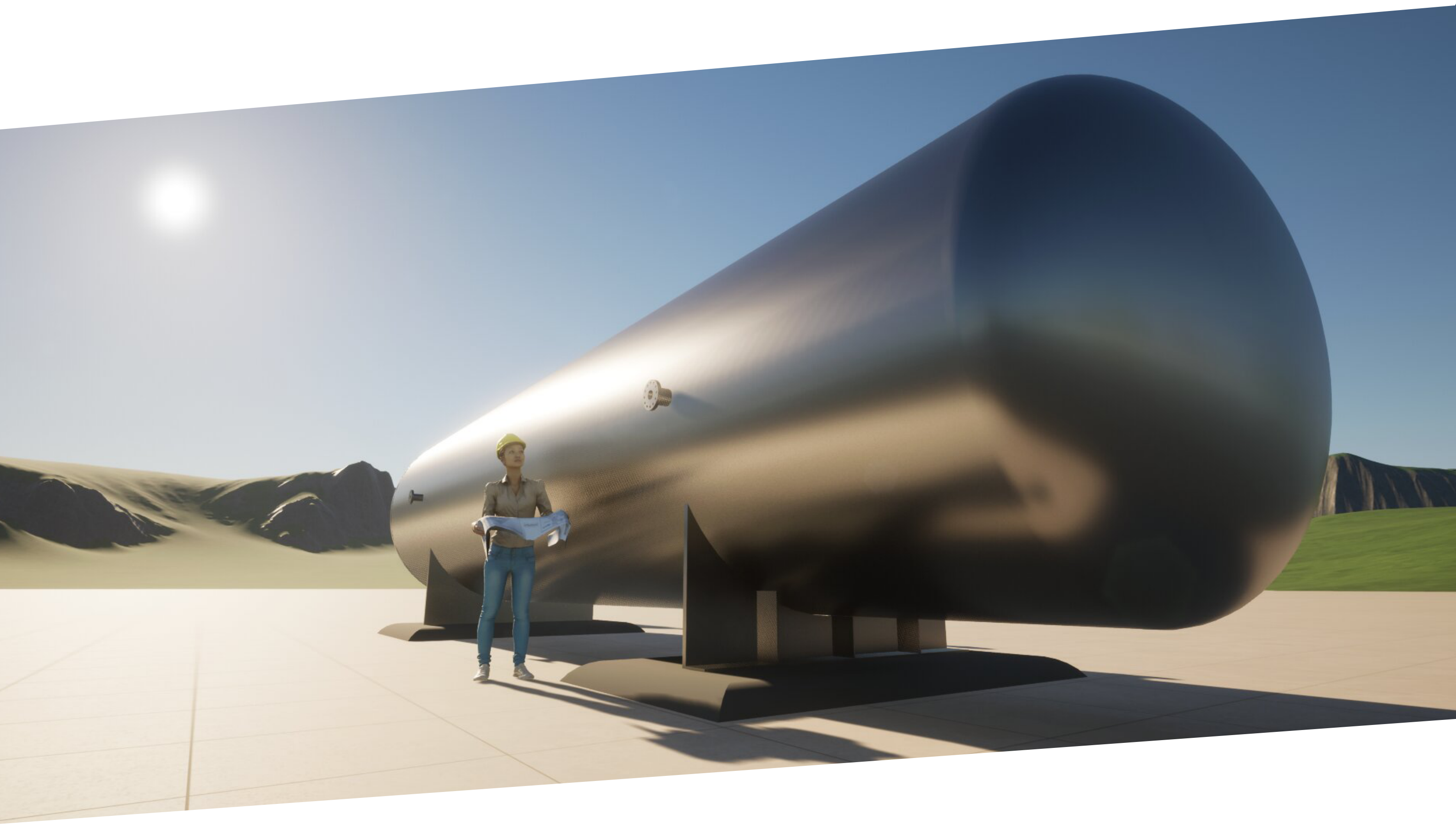
Design and Construction of Equipment

Part one: Design specifications and drawings of Methane Storage Tanks

Almaz Khalilov



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Executive Summary

This report outlines the specifications, drawings and calculations for the T2 pressure vessel tank farm pertaining to the storage of methane from the waste stream of the design problem.

The first part of the report focused on defining the project and scope, and discussion of the standards that needed to be adhered to. We drew a process flow diagram of the entire process, and understood the problem in its entirety.

The overall daily flow rate of methane to the storage tanks was calculated, and from it, the volume of methane that was required to be stored daily was 1185.12 kmol, this assumed that the plant was operating 24 hours per day.

A volume requirement of 1424.78 m^3 was calculated via the ideal gas law and the compressibility factor, and was checked for correctness with Hysis.

Three main guidelines were followed to size each individual tank, where the head diameter needed to be less than 3 meters, the aspect ratio was constrained to 4, and the tanks were horizonal. The heads were decided to be ellipsoidal due to their superior strength and cost effectiveness, and hence the diameter of the vessel was chosen to be the same as the head diameter. We consulted the brochure of Australian Pressure vessel heads PTY, and arrived at a head diameter of 2731mm. We arrived at a final vessel volume of 69.33 m^3, and 21 vessels were needed. One was added in case of delays.

The design pressure and temperature were chosen based on the conditions. Our surge pressure was 2200Kpa, and hence a design pressure of 2420Kpa was chosen complying with a 10% allowance, as standard. A design temperature of 80 degrees C was chosen, based on the maximum recorded temperature in Victoria, plus an allowance.

The material chosen was 25mm thickness AS 1548-7-460 Steel, for both the vessel heads and body, as it was the most cost effective, was the standard material of construction for pressure vessel heads, and hence readily available, and its thickness was compliant with a Class 2B construction.

Next, saddle supports were chosen, as they complied with standards and had superior strength. In terms of the nozzles, three nozzles for each tank were added. One was a manhole, to comply with standards, one was the input of the methane gas, which had DN 50, and an output nozzle of DN65. A nozzle of DN20 was also added for the pressure relief valve, which is to relief pressure at 105% the surge temperature. Specifications also set out to perform a hydrostatic test at 3868 kpa, to ensure the integrity of the vessel and to comply with standards.

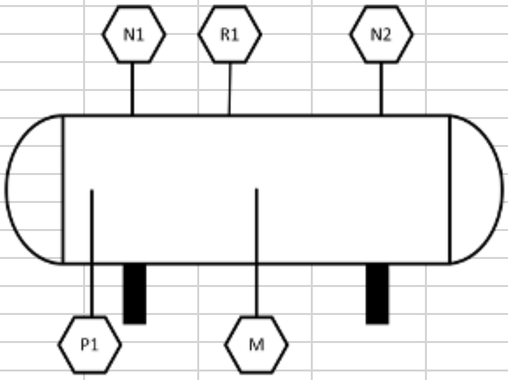
Lastly, the report highlights managerial and safety aspects such as to conduct risk assessment, provide regular safety training for personnel, and to implement maintenance and inspection schedules, maintaining accurate records of maintenance activities and repairs, and developing clear operational procedures. These measures will help ensure the safe and efficient operation of pressure vessels.

