ENGINEERING DESIGN PROJECT

FULL PIPELINE SYSTEM DESIGN OF A MANUFACTURING PLANT FOR CONTINENTAL AG

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OLD DOMINION UNIVERSITY

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1. ABSTRACT

Continental AG is planning to build a new manufacturing facility in Dayton, Ohio. The manufacturing plant will consist of an automated machining line with five machines. Each machine is to be designed to have its coolant supplied from a single reservoir tank. The project will cover the design of the entire coolant system, starting with the receival of new coolant by railcar all the way to the disposal of used and waste coolant by trucks.

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5. REPORT

I. Job Site Location

The manufacturing facility for Continental AG will be built in Dayton, Ohio. The machine coolant will be delivered by railroad that is positioned 600 feet from the machining area. The used and waste coolant will be stored in a separate tank and will be unloaded by trucks who enter the site via highway located 200 feet from the building.

II. Specifications and Design Philosophy

The coolant supply system will be designed to be financially beneficial with the system's efficiency taking part. The manufacturing plant will be in operation for 2 shift per day, 7 days per week. New coolant will be delivered in 15,000-gallon tanker cars. The new coolant is to be stored in a 15,500-gallon clean reservoir tank. The clean coolant storage tank will be fed to a 1,100-gallon tank that will feed the machine room and all five machines. Under normal operating conditions, the 1,100-gallon tank is emptied once per week. However, if an emergency situation may arise, the plant has the option to dump the tank to a used/waste coolant tank with a 5,000-gallon capacity. The 5,000-gallon tank will be emptied once per month by way of trucks.

III. Sources

Milbury, Matt. "Flange - Blind, ANSI Class 150, B16.5, RF (in)." Piping Designer, https://www.piping-designer.com/index.php/datasheets/flange-datasheets/194-blind-flange-datasheet/1178-flange-blind-ansi-class-150-asme-b16-5-1-16-raised-face-in.

Mott, R., Untener, J., "Applied Fluid Mechanics," 7th Edition. Pearson Education, Inc., (2015)

Spotts, M. F., Shoup, T. E., & Hornberger, L. E. (2004). Design of Machine Elements (8th ed.). Pearson Education Inc.

S, Werner. "Pressure Classes of Flanges." Explore the World of Piping, https://www.wermac.org/flanges/flanges_pressure-temperature-ratings_astm_asme.html#:~:text=For%20example%2C%20a%20Class%20150,at%20approximately%20800%C2%B0F.

"Vertical Storage Tanks." Norwesco, https://norwesco.com/products/above-ground-tanks/vertical-storage.

IV. Materials and Specifications

The entire piping system will be constructed from SCH 40 steel pipe. The selection of the pipe is based on typical industry standard availability. All tanks will be sourced from Norwesco, an industrial tank manufacturer located in Saint Bonifacius, Minnesota. The tanks are to be manufactured from high-density polyethylene resin.

The coolant used in the plant will be a solution of water and soluble oil with a specific gravity of 0.94. The coolant has a freezing point of 0°F. The coolant's corrosiveness is approximately the same as that of water.

V. Preliminary Drawings and Sketches

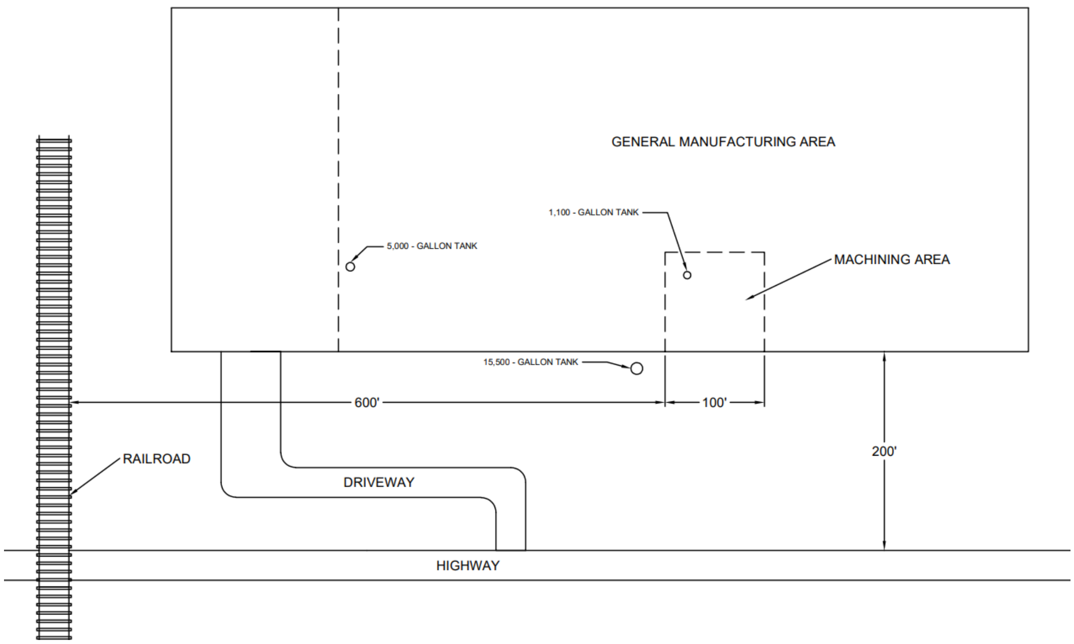


Figure V-1: Preliminary Plot Plan

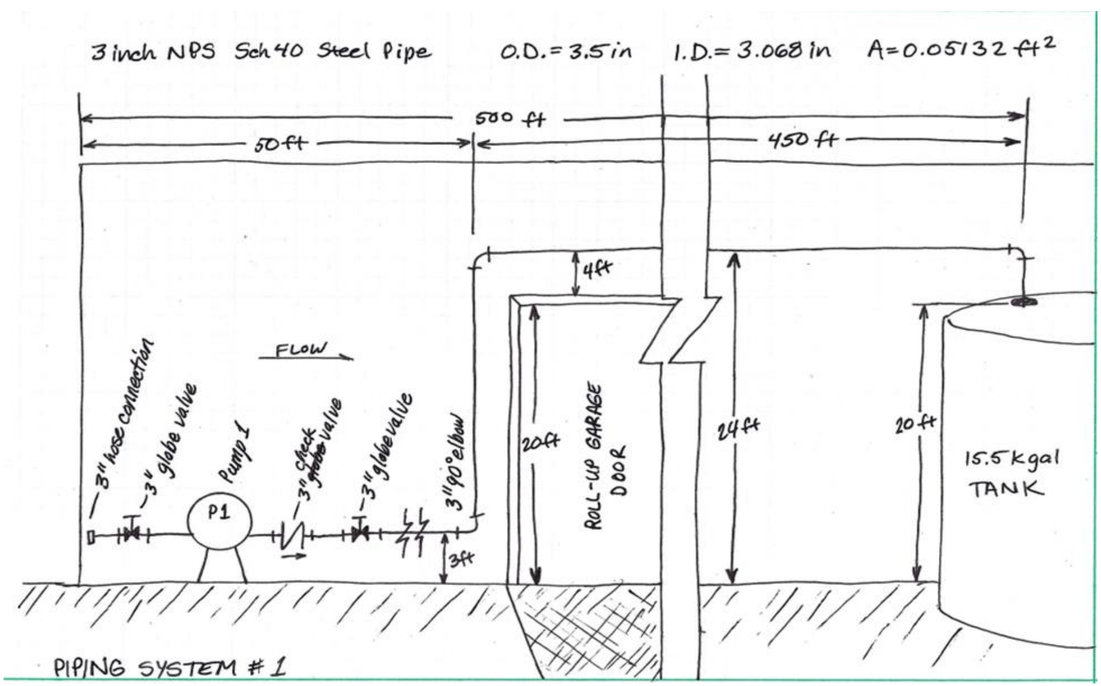


Figure V-2: System 1 Elevation

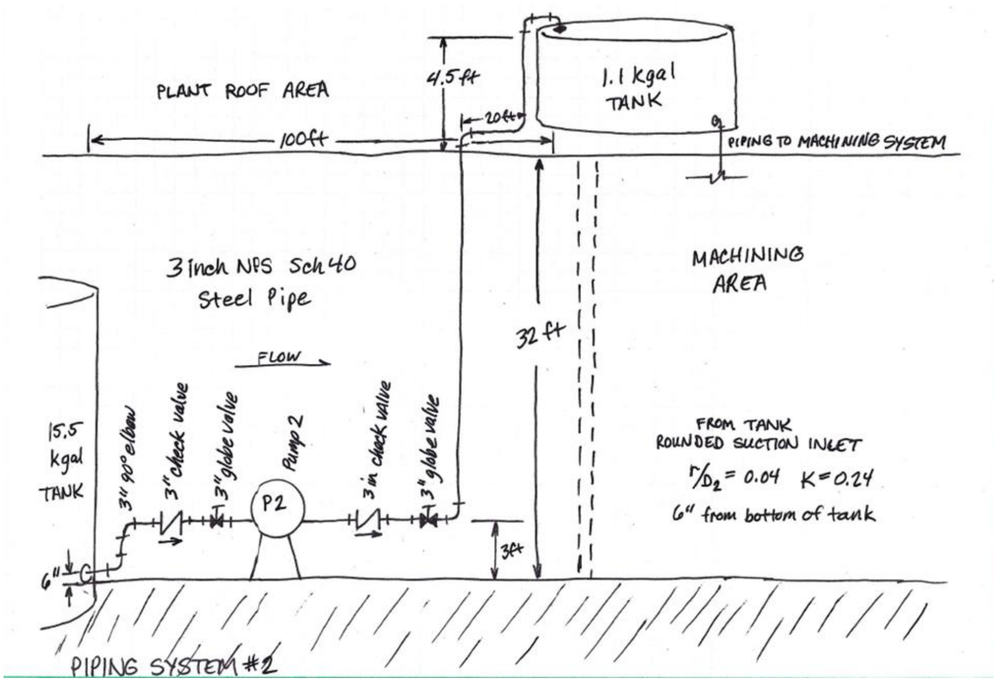


Figure V-3: System 2 Elevation

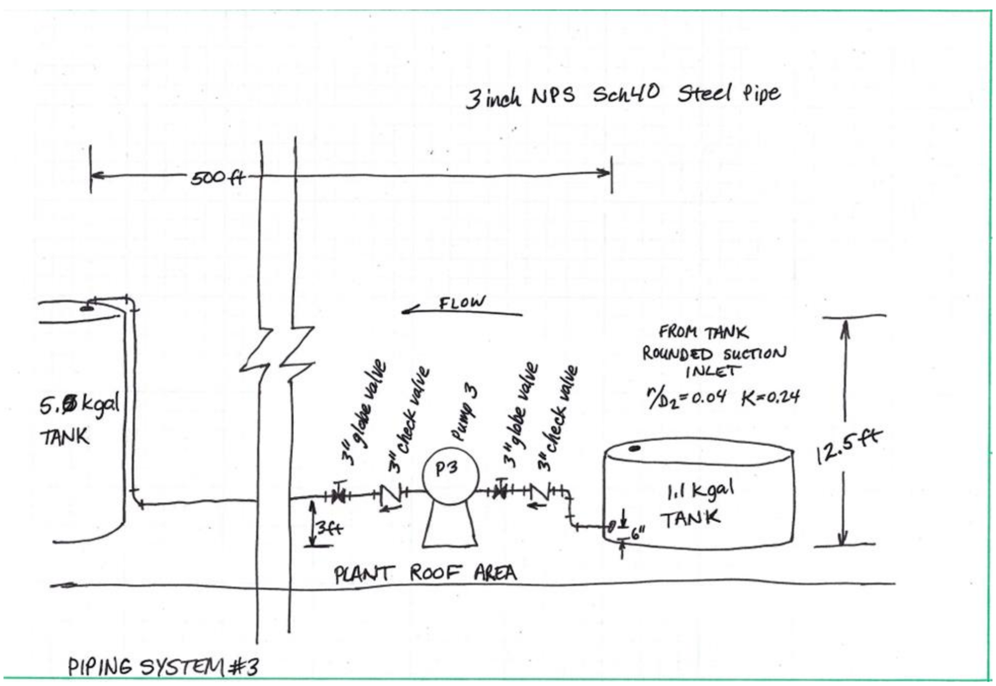


Figure V-4: System 3 Elevation

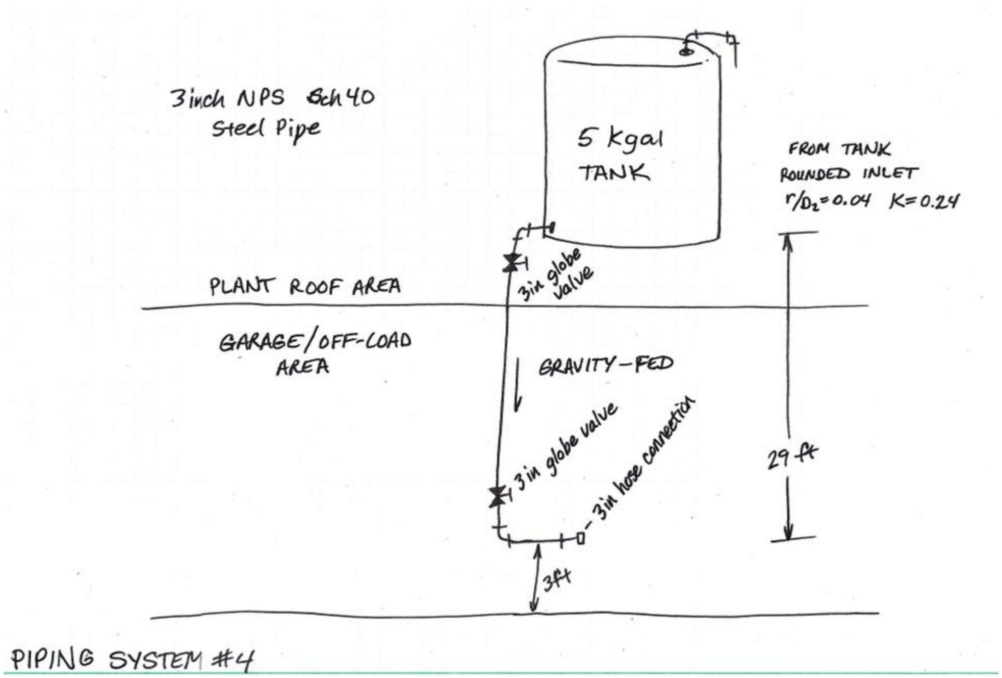


Figure V-5: System 4 Elevation

VI. Design Calculations

Tank specifications

Tank Size and Location

Purpose:

To specify the size and location of all storage tanks.

Drawings and Diagrams:

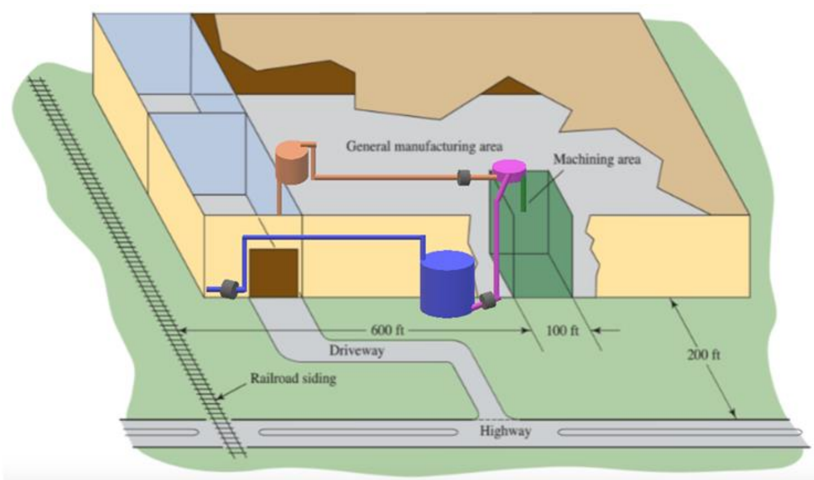


Figure VI-1: Tank Locations

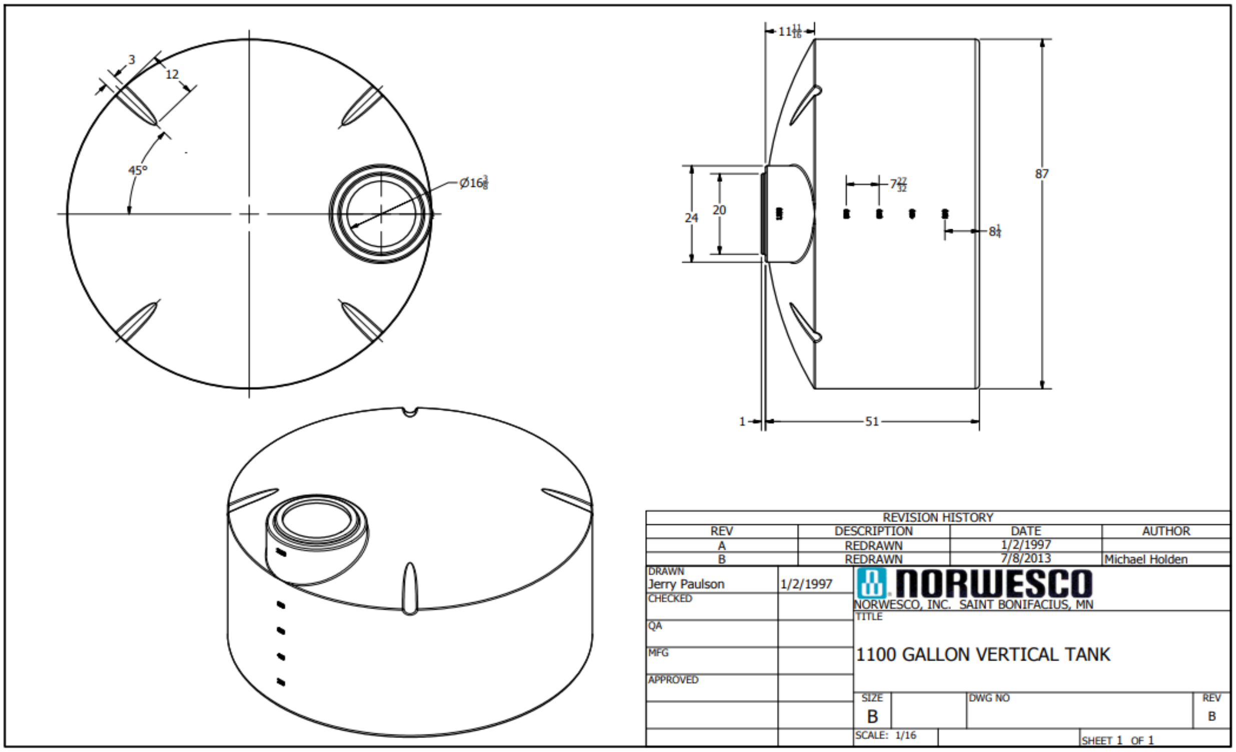


Figure VI-2: 1,100 Gallon Vertical Storage Tank

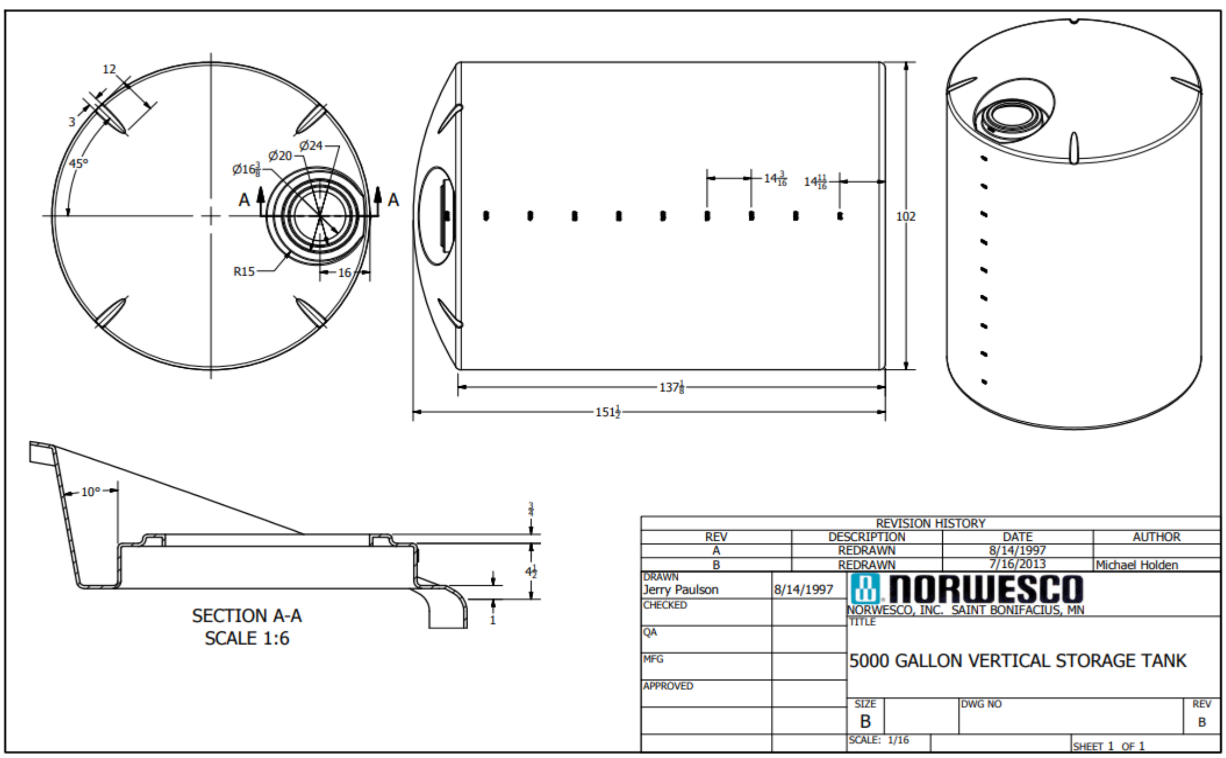


Figure VI-3: 5,000 Gallon Vertical Storage Tank

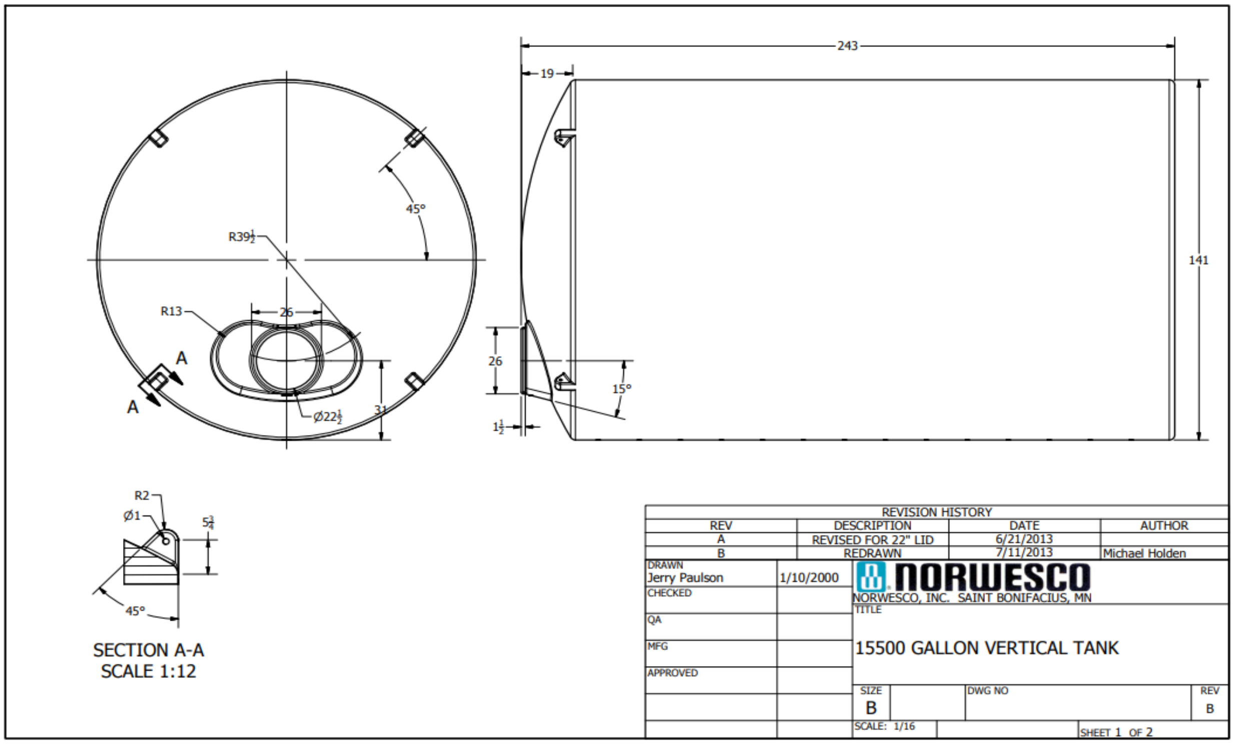


Figure VI-4: 15,500 Gallon Vertical Storage Tank (A)

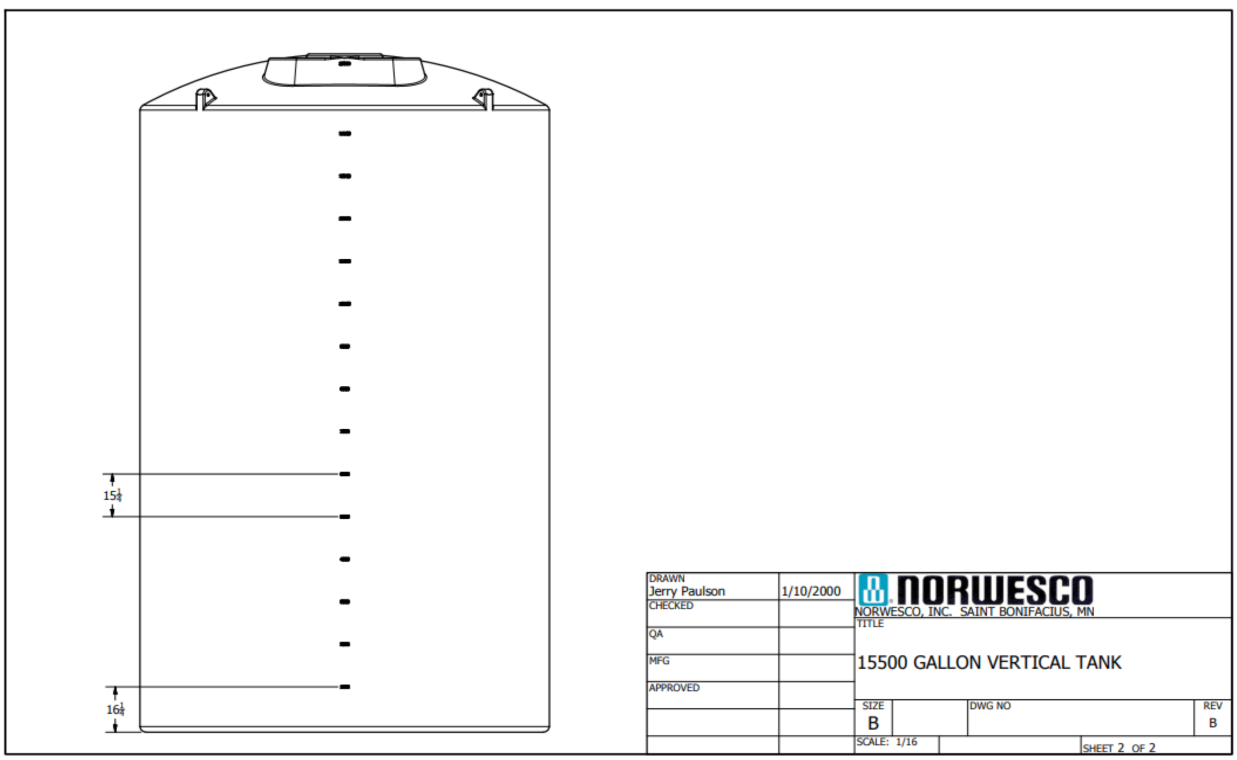


Figure VI-5: 15,500 Gallon Vertical Storage Tank (B)

Sources:

Milbury, Matt. “Flange - Blind, ANSI Class 150, B16.5, RF (in).” Piping Designer, https://www.piping-designer.com/index.php/datasheets/flange-datasheets/194-blind-flange-datasheet/1178-flange-blind-ansi-class-150-asme-b16-5-1-16-raised-face-in.

