MET 330 FLUID MECHANICS

FULL PIPELINE SYSTEM DESIGN OF A MANUFACTURING PLANT FOR CONTINENTAL AG

LE – PEAK - HOKE

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Abstract

Continental AG is in the planning phase for a new manufacturing facility in Dayton, Ohio. The plant will feature an automated machining line where five machines will be supplied coolant from a single reservoir tank. The project will cover the design of the entire coolant system, from reception of clean coolant from a railroad tank car to disposal of contaminated coolant by trucks from a contracted firm.

Design Philosophy and Specifications

The system should be designed to be as financially responsible and efficiently laid out as possible. The dirty coolant, given as 1,000 gallons, is moved from the reservoir tank to the dirty coolant storage tank every week. If it is assumed that there is four weeks in a month, then 4,000 gallons of coolant is moved in a month's time to the dirty coolant storage tank. It is also stated that emergency movement of used coolant may be made. If Continental AG makes one 1,000 gallon emergency dump per month, a total amount of 5,000 gallons of dirty coolant must be stored in a tank with such volume. As dirty coolant is picked up, it must be replaced with an equal amount of clean coolant. Since the train visits every three months, 15,000 gallons of new coolant will be added and used in the reservoir in that time span. Should the factory not make emergency dumps of coolant each month, there would be 3,000 gallons of unused coolant, bringing the grand total of volume for the clean coolant storage tank to 18,000 gallons.

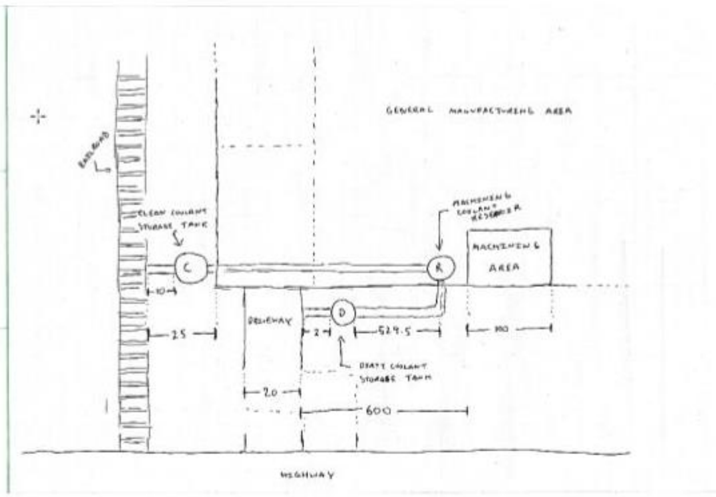
Sources

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

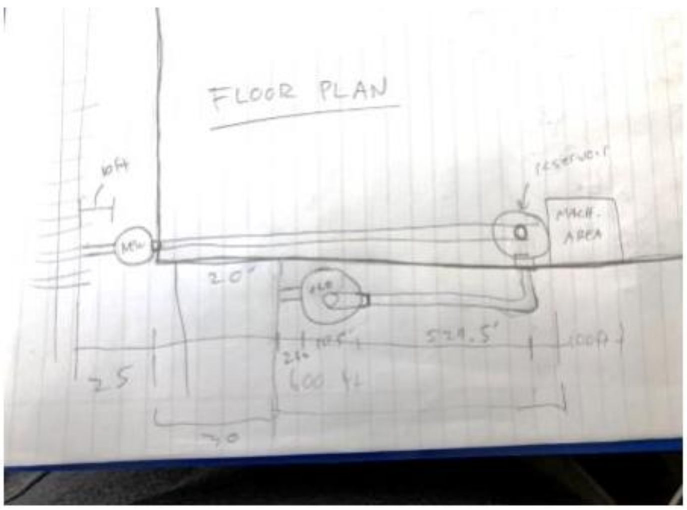
Materials and Specifications

All pipes are commercial-grade seamless carbon steel pipe. The pipe is selected based on typical industry standard availability. All tanks are high-density polyethylene with UV-resistant coatings. The coolant is a solution of water and soluble oil with a specific gravity of 0.94 and a freezing point of 0°F.

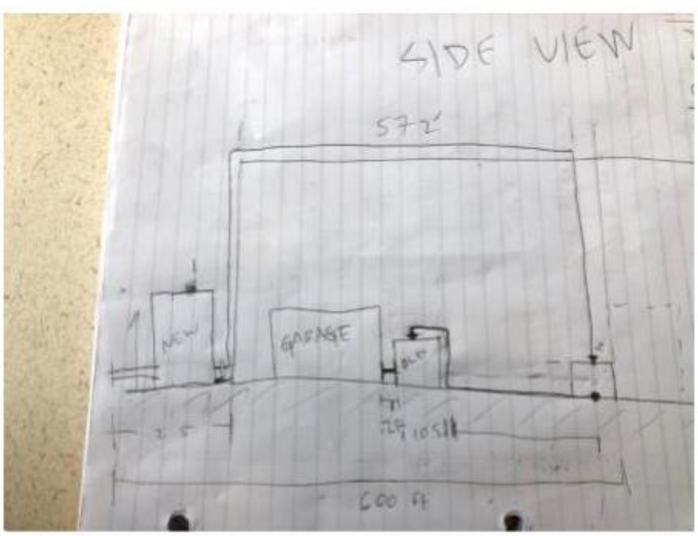
Preliminary Drawings



Preliminary Floor Plan 1



Preliminary Floor Plan 2



Preliminary Elevation Plan

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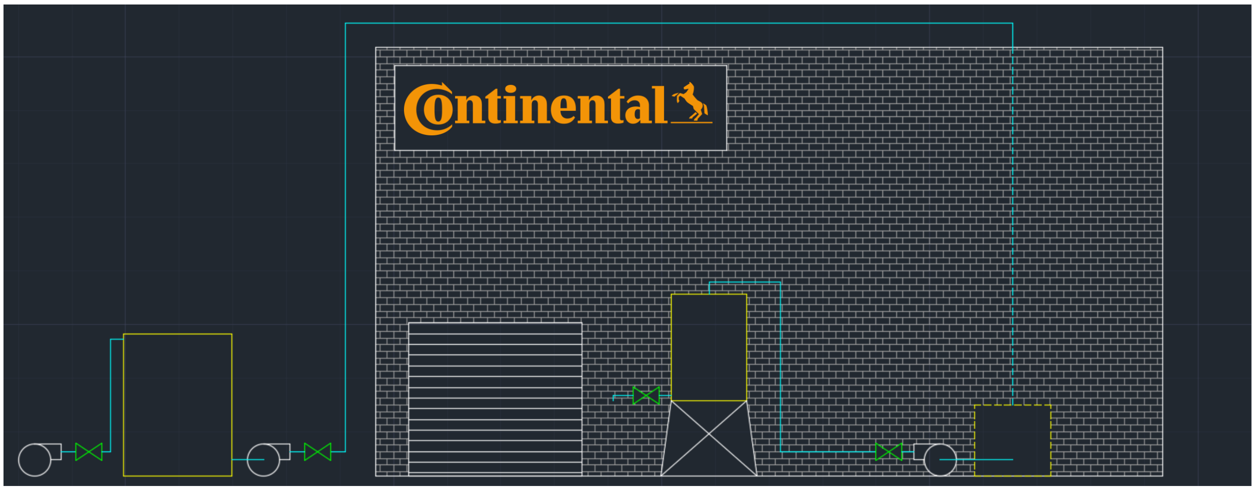
Location & Size Design

