Introduction to Pressure Vessels

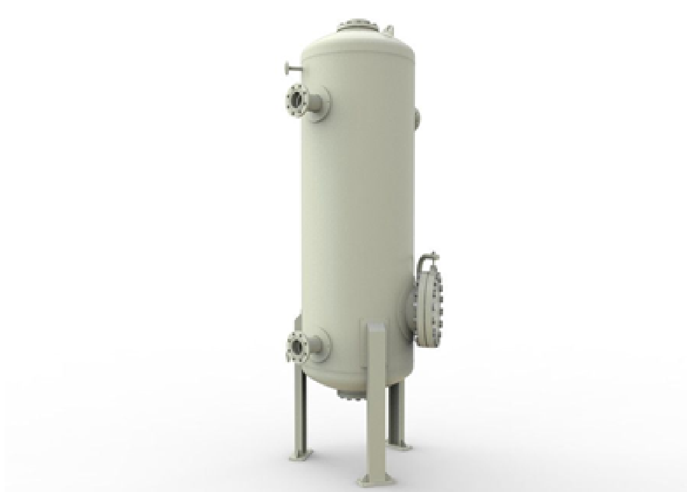
A pressure vessel is a closed vessel designed to hold a gas or liquid at a pressure other than ambient pressure. The end cap attached to the barrel is called the head. A pressure vessel can be defined as a pressure component of relatively large capacity (such as a spherical or cylindrical vessel) that has a greater cross section than the associated pipe or tube.

In the industrial field, pressure vessels are technically designed to operate safely at specif-ic pressures and temperatures called "Design pressure" and "Design temperature". An im-properly designed container for handling high pressure is a very serious safety issue. Therefore, the pressure vessel design and certifications are strictly complied by ASME North America Boiler and Pressure Vessel Code, EU Pressure Equipment Directive (PED), Japanese Standard Industrial Standard (JIS), Canadian CSA B51, AS1210 in Australia and other international standards such as Lloyd's, Germanischer Lloyd, Det Norske Veritas and Stoomwezen.
Common connections for pressure vessels include NPT, British Standard pipe threads, met-ric pipe threads, flanges, welds, and tubular or pipe fittings. National Pipe Thread can be male threads (NMPT or MPT) or female threads (NFPT or FPT). The flanges can be ANSI or SAE rated flanges. Soldering is a set of terms used in the joining process that uses filler metal (solder). Tubular joints are similar to hose rods, barbed or ridged and are designed to hold a connecting pipe by pushing and retaining by it.

Types of Pressure Vessels

There are a few types of pressure vessels:

Thinned wall: These are the most common types. A thinned walled pressure vessel is a pressure vessel with a thinned wall if the diameter is 10-times or more of the thickness.
Thick walled: These pressure vessels are rarely common. A thick walled pressure vessel is any cylinder [shell] ratio that is 10% or more of the ratio of thickness to inside diameter of a vessel.
Storage tanks: Storage tanks are under the category of thin walled pressure ves-sels and are typically under 15-psi. They are normally super thin compared to the ratio mentioned above.
Transportable Containers: These are the most commonly used pressure vessels and po-tentially the most ignored. These are mass produced and required to audit every 10 years for propane and gas.
Propane bottles: forklifts, cooking gas kitchen, BBQs
Gas cylinders: CO2, O2 etc



Load definition and sizing of principal components

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Allocated student number:

X = **5**
Y = **9**

Fettling shop

Description of tools	No of tools	FAD per tool (l/sec)	Total FAD for specific tool	Load factor	Air Consumption (l/sec)
Grinders	8	29.63	237.04	0.5	118.52
Chippers	5	5	25	0.5	12.5
Medium Hoist	2	20	40	0.1	4
Total air consumption for fettling shop (l/sec)					**135.02**

Given Y = **9**

Grinders:

Air consumption rate per tool =24+(Y/1.6)
=24+(9/1.6)
= **29.63 l/sec**

Air consumption = 29.63*8*0.5
= **118.52 l/s**

Chippers:

Air consumption rate per tool =14-Y
=14-9
= **5 l/sec**

Air consumption = 5*5*0.5
= **12.5 l/sec**

Medium hoist:

Air consumption rate per tool = **20 l/s**

Air consumption = 20*2*0.1
= **4 l/sec**

Machine shop

Description of tools	Total FAD for specific tool (l/sec)	Load factor	Air consumption (l/sec)
Blowguns	70	0.5	35
Air chucks			
Air vices			
Total air consumption for machine shop (l/sec)			**35**

Blow guns, air chucks and air vices uses **70 l/s** of air in total.

Air consumption = 70 x 0.5
= **<u>35 l/sec</u>**

Assembly shop:

Description of tools	No of tools	FAD per tool (l/sec)	Total FAD for specific tool(l/sec)	Load factor	Air Consumption (l/sec)
Small screwdrivers	25	7	175	0.40	70
Nut setters	8	20	160	0.25	40
Steel drills	12	12.67	152.04	0.50	76
Total air consumption for assembly shop (l/sec)					**186**

Given X = **<u>8</u>**

Small screwdrivers:
Air consumption rate per tool = **7 l/s**

Air consumption = 7 x 25 x 0.4
= **<u>70 l/sec</u>**

Nut setters:
Air consumption rate per tool = 16 + (X / 2)
= 16 + (**8** / 2)
= **<u>20.0 l/sec</u>**

Air consumption = 20 x 8 x 0.25
= **<u>40 l/sec</u>**

Steel drills:

Air consumption rate per tool = 14 - (X / 6)
= 14 - (**8** / 6)
= **12.67 l/sec**

Air consumption = 12.67 x 12 x 0.50
= **76 l/sec**

Paint shop:

Description of tools	No of tools	FAD per tool (l/sec)	Total FAD for specific tool(l/sec)	Load factor	Air Consumption (l/sec)
Polishers	22	9.45	207.9	0.50	104
Spray painting guns	20	4	80	0.50	40
Spray painting masks	20	3.6	72	0.50	36
Total air consumption for paint shop (l/sec)					**180**

Given Y = **6**

Polishers:

Air consumption rate per tool = 10 - (Y / 11)
= 10 - (**6** / 11)
= **9.45 l/sec**

Air consumption = 9.45 x 22 x 0.50
= **104 l/sec**

Spray painting guns:

Air consumption rate per tool = **4 l/s**

Air consumption = 4 x 20 x 0.50
= **40 l/sec**

Spraying painting masks:

Air consumption rate per tool = 3 + (Y / 10)
= 3 + (**6** / 10)
= **3.6 l/sec**

Air consumption = 3.6 x 20 x 0.50
= **36 l/sec**

Packaging department

Description of tools	No of tools	FAD per tool (l/sec)	Total FAD for specific tool(l/sec)	Load factor	Air Consumption (l/sec)
Wood borers	10	15	150	0.30	45
Large screwdrivers	8	13	104	0.25	26
Heavy hoists	3	20	60	0.10	6
Total air consumption for packaging department (l/sec)					**77**

Wood borers:

Air consumption rate per tool = **15 l/s**

Air consumption = 15 x 10 x 0.30
= **45 l/sec**

Large screwdriver:

Air consumption rate per tool = **13 l/s**

Air consumption = 13 x 8 x 0.25
= **26 l/sec**

<u>Heavy hoists:</u>

Air consumption rate per tool = **20 l/s**

Air consumption = 20 x 3 x 0.10
= **<u>6 l/sec</u>**

<u>Forge</u>

Swept volume of double acting cylinder per stroke:

$$= [(\pi D^2) / 4] \times L + \{[(\pi D^2)/ 4] - [(\pi D^2) / 4)]\} \times L$$
$$= [(\pi \times 0.5^2) / 4] \times 1.075 + \{[(\pi \times 0.5^2) / 4] - [(\pi \times 0.125^2) / 4]\} \times 1.075$$
=**<u>0.409 m^3 / stroke</u>**

Total stroke	= 10
Time taken for stroke	= 100 sec
Hence 1 stroke	= 100 / 10
	= 10 sec

Volume flow rate = 0.409 / 10
= **<u>0.0409 m^3 / sec</u>**

Absolute pressure = atmospheric pressure + gauge pressure
= 101300 + 590000
= **691300 N / m^2**

Patm x Vfad	= P compressed air x V compressed air
101300 x Vfad	= 691300 x 0.0409
Vfad	= 0.279 m^3 /sec
	= **<u>279 l / sec</u>**

Total air consumption rate for the factory

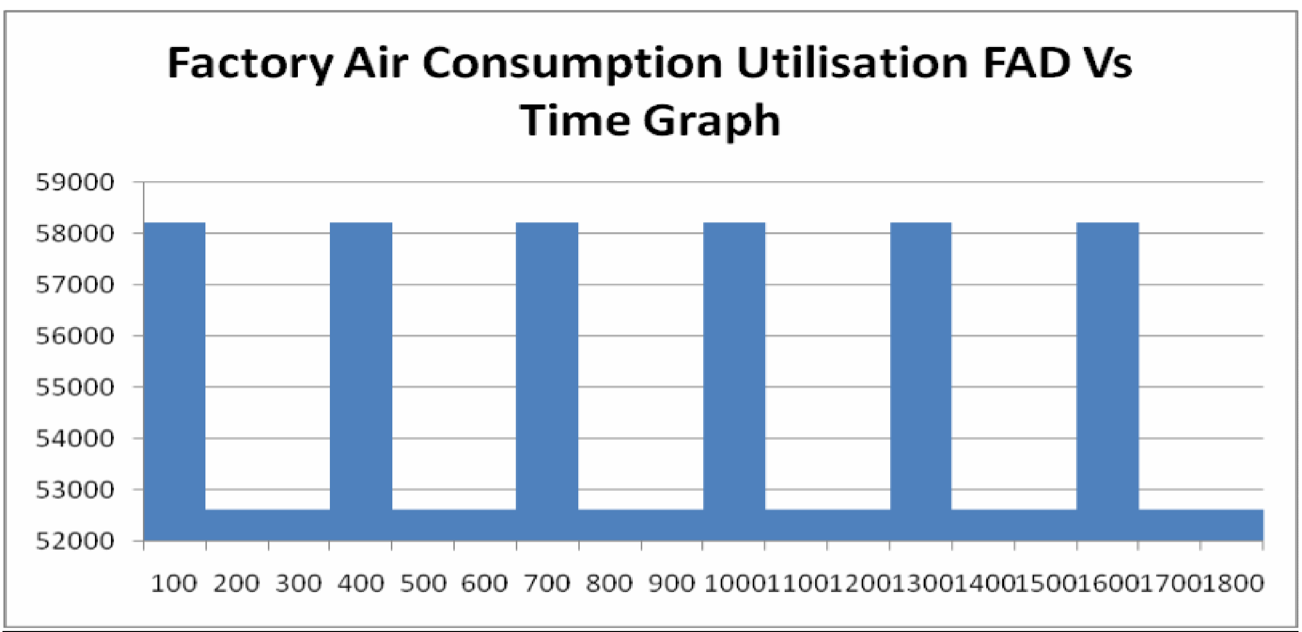
Departments	Air consumption rate with load factor (l /sec)
Fettling shop	329
Machine shop	35
Assembly shop	186
Paint shop	180
Packaging department	77
Forge	279
Overall leak	70
Total air consumption	**1156**
Total air consumption without including forge	**877**

Hence the total air consumption = **1156 l/ sec**
= **69.36 m^3 / min**

Hence the total air consumption Without including the forge = **877 l / sec**
= **52.62m^3 / min**

Total Airflow required for factory **with** the **forge** in operation is **1156 litres/sec.**
Hence in **1 min** the factory total free air consumption is $1156 \times 60 = 69360\ litres$

Total Airflow required for factory **without** the **forge** in operation is **877 litres/sec**
Hence in **1 min** the factory total free air consumption is $877 \times 60 = 52620\ litres$



Question 2a

Mean FAD(Free air delivery) Calculation:

$Total\ area\ under\ the\ graph = 69360 \times 30 = 2080800\ l/\min$

$Area\ excluded\ from\ the\ graph = (69360 - 52620) \times 6 \times 3.33 = 334465.2\ l/\min$

$$Mean\ FAD\ over\ 30\ \min\ Period = \frac{2080800 - 334465.2}{30} = 58211.16\ l/\min \quad or = 58.211\ \frac{m^3}{\min}$$

The peak requirement of airflow occurs when Forge is in operation which last for 100 seconds out of a 300 seconds. Cycle, as illustrated by the Air consumption rate load profile graph. Selecting a compressor that equal to peak flow will not be necessary as this will incur higher running cost and capital expense. Instead a compressor and receiver combination is designed.

Assuming a storage receiver will be used, there will not be a need to select a very high capacity compressor therefore the mean FAD is used as a guide for selecting the most suitable pump from the catalogue data provided (on pp.2.23-2.26 in the project resource material)

Based on the calculations above, type **ER8** was selected as it suited the FAD required.

Question 2b

From the catalogue data provided on pp2.23-2.26 in the project resource material, select a compressor that allows 5% possible future decay in free air delivery capability as the compressor wears.

The 5% decay is based on the chosen compressor, **ER8**.

FAD of **ER8** compressor = **63 m^3/min**
Therefore 5% decay = 63 x 0.95
= **59.85 m^3/min**

From this calculation, it is clear that with a possible 5% decay due to the wearing of the compressor, the FAD is still within the tolerance of the mean FAD **58.202 m^3/min** at **59.85 m^3/min.**
From the **ER8** compressor specifications table, we can see the normal working pressure is 100 psi and the maximum working psi is 125 psi. Therefore the selected compressor **ER8** meets the factory's normal working pressure of 100 psi and the maximum working pressure of 115psi.

The advantages of choosing **ER8** compressor, is that changes are not required even with the possible 5% future decay. Furthermore there is also no additional cost incurred with this selected compressor. However, the disadvantages of the **ER8** compressor, is that the amount of compress air produced by the compressor can only meet the required working pressure of the factory to function and not the projected demand. If the factory manager decided to add in any extra tools in the future, the **ER8** compressor may not be sufficient to meet the demands. If there is another possibility further decay of the compressor, it also may not be able to meet the factory demand.

Question 2c

Volumetric efficiency η_v = (Vol. of air delivered measured at free air pressure / swept vol. of cylinder)

From the **ER8** compressor specifications:

FAD = 63 m^3/min
Piston displacement = 73.7 m^3/min

η_v = (63/73.7) x 100% = **85.482%**

Other useful efficiency of a compressor is driven by electric motors and the power is transmitted through belt drive or gear box. To estimate the electrical power input the efficiency of the drive mechanism and that of the electric motor should be taken into account.

Isothermal Efficiency allows us to know how much heat is being lost and also can make use of the heat by recycling it for other uses; this in term saves energy and thus saves costs.

Mechanical Efficiency allows us to know how much energy is being lost in overcoming the resistances (Eg. Piston and bearing friction) in the mechanism of the compressor and also how good is the design. These enable us to keep track of the life span of the compressor's mechanism due to wear and tear, and this upgrading it to further overcome and reduce the resistances in order to save energy and costs.

Transmission Efficiency allows us to know whether the compressor's power transmitted through a belt drive or gear box, which is driven by electric motors, is efficient to transmit to the system. Hence, efficiency of transmission = compressor shaft power / motor shaft power.

Motor Efficiency allows us to know whether the compressor's motor is efficient to transmit power to the system with an amount of electrical power input. Thus enables us to monitor and estimate how much electrical power input is needed to activate the compressor's motor and thus saves costs. Hence, efficiency of motor = motor shaft power / electrical power input.

For system without storage receiver we have to take the peak air consumption rate for consideration of compressor. Based on air consumption rate, we can conclude that the production is contributing the peak demand due to the forge operation. Based on the factory air consumption rate of 58.203 m^3min, from the resource material catalogue the chosen compressor type is **ER8**. This type of compressor is chosen because its FAD of 63 m^3/min is greater than the demand of 58.203 m^3/min.

For the above scenarios, for a system without an air receiver, it will generally require a higher rating compressor and this means that the power consumption will be much higher than a system with air receiver. Increase in power consumption also means an increase in electrical cost. Using a higher crating compressor also increase the overall capital cost of the system. Operating the system at full load is good in the sense that it minimises the idling time of the equipment and it maximise the profit the equipment contributes to. When designing a compressed air system, a realistic assessment of the number of tools likely to be used at any one time and the level of power at which they operate when being used must be determined. Overestimating will result in unnecessary capital expenditure on compressor, motor and wiring and if no receiver has to run all this time.

Question 3a

(Compressor type ER8)

Specification for the compressor

Weight : 5300 kg
Length : 95 inch
Width : 64 inch
Height : 93 inch
Liquid cooled : Cooling water required at 15 degrees celcius.

Max. working pressure: $8.8\ kg/cm^2\ or\ 125\ psi$

Piston displacement: $73.7\ m^3/\min$

FAD: $63.0\ m^3/\min$

Based on 5% possible future decay in delivery capability:

$$63.0 \times \frac{100-5}{100}$$

$$= 59.85\ m^3/\min$$

The normal operating pressure requirement in the factory disregarding the need to consider for future extra tools and increase in load factor is 690 kPa

Conversion of kilopascal to pound per square inch:

$690\, kPa \cdot g \cdot = \left(690 \times 10^{3}\right) \times 1.4510^{-4} = 100\, psi$

Compressor type ER8 normal working pressure is 100 Psi and Max. Working pressure of 125 Psi therefore it will be able to handle the factory normal operating pressure requirement of 100 Psi

<u>Part A (i)</u>

Receiver Size when the compressor is new

Taking into consideration that the peak demand occurs when the Forge is in operation, by which the duration is 1.667 min in a 5 min cycle. The time taken for the pressure drop from 100 psi (690 kPa.g.) to 90 psi (620 kPa.g.) should not be more than this 1.667 min duration.

Normal Working Pressure from Compressor: 100 Psi (690 kPa.g.)

Pressure in Receiver cannot fall below: 90 Psi (620 kPa.g.)

Assume time elapsed, $\Delta time = 1.667 \min$

Initial Absolute Pressure, $P_1 = 690\, kPa \cdot g \cdot$

Final Absolute Pressure, $P_2 = 620\, kPa \cdot g \cdot$

Atmospheric Pressure, $P_f = 101.3\, kPa \cdot g \cdot$

Initial FAD Volume Flow Rate, $\dot{V}_{fi} = 63.0$ m^3/min

Final FAD Volume Flow Rate, $\dot{V}_{fo} = 69.36$ m^3/min

Using the law of mass conservation: Change in store mass equal to the net inflow of mass.

Taking air to be a prefect gas, the mass flows may be related to the free air volume flows by the prefect gas law.

Hence assuming the temperature of the air in the receiver is constant, and equal to ambient, the following equation is arranged,

$$\Delta time = \frac{(p_2 - p_1)V_{rr}}{p_f\left(V_{fi} - V_{fo}\right)}$$ (Contd. below)

$$1.667 = \frac{\left(620 \times 10^3 - 690 \times 10^3\right) V_{rr}}{101.3 \times 10^3 (63.0 - 69.36)}$$

$$\therefore V_{rr} = 15.340 m^3$$

Hence volume of the receiver to cope with the demand when the compressor is new $= 15.340 m^3$

Part A (ii)

Receiver Size when the compressor delivery capability has decayed by 5%

Solving this problem in similar manner, taking into consideration that the peak demand occurs when the Forge is in operation, by which the duration is 1.667 min in a 5 min cycle. The time taken for the pressure drop from 100 Psi (690 kPa.g.) to 90 Psi (620 kPa.g.) should not be more than this 1.667 min duration.