Mott, R., Untener, J., “Applied Fluid Mechanics,” 7th Edition. Pearson Education, Inc., (2015)

“Vertical Storage Tanks.” Norwesco, https://norwesco.com/products/above-ground-tanks/vertical-storage.

Design Considerations:

Norwesco was decided upon as the supplier due to their large selection of prefabricated tanks**.** The catalog link, https://norwesco.com/products/above-ground-tanks/vertical-storage, was used for the selection of tanks. The reduction of pump usage was paramount.

Data and Variables:

Tank Specifications					
Use	**Diagram Color**	**Location**	**Capacity (gallons)**	**Diameter (inches)**	**Height (inches)**
Storage	Blue	Ground level b/w driveway & machining area	15,500	141	244
Supply to Machining Area	Pink	Roof above machining area	1,100	87	53
Waste Storage	Orange	Roof above garage	5,000	102	152

Table 1: Tank Size and Location

Procedure:

The capacity of each tank was determined based upon the design parameters. An online supplier was chosen and a tank that met each capacity requirement was selected for each location. The dimensions of the tank were provided by the supplier. The location of each tank was determined based upon the numerous factors which will be outlined in the summary below.

Calculations:

Volume of Storage Tank:

$V = \pi r^2 h$

$V = \pi \left(\frac{141in}{2}\right)^2 (244\ in) = 3809938.216\ in^3 \left(\frac{1gal}{231in^3}\right)$

$\boxed{V = 16493.24\ gal}$

Volume of Supply Tank to Machining Area:

$V = \pi r^2 h$

$V = \pi \left(\frac{87in}{2}\right)^2 (53\ in) = 315067.97\ in^3 \left(\frac{1gal}{231in^3}\right)$

$\boxed{V = 1363.9306\ gal}$

Volume of Waste Storage Tank:

$V = \pi r^2 h$

$V = \pi \left(\frac{102in}{2}\right)^2 (152\ in) = 1242034.939\ in^3 \left(\frac{1gal}{231in^3}\right)$

$\boxed{V = 5376.775\ gal}$

Summary:

The calculated volumes differ from the manufacturer supplied volumes due to the fact that the calculations account for the tanks to be filled 100%. In a real-world application, this will never exist. Below, a table shows the calculated volumes versus the manufacturer specified volumes.

Vertical Storage Tank Volumes		
Tank Use	**Manufacturer Listed Capacity (gallons)**	**Calculated Capacity (gallons)**
Storage	15,500	16,493.24
Supply to Machining Area	1,100	1,363.93
Waste Storage	5,000	5,376.78

Table 2: Vertical Storage Tank Volumes

The location of each tank was decided upon based on reducing the usage of pumps. The pumps selected will only need to be used when filling and emptying the tanks, not during regular operations. The system from the supply tank to the machining area and the system from the waste storage to the trucks will be gravity fed.

The size of each tank was determined based on the provided specifications. The storage tank will allow for a full train car to be emptied when a delivery is needed. The 1,100-gallon supply tank will allow for a full week of operation. The 5,000-gallon waste tank allows for a normal month's worth of usage with the ability to handle an emergency dump of 1,000 gallons.

Materials:

Three tanks were used: 15,500-gallon vertical storage tank (item # 43943), a 5,000-gallon vertical storage tank (item # 40166), and a 1,100-gallon vertical storage tank (item # 40081).

Analysis:

Tank size was determined based upon the requirements of Continental AG and the capacity of the rail car for coolant deliveries. The 15,500-gallon capacity tank can fully empty the rail tank car and will provide more than enough coolant to last until the next delivery. The 1,100-gallon capacity tank will provide enough coolant for a full week of operation. The 5,000-gallon capacity tank will provide storage for four weeks of dirty coolant as well as capacity for any emergency dumping that will be needed.

Tank Material and Thickness (task 2)
Purpose:

To determine the tank material and wall thickness for each tank in the system.

Drawings and Diagrams:

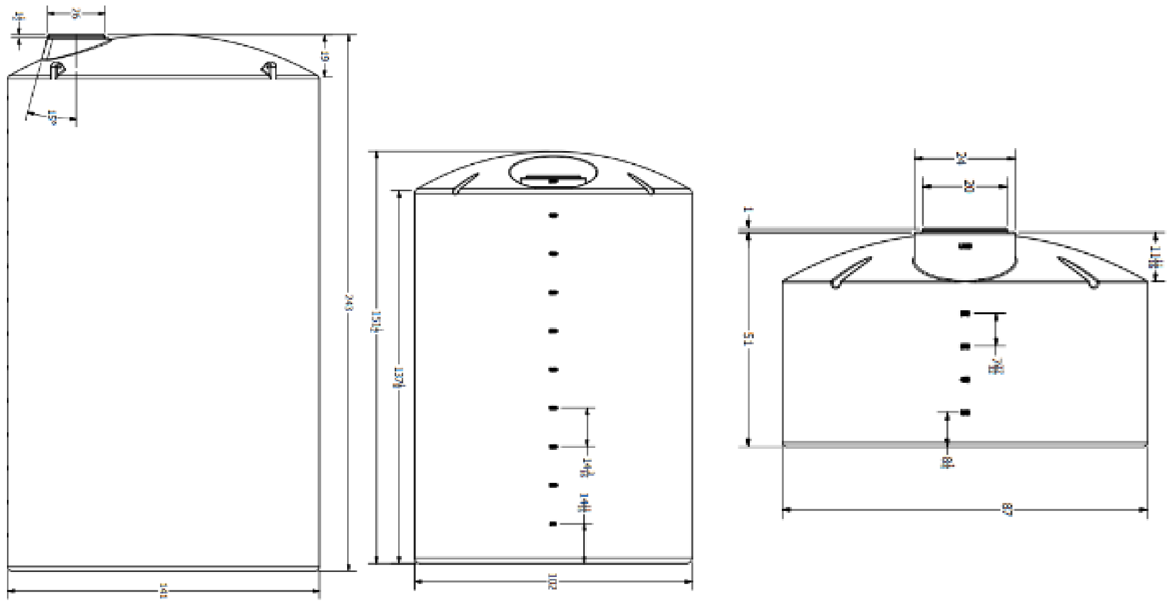


Figure VI-6: 15,500-Gallon, 5,000-Gallon, and 1,100-Gallon Storage Tanks

Sources:

Milbury, Matt. "Flange - Blind, ANSI Class 150, B16.5, RF (in)." Piping Designer, https://www.piping-designer.com/index.php/datasheets/flange-datasheets/194-blind-flange-datasheet/1178-flange-blind-ansi-class-150-asme-b16-5-1-16-raised-face-in.

Mott, R., Untener, J., "Applied Fluid Mechanics," 7th Edition. Pearson Education, Inc., (2015)

Spotts, M. F., Shoup, T. E., & Hornberger, L. E. (2004). Design of Machine Elements (8th ed.). Pearson Education Inc.

Design Considerations:

All tanks will be made of high-density polyethylene resin. The tanks will be UV stabilized to withstand outdoor conditions.

E is assumed to be 0.80

Y is assumed to be 0.40

Data and Variables:

$\mathrm{D}_{15500\ \mathrm{gal}} = 141"$ $\mathrm{h}_{15500\ \mathrm{gal}} = 244"$

$\mathrm{D}_{5000\ \mathrm{gal}} = 102"$ $\mathrm{h}_{5000\ \mathrm{gal}} = 152"$

$\mathrm{D}_{1100\ \mathrm{gal}} = 87"$ $\mathrm{h}_{1100\ \mathrm{gal}} = 53"$

TABLE 15-2 PROPERTIES OF COMMON PLASTIC MATERIALS†

	Common Trade Names	Tensile Yield Strength MPa	Tensile Modulus of Elasticity GPa	Izod Impact Strength J/m	Glass Transition Temperature °C	Melt Temperature °C
THERMOPLASTICS						
Amorphous[1]						
ABS (Acrylonitrile Butadiene Styrene)	Cyclolac, Lustran	34–50	2.1–2.8	164–524	102–115	-
PMMA (acrylic) (Polymethylmethalcrylate)	Lucite, Plexiglas	53–69	2.2–3.24	11–22	85–105	-
Polycarbonate	Lexan, Makrolon	62	2.4	655–982*	150	-
Polystyrene		83–90**	2.3–3.3	19–25	74–105	-
Semi-crystalline[1]						
Acetal (Homopolymer)	Delrin, Celcon	66–83	2.4–3.4	60–126		172–184
High Density Polyethylene	Alathon	26–33	0.9–1.1	22–218		130–137
Low Density Polyethylene		9–14	0.1–0.3	no break	−25	98–115
Polyamide (Nylon 6/6)	Zytel, Nylon	55–83+	2.3–3.8+	30–55		255–265
Polypropylene		31–37	1.14–1.6	22–76	−20	160–175
Polyethylene Tetrafluoride	Teflon	<7	0.4–0.55	160		327
THERMOSETS						
High cross-link density						
Epoxy (Casting Resin)	Epon	28–90**	2.4	11–55	-	-
Polyester (Cast)		4–90**	2.1–2.8	11–22	-	-
Phenolic (Cotton filled)	Bakelite, Durez	45–52**	7.6–9.7	16.4–103	-	-
Low cross-link density						
Natural Rubber	Latex	24**	***	No break	−68	-
Polychloroprene	Neoprene, Bayprene	22**	***	No break	−45	-
Silicone (Casting Resin)	Silastic	2–7**	***	No break	−127	-

[1]Properties of unfilled injection molding grades from Kaplan. Properties are averages of several grades. Consult manufacturers for specific design data.
*Below 4mm in thickness
**Ultimate Strength
***Material is not linearly elastic
†Properties listed are for comparison purposes only. For design data, consult the manufacturer of the material.

Figure VI-7: Properties of Tank Material

Procedure:

The material of the tank was given by the manufacturer as is stated to be high-density polyethylene resin. The thickness of the tank was calculated using the equation for thickness of a pipe as an approximation. The design pressure was determined to be the pressure due to the weight of the coolant inside the tank. The outside diameter was supplied by the manufacturer. The yield strength of the high-density polyethylene resin was found in Figure F-7 and a safety factor of four was used. The longitudinal joint quality factor and correction factor were assumed.

Calculations:

Wall Thickness Equation: $t = \frac{pD}{2(SE+pY)} = \frac{\gamma hD}{2(SE+\gamma hY)}$

<u>15,500-Gallon Capacity Vertical Storage Tank:</u>

$$t = \frac{\gamma h_{15500} D_{15500}}{2(SE + \gamma h_{15500} Y)}$$

$$t = \frac{(0.0361\ lb/in^3)(244\ in)(141\ in)}{2\left(\frac{3770.98\ psi}{4}(0.8) + (0.94)(0.0361\ lb/in^3)(244\ in)(0.40)\right)}$$

$\boxed{t = 0.82\ in}$

5,000-Gallon Capacity Vertical Storage Tank:

$$t = \frac{\gamma h_{5000} D_{5000}}{2(SE + \gamma h_{5000} Y)}$$

$$t = \frac{(0.0361\ lb/in^3)(152\ in)(102\ in)}{2\left(\frac{3770.98\ psi}{4}(0.8) + (0.94)(0.0361\ lb/in^3)(152\ in)(0.40)\right)}$$

$\boxed{t = 0.37\ in}$

1,100-Gallon Capacity Vertical Storage Tank:

$$t = \frac{\gamma h_{1100} D_{1100}}{2(SE + \gamma h_{1100} Y)}$$

$$t = \frac{(0.0361\ lb/in^3)(53\ in)(87\ in)}{2\left(\frac{3770.98\ psi}{4}(0.8) + (0.94)(0.0361\ lb/in^3)(53\ in)(0.40)\right)}$$

$\boxed{t = 0.11\ in}$

Summary:

The equation for pipe thickness was used to approximate the wall thickness of the storage tanks. The allowable stress in tension was determined from the tensile yield divided by a safety factory of four based on the recommendation of the Design of Machine Elements text. The values for longitudinal joint quality factor and correction factor based on the type of material and temperature were assumed to be as low as possible to ensure the tank will be thick enough to contain the full volume of coolant. The value for the longitudinal stress joint quality is based upon welds and most likely could have been assumed to be 1.0 since the tanks are of cast resin construction. However, a factor of 0.8 was used in an overabundance of caution. A table of the calculated thicknesses can be seen below.

Vertical Storage Tank Wall Thickness	
Vertical Storage Tank	**Tank Wall Thickness (in)**
15,500-gallon	0.82
5,000-gallon	0.37
1,100-gallon	0.11

Table 3: Storage Tank Wall Thickness

Materials:

Three tanks were used: 15,500-gallon vertical storage tank (item # 43943), a 5,000-gallon vertical storage tank (item # 40166), and a 1,100-gallon vertical storage tank (item # 40081).

Analysis:

The material that was chosen, high-density polyethylene resin, is relatively elastic and does not require an extremely thick wall to hold the fluid. Per the manufacturer of the selected tanks, the wall thickness does not vary across the vertical change of the tanks.

Future Drain Connection – Blind Flange (task 3)
Purpose:

To design a blind flange connection to the 1,100-gallon storage tank in preparation for possible future connections.

Drawings and Diagrams:

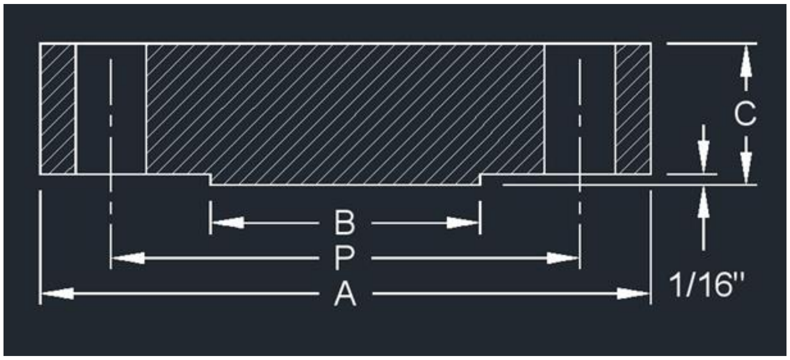


Figure VI-8: Blind Flange Class 150

Flange Specifications					
Pipe Size (in)	Outside Diameter, A (in)	Diameter of B (in)	Thickness, C (in)	Number of Bolts	Diameter (in)
3	7.5	5	0.9375	4	0.75

Table 4: Flange Specifications

Sources:

Milbury, Matt. “Flange - Blind, ANSI Class 150, B16.5, RF (in).” *Piping Designer*, https://www.piping-designer.com/index.php/datasheets/flange-datasheets/194-blind-flange-datasheet/1178-flange-blind-ansi-class-150-asme-b16-5-1-16-raised-face-in.

Mott, R., Untener, J., “Applied Fluid Mechanics,” 7th Edition. Pearson Education, Inc., (2015)

S, Werner. “Pressure Classes of Flanges.” *Explore the World of Piping*, https://www.wermac.org/flanges/flanges_pressure-temperature-ratings_astm_asme.html#:~:text=For%20example%2C%20a%20Class%20150,at%20approximately%20800%C2%B0F.

“Vertical Storage Tanks.” *Norwesco*, https://norwesco.com/products/above-ground-tanks/vertical-storage.

Design Considerations:

Per ASME B16.5, class 150 blind flanges have a pressure rating of 270 psig at ambient temperature. The tank is assumed to be at full capacity.

Data and Variables:

1,100-gallon capacity

87” diameter

53” height

Procedure:

The highest pressure in the tank we be experienced at the bottom. It was found using the “gamma-h” equation. A flange was selected based upon ASME requirements that can withstand the calculated pressure.

Calculations:

$$P = \gamma_{coolant} h = 62.4\ lb/ft^3 (0.94)(4.42\ ft) = 259.26\ lb/ft^2$$

$$P = 259.26 \frac{lb}{ft^2} \times \left(\frac{1\ ft^2}{144\ in^2}\right)$$

$$\boxed{P = 1.8\ psi}$$

Summary:

The pressure in the storage tank was fairly low at 1.8 psi. The lowest class of blind flange available was 150 per B16.5. These flanges are rated for 270 psig; therefore, this became the selected flange.

Materials:

1,100-gallon capacity vertical storage tank

4, ¾ inch diameter bolts

Class 150 flange with aforementioned specifications

Analysis:

Wind Load and Weight (task 11)
Purpose:

To determine the weight of each of the coolant storage tanks and their associated wind load conditions for the civil engineer's calculations.

Drawings and Diagrams:

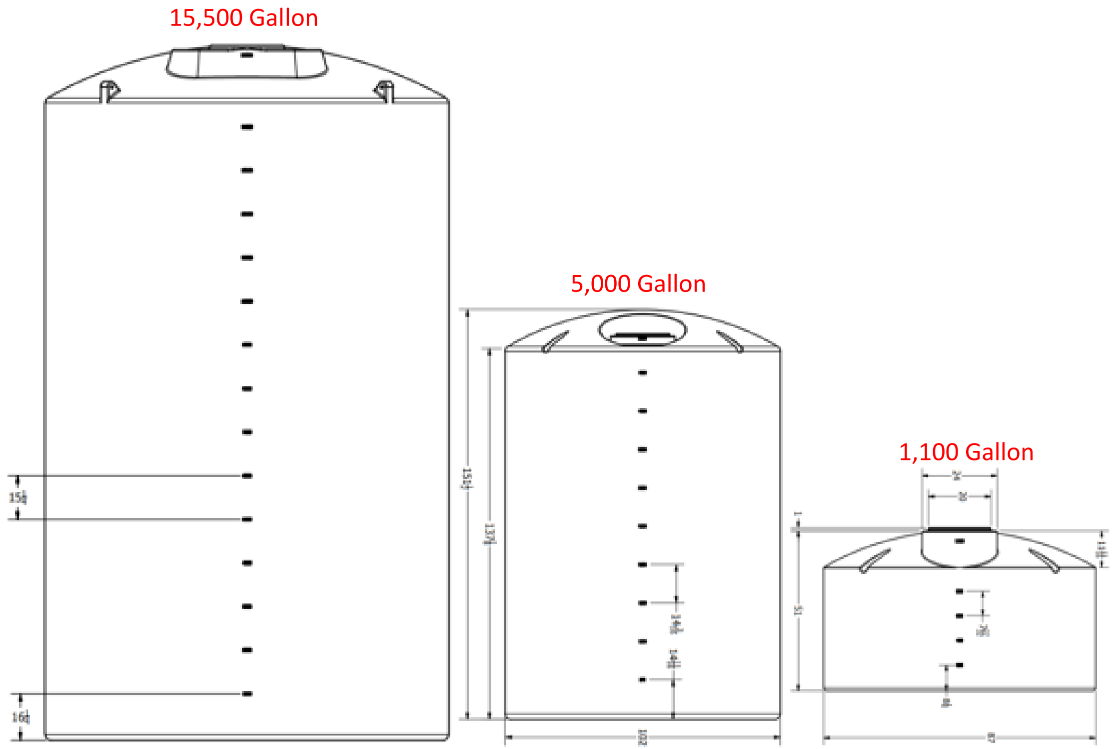


Figure VI-9: Tank Dimensions for Wind Load Calculations

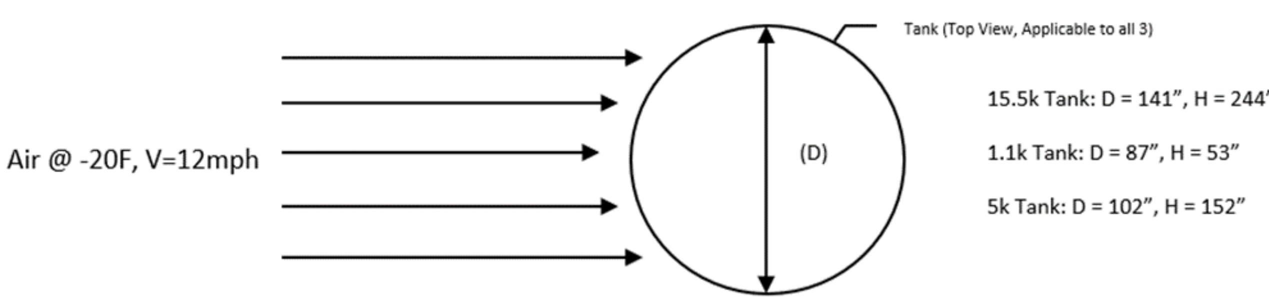


Figure VI-10: Wind Load Diagram

Sources:

Mott, R., Untener, J., "Applied Fluid Mechanics," 7th Edition. Pearson Education, Inc., (2015)

"15500 Gallon NORWESCO Plastic Potable Water Storage Tank." *Plastic*, https://www.plastic-mart.com/product/15816/15500-gallon-plastic-water-tank-norwesco-44814-44816.

"5000 Gallon Norwesco Plastic Potable Water Storage Tank." *40870 41375 40641*, https://m.plastic-mart.com/product/124/5000-gallon-plastic-water-storage-tank-40641.

"1100 Gallon NORWESCO Plastic Potable Water Storage Tank." *Plastic*, https://m.plastic-mart.com/product/2726/1100-gallon-plastic-water-storage-tank-40704.

Design Considerations:

All tanks are assumed to be at maximum capacity

High-density polyethylene resin tanks

Incompressible fluids

Tanks are vented to atmospheric pressure

Air is denser at colder temperatures

Projected area of cylindrical tank would be a rectangle (base x height)

Data and Variables:

Gravity, g = $9.81\ m/s^2$

Specific Gravity (coolant), sg = 0.94

Density of Water, $\rho = 1{,}000\ kg/m^3$

Weight of 15,500-gal Tank (empty), $W1 = 3{,}307\ lbs$

Weight of 5,000-gal Tank (empty), $W2 = 844\ lbs$

Weight of 1,100-gal Tank (empty), $W3 = 169\ lbs$

Density of Air @ -20F, $\rho = 1.4456\ kg/m^3$ (calculated below)

Drag Coefficient of a Cylinder, $C_D = 1.12$

Velocity of Air, V = 12 mph (see wind gust value from Accuweather below)

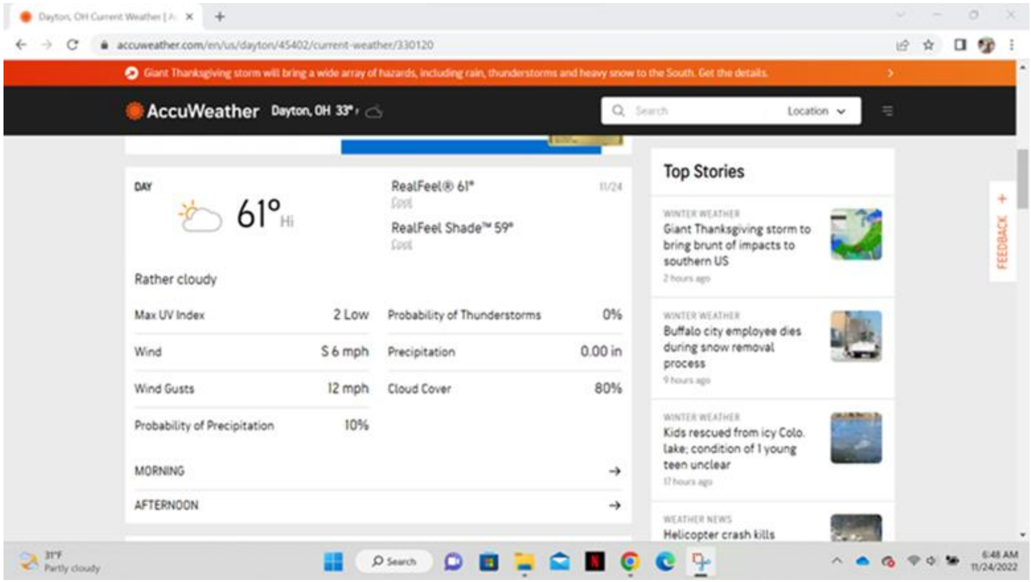


Figure VI-11: Jobsite Weather Information

Procedure:

Start by computing the weight of the coolant for each tank when at maximum capacity and add this to the weight supplied by the manufacturer of the empty tank. This will be given to the civil engineers to use for their designs.

For wind load, the equation for calculating drag force will be used. The drag coefficient for each tank, the density of air at the given temperature, velocity of air, and projected area of tank that will experience the wind load must be determined. After all variables are determined, they can be substituted into the drag force equation to find the resultant wind load.

Calculations:

<u>Weight of 15,500-gallon tank:</u>

From Norwesco tank specs, $W_{T1} = 3{,}307\ lbs$

$$V_C = 15500\ gal\left(\frac{1\ m^3}{264.2\ gal}\right) = 58.6677\ m^3$$

$$\rho_C = sg_C \times \rho_W \rightarrow 0.94\left(1000\frac{kg}{m^3}\right) \rightarrow \rho_C = 940\ kg/m^3$$

$$m_C = \rho_C V_C \rightarrow 940\frac{kg}{m^3}(58.6677\ m^3) \rightarrow m_C = 55147.6\ kg$$

$$W_C = m_C \cdot g \rightarrow 55147.6\ kg\left(9.81\frac{m}{s^2}\right) \rightarrow W_C = 540998\ N$$

$$W_C = 540998\ N\left(\frac{1\ lbf}{4.448\ N}\right) \rightarrow W_C = 121627\ lb$$

$W_{TOTAL} = W_{T1} + W_C \rightarrow W = 3307\ lb + 121627\ lb = 124934\ lb$

Therefore, the total weight of the 15,500-gallon storage tank at maximum capacity is 124,934 lbs.