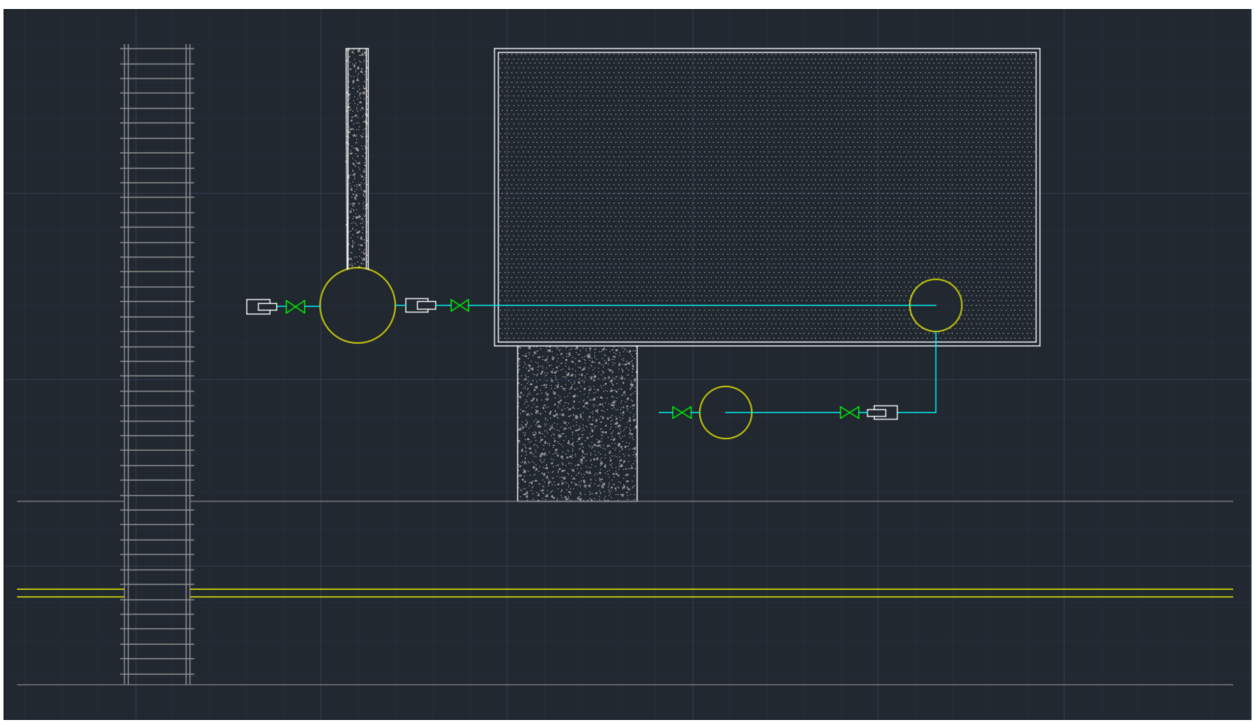
Purpose

The purpose of tank size design is to determine the location and critical dimensions of the clean coolant tank, dirty coolant tank, and machining coolant reservoir.

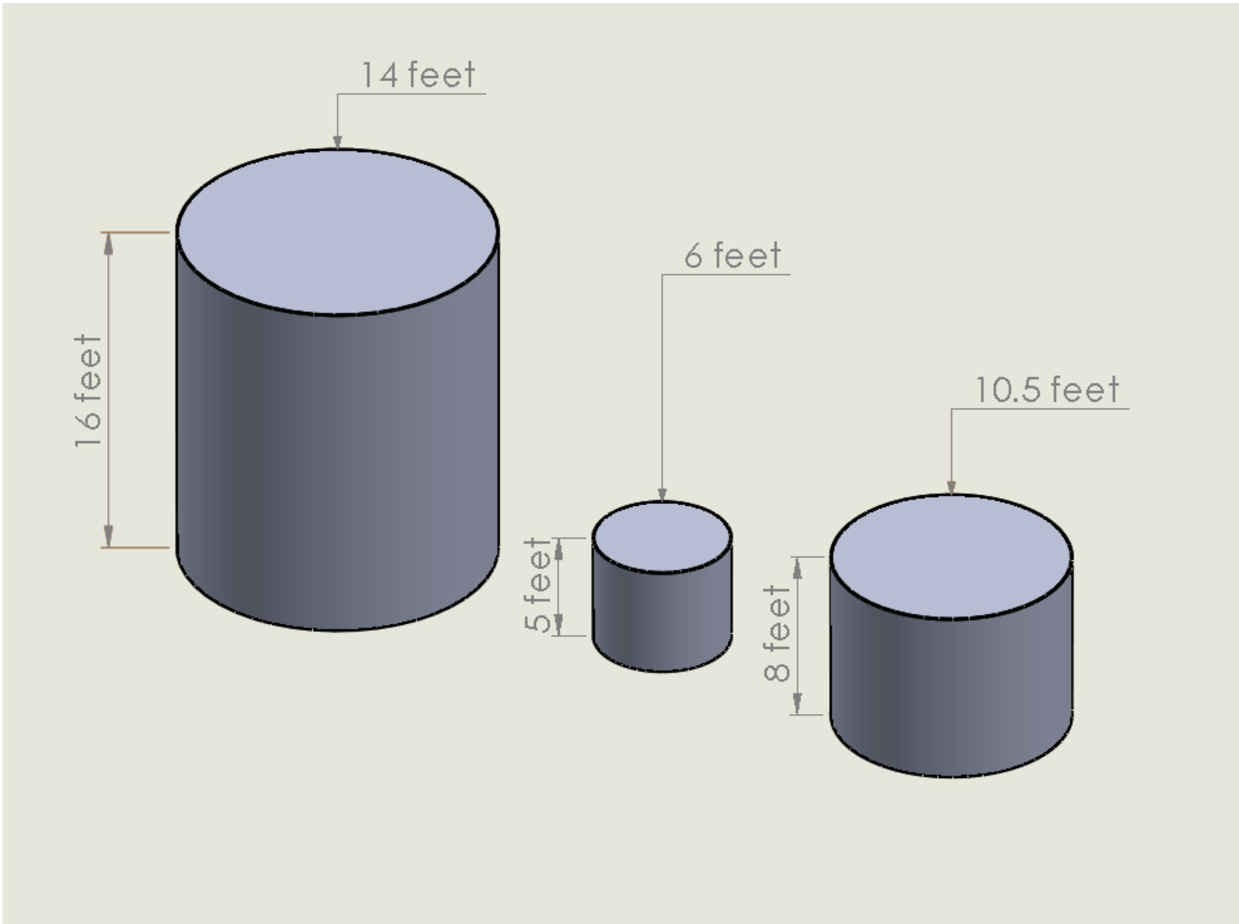
Drawings and Diagrams



Elevation View of Facility



Plot View of Facility with Railway and Road



Tank Dimensions (from left to right: clean coolant, coolant reservoir, dirty coolant)

Sources

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

Design Considerations

- Incompressible fluid
- Steady state system
- Tanks are cylindrical and stable (not too narrow)

Data and Variables

V = volume of each respective tank

d = diameter of each respective tank

h = height of each respective tank

Procedure

The size of each storage tank was determined using known parameters. After finding the optimal size for the dirty coolant storage tank, the sizing for the clean coolant storage tank could be determined. After finding the volume of each tank, calculations can be made about the height and diameter of each tank.