Equipment specification sheet

	DOCUMENT NO.	VESSEL PROCESS DATA SHEET					SHT : 1
	1	PROJECT :	Part 1			LOCATION : VICTORIA	
	Pressure VESSEL						
		20/8/2023	For Inq.	0	RG	DA	KM
		DATE	ISSUE	REVISION	PREP	CHKD	APPD
EQUIPMENT NAME :	Methane Storage Vessel					QTY	22
SUPPLIER	Custom		TYPE :	Horizontal, Ellipsoidal Heads			

OPERATING / MECHANICAL DATA

FLUID		Methane	Unit
Liquid :	Flow	0	kg/h
	Density	-	kg/m3
Vapor:	Flow	792.06	kg/h
	Mol. Weight	16.04	g/mol
	Density	-	kg/m3
Operating Temperature		25	°C
Operating Pressure		2000	Kpa
Design Temperature		80	°C
Design Pressure		2420	Kpa
Hydrotest Pressure		3868.03	Kpa
Total Volume		69.32	m3
Working Volume		69.32	m3
Shell ID		2731	mm
Length, TL-TL		12.2895	m
Jacket ID		-	mm
Jacket Height from Bottom TL		-	mm
Corrosion Allowance		2	mm
Insulation Thikness		0	mm
Shell Thickness		25	mm
Dish Thickness **(Left / Right)**		25 / 25	mm
Jacket Shell Thk		-	mm
Empty Wt.		24.0659	tonnes
Number of supports		2	
Type of support		Saddle	
Size of support		250	mm
Orientation		horizontal	
Demister		Not Reqd.	
Construction Code		AS1210	
Stress Relieving		Acc. to code	
Joint Efficiency		80%	
Radiography		Nil	
Remarks:			



MATERIAL OF CONSTRUCTION

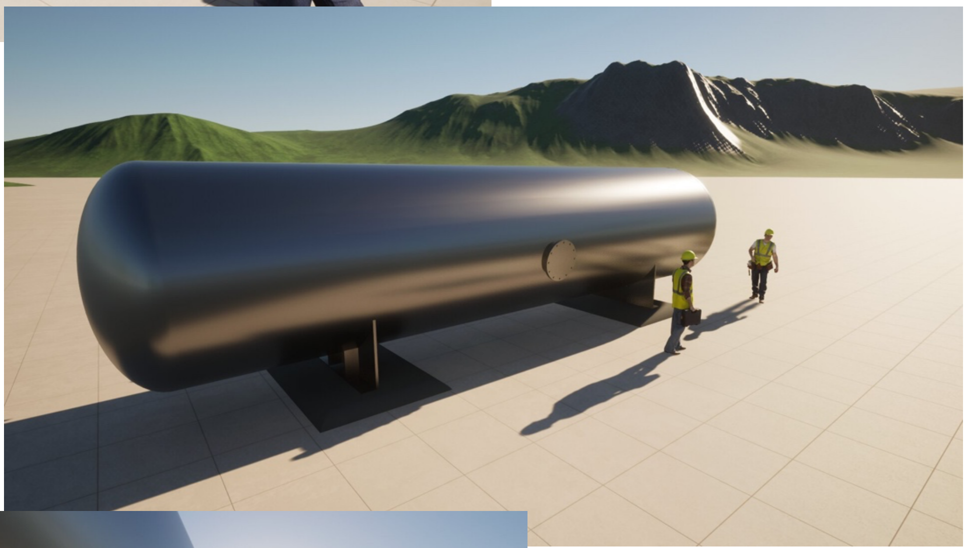
Main Shell	AS 1548-7-460 Steel
Dish Ends	AS 1548-7-460 Steel
Nozzle Flanges	AS 1548-7-460 Steel
Nozzle Pipes	AS 1548-7-460 Steel
Support	AS 1548-7-460 Steel
Nut & Bolts	AS 1548-7-460 Steel
Gasket	TBD

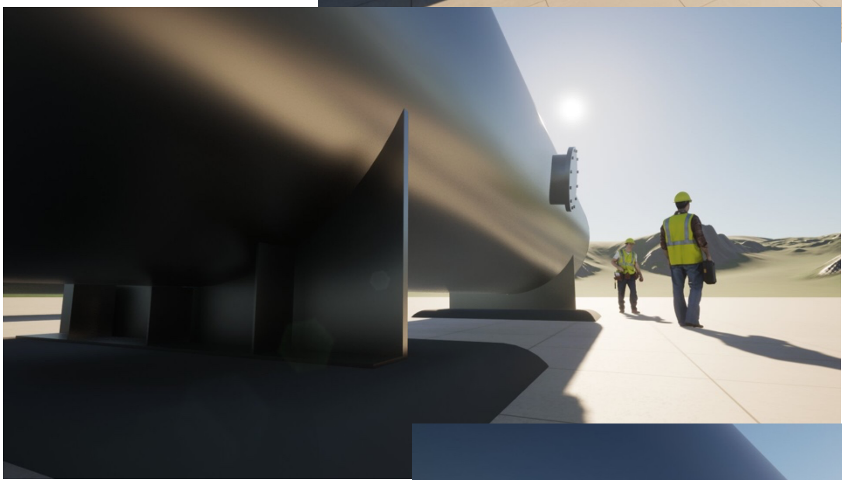
NOZZLE SCHEDULE (ANSI B16.5, SCH 40,150#)

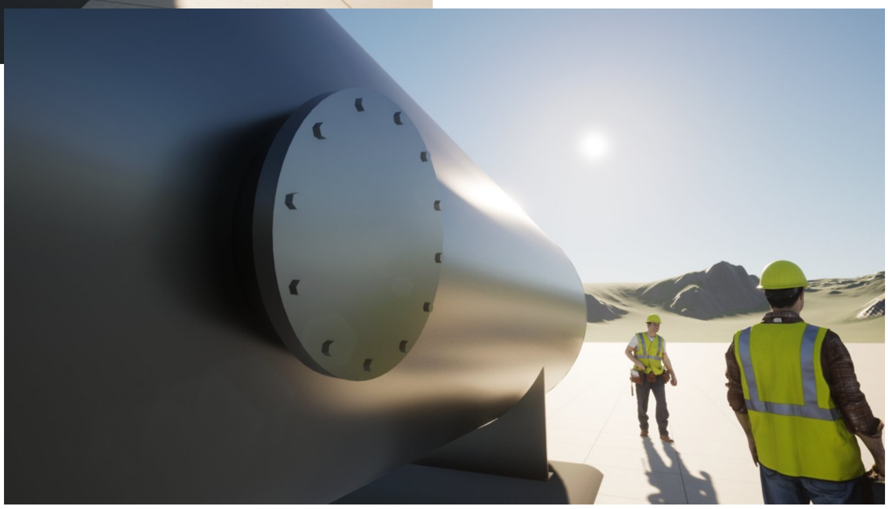
NOZZLE	SIZE (DM)	SERVICE	QTY
N1	50	Inlet	1
N2	65	Outlet	1
R1	20	Pressure Release Valve	1
P1	25	Pressure gauge	1
M	500	Manhole + Blind Flange	1

Equipment drawings 2D

Equipment Renders 3D







Calculations and Discussion

Defining Project and scope

Firstly, we need to define the project scope and what it entails. A Block flow diagram is drawn for the whole process to see the entire process.

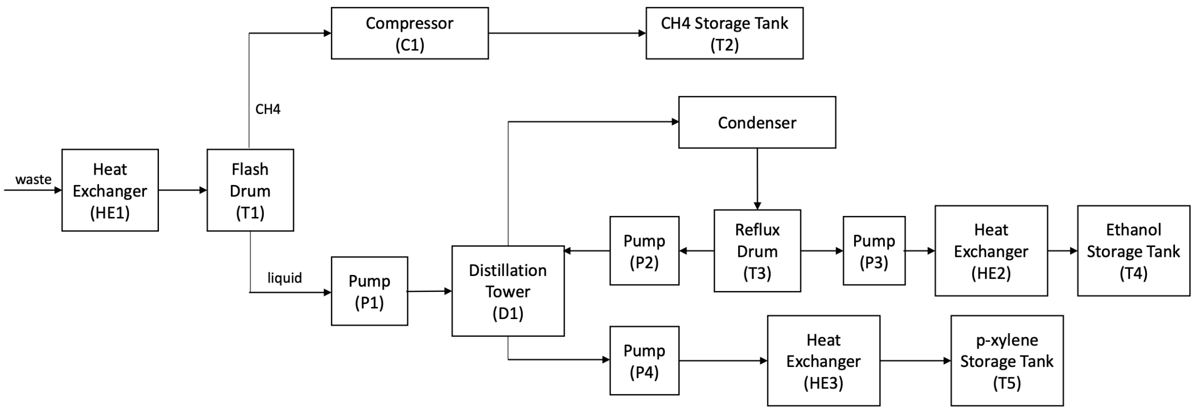


Figure 1 – Block flow diagram of the entire process (D Wang, 2023)

This report primarily concerns itself with the methane storage tank (T2) section, however the compressor (C1), can influence the design of the storage tank

An important assumption is that separation in the flash drum is complete, hence there is no methane in the liquid stream, and no p-xylene or ethanol in the gas stream.