Normal Working Pressure from Compressor: 100 Psi (690 kPa.g.)

Pressure in Receiver cannot fall below: 90 Psi (620 kPa.g.)

Assume time elapsed, $\Delta time = 1.667 \min$

Initial Absolute Pressure, $P_1 = 690\, kPa \cdot g \cdot$

Final Absolute Pressure, $P_2 = 620\, kPa \cdot g \cdot$

Atmospheric Pressure, $P_f = 101.3\, kPa \cdot g \cdot$

Initial FAD Volume Flow Rate, $\dot{V}_{fi} = 59.85$ m^3/min (95% of 63.0 m^3/min)

Final FAD Volume Flow Rate, $\dot{V}_{fo} = 69.36$ m^3/min

Using the law of mass conservation: Change in store mass equal to the net inflow of mass

Taking air to be a prefect gas, the mass flows may be related to the free air volume flows by the prefect gas law.

Hence assuming the temperature of the air in the receiver is constant, and equal to ambient, the following equation is arranged,

$$\Delta time = \frac{(p_2 - p_1) V_{rr}}{p_f \left(V_{fi} - V_{fo}\right)}$$

$$1.667 = \frac{\left(620\times10^{3} - 690\times10^{3}\right)V_{rr}}{101.3\times10^{3}(59.85 - 69.36)}$$

$$\therefore V_{rr} = 22.987 m^{3}$$

Hence volume of the receiver to cope with the demand when the compressor delivery capability has decayed by 5% $= 22.987\, m^{3}$

Question 3b(i)

The compressors' on time span and off time span both for the compressor in the as new state.

Forge is **ON** (start of new work cycle)
Compressor is **ON**

V_{rr}	The larger values of volume determined in Q3bii	22.988 m^3
V_{fo}	The compressed air needed by the factory	69.362 m^3/min
V_{fi}	The FAD of the compressor	63 m^3/min
P_1	Initial pressure	690 kPa.g.
Δt	Time difference	1.67 *min*
P_f	Atmospheric Pressure	101.3 kPa

By applying conservation of mass:

$$\Delta t = \frac{(P_2 - P_1)V_{rr}}{P_f\left(V_{fi} - V_{fo}\right)}$$

$$1.67 = \frac{(P_2 - 690\times10^{3})22.988}{101.3\times10^{3}(63 - 69.362)}$$

$$P_2 = 643.28\ kPa.g$$

Forge is **OFF** (100 sec work cycle is completed, 1.67 min has passed)
Compressor is **ON** (pressure in the receiver has reached 690kPa.g)

V_{rr}	The larger values of volume determined in Q3bii	22.988 m^3
V_{fo}	The compressed air needed by the factory	52.62 m^3/min
V_{fi}	The FAD of the compressor	63 m^3/min
P_1	Initial pressure	643.28 kPa.g.
P_2	Final Pressure	790 kPa.g.
P_f	Atmospheric Pressure	101.3 kPa

By applying conservation of mass:

$$\Delta t = \frac{(P_2 - P_1)V_{rr}}{P_f(V_{fi} - V_{fo})}$$

$$\Delta t = \frac{(790 \times 10^3 - 643.28 \times 10^3)22.988}{101.3 \times 10^3(63 - 52.62)}$$

$$\Delta t = 3.208\ min$$

Forge is **OFF** (4.875 min has passed)
Compressor is **OFF** (pressure in the receiver has reached 790kPa.g)

V_{rr}	The larger values of volume determined in Q3bii	22.988 m^3
V_{fo}	The compressed air needed by the factory	52.62 m^3/min
V_{fi}	The FAD of the compressor	0 m^3/min
P_1	Initial pressure	790 kPa.g.
Δt	Time difference	$0.125\ min$
P_f	Atmospheric Pressure	101.3 kPa

By applying conservation of mass:

$$\Delta t = \frac{(P_2 - P_1)V_{rr}}{P_f(V_{fi} - V_{fo})}$$

$$0.125 = \frac{(P_2 - 790 \times 10^3)22.988}{101.3 \times 10^3(0 - 52.62)}$$

$$P_2 = 761.015\ kPa.g$$

Forge is **ON** (start of new cycle, 5 min has passed)
Compressor is **OFF** (pressure in the receiver has reached 790kPa.g)

V_{rr}	The larger values of volume determined in Q3bii	22.988 m^3
V_{fo}	The compressed air needed by the factory	69.362 m^3/min
V_{fi}	The FAD of the compressor	0 m^3/min
P_1	Initial pressure	761.015 kPa.g.
P_2	Final Pressure	690 kPa.g.
P_f	Atmospheric Pressure	101.3 kPa

By applying conservation of mass:

$$\Delta t = \frac{(P_2 - P_1)V_{rr}}{P_f(V_{fi} - V_{fo})}$$

$$\Delta t = \frac{(690 \times 10^3 - 761.015 \times 10^3)22.988}{101.3 \times 10^3(0 - 69.362)}$$

$$\Delta t = 0.232\ min$$

Forge is **ON** (5.232 min has passed)
Compressor is **ON** (pressure in the receiver has reached 690kPa.g)

V_{rr}	The larger values of volume determined in Q3bii	22.988 m^3
V_{fo}	The compressed air needed by the factory	69.362 m^3/min
V_{fi}	The FAD of the compressor	63 m^3/min
P_1	Initial pressure	690 kPa.g.
Δt	Time difference	$1.434 min$
P_f	Atmospheric Pressure	101.3 kPa

By applying conservation of mass:

$$\Delta t = \frac{(P_2 - P_1)V_{rr}}{P_f(V_{fi} - V_{fo})}$$

$$1.434 = \frac{(P_2 - 690 \times 10^3)22.988}{101.3 \times 10^3(63 - 69.362)}$$

$$P_2 = 649.810\ kPa.g$$

Forge is **OFF** (6.666 min has passed)
Compressor is **ON** (pressure in the receiver has reached 690kPa.g)

V_{rr}	The larger values of volume determined in Q3bii	22.988 m^3
V_{fo}	The compressed air needed by the factory	52.62 m^3/min
V_{fi}	The FAD of the compressor	63 m^3/min
P_1	Initial pressure	649.810 kPa.g.
P_2	Final Pressure	790 kPa.g.
P_f	Atmospheric Pressure	101.3 kPa

By applying conservation of mass:

$$\Delta t = \frac{(P_2 - P_1)V_{rr}}{P_f(V_{fi} - V_{fo})}$$

$$\Delta t = \frac{(790 \times 10^3 - 649.810 \times 10^3)22.988}{101.3 \times 10^3(63 - 52.62)}$$

$$\Delta t = 3.065\ min$$

Forge is **OFF** (9.731 min has passed)
Compressor is **OFF** (pressure in the receiver has reached 790kPa.g)

V_{rr}	The larger values of volume determined in Q3bii	22.988 m^3
V_{fo}	The compressed air needed by the factory	52.62 m^3/min
V_{fi}	The FAD of the compressor	0 m^3/min
P_1	Initial pressure	790 kPa.g.
Δt	Time difference	$0.2681\ min$
P_f	Atmospheric Pressure	101.3 kPa

By applying conservation of mass:

$$\Delta t = \frac{(P_2 - P_1)V_{rr}}{P_f(V_{fi} - V_{fo})}$$

$$0.2681 = \frac{(P_2 - 790 \times 10^3)22.988}{101.3 \times 10^3(0 - 52.62)}$$

$$P_2 = 727.833\ kPa.g$$

Forge is **ON** (start of new cycle, 10 min has passed)
Compressor is **OFF** (pressure in the receiver has reached 790kPa.g)

V_{rr}	The larger values of volume determined in Q3bii	22.988 m^3
V_{fo}	The compressed air needed by the factory	69.362 m^3/min
V_{fi}	The FAD of the compressor	0 m^3/min
P_1	Initial pressure	727.833 kPa.g.
P_2	Final Pressure	690 kPa.g.
P_f	Atmospheric Pressure	101.3 kPa

By applying conservation of mass:

$$\Delta t = \frac{(P_2 - P_1)V_{rr}}{P_f(V_{fi} - V_{fo})}$$

$$\Delta t = \frac{(690 \times 10^3 - 727.833 \times 10^3)22.988}{101.3 \times 10^3(0 - 69.362)}$$

$$\Delta t = 0.124\ min$$

Forge is **ON** (10.124 min has passed)
Compressor is **ON** (pressure in the receiver has reached 690kPa.g)

V_{rr}	The larger values of volume determined in Q3bii	22.988 m^3
V_{fo}	The compressed air needed by the factory	69.362 m^3/min
V_{fi}	The FAD of the compressor	63 m^3/min
P_1	Initial pressure	690 kPa.g.
Δt	Time difference	$1.5432\ min$
P_f	Atmospheric Pressure	101.3 kPa

By applying conservation of mass:

$$\Delta t = \frac{(P_2 - P_1)V_{rr}}{P_f(V_{fi} - V_{fo})}$$

$$1.5432 = \frac{(P_2 - 690 \times 10^3)22.988}{101.3 \times 10^3(63 - 69.362)}$$

$$P_2 = 646.749\ kPa.g$$

Forge is **OFF** (11.667 min has passed)
Compressor is **ON** (pressure in the receiver has reached 690kPa.g)

V_{rr}	The larger values of volume determined in Q3bii	22.988 m^3
V_{fo}	The compressed air needed by the factory	52.62 m^3/min
V_{fi}	The FAD of the compressor	63 m^3/min
P_1	Initial pressure	646.749 kPa.g.
P_2	Final Pressure	790 kPa.g.
P_f	Atmospheric Pressure	101.3 kPa

By applying conservation of mass:

$$\Delta t = \frac{(P_2 - P_1)V_{rr}}{P_f(V_{fi} - V_{fo})}$$

$$\Delta t = \frac{(790 \times 10^3 - 646.749 \times 10^3)22.988}{101.3 \times 10^3(63 - 52.62)}$$

$$\Delta t = 3.132\ min$$

Forge is **OFF** (14.798 min has passed)
Compressor is **OFF** (pressure in the receiver has reached 790kPa.g)

V_{rr}	The larger values of volume determined in Q3bii	22.988 m^3
V_{fo}	The compressed air needed by the factory	52.62 m^3/min
V_{fi}	The FAD of the compressor	0 m^3/min
P_1	Initial pressure	790kPa.g.
Δt	Time difference	$0.2012\ min$
P_f	Atmospheric Pressure	101.3 kPa

By applying conservation of mass:

$$\Delta t = \frac{(P_2 - P_1)V_{rr}}{P_f(V_{fi} - V_{fo})}$$

$$0.2012 = \frac{(P_2 - 790 \times 10^3)22.988}{101.3 \times 10^3(0 - 52.62)}$$

$$P_2 = 743.346\ kPa.g$$

Forge is **ON** (start of new work cycle, 15 min has passed)
Compressor is **OFF** (pressure in the receiver has reached 790kPa.g)

V_{rr}	The larger values of volume determined in Q3bii	22.988 m^3
V_{fo}	The compressed air needed by the factory	69.362 m^3/min
V_{fi}	The FAD of the compressor	0 m^3/min
P_1	Initial pressure	743.346kPa.g.
P_2	Final Pressure	690 kPa.g.
P_f	Atmospheric Pressure	101.3 kPa

By applying conservation of mass:

$$\Delta t = \frac{(P_2 - P_1)V_{rr}}{P_f(V_{fi} - V_{fo})}$$

$$\Delta t = \frac{(690 \times 10^3 - 743.346 \times 10^3)22.988}{101.3 \times 10^3(0 - 69.362)}$$

$$\Delta t = 0.175\ min$$

Forge is **ON** (15.175 min has passed)
Compressor is **ON** (pressure in the receiver has reached 690kPa.g)

V_{rr}	The larger values of volume determined in Q3bii	22.988 m^3
V_{fo}	The compressed air needed by the factory	69.362 m^3/min
V_{fi}	The FAD of the compressor	63 m^3/min
P_1	Initial pressure	690 kPa.g.
Δt	Time difference	$1.49247\ min$
P_f	Atmospheric Pressure	101.3 kPa

By applying conservation of mass:

$$\Delta t = \frac{(P_2 - P_1)V_{rr}}{P_f(V_{fi} - V_{fo})}$$

$$1.49247 = \frac{(P_2 - 690 \times 10^3)22.988}{101.3 \times 10^3(63 - 69.362)}$$

$$P_2 = 648.171\ kPa.g$$

Forge is **OFF** (16.667 min has passed)
Compressor is **ON** (pressure in the receiver has reached 690kPa.g)

V_{rr}	The larger values of volume determined in Q3bii	22.988 m^3
V_{fo}	The compressed air needed by the factory	52.62 m^3/min
V_{fi}	The FAD of the compressor	63 m^3/min
P_1	Initial pressure	648.171 kPa.g.
P_2	Final Pressure	790 kPa.g.
P_f	Atmospheric Pressure	101.3 kPa

By applying conservation of mass:

$$\Delta t = \frac{(P_2 - P_1)V_{rr}}{P_f(V_{fi} - V_{fo})}$$

$$\Delta t = \frac{(790 \times 10^3 - 648.171 \times 10^3)22.988}{101.3 \times 10^3(63 - 52.62)}$$

$$\Delta t = 3.101\ min$$

Forge is **OFF** (19.768 min has passed)
Compressor is **OFF** (pressure in the receiver has reached 790kPa.g)

V_{rr}	The larger values of volume determined in Q3bii	22.988 m^3
V_{fo}	The compressed air needed by the factory	52.62 m^3/min
V_{fi}	The FAD of the compressor	0 m^3/min
P_1	Initial pressure	790 kPa.g.
Δt	Time difference	$0.2323\ min$
P_f	Atmospheric Pressure	101.3 kPa

By applying conservation of mass:

$$\Delta t = \frac{(P_2 - P_1)V_{rr}}{P_f(V_{fi} - V_{fo})}$$

$$0.2323 = \frac{(P_2 - 790 \times 10^3)22.988}{101.3 \times 10^3(0 - 52.62)}$$

$$P_2 = 736.134\ kPa.g$$

Forge is **ON** (20 min has passed)
Compressor is **OFF** (pressure in the receiver has reached 790kPa.g)

V_{rr}	The larger values of volume determined in Q3bii	22.988 m^3
V_{fo}	The compressed air needed by the factory	69.362 m^3/min
V_{fi}	The FAD of the compressor	0 m^3/min
P_1	Initial pressure	736.134 kPa.g.
P_2	Final Pressure	690 kPa.g.
P_f	Atmospheric Pressure	101.3 kPa

By applying conservation of mass:

$$\Delta t = \frac{(P_2 - P_1)V_{rr}}{P_f(V_{fi} - V_{fo})}$$

$$\Delta t = \frac{(690 \times 10^3 - 736.134 \times 10^3)22.988}{101.3 \times 10^3(0 - 69.362)}$$

$$\Delta t = 0.151\ min$$

Forge is **ON** (20.151 min has passed)
Compressor is **ON** (pressure in the receiver has reached 690kPa.g)

V_{rr}	The larger values of volume determined in Q3bii	22.988 m^3
V_{fo}	The compressed air needed by the factory	69.362 m^3/min
V_{fi}	The FAD of the compressor	63 m^3/min
P_1	Initial pressure	690 kPa.g.
Δt	Time difference	$1.5161\ min$
P_f	Atmospheric Pressure	101.3 kPa

By applying conservation of mass:

$$\Delta t = \frac{(P_2 - P_1)V_{rr}}{P_f(V_{fi} - V_{fo})}$$

$$1.5161 = \frac{(P_2 - 690 \times 10^3)22.988}{101.3 \times 10^3(63 - 69.362)}$$

$$P_2 = 647.509\ kPa.g$$

Forge is **OFF** (21.667 min has passed)
Compressor is **ON** (pressure in the receiver has reached 690kPa.g)

V_{rr}	The larger values of volume determined in Q3bii	22.988 m^3
V_{fo}	The compressed air needed by the factory	52.62 m^3/min
V_{fi}	The FAD of the compressor	63 m^3/min
P_1	Initial pressure	647.509 kPa.g.
P_2	Final Pressure	790 kPa.g.
P_f	Atmospheric Pressure	101.3 kPa

By applying conservation of mass:

$$\Delta t = \frac{(P_2 - P_1)V_{rr}}{P_f(V_{fi} - V_{fo})}$$

$$\Delta t = \frac{(790 \times 10^3 - 647.509 \times 10^3)22.988}{101.3 \times 10^3(63 - 52.62)}$$

$$\Delta t = 3.115\ min$$

Forge is **OFF** (24.782 min has passed)
Compressor is **OFF** (pressure in the receiver has reached 790kPa.g)

V_{rr}	The larger values of volume determined in Q3bii	22.988 m^3
V_{fo}	The compressed air needed by the factory	52.62 m^3/min
V_{fi}	The FAD of the compressor	0 m^3/min
P_1	Initial pressure	790 kPa.g.
Δt	Time difference	$0.218\ min$
P_f	Atmospheric Pressure	101.3 kPa

By applying conservation of mass:

$$\Delta t = \frac{(P_2 - P_1)V_{rr}}{P_f(V_{fi} - V_{fo})}$$

$$0.218 = \frac{(P_2 - 790 \times 10^3)22.988}{101.3 \times 10^3(0 - 52.62)}$$

$$P_2 = 739.45\ kPa.g$$

Forge is **ON** (25 min has passed)
Compressor is **OFF** (pressure in the receiver has reached 790kPa.g)

V_{rr}	The larger values of volume determined in Q3bii	22.988 m^3
V_{fo}	The compressed air needed by the factory	69.362 m^3/min
V_{fi}	The FAD of the compressor	0 m^3/min
P_1	Initial pressure	739.45 kPa.g.
P_2	Final Pressure	690 kPa.g.
P_f	Atmospheric Pressure	101.3 kPa

By applying conservation of mass:

$$\Delta t = \frac{(P_2 - P_1)V_{rr}}{P_f(V_{fi} - V_{fo})}$$

$$\Delta t = \frac{(690 \times 10^3 - 739.45 \times 10^3)22.988}{101.3 \times 10^3(0 - 69.362)}$$

$$\Delta t = 0.162\ min$$

Forge is **ON** (25.162 min has passed)
Compressor is **ON** (pressure in the receiver has reached 690kPa.g)

V_{rr}	The larger values of volume determined in Q3bii	22.988 m^3
V_{fo}	The compressed air needed by the factory	69.362 m^3/min
V_{fi}	The FAD of the compressor	63 m^3/min
P_1	Initial pressure	690 kPa.g.
Δt	Time difference	$1.5052\ min$
P_f	Atmospheric Pressure	101.3 kPa

By applying conservation of mass:

$$\Delta t = \frac{(P_2 - P_1)V_{rr}}{P_f(V_{fi} - V_{fo})}$$

$$1.5052 = \frac{(P_2 - 690 \times 10^3)22.988}{101.3 \times 10^3(63 - 69.362)}$$

$$P_2 = 647.814\ kPa.g$$

Forge is **OFF** (26.667 min has passed)
Compressor is **ON** (pressure in the receiver has reached 690kPa.g)

