.5mm**, the number of rivets required to secure the tank, **n = 93** and the minimum size of flange, **2A = 60mm**.

5.0 BOLT ANALYSIS

i. The SAE class steel to be used and number of bolts required to secure the tank
SAE class 8.8, n = 39 bolts

ii. Minimum size of flange, 2A (related to size of the bolt chosen)
2A = 60 mm

$\tau = 120\ N/mm^2$
P = 1 MPa
d = 27 mm
D = 3 m
A = 459 mm^2 (taking the nominal diameter of 27mm)
SF = 1.5

Sample calculation

$$F_{tank} = P \times A$$
$$= (1 \text{x } 10^6) \text{ x } (\pi \text{ x } 1.5^2)$$
$$= 7.07 \text{ x } 10^6 \text{ N}$$

Assuming the SAE class of 8.8 and nominal diameter of 27 mm:

$$F_{bolt} = S_P \times A_t$$
$$= (600 \text{ x } 10^6) \text{ x } (459 \text{ x } 10^{-6})$$
$$= 2.754 \text{ x } 10^5 \text{ N}$$

Assuming a safety factor of 1.5,

$$F_{bolt} = \frac{2.754 \times 10^5}{1.5}$$
$$= 1.836 \text{ x } 10^5$$

$$n = \frac{F_{tank}}{F_{bolt}}$$
$$= \frac{7.07 \text{ x } 10^6}{1.836 \text{ x } 10^5}$$
$$= 38.51$$

$\approx$ **39 number of bolts**

Radius of the cap = 1.5 + 0.02 + 0.03
= 1.55 m

Circumference of the cap = $2\pi r$

$= 2 \times \pi \times 1.55$

$= 9.739$ m

Clearance = $\frac{circumference\ of\ the\ cap}{number\ of\ bolts}$

$= \frac{9.739}{39}$

= **249.7 mm**

Explanation

Coarse thread was chosen instead of fine thread because it is considered to be a more common practice as it is easier to obtain compared to fine thread bolts. Fine thread bolts are more prone to damage due to its small pitch. The reason being is that the fine thread bolts will generate excess friction because it requires more turns per inch which will eventually gall the fasteners.

Based on the standard table of bolts specification as shown below, SAE class 8.8 bolt was chosen where it has a proof load strength of 600 MPa. The grade 8.8 bolts are manufactured using medium carbon steel which displays a good yield and tensile strength. These grade 8.8 bolts are also commonly found commercially and it performs very well in most environments. Although there are higher grades of bolts, the grade 8.8 bolts were still chosen as it is more affordable and is capable enough to withstand the high pressure in the tank. On the other hand, the SAE class of bolts below 8.8 was not chosen because it has a low proof load strength and it might not be able to withstand the required pressure in the tank.

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

Figure 14 Obtaining proof load strength for chosen SAE class

Based on the ISO metric table as shown below, the nominal diameter of 27 mm with a stress area of 459 mm^2 was chosen because it is closest to the diameter of rivet hole of 28.5 mm. In this case, a fair comparison can be made between the rivet and bolt. Despite the fact that the 30 mm was also the next closest diameter to the rivet hole diameter, it was not picked because it will increase the clearance/gap between the bolts. If the clearance between the bolts is too big, fracture on the certain part of the bolts might happen in the long run as there will be more stress acting on the bolts.

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

Figure 15 Obtaining stress area for the chosen nominal diameter

Therefore, it can be concluded that the cap of the cylindrical tank will be more suitable to be fabricated by bolting than riveting. This is because the number of bolts required is much lesser than the number of rivets required to withstand the amount of pressure in the tank. Based on the calculations, only 39 bolts are required to withstand the amount of tank pressure with a clearance of 249.7 mm between the bolts. With a good amount of gap, the number of bolts can also be increased without exceeding the clearance to provide extra strength for the cap. In a way, this also helps to reduce the cost of building it with a lesser number of fasteners required to be purchased.

6.0 POWER SCREW ANALYSIS

i. Determine the thread depth and helix angle
d = 3 mm, λ = 6.6°
ii. Estimate the starting torque for raising and for lowering the load
$T_{raising}$ = 23020.25 N.mm, $T_{lowering}$ = 15223.03 N.mm
iii. Estimate the efficiency of the jack when raising the load:
e = 16.28 %

p = 6 mm

$d = \frac{p}{2}$

$= 3$ mm

d_{major} = 36 mm

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

$= 36 - 3$

$= 33$ mm

For starting torque,

$f_c = \frac{4}{3} \times f_c$

$= \frac{4}{3} \times 0.12$

$= 0.16$

$f = \frac{4}{3} \times f$

$= \frac{4}{3} \times 0.15$

$= 0.2$

Since the square thread is the recommended screw type, the differences between single and double thread were compared. While the thread angle α_n for both are equal to 0, the lead L and helix angle λ are dissimilar as demonstrated below:

Table 2 Comparison of helix angle

Square Thread	
Single	Double
$tan\,\lambda = \frac{L}{\pi d_m} = \frac{p}{\pi d_m} = \frac{6}{\pi(33)}$ $\lambda = 3.3°$	$tan\,\lambda = \frac{L}{\pi d_m} = \frac{2p}{\pi d_m} = \frac{12}{\pi(33)}$ $\lambda = 6.6°$

Next, the torque required to lift the load in both cases were calculated using all constant variables except for L. This is necessary to calculate the efficiency of both thread types which are summarised below:

Table 3 Comparison of efficiency

	Single thread	Double thread
$T_{raising}$ (N.mm)	21002.47	23020.25
Efficiency (%)	8.91	16.28

Sample calculation

Assuming d_c = 80 and double thread conditions,

$\tan \alpha_n = \tan \alpha \cos \lambda$

$\alpha_n = \tan^{-1}(\tan 0 \cos 6.6) = 0°$

Raising the load (Starting)

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

$$= \frac{200 \times 9.81 \times 33}{2} \frac{(0.2 \times \pi \times 33) + 12 \cos 0°}{(\pi \times 33 \times \cos 0°) - (0.2 \times 6)} + \frac{200 \times 9.81 \times 0.16 \times 80}{2}$$

= 23020.25 N.mm

Lowering the load (Starting)

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

$$= \frac{200 \times 9.81 \times 33}{2} \frac{(0.2 \times \pi \times 33) - 12 \cos 0°}{(\pi \times 33 \times \cos 0°) + (0.2 \times 6)} + \frac{200 \times 9.81 \times 0.16 \times 80}{2}$$

= 15223.03 N.mm

Efficiency of the jack when raising the load

Raising the load (Running)

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

$$= \frac{200 \times 9.81 \times 33}{2} \frac{(0.15 \times \pi \times 33) + 12 \cos 0°}{(\pi \times 33 \times \cos 0°) - (0.15 \times 6)} + \frac{200 \times 9.81 \times 0.12 \times 80}{2}$$

$$= 18095.97 \text{ N.mm}$$

$$e = \frac{WL}{2\pi T_{raising}} \text{ x } 100\%$$

$$= \frac{200 \times 9.81 \times 12}{2 \times \pi \times 18095.97} \text{ x } 100\%$$

$$= 20.71 \ \%$$

Explanation

With a purpose of comparing both single and double square threads, a constant value of d_c was set. In the case where all other variables are constant, the increase in d_c will it increase the torque when raising the load and consequently decreases the efficiency of raising the load.

After calculating the efficiency of the jack when raising the load for both single and double thread, it was found that double thread has a better efficiency. Therefore, double and square thread screw was chosen. With better efficiency, the time required for the jack to raise the load will be lesser and this will be helpful when performing any daily operational or maintenance work. In a way, the productivity rate of the workers will increase.

7.0 CONCLUSION AND RECOMMENDATIONS

In conclusion, the design of the whole system can be divided into 2 parts. The first design used springs, rivets and power screw while the other design used the same components except for bolts instead of rivets. It can be seen from the analysis conducted that lesser amount of bolts were required to secure and withstand the same total load of the whole tank system as compared to rivets. Hence, this makes the design using bolts the more practical and optimum option. Lesser amount of bolts used will sum up to a lesser total cost. Thus, making this design more cost efficient.

In terms of the constant components (spring & power screw), the analysis carried out by the team was in depth and strongly justifies the team's selected specifications. For the spring, the team decided to use 2 springs as this will divide the amount of force experienced by each spring. Thus, decreasing tension. Ground ends spring was selected as the type of spring to be used based on the calculations done. This type of spring is also more commonly used and inexpensive in price.

As for power screw, a square thread was suggested to be used by the module coordinator. Therefore, the team calculated whether single or double square threads were to be used. Based on the analysis completed, double square threads were selected as the power screw design. This is because double thread screws have generally higher efficiency in addition to being able to secure objects firmly.

As for future recommendations, the team decided that certain aspects of the design could be improved on. Firstly, more amount of springs could be used instead of just 2. This will further decrease the amount of stress being applied on each spring. To overcome the cost of more springs, the team could use springs that are of different S.W.G value or smaller diameter springs as this will be a cheaper alternative and overcomes the cost issue. Besides that, other types of springs such as squared and ground ends could also be used instead as they may be able to withstand more tension per spring as compared to ground ends. Thus, lesser amount of springs required.

Next, higher quality rivets could be used instead of bolts. This will decrease the amount of rivets required and further decrease the total price of the design. This is because rivets are cheaper than bolts in general. Furthermore, different SAE Class bolts that have higher proof strength could also be used to secure the cover of the tank. This will decrease the amount of bolts needed as each bolt can withstand a higher pressure. Thus, saving cost. In addition to these, a different type of power can also be used to raise and lower the tank. For example, acme or modified square head screws. These screws have a higher efficiency than the square head screw. They also secure objects more firmly. Hence, a higher efficiency when lowering and raising the tank will result in greater efficiency of workers.