As for the location of the plant, we will assume that the plant is located in rural Victoria, hence the methane pressure vessel will need to comply with AS1210

Calculations of Molar Flowrate

Firstly, we must start with the specifications as laid out by the design problem. We know that the waste stream flowrate is $22{,}000\ kg/hr$, which contains $18\ mol\%\ methane$, all of which leaves into the storage tanks.

We also know that the

- molar mass of methane is 16.04g/mol.
- Molar mass of ethanol is 46 g/mol
- Molar mass of p-xylene is 106 g/mol

To proceed, we use the equation to calculate the average molar mass of the mixture:

$$M_{\text{avg}} = n\backslash\%_1 \cdot M_1 + n\backslash\%_2 \cdot M_2 + n\backslash\%_3 \cdot M_3$$

Hence the average molar mass is $80.2\ kg/kmol$

Now, to determine the molar flowrate, we divide the mass flowrate by the average molar mass:

$$\dot{N} = \frac{\dot{M}}{M_{\text{avg}}}$$

Hence the molar flowrate of methane to the storage tanks is $49.38\ kmol/hr$

We know therefore that $49.38\ kmol/hr$, or $792.06\ kg/hr$ of methane is entering the storage tanks.

Determination of Volume Requirements

Firstly, to determine the volume requirement, we use our normal operating conditions of $2000\ kPa$ and $25\ degrees\ C$ to find the required volume of methane that needs to be stored per day.

We do not know our plant operation time. A facility operating only 8 hours a day requires vastly different storage requirements than one operation 24 hours. We will assume this facility operates non-stop, and only stopping rarely due to maintenance or holidays, hence our assume plant operates the full $24\ hours\ per\ day$.

We calculate the total volume requirements using the ideal gas law, with a compressibility factor Z, which we find to be 0.97 at the specified conditions.

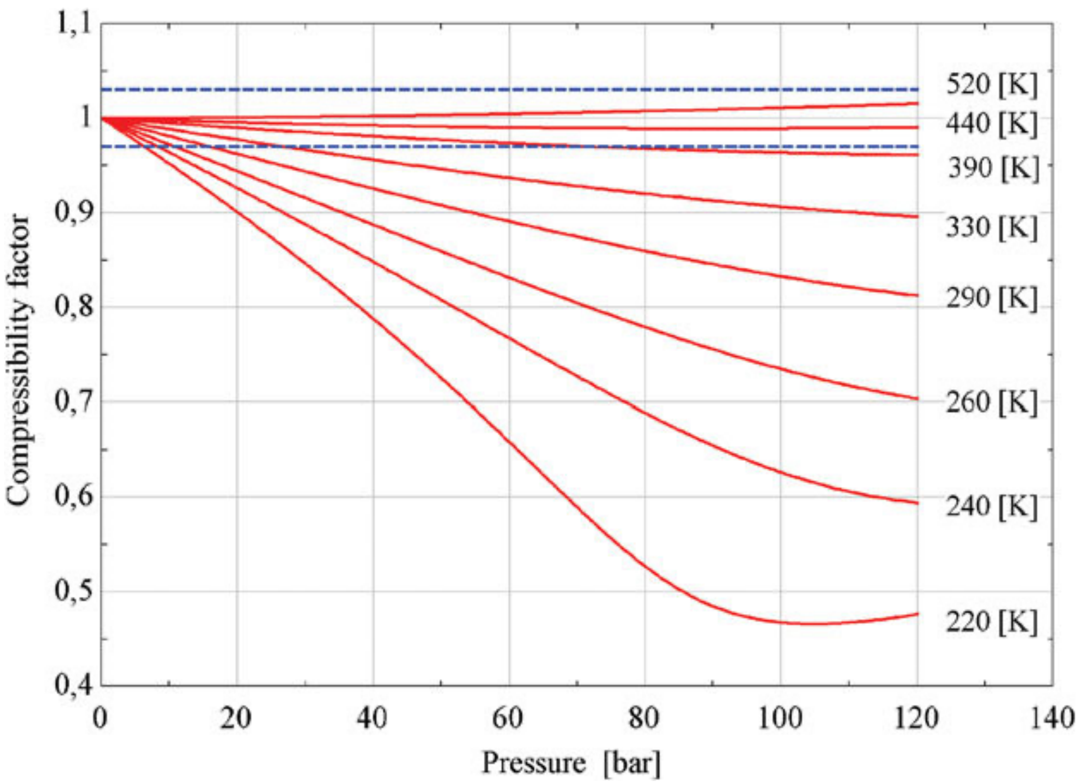


Figure 2 - Compressibility factor chart for methane (Almeida et al, 2014)

Hence calculating the volume required using the ideal gas law, we get a value of $1424.79m^3$.

The volume requirements are double checked using Hysis, using the Peng Robinson equation of state for accuracy.

Hysis outputs the volume requirements for the vapor phase below as 58.59m^3 per hour. Which is 1406.16 m^3 per day, which is very close to our calculations, and hence we can proceed.

Stream Name	2	Vapour Phase
Molecular Weight	16.04	16.04
Molar Density [kgmole/m3]	0.8428	0.8428
Mass Density [kg/m3]	13.52	13.52
Act. Volume Flow [m3/h]	58.59	58.59
Mass Enthalpy [kJ/kg]	-4691	-4691
Mass Entropy [kJ/kg-C]	9.840	9.840
Heat Capacity [kJ/kgmole-C]	38.04	38.04
Mass Heat Capacity [kJ/kg-C]	2.371	2.371

Figure 3 - Hysis output with Peng-Robinson fluid package

Calculation of pressure vessel volume

To calculate the pressure vessel volume, it is decided to first find the pressure vessel head diameter, and size the vessel according to that. Additionally, we will constrain our dimensions to comply with best practises so that:

1. The head diameter must be less than 3m
2. Aspect ratio is locked to be 4
3. Tank must be vertical for volumes less than 4000L, horizontal for more than 4000L

We consult Australian pressure heads PTY's brochure of standard options (APV, 2023), and we inspect the available standards. Although it is possible to manufacture a custom side and dimensions of the head, using standards that are made by manufacturers is cheaper, and quicker due to the fact that there is likely stockpile available.

We decide to use ellipsoidal heads, as they can withstand our required pressure, but are not as expensive as spherical heads to manufacture. We see that there are two options for ellipsoidal heads, hot pressed, and butt weld caps. The butt weld caps, do not have a large enough diameter to consider, hence we will consider only the hot-pressed heads.

Smaller tank diameters were considered at the start, as it is advantageous to have a smaller tank because less wall thickness is required, it may be able to comply with a lower scrutiny class (class 3), and transportation is easier, because oversize load permits are not required to transport vessels under 2.5m width in Victoria (Vicroads, 2023).

Transporting an extremely large vessel in requires permits, which increases both costs and time to transport vessels. In Victoria the maximum width to transport items without a permit is 2.5 meters wide, height of 4.3 meters high, and a length of 12.5 for a rigid truck. Our constraining factor is the limit of 2.5 meters wide.