V_{rr}	The larger values of volume determined in Q3bii	22.988 m^3
V_{fo}	The compressed air needed by the factory	52.62 m^3/min
V_{fi}	The FAD of the compressor	63 m^3/min
P_1	Initial pressure	647.814 kPa.g.
P_2	Final Pressure	790 kPa.g.
P_f	Atmospheric Pressure	101.3 kPa

By applying conservation of mass:

$$\Delta t = \frac{(P_2 - P_1)V_{rr}}{P_f(V_{fi} - V_{fo})}$$

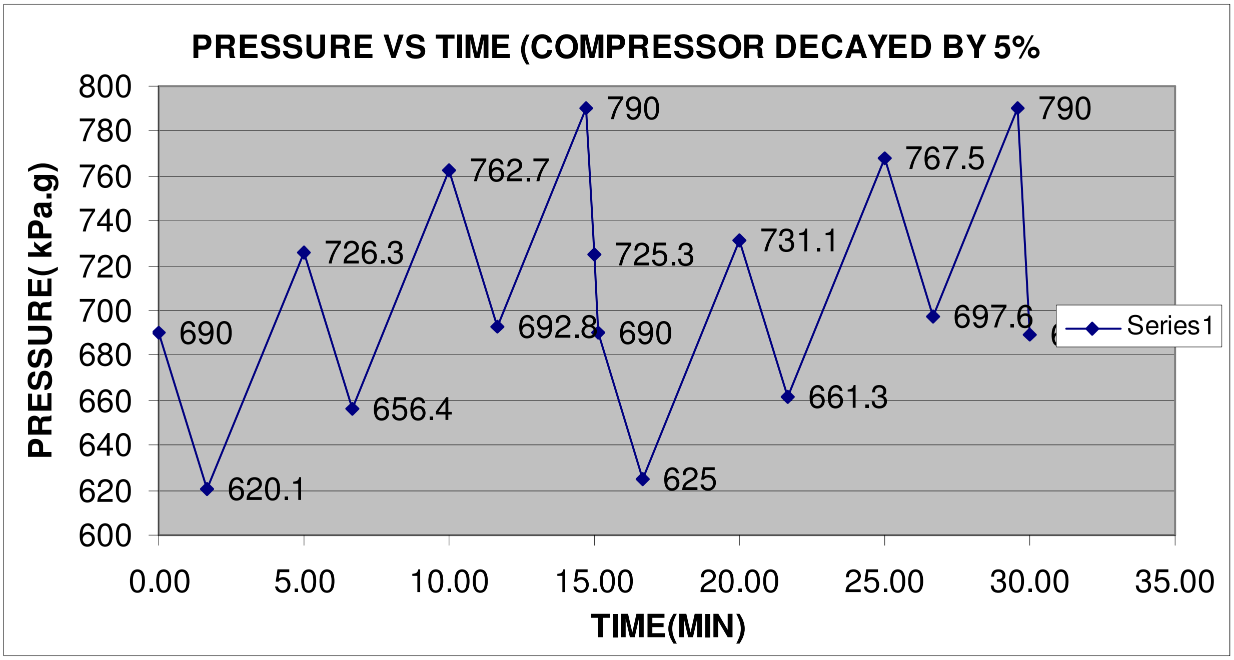
$$\Delta t = \frac{(790 \times 10^3 - 647.814 \times 10^3)22.988}{101.3 \times 10^3(63 - 52.62)}$$

$$\Delta t = 3.108\ min$$

Summary of New Compressor

Time(min)	Pressure(kPa.g	Forge status	Compressor status
0.00	690	ON	ON
1.67	643.3	ON	ON
4.88	790	OFF	OFF
5.00	761	OFF	ON
5.23	690	ON	ON
6.67	649.8	ON	ON
9.73	790	OFF	OFF
10.00	727.8	OFF	ON
10.13	690	ON	ON
11.66	646.7	ON	ON
14.80	790	OFF	OFF
15.00	743.3	OFF	ON
15.17	690	ON	ON
16.66	648.2	ON	ON
19.76	790	OFF	OFF
20.00	736.1	OFF	ON
20.14	690	ON	ON
21.66	647.5	ON	ON
24.77	790	OFF	OFF
25.00	739.5	OFF	ON
25.16	690	ON	ON
26.66	647.8	ON	ON
29.77	790	OFF	OFF
30.00	737.9	OFF	ON
Total ON/OFF for compressor for 30 min			6

In 15 mins the compressor starts 3 times
In 30 mins the compressor would have started 6 times
From the above data the compressor selected is suitable for the factory's usage and requirements.



Question 3b(ii)

The compressors' on time span and off time span both for the compressor decayed at 5%.

Forge is **ON** (start of cycle)
Compressor is **ON** (pressure in the receiver has reached 690kPa.g)

V_{rr}	The larger values of volume determined in Q3bii	22.988 m^3
V_{fo}	The compressed air needed by the factory	69.36 m^3/min
V_{fi}	The FAD of the compressor	59.85 m^3/min
P_1	Initial pressure	690 kPa.g.
Δt	Time difference	$1.667\ min$
P_f	Atmospheric Pressure	101.3 kPa

By applying conservation of mass:

$$\Delta t = \frac{(P_2 - P_1)V_{rr}}{P_f(V_{fi} - V_{fo})}$$

$$1.667 = \frac{(P_2 - 690 \times 10^3)22.988}{101.3 \times 10^3(59.85 - 69.36)}$$

$$P_2 = 620.140\ kPa.g$$

Forge is **OFF** (start of cycle, 1.667 min has passed)
Compressor is **ON** (pressure in the receiver has reached 690kPa.g)

V_{rr}	The larger values of volume determined in Q3bii	22.988 m^3
V_{fo}	The compressed air needed by the factory	52.62 m^3/min
V_{fi}	The FAD of the compressor	59.85 m^3/min
P_1	Initial pressure	620.140 kPa.g.
Δt	Time difference	$3.333\ min$
P_f	Atmospheric Pressure	101.3 kPa

By applying conservation of mass:

$$\Delta t = \frac{(P_2 - P_1)V_{rr}}{P_f(V_{fi} - V_{fo})}$$

$$3.333 = \frac{(P_2 - 620.140 \times 10^3)22.988}{101.3 \times 10^3(59.85 - 52.62)}$$

$$P_2 = 726.33\ kPa.g$$

Forge is **ON** (start of cycle, 5 min has passed)
Compressor is **ON** (pressure in the receiver has reached 690kPa.g)

V_{rr}	The larger values of volume determined in Q3bii	22.988 m^3
V_{fo}	The compressed air needed by the factory	69.36 m^3/min
V_{fi}	The FAD of the compressor	59.85 m^3/min
P_1	Initial pressure	726.33 kPa.g.
Δt	Time difference	$1.667\ min$
P_f	Atmospheric Pressure	101.3 kPa

By applying conservation of mass:

$$\Delta t = \frac{(P_2 - P_1)V_{rr}}{P_f(V_{fi} - V_{fo})}$$

$$1.667 = \frac{(P_2 - 726.33 \times 10^3)22.988}{101.3 \times 10^3(59.85 - 69.36)}$$

$$P_2 = 656.47\ kPa.g$$

Forge is **OFF** (6.667)
Compressor is **ON** (pressure in the receiver has reached 690kPa.g)

V_{rr}	The larger values of volume determined in Q3bii	22.988 m^3
V_{fo}	The compressed air needed by the factory	52.62 m^3/min
V_{fi}	The FAD of the compressor	59.85 m^3/min
P_1	Initial pressure	690 kPa.g.
Δt	Time difference	$3.333\ min$
P_f	Atmospheric Pressure	101.3 kPa

By applying conservation of mass:

$$\Delta t = \frac{(P_2 - P_1)V_{rr}}{P_f(V_{fi} - V_{fo})}$$

$$3.333 = \frac{(P_2 - 690 \times 10^3)22.988}{101.3 \times 10^3(59.85 - 52.62)}$$

$$P_2 = 762.666\ kPa.g$$

Forge is **ON** (start of cycle, 10 min has passed)
Compressor is **ON** (pressure in the receiver has reached 690kPa.g)

V_{rr}	The larger values of volume determined in Q3bii	22.988 m^3
V_{fo}	The compressed air needed by the factory	69.36 m^3/min
V_{fi}	The FAD of the compressor	59.85 m^3/min
P_1	Initial pressure	762.666 kPa.g.
Δt	Time difference	$1.667\ min$
P_f	Atmospheric Pressure	101.3 kPa

By applying conservation of mass:

$$\Delta t = \frac{(P_2 - P_1)V_{rr}}{P_f(V_{fi} - V_{fo})}$$

$$1.667 = \frac{(P_2 - 762.666 \times 10^3)22.988}{101.3 \times 10^3(59.85 - 69.36)}$$

$$P_2 = 692.801\ kPa.g$$

Forge is **OFF** (11.667 min has passed)
Compressor is **ON** (pressure in the receiver has reached 690kPa.g)

V_{rr}	The larger values of volume determined in Q3bii	22.988 m^3
V_{fo}	The compressed air needed by the factory	52.62 m^3/min
V_{fi}	The FAD of the compressor	59.85 m^3/min
P_1	Initial pressure	692.801 kPa.g.
P_2	Final Pressure	790 kPa.g.
P_f	Atmospheric Pressure	101.3 kPa

By applying conservation of mass:

$$\Delta t = \frac{(P_2 - P_1)V_{rr}}{P_f(V_{fi} - V_{fo})}$$

$$\Delta t = \frac{(790 \times 10^3 - 692.801 \times 10^3)22.988}{101.3 \times 10^3(59.85 - 52.62)}$$

$$\Delta t = 3.051\ min$$

Forge is **OFF** (14.718 min has passed)
Compressor is **OFF** (pressure in the receiver has reached 790kPa.g)

V_{rr}	The larger values of volume determined in Q3bii	22.988 m^3
V_{fo}	The compressed air needed by the factory	52.62 m^3/min
V_{fi}	The FAD of the compressor	0 m^3/min
P_1	Initial pressure	790 kPa.g.
Δt	Time difference	$0.279\ min$
P_f	Atmospheric Pressure	101.3 kPa

By applying conservation of mass:

$$\Delta t = \frac{(P_2 - P_1)V_{rr}}{P_f(V_{fi} - V_{fo})}$$

$$0.279 = \frac{(P_2 - 790 \times 10^3)22.988}{101.3 \times 10^3(0 - 52.62)}$$

$$P_2 = 725.306\ kPa.g$$

Forge is **ON** (15 min has passed)
Compressor is **OFF** (pressure in the receiver has reached 790kPa.g)

V_{rr}	The larger values of volume determined in Q3bii	22.988 m^3
V_{fo}	The compressed air needed by the factory	69.36 m^3/min
V_{fi}	The FAD of the compressor	0 m^3/min
P_1	Initial pressure	725.306 kPa.g.
P_2	Final Pressure	790 kPa.g.
P_f	Atmospheric Pressure	101.3 kPa

By applying conservation of mass:

$$\Delta t = \frac{(P_2 - P_1)V_{rr}}{P_f(V_{fi} - V_{fo})}$$

$$\Delta t = \frac{(790 \times 10^3 - 725.306 \times 10^3)22.988}{101.3 \times 10^3(0 - 69.36)}$$

$$\Delta t = 0.116\ min$$

Forge is **ON** (15.116 min has passed)
Compressor is **ON** (pressure in the receiver has reached 690kPa.g)

V_{rr}	The larger values of volume determined in Q3bii	22.988 m^3
V_{fo}	The compressed air needed by the factory	69.36 m^3/min
V_{fi}	The FAD of the compressor	59.85 m^3/min
P_1	Initial pressure	690 kPa.g.
Δt	Time difference	1.552 min
P_f	Atmospheric Pressure	101.3 kPa

By applying conservation of mass:

$$\Delta t = \frac{(P_2 - P_1)V_{rr}}{P_f(V_{fi} - V_{fo})}$$

$$1.552 = \frac{(P_2 - 690 \times 10^3)22.988}{101.3 \times 10^3(59.85 - 69.36)}$$

$$P_2 = 624.96\ kPa.g$$

Forge is **OFF** (16.667 min has passed)
Compressor is **ON** (pressure in the receiver has reached 690kPa.g)

V_{rr}	The larger values of volume determined in Q3bii	22.988 m^3
V_{fo}	The compressed air needed by the factory	52.62 m^3/min
V_{fi}	The FAD of the compressor	59.85 m^3/min
P_1	Initial pressure	624.96 kPa.g.
Δt	Time difference	$3.333\ min$
P_f	Atmospheric Pressure	101.3 kPa

By applying conservation of mass:

$$\Delta t = \frac{(P_2 - P_1)V_{rr}}{P_f(V_{fi} - V_{fo})}$$

$$3.333 = \frac{(P_2 - 624.96 \times 10^3)22.988}{101.3 \times 10^3(59.85 - 52.62)}$$

$$P_2 = 731.149\ kPa.g$$

Forge is **ON** (start of cycle, 20 min has passed)
Compressor is **ON** (pressure in the receiver has reached 690kPa.g)

V_{rr}	The larger values of volume determined in Q3bii	22.988 m^3
V_{fo}	The compressed air needed by the factory	69.36 m^3/min
V_{fi}	The FAD of the compressor	59.85 m^3/min
P_1	Initial pressure	731.149 kPa.g.
Δt	Time difference	$1.667\ min$
P_f	Atmospheric Pressure	101.3 kPa

By applying conservation of mass:

$$\Delta t = \frac{(P_2 - P_1)V_{rr}}{P_f(V_{fi} - V_{fo})}$$

$$1.667 = \frac{(P_2 - 731.149 \times 10^3)22.988}{101.3 \times 10^3(59.85 - 69.36)}$$

$$P_2 = 661.29\ kPa.g$$

Forge is **OFF** (21.667 min has passed)
Compressor is **ON** (pressure in the receiver has reached 690kPa.g)

V_{rr}	The larger values of volume determined in Q3bii	22.988 m^3
V_{fo}	The compressed air needed by the factory	52.62 m^3/min
V_{fi}	The FAD of the compressor	59.85 m^3/min
P_1	Initial pressure	661.29 kPa.g.
Δt	Time difference	3.333 min
P_f	Atmospheric Pressure	101.3 kPa

By applying conservation of mass:

$$\Delta t = \frac{(P_2 - P_1)V_{rr}}{P_f(V_{fi} - V_{fo})}$$

$$3.333 = \frac{(P_2 - 661.29 \times 10^3)22.988}{101.3 \times 10^3(59.85 - 52.62)}$$

$$P_2 = 767.479\ kPa.g$$

Forge is **ON** (start of cycle, 25 min has passed)
Compressor is **ON** (pressure in the receiver has reached 690kPa.g)

V_{rr}	The larger values of volume determined in Q3bii	22.988 m^3
V_{fo}	The compressed air needed by the factory	69.36 m^3/min
V_{fi}	The FAD of the compressor	59.85 m^3/min
P_1	Initial pressure	767.479 kPa.g.
Δt	Time difference	1.667 min
P_f	Atmospheric Pressure	101.3 kPa

By applying conservation of mass:

$$\Delta t = \frac{(P_2 - P_1)V_{rr}}{P_f(V_{fi} - V_{fo})}$$

$$1.667 = \frac{(P_2 - 767.479 \times 10^3)22.988}{101.3 \times 10^3(59.85 - 69.36)}$$

$$P_2 = 697.62\ kPa.g$$

Forge is **OFF** (26.667 min has passed)
Compressor is **ON** (pressure in the receiver has reached 690kPa.g)

V_{rr}	The larger values of volume determined in Q3bii	22.988 m^3
V_{fo}	The compressed air needed by the factory	52.62 m^3/min
V_{fi}	The FAD of the compressor	59.85 m^3/min
P_1	Initial pressure	697.62 kPa.g.
P_2	Final Pressure	790 kPa.g.
P_f	Atmospheric Pressure	101.3 kPa

By applying conservation of mass:

$$\Delta t = \frac{(P_2 - P_1)V_{rr}}{P_f(V_{fi} - V_{fo})}$$

$$\Delta t = \frac{(790 \times 10^3 - 697.62 \times 10^3)22.988}{101.3 \times 10^3(59.85 - 52.62)}$$

$$\Delta t = 2.9\ min$$

Forge is **OFF** (29.567 min has passed)
Compressor is **OFF** (pressure in the receiver has reached 790kPa.g)

V_{rr}	The larger values of volume determined in Q3bii	22.988 m^3
V_{fo}	The compressed air needed by the factory	52.62 m^3/min
V_{fi}	The FAD of the compressor	0 m^3/min
P_1	Initial pressure	790 kPa.g.
Δt	Time difference	0.4335 min
P_f	Atmospheric Pressure	101.3 kPa

By applying conservation of mass:

$$\Delta t = \frac{(P_2 - P_1)V_{rr}}{P_f(V_{fi} - V_{fo})}$$

$$0.4335 = \frac{(P_2 - 790 \times 10^3)22.988}{101.3 \times 10^3(0 - 52.62)}$$

$$P_2 = 689.48\ kPa.g$$

Forge is **ON** (start of new cycle, 30 min has passed)
Compressor is **OFF** (pressure in the receiver has reached 790kPa.g)

V_{rr}	The larger values of volume determined in Q3bii	22.988 m^3
V_{fo}	The compressed air needed by the factory	69.36 m^3/min
V_{fi}	The FAD of the compressor	0 m^3/min
P_1	Initial pressure	689.48 kPa.g.
P_2	Final Pressure	690 kPa.g.
P_f	Atmospheric Pressure	101.3 kPa

By applying conservation of mass:

$$\Delta t = \frac{(P_2 - P_1)V_{rr}}{P_f(V_{fi} - V_{fo})}$$

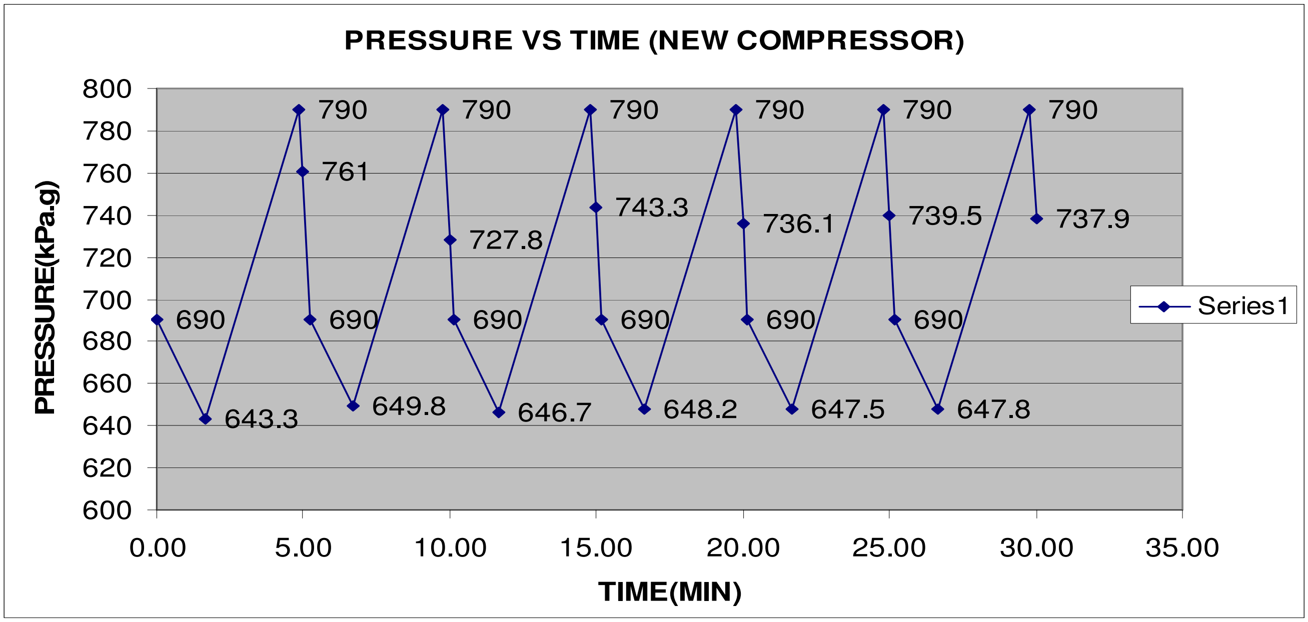
$$\Delta t = \frac{(690 \times 10^3 - 689.48 \times 10^3)22.988}{101.3 \times 10^3(0 - 69.36)}$$

$$\Delta t = 0.001\ min$$

Summary of Compressor Decayed by 5%

Time (min)	Pressure (Kpa.g	Forge Status	Compressor status
0.00	690	ON	ON
1.67	620.1	OFF	ON
5.00	726.3	ON	ON
6.67	656.4	OFF	ON
10.00	762.7	ON	ON
11.67	692.8	OFF	ON
14.71	790	OFF	OFF
14.99	725.3	ON	ON
15.11	690	ON	ON
16.66	625	OFF	ON
20.00	731.1	ON	ON
21.67	661.3	OFF	ON
25.00	767.5	ON	ON
26.67	697.6	OFF	ON
29.57	790	OFF	OFF
30.00	689.5	ON	ON
Total ON/OFF for compressor for 30 min			2

From the above data, the compressor selected is still suitable for the factory's usage and requirements.