Calculations

For stability, it was recommended that the tank have as close to a 1:1 ratio of height and diameter as possible. The equation for volume of a cylinder is:

$$Volume, V_{tank} = \frac{\pi (d_{tank})^2 (h_{tank})}{4}$$

To find the dimensions for the clean coolant storage tank:

$$V_{clean} = \frac{\pi (d_{clean})^2 (h_{clean})}{4} = \frac{\pi (14\, ft.)^2 (16\, ft.)}{4} = 18{,}000\ gallons$$

To find the dimensions for the machining coolant reservoir:

$$V_{reservoir} = \frac{\pi (d_{reservoir})^2 (h_{reservoir})}{4} = \frac{\pi (6\, ft.)^2 (5\, ft.)}{4} = 1{,}000\ gallons$$

To find the dimensions for the dirty coolant storage tank:

$$V_{dirty} = \frac{\pi (d_{dirty})^2 (h_{dirty})}{4} = \frac{\pi (10.5\, ft.)^2 (8\, ft.)}{4} = 5{,}000\ gallons$$

Summary

Tank Data	Tank 1 Volume	18000	gal	68.13738	m^3
	Tank 2 Volume	5000	gal	18.92705	m^3
	Tank 3 Volume	5000	gal	18.92705	m^3
	Tank 1 Height	16	ft	4.8768	m
	Tank 2 Height	5	ft	1.524	m
	Tank 3 Height	8	ft	2.4384	m
	Tank 1 Diameter	14	ft	4.2672	m
	Tank 2 Diameter	6	ft	1.8288	m
	Tank 3 Diameter	10.5	ft	3.2004	m

The clean coolant storage tank has a volume of 18,000 gallons, a height of 16 feet, and a diameter of 14 feet. The dirty coolant storage tank has a volume of 5,000 gallons, a height of 8 feet, and a diameter of 10.5 feet. The machining coolant reservoir has a volume of 1,000 gallons, a height of 5 feet, and a diameter of 6 feet.

Materials

- Polyethylene water storage tank

Analysis

The location of the tanks was done to maximize the efficiency of unloading new coolant from a railroad car and the offloading of dirty coolant onto a truck. The railroad car, outfitted with 15,000 gallons of new coolant, visits every three months. The truck from a contracted firm visits the factory once a month to collect all the dirty coolant. The clean coolant storage tank will be placed on the side of the factory between the building and the railroad for ease of access to offload the new coolant. The dirty coolant tank will be placed adjacent to the driveway between the garage and the coolant reservoir. Like the other tanks, placement is optimized for maximum efficiency, in this case to make it easier for the truck driver to collect the dirty coolant.

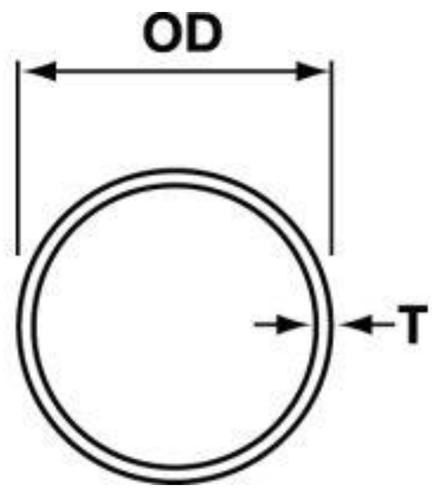
A cylinder was selected as the tank shape. This shape is optimal for overall storage area, space considerations, and overall stability. Steel, stainless steel, and plastic (polyethylene) coolant tanks were considered. The polyethylene tank was selected as the material of choice for several reasons. The price of polyethylene tanks was found to be approximately 50% less than steel and 75% less than stainless steel counterparts. Polyethylene tanks are more durable, lasting more than 20 years, can withstand impacts as well as natural elements, such as the sun, with UV-stabilizers mixed into the molding process. In addition, the plastic tanks would be easier to upgrade, relocate, or expand if needed in the future. Lastly, because the tank will be operating in temperatures with extremes of -20°F, a heater must be purchased and used during the winter months to prevent freezing of coolant and clogged piping. The design of this tank may change based on the fluid and volume requirement.

Tank Thickness

Purpose

The purpose of this section is to determine the minimum wall thickness for the large coolant tank.

Drawings and Diagrams



Pipe Thickness and Outer Diameter

Sources

Mott, R., Untener, J., "Applied Fluid Mechanics," 7^{th} edition Pearson Education, Inc., (2015)

Design Considerations

- Coolant is incompressible
- System is in a steady state

Data and Variables

t = wall thickness

σ = Yield strength of HDPE (high-density polyethylene): 25.9 MPa or 3756.477 psig

γ = Specific weight of coolant: 58.656 lb/ft^3

h = Height of tank: 16 ft

D = Diameter of tank: 14 ft (13.838 rounded up for ease of manufacturing)

Procedure

To find the minimum wall thickness, the yield strength of the high-density polyethylene plastic, specific weight of the coolant, height of the tank, and diameter of the tank are required. Using the formula provided below, the wall thickness is equal to the maximum pressure times the diameter of the tank divided by two times the yield strength.

Calculations

Where $\Delta p = \gamma h$

$$Minimum\ Thickness, t = \frac{\Delta pD}{2\sigma} = \frac{(58.656\ lb/ft^3)(16\ ft)(14\ ft\ )}{(2)(3756.477\ psig)} = 0.01200ft = 0.144in$$

Summary

The minimum wall thickness of the largest tank, the 18,000-gallon clean coolant storage tank, is 0.144 in or 3.675 mm. This value may seem small, but the pressure found at the bottom of the tank was only 938.496 lb/ft^2 or 6.517 psi.

Materials

- Polyethylene water storage tank

Analysis

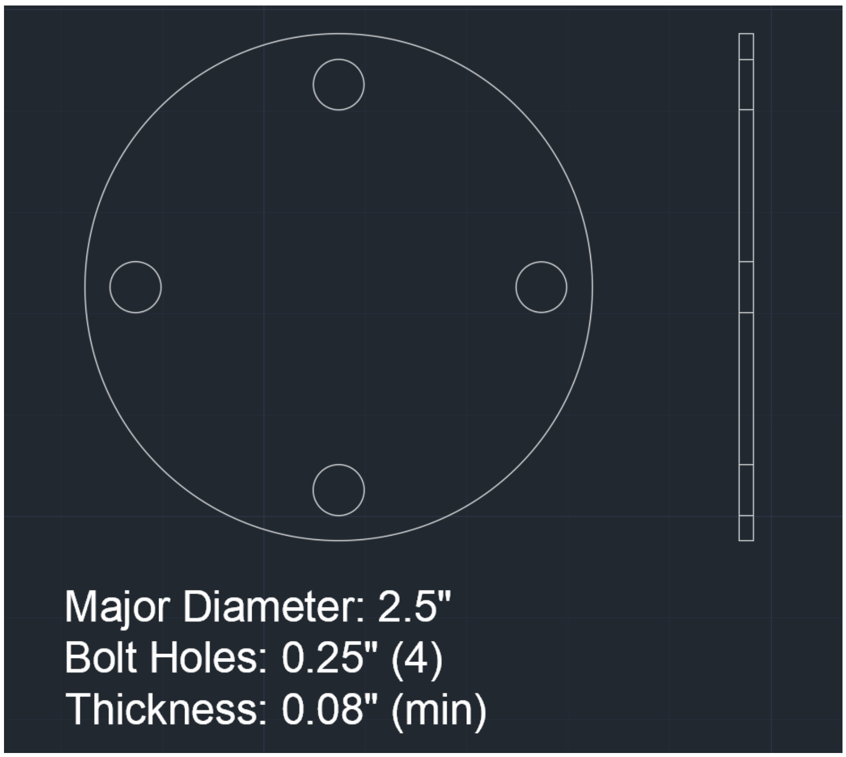
Upon further research, the industry standards (ASTM) requires a polyethylene tank have a wall thickness of 0.187 inches or greater for a tank greater than 15,000 gallons. Therefore, a tank built to ASTM specifications should be selected and will be more than sufficient for the factory's storage purposes. This tank may change based on the volumetric requirement of the manufacturer.