Trying a head diameter of 2438mm, results in a requirement of 29 tanks

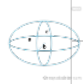
				Volume of Pressure Vessel				
D	Inside Diameter of pressure vessel	2438	mm		2.438	m		
r	Radius of pressure vessel	1.219	m					
A	Aspect ratio of pressure vessel	4					We calculate volume with a fixed ratio of 4	
h	Length of pressure vessel	9.752						$h = A * D$
V_body	Volume of main part of pressure vessel	45.52510714	m^3				Calculate using the volume for a cylinder	$V = \pi \cdot r^2 \cdot h$
	Ellipsoidal head ratio	2					Standard production values use a value of 2	
a	a value for ellipsoidal head	0.6095					ellipsoid lengths 	
b	b value for ellipsoidal head	1.219						
V_ellipsoid	Calculated Volume of full ellipsoid	3.793758928	m^3				Volume of both heads (full spherical ellipsoid)	$V = \frac{4}{3} \cdot \pi \cdot a \cdot b^2$
V_ellipsoid_supplier	Manual volume of elipse from supplier	3794	L		3.794	m^3	Manual lookup of head volume from supplier's datasheet. Used to double check calculations	
V_vessel	Votal volume of pressure vessel	49.31910714	m^3				Add the volumes of the full ellipsoid and body. Uses the ellipsoid volume from supplier, if not avaliable, use the calculated volume	
				Calculate the number of Pressure Vessels required				
Tanks	fractional number of storage tanks required	28.88907884	tanks					$Tanks = \frac{V}{V_{vessel}}$
Tanks_int	integer number of storage tanks required	29					rounded to highest integer value	

Figure 4 – Tank volume calculations for 2438 diameter tank

However, when trying a tank diameter of 2731mm, only 21 tanks a required, a staggering difference of 8 tanks.

It was decided that the benefits of smaller tanks do not outweigh the cost of extra tanks, that involve extra materials, supports, nozzles, and production costs. Therefore, it was decided to go with a tank diameter of 2731mm for both the body and the head of the tank.

We calculate the main body of the vessel using the standard equation of a cylinder:

$$V = \pi \cdot r^2 \cdot h$$

To get a volume of $63.99\ m$^3.

We calculate the volume of the ellipsoidal heads using the ellipsoid ratio of 2, which is the standard production ratio (APV, 2023). Using the a and b lengths of a circular ellipsoid. We calculate the volume to be $5.33\ m$^3 using the equation for the volume of the ellipsoid:

$$V = \frac{4}{3} \cdot \pi \cdot a \cdot b^2$$

Lastly, we double check the volume calculation using the manufacturer's data, if it is available, using a VLOOKUP excel function from the manufacturer's brochure.

Hence adding up the volume of the body, and the head we receive our tank volume of $\mathbf{69.32}m^3$. Dividing the volume required, by this volume, we receive our fractional number of tanks required of $20.55\ Tanks$. Rounding up to the nearest integer corresponds to a requirement of $21\ Tanks$.

Methane Removal

Now we know that methane in the storage tanks gets transported out from the facility once per day.

Considering redundancies, it is possible that the entire process of the plant may have to stop or get disrupted if methane tanks fill up. Additional spare capacity was considered in cases where trains may be late or did not arrive. A loading/unloading buffer is required, as filling a tank at the same time as it is unloading is not safe, and introduces potential operator errors.

It was considered to instead flare the gas in the case that trains get delayed, as methane is an inexpensive gas, however that was decided against, as it introduces additional safety issues, and introduces more compliance and protocols to adhere to.

It was also considered to simply increase the operating pressure to the surge pressure, closer to the design conditions, and hence we will be able to store more methane, while tanks are unloading, or if the train is late. However, this introduces safety issues, and introduces the risk of overpressure, potentially requiring flaring or stopping he process altogether.

Ultimately however, these strategies involve a larger amount of risk than simply adding another tank. And hence it was decided to construct an additional tank. It was calculated that an additional tank provides for $1.17\ hours$ of operation time, and along with the rounded-up volume of 0.45 that we have from rounding up the tank to an integer value, we have a spare capacity filling time of $1.69\ hours$ (at standard operating flowrate).

We think that is sufficient as a loading/unloading buffer, and in cases where the train arrives late.

Determination of Design pressure and Temperature

Firstly, from the problem our normal operating pressure is 2000kpa. (Normal operating pressure). Now our maximum operating pressure is 2200kpa (accounting for surges).

According to the equipment sizing guidelines provided by Northwestern University (Equipment sizing, 2023), our design pressure should be set higher than the maximum operating pressure. Specifically, we must add the greater of a 10% margin or 25 kPa. Following this guideline, we set our design pressure at 2420 kPa.

Now for temperature, we start with the minimum design temperature, which is based on the environmental conditions. According to records (Climate Extremes and Records, Bureau of Meteorology, 2023), the minimum temperature in Australia is -11.7 °C. To this, we add an allowance of 25 °F, equivalent to 13.89 °C, as suggested by the guidelines. This results in a minimum design temperature of -51 °C.

For the maximum design temperature, we need to make some assumptions and consider the process dynamics. When compressing methane, the gas naturally heats up. Typically, a multi-stage compressor is used in these processes, which means the gas doesn't instantaneously heat up. Using HYSIS to analyse this scenario, we get an outlet temperature of around 350 °C. However, this scenario does not seem realistic and is potentially dangerous, and would likely require an exceptionally large compressor.

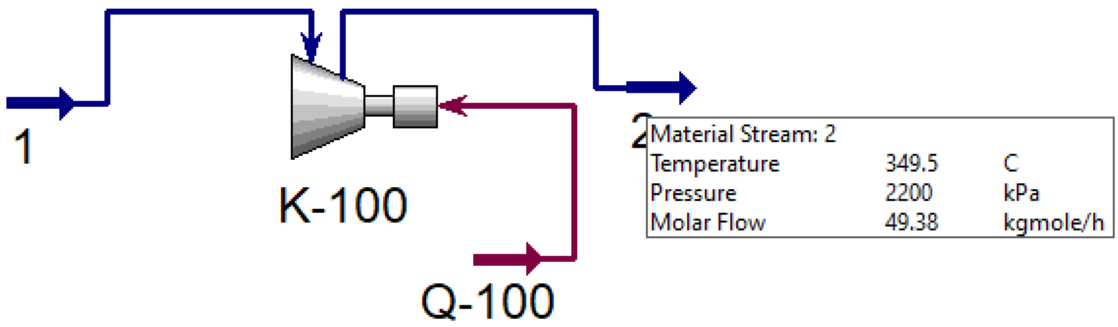


Figure 5 – Hysis Compressor simulation

We assume that a multi-stage compressor is used, which compresses and cools the gas at each stage. Additionally, considering the safety guideline that storage tanks must be placed 100 meters away from other operating units, there should be adequate time for the gas to cool down and return to ambient temperature. Therefore, we use normal operating temperature of 25 degrees, and assume a maximum design pressure as the maximum ambient temperature in Victoria, plus a margin.

The facility is assumed to be located in Victoria, where the maximum recorded temperature is 48.8 °C (Climate Extremes and Records, Bureau of Meteorology, 2023). Following general guidelines, we then add a margin of 50 °F, or 27.78 °C. With this, we round our maximum design temperature to 80 °C.

In future work, we may also explore the potential impact of sunlight on the tank temperature, considering the tank's exposure to open outside air. There are additional standards that set out the calculation due to sun exposure such as AS2872. However, for the current design, we assume that the temperature increase due to sunlight is negligible, and that measures are in place to shield the tank from direct sunlight.

Material Selection

Various materials exist for the construction of pressure vessels, however the search was narrowed down to two materials, AS 1548-7-460 carbon steel and a stronger AS 1548-7-490 steel.