<u>Weight of 5,000-gallon tank:</u>

From Norwesco tank specs, $W_{T2} = 844\ lbs$

$$V_C = 5000\ gal\ \left(\frac{1\ m^3}{264.2\ gal}\right) = 18.9251\ m^3$$

$$\rho_C = sg_C \times \rho_W \rightarrow 0.94\left(1000\frac{kg}{m^3}\right) \rightarrow \rho_C = 940\ kg/m^3$$

$$m_C = \rho_C V_C \rightarrow 940\frac{kg}{m^3}(18.9251\ m^3) \rightarrow m_C = 17789.6\ kg$$

$$W_C = m_C \cdot g \rightarrow 17789.6\ kg\left(9.81\frac{m}{s^2}\right) \rightarrow W_C = 174516\ N$$

$$W_C = 174516\ N\left(\frac{1\ lbf}{4.448\ N}\right) \rightarrow W_C = 39234.7\ lb$$

$$W_{TOTAL} = W_{T2} + W_C \rightarrow W = 844\ lb + 39234.7\ lb = 40078.7\ lb$$

Therefore, the total weight of the 5,000-gallon storage tank at maximum capacity is 40,078.7 lbs.

<u>Weight of 1,100-gallon tank:</u>

From Norwesco tank specs, $W_{T3} = 169\ lbs$

$$V_C = 1100\ gal\ \left(\frac{1\ m^3}{264.2\ gal}\right) = 4.1635\ m^3$$

$$\rho_C = sg_C \times \rho_W \rightarrow 0.94\left(1000\frac{kg}{m^3}\right) \rightarrow \rho_C = 940\ kg/m^3$$

$$m_C = \rho_C V_C \rightarrow 940\frac{kg}{m^3}(4.1635\ m^3) \rightarrow m_C = 3913.69\ kg$$

$$W_C = m_C \cdot g \rightarrow 3913.69\left(9.81\frac{m}{s^2}\right) \rightarrow W_C = 38393.3\ N$$

$$W_C = 38393.3\ N\left(\frac{1\ lbf}{4.448\ N}\right) \rightarrow W_C = 8631.59\ lb$$

$$W_{TOTAL} = W_{T3} + W_C \rightarrow W = 169\ lb + 8631.59\ lb = 8800.59\ lb$$

Therefore, the total weight of the 1,100-gallon storage tank at maximum capacity is 8,800.59 lbs.

Wind load on 15,500-gallon tank:

$$C_D = 1.12\ for\ semitubular\ cylinders$$

$$V_{AIR} = 12\ mph\ \left(\frac{5280\ ft}{1\ mile}\right)\left(\frac{1\ h}{3600\ s}\right)\left(\frac{1\ m}{3.281\ ft}\right) = 5.364\ m/s$$

$$A = BH \rightarrow H = 244\ in\ \left(\frac{1ft}{12in}\right)\left(\frac{1m}{3.281ft}\right) = 6.197\ m$$

$$B = 141\ in\left(\frac{1ft}{12in}\right)\left(\frac{1m}{3.281ft}\right) = 3.58123\ m$$

$$\therefore A = 6.197\ m \times 3.58123\ m = 22.194\ m^2$$

Since there is not a given density value for air at -20F, interpolation methods are used to get an accurate value.

$$-20°F \rightarrow °C = (-20 - 32)\left(\frac{5}{9}\right) = -28.8889°C$$

Using table E.1, values above and below the temperature,

$x = -28.8889°C$ $\qquad$ $y = \rho_{AIR}\ @ - 28.8889°C$

$x_1 = -30°C$ $\qquad$ $y_1 = 1.452\ kg/m^3$

$x_2 = -20°C$ $\qquad$ $y_2 = 1.394\ kg/m^3$

$$y = y_1 + \left[\left(\frac{x - x_1}{x_2 - x_1}\right)(y_2 - y_1)\right] \rightarrow y = 1.452\frac{kg}{m^3}\left[\left(\frac{-28.8889 - (-30)}{-20 - (-30)}\right)(1.394 - 1.452)\right]$$

$$y = 1.4456\frac{kg}{m^3}$$

$$\rho_{AIR} = 1.4456\frac{kg}{m^3}\ @ - 28.8889°C\ (or - 20°F)$$

Using the drag force equation, plug in all variables

$$F_D = C_D\frac{\rho_{AIR}V^2}{2} \cdot A \rightarrow (1.12)\left(\frac{1.4456\frac{kg}{m^3}\left(5.364\frac{m}{s}\right)^2}{2}\right)(22.194\ m^2)$$

$$F_D = 516.951\ N\ \left(\frac{1lb}{4.448\ N}\right)$$

$$\boxed{F_D = 116.221\ lbs}$$

Wind load on 5,000-gallon tank:

$C_D = 1.12$

$V_{AIR} = 5.364 \dfrac{m}{s}$

$A = BH \rightarrow H = 152\ in \left(\dfrac{1ft}{12in}\right)\left(\dfrac{1m}{3.281ft}\right) = 3.86061\ m$

$B = 102\ in \left(\dfrac{1ft}{12in}\right)\left(\dfrac{1m}{3.281m}\right) = 2.59067\ m$

$\therefore A = 2.59067\ m \times 3.86061\ m = 10.0016\ m^2$

$\rho_{AIR} = 1.4456\ kg/m^3$

$$F_D = C_D \frac{\rho_{AIR} V^2}{2} \cdot A \rightarrow (1.12)\left(\frac{1.4456 \frac{kg}{m^3}\left(5.364 \frac{m}{s}\right)^2}{2}\right)(10.0016\ m^2)$$

$F_D = 232.961\ N\ \left(\dfrac{1lb}{4.448\ N}\right)$

$\boxed{F_D = 52.3743\ lbs}$

<u>Wind load on 1,100-gallon tank:</u>

$C_D = 1.12$

$V_{AIR} = 5.364 \dfrac{m}{s}$

$A = BH \rightarrow H = 53\ in \left(\dfrac{1ft}{12in}\right)\left(\dfrac{1m}{3.281ft}\right) = 1.34613\ m$

$B = 87\ in \left(\dfrac{1ft}{12in}\right)\left(\dfrac{1m}{3.281m}\right) = 2.20969\ m$

$\therefore A = 2.20969\ m \times 1.34613\ m = 2.97453\ m^2$

$\rho_{AIR} = 1.4456\ kg/m^3$

$$F_D = C_D \frac{\rho_{AIR} V^2}{2} \cdot A \rightarrow (1.12)\left(\frac{1.4456 \frac{kg}{m^3}\left(5.364 \frac{m}{s}\right)^2}{2}\right)(2.97453\ m^2)$$

$F_D = 69.2839\ N\ \left(\dfrac{1lb}{4.448\ N}\right)$

$\boxed{F_D = 15.5764\ lbs}$

Summary:

The weights of the tanks were calculated using the assumption that they are at maximum capacity. The wind loads were calculated using the current wind speeds as shown by Accuweather. The results can be seen in the table below.

Weight and Wind Loads for Storage Tanks		
Vertical Storage Tank	**Weight at Maximum Capacity (lbs)**	**Wind Load at 12 mph (lbs)**
15,500-gallon	124,934	116.221
5,000-gallon	40,079	52.374
1,100-gallon	8,801	15.576

Table 5: Weight and Wind Loads for Storage Tanks

Materials:

High-density polyethylene resin tanks

Coolant (-20F to +105F)

Air @ -20F (coldest temperature in Dayton, Ohio)

Analysis:

When calculating the weight of the tanks, they are assumed to be at maximum capacity to yield worst-case-scenario results. While these will not usually be normal operating conditions, if the maximum weight values were not used, the foundations of the tanks may not be designed to adequately support the full load of the tank and coolant. Even though the civil engineers will use their own set of safety factors, the weight at maximum capacity should still be used when determining the weight of the tanks.

Open Channel for Drainage (task 12)
Purpose:

The purpose of this is to determine the minimum dimensions required for an open channel used to drain the clean coolant tank in case of an emergency.

Drawings and Diagrams:

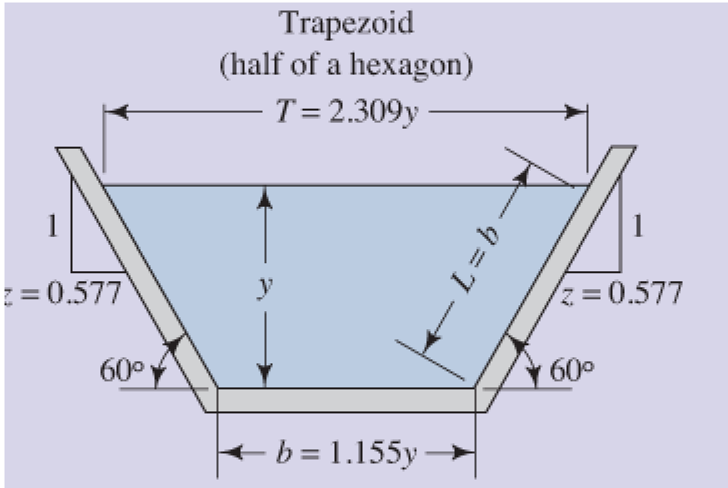


Figure VI-12: Open Channel Cross-Section Sketch

Sources:

Mott, R., Untener, J., "Applied Fluid Mechanics," 7th Edition. Pearson Education, Inc., (2015)

Design Considerations:

Steady state flow

Flow rate will be the same as it is in the system

Trapezoidal profile for the open channel

Concrete is assumed to be the channel material due to cost and availability

Channel is designed for maximum efficiency

Data and Variables:

Manning's Constant, n = 0.013

Slope = 0.001

Dimensions shown in sketch

Procedure:

Manning's equation is used to calculate the depth of the trapezoidal channel designed for maximum efficiency. The breadth, slope, wet perimeter, and hydraulic radius will be substituted into the equation and the iterative process is used to solve for the depth.

Calculations:

Manning's equation states:

$$(b+zy)y \times \left(\frac{b+zy}{b+2y \times \sqrt{1+z^2}}\right)^{\frac{2}{3}} = \frac{Qn}{1.49 \times \sqrt{S}}$$

Using excel and the iterative process to calculate depth, the following is calculated:

Q=	62.5	gpm	0.13925	ft^3/s
n=	0.013			
S=	0.10%	ft/ft		
A=	0.89366	ft^3		
R=	0.35915	ft		
W=	0.82942	ft		
T=	1.65884	ft		
WP=	2.48826	ft		

D (ft)	Q (ft^3/s)	% diff
0.7183	0.1393	0.0112%
0.7183		

D (in)	Q (gpm)
8.6196	62.507

Iterative Calculations				
Iteration	Guess D (C11)	Calculated Q (ft^3/s)	Converted Q (gpm)	% diff
1	1.00	0.5231453	234.8039001	275.69%
2	0.9	0.3432356	154.0548388	146.5%
3	0.80	0.2142803	96.17567747	53.88%
4	0.75	0.1655264	74.2934215	18.87%
5	0.72	0.1405893	63.10086199	0.96%
6	0.719	0.1398099	62.7510313	0.40%
7	0.7185	0.1394214	62.57666244	0.12%
8	0.7183	0.1392662	62.50701675	0.0%

Summary:

The following results show the construction of the trapezoidal channel

Trapezoidal Channel Data		
Variables	**Value**	
Flow Rate, Q	62.5	gpm
Slope, S	0.1	%
Area, A	0.89366	ft
Wet Perimeter, WP	2.48826	ft
Hydraulic Radius, R	0.35915	ft
Width, B	0.82942	ft
Taper, Z	0.35915	ft
Depth, D	0.7183	ft

Table 6: Open Channel Data

Materials:

Concrete channel

Coolant

Analysis:

For the open channel design, the calculation indicate that channel should have a width of 0.82942 feet, taper of 0.35915 feet, and a depth of 0.7183 feet. The actual design could constitute for a deeper depth or larger size, these calculations just determine the bare minimum requirements. If a catastrophic spill were to occur, the minimum open channel designs may not be able to withstand the drastic increase in fluid it is trying to contain.

Flow Rate

Tank Fill/Empty Times and Flow Rate (task 4)
Purpose:

Flow rate is needed to determine the fill and empty times of each vertical storage tank. The volumetric flow rate is essential for many subsequent calculations for the systems in the design.

Drawings and Diagrams:

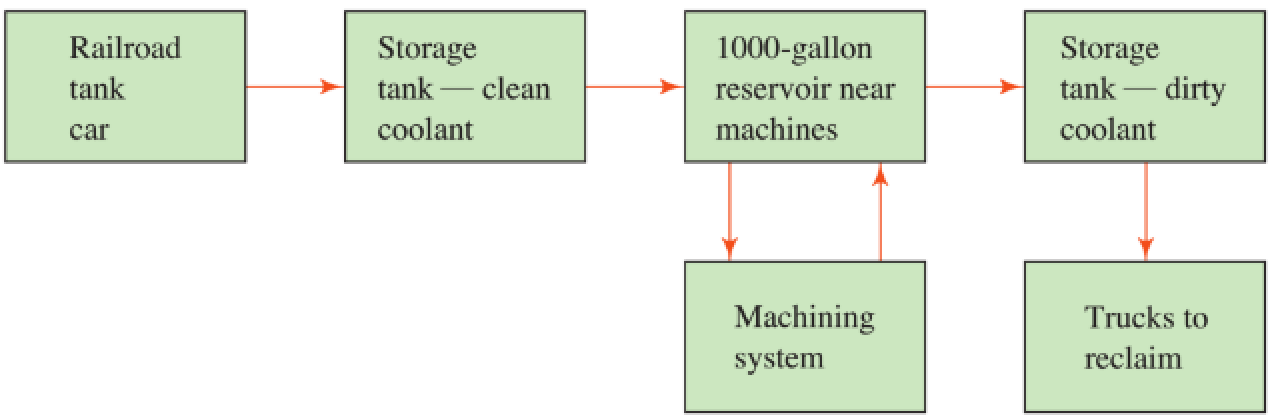


Figure VI-13: Diagram of Coolant System

Sources:

Mott, R., Untener, J., "Applied Fluid Mechanics," 7th Edition. Pearson Education, Inc., (2015)

Design Considerations:

Incompressible fluid

Steady state

Data and Variables:

$V_{Main\ Reservoir} = 15500\ gallons$

$V_{Waste} = 5000\ gallons$

$V_{Machine} = 1100\ gallons$

Procedure:

In order to determine the fill and empty times for the reservoirs, a fixed flow rate needs to be defined. The flow rate is found by dividing the volume of the tank by the time to fill and empty said volume. Once the flow rate is determined, it can be used to calculate the fill/empty times for each tank.

Calculations:

We decided upon and arbitrary flow rate of 62.5 GPM as it is pretty industry standard for cost efficient pumped systems.

Therefore, the time to fill and empty the 15,500-gallon tank can be determined using:

$$t = \frac{15500\ gal}{62.5\ GPM}$$

$$\boxed{t = 248\ minutes}$$

5,000-gallon tank fill/empty time:

$$t = \frac{5000\ gal}{62.5\ GPM}$$

$$\boxed{t = 80\ minutes}$$

1,100-gallon tank fill/empty time:

$$t = \frac{1100\ gal}{62.5\ GPM}$$

$$\boxed{t = 17.6\ minutes}$$

Summary:

The fill and empty times along with flow rate can be seen in the table below.

Storage Tank Fill/Empty Times and Flow Rate

Tank Capacity (gallons)	Time to Fill/Empty (minutes)	Flow Rate (GPM)
15,500	248	62.5
5,000	80	62.5
1,100	17.6	62.5

Table 7: Storage Tank Fill/Empty Times and Flow Rate

Materials:

Coolant

15,500-gallon storage tank

5,000-gallon storage tank

1,100-gallon storage tank

Analysis:

When determining the fill and empty times, it was assumed that the tanks were using their manufacturer-specified maximum capacity. These times determined do not pose any issue to the operation of the facility. The 17.6-minute time for the machine supply tank is reasonable when taking into consideration the need for emergency emptying of the tank. If a problem arises and the tank needs to be emptied, it is paramount that it be done in a rapid manner to ensure any debris or particulates that are the cause for the emergency dumping do not have time to make their way throughout the entire system.

Pipe Sizing

Piping Layout, Diameters, and Lengths (task 5)
Purpose:

To specify the layout, sizing, and material of the piping systems for the manufacturing facility.

Drawings and Diagrams:

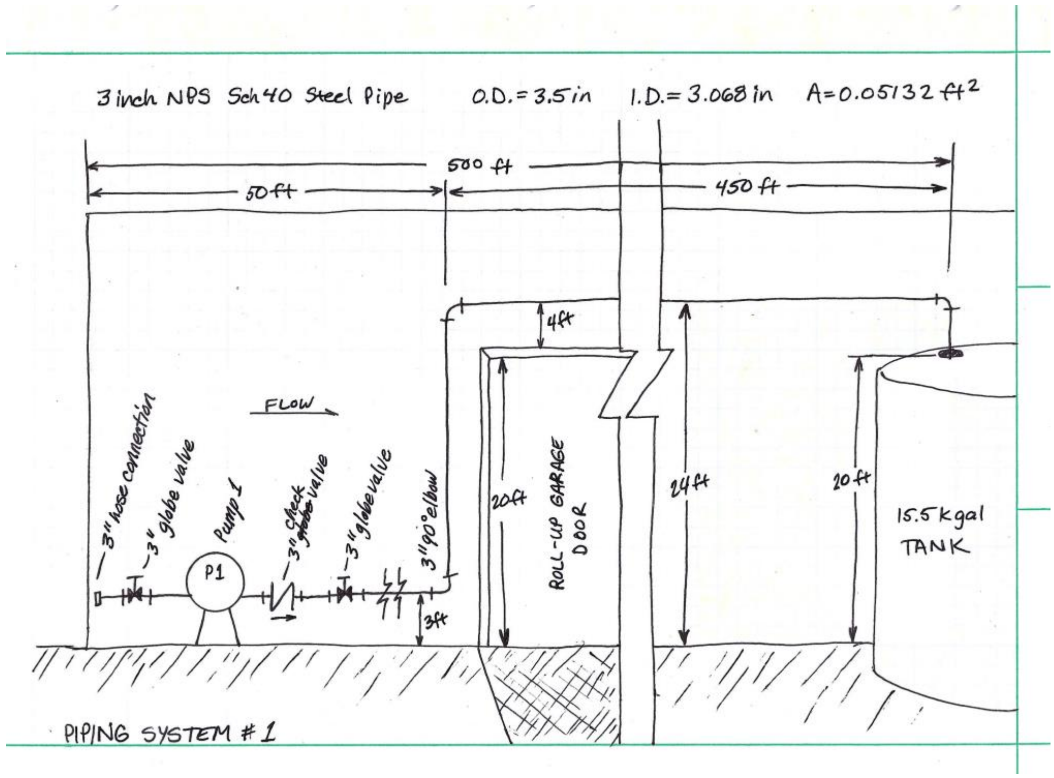


Figure VI-14: Piping System #1

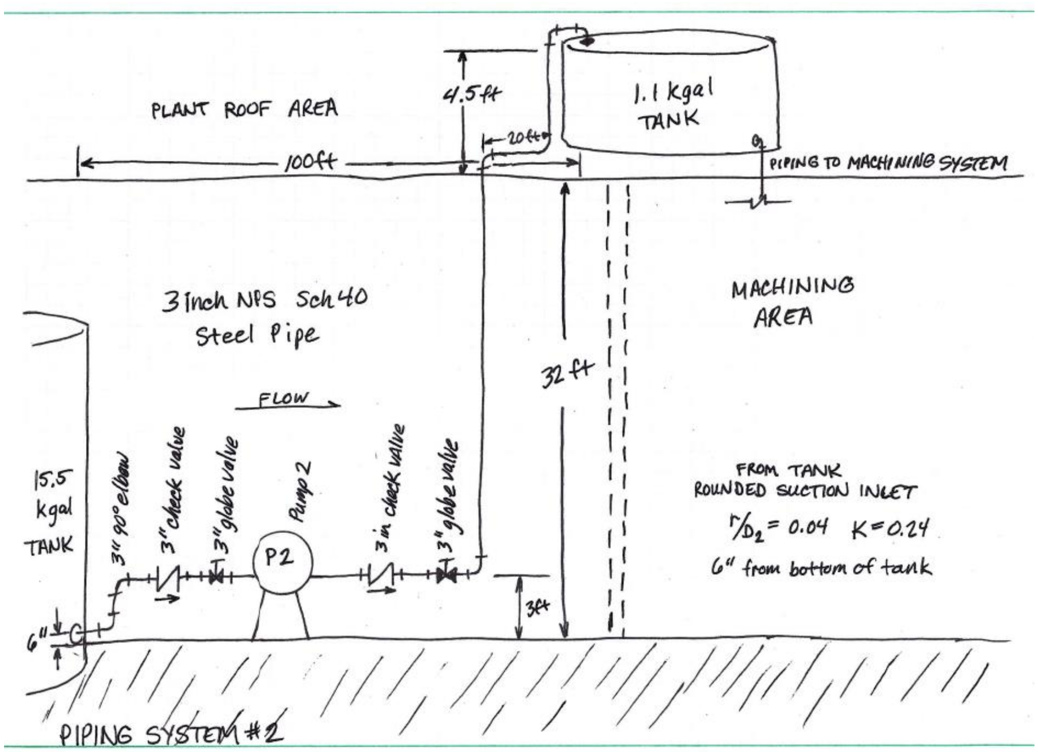


Figure VI-15: Piping System #2

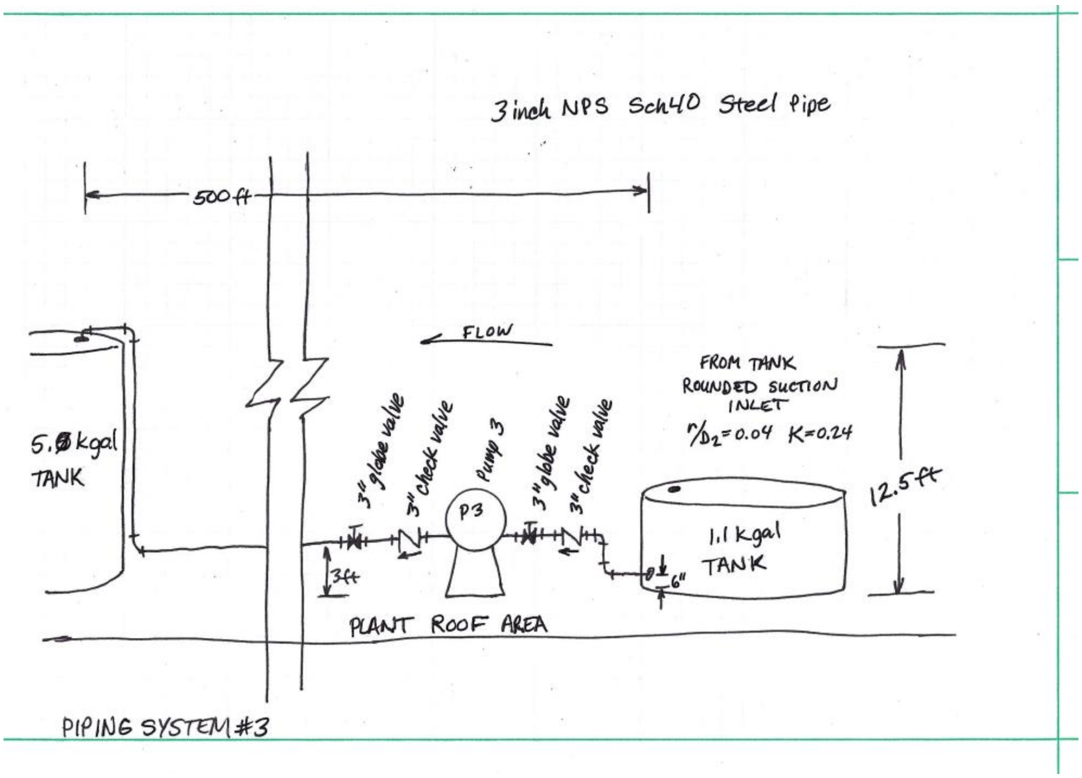


Figure VI-16: Piping System #3

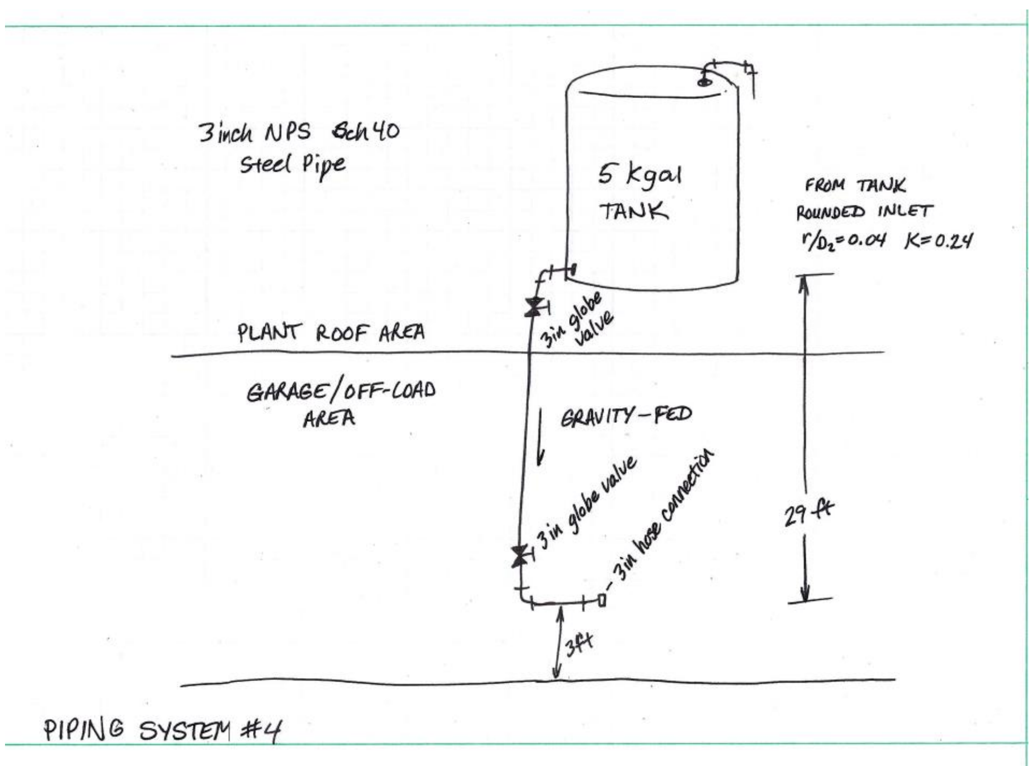


Figure VI-17: Piping System #4

Sources:

Mott, R., Untener, J., “Applied Fluid Mechanics,” 7th Edition. Pearson Education, Inc., (2015)

Design Considerations:

The piping systems will be outside, exposed to the elements. The fluid coolant flowing through the system has a freezing point higher than the lowest temperatures experienced at the facility’s location. The piping system supplying the machines is gravity fed.

Data and Variables:

Drawings and dimensions shown above

Procedure:

The flow rate equation is used to determine the inside diameter of piping needed for the system. The layout drawings are used to calculate the lengths of pipe needed.

Calculations:

Length of Pipe Needed:

$L_1 = 50\ ft + 21\ ft + 450\ ft$

$L_1 = 521\ ft$

$L_2 = 100\ ft + 29ft + 4.5\ ft$

$L_2 = 133.5\ ft$

$L_3 = 500\ ft + 12.5\ ft$

$L_3 = 512.5\ ft$

$L_4 = 29\ ft$

$L_{total} = 521\ ft + 133.5\ ft + 512.5\ ft + 29\ ft$

$\boxed{L_{total} = 1196\ ft}$

Summary:

For the complete facility piping system, 1,196 feet of 3-inch NPS Schedule 40 pipe is to be used. The inside diameter of this pipe is 3.068 inches.