Future Drain Connection

Purpose

The purpose of this section is to determine blind flange thickness and number of bolts needed to incorporate a future drain connection.

Drawings & Diagrams



Blind Flange Front and Side View

Sources

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

“ASME - STANDARDS - Process Piping.” ASME.org, ASME, 2016, www.asme.org/products/codes-standards/b313-2016-process-piping#pdf.

Design Considerations

- New tank is filled to its maximum capacity (18,000 gal)
- Pressure in tank is only the result of gravity acting on the fluid

- Factor of safety for allowable stress in flange is 1.5, since there is no foreseeable circumstance where the tank will see 1.5 times the maximum pressure.

Data and Variables

- Drain diameter d = 1.5 in.
- Specific weight of coolant γ = 58.656 lb/ft^3
- Height of new coolant tank h = 16 ft.
- Bolt stress area A_{Bolt} = 0.0318 in^2
- Bolt (0.25") Proof Load = 55,000 psi

Procedure

The maximum pressure in the tank (P) will be calculated with the relationship γh. The pressure can then be applied to the tank drain and used to specify the thickness and number of bolts needed to secure the blind flange to the tank.

Calculations

$$Pressure, P = \gamma * h = 58.656\, lb/ft^3 * 16ft = 938.469 \frac{lb}{ft^2} = 6.517psi$$

$$Area\ of\ Drain, A_D = \left(\frac{\pi}{4}\right) * D^2 = \left(\frac{\pi}{4}\right) * 1.5in = 1.767in^2$$

$$Force\ on\ Flange, F_{Fl} = \frac{P}{A_D} = \frac{6.517psi}{1.767in^2} = 11.517lbf$$

$$Axial\ Stress, \sigma_{Axial} = \frac{F}{A_{Bolt}} = \frac{11.517lbf}{0.0318in^2} = 362.169psi$$

$$Static\ bolt\ load = \sigma_{Axial} * A_{Bolt} = 362.169psi * 0.0318in^2 = 11.516lb$$

$$Static\ Bolt\ Load < Bolt\ Proof\ Load$$

$$T_{Flange,steel} = \sqrt{\frac{3 * P}{16 * \sigma_{Axial} * F_{safety}}} * d_{Drain} = \sqrt{\frac{3 * 6.517psi}{16 * 362.169psi * 1.5}} * 1.5in = 0.0761in$$

Summary

The total static bolt load will be 11.516lb assuming a 0.25in bolt will be used. Distributing this load amongst 4 bolts will result in an individual bolt load of 2.879lb. The minimum flange thickness is 0.08in (rounded up).

Materials

- Coolant
- Tank with 1.5in drain hole

Analysis

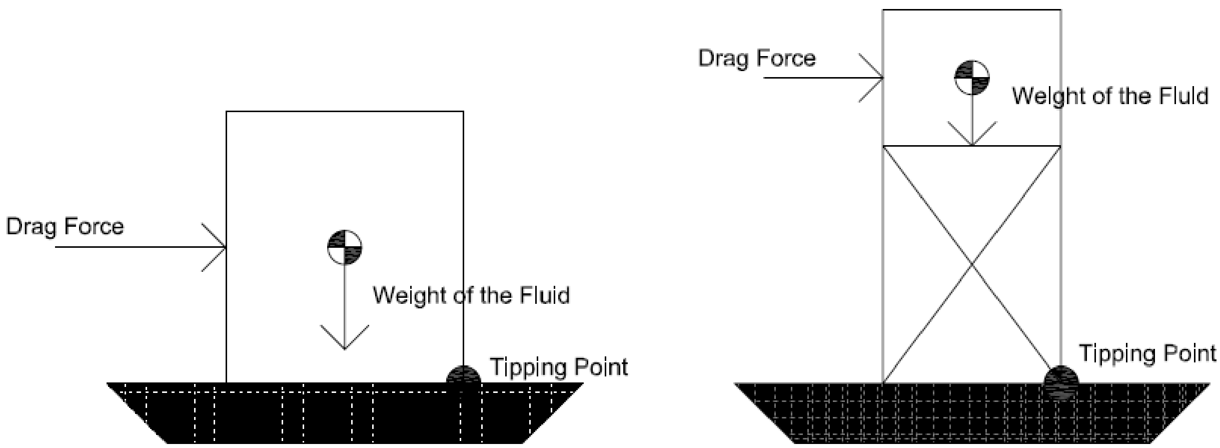
Per the calculations shown above, we have decided to use four 0.25-inch bolts. The blind flange thickness will be 0.08 inch. The standard industry (ASTM A-197, type #150) thickness for a 1.5-inch pipe diameter blind flange is 0.62 inch. In addition, for this particular blind flange, the smallest bolt size would be 0.625-inch bolts. Therefore, it was concluded that the developer for this factory should purchase the industry standard blind flange made for a pipe with dimensions of 1.5-inch diameter and 0.625-inch thickness. It would be cheaper to purchase a mass-produced industrial flange, rather than fabricating a custom part. This would be more than sufficient to withstand the maximum pressure and still allow ease of access for drainage, as well as future expansion. Factors that would influence the thickness of this flange would be tank height, specific gravity of the fluid, and diameter. If any of these characteristics change, this calculation should be repeated to find the new minimum thickness.

Wind Loading and Weight

Purpose

The purpose of this calculation is to verify the tanks will not tip when exposed to high winds.

Drawings and Diagrams



Wind Force and Coolant Tank Diagrams

Sources

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

"Dayton, OH Weather Forecast and Conditions - The Weather Channel." The Weather Channel, TWC Product and Technology LLC, 27 Apr. 2018, weather.com/weather/today/l/Dayton OH USOH0245:1:US.