Question 4a

Conversion of pound per square inch to kilopascal:

Max. working pressure of type **ER8** compressor: $\dfrac{125}{1.4510^{-4}} = 861.845\ KPa \cdot g \cdot$

Type **ER8** compressor FAD: $63.0\ m^3 / \min = \dfrac{63.0 \times 1000}{60} = 1050\ l/\sec$

Designed operating pressure regime:

Max. Operating pressure: 790 KPa.g.

Min. operating pressure: 690 KPa.g

Based on the catalogue data provided in the project resource material on p.3.3 and comparing it against the data stated above.

The most **suitable safety valve** is the one with the following specifications:

Set pressure: 827 KPa.g. Because < 861.845 KPa.g. And > 790 KPa.g.

Inlet size: 50 mm

Discharge area: 830 mm^2

Flow (FAD at 15 deg C): 1572 l/sec because > 1050 l/sec

Question 4b

(i) For the situation of a 'mild' fire, determine the required rate of discharge of a pressure activated relief valve.

Given : Wall of thickness of Vessel t=12mm (worst case)
Minimum operating temperature $T_0 = 12^o C\,(285K)$
Vessel made of low carbon manganese steel.

Using reference from AS1210, equation 8.6.2.4(2)
Overpressure protection initial flow of gas:
$m' = mY_t + m'_p$

1) Finding, Y_p where $C_w = 3550\,for\,steel,\,t = 12mm,\,T_0 = 285K\,(From\,AS1210)$

$$Y_p = \frac{10000}{C_w \cdot t \cdot T_0}$$

$$Y_p = \frac{10000}{3550}(12)(285) \quad = 8.237 \times 10^{-4}$$

2) Find m, where R=287 J/kgk (gas constant), $V_{rr} = 22.987m^3$, $T_0 = 285K$

Mass of air in the receiver,

$$P_{max} \times V_{rr} = m \times R \times T_0 \left(perfect\ gas\ law\right)$$
$$(790 + 101.3) \times (22.987) = m \times 0.287 \times 285$$
$$m = 250.49kg$$

3) Find m', $where\ R = 287\,\frac{J}{kgk}\,(gas\ constant),\,V_c = 1.05\,\frac{m^3}{s},\,T_0 = 285k$

$$P_{atm} \times V_c = m'_p \times R \times T_0 \left(perfect\ gas\ law\right)$$
$$(101.3) \times (1.05) = m'_p \times 0.287 \times 285$$
$$m'_p = 1.300\,\frac{kg}{s}$$

Therefore sub values of Y_p, $m\ and\ m'_p\ to\ find\ m'$.

$$m' = mY_p + m'_p$$
$$= (250.49) \times \left(8.237 \times 10^{-4}\right) + 1.3$$
$$m' = 1.506\,\frac{kg}{s}$$

Converting to free air volume flow rate,

(eqn. 8.6.2.3(2) in AS1210)
1) Where M is molecular weight of air = 28.97 kg/kmole.

2) K is the isentropic exponent of air,

$$K = \frac{C_p}{C_v} = 1.4$$

3) C is the constant of gas.(in AS1210)

$$C = 3.948\left[k\left[\frac{2}{k+1}\right]^{\frac{k+1}{k-1}}\right]^{\frac{1}{2}}$$

$$C = 3.948\left[1.4\left[\frac{2}{1.4+1}\right]^{\frac{1.4+1}{1.4-1}}\right]^{\frac{1}{2}} = 2.7$$

T_r is relieving temperature in Kelvin corresponding to:
1.21P + 0.1Mpa.abs = 1.21 x 861.845 + 100 = 1142.83 kPa
= 11.43 bar

from AS1210 Table 3.3.1(A)

therefore,

$$\frac{11.43 - 10.2}{12.54 - 10.2} = \frac{T_r - 180}{190 - 180}$$

$$T_r = 185.3^0\, C \; or \; 458.3K$$

Z is the compressibility of gas relieving conditions =1 (ideal gas)

Therefore,

$$V_f = 41.44 \times \frac{1.506}{2.7}\left[\frac{T_r,\,(1)}{28.97}\right]^{\frac{1}{2}}$$

$V_f = 91.94 \dfrac{m^3}{\min}$

$\boldsymbol{V_f = 1532.25 \dfrac{kg}{\sec}}$

4B (ii) (For the situation of a severe fire, determine the required rate of discharge of a pressure activated relief valve.

Given:
Wall thickness of vessel, t=12mm(worst case)
Minimum operating temperature $T_0 = 100^0 C\,(373K)$
Vessel made of low carbon manganese steel.

From the table 3.3.1(A) in the AS1210 and with the given temperature T_0, we found that the Design Tensile Strength 'f' is 115Mpa. Therefore the temperature relief valve opens at a pressure of,

$\dfrac{Z}{1.21} \cdot f$

$= \dfrac{1}{1.21} \cdot \left(115 \times 10^6\right)$

= 95.04MPa

Referring to table 3.3.1(A) in AS1210 and by interpolation to obtain T_r.

$\dfrac{99-79}{95.04-99} = \dfrac{400-425}{T_r - 400}$

$T_r = 404.95^0 C$

$T_r = 677.95K$

Using reference from AS1210 equation 8.6.2.4(2)

Over-temperature protection initial flow of gas:

$m' = mY_t + m'_p$

1) Find Y_t, $where\, C_w = 3550\, for\, steel,\, t = 12mm,\, T_r = 677.95K \left(from\, AS1210\right)$

$Y_t = \dfrac{110{,}000}{C_W \times t \times T_r}$

$$Y_t = \frac{110{,}000}{3550 \times 12 \times 677.5}$$

$$= 3.81 \times 10^{-3}$$

2) Find m, where R=287J/kgk (gas constant), $V_{rr} = 22.987m^3$, $T_0 = 285K$.

Mass of air in the receiver.

$$P_{\max} \times V_{rr} = m \times R \times T_0 \left(perfect\ gas\ law\right)$$
$$(790 + 101.3) \times (22.987) = m \times 0.287 \times 285$$
$$m = 250.49kg$$

3) Find m', $where\ R = 287\ \frac{J}{kgk}\ (gas\ constant), V_c = 1.05\ \frac{m^3}{s}, T_0 = 285k$

$$P_{atm} \times V_c = m_p' \times R \times T_0 \left(perfect\ gas\ law\right)$$
$$(101.3) \times (1.05) = m_p' \times 0.287 \times 285$$
$$m_p' = 1.300\ \frac{kg}{s}$$

Therefore sub values of Y_p, $m\ and\ m_p'\ to\ find\ m'$.

$$m' = mY_p + m_p'$$

$$= (250.49) \times \left(3.81 \times 10^{-3}\right) + 1.3$$
$$m' = 2.25\ \frac{kg}{s}$$

Converting to free air volume flow rate,

(eqn. 8.6.2.3(2) in AS1210)
1) Where M is molecular weight of air = 28.97 kg/kmole.

2) K is the isentropic exponent of air,

$$K = \frac{C_p}{C_v} = 1.4$$

3) C is the constant of gas.(in AS1210)

$$C = 3.948 \left[k \left[\frac{2}{k+1} \right]^{\frac{k+1}{k-1}} \right]^{\frac{1}{2}}$$

$$C = 3.948\left[1.4\left[\frac{2}{1.4+1}\right]^{\frac{1.4+1}{1.4-1}}\right]^{\frac{1}{2}} = 2.7$$

$$T_r = 404.95^0\, C\, or\, 677.95K$$

Z is the compressibility of gas relieving conditions =1 (ideal gas)

Therefore,

$$V_f = 41.44 \times \frac{2.25}{2.7}\left[\frac{677.95\,(1)}{28.97}\right]^{\frac{1}{2}}$$

$$V_f = 167.06\,\frac{m^3}{\min}$$

$$\boldsymbol{V_f = 2784.33\,\frac{kg}{\mathbf{sec}}}$$

Question 5

a) Select an after cooler to match the chosen compressor and confirm that it is suitable and safe to use.

From the table of water-cooled after coolers(Pg2.62 of resource material) the HD64 after cooler is selected because the specifications of ER8 two-stage compressor's free air delievery(FAD) of
$63.0 m^3$ /min and maximum working pressure of 8.8kg/ cm^3(refer to pg 2.26 of resource material) are within the specifications of HD64 after-cooler's free air delievery(FAD) of $63.0 m^3$ /min and maximum working pressure of 10.5kg/ cm^3. As stated in the water-cooled after coolers catalogue(refer to 22.62 of resource material), it is confirmed that HD64 after-cooler is suitable & safe to use under the conditions we believed in.

Since the after cooler's temperature of discharge air is above that of inlet water, therefore the temperature of air leaving it $=15^0 C + 10^0 C = 25^0 C$.

Therefore the temperature of air leaving it will be $\mathbf{25^0 C}$.

b) Determine what percentage of the hot water demand can be met by the heat scavenging.

From the resource material for the compressor the ER8 compressor needs minimum power of 450hp to power up the motor of the compressor

Converting hp to kW,

= 450hp x 0.746

= 335.7kW

Assuming large electric motor has 93% efficiency,

= 355.7 x 0.93

= 312.20kW

State Electricity Commission of Victoria advices that heat energy can be equivalent to a maximum of 80% of the compressor's electrical energy input when it is operating on load.

= 312.20 x 0.80

= 249.76kW

From the details given in question 1 the electroplating shop requires a volumetric flow rate of $9.0\,m^3$/hr of water at $55^0\,C$. Also, water enters the premises at $15^0\,C$. Density of water is 1000kg/m^3.

Converting mass flow rate,

$$9.0 \times \frac{P_{water}}{3600}$$

$$= 9.0 \times \frac{1000}{3600}$$

$$= 2.5 kg/\mathrm{s}$$

Therefore the heat energy needed to heat up the water at $60^0\,C$,

$$C_p\left(\textit{specific heat capacity of water}\right) \textit{ is } 4.18\,\frac{KJ}{Kg^0\,C}$$

$$= m \times C_p \times \Delta t$$

$$= 2.5 \times 4.18 \times (55 - 15)$$

$$= 418 kW$$

Heat energy recovered from compressor

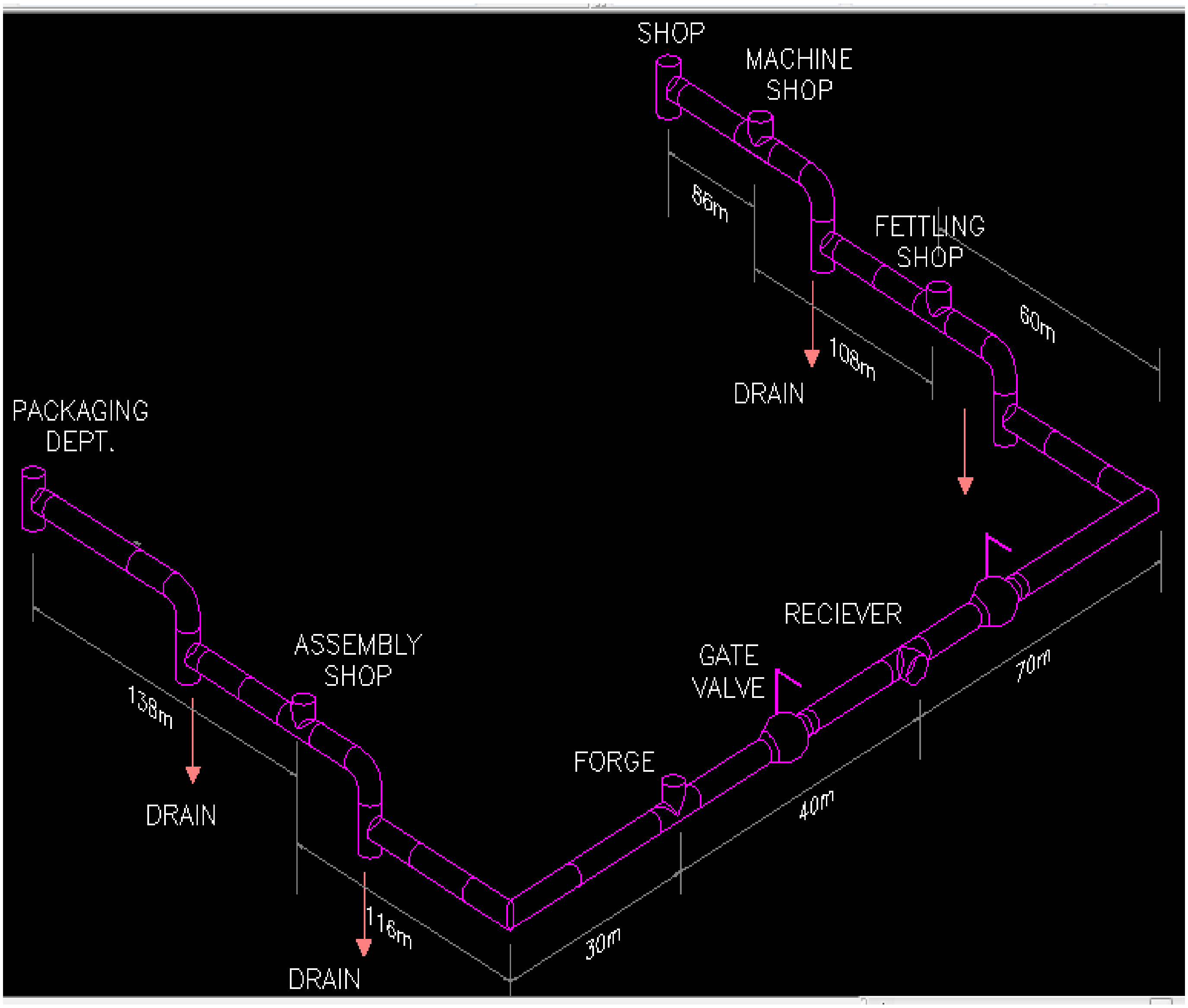
249.76 x 0.93 = 232.28 kW

Therefore, the percentage of hot water met by heat scavenging,

$$\frac{232.28}{418} \times 100\% = 55.6\%$$

Question 6A

Distribution of compressed air to the tools



Question 6b

From question 4, we have determined that the design pressure is 861.845Pa with the vessel temperature of $100^{0}C$ (373K). Referring to the resource material on page 4.22 table 6.7.6 AS1074 medium and
heavy tube, provides the following information:

For threaded joints:

In order to satisfy the design pressure and the temperature above, the suitable nominal size range from diameter 8mm to 40mm for both "medium" and "heavy" class. However when it comes into concern, "medium" class is preferred as they have a lower cost.

For Welded joints:

In order to satisfy the design pressure and the temperature above, the suitable nominal size range from 40mm and above. It can be used for both "medium" and "heavy" class.

In order to prevent corrosion, steel pipe to AS1074 with galvanized provides protection from corrosion caused by condensation. It is also the easiest to be welded and the relative roughness:

$\varepsilon = 0.15mm$ (Refer to page 418 of reference text, Engineering Fluid Mechanics 7^{Th} Edition, Clave Elger Roberson)

Question 6d

Calculating the Fluid Flow rate and internal diameter for each section

Partner 1: Gary **N**adhan (3181**8**00)

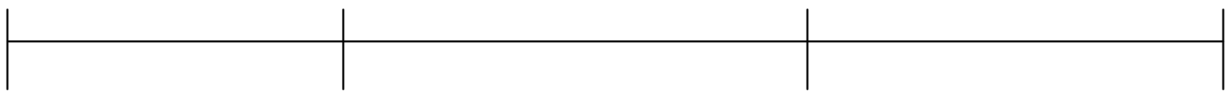
Partner 2: Ravi **S**hankar (318182**6**)

As allocated base on the student number:

X= 8

Y= 6

Paint shop "R" — Machine shop — Fettling shop



Section f — Section e — Section d

Friction pressure drop <= 20kPa

Design Specification

- Leakage at each work shop = 11.71 l/s (70l/s overall)
- Velocity of main line not exceeding 6m/s
- Frictional pressure drop not exceeding 20KPa
- Pressure at "R" = 620KPa
- Allowable pressure at main pipe, P=610KPa

Fettling Shop

Volume flow rate = FAD (fettling) + FAD (machine) + FAD (Paint) + leakage

= 329 + 35 + 180 +11.71

= 555.71 l/s

Using perfect gas law equation to find the actual or restraint volume flow rate, V_2

$$P_1 \times V_1 = P_2 \times V_2$$

$$101.3 \times 555.71 = 711.3 \times V_2$$

$$\mathbf{V_2 = 79.14 \frac{l}{s}}$$

Note that:

$P_1 = P_{atm}$

$= 101.3 kPa$

$V_1 = 555.71 \frac{l}{s}$

[Average frictional pressure drop if 10Kpa is 610kpa]

$P_2 = (610 + 101.3)\ Kpa$

$= 711.3\ kPa$

Referring to equation P12 in page 4.9 of resource material to determine the minimum bore size.

$V_2 = C \times A$

$$\frac{79.14}{1000} = 6\left(\frac{\pi \times d^2}{4}\right)$$

$d = 0.130m$

Referring to page 4.22 table 6.7.6 AS1074 medium and heavy tube.

Nominal size 150mm is selected

Outside diameter (OD) = 165.1mm

Pipe diameter (P) = 4.9mm

Inner diameter (ID) = 165.1-(4.9x2)

= 155.3mm

To check if selected pipe size satisfies the velocity and pressure drop requirement.