Materials:

Schedule 40 Steel Pipe

Analysis:

The inside diameter of the pipe came from the table once 3-inch NPS Schedule 40 steel pipe was selected. The selected pipe size will be adequate to deliver the desired flow rate for the system.

Pipe Thickness (task 9)
Purpose:

To determine the appropriate wall thickness of the pipes in each system to ensure the pipes will withstand the pressures resulting from the operational process.

Drawings and Diagrams:

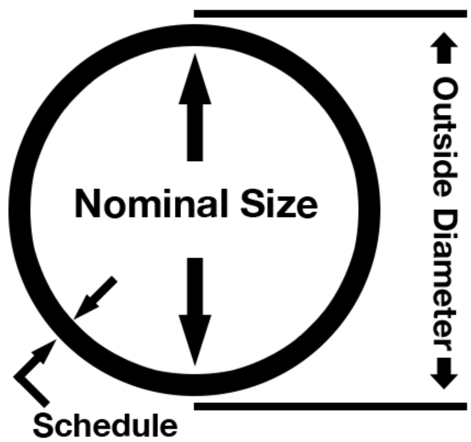


Figure 18: Pipe Thickness

Sources:

Mott, R., Untener, J., "Applied Fluid Mechanics," 7th Edition. Pearson Education, Inc., (2015)

Design Considerations:

Incompressible fluid

Steady state

Data and Variables:

Inside Diameter = 2.9 inches

Outside Diameter = 3.5 inches

Procedure:

An equation can be used to determine wall thickness, but the project design asks for readily available materials to be used. Therefore, the table from the book (located in the appendix) will be used to determine the wall thickness of the pipe.

Calculations:

There are no calculations for this task.

Summary:

The specified wall thickness for 3-inch Schedule 40 Steel Pipe is stated to be 0.3 inches.

Materials:

3-inch Schedule 40 Steel Pipe

Analysis:

The wall thickness chosen was taken from the textbook's supplied chart for schedule 40 steel pipe.

Fittings (task 6)
Purpose:

To determine the total type and quantity of fittings needed for the complete system.

Drawings and Diagrams:

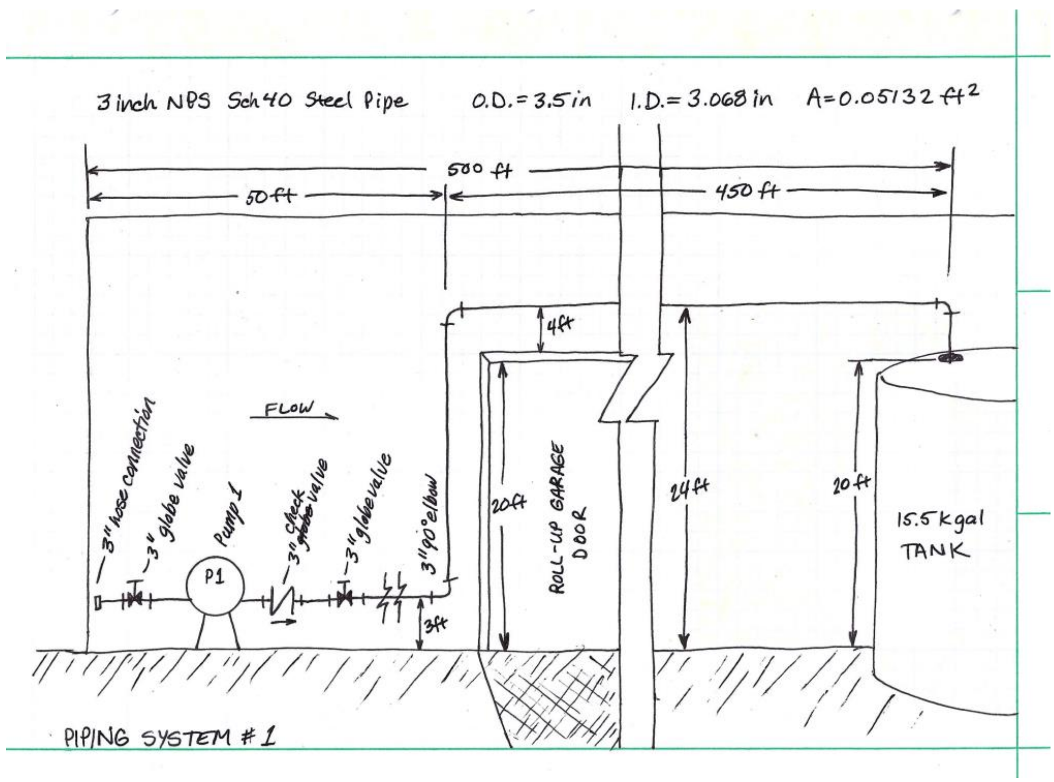


Figure VI-19: Piping System #1 with Fittings

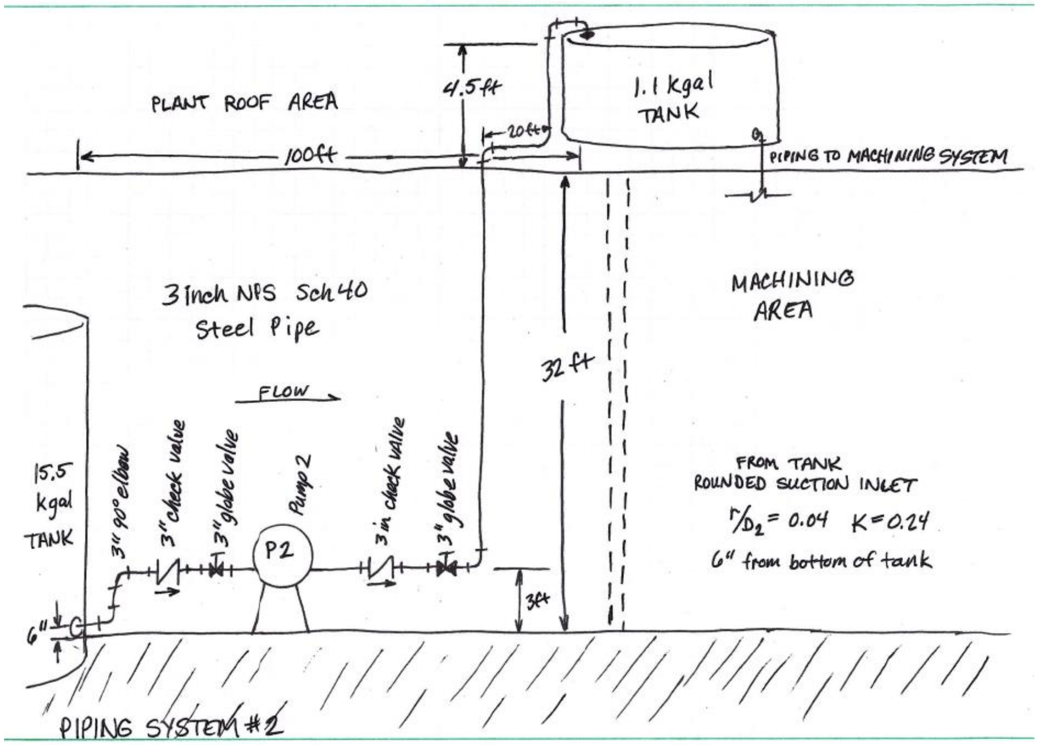


Figure VI-20: Piping System #2 with Fittings

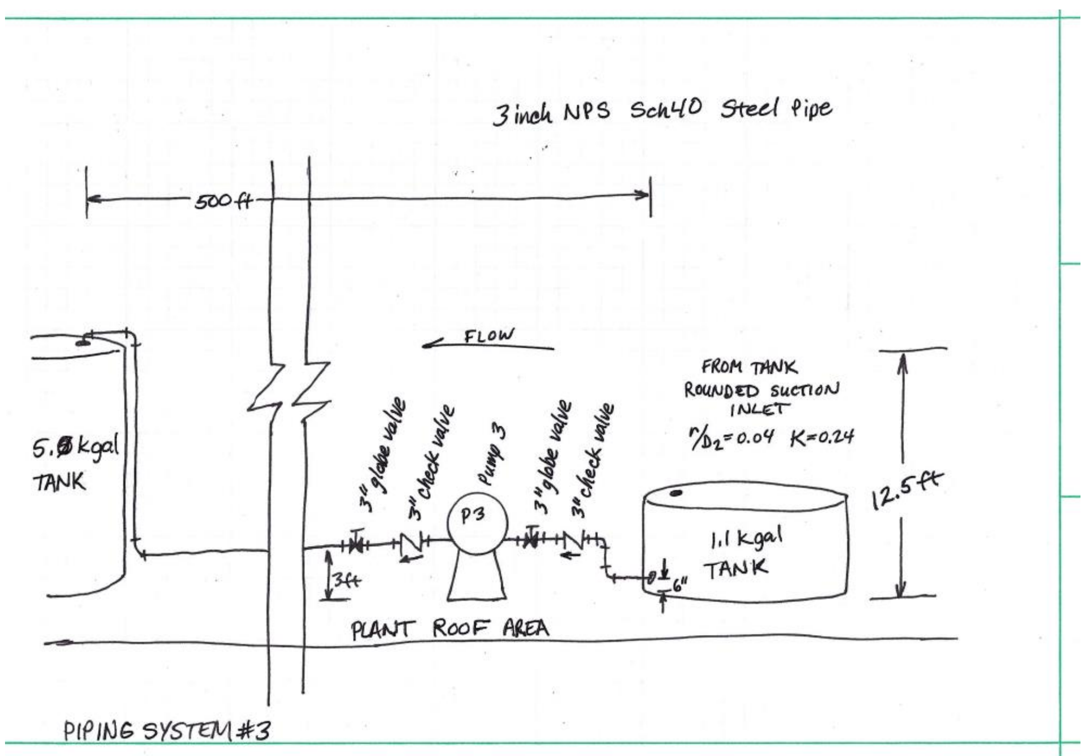


Figure VI-21: Piping System #3 with Fittings

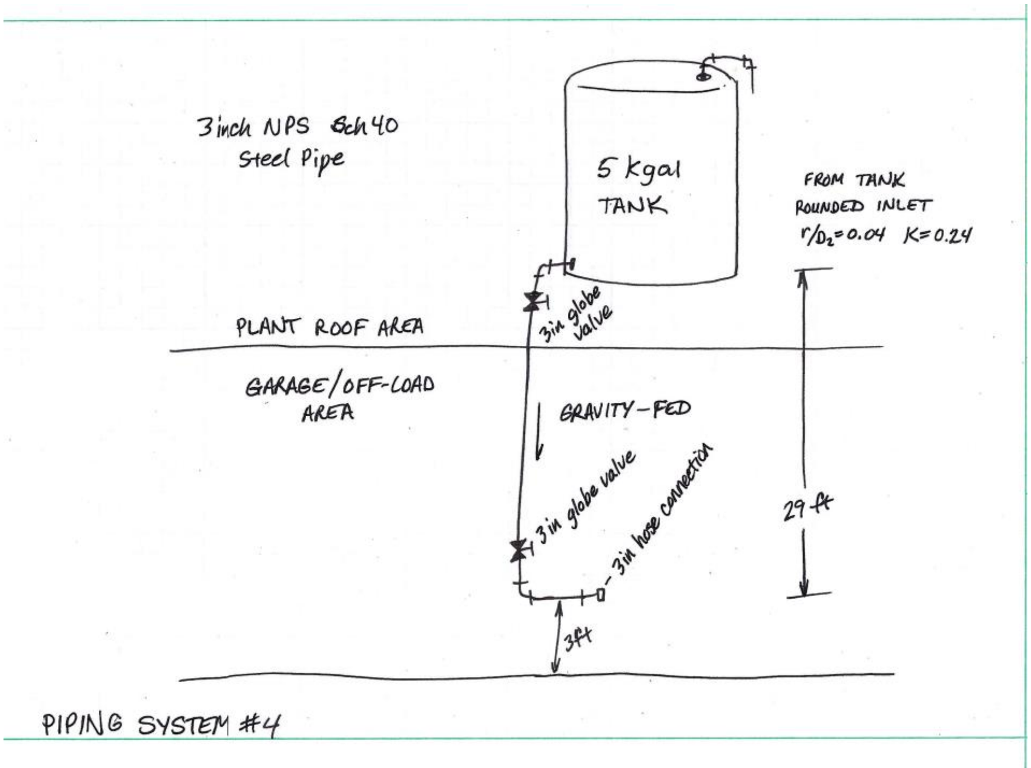


Figure VI-22: Piping System #4 with Fittings

Sources:

Mott, R., Untener, J., "Applied Fluid Mechanics," 7th Edition. Pearson Education, Inc., (2015)

Design Considerations:

Pipe sizes and locations are unchangeable. Fittings should be ordered to adhere to the design parameters.

Data and Variables:

Reference drawings above.

Procedure:

Using the aforementioned diagrams, valves and other pipe fittings were positioned to account for routine maintenance on each system as well as proper jointing of the pipes.

Calculations:

Calculations are not needed for this design task.

Summary:

A table of all necessary fittings can be seen below.

System Fitting Selection			
Type	**Quantity**	**Material**	**Size (inches)**
Hose Connection	2	Steel	3
Globe Valve	8	Steel	3
Check Valve, Swing Type	5	Steel	3
90° Elbow	16	Steel	3
Rounded Inlet	3	Steel	3

Table 8: Fitting Selection

Materials:

3-inch NPS Schedule 40 Steel Pipe

Analysis:

After designing the final layout of the piping system, elbows, valves, inlets, and hose connections were needing to allow the system to be functional. At each tank, an inlet is needed to connect the pipe. Hose connectors are needed for the train to offload coolant as well as for the trucks to haul coolant away. Elbows are needed wherever the pipes make a turn. Globe valves are needed for throttling and isolation purposes. Check valves are need to ensure the coolant can only flow through the system in the desired direction.

Water Hammer (task 10)
Purpose:

To determine if the system could suffer from issues due to water hammer.

Drawings and Diagrams:

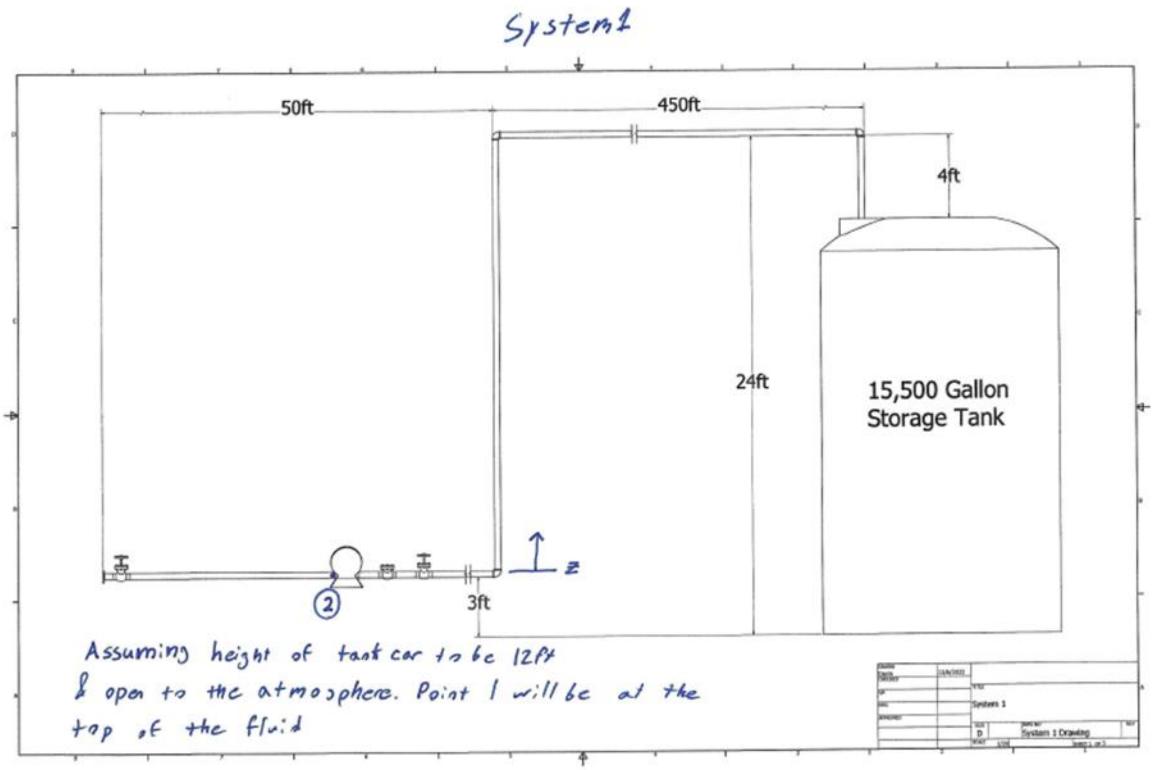


Figure VI-23: System 1 Layout for Water Hammer Calculations

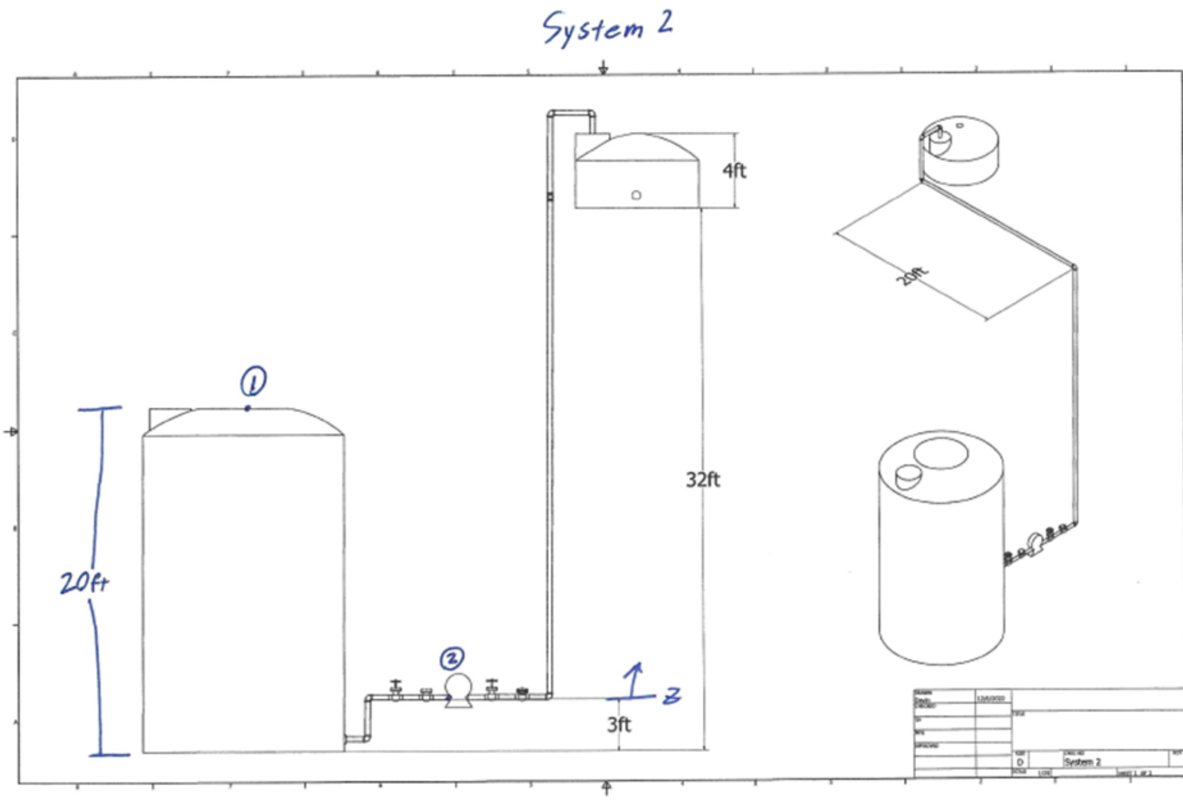


Figure VI-24: System 2 Layout for Water Hammer Calculations

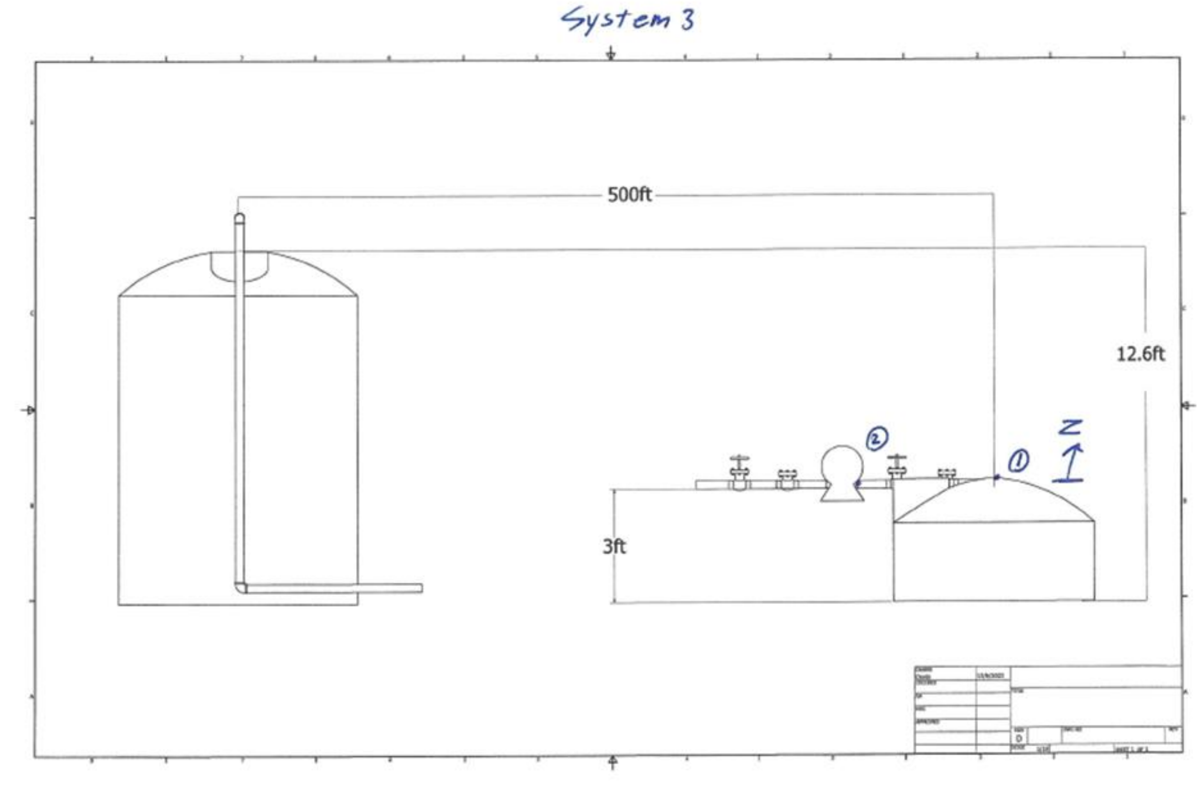


Figure VI-25: System 3 Layout for Water Hammer Calculations

Sources:

Mott, R., Untener, J., "Applied Fluid Mechanics," 7th Edition. Pearson Education, Inc., (2015)

Design Considerations:

All piped systems have the same flow rate and area; therefore, the velocity will be the same throughout.

Data and Variables:

$E_0 = 316000\ psi$

$D_1 = 3.068\ in$

$\rho_{coolant} = 0.94\left(62.4\frac{lb}{ft^3}\right) = 68.656\frac{lb}{ft^3} = 0.0339\frac{lb}{in^3}$

$E = 30000000\ psi$

$\delta = 0.216\ in$

Procedure:

The pressure change due to possible water hammer/cavitation is calculated. The highest pressure in each system was calculated using Bernoulli's. The pressure change was added to the highest pressure, including the atmospheric pressure. This, along with the boiling point at that pressure was taken into consideration to determine the risk for water hammer and/or cavitation.

Calculations:

$$C = \frac{\sqrt{\frac{E_0}{\rho}}}{\sqrt{1 + \frac{E_0 D}{E\delta}}} = \frac{\sqrt{\frac{316000\ psi}{0.339\frac{lb}{in^3}}}}{\sqrt{1 + \frac{(316000\ psi)(3.068\ in)}{(30000000\ psi)(0.216\ in)}}} = 2847.528\frac{in}{s}$$

$$C = 237.294\frac{ft}{s}$$

$$\Delta P = \rho VC = \left(68.656\frac{lb}{ft^3}\right)\left(2.7134\frac{ft}{s}\right)\left(237.294\frac{ft}{s}\right) = 44205.782\frac{lb}{ft \cdot s^2}$$

$$\Delta P = \frac{\left(44205.782\frac{lb}{ft \cdot s^2}\right)}{32.2\ ft/s^2} = 1372.85\frac{lb}{ft^2} = 9.834\ psi$$

System 1, Highest Pressure Calculation:

$$\frac{P_1}{\gamma} + \frac{V_1^2}{2g} + z_1 = \frac{P_2}{\gamma} + \frac{V_2^2}{2g} + z_2 + h_L$$

$$z_1 = \frac{P_2}{\gamma} + \frac{V_2^2}{2g} + h_L$$

$$h_L = \left(f\frac{L}{D}\frac{V_2^2}{2g}\right) + 340\frac{V_2^2}{2g} = \frac{V_2^2}{2g}\left(f\frac{L}{D} + 340\right) = \frac{0.544\frac{ft^2}{s}}{2\left(32.2\frac{ft}{s^2}\right)}\left(0.018\frac{10\ ft}{0.256\ ft} + 340\right) = 1.564\ ft$$

$$P_2 = \gamma\left(z_1 - \frac{V_2^2}{2g} - h_L\right) = 0.94\left(62.4\frac{lb}{ft^3}\right)\left(12\ ft - \frac{0.544\frac{ft^2}{s}}{2(32.2\ ft/s^2} - 1.564\ ft\right)$$

$$\boxed{P_2 = 0.7204\ psi}$$

System 2, Highest Pressure Calculation:

$$\frac{P_1}{\gamma} + \frac{V_1^2}{2g} + z_1 = \frac{P_2}{\gamma} + \frac{V_2^2}{2g} + z_2 + h_L$$

$$z_1 = \frac{P_2}{\gamma} + \frac{V_2^2}{2g} + h_L$$

$$h_L = \left(f\frac{L}{D}\frac{V_2^2}{2g}\right) + 340\frac{V_2^2}{2g} + 100\frac{V_2^2}{2g} = \frac{V_2^2}{2g}\left(f\frac{L}{D} + 340 + 100\right) = \frac{0.544\frac{ft^2}{s}}{2\left(32.2\frac{ft}{s^2}\right)}\left(0.018\frac{3\,ft}{0.256\,ft} + 340 + 100\right) = 2.02\,ft$$

$$P_2 = \gamma\left(z_1 - \frac{V_2^2}{2g} - h_L\right) = 0.94\left(62.4\frac{lb}{ft^3}\right)\left(17\,ft - \frac{0.544\frac{ft^2}{s}}{2(32.2\,ft/s^2} - 2.02\,ft\right)$$

$$\boxed{P_2 = 6.11\,psi}$$

System 3, Highest Pressure Calculation:

$$\frac{P_1}{\gamma} + \frac{V_1^2}{2g} + z_1 = \frac{P_2}{\gamma} + \frac{V_2^2}{2g} + z_2 + h_L$$

$$z_1 = \frac{P_2}{\gamma} + \frac{V_2^2}{2g} + h_L$$

$$h_L = 340\frac{V_2^2}{2g} + 100\frac{V_2^2}{2g} = \frac{0.544\frac{ft^2}{s}}{2\left(32.2\frac{ft}{s^2}\right)}(340 + 100) = 2.0196\,ft$$

$$P_2 = \gamma\left(z_1 - \frac{V_2^2}{2g} - h_L\right) = 0.94\left(62.4\frac{lb}{ft^3}\right)\left(3\,ft - \frac{0.544\frac{ft^2}{s}}{2(32.2\,ft/s^2} - 2.0196\,ft\right)$$

$$\boxed{P_2 = 0.4\,psi}$$

Summary:

Based on the yielded results, system 2 experiences the highest possible system pressure at 6.11 psi. With the additional pressure caused by rapidly closing a valve, 9.534 psi, in addition to atmospheric pressure, the highest possible pressure in the system would be 30.56 psia. At 30.56 psi, water has a boiling point of 250F, therefore, there is no risk of water hammer in this system. The highest pressures for each system can be seen in the table below.