This is firstly due to the properties of methane and our operating conditions. Methane is not corrosive to carbon steel, and hence highly resistant materials are not necessary (“Compatibility charts for methane”, 2023). Additionally, the operating pressure and temperature is not very high and hence materials resistant to extreme conditions are not needed.

Additionally, 460 and 490 carbon steel is readily available by manufacturers, and hence will not require extended waiting periods to manufacture.

For each of the two materials we first obtain the design tensile strength of the material at the specified design conditions using Table B1 from AS1210, starting initially with a thickness of 16-40mm, and using a temperature of 100 degrees C, iterating values as needed.

Grade	Thickness mm	Steel group	R_m MPa	Notes	Design tensile strength, MPa (See Notes)													
					Temperature, °C													
					50	100	150	200	250	300	325	350	375	400	425	450	475	500
PT430	3≤t≤16	A1	430	—	**185**	**185**	173	162	151	140	135	130	127	122	119	—	—	—
	16<t≤40				185	174	161	151	141	131	126	121	118	114	111	—	—	—
	40<t≤80				178	168	156	146	136	126	128	117	114	110	107	—	—	—
	80<t≤150				165	156	144	135	126	117	113	108	106	102	99	96	—	—
	All							Creep	Life:		Indefinite		—	*115*	*79*	*52*	*35*	—
											10 000	hrs	—	*179*	*141*	*105*	*74*	*52*
											30 000	hrs	—	*158*	*118*	*82*	*56*	*39*
											100 000	hrs	—	*133*	*91*	*60*	*40*	—
											150 000	hrs	—	*123*	*82*	*53*	*33*	—
											250 000	hrs	—	*114*	*73*	*48*	—	—
PT460	8≤t≤16	A1	460	—	**198**	189	176	165	150	143	138	132	128	124	120	116	—	—
	16<t≤40				195	183	170	159	148	138	133	128	124	120	116	112	—	—
	40<t≤80				182	170	158	148	138	128	124	119	116	112	108	104	—	—
	80<t≤150				175	165	152	142	133	124	120	115	111	107	99	—	—	—
	All							Creep	Life:		Indefinite		—	*115*	*79*	*52*	*35*	—
											10 000	hrs	—	*179*	*141*	*105*	*74*	*52*
											30 000	hrs	—	*158*	*118*	*82*	*56*	*39*
											100 000	hrs	—	*133*	*91*	*60*	*40*	—
											150 000	hrs	—	*123*	*82*	*53*	*33*	—
											250 000	hrs	—	*114*	*73*	*48*	—	—

Figure 6 – Table B1 extract (AS1210, 2023)

We use a design tensile strength of 183 for the 460 carbon steel, and 211 for the 490 carbon steel.

Initially, we used a welding efficiency of 0.7, to comply with class 3 regulations. However, upon iterating it was discovered that the minimum thickness does not comply with class 3 regulations, and hence a welding efficiency of 0.8 was used to comply with a class 2B vessel.

Lastly, or corrosion allowance was set to 2mm, and hence assuming a corrosion of 0.1mm per year, our tank life is expected to be 20 years, which is the standard for a pressure vessel.

Now to calculate the minimum thickness required of the pressure vessel for a specified material, the equation

$$t = \frac{P \cdot D}{4 \cdot f \cdot n - P}$$

Was used. It describes the circumferential stress on the pressure vessel. The circumferential stress is the limiting stress on the pressure vessel, as the tangential stress is always 2 times less than the tangential stress for our pressure vessel shape.

Additionally, the minimum thickness from the stress on the ellipsoidal heads is the same as the circumferential stress, due to the fact that our ellipsoid ratio is 2, therefore we have a K constant of one, yielding the same stress.

$$t = \frac{P \cdot D \ \mathrm{K}}{4 \cdot f \cdot n - P}$$

We hence get a minimum thickness of 22.76mm for the 460 steel, or 19.72mm for the 490 steel. Adding the corrosion allowance, the minimum thickness is 24.76mm, and 21.72mm respectively.

Referring to the APV pressure heads website, the standard sizes available are 20mm, 25mm, and 32mm. As a result, our choice for steel would need to be rounded to 25mm. Given this, selecting the 490 steel wouldn't be practical since it's not only pricier but also not readily available. We'll opt for the 460 steel head, which has a thickness of 25mm.

Looking at the Bluescope steel website (Bluescope, 2023), we've chosen the XLERPLATE® steel for the main body of the vessel. This selection aligns with the standard and is readily available. Specifically, we've opted for the AS1548-PT460NR from the standard range, which matches the material of the heads and is the most accessible. The chosen steel is 25mm thick, 3100mm wide, and 9.6m in length.

For our requirements, this is ideal. The necessary circumference for a circular section is 8.58m, and the main pressure vessel's length is 10.924m. This means we'll need four plates to construct the body of a single pressure vessel.

Joint selection

To select joints, we refer to AS1210 for guidance. We will consider joints that have a weld thickness sufficient with our material thickness. Our primary considerations are:

- Double-welded single V butt joint
- Single-welded single V butt joint with a backing strip.
- Single-welded single V butt joint
- Double-welded double V butt joint
- Double-welded single U butt joint

Referring to table 3.5.1.7, only the double welded butt joint, and the single welded but joint with backing strip allows welded joints at every location on the vessel. Selecting any other weld type to do different types of welds introduces operational complexity, slows down the production process, and may introduce operational errors, and hence only one type of weld will be used for all joints in the vessel. The single-welded single V-butt joint will also not be considered.

Additionally, we can see that if using a single-welded joint with a backing strip, makes the welding efficiency 0.75 instead of 0.8 for the double welded butt joint. Reviewing the material calculations with a welding efficiency of 0.75 results in a higher minimum thickness requirement of 32 instead of 25, and hence the single-welded butt joint with a backing strip will also not be considered.

Double welded joints are the strongest joints available, however this joint presents difficulty due to the accessibility issues in the tank. A double welded joint requires personnel to weld both inside and outside the tank, and hence proves difficult in the final stage of welding the vessel. However this can be mitigated via access from the manhole when welding the final stages of the vessel, and hence can be used.

Now selecting for a V vs U joint presents both advantages and disadvantages. U joints are typically smoother, and can reduce fatigue stress, however preparation is more complex and more expensive. V joints on the other hand are quicker and easier to prepare, at the cost of potentially higher stress concentrations and more filler material required (Butt Welds Explained, 2022).

Finally choosing between the Double-welded Single V and Double-welded Double V butt joint, the double V butt joint has uniform penetration, and can reduce the distortion and warping, at a cost of higher material cost, more complex preparation and more labour time required for machining.

Both joints have the same weld efficiency, hence we will pick the Double welded single V butt joints to weld all the welds in the tank.

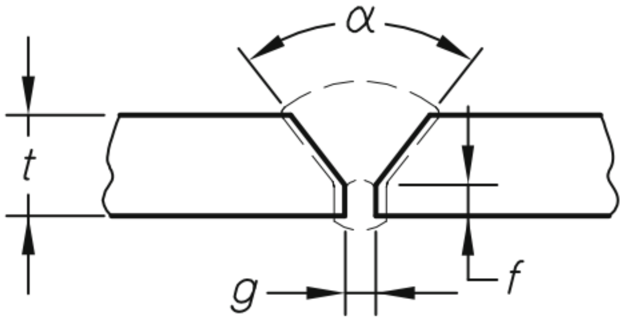


Figure 7 - Double Welded single V butt joint selected (AS1210, 2023)

Legs and support

For our pressure vessel, AS1210 standards set out a specification of saddle supports, Section 3.24.4 explicitly states: "Vessels with a diameter exceeding 1 meter must be equipped with saddle supports that subsume a minimum of 120° of the shell circumferences continuously. Hence, we will need to utilise saddle supports.