$V_2 = C \times A$

$$\frac{79.14}{1000} = C\left(\frac{\pi \times 0.1553^2}{4}\right)$$

$$\boldsymbol{C = 4.18 \frac{m}{s} \left(fulfilled\right)}$$

Machine Shop

Volume flow rate = FAD (machine) + FAD (Paint) + leakage

= 35 + 180 +11.71

= 226.71 l/s

Using perfect gas law equation to find the actual or restraint volume flow rate, V_2

$$P_1 \times V_1 = P_2 \times V_2$$

$$101.3 \times 226.71 = 711.3 \times V_2$$

$$\boldsymbol{V_2 = 32.28\,\frac{l}{s}}$$

Note that:

$$P_1 = P_{atm}$$

$$= 101.3kPa$$

$$V_1 = 226.71\,\frac{l}{s}$$

[Average frictional pressure drop if 10Kpa is 610kpa]

$$P_2 = (610 + 101.3)\ Kpa$$

$$= 711.3\ kPa$$

Referring to equation P12 in page 4.9 of resource material to determine the minimum bore size.

$$V_2 = C \times A$$

$$\frac{32.28}{1000} = 6\left(\frac{\pi \times d^2}{4}\right)$$

$$d = 0.0828m$$

Referring to page 4.22 table 6.7.6 AS1074 medium and heavy tube.

Nominal size 90mm is selected

Outside diameter (OD) = 101.6mm

Pipe diameter (P) = 4.0mm

Inner diameter (ID) = 101.6-(4.0x2)

= 93.6mm

To check if selected pipe size satisfies the velocity and pressure drop requirement.

$V_2 = C \times A$

$$\frac{32.28}{1000} = C\left(\frac{\pi \times 0.0936^2}{4}\right)$$

$$\boldsymbol{C = 4.69\,\frac{m}{s}\left(fulfilled\right)}$$

<u>Paint shop</u>

Volume flow rate = FAD (Paint) + leakage

= 180 +11.71

= 191.71 l/s

Using perfect gas law equation to find the actual or restraint volume flow rate, V_2

$P_1 \times V_1 = P_2 \times V_2$

$101.3 \times 191.71 = 711.3 \times V_2$

$$\boldsymbol{V_2} = 27.30\,\frac{\boldsymbol{l}}{\boldsymbol{s}}$$

Note that:

$P_1 = P_{atm}$

$= 101.3kPa$

$V_1 = 191.71\,\frac{l}{s}$

[Average frictional pressure drop if 10Kpa is 610kpa]

$P_2 = (610 + 101.3)\ Kpa$

$= 711.3\,kPa$

Referring to equation P12 in page 4.9 of resource material to determine the minimum bore size.

$V_2 = C \times A$

$$\frac{27.3}{1000} = 6\left(\frac{\pi \times d^2}{4}\right)$$

$d = 0.076m$

Referring to page 4.22 table 6.7.6 AS1074 medium and heavy tube.

Nominal size 80mm is selected

Outside diameter (OD) = 88.9mm

Pipe diameter (P) = 4.0mm

Inner diameter (ID) = 88.9-(4.0x2)

= 80.9mm

To check if selected pipe size satisfies the velocity and pressure drop requirement.

$$V_2 = C \times A$$

$$\frac{27.3}{1000} = C\left(\frac{\pi \times 0.0809^2}{4}\right)$$

$$\boldsymbol{C = 5.31\,\frac{m}{s}\left(fulfilled\right)}$$

<u>Pressure Loss Calculation for section (d), (e), (f)</u>

Using perfect gas law to determine the fluid density

P = 711.3kPa

$R_{air} = 287$ J/kgK

T = 15^0C + 273K = 288K

$$\rho = \frac{m}{v}$$

$$P_v = m \times R \times T$$

$$\rho = \frac{P}{R \times T}$$

$$\rho = \frac{711.3}{287}(15 + 273)$$

$$= \mathbf{8.606\,\frac{kg}{m^3}}$$

Section (d) fettling

$\rho = 8.606\,\frac{kg}{m^3}$

$\mu = 1.78 \times 10^{-5}$(from table A3, page A-12 of Engineering Fluids Mechanics textbook)

C = 4.18m/s

d = 155.3mm

$\varepsilon = 0.15mm$ (from page 418 of Engineering Fluids Mechanics textbook)

Reynolds number, $Re = \frac{\rho \times C \times d}{\mu}$

$$= \frac{8.606 \times 4.18 \times 0.1553}{1.78 \times 10^{-5}}$$

$$= \mathbf{3.14 \times 10^{5}}$$

Relative Roughness $= \frac{\varepsilon}{d}$

$$= \frac{0.15}{155.3}$$

$$= \mathbf{9.659 \times 10^{-4}}$$

Using the Moody's diagram on Engineering Fluids Mechanics textbook page 418

Resistance coefficient, $f = 0.025$

Applying Daray-Weisbach equation on fraction pressure loss

Where L is 60m

$$\Delta P_f = \frac{f \times L \times \rho \times V^2}{2d}$$

$$\Delta P_f = \frac{0.025 \times 60 \times 8.606 \times 4.18^2}{2 \times 0.1553}$$

$$= \mathbf{726.2 Pa}$$

Section (e) Machine Shop

$\rho = 8.606\,\dfrac{kg}{m^3}$

$\mu = 1.78 \times 10^{-5}$(from table A3, page A-12 of Engineering Fluids Mechanics textbook)

C = 4.69m/s

d = 93.6mm

$\varepsilon = 0.15mm$ (from page 418 of Engineering Fluids Mechanics textbook)

Reynolds number, $Re = \dfrac{\rho \times C \times d}{\mu}$

$$= \frac{8.606 \times 4.69 \times 0.0936}{1.78 \times 10^{-5}}$$

$$= \mathbf{2.12 \times 10^{5}}$$

Relative Roughness $= \dfrac{\varepsilon}{d}$

$$= \frac{0.15}{93.6}$$

$$= \mathbf{0.0016}$$

Using the Moody's diagram on Engineering Fluids Mechanics textbook page 418

Resistance coefficient, $f = 0.029$

Applying Daray-Weisbach equation on fraction pressure loss

Where L is 108m

$$\Delta P_f = \frac{f \times L \times \rho \times V^2}{2d}$$

$$\Delta P_f = \frac{0.029 \times 108 \times 8.606 \times 4.69^2}{2 \times 0.0936}$$

$$= \mathbf{3167.1\,Pa}$$

Section (f) Paint Shop

$\rho = 8.606 \dfrac{kg}{m^3}$

$\mu = 1.78 \times 10^{-5}$(from table A3, page A-12 of Engineering Fluids Mechanics textbook)

C = 5.31m/s

d = 80.9mm

$\varepsilon = 0.15mm$ (from page 418 of Engineering Fluids Mechanics textbook)

Reynolds number, $Re = \dfrac{\rho \times C \times d}{\mu}$

$$= \frac{8.606 \times 5.31 \times 0.0809}{1.78 \times 10^{-5}}$$

$$= \mathbf{2.08 \times 10^5}$$

Relative Roughness $= \dfrac{\varepsilon}{d}$

$$= \frac{0.15}{80.9}$$

$$= \mathbf{0.0019}$$

Using the Moody's diagram on Engineering Fluids Mechanics textbook page 418

Resistance coefficient, $f = 0.03$

Applying Daray-Weisbach equation on fraction pressure loss

Where L is 86m

$$\Delta P_f = \frac{f \times L \times \rho \times V^2}{2d}$$

$$\Delta P_f = \frac{0.03 \times 86 \times 8.606 \times 5.31^2}{2 \times 0.0809}$$

$$= \mathbf{3869.3\,Pa}$$

Pressure loss through transitions and fittings

1 x Tee joint through flow, $Kt = 0.4$

1 x Gate valve (wide open), $Kv = 0.2$

1 x 90^0 milter bend with vanes, $Ke = 0.2$

2 x Contraction D2/D1, $Kc = 0.06$

2 x Tee joint side outlet flow, $Kt = 1.8$

Pressure Loss, $\Delta P_f = \dfrac{K \times \rho \times C^2}{2}$

Section(d) fettling

Total K = 0.4+0.2+0.2+0.06

= **0.86**

$$\Delta P_f = \frac{0.86 \times 8.606 \times 4.18^2}{2}$$

= **64.7 Pa**

Section(e) Machine shop

Total K = 0.86+1.8+0.06

= **2.72**

$$\Delta P_f = \frac{2.72 \times 8.606 \times 4.69^2}{2}$$

= **257.4 Pa**

Section(f) Paint shop

Total K = 2.72+1.8

= **4.52**

$$\Delta P_f = \frac{4.52 \times 8.606 \times 5.31^2}{2}$$

= **548.4 Pa**

<u>Total Pressure loss</u>

The total pressure loss in section (d),(e) and (f) is

$$\Delta P_f = (726.2 + 64.7) + (3167.1 + 257.4) + (3869.3 + 548.4)$$

= 8633.1 Pa

Taking Pulsing flow due to water hammering into consideration, the total pressure loss shall be multiplied by 150%

$$\Delta P_f = 8633.1 \times 1.5$$

= 12949.65 Pa or 12.95 kPa

The total pressure loss in section (d),(e) and (f) is 12.95kPa, which is less than 20kPa, thus the nominal bore size of the pipes can be used.

Summary

Overall air consumption

Including Forge = 1156 l/s = 69.36 m^3/min

Excluding Forge = 877 l/s = 52.62 m^3/min

Selected Compressor

Type = ER8

Delivery capacity when NEW = 63.0 m^3/min

Delivery capacity when decayed 5% = 59.85 m^3/min

Volumetric Efficiency

η_v = 85.481%

Volume of receiver needed

V_{rr} = 22.987 m^3

Total number of ON/ OFF time for a span of 30 minutes

When compressor is NEW = 6 occurrence

When compressor has Decayed 5% = 2 occurrence

Pressure relief valve specifications

Set Pressure = 827 KPa.g

Inlet Diameter = 50mm

Discharge = 830 mm^2

Discharge rate at 15^{0}C = 1572 l/s

Required rate of discharge of a pressure activated relief valve

During the situation of Mild Fire =1532.25 Kg/s

During the situation of Severe Fire = 2784.33 Kg/s

Selected after cooler

Type = HD64

Delivery capacity = 63.0 m^3/min

Discharged air temperature

Input temperature = 15^{0}C

Output temperature = 25^{0}C

Electroplating hot water demand

Percentage of heat that can be scavenge = 55.6%

Range of sizes for fittings(nominal sizes)

Class = Medium and Heavy Tube

Threaded Joints = 8mm to 40mm diameter

Welded joints = 8mm to 150mm diameter

Nominal pipe size

Section D (Fettling)	Nominal Bore size	= 150mm
	Outside diameter	= 165.1mm
	Pipe Thickness (medium)	= 4.9mm
	Inner diameter	= 155.3mm
Section E (Machine shop)	Nominal Bore size	= 90mm
	Outside diameter	= 101.6mm
	Pipe Thickness (medium)	= 4.0mm
	Inner diameter	= 93.6mm
Section F (Packaging)	Nominal Bore size	= 80mm
	Outside diameter	= 88.9mm
	Pipe Thickness (medium)	= 4.0mm
	Inner diameter	= 80.9mm

Summary of pressure

Section D (fettling)	Pressure in pipe	= 726.2 Pa
Section E (machine shop)	Pressure in pipe	= 3167.1 Pa
Section F (Paint)	Pressure in pipe	= 3869.3 Pa
Total	Pressure in pipe	= 7762.6 Pa

Pressure drop through transitions and fittings

Section D (fettling)	Pressure drop	= 64.7 Pa
Section E (machine shop)	Pressure drop	= 257.4 Pa
Section F (Paint)	Pressure drop	= 548.4 Pa
Total	Pressure drop	= 870.5 Pa

Conclusion

Section (a) of this project has widened our knowledge on pressure vessels and it's uses. We have learnt how to find out and calculate suitable compressors and receiver sizes for different types of uses and also finding out its suitable after-coolers. The many nights spent together on doing this project with my partner have brought closer our friendship and through assigning different tasks to each other, this project also builds up our teamwork. Completing section (a) of this project has boosted our morale and confidence in our work and thus prepares us for future challenges.

References

1. RMIT student resource material volume 1 and 2.
2. Engineering Fluids Mechanics (7^{th} Edition by Crowe,Elger and Roberson)
3. Australia Standard AS1210 – 1997 Pressure Valve

Considerations of Class and Testing of Vessel

PART B OF ASSIGNMENT
Question 1

a) **What weld efficiencies may be assumed on longitudinal welds in a cylinder if class 2A construction is adopted?**

From [Table 1.6], referring to the column ‘Class 2 Vessels’, sub-column ‘2A’, regarding longitudinal welds, we are given three choices for types of welding.

Double Welded Butt Joint or equivalent	Efficiency: 0.85
Single Welded Butt Joint with Backing Strips	Efficiency: 0.80
Single Welded Butt Joint without Backing Strips	Efficiency: 0.70

b) **Give the title and numbers of the two Australian standards that need to be referred to, to find out more about the testing of weld tests plates and to find out the required extend of non-destructive examination of welds on the vessel.**

The title and number of the two Australian Standards that need to be referred to, to find out more about the testing of weld tests plates and to find the required extend of non destructive examination of the welds on the vessel are listed below.

AS 3992
Boilers and Pressure Vessels – Welding and brazing qualification

AS 4037
Boilers and Pressure Vessels – Examination and Testing

c) **Determine the required hydrostatic test pressure given that it will be carried out at 20^0C.**
From [Table 3.3.1 (A)]: Design Tensile Strength. Under classification; Plate, Sheet and Strip; type: C, C-Mn; Material Specification: AS 1548;
Grade: 7-460R at 50^0C, the design tensile strength is stated at 115MPa. Since there is no record for below 50^0C, we assume that the design tensile strength is at 115 MPa too.

From Assessment Part A we have Chosen ER8 as out Compressor. It has a working pressure of 125 Psi.

Design Pressure of the Vessel (P)	=	125 Psi
	=	125 Psi x 6994 Pa
	=	0.86175 MPag

Using the formulae below

P_h = $1.5\,P \times f_h/f$

= $1.5\,(0.86175) \times (115\text{ MPa} / 115\text{ MPa})$

= 1.293 MPa

- P_h : Hydrostatic Test Pressure in MPa
- P : Design Pressure of the Vessel in MPa
- f_h/f : Lowest Ration (For materials of which the vessel is constructed of)

<u>Design Strength of Test Temperature (MPa)</u>
Design Strength at Design Temperature (MPa)

Consideration of Material Suitability

Question 2

a) Show that the design tensile strength Data provided is consistent with the mechanical properties data provided from BHP in Appendix 5A.

From [Table 3.3.1 (A)] for Carbon Manganese Steel to for material type AS1548 – 7 – 460R at temperature 500C, the design tensile strength is stated at 115 MPa. Referring to [Appendix A Cl A3.2 & A2], it explains that for design temperature up to and including 500C, the design strength is the lower of Rm/4 and Re/1.5.

Note:

- *Rm = Specified minimum tensile strength for the grade of material concerned at room temperature (tested in accordance with AS 1391 or equivalent)*

- *Re = Specified minimum yield strength for the grade of material concerned at room temperature (tested in accordance with AS 1391 or equivalent)*

In order to calculate this, we referred to Appendix 5A of the project resource material under mechanical properties; the minimum specified tensile strength is 460 MPa (Rm). The minimum specified yield strength is 305 MPa (Re).

Calculating

Rm/4 = 460 MPa/4 = 115 MPa < Re/1.5 = 203.33 MPa

Therefore

Rm/4 < Re/1.5

As mentioned earlier, the design strength is the lower of the two. Therefore design strength is 115 MPa. This compares consistently with the value stated in [Table 3.3.1 (A)] with design tensile strength of 115 MPa at 50^0C.

b)(i) What is the upper limit of design temperatures that may be used with this steel?

From [Cl 2.7.1], states that materials of pressure parts of the vessels shall not be used at operating temperatures in excess of the highest design temperature for which strengths are shown in [Table 3.3.1]. Referring to the table, the highest design temperature for which strength is shown is 475^0C with strength of 35 MPa.

(ii) What is the lower limit of design temperature that may be used with this steel when it is used in an as welded and non-impact tested assembly?

Referring to [Table 2.6.2] and noting [Note 2d], we have chosen to categorize our material under Curve B because of two reasons. Firstly AS 1548 Type 7 is considered a fine grain steel with grain refining elements added, Secondly it will be used as a non-impact tested assembly, otherwise meaning NO TEST. Referring to [Fig 2.6.2 (A)], for the shell and pipe with thickness ranging between 10 – 16 mm, the minimum design temperature is 25^0C. As for the Flange with thickness ranging between 22-48 mm, the minimum design temperature is -7^0C.

c) Check if the corrosion allowance complies with the minimum corrosion allowance permitted by the standard for compressed air.

Corrosion allowance	=	(34.X) x 0.125	where X=8
	=	(34.8) x 0.125	
	=	4.35 mm	

From [Cl 3.2.4.2], the allowable permitted minimum corrosion is stated at 0.75 mm. our calculated corrosion allowance is 4.35 mm. This thus satisfies the requirement.

Determination of Vessel Diameter, Length and Thickness

Question 3

(a) (i) Determine the optimum diameter using volume of metal and the length of the cylindrical portion and the thickness of the cylinder and heads.

From the assignment 2A, the volume of the receiver is found to be 22.988 m^3. The cylindrical pressure vessel with 2:1 semi ellipsoid heads is used and with the corrosion allowance for 34.8 years as found in question 2(c) and using a weld efficiency of 85% to find the volume of metal, length and thickness of cylinder and heads.

To determine optimum diameter D:

The equation developed for the volume of metal Vm found in equation. V56 of page 5.53 of the project resource material.

$$Vm = (4V/D + 1.13D^2)\ (PD/2nf + C)$$

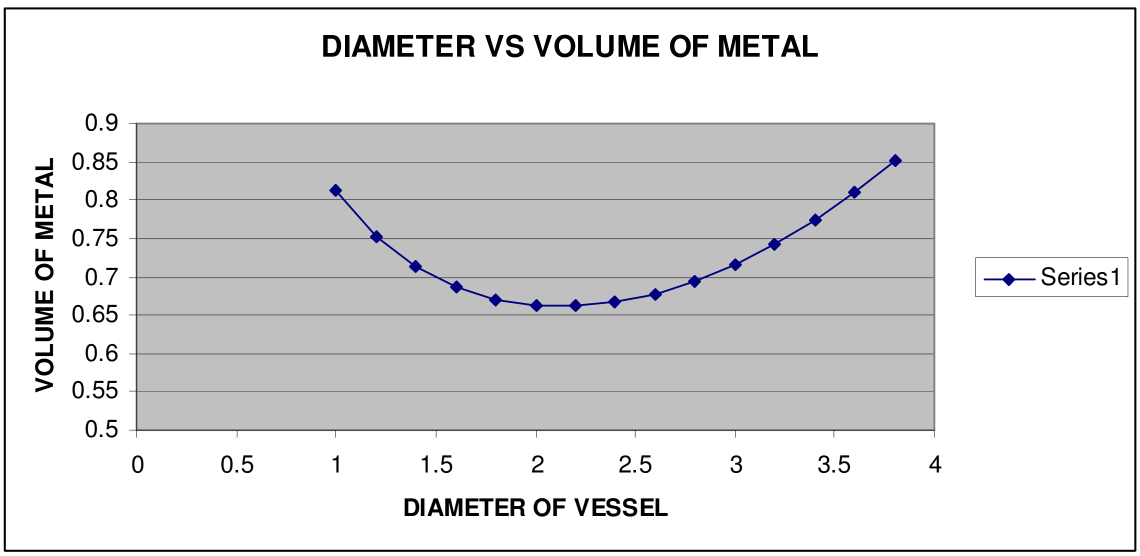
Where
n = 0.85, C = 4.35 mm, V = 22.98 m^3, f = 115 MPa, P = 0.86 MPa

$$Vm = \{(4(22.98)/D + 1.13\ D2\}\ \{\frac{(0.86 \times 10^6)\ D}{2(0.85)\ (115 \times 10^6)} + (4.35\}$$

$$Vm = 0.40367 + (0.04010/D) + 0.00496\ D^3 + 0.004916\ D^2$$

Therefore by using the derived formula of the volume of metal with respect of diameter of the vessel, a graph shall be plotted based on different diameter and finding the optimized one giving the least volume of metal.