Design Considerations

- Uniform wind load along the y axis
- The weight of the tank is neglected because the weight of the fluid is proportionally larger than the tanks own weight

Data and Variables

- Maximum recorded wind speed in Dayton, V = 56 mph
- Specific weight of the fluid, $\gamma = 9221.4\ N/m^3$
- Diameter of Clean Tank D_{clean} = 14 ft = 4.2672 m

- Height of Clean Tank, h_{clean} = 16 ft = 4.8768 m
- Diameter of Dirty Tank, D_{dirty} = 10.5 ft = 3.2004 m
- Height of Dirty tank, h_{dirty} = 8 ft = 2.4384 m
- Density of air, ρ = 1.292 kg/m^3
- Kinematic Viscosity of air, ν = 1.33 x 10^{-5} m^2/s

Procedure

Since there are two tanks that are subjected to the elements outside of the building, the drag force from the wind needs to be calculated for both tanks. Starting with the larger clean tank, the drag force requires that the coefficient of drag. To find this, the velocity, diameter and viscosity are needed to find the Reynolds number. After calculating Reynolds number, the coefficient of drag can be read from Graph 17.4. Substituting known values, the drag force can be found. Next the case where the tank tips needs to be checked to make sure the wind will not topple the tank. Taking moments about the rotational point for the tank, the moment created from the weight and the moment created by the wind force can be found.

Calculations

First the force due to drag,

$$Drag\ Force, F_D = \frac{C_D \rho V^2 A}{2}$$

Calculating the coefficient of drag,

$$Reynolds\ Nubmer, N_R = \frac{vD}{\nu} = \frac{25.0342\ \frac{m}{s} * 4.2672\ m}{1.33 * 10^{-5}\frac{m^2}{s}} = 8034025$$

Reading from Graph 17.4,

$$C_D = .3$$

Substituting known values into the drag force equation,

$$F_D = \frac{(0.3)\left(1.292\ \frac{kg}{m^3}\right)\left(25.0342\frac{m}{s}\right)^2 (4.8768\ m * 4.2672\ m\ )}{2} = 2527.5N$$

Calculating the weight of the fluid,

$$Weight\ of\ Fluid, W_f = \gamma V$$

$$W_f = \left(9221.4\ \frac{N}{m^3}\right)\left(\frac{\pi(4.2672\ m)^2}{4}\right)(4.8768\ m) = 643143.2\ N$$

Calculating for moments about the tipping point,

$$Moment\ due\ to\ Fluid\ Weight, M_f = W_f * h = 643143.2N * 2.1336m = 1372210.3\ N * m$$

$$Moment\ due\ to\ Wind, M_{F_D} = W_f * h = 2527.5\ N * 2.4384\ m = 6163.1\ N * m$$

Repeating the process for the dirty fluid tank,

$$N_R = \frac{\left(25.0342\frac{m}{s}\right)(3.2004\ m)}{1.33 * 10^{-5}\frac{m^2}{s}} = 6024019.1\ (Unitless)$$

Reading from the graph 17.4,

$$C_D = 0.3$$

Solving again for the drag force,

$$Drag\ Force, F_D = \frac{(.3)\left(1.292\frac{kg}{m^3}\right)\left(25.0342\frac{m}{s}\right)^2(2.4384\ m * 3.2004\ m)}{2} = 947.83\ N$$

$$Weight\ of\ Fluid, W_f = \left(9221.4\frac{N}{m^3}\right)\left(\frac{\pi(3.2004\ m)^2}{4}\right)(2.4384\ m) = 180884\ N$$

Since this tank is raised, the moment arms will change in this calculation,

$$M_f = 180884N(1.6002\ m) = 289450.6\ N * m$$

$$M_{F_D} = 947.83\ N(5.4864\ m) = 5200.174\ N * m$$

<u>Summary</u>

Since the moment created by the fluid is larger than the moment created by the wind force, the tanks will not topple.

<u>Materials</u>

- Coolant
- Tanks

Analysis

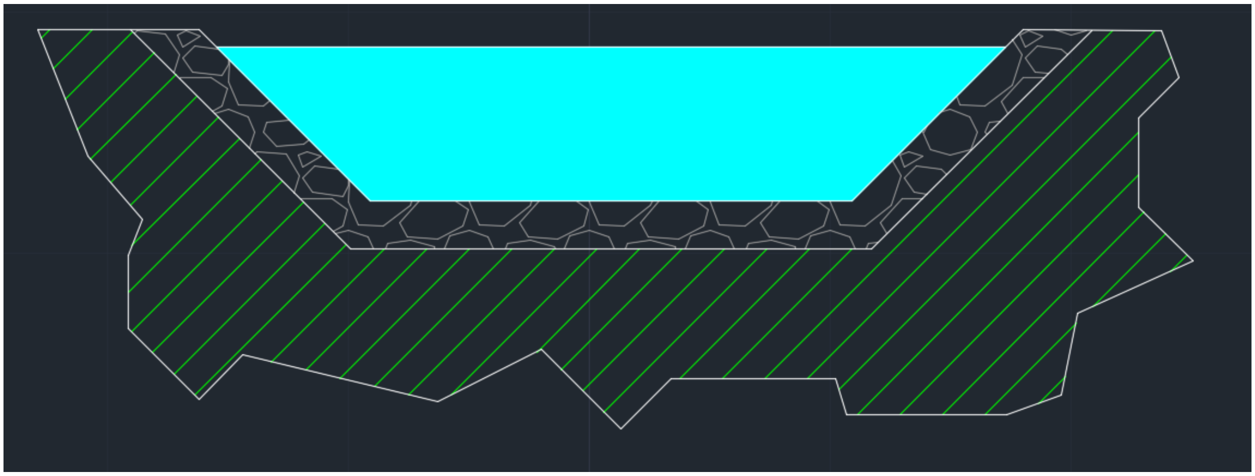
With the current design, the weight of the fluid is more than enough to counteract the moments due to the wind force. Even with the case of the elevated tank, the wind force is not capable of tipping the tanks. This may be subject to change if the fluid weight decreases or if the tank height increases. In this calculation the weight of the tanks was neglected in the calculations because the weight of the fluid is more than enough to prove that the tanks will not tip due to wind conditions. Including the weight of the tanks would only help the tanks resist the wind force more in the case of wind loading.

Open Channel for Drainage

Purpose

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

Drawings & Diagrams



Cross-Section of Open Channel in Ground

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 in the system (58 gpm)
- Trapezoidal profile
- Base is 1ft wide to allow crossing by foot without a bridge
- Concrete is assumed to be the channel material because it is relatively inexpensive, easy to apply, and readily available

Data & Variables

- Channel Breadth, B = 1 ft
- Channel Angle, z = 1 ft

- Manning's Constant for Concrete, n = 0.013
- Slope = 0.00015

Procedure

Manning's equation will be used to calculate the depth of a trapezoidal channel. This will be done by substituting the assumed values (breadth, slope, wet perimeter, hydraulic radius) and using the iterative process to solve for the depth.

Calculations

Manning's Equation,

$$(b+zy)y*\left(\frac{b+zy}{b+2y*\sqrt{1+z^2}}\right)^{\frac{2}{3}}=\frac{n*Q}{1.49*\sqrt{S}}\rightarrow(1y+y^2)*\left(\frac{1y+y}{1+2y*\sqrt{1}}\right)^{\frac{2}{3}}=0.0919$$

Then using the iterative process to equate the left and right hand side,

Height, Y	Left	Right	Error
0.35	0.112200347	0.0919	22.090%
0.34	0.105821689	0.0919	15.149%
0.33	0.099642678	0.0919	8.425%
0.32	0.093661683	0.0919	1.917%
0.31	0.087877107	0.0919	-4.377%
0.315	0.090744942	0.0919	-1.257%
0.316	0.091324371	0.0919	-0.626%
0.317	0.091905758	0.0919	0.006%
0.318	0.092489105	0.0919	0.641%
By iteration, ideal depth is		0.317	ft
		3.804	in

This iteration includes an error of 0.006%, which is assumed to be negligible.