Highest Possible System Pressure for Water Hammer Calculations	
System	**Highest Pressure (psi)**
1	0.7204
2	6.11
3	0.4

Table 9: Highest Possible System Pressures for Water Hammer Calculations

Materials:

All materials in the system. (See Bill of Materials)

Analysis:

When determining if a system has an issue with water hammer or cavitation, it is paramount to use the proper units when solving equations. A slight slip of a decimal on the wrong unit measurement can yield the results entirely inaccurate.

Provide pipeline support info

Type of Supports, Distance Between Supports, and Forces on Supports (task 13)
Purpose:

To determine the horizontal and vertical forces in piping system #1, the forces and deflection of pipe between supports, and the type of supports to be used by the civil engineers.

Drawings and Diagrams:

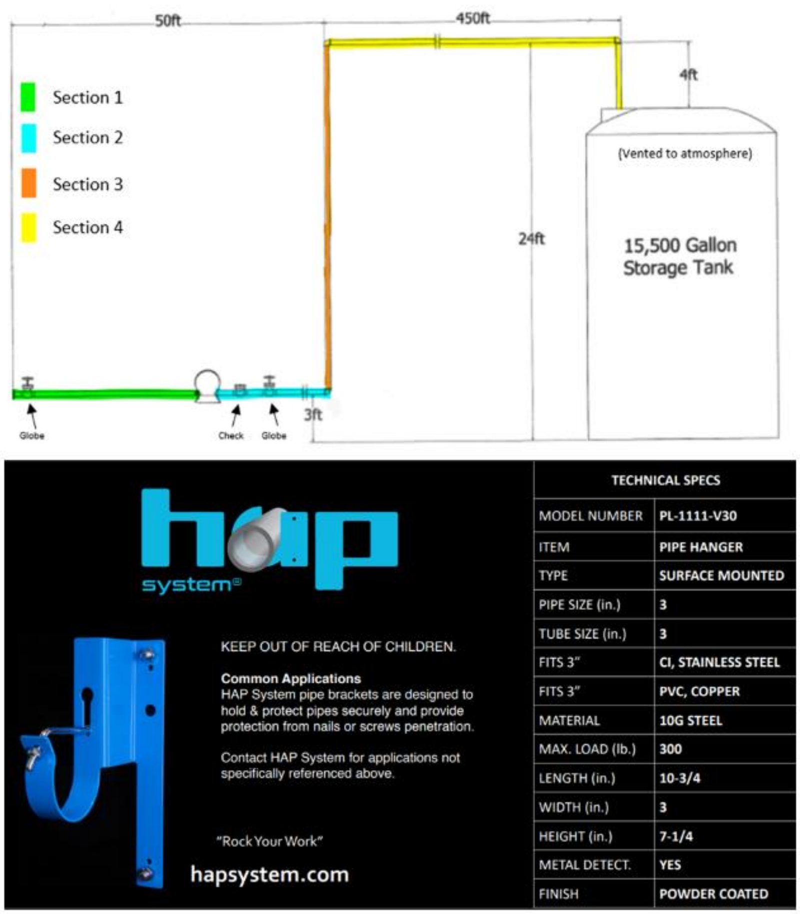


Figure VI-26: Pipeline Support Diagram

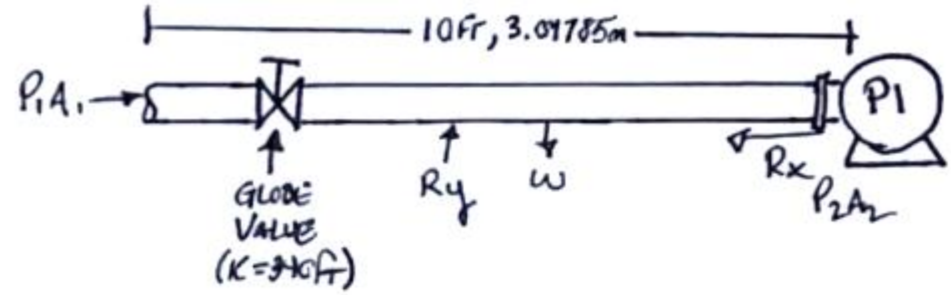


Figure VI-27: System 1 Pipe Support

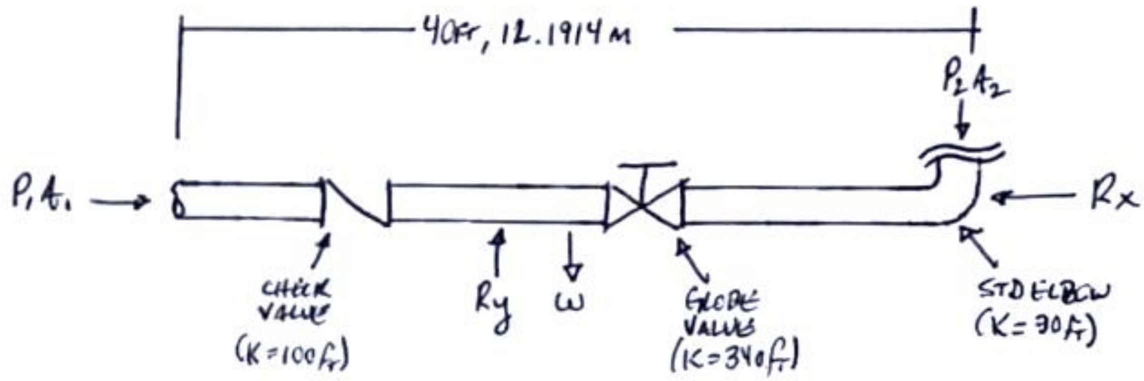


Figure VI-28: System 2 Pipe Support

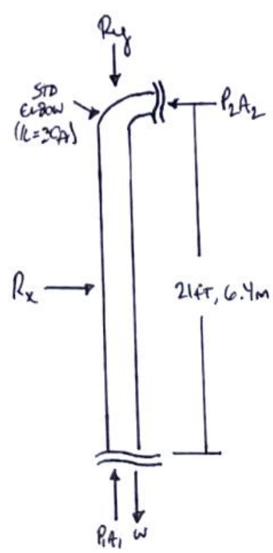


Figure VI-29: System 3 Pipe Support

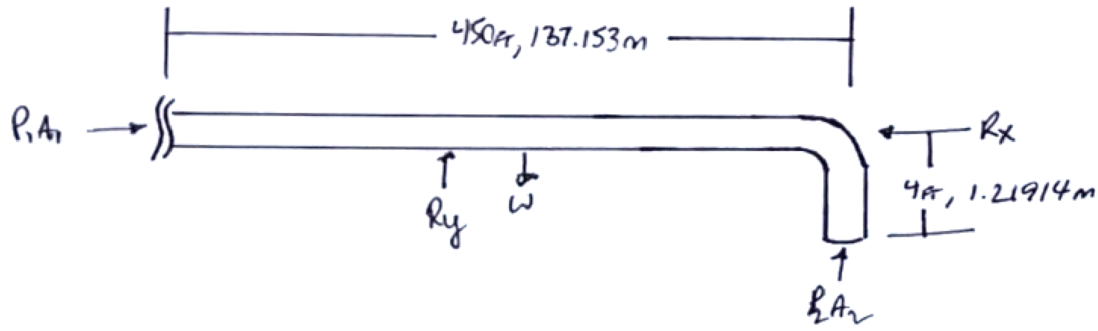


Figure VI-30: System 4 Pipe Support

Sources:

Mott, R., Untener, J., “Applied Fluid Mechanics,” 7th Edition. Pearson Education, Inc., (2015)

“3’ Lockable Hap Pipe Hanger.” GETPVF, https://getpvf.com/3-lockable-hap-pipe-hanger/.

Edge, Engineers. “Schedule 40 Steel Pipe Sizes & Dimensions Ansi.” Engineers Edge - Engineering, Design and Manufacturing Solutions, https://www.engineersedge.com/fluid_flow/steel-pipe-schedule-40.htm.

Design Considerations:

Steady State

Drawings and dimensions above

Data and Variables:

Flow Rate, $Q = 62.5\ GPM = 0.139198\frac{ft^3}{s} = 0.003942\frac{m^3}{s}$

Velocity, $V = 2.71235\frac{ft}{s} = 0.826762\frac{m}{s}$

Flow Area, $A = 0.05132\ ft^2 = 0.004768\ m^2$

Inside Pipe Diameter, $D = 3.068\ in = 0.0779\ m$

Outside Pipe Diameter, $D_O = 3.5\ in$

Pipe Friction Factor, $f = 0.018$ (from energy loss calculations)

Specific Weight (coolant), $\gamma_C = 9221.4\ N/m^3$

Density (coolant), $\rho_C = 940\ kg/m^3$

Young’s Modulus (SCH 40 Steel), $E = 29500000\ psi$

Procedure:

The forces for each section must be calculated first. The system was separated into three sections to make the calculations simpler, and they must be evaluated separately for deflection. This is due to the fact that there are various lengths of piping and different components for each which will require different support spacing.

To calculate the deflection in the piping, the total weight of each piping leg must be found by obtaining the sum of the vertical reaction forces and the weight of the piping itself. The empty weight per foot of 3" NPS SCH 40 Steel Pipe is 7.58 lbs. The formula for maximum deflection of a simply supported beam under uniform loading conditions is used. To determine the percentage of deflection relative to the overall pipe diameter of 3.5 inches with the goal of being less than 1% overall deflection between support hangers.

Calculations:

System 1:

$$\Sigma F = \rho Q(V_2 - V_1)$$

$$P_1 A_1 - R_x = \rho Q(V_2 - V_1)$$

$$\frac{P_1}{\gamma_c} = h_{L1-2} = f\frac{L}{D}\frac{V^2}{2g} + K_{qv}\frac{V^2}{2g} = (0.018)\left(\frac{3.04785m}{0.0779m}\right)\left(\frac{0.826762^2\frac{m}{s}}{2\left(9.81\frac{m}{s^2}\right)}\right) + (340 \times 0.018)\left(\frac{0.826762^2\frac{m}{s}}{2\left(9.81\frac{m}{s^2}\right)}\right)$$

$$\frac{P_1}{\gamma_c} = 0.237748\, m$$

$$P_1 = 0.237748\, m\,(\gamma_c) = 0.237748\, m\left(9221.4\frac{N}{m^3}\right)$$

$$P_1 = 2192.37\, N/m^2$$

$$R_x = P_1 A_1 \rho Q V_1 = \left(2192.37\frac{N}{m^2} \times 0.004768\, m^2\right) + \left(940\frac{kg}{m^3} \times 0.003942\frac{m^2}{s} \times 0.826762\frac{m}{s}\right)$$

$$R_x = 13.5468\, N \times \frac{1lb}{4.448N}$$

$$\boxed{R_x = 3.04\, lb}$$

$$R_y = \omega - \rho Q V_2 = \gamma_c A L - \rho Q V_2$$

$$R_y = \left(9221.4\frac{N}{m^3} \times 0.004768\, m^2 \times 3.04785\, m\right) - \left(940\frac{kg}{m^3} \times 0.003942\frac{m^3}{s} \times 0.826762\frac{m}{s}\right)$$

$$R_y = 130.943\, N \times \frac{1lb}{4.448N}$$

$$\boxed{R_y = 29.44\, lb}$$

System 2:

$P_1A_1 - R_x = \rho Q(V_2 - V_1)$

$\frac{P_1}{\gamma_c} = h_{L1-2} = f\frac{L}{D}\frac{V^2}{2g} + K_{cv}\frac{V^2}{2g} + K_{qv}\frac{V^2}{2g} + K_e\frac{V^2}{2g}$

$\frac{V^2}{2g} = \left(\frac{\frac{0.826762^2 m}{s}}{2\left(9.81\frac{m}{s^2}\right)}\right) = 0.034839\ m$

$\frac{P_1}{\gamma_c} = (0.018)\left(\frac{12.1914m}{0.0779m}\right)(0.034839\ m) + (100 \times 0.018)(0.034839\ m) + (340 \times 0.018)(0.034839\ m) + (30 \times 0.018)(0.034839\ m)$

$h_{L1-2} = 1.27615\ m \times 9221.4\frac{N}{m^3}$

$P_1 = 11767.9\frac{N}{m^2}$

$R_x = P_1A_1 + \rho QV_1 = \left(11767.9\frac{N}{m^2} \times 0.004768\ m^2\right) + \left(940\frac{kg}{m^3} \times 0.003942\frac{m^3}{s} \times 0.826762\frac{m}{s}\right)$

$\boxed{R_x = 13.3\ lb}$

$R_y = \omega - \rho QV_2 = \left(9221.4\frac{N}{m^3} \times 0.004768\ m^2 \times 12.1914\ m\right) - (940 \times 0.003942 \times 0.826762)$

$R_y = 119.821\ lb$

<u>System 3:</u>

$R_x - P_2A_2 = \rho Q(V_2 - V_1)$

$\frac{P_2}{\gamma_c} = h_{L1-2} = f\frac{L}{D}\frac{V^2}{2g} + K_e\frac{V^2}{2g} + z_2$ $\qquad \frac{V^2}{2g} = 0.034839\ m$

$= (0.018)\left(\frac{6.4m}{0.0779m}\right)(0.034839\ m) + (30 \times 0.018)(0.034839\ m) + (6.4\ m)$

$h_{L1-2} = 6.47033\ m$

$P_2 = 9221.4\frac{N}{m^3} \times 6.47033\ m$

$P_2 = 59665.5\ N/m^2$

$R_x = P_2A_2 + \rho Q(V_2) = \left(59665.5\frac{N}{m^2} \times 0.004768\ m^2\right) + \left(940\frac{kg}{m^3} \times 0.003942\frac{m^3}{s} \times 0.826762\frac{m}{s}\right)$

$\boxed{R_x = 64.6468\ lb}$

$$R_y = P_1 A_1 - \omega - \rho Q V_1 = \left(59665.5\,\frac{N}{m^2} \times 0.004768\ m^2\right) - \left(9221.4\,\frac{N}{m^3} \times 0.004768\ m^2 \times 6.4\ m\right) - \left(940\,\frac{kg}{m^3} \times 0.003942\,\frac{m^3}{s} \times 0.826762\,\frac{m}{s}\right)$$

$$\boxed{R_y = 0.0065\ lb}$$

<u>System 4:</u>

$$P_1 A_1 - R_x = \rho Q (V_2 - V_1)$$

$$\frac{P_1}{\gamma_c} = h_{L1-2} = f\,\frac{L}{D}\frac{V^2}{2g} + K_e\,\frac{V^2}{2g} \qquad \frac{V^2}{2g} = 0.034839\ m$$

$$= (0.018)\left(\frac{137.153\ m + 1.21914\ m}{0.0779m}\right)(0.034839\ m) + (30 \times 0.018)(0.034839\ m)$$

$$P_1 = 1.13271\ m \times 9221.4\ N/m^3$$

$$P_1 = 10445.2\ N/m^2$$

$$R_x = P_1 A_1 + \rho Q V_1 = \left(10445.2\,\frac{N}{m^2} \times 0.004768\ m^2\right) + \left(940\,\frac{kg}{m^3} \times 0.003942\,\frac{m^3}{s} \times 0.826762\,\frac{m}{s}\right)$$

$$\boxed{R_x = 11.885\ lb}$$

$$R_y = \omega - \rho Q V_1 = \left(9221.4\,\frac{N}{m^3} \times 0.004768\ m^2 \times 138.372\ m\right) - \left(940\,\frac{kg}{m^3} \times 0.003942\,\frac{m^3}{s} \times 0.826762\,\frac{m}{s}\right)$$

$$\boxed{R_y = 1367.09\ lb}$$

For the total forces of the sections and pipe deflections, handwritten calculations are included below.

TOTAL FORCES OF SECTIONS (WEIGHT + FORCES)

SECTION 1: $R_y = 29.441 \text{ lbs}$; WEIGHT OF PIPE, $W_p = 10\text{ FT}(7.58 \frac{\text{lb}}{\text{FT}}) = 75.8 \text{ lbs}$

$$W_T = R_y + W_p = 29.441 \text{ lbs} + 75.8 \text{ lbs} = 105.241 \text{ lbs}$$

SECTION 2,3,4: $R_{y_T} = R_{y_2} + R_{y_3} + R_{y_4}$; WEIGHT OF PIPE, $W_{P_T} = W_{P_2} + W_{P_3} + W_{P_4}$

$= 119.821 + 0.0065 + 1367.09 \text{ lbs}$
$= 1486.92 \text{ lbs}$

$= (40\text{ FT} + 21\text{ FT} + 454\text{ FT})(7.58 \frac{\text{lb}}{\text{FT}})$
$= 3903.7 \text{ lbs}$

$$W_T = R_{y_T} + W_{P_T} = 1486.92 \text{ lbs} + 3903.7 \text{ lbs} = 5390.62 \text{ lbs}$$

DEFLECTION OF PIPING

DEFLECTION, $D = wL^4/384EI$

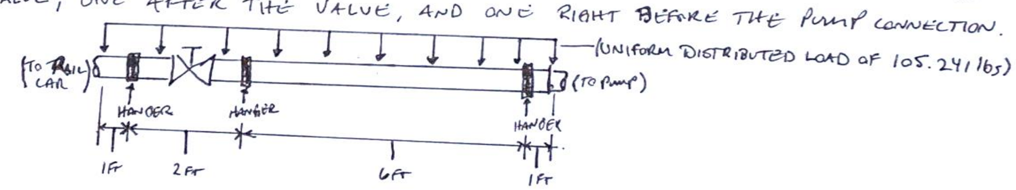
w = FORCE APPLIED PER UNIT LENGTH
L = LENGTH BETWEEN SUPPORTS
E = YOUNGS MODULUS
I = MOMENT OF INERTIA

FOR 3" NPS SCHEDULE 40, $I = \frac{\pi(D^4 - d^4)}{64} = \frac{\pi(3.5^4 - 3.068^4)}{64} = 3.017 \text{ in}^4$

$$E = 29,500,000 \text{ psi}$$

FOR SECTION 1, WE WILL USE FORMULA FOR FIXED SIMPLE BEAM UNDER UNIFORM DISTRIBUTED LOAD, THIS WILL GIVE US MAXIMUM DEFLECTION AT CENTER OF PIPE SECTION.

I WILL START BY PLACING 3 HANGERS ON THE PIPE, ONE BEFORE THE VALVE, ONE AFTER THE VALVE, AND ONE RIGHT BEFORE THE PUMP CONNECTION.



SINCE THE MOST DISTANCE BETWEEN HANGER SUPPORTS IS 6 FT, I WILL USE THIS FOR MY DISTANCE. $6\text{ FT} = 72 \text{ in}$; $\frac{105.241 \text{ lbs}}{120 \text{ in}} = 0.87701 \frac{\text{lb}}{\text{in}} \times 72 \text{ in} = 63.1446 \text{ lb}$

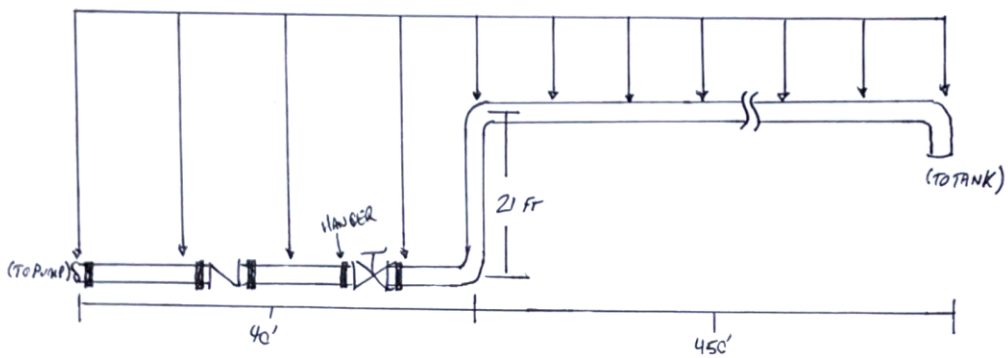
$$D = \frac{wL^4}{384EI} \rightarrow \frac{\left(\frac{63.1446 \text{ lb}}{72 \text{ in}}\right)(72 \text{ in})^4}{384(29,500,000 \frac{\text{lb}}{\text{in}^2})(3.017 \text{ in}^4)} = 0.00069 \text{ in}$$

$$\% \text{ DEFLECTION} = \frac{0.00069 \text{ in}}{3.5 \text{ in}} = 0.000197 \times 100 = 0.02\% \text{ DEFLECTION}$$

THEREFORE, THE SUPPORT LOCATIONS ARE ADEQUATE.

FOR THE OTHER SECTIONS DOWNSTREAM OF THE PUMP, I WILL GUESS THE SUPPORTS TO BE 12 FEET APART (EXCEPT FOR NEAR COMPONENTS AND ELBOWS).

$12\text{ FT} = 144\text{ in}; \quad \frac{5390.62\text{ lbs}}{6180\text{ in}} = 0.872269\frac{\text{lbs}}{\text{in}} \times 144\text{ in} = 125.607\text{ lbs}$



$$D = \frac{wL^4}{384EI} \longrightarrow \frac{\left(\frac{125.607\text{ lbs}}{144\text{ in}}\right)(144\text{ in})^4}{384\left(29{,}500{,}000\frac{\text{lb}}{\text{in}^2}\right)(3.017\text{ in}^4)} = 0.010974\text{ in}$$

$$\%\text{ DEFLECTION} = \frac{0.010974\text{ in}}{3.5\text{ in}} = 0.003135 \times 100 = 0.314\%$$ MAX DEFLECTION BETWEEN HANGERS

THE SUPPORT HANGER LOCATIONS CAN BE SPACED 12 FT APART OTHER THAN AT VALVES, CONNECTIONS, AND ELBOWS AND WILL ADEQUATELY SUPPORT THE SYSTEM FORCES.

Summary:

The forces for sections 1 and 2 were calculated separately due to the pump being mounted to a foundation on the ground. Total forces for sections 2, 3, and 4 were combined for the deflection calculations due to only being supported by hangers other than at the pump discharge flange.

For all pipe sections, support hangers will be installed within 1 ft of flanged connections and pipe elbows, as well as every 12 ft between these sections where applicable by mounting to the exterior walls of the building. This will limit pipe deflection between support hangers to <1% under normal system operating conditions.

Materials:

Coolant

3” SCH 40 Steel Pipe

10G Steel 3” HAP Pipe Hangers

Analysis:

The section with the greatest force was section 4 due to the overall length being the greatest. Given that the deflection was less than 1%, the distance of 12 ft between hangers was decided upon. It could have been adjusted to reduce the deflection even more, but the deflection was within tolerance and increasing the number of hangers would mean increasing costs for the customer.

Energy losses (task 7) make a table of all energy losses and analyze them

Purpose:

To determine the losses in the system for evaluation towards pump requirements.

Drawings and Diagrams:

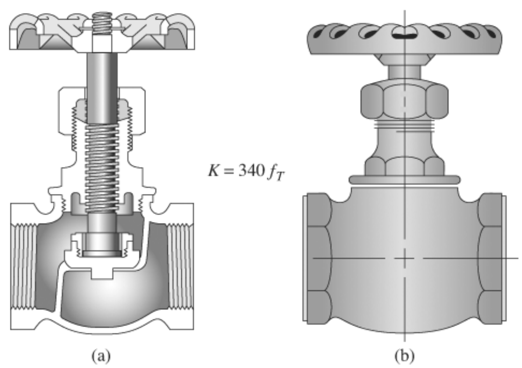


Figure VI-31: Globe Valve

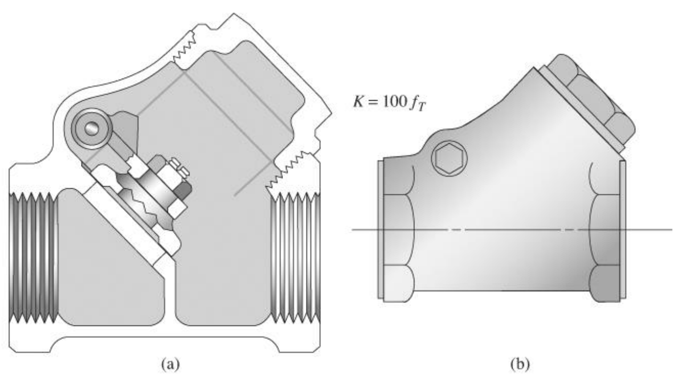


Figure VI-32: Check Valve, Swing Type

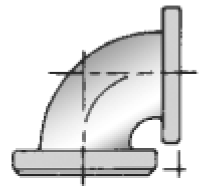


Figure VI-33: 90 deg Elbow

Sources:

Mott, R., Untener, J., "Applied Fluid Mechanics," 7th Edition. Pearson Education, Inc., (2015)

Design Considerations:

Incompressible fluids

Isothermal process

All minor losses

T_{AVG} > 50F

Data and Variables:

K values given in above diagrams

Procedure:

Reynold's number must be calculated for the system to determine the losses. The Reynold's number will be used to determine the friction factor for the system. The friction factor is used to calculate the losses of each component, which will be added to the losses in the pipe to give the total system losses.

Calculations:

Calculations for this are required to be done by hand.