VOLUME OF VESSEL D(m)	Diameter of metal,Vm
0.813546	1
0.752653253	1.2
0.712629886	1.4
0.68657112	1.6
0.670746782	1.8
0.663014	2
0.662095702	2.2
0.667219867	2.4
0.677925274	2.6
0.693950503	2.8
0.715167333	3
0.74153912	3.2
0.773093859	3.4
0.809906231	3.6
0.852085318	3.8



When D = 2.2 m, Vm is calculated as 0.6621 m^3, and it is using the latest amount of material metal. Sample calculation is shown below using D = 1.4 m.

$$Vm = 0.40367 + (0.4/2.2) + 0.00496\,(2.2)^3 + 0.004916\,(2.2)^2$$
$$= \mathbf{0.6621\ m^3}$$

Thickness of Cylinder using:
P = 0.86 MPa, D = 2.2 m, η = 0.85, C = 0.004350 m, f = 115 MPa

$$t = \{PD/\,(2\eta f - P)\} + C$$
$$= \{0.86(2.2)\}/\,\{2(0.85)(115)\} - (0.86) \quad + 0.004350$$
$$= \mathbf{0.01407m}$$

Thickness of head; Th, D= 2.2 m, K = 1, P = 0.86MPa, C = 0.004350

Th = {(PDK/2nf – 0.2P) + C}
= [(0.86)(2.2)(1)/{2(1)(115) – 0.2(0.86)} + 0.004350
= **0.01264 m**

Length of the cylinder

L = 4[Vr – 0.2618 D3]/π D2
= 4[(22.98) – 0.2618 $(2.2)^3$]/π $(2.2)^2$
= **5.31 m**

a)(ii) Determine the optimum diameter using the volume of metal and the length of the cylindrical portion and the thicknesses of the cylinder and heads when the restriction of 5.0 m is dictated. Comment of other issues that may affect a designer's choice of diameter and length, and comment on how the graph in (i) can help the designer in his/her deliberations on such issues.

Using D = 2.2m, the length was found to be 5.31 m, thickness of cylinder was 14.07m and thickness of head was 12.64 m. But referring to the question, there is a clause, which states that there is a height restriction in the factory, which dictates that the length of the vessel from the top of the upper semi-ellipsoid to the bottom of the other cannot exceed 7.0 meters. Due to this restriction, an internal diameter of 1.4 m would be insufficient. Referring back to the table of 'Heads of Pressure vessels, the next best would have to be an internal diameter of 2.134 m.

Using D = 2.134 m

Length of the cylinder

L = 4[Vr – 0.2618 D^3]/π D2
= 4[22.988 – 0.2618$(2.134)^3$]/π $(2.134)^2$
= 5.72 m

Thickness of the cylinder using:

P = 0.86 MPa, D = 2.134 m, η = 0.85, C = 0.004350 m, f = 115 MPa

t = {PD/(2nf – P)} + C
= {(0.86) (2.134)} / {(2) (0.85) (115) – (0.86)} + 0.004350
= 0.0138 m

Thickness of Head; Th, K = 1, D = 1.981 m, P = 0.86 MPa,
C = 0.004350, f = 115 MPa

$$Th = \{(PDK/2nf - 0.2P) + C\}$$
$$= [(0.86)(2.134)(1)/\{2(1)(115) - 0.2(0.86)\} + 0.004350$$
$$= \mathbf{0.0124\ m}$$

Therefore the internal diameter = 2.134 mm, length of straight Flange = 76mm

Total Length of the Vessel (L_{Total})
L_{Total} = L + 2[(internal height of head) + (Head Thickness) + (Flange length)
= 5.72 + 2[(0.5335) + (0.0124) + (0.076)]
= **6.9638m**

Since 6.9638m < 7m
This is acceptable.

The optimum specifications are as follows

Cylinder Diameter	=	2.134m
Cylinder Length	=	5.72m
Cylinder Thickness	=	13.8mm
Head Thickness	=	12.4mm

The other issues that may affect a designer's choice would probably be the design restrictions (i.e. space constraints height restrictions), safety standards, external loads (i.e. Wind, earthquake) location (i.e. indoor or outdoor) and corrosion factor.

The graph used in (i) explains how the diameter of vessel directly affects the volume of metal required. Properties such as weld efficiencies, corrosion factor and design pressure adversely affects the choice.

b) Determine the required thickness of the cylinder and head (including corrosion allowance). Refer to appendix 5A of the project resource material and select semi-ellipsoidal heads and sheet sizes that will give close to your optimal dimensions.

Referring to section 5.6.4 and Appendix 5A of the project resource material, the thickness of the semi-ellipsoidal head of 12mm is selected based on the internal diameter of 2134mm.

Circumference of the cylinder	=	π x D
	=	π (2134)
	=	6704.2 mm

Referring to Appendix 5a, quality of 2 pieces of sheet metal with length of 3200mm and a thickness of 16mm shall be used for the construction of the cylinder vessel section.

3 x 2300 = 6900 mm > 6704.2 mm

From [Cl 3.7 and 3.12] using equations found in [Cl 3.7.3 Cylindrical Shell]

Based on circumferential stress (longitudinal joints)

Th = {(PDK/2nf – 0.2P) + C}
= [(0.86) (2.134) / {2(0.85) (115) – (0.86)} + 0.004350

= 0.0138 m

Based on Longitudinal stress (circumferential joints)

Th = {(PDK/4nf – 0.2P) + C}
= [(0.86) (2.134) / {4(0.85) (115) – (0.86)} + 0.004350

= 0.00905 m

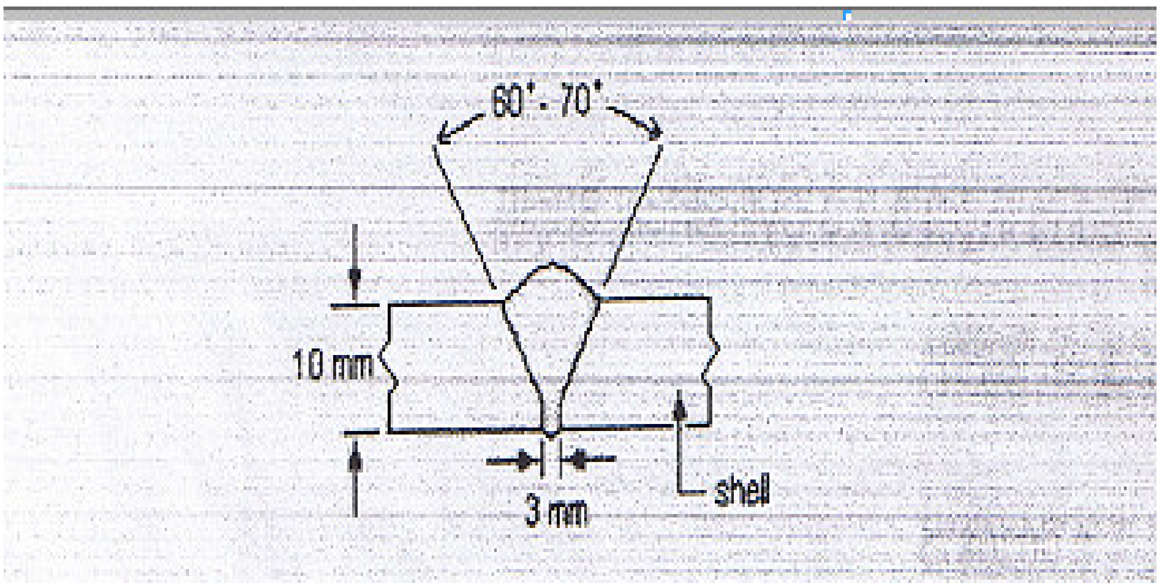
The thickness of 16mm selected is greater than both thicknesses calculated based on the circumferential and longitudinal stress acting on the cylinder.

Longitudinal and Circumferential Welds

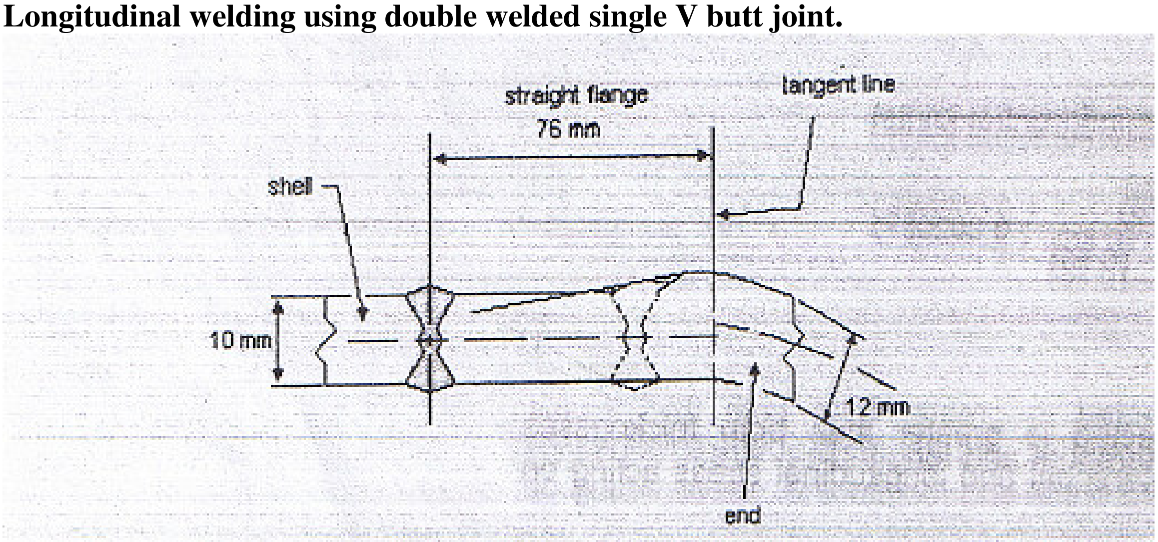
Question 4

(i) Decide on an appropriate type of weld for the longitudinal and circumferential seams given then a weld efficiency of 85% is to be achievable and provide a dimensional cross sectional sketch.

As done in question 1, we have identified three types of welds that can be used for longitudinal welding. The only one that can be used within an efficiency of 0.85% is double welding butt joint.



Longitudinal welding using double welded single V butt joint.



Circumferential welding using double-welded double V butt joint.

Inspection Opening

Question 5

a)It is proposed to have one-person hole for the inspection opening. Is one such opening sufficient? Is this a sufficient large inspection opening size for a vessel of this size? Can a hole of this size be dealt with by the rules for reinforcement given in AS1210?

From Question 3, we found that our vessel internal diameter is **2134mm** and according to the table 3.20.4 (Inspection openings for general purpose vessels) states that the inside diameter of the vessel > 1500mm, the minimum number of openings is one for shells of any length and minimum clearance size of openings elliptical manhole or equivalent.

Therefore one manhole opening is sufficient for the selected inside diameter of 2134mm of the vessel.

The proposed inspection opening is to be via a short pipe of 610+Y mm outside diameter. Where Y = 6

Therefore D = 610 + 6 = **616mm**

It is stated in [Cl 3.18.4.1] that the application of the requirements in the AS standard for the reinforcement of openings is intended to cover the following.

1) For Vessel equal to or less than 1500mm inside diameter: one half of the vessel diameter but not to exceed 500mm.
2) For Vessel greater than 1500mm inside diameter: one third of the vessel diameter but not to exceed 1000mm.

The vessel inside diameter is 2134mm. One third of the diameter is 660mm.
Therefore according to (b), the inspection opening size is sufficient for the vessel.
616 < 711.33mm

(ii) Determine the minimum branch thickness permitted and select an appropriate pipe thickness from appendix 6C of project resource material.

In the [Cl 3.19.10.2], Branch thickness

The minimum thickness of the branch after fabrication, up to the connection to the external piping shall be the greater of –

a) The thickness to withstand the calculation pressure and other loadings plus corrosion and
d) The smaller of the required thickness of the vessel wall (including allowances in clause 3.4.2) at the point of attachment, and the following thickness plus corrosion.

Outside diameter, mm	21.3	26.7	33.4	48.3	60.3	88.9	114.3	168.3	219.1	273	>273
DN	15	20	25	40	50	80	150	150	200	250	>250
Minimum thickness, mm	2.4	2.5	2.9	3.2	3.4	4.8	5.2	6.2	7.1	8.1	8.3

Therefore with an outside diameter of 616mm, the thickness given in the table above is 8.3mm and adding the corrosion allowances of 4.35mm, the minimum thickness permitted is 12.65mm.

From appendix 6C of the project resource material, the ANSI pipe size data is given for selection of the thicknesses of ANSI B36.10.

The nearest pipe size to the outside diameter of 616mm is 610 mm and there are two types.

Standard : 9.53mm
Xstrong : 12.7mm

Therefore the selection is XStrong **<u>12.7mm</u>** to accommodate the minimum thickness of 12.65mm.

c)(i) Calculate the 'missing' area A to be compensated for the inspection opening.

In 3.18.7.2, The total cross sectional area of reinforcement A, in square millimeters, required in any given plane for a vessel subject to internal pressure shall be no less than the following.

$A \quad = \quad dtF + 2T_{bl}tF\,(1-f_{rl})$

Given that the branch material is the same as the vessel and therefore has the same design tensile strength. (f_{r1} = 1). F is a correction factor and it is a value of 1 for all configurations. Therefore the equation had been simplified to the following.

$A \quad = \quad dtF$

Where t is the thickness of vessel, P = 0.86MPa, D = 1981mm, f = 115MPa

Th = 2PD/(2nf – P)

Th = (0.86 x 2134)/{2(1)(115) – 0.86}

Th = 8.04mm

Where d is the diameter of the finished opening +2 corrosion allowances.

D = 616 – (2x12.7) + (2 x 2.4.35)
D = 599.3mm

Therefore missing area A:

A = dfF
A = (599.3)(8.04)(1)
A = **4818.37 mm²**

(ii) Sketch in a A4, and include as part of the calculations, a cross section area of the pipe going through the vessel wall and also include a sketch showing weld dimensions.

The relevant thickness used in the sketch:

T1 = Normal Vessel wall thickness without corrosion allowances
= 16mm – 4.35mm
= 11.65mm
= Calculated vessel wall thickness of a seamless shell

Th = PD/(2nf – P)

Th = [(0.86 x 2134)/{2(1)(115) – 0.86}]

Th = 8.04 mm

Tb1 = nominal branch wall thickness without corrosion allowances
= 12.7 – 4.35
= 8.35 mm

Tb = calculated branch wall thickness of a seamless shell

Tb = 2PD/(2nf – P)

Tb = [(0.86 x 599.3)/{2(1)(115) – 0.86}]

= 2.249 mm

Tr1 = nominal thickness of reinforcing element.
(Recommended to be 0.75 to 1.5 times of the vessel wall thickness)

= (1)(16)
= 16.0 mm

Sketch of cross-sectional area of the vessel wall

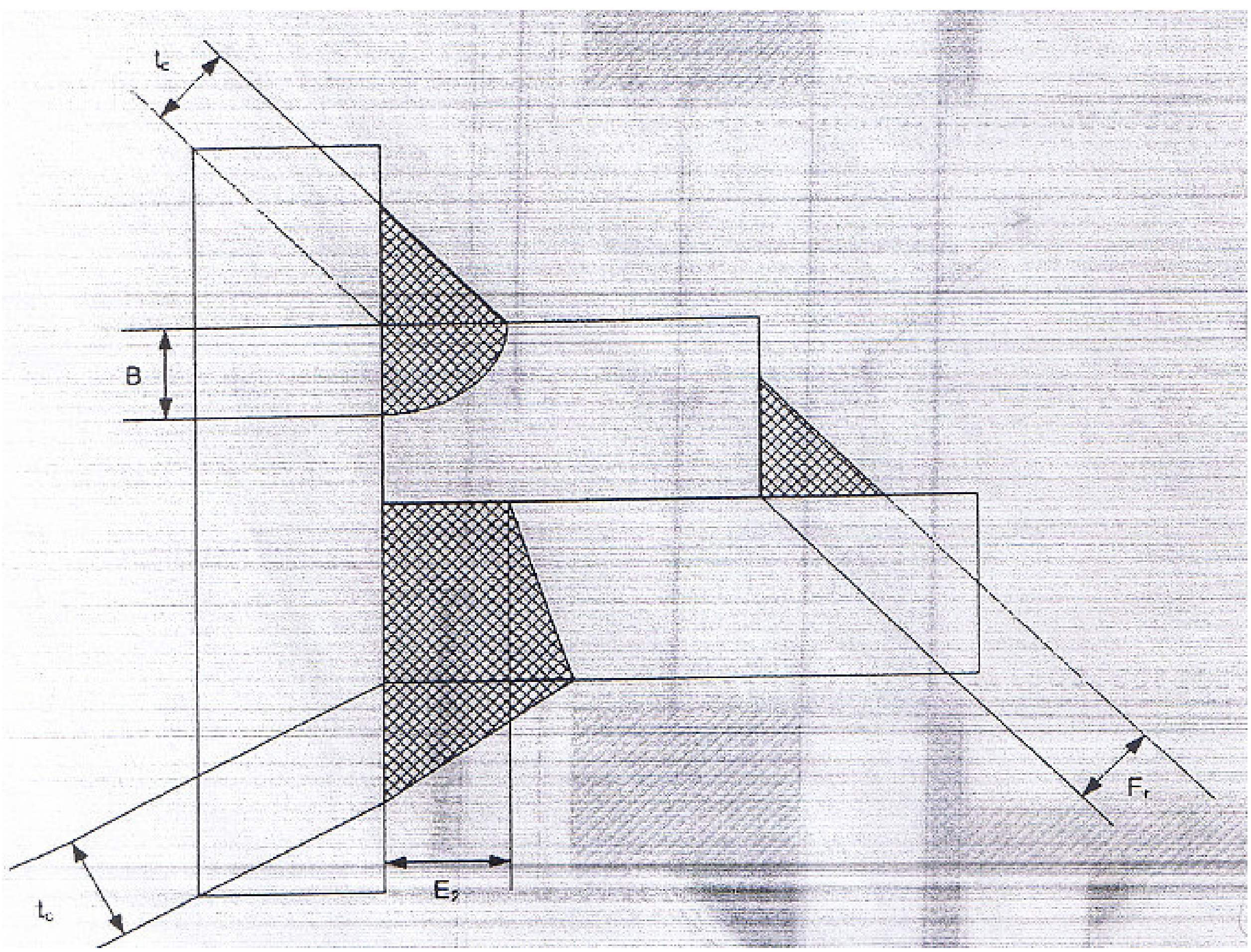
Given in figure 3.19.3 (C) guidelines for welding throat thickness

t_s = nominal thickness of vessel wall, in millimeters
t_n = nominal thickness of branch wall, less specified under tolerance, in millimeters
C = corrosion allowance, in millimeters
t = ts – C mm
tb = tn – C mm
tr = nominal thickness of height of reinforcing element, in millimeters

tc >= 0.7t, 0.7tb or 6 mm, whichever is least
0.7t = (0.7)(16 – 4.35) = 8.155 mm
0.7t b = (0.7)(12.7 – 4.35) = 5.845 mm
Therefore 0.7t can be used.