Stresses and the weight of the vessel need to be considered to properly support our vessel and ensure that fatiguing does not occur due to improper support. We see that the standard only lays out that the maximum overhang from the support must be lower than 10 times the thickness, and the support should have an angle of 120 degrees at a minimum.

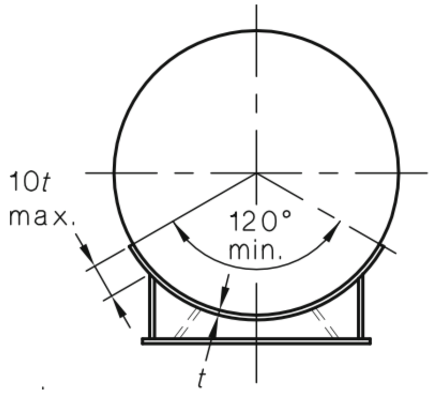
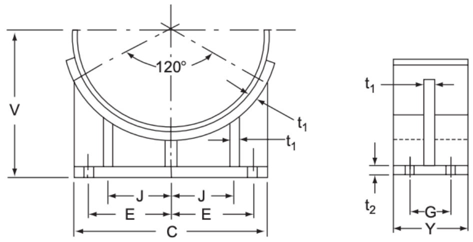


Figure 8 – Saddle support specifications (AS1210, 2023)

Determining whether to put 2 or 3 supports is an important question. Given the vessel's anticipated weight, it might be considered to add an extra support to alleviate stress in the vessel's centre. However, after careful consideration, we've decided against this. The textbook: “Chemical Engineering Design” states that "A horizontal vessel is typically supported at two cross-sections. Employing more than two saddles introduces uncertainty in load distribution"(Towler & Sinnott, 2008). Furthermore, the weight of the methane when the vessel is full constitutes only a minor portion of the vessel's total weight. This means the bending stresses introduced are minimal. By limiting the design to two supports, we can confidently predict and manage the stress distribution.

Additionally, the Chemical Engineering design textbook sets out the design specifications according to vessel size, and the maximum weight tolerated.



Vessel diam. (m)	Maximum weight (kN)	Dimensions (m)						mm			
		V	Y	C	E	J	G	t_2	t_1	Bolt diam.	Bolt holes
1.4	230	0.88	0.20	1.24	0.53	0.305	0.140	12	10	24	30
1.6	330	0.98	0.20	1.41	0.62	0.350	0.140	12	10	24	30
1.8	380	1.08	0.20	1.59	0.71	0.405	0.140	12	10	24	30
2.0	460	1.18	0.20	1.77	0.80	0.450	0.140	12	10	24	30
2.2	750	1.28	0.225	1.95	0.89	0.520	0.150	16	12	24	30
2.4	900	1.38	0.225	2.13	0.98	0.565	0.150	16	12	27	33
2.6	1000	1.48	0.225	2.30	1.03	0.590	0.150	16	12	27	33
2.8	1350	1.58	0.25	2.50	1.10	0.625	0.150	16	12	27	33
3.0	1750	1.68	0.25	2.64	1.18	0.665	0.150	16	12	27	33
3.2	2000	1.78	0.25	2.82	1.26	0.730	0.150	16	12	27	33
3.6	2500	1.98	0.25	3.20	1.40	0.815	0.150	16	12	27	33

All contacting edges fillet welded.

(b)

Figure 8 – Specifications of saddle supports (Towler & Sinnott, 2008)

We need to firstly calculate the weight of the methane and vessel. We use the flowrate of methane, and the time it takes to fill up one tank, netting 0.93 tonnes per tank of methane.

To find the weight of the vessel, we consult the XLERPLATE datasheet to find the density of our steel. According to manufacturer data, the density of steel is 7.85 tonnes per cubic metre (Edcon Steel, 2019).

For our calculations, we'll approximate the need for 3.5 plates to encase the entire vessel. Additionally, the head weighs kg/10mm in thickness. With the 2731 plate being 25mm thick, its weight is 1.82 tonnes. Since there are two of these plates, their combined weight is 3.63 tonnes. This brings our total weight to approximately 25 tonnes, and hence the force is approximately 236 kN.

We can see this is way less than the maximum weight supported from the figure above, and hence 3 supports prove to be unnecessary.

Constructing the vessel, we choose parameters for a vessel diameter of 3m (for ease of construction, and gain some ground clearance), and with some dimensions slightly tweaked to make use of less materials, and to bring down the incidence angle to utilise less material.

Regarding the vessel's lateral positioning, a uniformly loaded beam's optimal position is 21% from each end (Towler & Sinnott, 2008). This gives us a clear placement for the supports.

Manhole

According to 3.20.1 of AS1210, our vessel needs a manhole: “All vessels, except those permitted by Clauses 3.20.5 and 3.20.6, shall be provided with suitable inspection openings to permit visual examination and cleaning of internal surfaces.”

For practicality, we prefer the manhole to be located on the vessel's side rather than the top. Positioning it on the top would necessitate additional equipment like ladders for access, complicating the design and maintenance process.

Our vessel doesn't fall under the exemptions of Clauses 3.20.5 and 3.20.6, primarily because it's susceptible to corrosion and thus mandates an inspection opening.

According to table 3.20.4, we need an elliptical manhole (circular), since our vessel is over 1500mm in diameter:

TABLE 3.20.4

INSPECTION OPENINGS FOR GENERAL PURPOSE VESSELS

Inside diameter of vessel mm	Minimum clearance size of openings (Note 1) mm	Minimum number of openings (Note 2)	Location of openings
≤315	30 dia.	One for shells up to and including 900 mm long	In end, or where this is not practicable, in the shell near end.
		Two for shells over 900 mm long	—
>315 ≤460	40 dia.	Two for shells of any length	One in each end, or where this is not practicable, in the shell near each end
>460 ≤920 (Note 3)	50 dia.		
>920 ≤1500† (Note 3)	Handhole 150 diameter or 180 × 120	Two for shells up to 3000 mm long (Note 4)	One in each end or in the shell near each end
	Headhole 290 diameter	One for shells up to 3000 mm long (Note 4)	In the central third of the shell (Note 5)
>1500	Elliptical manhole or equivalent*	One for shells of any length	In the shell or end to give ready ingress and egress

* See Table 3.20.9.

Figure 9 – Inspection opening specifications (AS1021, 2023)

Hence we only need on manhole, in the body or the shell. Now we use table 3.20.9 to designate manhole size:

TABLE 3.20.9

SIZE OF INSPECTION OPENINGS

millimetres

Type	Circular openings (see Note 1)	Equivalent elliptical openings— (major × minor axes)	Maximum depth of opening (see Note 2)
Sighthole	50	—	50
	40	—	40
	30	—	30
Handhole	200	225 × 180	100 Pads
	150	180 × 120	75 Pads
	125	150 × 100	63 Pads
	100	115 × 90	50 Pads
	75	90 × 63	50 Pads
Headhole	300 max.	320 × 220 max.	100
	290 min.	310 × 210 min.	
Manhole (see Note 3)	550 (Preferred)	—	500
	500 (Min. recommended)	—	300
	450 (Exceptional)	450 × 400	245

Figure 10 – manhole size requirements (AS1021, 2023)

We therefore pick a manhole with a 500mm circular opening, (we will rarely assume that we will need to get in and out of the vessel). Hence a maximum depth of 300mm.

Nozzles and pipe sizing

We begin by consulting the AS 4041 and AS4343 standards. According to AS4343, methane is classified as a VHG (Very Harmful Gas).

Turning to AS4041, it's specified that the pressure limit for a class 3 pipe is 2 Mpa, while for a class 2A pipe, it's 10.4 Mpa.