$f = 0.018$

— SYSTEM 1

$$h_{A_1} = (\cancel{h_{ent_S}}^{\varnothing} + 2h_{v_{gl}} + 3h_{elbow} + h_{v_{ch}} + h_{pipe_1} + \cancel{h_{exit_D}}^{\varnothing}) + Z_2 - Z_1$$

$$\cancel{h_{ent_S}}\quad h_{v_{globe}} = 340 \cdot \frac{v^2}{2g} = 0.716\text{ ft}$$

$$h_{elbow} = 30 \cdot \frac{v^2}{2g} = 0.063\text{ ft} \qquad \Delta Z_1 = 17\text{ ft}$$

$$h_{v_{check}} = 100 \cdot \frac{v^2}{2g} = 0.210\text{ ft}$$

$$h_{pipe_1} = f_T \cdot \frac{525\text{ ft}}{0.256\text{ ft}} = 0.018 \cdot \frac{525}{0.256} = 36.91$$

$$\cancel{h_{A_1} = (0.063\text{ ft} + 0.210\text{ ft} +}$$

$$h_{A_1} = 3(0.063\text{ ft}) + 2(0.716\text{ ft}) + (0.210\text{ ft}) + (36.91) + 17\text{ ft}$$

$$h_{A_1} = 55.745\text{ ft}$$

— SYSTEM 2

$$h_{A_2} = (h_{ent_S} + 7h_{elbow} + 2h_{v_{ch}} + 2h_{v_{gl}} + h_{pipe_2}) + Z_2 - Z_1$$

$$h_{ent_S} = 0.24 \cdot \frac{v^2}{2g} = 0.051\text{ ft}$$

$$h_{elbow} = 30 \cdot \frac{v^2}{2g} = 0.063\text{ ft} \qquad \Delta Z_2 = 39\text{ ft}$$

$$h_{v_{check}} = 100 \cdot \frac{v^2}{2g} = 0.210\text{ ft}$$

$$h_{v_{globe}} = 340 \cdot \frac{v^2}{2g} = 0.716\text{ ft}$$

$$h_{pipe_2} = f_T \cdot \frac{156\text{ ft}}{0.256\text{ ft}} = 0.018 \cdot \frac{156}{0.256} = 10.98\text{ ft}$$

$$h_{A_2} = (0.051\text{ ft}) + 7(0.063\text{ ft}) + (0.210\text{ ft})2 + 2(0.716\text{ ft}) + 10.98\text{ ft} + 39\text{ ft}$$

$$h_{A_2} = 52.33\text{ ft}$$

– SYSTEM 3

$$h_{A_3} = (h_{ent_s} + 5h_{elbow} + 2h_{v_{ch}} + 2h_{v_{gl}} + h_{pipe_3}) + Z_2 - Z_1$$

same values as System 1 & 2 except for h_{pipe} & ΔZ_3

$$h_{A_3} = (0.051\,ft) + 5(0.063\,ft) + 2(0.210\,ft) + 2(0.716\,ft) + 36.26\,ft + 12\,ft$$

$$h_{A_3} = 50.48\,ft$$

Summary:

A table summarizing the losses for all systems can be seen below.

Total System Losses	
System Number	**Total Losses (ft)**
1	55.745
2	52.33
3	50.48

Table 10: System Energy Losses

Materials:

3-inch NPS Schedule 40 Steel Pipe

Gate Valves

Elbows

Check Valves

Coolant

Analysis:

When calculating the losses for each system, the fourth was neglected due to it being gravity fed. The three calculated systems were shown to have very similar losses compared to each other. The losses are going to be used when determining the selection of pumps.

Pump selection

Pump Requirements (tasks 8 and 15)

Purpose:

To determine the least amount of required power the system needs to be able to transport the coolant to all appropriate areas based on design criteria.

Drawings and Diagrams:

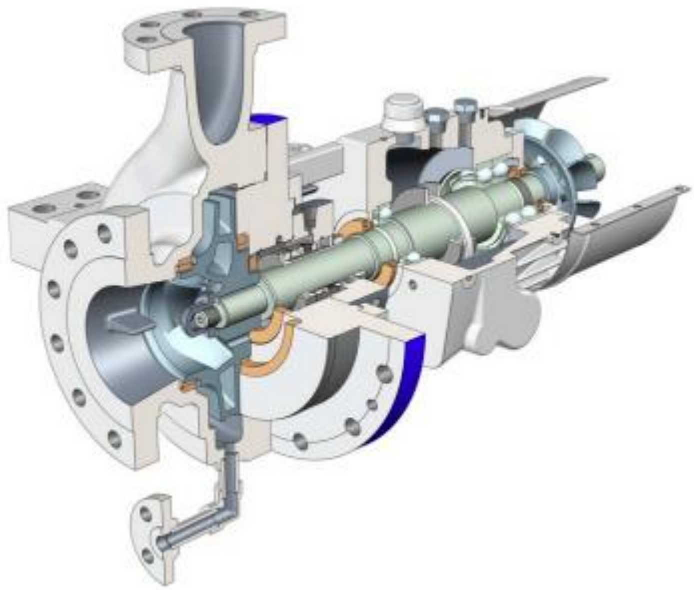


Figure 34: Sulzer Pump

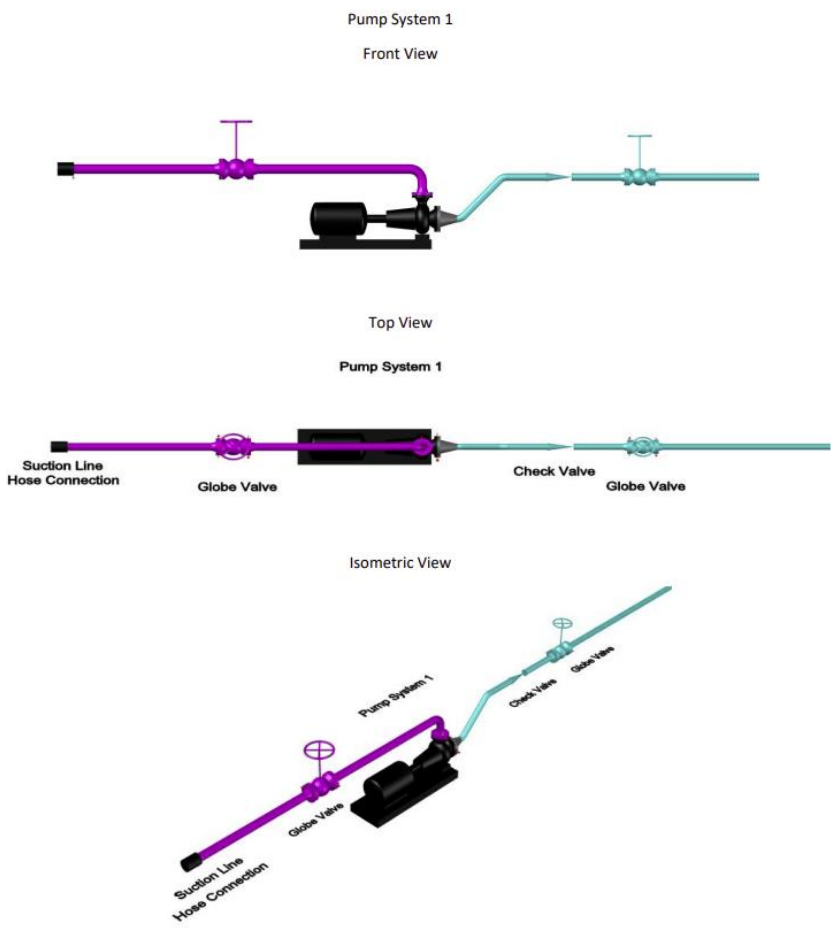


Figure 35: Pump System 1 Views

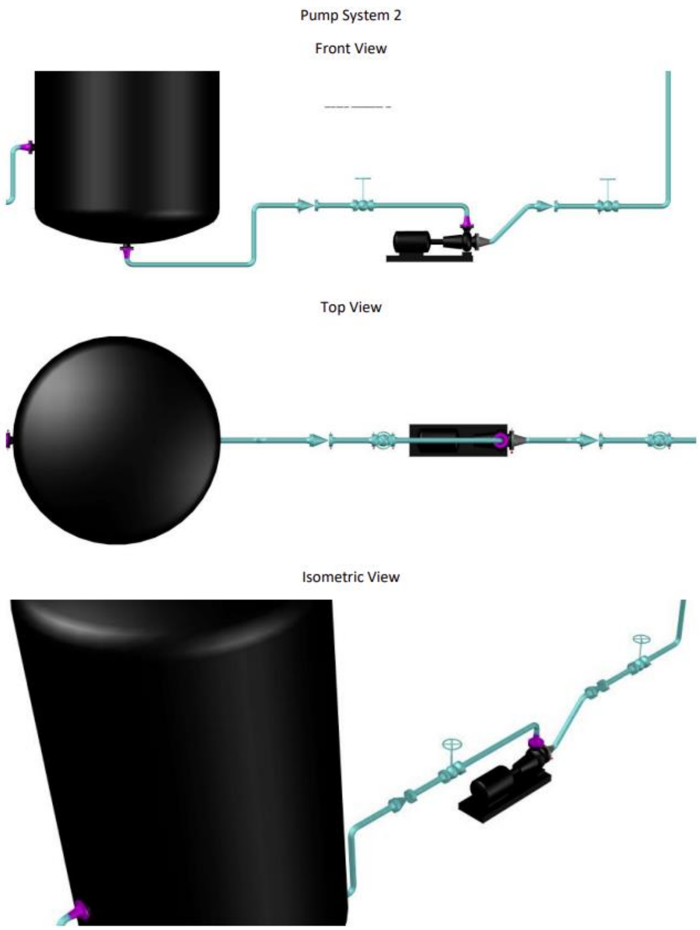


Figure 36: Pump System 2 Views

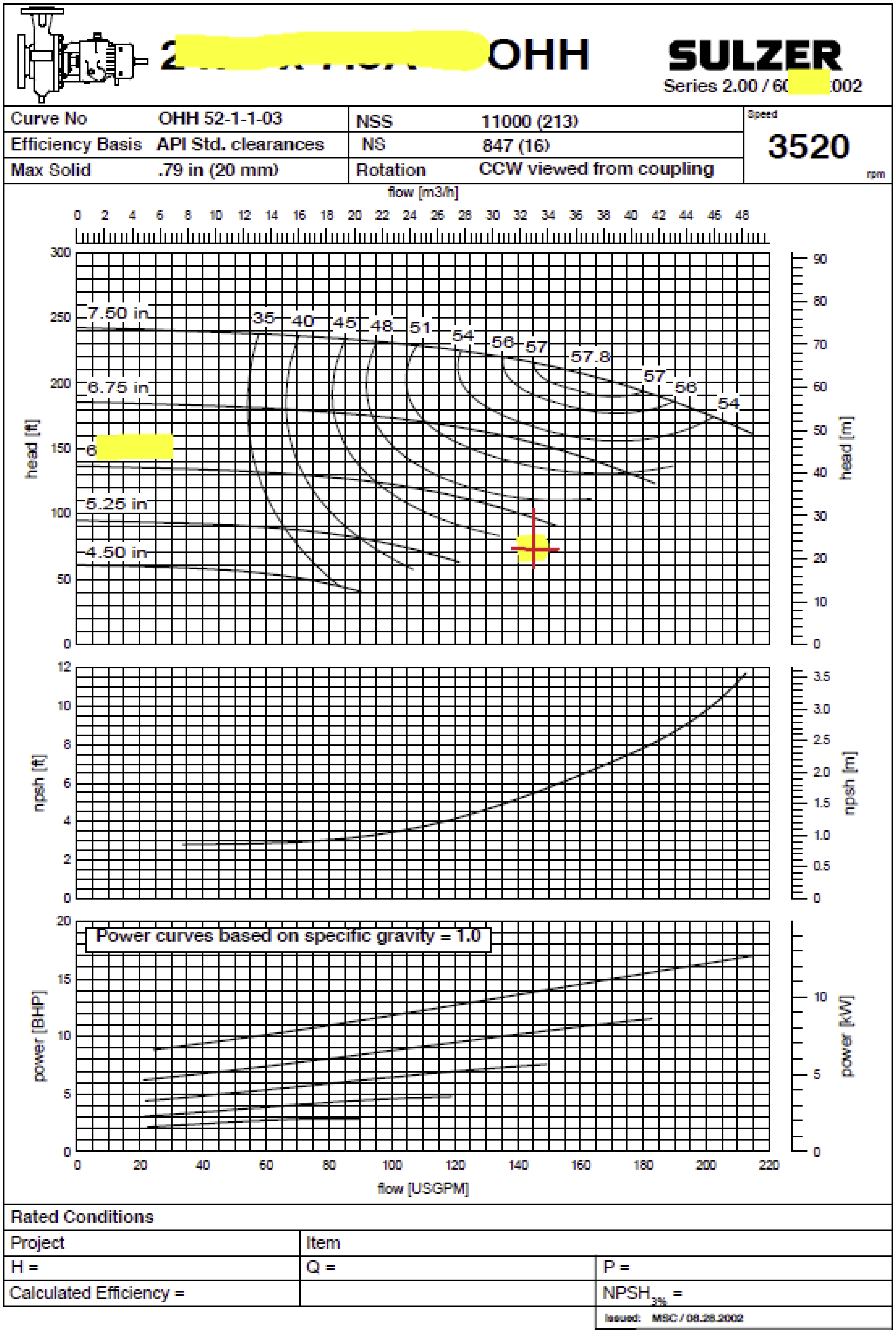


Figure 37: Sulzer 1x2x7.5 Pump

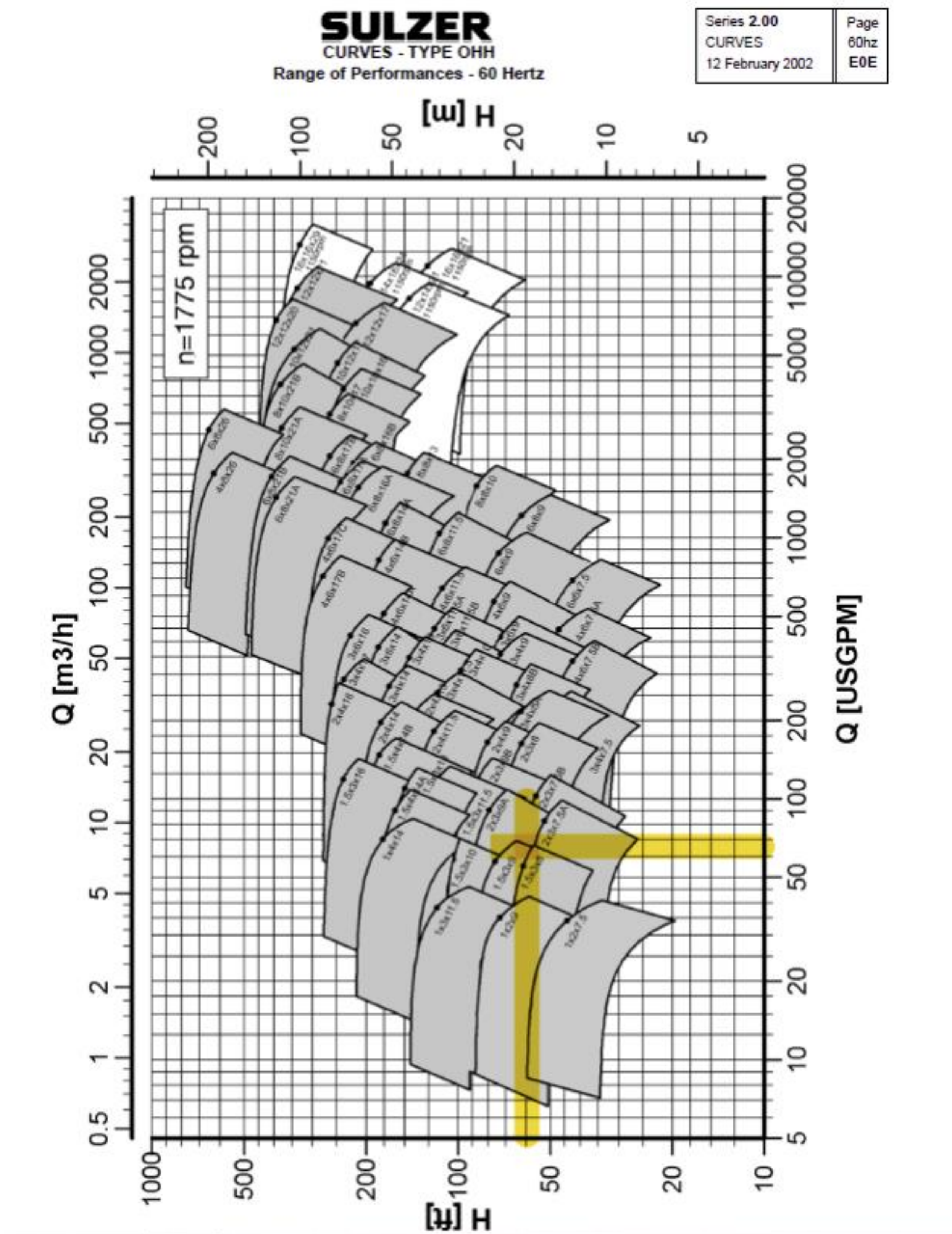


Figure 38: Pump Impeller Design

Sources:

Mott, R., Untener, J., “Applied Fluid Mechanics,” 7th Edition. Pearson Education, Inc., (2015)

Design Considerations:

Isothermal process

Steady state

Data and Variables:

Drawings shown in the figures above.

Procedure:

Using the predetermined fluid characteristics, various tables and charts from the Sulzer manual were used to adequately size the pumps.

Calculations:

Calculations from previous tasks were used to determine the Q and H values. The supplied Sulzer manual was used to determine the pump information needed.

Summary:

Based on the charts from the Sulzer manual, we are using three, 2X3X7.5 pumps with a 6-inch impeller on each. Once again, using the supplied manual, a 2 HP motor with a NEMA frame size of 324T-405T is required to power the pumps.

Materials:

Sulzer Pumps

Coolant

Electric Motor

Analysis:

When calculating the pumps and motors, manuals can be of great assistance if they can be interpreted correctly. While a 2 HP motor was sized for the pump, they may not be as readily available as say a 5 HP motor which would certainly work for the application providing the frame size matches or adequate adaptor rails are constructed to allow them to marry up.

Instrumentation Selection

Flow Nozzle with Manometer (task 14)
Purpose:

To perform calculations for a flow nozzle with a manometer in piping system #2. The flow nozzle will have a nozzle diameter to pipe diameter ratio of 0.5. The manometer will be used to measure the pressure drop across the nozzle and the manometric fluid will be mercury. The device will be installed near ground level on the vertical stretch of pipe that supplies the 1,100-gallon tank, making it easy to read from the ground.

Drawings and Diagrams:

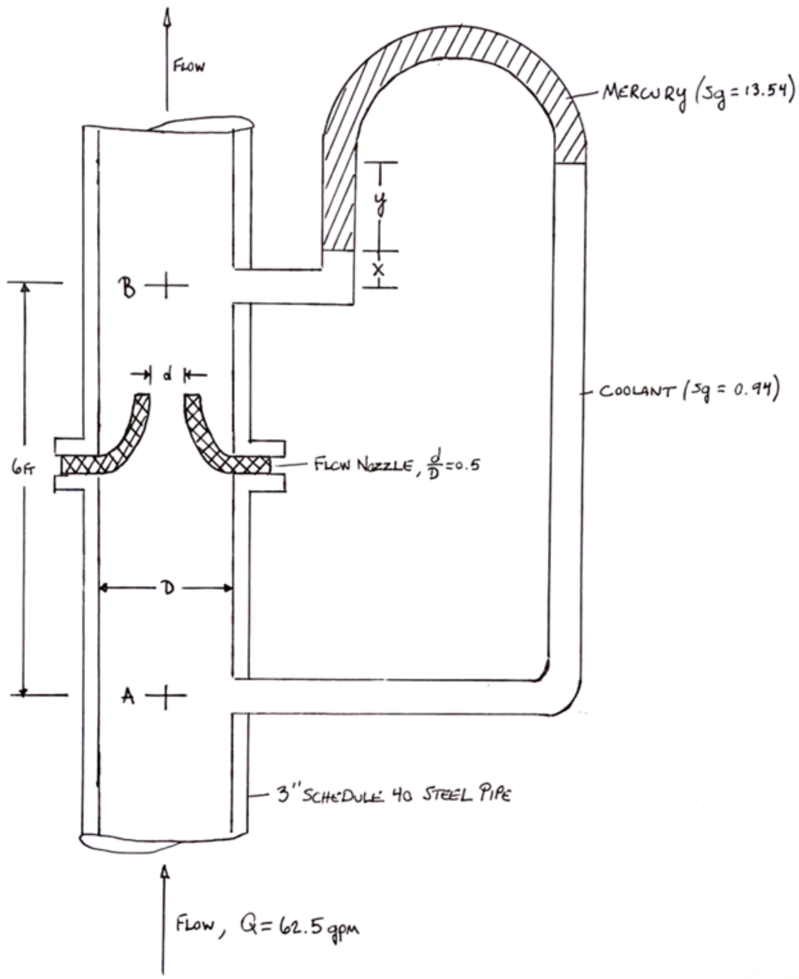


Figure VI-39: Flow Nozzle with Manometer

Sources:

Mott, R., Untener, J., "Applied Fluid Mechanics," 7th Edition. Pearson Education, Inc., (2015)

Design Considerations:

3" SCH 40 Steel Pipe

Incompressible fluids

T = 60F

Data and Variables:

Gravity, g = $32.2\ ft/s^2$

Specific Gravity (coolant), SG_{COOL} = 0.94

Specific Gravity (Mercury), SG_{HG} = 13.54

Specific Weight (water), $\gamma_W = 62.4\ lb/ft^3$

Flow Rate, Q = 62.5 GPM

Flow Area of Pipe, $A_1 = 0.05132\ ft^2$

Inside Pipe Diameter, D = 3.068 in

Kinematic Viscosity, $v = 1.21 \times 10^{-5} ft^2/s$

d/D = 0.5

Procedure:

First, the change in pressure across the flow nozzle must be computed by rearranging the equation for flow rate through a flow meter. To do this, the diameter for the flow nozzle needs to be determined. Also, Reynold's number for the larger pipe needs to be found. The Reynold's number will be used to find the discharge coefficient, C. Using this data along with other variables, the pressure drop across the flow nozzle will be found. The calculated pressure drop across the flow nozzle is used to determine the deflection range of the manometer as well as the minimum length of the manometer. This is found by applying the manometric equation to the proposed manometer.

Calculations:

Pressure drop across the flow nozzle:

$$Q = 62.5\ GPM \left(\frac{1\frac{ft^3}{s}}{449\ GPM}\right) = 0.139198 \frac{ft^3}{s}$$

$$A_1 = 0.05132\ ft^2$$

$$D_1 = 3.068\ in \left(\frac{1ft}{12in}\right) = 0.255667\ ft$$

$$V = \frac{Q}{A} \rightarrow \frac{0.139198\frac{ft^3}{s}}{0.05132\ ft^2} = 2.71235 \frac{ft}{s}$$

$$\gamma_C = (0.94)\left(62.4 \frac{lb}{ft^3}\right) = 58.656$$

$R_e = \frac{VD}{v} \rightarrow \frac{\left(2.71235\frac{ft}{s}\right)(0.2255667\ ft)}{1.21\times10^{-5}} \rightarrow R_e = 57310.6$

Equation for Flow Meter, $Q = CA_1\sqrt{\frac{2g\left(\frac{\Delta P}{\gamma}\right)}{\left(\frac{A_1}{A_2}\right)^2-1}}$

$\rightarrow \Delta P = \frac{\left(\frac{Q}{CA_1}\right)^2\left[\left(\frac{A_1}{A_2}\right)^2-1\right](\gamma_C)}{2g}$

Since d/D = 0.5, $\frac{A_1}{A_2} = \left(\frac{D}{d}\right)^2$ → the flow nozzle diameter will be half of the inside diameter of pipe

Therefore, $d = \frac{D_1}{2} = \frac{0.255667\ ft}{2} = 0.127833\ ft$

$\frac{A_1}{A_2} = \left(\frac{D}{d}\right)^2 = \left(\frac{0.255667\ ft}{0.127833\ ft}\right)^2 = 4$

Solving for pressure drop across the flow nozzle,

$$\Delta P = \frac{\left(\frac{0.139198\frac{ft^3}{s}}{0.975(0.05132\ ft^2}\right)^2[(4)^2-1]\left(58.656\frac{lb}{ft^3}\right)}{2\left(32.2\frac{ft}{s^2}\right)} = 105.731\ lb/ft^2\left(\frac{1ft^2}{144\ in^2}\right)$$

$\boxed{\Delta P = 0.7342\ psi}$

<u>Manometer deflection and length:</u>

$P_B - \gamma_c \cdot x - \gamma_{HG} \cdot y + \gamma_C(y + x + 6\ ft) = P_A$

$\rightarrow P_A - P_B = -\gamma_{HG} \cdot y + \gamma_C \cdot y + \gamma_C \cdot 6\ ft \qquad (\gamma_{HG} = SG_{HG} \cdot \gamma_W)$

$\rightarrow \Delta P = (\gamma_C - SG_{HG} \cdot \gamma_W)y + \gamma_C \cdot 6\ ft$

$$\rightarrow y = \frac{\Delta P - \gamma_C \cdot 6\ ft}{\gamma_C - (SG_{HG} \cdot \gamma_W)} = \frac{105.731\frac{lb}{ft^2} - \left(58.656\frac{lb}{ft^3} \times 6\ ft\right)}{58.656\frac{lb}{ft^3} - \left(13.54 \times 62.4\frac{lb}{ft^3}\right)}$$

$y = 0.3131\ ft + 6\ ft$

$\boxed{y = 6.3131\ ft}$

Summary:

The distance "x" was cancelled in the manometric equation. The deflection of the manometer for the calculated pressure drop is 0.3131 feet. The vertical section on the left-hand side of the manometer should be at least 0.3131 feet long while the vertical section on the right-hand side of the manometer should be at least 6.3131 feet long.

Materials:

Mercury (manometric fluid)

Coolant

3” SCH 40 Steel Pipe

Analysis:

The flow nozzle diameter will be half of the inside pipe diameter. This could potentially be adjusted if the client would like the manometer to yield a larger deflection, or a different manometric fluid could be used to produce the same outcome. For this application, Mercury was chosen as the manometric fluid because it has a much higher specific gravity that other liquids. In the calculation for the manometer deflection and length, the distance “x” was found to be negligible by simplification of the manometric equation, meaning that it will not have any effect on the deflection of the manometer for the given pressure difference. If the overall pressure difference was large enough, paired with a different manometric fluid being used, a reassessment would be required to ensure that the left side of the manometer would have enough length to prevent the gage fluid from entering the downstream sensing line.