Fr >= 0.5t, 0.5t or 10 mm, whichever is the least, but sufficient to comply with clause 3.19.3.5

0.5tr = (0.5)(16) = 8mm
0.5t = (0.5)(16 – 4.35) = 5.825
Therefore 0.5t can be used.



(iii) Determine the limiting distances of the reinforcement, and the available reinforcing area in the wall (A1), in the branches (A2 and A3), and the welds (A4) and hence the cross sectional area (A5) of the reinforcing pad required. (if any).

Limits of Reinforcement

The limiting distance parallel to the vessel wall shall be at a distance, on each side of the axis of the opening, equal to the greater of:

a) The diameter of the finished opening plus twice the corrosion allowance
= 599.3 mm

b) The radius of the finished opening plus the corrosion
allowance plus the corroded thickness of the vessel wall, plus the thickness of the nozzle wall. [(0.5)(599.3) + (11.65+8.35)]=319.65mm

Therefore case (a) is used

The limiting distance normal to the vessel wall shall conform to the contour of the surface at a distance from each surface equal to the smaller of:

a) 2.5 times the nominal to the shell thickness less corrosion
Allowance =2.5(11.65)= 29.125 mm

b) 2.5 times the nozzle-wall thickness less corrosion allowance,
plus the thickness of added reinforcement exclusive of weld metal on the side of the shell under consideration.
= 2.5(8.35) +16= 36.875mm

Therefore case (a) is used

Reinforcing Metal

Reinforcing Area A1 shall be larger of the formulae in 3.18.10.4 (a)

A1 $= (nT - Ft)\, d - 2T_{b1}\, (nT1 - Ft)\, (1 - fn)$
A1 $= (11.65 - 8.04)\, (599.3) - (0)$
A1 $= 2163.47\ mm^2$

Reinforcing Area A2 shall be smaller of the formulae in 3.18.10.4 (b)

A2 $= (T_{bt} - t_b)\ 5\ T_1 f_2$
A2 $= (8.35 - 2.249)\ 5\ (11.65)\ (1)$
A2 $= 355.383\ mm^2$

Reinforcing Area A3

A3 $= 2(Tb1 - C) \times 0.8 \times \sqrt{dTb1} - C$
A3 $= 2(8.35 - 4.35)\ 0.8 \times \sqrt{599.3} - (8.35 - 4.35)$
A3 = 626.7 mm2

Reinforcing Area A4 according to figure 3.19.3 (c):

A4 $= 4(Tc^2) + 2(Fr^2)$
A4 $= 4(8.155) + 2(5.828)^2$
A4 $= 333.88\ mm^2$

Total reinforcing Areas of A1+A2+A3+A4

$= 2163.47 + 355.383 + 626.7 + 333.88$
$= 3479.433\ mm^2$

Therefore the sum of these areas shall be greater than the missing area A calculated earlier. If not, a reinforcing ring, A5 will be necessary to compensate the missing area.

Missing area A = $\mathbf{4818.37\ mm^2}$

Total Reinforcing Area A1 to A4 = $3479.33 mm^2$

$4818.37\ mm^2 \quad > \quad 3479.433\ mm^2$

Therefore a reinforcing ring, A5 is needed.

A5 $= 4818.37 - 3479.433$
A5 $= 1338.94\ mm^2$

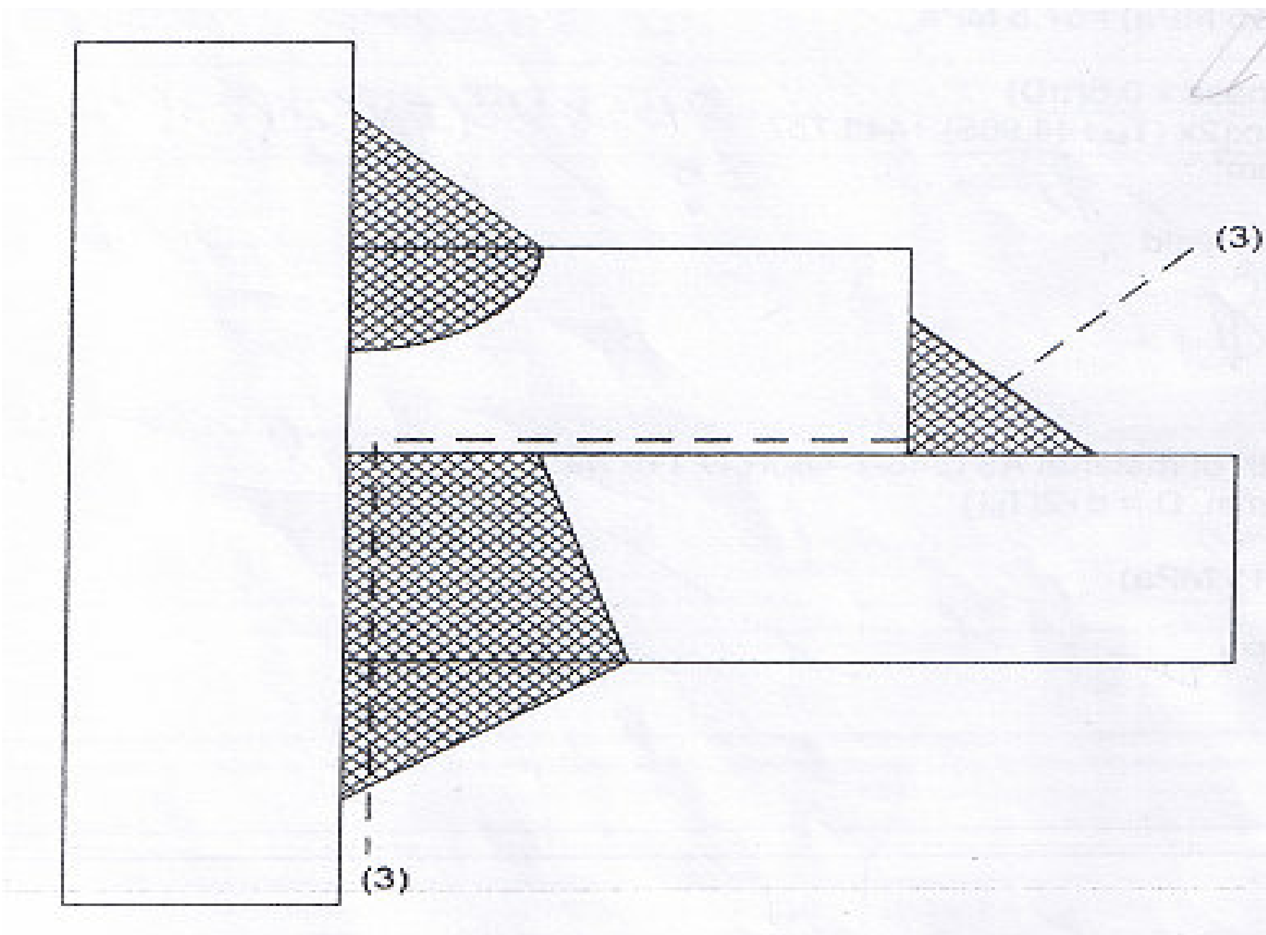
(iv) Determine the strength in the tension (in Newton's) of the combination of all the elements contributing reinforcement to the shell, whose contribution can only be made if the welds holding them to the shell are adequate.

The strength of the combination of elements contributing reinforcement to the shell except the reinforcement area in the vessel wall consists of the branch, the welds and the reinforcing ring.

Therefore the tension force = Area of contribution X design tensile strength of the material.

Ft = (A2+A3+A4+A5) f
= (355.383 + 626.7 + 333.88+1338.94) (115 x 10^6)
= 305.314KN

(v) Determine the strength (in Newton's) of the welds on the failure path (3)…. (3) On Fig 3.19.2 (c). Sketch the failure path as part of the calculations. Check that these welds combined strength is adequate.



Given in 3.19.3.5 the strength of a weld shall be based on the nominal throat of the weld times the length of the weld measured on its inner periphery times the maximum allowable stress of the weld. The allowable stress in the weld and in the component, which may be included in the possible path of failure, shall be the following percentage of the design tensile strength for the appropriate material. (See table 3.3.1)

(a) Fillet weld in end shear equals 50 percent end shear
(b) Butt weld in tension equals 74 percent
(c) Butt weld in shear equals 60 percent.
(d) Components in shear equal 70 percent.

Where the load on a weld varies from side to end shear or shear to tension, the lower of the above values shall be used.

From the design of the inspection opening, there are 2 forces holding the branch. One of a fillet weld and other of a butt shear weld.

Force produced by fillet weld

From Table 3.3.1, the design strength if material AS 1548 – 460 R is 115 MPa.

Where t_c = 8.155 mm, D = d + 2(T_{b1} +Length of the ring)

1) Fillet weld in end shear = (0.5)(115 MPa) = 57.5 MPa

2) Area of Fillet weld	= Throat Thickness x 0.5(πD)
	= 8.155 x 0.5{π(2x(8.35 + 63.25) + 599.3
	= 4278.015mm2
Therefore Ff	= Fillet strength x Area of the weld.
	= 57.5 MPa x 4278.015mm2
	= 245.985KN

Force produced by butt shear weld:

From Table 3.3.1 the design strength of material AS 1549 – 7 – 460R is 115 MPa
Where thickness of the vessel = 11.65 mm, $D = d + 2(T_{b1})$

3) Fillet weld in end shear	=	0.6(115 MPa)
	=	69.0 MPa
Area of the Fillet weld	=	Thickness x 0.5(πD)
	=	11.65(0.5{π(2x(8.35)+599.3
	=	11272.66 mm2
Therefore Fr	=	Fillet strength x area of weld.
	=	69 MPa x 11272.66
	=	777.813 KN

Sum of forces of reinforcement in the weld = 245.985+777.813
= 1023.8 KN

Therefore the sum of forces is greater than the force of reinforcement elements in (v) and it is adequate enough to hold the branch onto vessel wall.

(d) The inspection opening is to be in the cylindrical shell as low as is permitted to enable ease of access. Establish how long it may go.

From 3.18.5.1 states that the opening in the dished ends shall be located clear of structural discontinuities, e.g. supports, junction between conical and cylindrical section, by the least three times the shell or end thickness except where the design of the opening is proved adequate.

Therefore the safety distance to establish from the joints of the weld:

$$3T_1 = 3 \times 11.65$$
$$= 34.95 \text{ mm}$$

e) Select an appropriately sized slip on plate flange, blind plate and associated bolts.

The design pressure for this pressure vessel is found to be 0.86 MPa = 860 KPa and operates at a design temperature of 100^0C. With this information, the appropriate size of a slip on plate flange shall be selected. Referring to the Temperature/Pressure ratings table for steel flanges in chapter 6, Page 6.8 of resource material. Flange table **'E'** is selected which can withstand up to 1400 KPa and operates in a temperature range of -50^0C to 232^0C.

As also stated that the branch will have an outside diameter of 616mm and an inside diameter of 581.9 mm. From the commercial catalogue in Appendix 6C, on fittings of backwoods Flange Tables on page 6.56 of resource material, the nominal Bore diameter of 600mm is selected. It is the closest to the outside diameter of the branch. The flange will have to bore to the outside diameter of 616mm. Therefore using table E, the specification is of the following.

Standard	Table	Diameter of the Flange (mm)	PCD	No of Bolts	Diameter of bolts (mm)	Diameter of holes (mm)	Thickness of the flange (mm)
AS 2129	E	825	756	16	30	33	48

(f)(i) Check that the center spacing of the bolt on the pitch circle diameter is not too great. Assuming steel is the same of the vessel. And for properties of gasket materials of gasket materials when used on flanges with full face gasket.

Given that the pitch centre diameter is 756 mm.

Therefore the spacing of bolts from center to center.

Π (756) / 16 = 148.44 mm

From 3.21.4.1, the max center to center spacing of the blot with full-face gasket.

P_b max = $2D_b + 6t/ (m+0.5) (E/200000)^{1/4}$

Where Db = 30, t = 48, m = 0.8 (Table 3.21.11.4)
E = 197 GPa (At 100^0C)

Pb max = $2(30) + 6(48)/(0.8+0.5)((197 \times 10^9)/200000)^{1/4}$

Pb max = 208.72mm > 152.89mm acceptable

(F)(ii) Check that there is adequate spanner room there are some dimensions to considerate:

For checking of adequate spanner room there are some dimensions to considerate.

(a) The distance between the P.C.D to the inside diameter = (756 – 616)/2 = 70 mm

(b) The pitching between bolt holes = 148.44 mm

(c) The distance between the PCD to the outside diameter = (825 – 756)/2 = 34.5 mm

F)(iii) Determine the required bolt forces both for the operation condition, and for the gasket seating condition.

To find the bolt forces for both conditions, the equations from 3.21.11.2 will be used to find b, hg, h'g and G. And from the above, A = 825 mm, B = 616 mm, C = 756 mm.

B = effective gasket or joint-contact-seating width

b = (C-B)/4
b = (756-616)/4
b = 35 mm

hg = radial distance from the bolt circle to the reaction of that portion of the gasket-force between the bolt circle and the inside of the flange.

$hg \quad = (C - B)(2B + C)/6(B + C)$
$hg \quad = (756 – 616)\ (2 \times 616 + 756) / [6\ (616 + 756)]$
$hg \quad = 33.8$ mm

$h'g \quad$ = Radial distance from the bolt center to the reaction of that portion of the gasket force between the bolt circle and the outside of the flange,

$h'g \quad = (A – C)\ (2A + C) / 6\ (A + C)$
$h'g \quad = (825 – 756)\ (2 \times 825 + 756) / 6\ (825 + 756)$
$h'g \quad = 17.5$ mm

$G \quad$ = diameter at the location of that portion of the gasket reaction between the bolt circle and the internal diameter of the flange.

$G \quad = C – 2\ hg$
$G \quad = 756 – 2\ (33.81)$
$G \quad = 688.38$ mm

The required bolt force for the opening conditions W_{m1}

$W_{ml} \quad = H + Hp$
$\quad = 0.785\ G^2\ P \quad + 2b\pi GmP\ (1 + hg/h'g)$

$W_{ml} \quad = 0.785\ (0.688)^2\ (0.86 \times 10^6) + 2\ (0.0035)\ \pi\ (0.688)\ (0.8)\ (0.86 \times 10^6)$
$\quad x(1+ (0.0338/0.0175)$

$W_{ml} \quad = 350.068$ KN

The minimum initial bolt force for gasket seating-force W_{m2}

$W_{m2} \quad = b\pi Gy\ \{1 + (hg/h'g)$

Where y = 2.9 MPa (Table 3.21.11.4)

$W_{m2} \quad = (0.0035)\ \pi\ (0.688)\ (2.9 \times 106)(1 + \quad (0.0338/0.0175))$

$W_{m2} \quad = 643.107$ KN

f)(iv) Confirm the adequacy of the total bolt core area assuming precision metric steel bolts to AS 1110 of grade 5.8 or 8.8 are used.

In 3.21.6.4.3 states that the total cross-sectional area of bolts Am, required for both the operating conditions and gasket seating is the greater of the values for A_{m1} and A_{m2}. A selection of bolts to be used shall be made such that the actual total cross-sectional area of bolt A_{b1} will not be less than Am.

Therefore by using the greater of Wm, the greater of Am will be obtained. Wm2 of 643.107KN will be used. From then table 3.21.5 selecting steel bolts to AS1110 of grade 8.8, the design strength at atmosphere Sa and design Sb condition and at 100degree/celcius is 160Mpa

A_{m2} $= W_{m2} / Sa$

A_{m2} $= 643.107 / 160 \times 10^6$

A_{m2} $= 0.00402 \text{ m}^2$

From 3.21.5.3.6 the core area of the steel bolts to AS 1110 of grade 8.8 is determined from the table. With a nominal diameter of M24, the core area is 324 mm^2

Therefore the total core area for 16 bolts:

Ab $= 16 \times 324 \times 10^{-6}$
Ab $= 0.00518 \text{ m}^2$

It is confirmed that the total bolt core area is adequate due Ab > A_{m2}.

g(i) Regarding Flange stresses, determine the bolt force that may exist during seating and hence radial flange stress during gasket seating. Check that the stress is not excessive.

Using the equation 3.21.6.4.4(2), to find W

W $= (Am + Ab)\, Sa/2$
W $= (0.00402 + 0.00518)\,(160 \times 10^6) / 2$
W $= 736.0$ KN

Total hydrostatic end-force, H

H $= 0.785\, G^2P$
H $= 0.785\,(0.688)^2\,(0.86 \times 10^6)$
H $= 319.554$ KN

Using equation 3.21.11.5 to find total flange moment

Mo = $(W - H)/ \{(1/hg) + (1/h'g)\}$

Mo = $\dfrac{\{(736 \times 10^3) - (319.554 \times 10^3)\}}{\{(1/0.0338) + (1/0.0175)\}}$

Mo = 4801.7 Nm

Hence using the equation 3.21.11.6 (2), to find Radial flange stress during gasket seating.

Sr = $6Mo / t^2 (\pi C - nD)$

Sr = $6 (4801.7) / (0.048)^2 \{\pi (0.756 - (16) (0.030)\}$

Sr = 6..59 MPa

According to 3.21.11.7 the flange stress shall be no greater than the design tensile strength of 115 MPa. Therefore it is under control.

(g) (ii) Regarding Flange stresses, determine the bolt force allowed for during operating conditions and hence the radial flange stress. Determine the overall tilting moment Mo and hence the tangential flange stress. Check these two stresses are not excessive.

Using equation 3.21.6.4.4(1) to find the bolt force allowed for during operating conditions.