The configuration of our pipe network leading to the pressure vessel is important and will determine pipe size. One approach could be to use a class 2A pipe directly from the compressor, assuming simultaneous filling of all tanks, thereby distributing the pressure across them, and class 3 pipes could branch from each tank. However this approach isn't the most logical, as we are using two different classes of pipe, and additionally, introduces operational and safety issues. A better assumption would be to allow each tank to be be filled independently.

Our normal operating pressure is 2Mpa, therefore we must go with a class 2A pipe network.

Before proceeding further, it's essential to size our pipes. The first step is to determine the volumetric flow rate, which can be calculated from the mass of methane in transit divided by its density. At standard operating conditions, methane's density is 13.4 kg/m^3 (Methane Density Calculator, 2023).

Now to determine our cross-sectional area of the pipe, we must first determine the velocity of the methane travelling to the tanks. We refer to the guidelines set out by Chemical engineering design book, section 5.6:

Simpson (1968) gives values for the optimum velocity in terms of the fluid density. His values, converted to SI units and rounded, are

Fluid Density kg/m^3	Velocity m/s
1600	2.4
800	3.0
160	4.9
16	9.4
0.16	18.0
0.016	34.0

The maximum velocity should be kept below that at which erosion is likely to occur. For gases and vapors, the velocity cannot exceed the critical velocity (sonic velocity) and would normally be limited to 30% of the critical velocity.

Figure 11 – recommended fluid velocities (Towler & Sinnott, 2008)

Our methane density is approximately 13.4 kg/meters cubed at the operating conditions. Hence, we use the closest value of 9.4meters/second to calculate the cross sectional area of the pipe, and hence the diameter of the pipe.

We hence arrive at a value of 46.19mm inside diameter of pipe, rounding up, a pipe of DN50 is sufficient.

We will check that with the second statement, that the maximum fluid velocity should be limited to 30% of the critical velocity (sonic velocity). The sonic velocity of methane is 465.14 m/s, so we should be well within that range. Hence, we set our fluid velocity to 9.4 meters per second.

Hence, we choose DN50 pipe for the input nozzle.

Output nozzle

Firstly, to deliver the methane to the train, a compressor would need to be used to achieve the desired flowrate, especially at low pressure levels. Additionally, since we are using a compressor, there could be a possibility of negative pressure, however this can be mitigated with electronic control valves, and the hence an under-pressure sensor will not be used.

Now, for the output nozzle. There are many ways to set up a piping system, one way that is possible is to not have an output nozzle. Because we are storing compressed gas, it is possible to use singular common input/output pipes. This would save on piping costs, and equipment design costs.

This Idea was considered; however, it presented a number of drawbacks. Firstly, the flow of gas would be poorly controlled. As there is methane coming in from the compressor, and going during unloading, this results in uncontrolled flow rates, and hence could lead to unpredictable behaviour and dangerous pressure fluctuations.

There also exits a higher potential for backflow, compromising the quality of the methane, and causing damages to storage tanks. Therefore, two nozzles and two set of pipes were assumed to be operating.

Now to size the nozzle, we must take into account the time it takes to empty the vessels. If using the same diameter as the input nozzle, and emptying them one at a time, it would take 24 hours, the same time that the plant is operating, which is obviously unfeasible.

An unloading time of 1 hour seems feasible, and consistent with industry standards for other commodities (Grainscorp, 2021). Assuming tanks get filled one at a time, the time to unload a tank is just under 3 minutes, and the calculated nozzle diameter is over 200mm. This value seems however seems ridiculous, and unfeasible. In addition, constantly starting and stopping the process and the compressor, may put wear on the pressure vessel and compresser. In addition, there may be a time delay to spin up and get the compressor going, so the switchover period to other tanks may take additional time.

A solution is to empty the tanks in parallel, and hence ensure uninterrupted flow from all tanks at once. Remote valves can be used to shut off any tanks that are not required to be empty, or are in the process of being filled. A much lower pipe diameter can also be used to ensure cost savings on material costs.

We will size our exit nozzle so that all 22 tanks can be unloaded within a span of 1 hour. Hence, we calculate a nozzle diameter of 65.

Pressure Relief Devices

According to AS1021, we are required to have an overpressure valve. This device should prevent the pressure from exceeding 110% of the vessel's design pressure. A spring-loaded type should be used.

Additionally, a thermal relief device may be required for our vessel, as the contents are flammable. However, we can use a singular pressure relief valve for both overpressure and fire scenarios, complying with clause 8.2.

We will consult the pressure systems brochure to find a pressure relieving valve we can use for our vessel (Flanged Safety Relief Valves | Pressure Systems, 2023). We choose a LESER type 437 valve, as it adheres to our pressure rating. We choose an inlet size of DN20, as it is in the medium range of what is available. To explore requirements of flowrate and pressure, we could use equations from section 8.6 to find the discharge capacity for pressure relief and fire conditions

Figure 12 – Selected pressure relief valve (Flanged Safety Relief Valves | Pressure Systems, 2023).

Additionally, to comply with clause 9.2, we need discharge location which is 2m away from the vessel and contained, as it is a flammable gas. As methane is an inexpensive gas, it may be possible to simply flare and burn it off in the event of an overpressure scenario.

Additionally, we will set the valve to discharge at a pressure of 105% of the maximum operating pressure. This gives us a buffer for extreme surges, while also being safe, and not too near the design pressure, as recommended by the following figure.

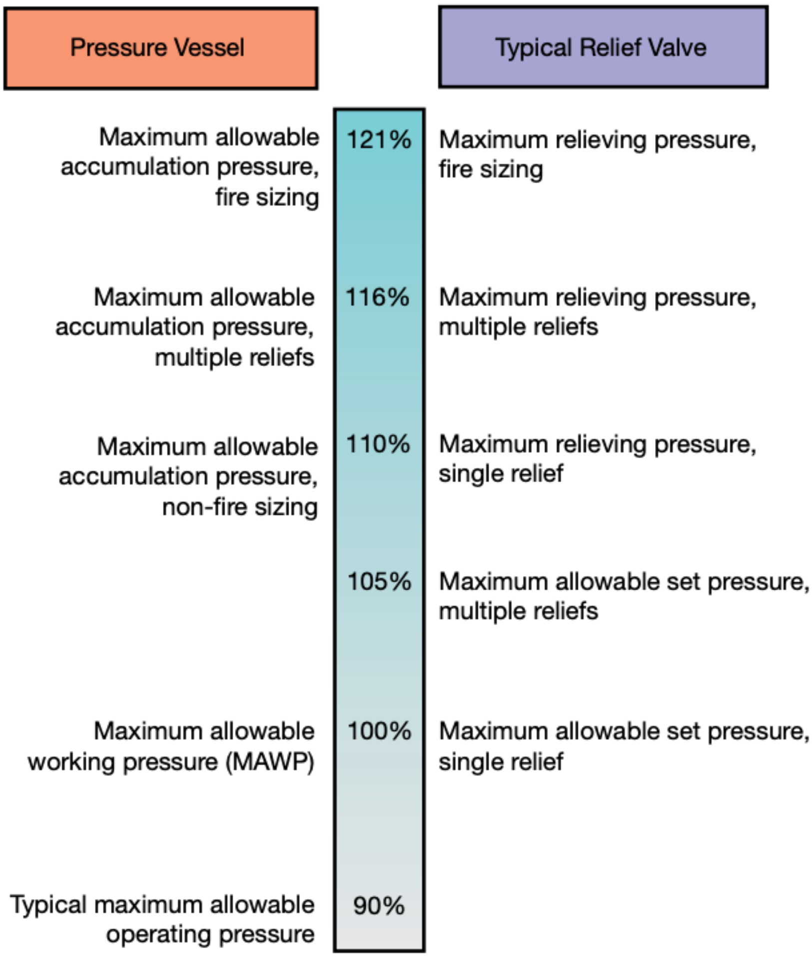


Figure 13 - Typical relief valve conditions (CEP, 2013)

Other Valves and Gauges

According to Clause 8.13, each vessel equipped with a pressure relief device is mandated to have a pressure gauge. Additionally, shut-offs are recommended to be put in place between the vessel and the gauge.