Summary

The results of this calculation are shown in the table below.

Trapezoidal Channel Data			
Variable	Value	Unit	Equation
Flow Rate, Q	58.00	gpm	Assumed
Area, A	0.42	ft^2	(B+ZY)Y
Wet Perimeter, WP	1.90	ft	B+2Y*SQRT(1+Z^2)
Hydraulic Radus, R	0.22	ft	A/WP
Width, B	1.00	ft	Assumed
Taper, Z	1.00	none	Assumed
Height, Y	0.32	ft	Calculated
Average Slope, S	0.00	Slope	Assumed

Materials

- Concrete
- Coolant

Analysis

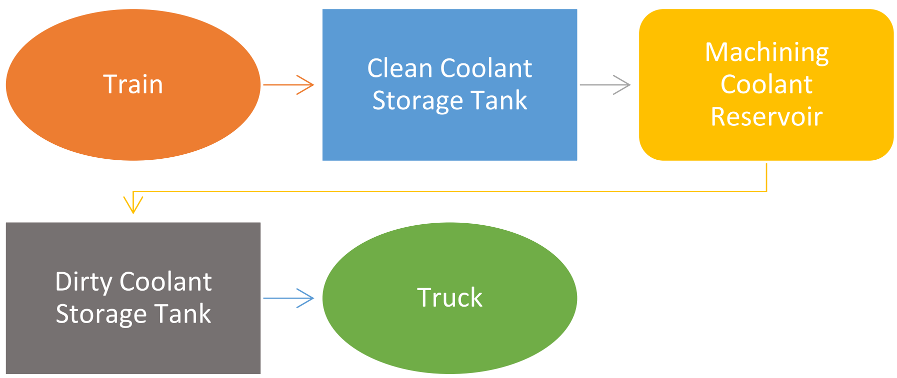
The calculations of this section indicate the height of the channel should be 3.804in, although it would be reasonable to say the height should be 5in. The reason for this is to compensate for rain or leaks in the tank which may make the total flowrate exceed 58gpm. This depth is dependent on many factors such as flow rate, channel material, and average slope of the surrounding land. If any of these critical factors change, this procedure should be re-examined to adjust the channel depth.

Flow rate

Purpose

The purpose of calculating flow rate is to determine the desired fill and empty times of each coolant tank.

Drawings and Diagrams



Delivery System Flowchart

Sources

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

Design Considerations

- Coolant is incompressible
- System is in a steady state

Data and Variables

$V_{Clean\ Coolant}$ = 15,000 gallon

$V_{Dirty\ Coolant}$ = 5,000 gallons

$V_{Coolant\ Reservoir}$ = 1,000 gallons

T_{Shift} = 4 hours

Procedure

To calculate the fill and empty times of each tank, the fixed volumetric flow rate needed to be calculated first. The flow rate was calculated by dividing tank volume by time. That fixed rate was then substituted into the same equation for each different storage tank volume to calculate the time for each tank to fill and empty.

Calculations

$$Q = \frac{tank\ volume}{time} = volumetric\ flow\ rate$$

$$Q = \frac{15{,}000\ gallons}{4\ hours} * \frac{1\ hour}{60\ minutes} = 62.5\ gallons\ per\ minute$$

$$Q = 62.5\ gallons\ per\ minute$$

The desired flow rate for all tanks is 62.5 gallons per minute.

$$volumetric\ flow\ rate = \frac{volume}{time}$$

$$time = \frac{volume}{volumetric\ flow\ rate}$$

From the train with new coolant to the clean coolant tank:

$$T = \frac{15{,}000\ gallons}{62.5\ gallons\ per\ minute}$$

$$T = 240\ minutes\ or\ 4\ hours$$

From the clean coolant tank to the machining coolant reservoir:

$$T = \frac{1{,}000\ gallons}{62.5\ gallons\ per\ minute}$$

$$T = 16\ minutes$$

From the machining coolant reservoir to the dirty coolant tank:

$$T = \frac{1{,}000\ gallons}{62.5\ gallons\ per\ minute}$$

$$T = 16\ minutes$$

From the dirty coolant tank to the truck:

$$T = \frac{5{,}000\ gallons}{62.5\ gallons\ per\ minute}$$

$$T = 80\ minutes\ or\ 1.333\ hours$$

Summary

The calculated volumetric flow rate was calculated using a fixed time interval of 4 hours to get an ideal fill/empty time. However, due to the pump selection, we have found that the maximum flow capacity of the pump will the 58 gpm. The table below summarizes the ideal and pumped values for fill and empty time.

Filling and Emptying Times		
System	Ideal	Actual
To Tank 1	240 min	258 min
To Tank 2	16 min	17.24 min
To Tank 3	16 min	17.24 min
To Truck	80 min	86.2 min

Materials

- Polyethylene water storage tank

Analysis

The estimations for each tank was done assuming that each tank is filled to their maximum capacity. In addition, all tasks were purposely limited to a maximum 4-hour runtime to ensure that only one technician starts and completes the task, minimizing human error. Upon applying the pumped flow rate, the time to completely fill the new coolant tank changed to 4 hours and 18 minutes. This additional time does not impede the functionality of the system. Several factors influence this time such as pump capacity, flow rate, and tank volume. This calculation must be revisited when changing any of these properties.

Pipe sizing

Purpose

The purpose of this section is to specify the layout of the piping system, pipe material and sizing.