6. Final Drawings

I. Plot Plan (task 19)

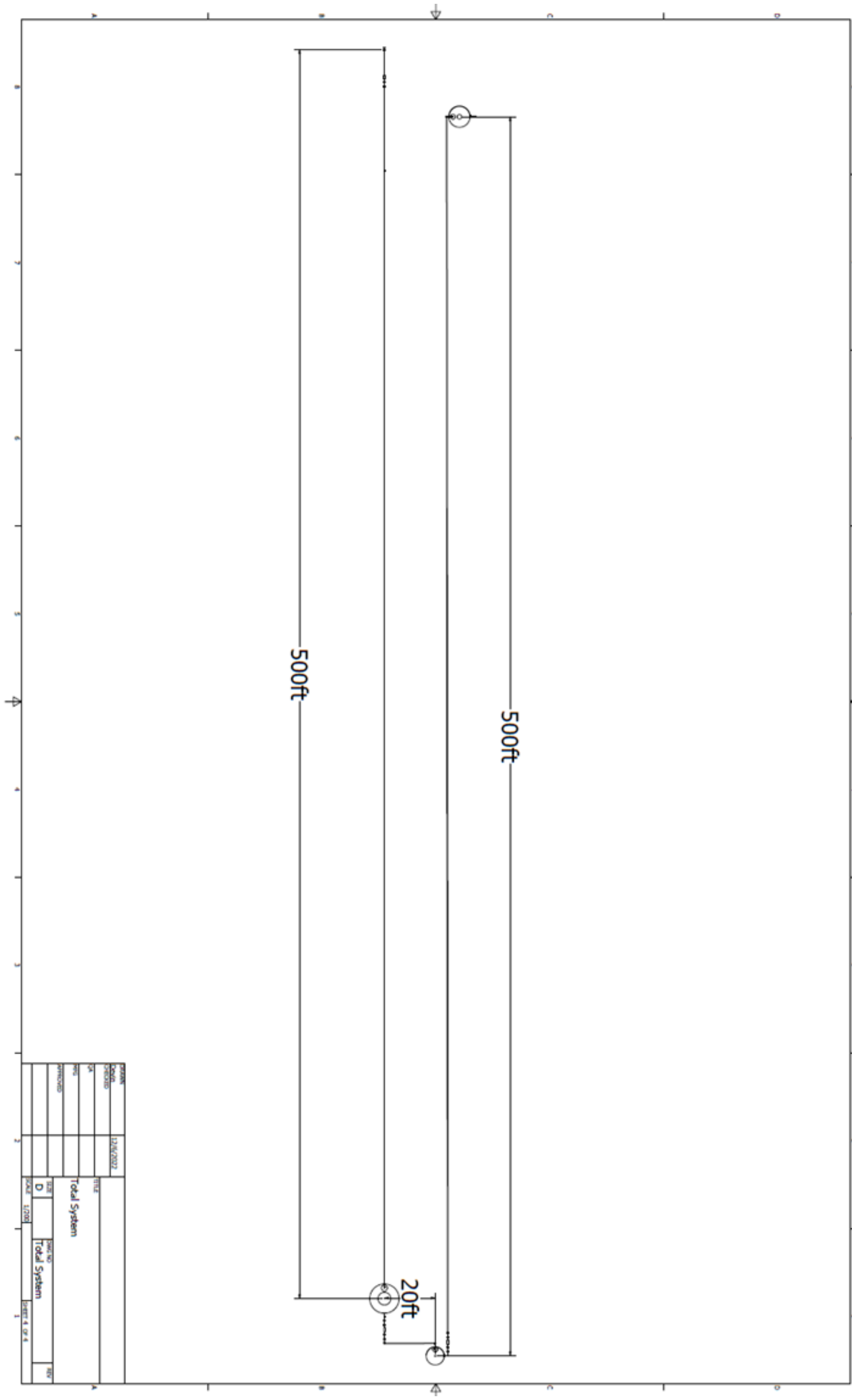


Figure I-1: Plot Plan

II. Elevation View (task 19)

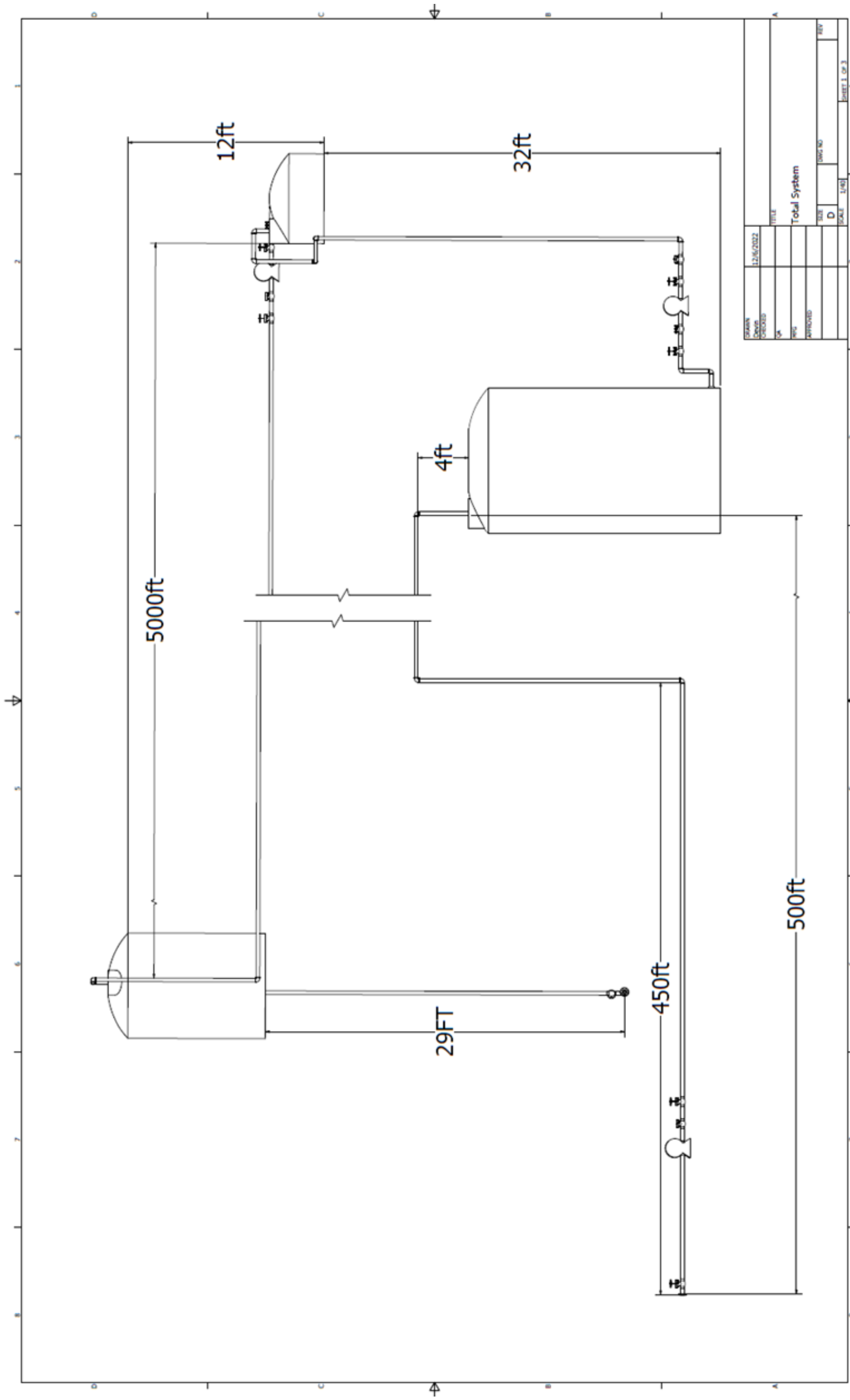


Figure II-1: Elevation View

III. Isometrics (task 19)

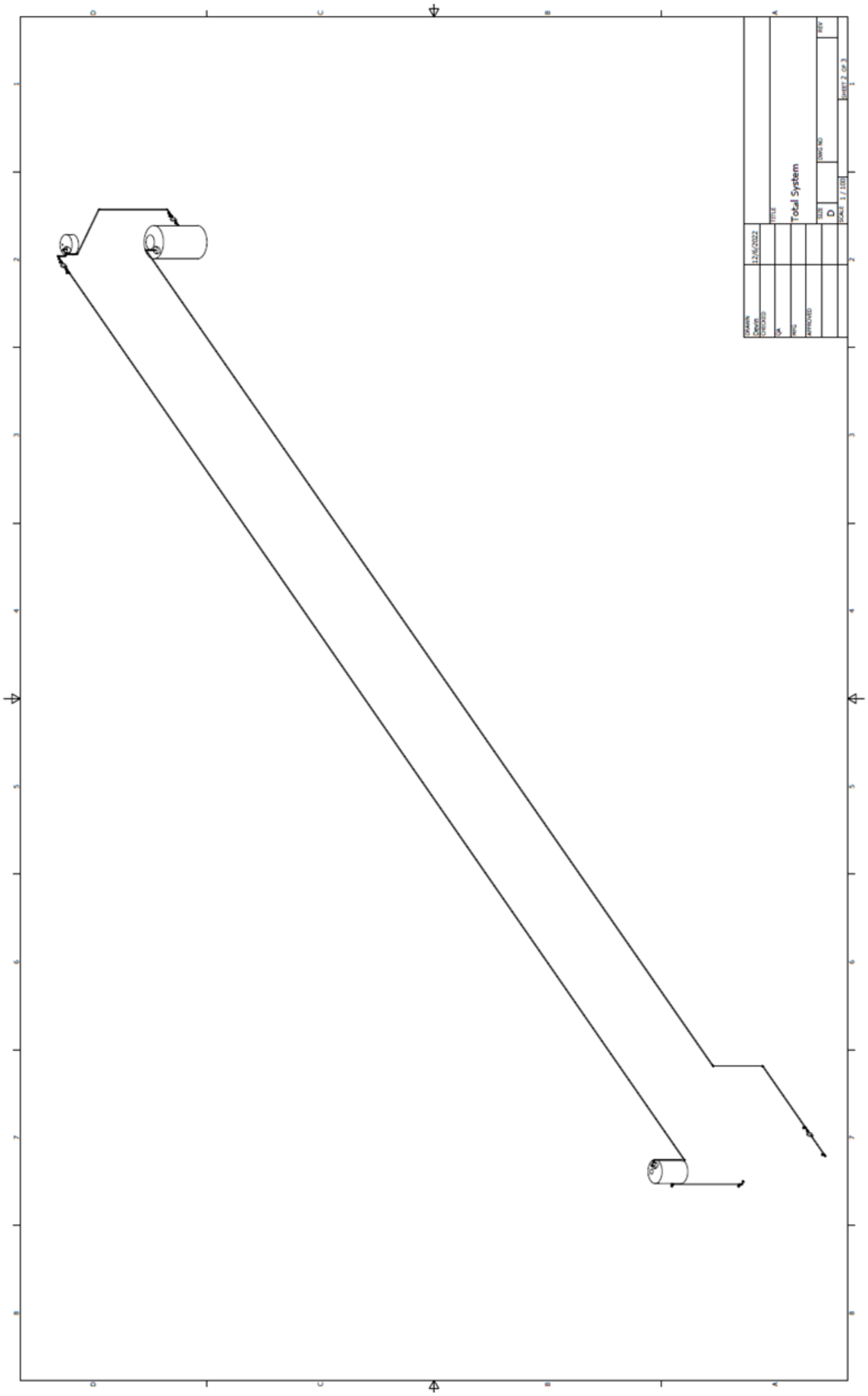


Figure III-1: Isometric View (A)

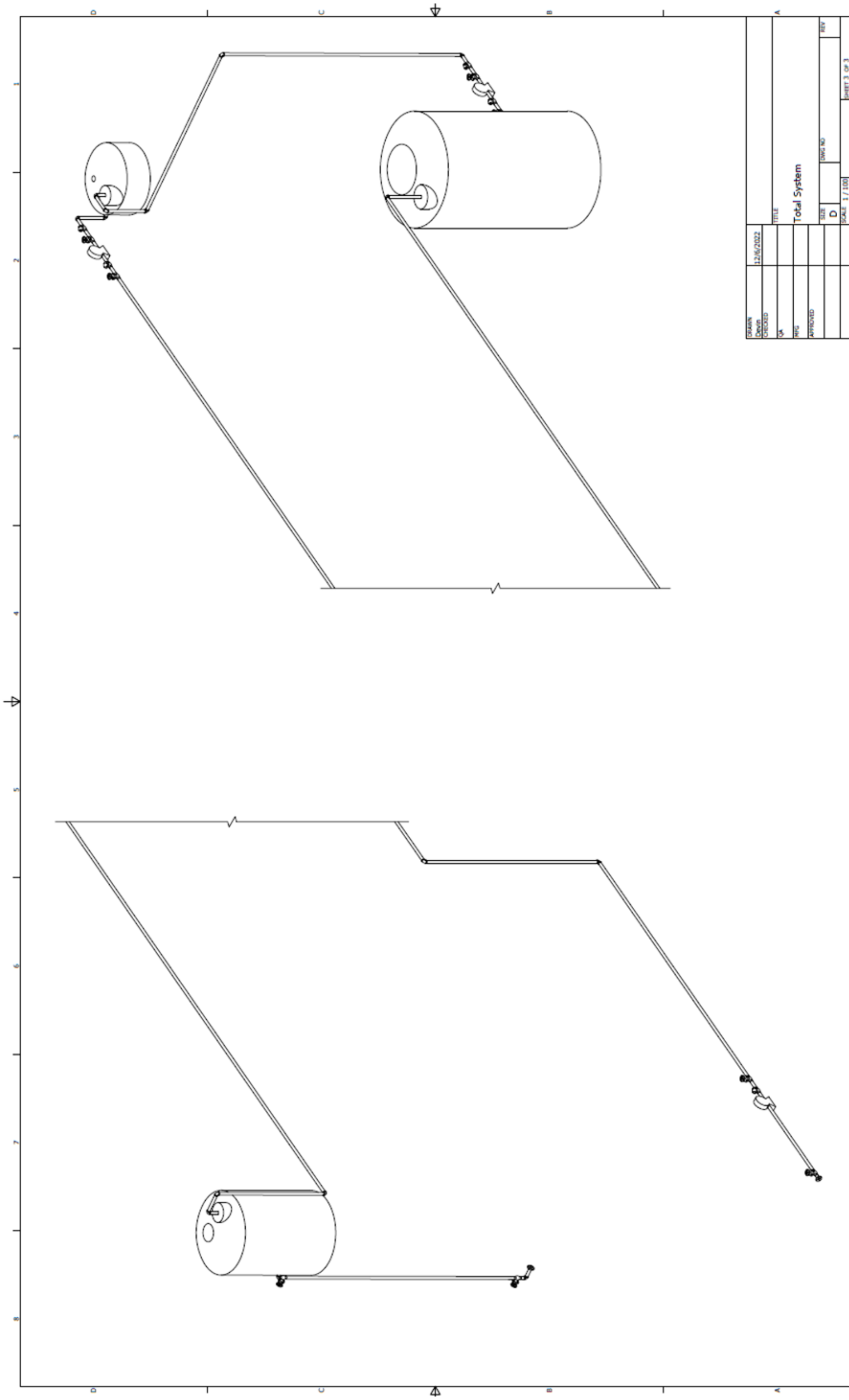


Figure III-2: Isometric View (B)

7. Bill of Materials and Equipment List (Task 20)

Bill of Materials		
Item	**Quantity**	
3-in Schedule 40 Steel Pipe	**1196**	ft
3-in 90° Elbow; Steel	**16**	each
3-in Check Valve, Swing Type; Steel	**5**	each
3-in Globe Valve; Steel	**8**	each
3-in Hose Connections; Steel	**2**	each
3-in Rounded Inlet; Steel	**2**	each
Norwesco 15,500-gal Poly Resin Tank	**1**	each
Norwesco 5,000-gal Poly Resin Tank	**1**	each
Norwesco 1,100-gal Poly Resin Tank	**1**	each
Manometer	**1**	each
Flow Nozzle	**1**	each
2x3x7.5 Pump	**3**	each
2 HP Motor	**3**	each

Table 11: Bill of Materials

8. Final Remarks (Task 21)

This engineering design project is extremely detailed and contains nitty-gritty information. Numerous inputs, calculations, iterations, and derivations were used to compute the piping, fittings, fluid properties, and materials used for the design of the cooling system. The system is designed to offload coolant from a railroad tanker car to a 15,500-gallon vertical storage tank. From there, a pump and piping system is designed to deliver coolant to a 1,100-gallon tank that will feed five machines inside the manufacturing facility's machining area. A separate, 5,000-gallon holding tank was designed to store any waste or used oil that will be picked up via truck. All three tanks and the piping system were designed to be as economically efficient as reasonably possible. In addition, a part of the design was to use a minimum amount of real estate when designing the layout and taking into consideration the potential for additional projects down the road. To reduce operational downtown, parts were selected from a pre-manufactured supplier catalog where applicable. This will instill some sense of conformity throughout the plant and will make things such as maintenance an easier task to handle, while maintaining a relatively low cost. All pumps in the design were chose to provide the system with the designed flow rates and an added expansion factor was considered if there ever arose a need for additional tanks or systems in the future.

9. APPENDIX

Reflections:

Giacomo's Reflection:

This project came off as extremely intimidating at first. Albeit it was rather intimidating throughout as well. That being said, this project was responsible for teaching me tremendously more about the principles of fluid dynamics than I originally anticipated. This project came off as a great deal of "busy work" but ended up having a great deal to do with the course, and MET degree, and many future career paths for students. I found myself taking many key principles I use in my professional career and applying them to the project. The material learned through this course and through this project is definitely important to professional career development in the sense that I already work as an engineer. The material and methodologies learned through this course will most certainly be applied to my day-to-day life at the office. I think a strength of mine that conveyed to the group was my particularness and my organization. On the flip side, I believe this was also my weakness because I like to spend countless hours trying to perfect things that only I will notice. In addition, I believe my time management sometimes negatively impacted my group. We are a group comprised of all full-time adults with full-time professions. Sometimes life gets in the way and it is very easy to fall behind in this course. I think our project is very strong in a technical sense due to the fact that a majority of our group has real-world experience with similar situations to the one of that proposed in this design project. On the other hand, I think that played to our detriment in certain areas where we would get caught up in the weeds about an issue that applies to a real-world scenario and not to this particular in-class assignment. If I were to retake this course, I would certainly try to manage my time more efficiently, where applicable, and attempt to get as far ahead as possible. I would also potentially take a lighter courseload knowing the demand of this class and the struggle to balance it with others, working, and a healthy lifestyle.

Austin's Reflection:

Devin's Reflection:

Erik's Reflection:

To be completely honest, I'm collecting my military retirement pay and now I'm almost a decade into my second career and do not intend to ever change employers before finally retiring. However, stuff happens, right? If I were to enter into a job interview this class would probably be at the bottom of my portfolio. I would describe the amount of teamwork involved and group problem-solving that, I feel, was the most beneficial for the team while in this class. My strengths would probably be team organization, advising the overall of the project, and the computer-aided design portion. After all, this is my second go-around with the project and I was aware of the final outcome. My weaknesses, as they were also in the Summer semester, would be clear communication of expectations with Dr. Ayala, as well as the iterations of calculations - but I'm much better with excel this time which made a definite difference. If I were starting this course over again I would first make certain that I was not enrolled in any other courses as was my problem last semester, as well as that I would have to do better with relying more on my wife to handle some things around here. There simply was not enough time in my day to complete everything to my usual standards and unfortunately my time set aside for college falls nearly to the bottom of priorities in life..

Useful Material from Text:

TABLE A.2 U.S. Customary System units (14.7 psia)

Temperature (°F)	Specific Weight γ (lb/ft^3)	Density ρ ($slugs/ft^3$)	Dynamic Viscosity η ($lb\text{-}s/ft^2$)	Kinematic Viscosity ν (ft^2/s)
32	62.4	1.94	3.66×10^{-5}	1.89×10^{-5}
40	62.4	1.94	3.23×10^{-5}	1.67×10^{-5}
50	62.4	1.94	2.72×10^{-5}	1.40×10^{-5}
60	62.4	1.94	2.35×10^{-5}	1.21×10^{-5}
70	62.3	1.94	2.04×10^{-5}	1.05×10^{-5}
80	62.2	1.93	1.77×10^{-5}	9.15×10^{-6}
90	62.1	1.93	1.60×10^{-5}	8.29×10^{-6}
100	62.0	1.93	1.42×10^{-5}	7.37×10^{-6}
110	61.9	1.92	1.26×10^{-5}	6.55×10^{-6}
120	61.7	1.92	1.14×10^{-5}	5.94×10^{-6}
130	61.5	1.91	1.05×10^{-5}	5.49×10^{-6}
140	61.4	1.91	9.60×10^{-6}	5.03×10^{-6}
150	61.2	1.90	8.90×10^{-6}	4.68×10^{-6}
160	61.0	1.90	8.30×10^{-6}	4.38×10^{-6}
170	60.8	1.89	7.70×10^{-6}	4.07×10^{-6}
180	60.6	1.88	7.23×10^{-6}	3.84×10^{-6}
190	60.4	1.88	6.80×10^{-6}	3.62×10^{-6}
200	60.1	1.87	6.25×10^{-6}	3.35×10^{-6}
212	59.8	1.86	5.89×10^{-6}	3.17×10^{-6}

Figure 3: Properties of Water

Table F.1 Schedule 40

Nominal Pipe Size		Outside Diameter		Wall Thickness		Inside Diameter			Flow Area	
NPS (in)	DN (mm)	(in)	(mm)	(in)	(mm)	(in)	(ft)	(mm)	(ft^2)	(m^2)
⅛	6	0.405	10.3	0.068	1.73	0.269	0.0224	6.8	0.000 394	3.660×10^{-5}
¼	8	0.540	13.7	0.088	2.24	0.364	0.0303	9.2	0.000 723	6.717×10^{-5}
⅜	10	0.675	17.1	0.091	2.31	0.493	0.0411	12.5	0.001 33	1.236×10^{-4}
½	15	0.840	21.3	0.109	2.77	0.622	0.0518	15.8	0.002 11	1.960×10^{-4}
¾	20	1.050	26.7	0.113	2.87	0.824	0.0687	20.9	0.003 70	3.437×10^{-4}
1	25	1.315	33.4	0.133	3.38	1.049	0.0874	26.6	0.006 00	5.574×10^{-4}
1¼	32	1.660	42.2	0.140	3.56	1.380	0.1150	35.1	0.010 39	9.653×10^{-4}
1½	40	1.900	48.3	0.145	3.68	1.610	0.1342	40.9	0.014 14	1.314×10^{-3}
2	50	2.375	60.3	0.154	3.91	2.067	0.1723	52.5	0.023 33	2.168×10^{-3}
2½	65	2.875	73.0	0.203	5.16	2.469	0.2058	62.7	0.033 26	3.090×10^{-3}
3	80	3.500	88.9	0.216	5.49	3.068	0.2557	77.9	0.051 32	4.768×10^{-3}
3½	90	4.000	101.6	0.226	5.74	3.548	0.2957	90.1	0.068 68	6.381×10^{-3}
4	100	4.500	114.3	0.237	6.02	4.026	0.3355	102.3	0.088 40	8.213×10^{-3}
5	125	5.563	141.3	0.258	6.55	5.047	0.4206	128.2	0.139 0	1.291×10^{-2}
6	150	6.625	168.3	0.280	7.11	6.065	0.5054	154.1	0.200 6	1.864×10^{-2}
8	200	8.625	219.1	0.322	8.18	7.981	0.6651	202.7	0.347 2	3.226×10^{-2}
10	250	10.750	273.1	0.365	9.27	10.020	0.8350	254.5	0.547 9	5.090×10^{-2}
12	300	12.750	323.9	0.406	10.31	11.938	0.9948	303.2	0.777 1	7.219×10^{-2}
14	350	14.000	355.6	0.437	11.10	13.126	1.094	333.4	0.939 6	8.729×10^{-2}
16	400	16.000	406.4	0.500	12.70	15.000	1.250	381.0	1.227	0.1140
18	450	18.000	457.2	0.562	14.27	16.876	1.406	428.7	1.553	0.1443
20	500	20.000	508.0	0.593	15.06	18.814	1.568	477.9	1.931	0.1794
24	600	24.000	609.6	0.687	17.45	22.626	1.886	574.7	2.792	0.2594

Figure 4: Dimensions of Schedule 40 Pipe

Table 10.4 Resistance in valves and fittings expressed as equivalent length in pipe diameters, L_e/D

Type	Equivalent Length in Pipe Diameters L_e/D
Globe valve—fully open	340
Angle valve—fully open	150
Gate valve—fully open	8
—¾ open	35
—½ open	160
—¼ open	900
Check valve—swing type	100
Check valve—ball type	150
Butterfly valve—fully open, 2–8 in	45
—10–14 in	35
—16–24 in	25
Foot valve—poppet disc type	420
Foot valve—hinged disc type	75
90° standard elbow	30
90° long radius elbow	20
90° street elbow	50
45° standard elbow	16
45° street elbow	26
Close return bend	50
Standard tee—with flow through run	20
—with flow through branch	60
(Reprinted with permission from "Flow of Fluids Through Valves, Fittings and Pipe, Technical Paper 410" 2011. Crane Co. All Rights Reserved.)	

Figure 5: Equivalent Length for Pipe Fittings

Miscellaneous Hand Calculations:

Wind Load and Weight of Tanks:

WEIGHT OF STORAGE TANKS

15,500 gal TANK: FROM NORWESCO TANK SPECIFICATIONS: $w_{T_1} = 3307$ lbs

$$V_c = 15,500 \text{ GAL} \cdot \left(\frac{1 m^3}{264.2 \text{ GAL}}\right) = 58.6677 m^3$$

$$\rho_c = sg_c \cdot \rho_w \rightarrow 0.94 \left(1000 \frac{kg}{m^3}\right) \rightarrow \rho_c = 940 \frac{kg}{m^3}$$

$$m_c = \rho_c \cdot V_c \rightarrow 940 \frac{kg}{m^3} (58.6677 m^3) \rightarrow m_c = 55147.6 \text{ Kg}$$

$$w_c = m_c \cdot g \rightarrow 55147.6 \text{ Kg} \left(9.81 \frac{m}{s^2}\right) \rightarrow w_c = 540998 \text{ N}$$

$$w_c = 540998 \text{N} \cdot \left(\frac{1 \text{ lbf}}{4.448 \text{N}}\right) \rightarrow w_c = 121627 \text{ lbs}$$

$$W(\text{TOTAL}) = w_{T_1} + w_c \rightarrow w = 3307 \text{ lbs} + 121627 \text{ lbs} = 124934 \text{ lbs}$$

THEREFORE, THE WEIGHT OF THE 15,500 gal STORAGE TANK AT MAXIMUM CAPACITY IS 124,934 lbs.

1,100 gal TANK: FROM NORWESCO TANK SPECIFICATIONS: $W_{T_2} = 169$ lbs

$$V_c = 1,100 \text{ gal} \cdot \left(\frac{1 m^3}{264.2 \text{ gal}}\right) = 4.1635 m^3$$

$$\rho_c = 940 \frac{kg}{m^3}$$

$$m_c = \rho_c \cdot V_c \rightarrow 940 \frac{kg}{m^3} (4.1635 m^3) \rightarrow m_c = 3913.69 \text{ Kg}$$

$$w_c = m_c \cdot g \rightarrow 3913.69 \text{ Kg} \left(9.81 \frac{m}{s^2}\right) \rightarrow w_c = 38393.3 \text{ N}$$

$$w_c = 38393.3 \text{ N} \cdot \left(\frac{1 \text{ lbf}}{4.448 \text{ N}}\right) \rightarrow w_c = 8631.59 \text{ lbs}$$

$$W(\text{TOTAL}) = W_{T_2} + w_c \rightarrow 169 \text{ lbs} + 8631.59 \text{ lbs} = 8800.59 \text{ lbs}$$

THEREFORE, THE WEIGHT OF THE 1,100 gal STORAGE TANK AT MAXIMUM CAPACITY IS 8800.59 lbs.