Drainage is not required for our vessel as no liquid accumulation is expected. Additionally, a burst disk is not necessary as the conditions outlined in Clause 8.3 are not met. These conditions include rapid pressure rise, leakage concerns, or heavy deposits that could render a pressure-relief valve inoperative.

For the vacuum relief valve. Although there may be a compressor facilitating the removal of the gas, the differential pressure remains minimal, especially spread out across the pressure vessels in parallel. Additionally, vacuum mitigation measures such as vacuum relief in the pipe and automatic valve switching could be implemented to ensure that there will not be a vacuum in the tanks. Hence, we believe that the vacuum relief valve is not necessary.

Nozzle thickness

AS1210 sets out guides for the nozzle thickness in section 3.19.10.2. It states that it must be sufficient to withstand the wall's required thickness, however must be not greater than the outside diameter of the pipe to the power of ¼, and smaller than the stress at the point of attachment.

We first consult AS4041 pipe standards, and use the minimum pipe thickness equation of $t = \frac{P \times d}{2 \times f \times e \times M - P}$ to find the minimum thickness of the inlet and outlet and manhole. We get a minimum thickness of 4.36mm for the manhole, and 0.43mm, and 0.57mm for the inlet and outlet respectively.

These are much lower than realistically possible and feasible. Hence we set all our nozzle thickness to be 25mm, the same as the pressure vessel.

Bolted Flange connections

We will have all of our connections as bolted flange connections. To comply with section 3.21.

We will use Narrow-face flanges, as they are the most secure and reliable.

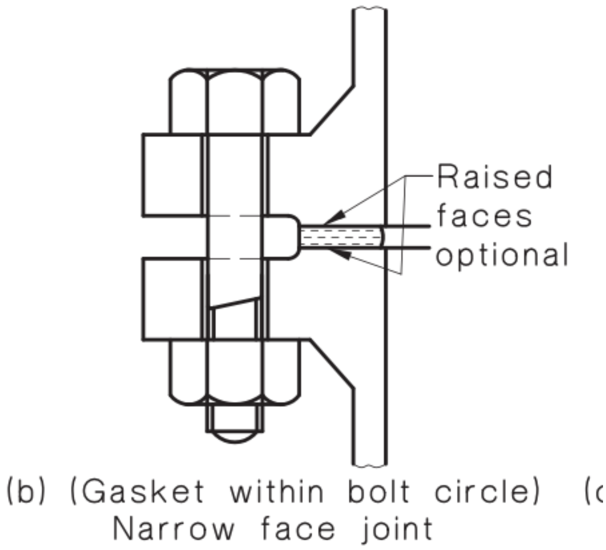
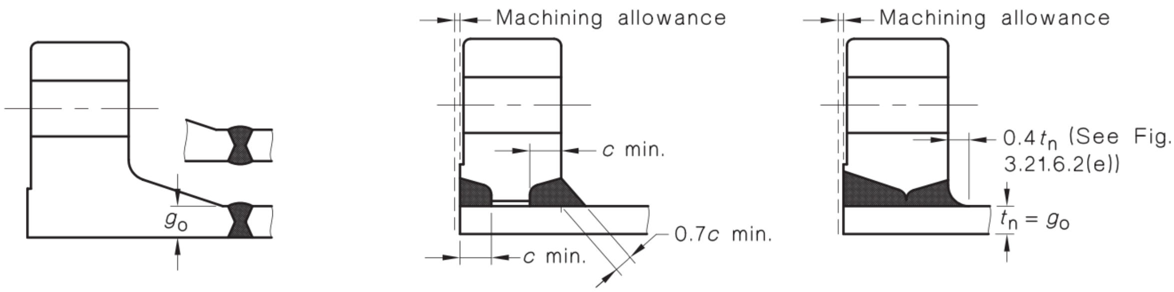


Figure 14 – Narrow face flange (AS1210, 2023)

We shall use the following types of welds to connect flanges, as there is no limits to the temperature and pressure, and they look to be the strongest:



(a) Weld neck flange

(b) Face and back welded-on flange

(c) Full penetration welded-on flange

Figure 15 – Flange welds (AS1210, 2023)

Additionally, we will also use carbon steel, the same steel as our pressure vessel, with a threaded and expanded method of attachment, as they are adequate for our conditions:

Material	Method of attachment	Maximum pressure MPa	Maximum temperature °C
Carbon and carbon-manganese steels	Threaded and expanded	3.1	371
	Taper to taper	2.1	260
	Taper to parallel	0.86	260

Figure 16 – Maximum temperature and pressure for attachments (As1210, 2023)

Nozzle padding

We must introduce padding at the meeting point of our vessel and the nozzle to reduce the stresses that occur. Generally, the added padding corresponds to the amount of material removed from the bore hole when drilling. We will calculate the Area requirement using the formula $A = dtF + 2T_{b1}tF(1 - f_{r1})$.

We find that the area of padding that is required to be compensated is 9937mm^2 for the manhole, 1065mm^2 for the inlet nozzle and 1361mm^2 for the outlet nozzle.

Integrity Tests

We need to perform pneumatic or hydrostatic tests to comply with regulations. However pneumatic tests were avoided due to the fact that they introduce unnecessary risks, and are more expensive to operate, additionally they are discouraged by regulations.

A vessel hydrostatic test pressure was calculated using the equation:

$$P_h = \frac{1.5P \times f_h}{f}$$

And hence a hydrostatic test pressure of 3868 Kpa was obtained.

Managerial Aspects

Safety issues

The main risk during normal operation is an escape of methane from containment. There is no risk of explosion, as long as there isn't any oxygen entering with the methane from the input stream. Operational procedures and additional sensors may be implemented in processes upstream such as the compressor or the flash drum, to mitigate that risk.

It is also important to set out procedures for loading and unloading the methane from the storage tanks. The pressure must be carefully managed so that there isn't any backpressure or oxygen introduced after unloading the tanks. A one-way check valve or other devices or thorough operator training procedures can be implemented to mitigate this risk.

Staff and operator training is also essential to ensure smooth operation of the methane storage tanks. Emergency procedures must be set in place to effectively plan for emergencies, and emergency response protocols must be taught to operators.

Equipment should also be maintained and repaired by qualified personnel, knowledgeable with industry best practises, and knowledgeable of the operation of the facility.

Lastly, additional factors and loads such as seismic and wind loads should be considered before production of the vessel to ensure that it remains stable and safe in the event of natural disasters.

Operational and managerial issues

A regular maintenance schedule should be put in place to ensure the pressure vessel is properly maintained and inspected. This includes tasks such as checking for corrosion, performing non-destructive testing, and verifying the integrity of components.

Accurate records of inspections, maintenance activities, and repairs conducted on the pressure vessel should be kept. This documentation will help demonstrate compliance with regulations and provide a historical record of the vessel's condition.

standard operating procedures (SOPs) for the safe and efficient operation of the pressure vessel should be implemented. Clearly define roles and responsibilities, start-up and shutdown procedures, and emergency response protocols.

Compliance with regulations

The design, fabrication, and operation of the pressure vessel should comply with relevant codes and standards, such as AS1210, AS4343, and local council regulations. Operators should stay updated with any changes in regulations to maintain compliance throughout the vessel's lifecycle.

Coordination should happen with regulatory bodies and authorized inspection agencies to schedule periodic inspections and obtain necessary certifications. This may include certifications for design and fabrication, as well as regular inspections for ongoing compliance.

Documentation, including design calculations, certificates, inspection reports, and permits should be kept. An organised document control system should be put in place to ensure quick retrieval of documents.

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