Drawings and Diagrams

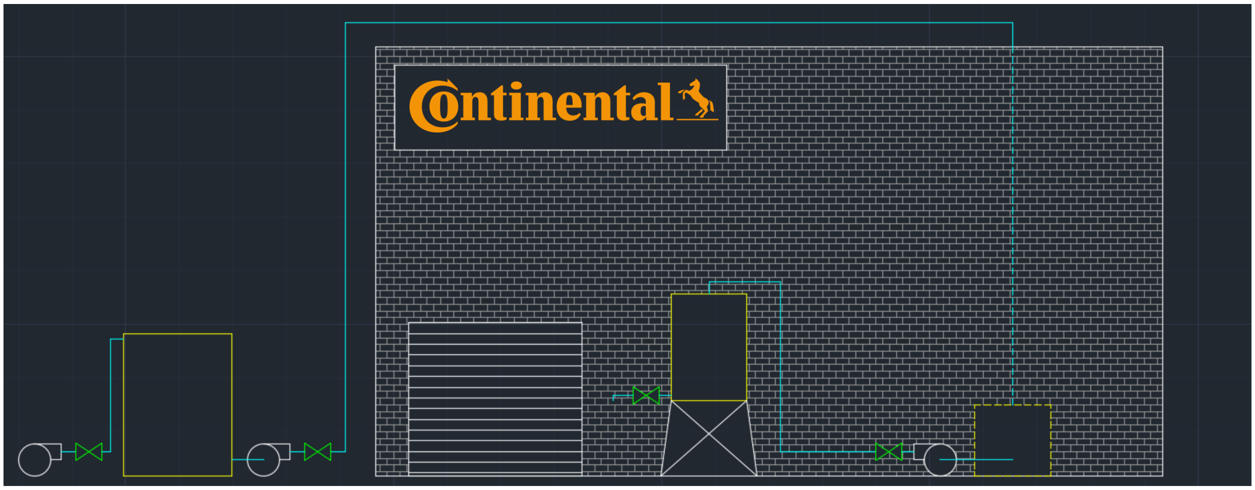


Diagram 1: Elevation View with Tanks and Pipelines

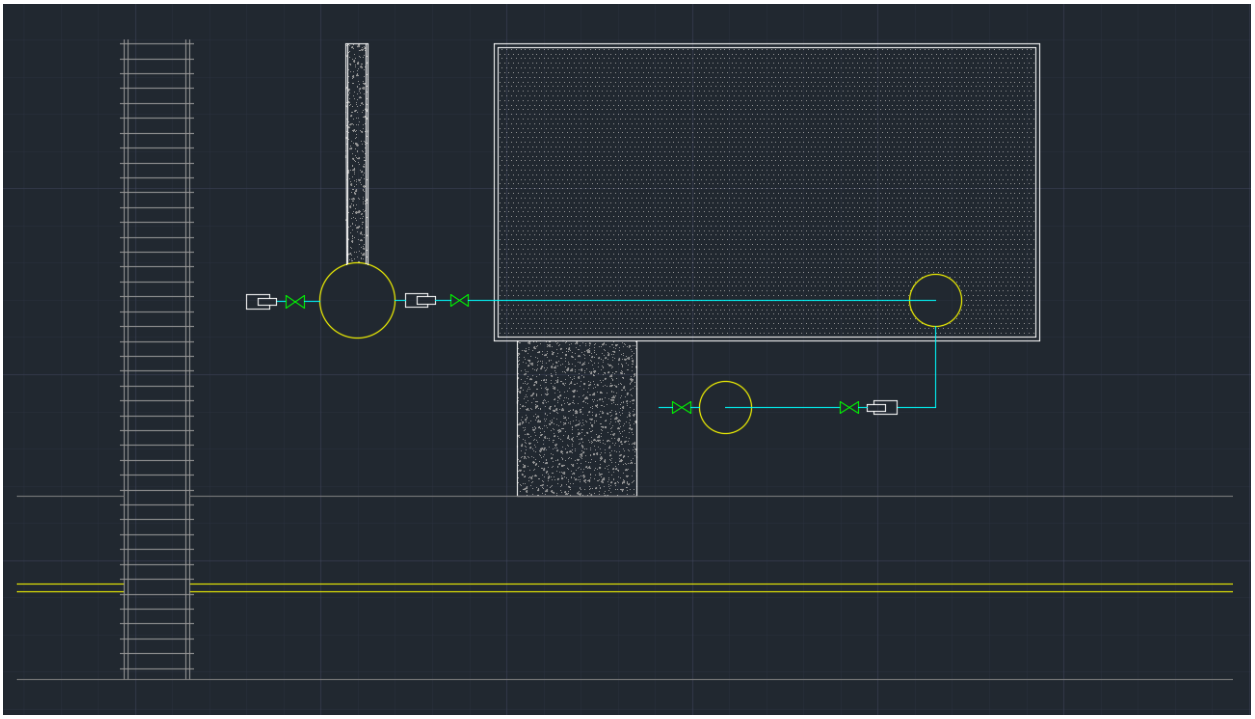


Diagram 2: Plot with Tanks and Pipelines

Sources

Mott, R., Untener, J., "Applied Fluid Mechanics," 7^{th} edition Pearson Education, Inc., (2015)

Design Considerations

The pipe will be outside, and the fluid flowing through the pipe has a higher freezing point then the location's lowest temperature. This means a heater must be applied to the fluid or some form of pipe insulation must be added.

Data and Variables

Length from train to clean tank, $L_{RR\text{-}New}$ = 18 feet

Length from clean tank to reservoir, $L_{New\text{-}Res}$ = 632 feet

Length from reservoir to dirty tank, $L_{Res\text{-}Old}$ = 557.75 feet

Length from dirty tank to truck, $L_{Old\text{-}Truck}$ = 5 feet

Procedure

To calculate the inside diameter of the pipe needed, the equation for flow rate will be used. Substituting in known values, it is possible to calculate the inside diameter of the pipe needed. To calculate the length of pipe necessary for the system, the layout of piping must be defined. The locations of the tanks and the building are also needed. The tanks were placed as close to the designated locations of loading and unloading to reduce the length of pipe necessary for the system. For calculating the length, the sum of all the lengths must be found.

Calculations

From the equation for flow rate,

$$Flow\ rate, Q = V * A$$

Expanding term for area,

$$Q = v * \frac{\pi * d^2}{4}\ ft^2$$

Solving in terms of d,

$$Internal\ Diameter, d = \sqrt{\frac{4 * Q}{v * \pi}} = \sqrt{\frac{4 * 0.00365}{2.78 * \pi}} = 0.0409m$$

For pipe length:

From Diagram 1 and 2,

$$Total\ Length\ of\ pipe, \Sigma L_{pipe} = L_{RR-New} + L_{New-Res} + L_{Res-Old} + L_{Old-Truck}$$

Where,

$L_{RR\text{-}New}$ = Length of pipe from the railroad car to the new tank

$L_{New\text{-}Res}$ = Length of pipe from the new tank to the, reservoir

$L_{Res\text{-}Old}$ = Length of pipe from the reservoir tank to the used oil tank

$L_{Old\text{-}Truck}$ = Length of pipe from the used oil tank to the truck offloading site

$$\Sigma L_{pipe} = 18\ ft + 632\ ft + 557.75\ ft + 5\ ft = 1212.75ft$$

<u>Summary</u>

The total length of pipe needed for the system is 1212.75 feet and the inside diameter of the pipe needed for the flow rate is 0.0409 meters or 1.61 inches.

<u>Materials</u>

- Schedule 40 Steel Pipe

<u>Analysis</u>

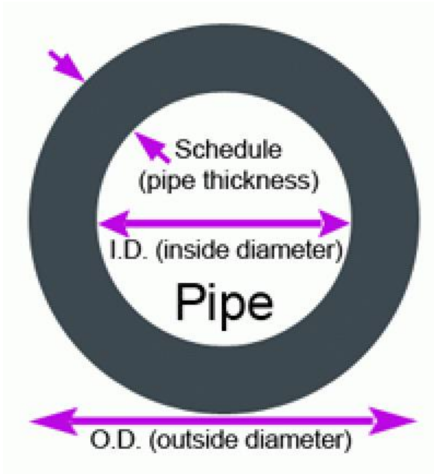
The inside diameter of the pipe calculated is the same as a 1.5” Schedule 40 steel pipe. This being the case, it is simple to see this pipe would be the simple and obvious choice for the pipe system. The two major factors that determine the internal diameter are velocity and flow rate. If either of these changes, internal diameter must be recalculated.

Pipe Thickness

Purpose

The purpose of finding pipe thickness is to determine a proper wall thickness so that the pipe should not fail while under operating conditions.