W = W_{m1}

W = 350.068 KN

Total joint contact surface compression force, Hp

Hp = $2b\pi GmP (1 + hg/h'g)$

Hp = $2 (0.0035) \pi (0.688) (0.8) (0.86 \times 10^6) (1+ (0.0338/0.0175))$

Hp = 334.884 KN

From operating conditions

Mg = $Hp / \{(1/hg) + (1/h'g)\}$

Mg = $(334.884) / \{(1/0.0388) + (1/0.0175)\}$

Mg = 4038.83 Nm

Hence using equation 3.21.11.6 (2), to find Radial Flange Stress

Sr = $6Mg / t^2 (\pi C - nD)$

Sr = $6(4038.83)/\{(0.048)^2 \{\pi (0.756 - (16) (0.030)\}$

Sr = 5.55 MPa < 115 MPa , Therefore under control

The hydrostatic force acting inside diameter Ho

H_D $= (\pi/4)\ B^2P$

Where B $= 616$ mm

H_D $= (\pi/4)\ (0.616)^2(0.86 \times 10^6)$
H_D $= 256.3$ KN

Total Hydrostatic end force during operating conditions

H $= (\pi/4)\ G^2P$
H $= (\pi/4)\ (0.688)^2(0.86 \times 10^6)$
H $= 319.716$ KN

Therefore H_T $= H - H_D$

H_T $= 319.716 - 256.300$
H_T $= 63.416$ KN

Value of h_D and h_T during operating conditions

h_D $= (C-B)/2$
h_D $= (0.756 - 0.616) / 2$
h_D $= 0.07$ m

h_T $= (H_D + H_G) / 2$
h_T $= (0.07 + 0.0388) / 2$
h_T $= 0.0544$ m

Therefore the overall tilting moment

Mo $= H_D\ h_D + H_T\ h_T$
Mo $= (256.3 \times 10^6)(0.07) + (63.416 \times 10^3)(0.0544)$
Mo $= 21390.8$ Nm

Using the moment Mo, obtain the tangential stress (3.21.11.6(1))

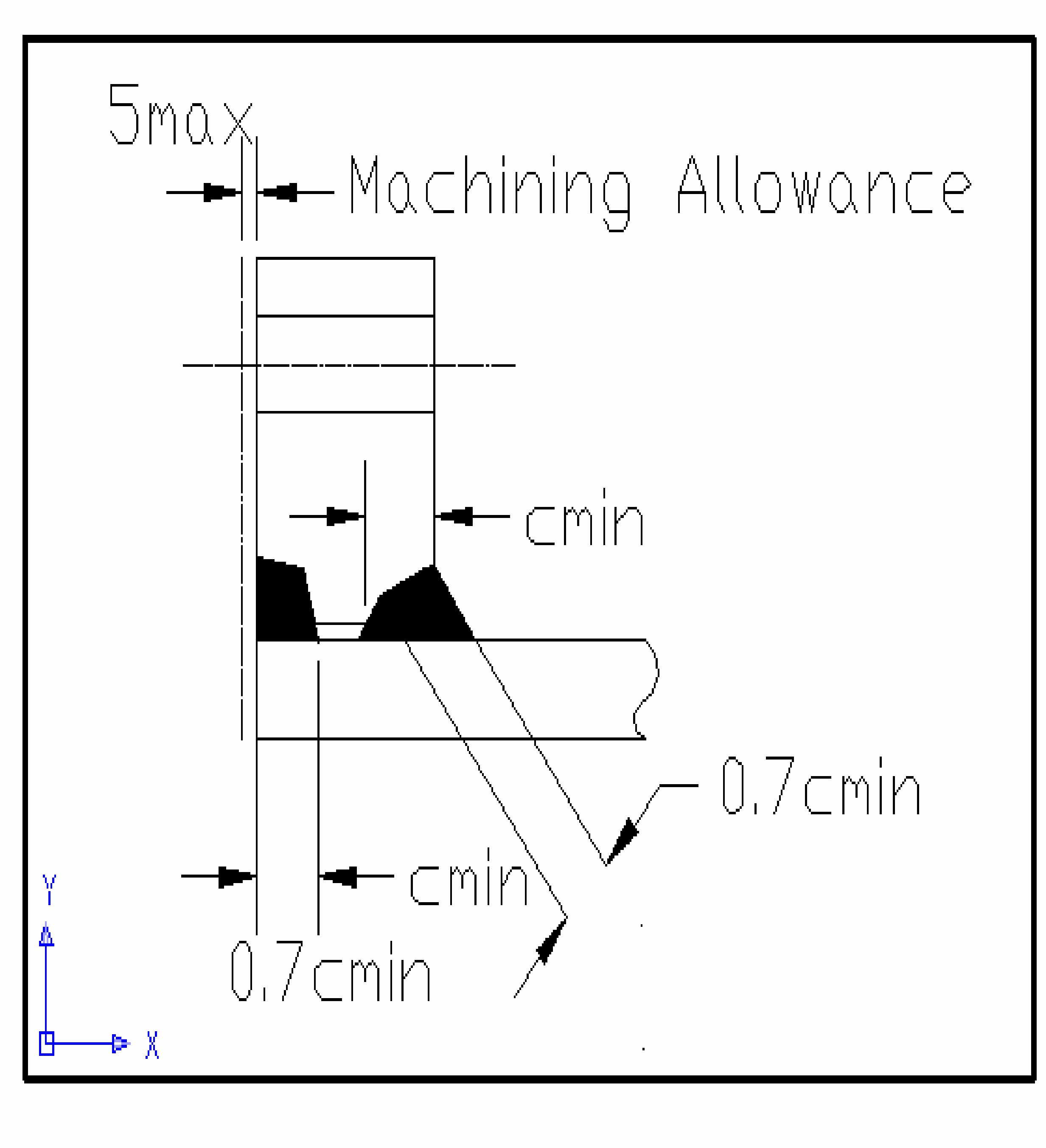
σT $= (Y'Mo) / (t^2B)$

Where Y is a function of A/B = 3.85 (Table 3.21.11.6(a))

σT $= (3.85)(21390.8) / (0.048^2)(0.616)$
σT $= 58.03$ MPa < 115 MPa

Therefore it is under control Both stress are not excessive

(g) (iii) Determine an appropriate welding arrangement for attaching the slip on the flange to the pipe piece.



(h) Confirm the adequacy of the cover plate

The minimum calculated thickness for circular ends in the equation 3.12.3(1)

$t \quad = D\ (P/Kfn)^{0.5}$

Where K = 4 (for full face gasket), And seamless, D = 584 mm

$t \quad = 756\ \{(0.86 \times 10^6) / (4)\ (115 \times 106)\ (1)\}^{0.5}$
$t \quad = 32.688$ mm

Therefore the given thickness is 48 mm and it is sufficient.

(i) Local from point view of the inspection opening looking towards the cover plate and the local part of the vessel wall, and showing the sectioning plane used to generate (ii).

See attached appendix

(ii) Cross section of branch, flange, cover plate and local part of the vessel wall, showing the welds. Since the opening is symmetrical only one side need to be shown

See attached appendix

Supports

Question 6

Cylindrical Shell

Outer Diameter	=	2154 mm
Inner Diameter	=	2134 mm
Length	=	5720 + 2 (76) due to straight flange
	=	5872 mm
Thickness	=	16 mm

Branch

Outer Diameter	=	616 mm
Inner Diameter	=	590.6 mm
Length	=	378 mm
Thickness	=	12.7 mm

Semi-Ellipsoid Head

Inner Diameter	=	2134 mm
Thickness	=	16 mm

(i) Select an appropriate sized lug and pad

Volume of hydraulic oil = Volume of cylinder + 2 (Volume of semi-ellipsoidal head)

$= (\Pi D^2 / 4) \times L + 0.2618 D^3$

$= (\Pi 2.134^2 / 4) \times 5.872 + 0.2618 \times (2.134)^3$

$= 23.546 \text{ m}^3$

Density of hydraulic oil is 910 + **X**0 kg/m^3 =910 +**8**0 kg/m^3
= 990 kg/m^3

Mass = Density x Volume
= 23.546 x 990
= 23310.54 Kg

The mass of water in the pressure vessel is 23310.54 Kg

Total Volume of Steel = Volume of Cylinder + Volume of Branch + Volume of semi-ellipsoidal Head

= $(\Pi\ D\ L\ t) + (\Pi\ D\ L\ t) + (2.18\ D^2\ t)$

= $(\Pi\ 2.134 \times 5.872 \times 0.016) + (\Pi\ 0.5906 \times 0.378 \times 0.0127) + (2.18 \times 2.134^2 \times 0.016)$

= $0.7976\ m^3$

Density of Steel is 7850 Kg/m^3

Mass	=	Volume x Density
	=	0.7976 x 7850
	=	6261.16 Kg
Converting to lbs	=	6261.16 Kg x 2.2046
	=	(Where 1 Kg = 2.2046 lbs)
	=	13803.35 lbs

The total mass of the steel is 6261.16 Kg

Total mass of pressure vessel filled with hydraulic oil

= 23310.54 + 6261.16

= 29571.7 Kg

The total mass of 11212.63 Kg is distributed among 4 lugs

Mass on one lug	=	29571.7 / 4
	=	7392.925 Kg
Converting lbs	=	7392.925 x 2.2046
	=	16298.44 lbs

With mass on one lug = 16298.44 lbs, the selected maximum allowable load on one lug is 22000 lbs (Pg. 5.78, resource material). Therefore this mass is acceptable.

Dimensions of the Reinforcement Pad

In [Cl 3.26.10.2], it states that

a) The pad shall have thickness not exceeding 1.5 times the thickness of the shell or head and not less than 5 mm (the throat thickness of the fillet weld joining pad to vessel shell shall not be more than the shell thickness.)

b) Pad shall extend at least 4 times its thickness in each direct beyond the toe of the weld attaching the support, but not less than 50 mm.

With the above two clauses as boundary conditions, the thickness of the pad inclusive of throat thickness shall be assumed to be 10mm < 16mm < 18mm

T pad = 16 mm (inclusive of throat thickness)
= 0.63 in

2A = 2(4 x 0.63) + 175 / h1
= 22.35 in

2B = 2(4 x 0.63) + bi
= 14.29 in

a) **(ii) Check the local stresses are not excessive in the new condition under hydrostatic test**.

Stresses in vessels due to the Lug Support

W	= 4840.53 + 300 (300 lbs due to cover plate, flanges and fittings etc)
	= 13803.35 lbs
n	= 4 number of lugs
Q	= W/n
	= 3450.83 lbs
R	= 41.35 in
H	= 6.5" + 0.63" + (0.63" / 2)
	= 7.445 in
t	= 0.63
P	= 1.5 x 0.86
	= 1.29 MPa
	= 187.23 Psi

Allowable stress value	= 115 MPa
	= 16690.86 PSI
Yield point	= 305 MPa
	= 44267.05 PSI (Appendix 5A)
joint Efficiency	= 0.85
Shape Factors	= 41.35 / 0.63
	= 65.63in

C1 = C2 = C3 = C4 = 0 (From table a Pg 5.84 of resource Material)

he K factors

B	=	A/R x $(B/A)^{1/3}$
	=	$(11.175/32.94)^{1/3}$
	=	0.34 x 0.86
	=	0.29
C	=	R/T
	=	41.35 / 0.63
	=	65.63in
K1	=	2.7
K2	=	0.022
K3	=	4.9
K4	=	0.015

Longitudinal Stresses

S1 = +- $(QH/\mathit{B}R^2t) \times [C_1K_1 + (6K_2R/C_2t) + (B/2(1.17 + B/A)) \times (R^2/HA)]$

$(3450.83 \times 7.445) / (0.29 \times 41.35^2 + 0.63) \times [1.0 \times 2.7 + ((6 \times 0.022 \times 41.35) / (1.0 \times 0.63)) + 0.29 / 2 \times (1.17 + 0.625) \times 41.35^2 / (7.445 + 11.175)$

= 782.2 Psi

Stress due to internal pressure PR / 2t = 187.23 x 52.94 / 2 x 0.63
= 7866 Psi

The sum of tensional stresses:

782.2 + 7866 = 8648.2 Psi

8648.2 Psi < 16690.86 Psi x 0.85 (Joint Efficiency)
8648.2 Psi < 14187.23 Psi

It does not exceed the stress value of the given seam.

Circumferential Stress

S2 = +- $(QH/\mathit{B}R^2t) \times [C_3K_3 + (6K_4R/C_2t) + (B/2(1.17 + B/A)) \times (R^2/HA)]$
= $(1285 \times 7.445) / (0.29 \times 32.94^2 + 0.63) \times [1.0 \times 4.9 + ((6 \times 0.015 \times 32.94) / (1.0 \times 0.63))$

= 463.57 Psi

Stress due to internal pressure

PR / 2t = 187.23 x 52.94 / 0.63
= 15733.26 Psi

The sum of tensional stresses:

463.57 + 15733.26 = 16196.83 Psi x 1.5

16196.83 Psi < 16690.86 Psi x 1.5
16196.83 Psi < 25036.29 Psi

It does not exceed the stress value of the shell material x 1.5

b) Select a square hollow section from the catalogue extracts provided to you and design a column that matches the lug and satisfies the required unity equations for combines loading that have been provided to you.

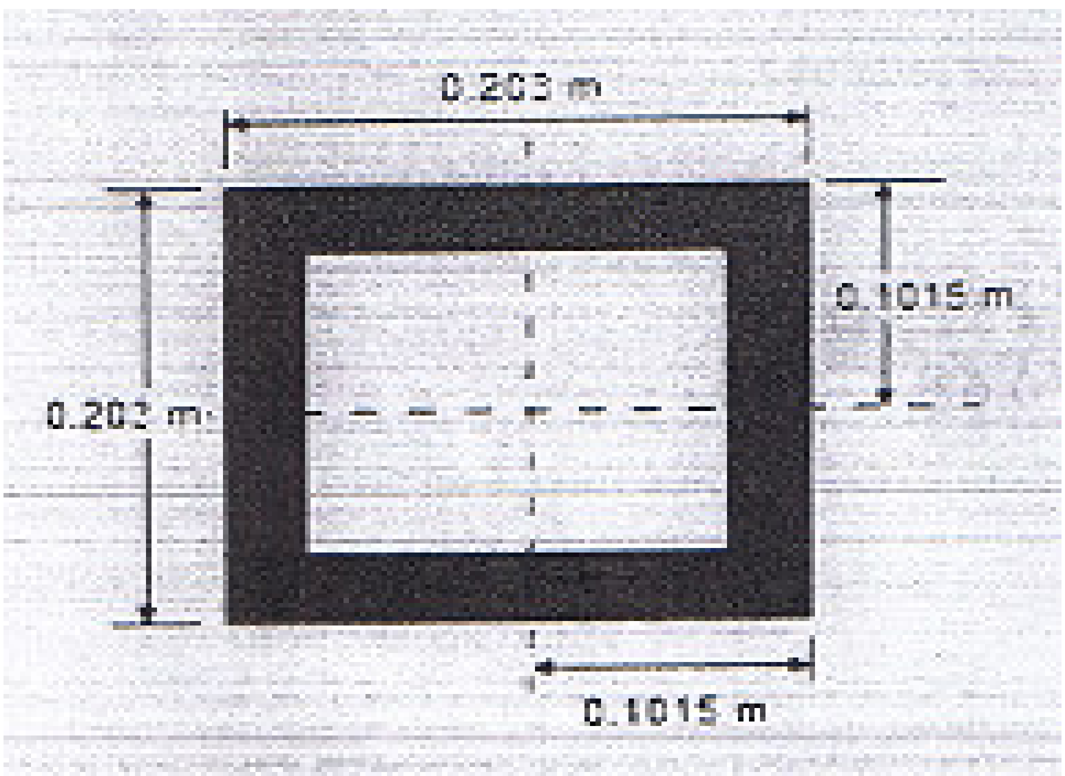
From (i) the chosen lug support has a base size of 10" x 10.25". 254 mm x 260.35 =66128.9 mm^2

The cross-sectional area of the column chosen must be smaller than the base area of the lug support chosen earlier so that the whole cross sectional area of the column can be fixed properly and fully to the base of the lug.

From the appendix 8B of the resource material, the column size 203 x 203 mm is chosen with thickness of 9.5 mm.
Cross sectional area of the column = 6030 mm^2

41209 mm^2 < 56693.435 mm^2

The eccentricities of load, e_x is equal to e_y all the sides if the cross sectional area of the columns are equal.



Therefore ex = ey = 0.203/2 = 0.1015 m

The corrosion allowance is assumed to be the same as the vesse lat 4.3125.
The corrosion allowance is given to the internal walls of the columns only.
Therefore the next thickness size down of 8 mm is used for calculations.

Length of the column, L = 2247 + XY mm
= 2247 + 86 mm
= 2.333 m

A flat ended column is less restrained at the end than required for full fix. The support leg can therefore be approximated by a column with end conditions between one end guided, the other end at the based fixed (Kc=1) and one end guided the other end hinged (Ke = 2). In this case the value of Ke is assumed to be 1.5

The effective length, I = KeL
= 1.5 (2.333)
= 3.4995 m

From appendix 8 B of resource material, r = 78.1 mm

Therefore Slenderness ratio = I/r
= 3.4995 / 0.0781
= 44.81

As stated in Table 4.6 Pg. 8.30, the maximum slenderness ratio must not exceed 180.

$$44.80 < 180$$

The modulus of elasticity of steel, $E = 207 \times 10^3$ MPa
The Euler's critical stress

$$F_{ocx} = F_{ocy} = \pi^2 E / (l/r)^2$$
$$= \pi^2 \times 207 \times 10^9 / (44.81)^2$$
$$= 1017.47 \text{ MPa}$$

From appendix 8 B of resource material,

Yield stress, FL = 350 MPa
$= 0.00003 (l/r)^2$
$= 0.00003 (44.81)^2$
$= 0,0602$

$$L = \{(350 + (I + 0.026)\, 2352.763) / 2\} - \{(Fy + (1 + \eta)\, Foc) / 2) - Fy\, Foc\}$$
$$= 337.95 \text{ MPa}$$

In AS 3390, $\Omega = 1.67$

Maximum permissible average compressive stress,

Fac	=	FL / Ω
	=	337.95 / 1.67
	=	202.365 MPa

(Mass acting on one lug + mass of lug) = (2803.1575 +28)
= 2831.1575 lbs

P	=	(2831.1575 / 2.2046) x 9.81
	=	12598 N

Pc	=	$\pi^2 E\, l / e^2$
	=	π^2 (207 x 10^9) (37.5 x 10^6) / 31.521^2
	=	77108.345 N

From appendix 8 B of resource material,
Cross-sectional area, A = 6.03 x 10^{-3} m^2

Fac	=	P / A
	=	12598 / (6.03 x 10^{-3})
	=	2.089 MPa

M	=	P x e
	=	12598 x 0.1015
	=	1278.697 Nm

The bending stress that exist in axial compressive load was not present, Fbc = M / z
From appendix 8 B, z = 369 x 10^{-6} m^3

Therefore Fbc	=	M/z
	=	1278.697/369 x 10^{-6}
	=	3.47 MPa

The maximum bending stress about x-axis, Fbcx is equal to the maximum bending stress about y-axis, Fbcy.

When the end moments are equal and opposite, B = 1 which represent, The bending is of single curvature.

Mmax	=	M / [1 – (P / Pc)]

Fbcx	=	fbs / [1 – (P / Pc)]
	=	15.4682 / [1 – (12598 / 77108.345)]
	=	184.89 MPa
Fbcx – Fbcy	=	184.89 MPa

Assume Cmx and Cmy is 1

Applying the equation 8.3.1 (a) from AS 3990 to verify whether the chosen column size is suitable.

{fac/Fac}+{[Cmx(1 / 1 – (fac / 0.6 Foc))] / Fbcx+{[Cmy(1 / 1 – (fac / Focy))] / Fbcy<= 1

{2.089/202.364} + {[1(1 / 1 – (2.089 / 0.6 x 2056.233))]x 57.085 / 184.89} +
{[1(1 / 1 – (203.4 / 2352.763))] x 18.489 / 184.89} <= 1

0.0103 + 0.309 + 0.109 <= 1
0.4283 <= 1

Therefore, the chosen column size of 203 mm by 203 mm with the thickness of 9.5 mm is suitable.

General Arrangement Drawing

Question 7

See Attached Appendix