5,000 gal TANK: FROM NORWESCO TANK SPECIFICATIONS: $W_{T_3} = 844 \text{ lbs}$

$$V_c = 5,000 \text{ gal} \cdot \left(\frac{1 m^3}{264.2 \text{ gal}}\right) = 18.9251 m^3$$

$$\rho_c = 940 \frac{kg}{m^3}$$

$$M_c = \rho_c \cdot V_c \rightarrow 940 \frac{kg}{m^3}(18.9251 m^3) \rightarrow M_c = 17789.6 \text{ kg}$$

$$W_c = M_c \cdot g \rightarrow 17789.6 \text{ kg}\left(9.81 \frac{m}{s^2}\right) \rightarrow W_c = 174516 \text{ N}$$

$$W_c = 174516 \text{ N} \cdot \left(\frac{1 \text{ lbf}}{4.448 \text{ N}}\right) \rightarrow W_c = 39234.7 \text{ lbs}$$

$$W(\text{TOTAL}) = W_{T_3} + W_c \rightarrow W = 844 \text{ lbs} + 39234.7 \text{ lbs} = 40078.7 \text{ lbs}$$

THEREFORE, THE WEIGHT OF THE 5,000 gal STORAGE TANK AT MAXIMUM CAPACITY IS 40,078.7 lbs.

WIND LOAD ON STORAGE TANKS

15,500 gal TANK: $C_D = 1.12$ FOR SEMITUBULAR CYLINDERS (TABLE 17.1)

$$V_{AIR} = 12 \text{ mph} \cdot \left(\frac{5280 \text{ ft}}{1 \text{ mile}}\right) \cdot \left(\frac{1 h}{3600 s}\right) \cdot \left(\frac{1 m}{3.281 \text{ ft}}\right) = 5.364 \frac{m}{s}$$

$$A = BH \rightarrow H = 244 \text{ in} \cdot \left(\frac{1 \text{ ft}}{12 \text{ in}}\right) \cdot \left(\frac{1 m}{3.281 \text{ ft}}\right) = 6.1973 m$$

$$B = 141 \text{ in} \cdot \left(\frac{1 \text{ ft}}{12 \text{ in}}\right)\left(\frac{1 m}{3.281 \text{ ft}}\right) = 3.58123 m$$

$$\rightarrow A = (3.58123 m)(6.1973 m) = 22.194 m^2$$

SINCE THERE IS NO GIVEN VALUE FOR DENSITY OF AIR AT −20°F, I WILL USE INTERPOLATION METHOD TO GET AN ACCURATE VALUE FOR DENSITY.

$$-20°F \rightarrow °C, \quad (-20°F - 32)\left(\frac{5}{9}\right) = -28.8889°C$$

USING TABLE E.1 VALUES ABOVE AND BELOW MY TEMPERATURE,

$X = -28.8889°C$ $\quad y = \rho_{AIR}$ @ $-28.8889°C$

$X_1 = -30°C$ $\quad y_1 = 1.452 \frac{kg}{m^3}$

$X_2 = -20°C$ $\quad y_2 = 1.394 \frac{kg}{m^3}$

$$y = y_1 + \left[\left(\frac{x - x_1}{x_2 - x_1}\right)(y_2 - y_1)\right] \rightarrow y = 1.452 \frac{kg}{m^3}\left[\left(\frac{-28.8889 - (-30)}{-20 - (-30)}\right)(1.394 - 1.452)\right]$$

$$y = 1.4456 \frac{kg}{m^3}, \quad \rho_{AIR} = 1.4456 \frac{kg}{m^3} \text{ @ } -28.8889°C \text{ (or } -20°F)$$

NOW THAT I HAVE ALL VARIABLES, I CAN CALCULATE THE DRAG FORCE ON THE TANK,

$$F_D = C_D \frac{\rho_{AIR} V^2}{2} \cdot A \rightarrow (1.12)\left(\frac{1.4456 \frac{kg}{m^3}\left(5.364 \frac{m}{s}\right)^2}{2}\right)(22.194 m^2)$$

$$F_D = 516.951 \text{ N} \cdot \left(\frac{1 \text{ lbf}}{4.448 \text{ N}}\right) = 116.221 \text{ lbs}$$

THEREFORE, THE WIND LOAD ON THE 15,500 gal STORAGE TANK FOR GIVEN CONDITIONS IS EQUIVALENT TO 116.221 lbs.

1,100 gal TANK: $C_D = 1.12$

$V_{AIR} = 5.364 \frac{m}{s}$

$A = BH \longrightarrow H = 53\,in \cdot \left(\frac{1\,ft}{12\,in}\right)\cdot\left(\frac{1\,m}{3.281\,ft}\right) = 1.34613\,m$

$B = 87\,in \cdot \left(\frac{1\,ft}{12\,in}\right)\cdot\left(\frac{1\,m}{3.281\,ft}\right) = 2.20969\,m$

$A = (2.20969\,m)(1.34613\,m) = 2.97453\,m^2$

$\rho_{AIR} = 1.4456 \frac{kg}{m^3}$

$F_D = C_D \frac{\rho_{AIR} V^2}{2} \cdot A \longrightarrow (1.12)\left(\frac{1.4456 \frac{kg}{m^3}(5.364 \frac{m}{s})^2}{2}\right)(2.97453\,m^2)$

$F_D = 69.2839\,N \cdot \left(\frac{1\,lbf}{4.448\,N}\right) = 15.5764\,lbs$

THEREFORE, THE WIND LOAD ON THE 1,100 gal TANK FOR GIVEN CONDITIONS IS ~~EQUIVALENT~~ TO 15.5764 lbs.

5,000 gal TANK: $C_D = 1.12$

$V_{AIR} = 5.364 \frac{m}{s}$

$A = BH \longrightarrow H = 152\,in \cdot \left(\frac{1\,ft}{12\,in}\right)\cdot\left(\frac{1\,m}{3.281\,ft}\right) = 3.86061\,m$

$B = 102\,in \cdot \left(\frac{1\,ft}{12\,in}\right)\cdot\left(\frac{1\,m}{3.281\,ft}\right) = 2.59067\,m$

$A = (2.59067\,m)(3.86061\,m) = 10.0016\,m^2$

$\rho_{AIR} = 1.4456 \frac{kg}{m^3}$

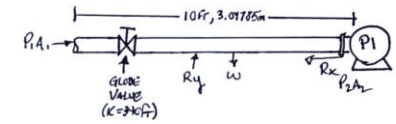
$F_D = C_D \frac{\rho_{AIR} V^2}{2} \cdot A \longrightarrow (1.12)\left(\frac{1.4456 \frac{kg}{m^3}(5.364 \frac{m}{s})^2}{2}\right)(10.0016\,m^2)$

$F_D = 232.961\,N \cdot \left(\frac{1\,lbf}{4.448\,N}\right) = 52.3743\,lbs$

THEREFORE, THE WIND LOAD ON THE 5,000 gal TANK FOR GIVEN CONDITIONS IS EQUIVALENT TO 52.3743 lbs.

Pipeline Support Information:

SECTION 1



$$\Sigma F = \rho Q(V_2 - V_1)$$

$$P_1A_1 - R_x = \rho Q(V_2 - V_1)$$

$$\frac{P_1}{\gamma_c} = h_{L_{1-2}} = f\frac{L}{D}\frac{V^2}{2g} + K_{gv}\frac{V^2}{2g}$$

$$= (0.018)\left(\frac{3.04785\,m}{0.0779\,m}\right)\left(\frac{0.826762^2\frac{m}{s}}{2(9.81\frac{m}{s^2})}\right) + (340 \times 0.018)\left(\frac{0.826762^2\frac{m}{s}}{2(9.81\frac{m}{s^2})}\right)$$

$$= 0.237748\,m$$

$$P_1 = h_{L_{1-2}} \cdot \gamma_c = (0.237748\,m)(9221.4\tfrac{N}{m^3})$$

$$= 2192.37\tfrac{N}{m^2}$$

$$R_x = P_1A_1 + \rho Q V_1 = (2192.37\tfrac{N}{m^2} \times 0.004768\,m^2) + (940\tfrac{kg}{m^3} \times 0.003942\tfrac{m^3}{s} \times 0.826762\tfrac{m}{s})$$

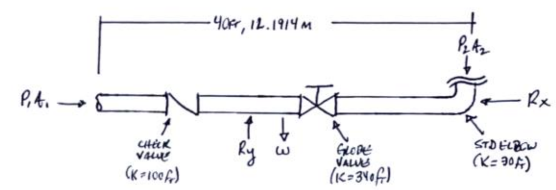
$$R_x = 13.5168\,N \cdot \frac{1\,lb}{4.448\,N} = 3.04\,lb$$

$$R_y = \omega - \rho Q V_2 = \gamma_c \cdot A \cdot L - \rho Q V_2$$

$$= (9221.4\tfrac{N}{m^3} \times 0.004768\,m^2 \times 3.04785\,m) - (940\tfrac{kg}{m^3} \times 0.003942\tfrac{m^3}{s} \times 0.826762\tfrac{m}{s})$$

$$R_y = 130.943\,N \cdot \frac{1\,lb}{4.448\,N} = 29.44\,lb$$

SECTION 2



$$P_1A_1 - R_x = \rho Q(V_2 - V_1)$$

$$\frac{P_1}{\gamma_c} = h_{L_{1-2}} = f\frac{L}{D}\frac{V^2}{2g} + K_{cv}\frac{V^2}{2g} + K_{gv}\frac{V^2}{2g} + K_e\frac{V^2}{2g} \qquad \frac{V^2}{2g} = \left(\frac{0.826762^2\frac{m}{s}}{2(9.81\frac{m}{s^2})}\right) = 0.034839\,m$$

$$= (0.018)\left(\frac{12.1914\,m}{0.0779\,m}\right)(0.034839\,m) + (100 \times 0.018)(0.034839\,m) + (340 \times 0.018)(0.034839\,m)$$
$$+ (30 \times 0.018)(0.034839\,m)$$

$$h_{L_{1-2}} = 1.27615\,m$$

$$P_1 = h_{L_{1-2}} \cdot \gamma_c = (1.27615\,m)(9221.4\tfrac{N}{m^3}) = 11767.9\tfrac{N}{m^2}$$

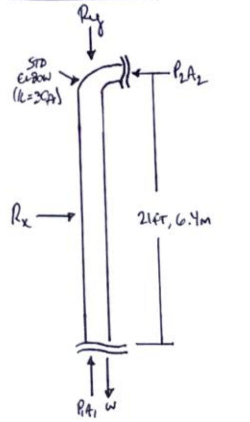
$$R_x = P_1A_1 + \rho Q V_1 = (11767.9\tfrac{N}{m^2} \times 0.004768\,m^2) + (940\tfrac{kg}{m^3} \times 0.003942\tfrac{m^3}{s} \times 0.826762\tfrac{m}{s})$$

$$R_x = 59.1729\,N = 13.3\,lb$$

$$R_y = \omega - \rho Q V_2 = (9221.4\tfrac{N}{m^3} \times 0.004768\,m^2 \times 12.1914\,m) - (940 \times 0.003942 \times 0.826762)$$

$$R_y = 532.963\,N = 119.821\,lb$$

SECTION 3



$$R_x - P_2A_2 = \rho Q(V_2 - V_1)$$

$$\frac{P_2}{\gamma_c} = h_{L_{1-2}} = f\frac{L}{D}\frac{V^2}{2g} + K_e\frac{V^2}{2g} + Z_2 \qquad \frac{V^2}{2g} = 0.034839\,m$$

$$= (0.018)\left(\frac{6.4\,m}{0.0779\,m}\right)(0.034839\,m) + (30\times 0.018)(0.034839\,m) + 6.4\,m$$

$$h_{L_{1-2}} = 6.47033\,m$$

$$P_2 = \gamma_c \cdot h_{L_{1-2}} = (9221.4\,\tfrac{N}{m^3})(6.47033\,m) = 59665.5\,\tfrac{N}{m^2}$$

$$R_x = P_2A_2 + \rho Q(V_2)$$

$$= (59665.5\,\tfrac{N}{m^2} \times 0.004768\,m^2) + (940\,\tfrac{kg}{m^3} \times 0.003942\,\tfrac{m^3}{s} \times 0.826762\,\tfrac{m}{s})$$

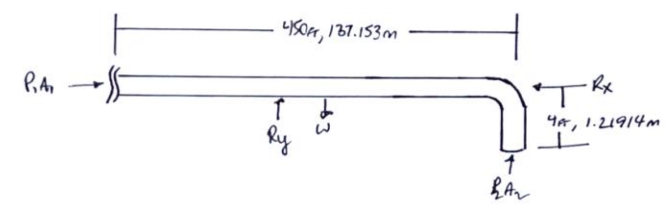
$$R_x = 287.549\,N = 64.646\,lb$$

$$R_y = P_1A_1 - W - \rho QV_1$$

$$= (59665.5\,\tfrac{N}{m^2} \times 0.004768\,m^2) - (9221.4\,\tfrac{N}{m^3} \times 0.004768\,m^2 \times 6.4\,m)$$
$$- (940\,\tfrac{kg}{m^3} \times 0.003942\,\tfrac{m^3}{s} \times 0.826762\,\tfrac{m}{s})$$

$$R_y = 0.028689\,N = 0.0065\,lb$$

SECTION 4



$$P_1A_1 - R_x = \rho Q(V_2 - V_1)$$

$$\frac{P_1}{\gamma_c} = h_{L_{1-2}} = f\frac{L}{D}\frac{V^2}{2g} + K_e\frac{V^2}{2g} \qquad \frac{V^2}{2g} = 0.034839\,m$$

$$= (0.018)\left(\frac{137.153\,m + 1.21914\,m}{0.0779\,m}\right)(0.034839\,m) + (30\times 0.018)(0.034839\,m)$$

$$= 1.13271\,m$$

$$P_1 = \gamma_c \cdot h_{L_{1-2}} = (9221.4\,\tfrac{N}{m^3})(1.13271\,m) = 10445.2\,\tfrac{N}{m^2}$$

$$R_x = P_1A_1 + \rho QV_1 = (10445.2\,\tfrac{N}{m^2} \times 0.004768\,m^2) + (940\,\tfrac{kg}{m^3} \times 0.003942\,\tfrac{m^3}{s} \times 0.826762\,\tfrac{m}{s})$$

$$R_x = 52.8663\,N = 11.885\,lb$$

$$R_y = W - \rho QV_2 = (9221.4\,\tfrac{N}{m^3} \times 0.004768\,m^2 \times 138.372\,m) - (940\,\tfrac{kg}{m^3} \times 0.003942\,\tfrac{m^3}{s} \times 0.826762\,\tfrac{m}{s})$$

$$R_y = 6080.83\,N = 1367.09\,lb$$

TOTAL FORCES OF SECTIONS (WEIGHT + FORCES)

SECTION 1: $R_y = 29.441 \text{ lbs}$; WEIGHT OF PIPE, $W_P = 10\text{FT}(7.58\frac{\text{lb}}{\text{FT}}) = 75.8 \text{ lbs}$

$W_T = R_y + W_P = 29.441 \text{ lbs} + 75.8 \text{ lbs} = 105.241 \text{ lbs}$

SECTION 2,3,4: $R_{y_T} = R_{y_2} + R_{y_3} + R_{y_4}$; WEIGHT OF PIPE, $W_{P_T} = W_{P_2} + W_{P_3} + W_{P_4}$

$= 119.821 + 0.0065 + 1367.09 \text{ lbs}$

$= 1486.92 \text{ lbs}$

$= (40\text{FT} + 21\text{FT} + 454\text{FT})(7.58\frac{\text{lb}}{\text{FT}})$

$= 3903.7 \text{ lbs}$

$W_T = R_{y_T} + W_{P_T} = 1486.92 \text{ lbs} + 3903.7 \text{ lbs} = 5390.62 \text{ lbs}$

DEFLECTION OF PIPING

DEFLECTION, $D = wL^4/384EI$

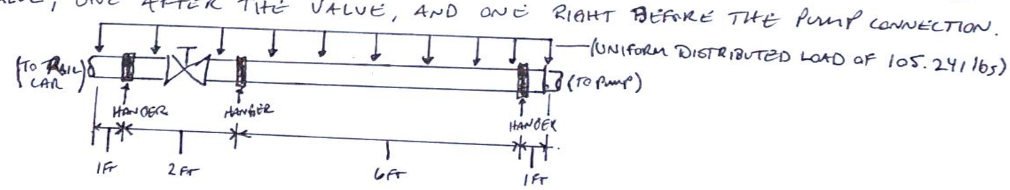
w = FORCE APPLIED PER UNIT LENGTH
L = LENGTH BETWEEN SUPPORTS
E = YOUNGS MODULUS
I = MOMENT OF INERTIA

FOR 3" NPS SCHEDULE 40, $I = \frac{\pi(D^4 - d^4)}{64} = \frac{\pi(3.5^4 - 3.068^4)}{64} = 3.017 \text{ in}^4$

$E = 29,500,000 \text{ psi}$

FOR SECTION 1, WE WILL USE FORMULA FOR FIXED SIMPLE BEAM UNDER UNIFORM DISTRIBUTED LOAD, THIS WILL GIVE US MAXIMUM DEFLECTION AT CENTER OF PIPE SECTION.

I WILL START BY PLACING 3 HANGERS ON THE PIPE, ONE BEFORE THE VALVE, ONE AFTER THE VALVE, AND ONE RIGHT BEFORE THE PUMP CONNECTION.



SINCE THE MOST DISTANCE BETWEEN HANGER SUPPORTS IS 6 FT, I WILL USE THIS FOR MY DISTANCE. $6\text{FT} = 72 \text{ in}$; $\frac{105.241 \text{ lbs}}{120 \text{ in}} = 0.87701 \frac{\text{lb}}{\text{in}} \times 72 \text{ in} = 63.1446 \text{ lb}$

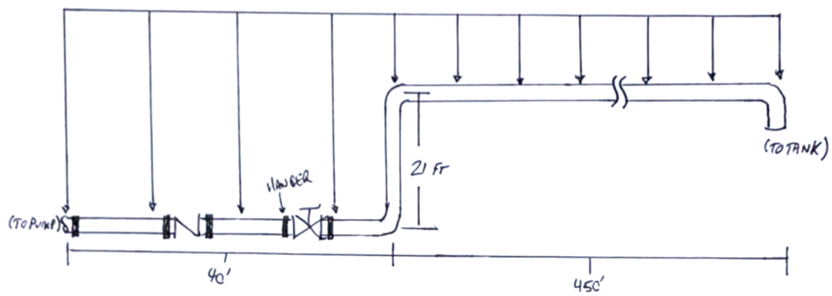
$$D = \frac{wL^4}{384EI} \rightarrow \frac{\left(\frac{63.1446 \text{ lb}}{72 \text{ in}}\right)(72 \text{ in})^4}{384(29,500,000\frac{\text{lb}}{\text{in}^2})(3.017 \text{ in}^4)} = 0.00069 \text{ in}$$

$$\% \text{ DEFLECTION} = \frac{0.00069 \text{ in}}{3.5 \text{ in}} = 0.000197 \times 100 = 0.02\% \text{ DEFLECTION}$$

THEREFORE, THE SUPPORT LOCATIONS ARE ADEQUATE.

FOR THE OTHER SECTIONS DOWNSTREAM OF THE PUMP, I WILL GUESS THE SUPPORTS TO BE 12 FEET APART (EXCEPT FOR NEAR COMPONENTS AND ELBOWS).

$12\text{ FT} = 144\text{ in}; \quad \frac{5390.62\text{ lbs}}{6180\text{ in}} = 0.872269\frac{\text{lbs}}{\text{in}} \times 144\text{ in} = 125.607\text{ lbs}$



$$D = \frac{wL^4}{384EI} \longrightarrow \frac{\left(\frac{125.607\text{ lbs}}{144\text{ in}}\right)(144\text{ in})^4}{384\left(29{,}500{,}000\frac{\text{lb}}{\text{in}^2}\right)(3.017\text{ in}^4)} = 0.010974\text{ in}$$

$$\%\text{ DEFLECTION} = \frac{0.010974\text{ in}}{3.5\text{ in}} = 0.003135 \times 100 = 0.314\%$$ MAX DEFLECTION BETWEEN HANGERS

THE SUPPORT HANGER LOCATIONS CAN BE SPACED 12 FT APART OTHER THAN AT VALVES, CONNECTIONS, AND ELBOWS AND WILL ADEQUATELY SUPPORT THE SYSTEM FORCES.

Energy Losses:

$f = 0.018$

— SYSTEM 1

$$h_{A_1} = (\cancel{h_{ent_s}} + 2h_{v_{gl}} + 3h_{elbow} + h_{v_{ch}} + h_{pipe_1} + \cancel{h_{exit_D}}) + Z_2 - Z_1$$

$$\cancel{h_{ent_s}} \quad h_{v_{globe}} = 340 \cdot \frac{v^2}{2g} = 0.716\ ft$$

$$h_{elbow} = 30 \cdot \frac{v^2}{2g} = 0.063\ ft \qquad \Delta Z_1 = 17\ ft$$

$$h_{v_{check}} = 100 \cdot \frac{v^2}{2g} = 0.210\ ft$$

$$h_{pipe_1} = f_T \cdot \frac{525\ ft}{0.256\ ft} = 0.018 \cdot \frac{525}{0.256} = 36.91$$

$$\cancel{h_{A_1} = (0.063\ ft + 0.210\ ft +}$$

$$h_{A_1} = 3(0.063\ ft) + 2(0.716\ ft) + (0.210\ ft) + (36.91) + 17\ ft$$

$$h_{A_1} = 55.745\ ft$$

— SYSTEM 2

$$h_{A_2} = (h_{ent_s} + 7h_{elbow} + 2h_{v_{ch}} + 2h_{v_{gl}} + h_{pipe_2}) + Z_2 - Z_1$$

$$h_{ent_s} = 0.24 \cdot \frac{v^2}{2g} = 0.051\ ft$$

$$h_{elbow} = 30 \cdot \frac{v^2}{2g} = 0.063\ ft \qquad \Delta Z_2 = 39\ ft$$

$$h_{v_{check}} = 100 \cdot \frac{v^2}{2g} = 0.210\ ft$$

$$h_{v_{globe}} = 340 \cdot \frac{v^2}{2g} = 0.716\ ft$$

$$h_{pipe_2} = f_T \cdot \frac{156\ ft}{0.256\ ft} = 0.018 \cdot \frac{156}{0.256} = 10.98\ ft$$

$$h_{A_2} = (0.051\ ft) + 7(0.063\ ft) + (0.210\ ft)2 + 2(0.716\ ft) + 10.98\ ft + 39\ ft$$

$$h_{A_2} = 52.33\ ft$$

— SYSTEM 3

$h_{A_3} = (h_{ent_s} + 5\,h_{elbow} + 2\,h_{v_{ch}} + 2\,h_{v_{gl}} + h_{pipe_3}) + Z_2 - Z_1$

same values as System 1 & 2 except for h_{pipe} & ΔZ_3

$h_{A_3} = (0.051\,ft) + 5(0.063\,ft) + 2(0.210\,ft) + 2(0.716\,ft) + 36.26\,ft + 12\,ft$

$h_{A_3} = 50.48\ ft$

Instrumentation Selection:

PRESSURE DROP ACROSS THE FLOW NOZZLE

$$Q = 62.5\ gpm \cdot \frac{1\ \frac{ft^3}{s}}{449\ gpm} = 0.139198\ \frac{ft^3}{s}$$

$$A_1 = 0.05132\ ft^2$$

$$D_1 = 3.068\ in \cdot \frac{1\ ft}{12\ in} = 0.255667\ ft$$

$$V = \frac{Q}{A} \rightarrow \frac{0.139198\ \frac{ft^3}{s}}{0.05132\ ft^2} = 2.71235\ \frac{ft}{s}$$

$$\gamma_c = (0.94)(62.4\ \tfrac{lb}{ft^3}) = 58.656$$

$$R_e = \frac{VD}{\nu} \rightarrow \frac{(2.71235\ \frac{ft}{s})(0.255667\ ft)}{1.21 \times 10^{-5}}$$

$$\rightarrow R_e = 57310.6$$

USING FIGURE 15.5, $C = 0.975$

EQUATION FOR FLOW METER, $Q = CA_1\sqrt{\frac{2g(\Delta P/\gamma)}{(A_1/A_2)^2 - 1}} \rightarrow \Delta P = \frac{\left(\frac{Q}{CA_1}\right)^2 \cdot \left[\left(\frac{A_1}{A_2}\right)^2 - 1\right] \cdot \gamma_c}{2g}$

SINCE $\frac{d}{D} = 0.5$, $\frac{A_1}{A_2} = \left(\frac{D}{d}\right)^2 \longrightarrow$ THE FLOW NOZZLE DIAMETER WILL BE HALF OF THE INSIDE DIAMETER OF PIPE. SO,

$$d = \frac{D_1}{2} = \frac{0.255667\ ft}{2} = 0.127833\ ft$$

THEREFORE, $\frac{A_1}{A_2} = \left(\frac{D}{d}\right)^2 = \left(\frac{0.255667\ ft}{0.127833\ ft}\right)^2 = 4$

SOLVING FOR PRESSURE DROP ACROSS THE FLOW NOZZLE,

$$\Delta P = \frac{\left(\frac{0.139198\ \frac{ft^3}{s}}{0.975(0.05132\ ft^2)}\right)^2 \cdot \left[(4)^2 - 1\right] \cdot (58.656\ \tfrac{lb}{ft^3})}{2(32.2\ ft/s^2)} = 105.731\ \frac{lb}{ft^2} \quad (\text{OR } 0.734\ psi)$$

MANOMETER DEFLECTION AND LENGTH

APPLYING THE MANOMETRIC EQUATION TO THE PROPOSED MANOMETER ABOVE,

$$P_B - \cancel{\gamma_c \cdot x} - \gamma_{Hg} \cdot y + \gamma_c(y + \cancel{x} + 6\ ft) = P_A$$

$$\rightarrow P_A - P_B = -\gamma_{Hg} \cdot y + \gamma_c \cdot y + \gamma_c \cdot 6\ ft \qquad (\gamma_{Hg} = sg_{Hg} \cdot \gamma_w)$$

$$\rightarrow \Delta P = (\gamma_c - sg_{Hg} \cdot \gamma_w)y + \gamma_c \cdot 6\ ft$$

$$\rightarrow y = \frac{\Delta P - \gamma_c \cdot 6\ ft}{\gamma_c - (sg_{Hg} \cdot \gamma_w)}$$

$$\rightarrow y = \frac{105.731\ \frac{lb}{ft^2} - (58.656\ \frac{lb}{ft^3} \cdot 6\ ft)}{58.656\ \frac{lb}{ft^3} - (13.54 \cdot 62.4\ \frac{lb}{ft^3})} = 0.3131\ ft \quad (\text{OR } 3.758\ in)$$

I WAS ABLE TO CANCEL THE DISTANCE "X" IN MY MANOMETRIC EQUATION, THEREFORE THAT MEANS IT WILL NOT HAVE ANY AFFECT ON THE DEFLECTION IN THE MANOMETER. THE DEFLECTION IN THE MANOMETER FOR THE CALCULATED PRESSURE DROP IS $y = 3.758$ in. THE VERTICAL SECTION ON THE LEFT-HAND SIDE OF THE MANOMETER SHOULD BE ATLEAST 3.758 inCHES LONG. THE VERTICAL SECTION ON THE RIGHT HAND SIDE OF THE MANOMETER SHOULD BE A MINIMUM OF 6 ft 3.758 in LONG.