Drawings and Diagrams



Outer Diameter and Wall Thickness

Sources

Mott, R., Untener, J., "Applied Fluid Mechanics," 7^{th} edition Pearson Education, Inc., (2015)

Design Considerations

- Coolant is incompressible
- System is in a steady state

Data and Variables

Inner Diameter of Pipe = 1.61 inches

Design Pressure, P = 17.14 psi

Outside Diameter, D = 1.5 inches

Allowable Stress, S= 20,000 psi

Longitudinal Joint Quality, E = 1

Correction Factor (Steel, Room Temperature) = 0.4

Corrosion Allowance, A = 0.08in

Procedure

An equation for basic wall thickness with a tolerance of +0/-12.5% is applied. After finding the minimum wall thickness, a chart for “Schedule 40 Steel Pipe” in the textbook, “Applied Fluid Mechanics,” 7th edition, is used to find the industrial standard for such piping material.

Calculations

$$t_{nominal} = 1.145\left[\frac{pD}{2(SE+pY)} + A\right] = 1.145\left[\frac{(17.14\,psi)(1.5\,in.)}{2[(20{,}000\,psi)(1)+(17.14)(0.4)]} + 0.08\,in.\right]$$
$$= 0.0806in$$

Summary

The minimum nominal wall thickness was found to be 0.0806 inches.

Materials

- 1.5-inch nominal Schedule 40 Steel Pipe

Analysis

A wall thickness of 0.0806 inches is not an industry standard, so a standardized pipe thickness needs to be applied. Using the referenced table, the selected pipe thickness for 1.5-inch schedule 40 steel pipe is 0.145 inches. Steel pipe was selected as it will not galvanize with fittings and other piping. This pipe thickness is more than sufficient for the low pressure this system will experience but adds a factor of safety against bursting. This thickness may change depending on the corrosive factor, weld joint quality, allowable stress, and design pressure.

Fittings

Purpose

The purpose of this section is to specify the quantity, type, and size of all valves, elbows, and fittings.

Drawings and Diagrams

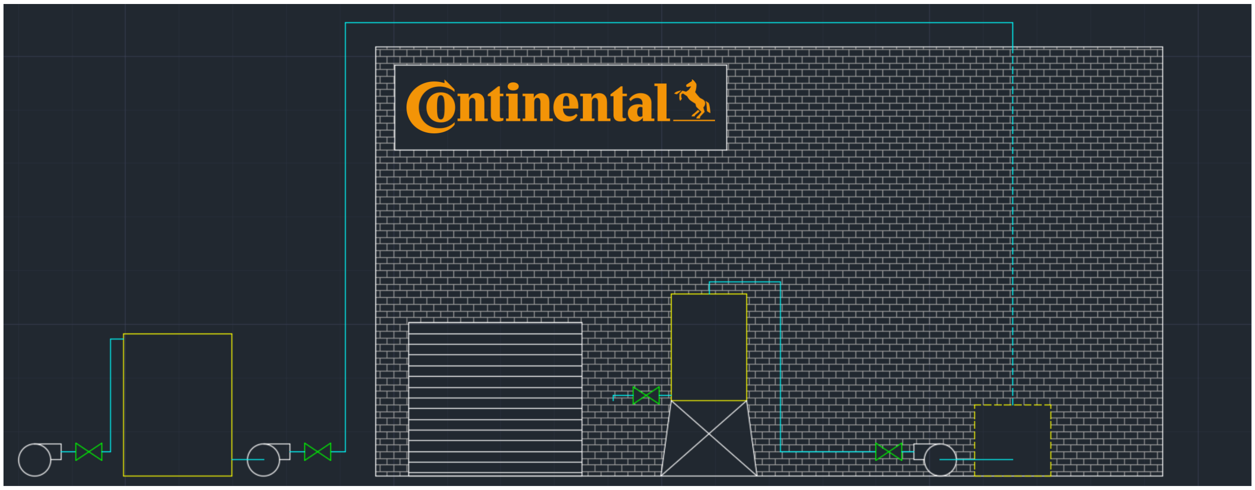


Diagram 1: Elevation View with Tanks and Pipelines

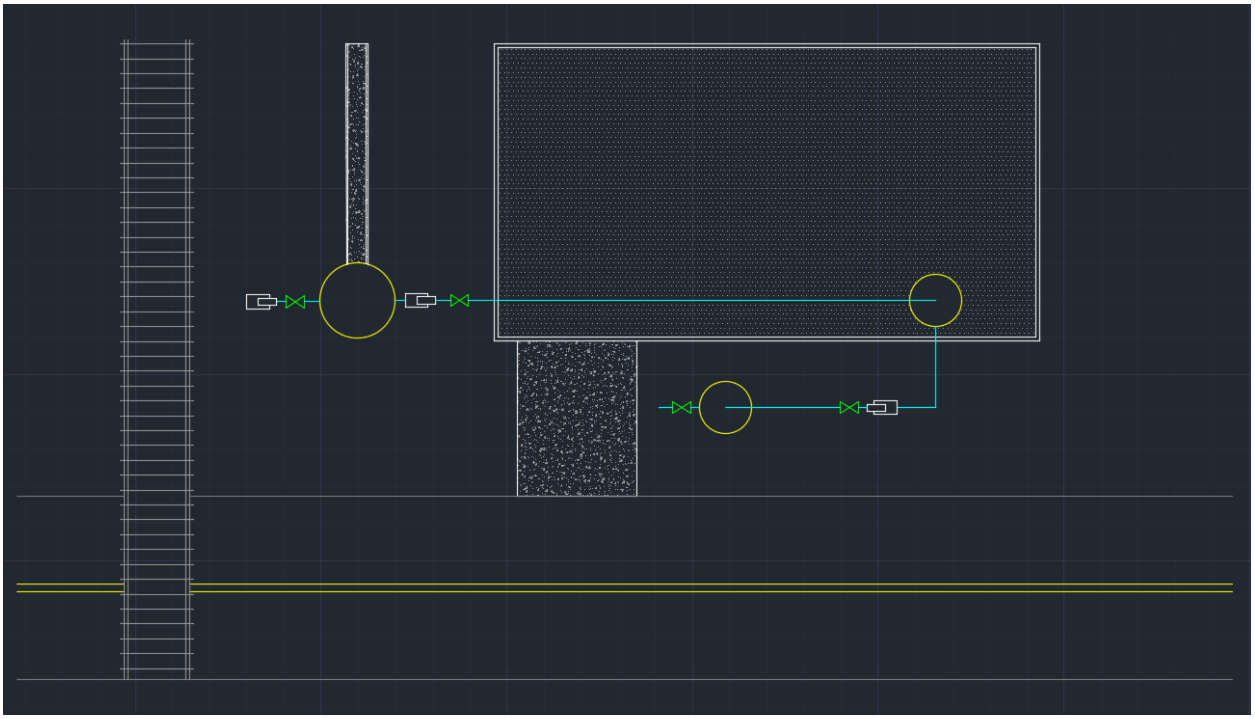


Diagram 2: Plot with Tanks and Pipelines

Sources

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

Design Considerations

The building location and size is fixed and pipes will need to be routed to accommodate for such parameters.

Data and Variables

Length from train to clean tank, $L_{RR\text{-}New}$ = 10 feet

Length from clean tank to reservoir, $L_{New\text{-}Res}$ = 632 feet

Length from reservoir to dirty tank, $L_{Res\text{-}Old}$ = 543.75 feet

Length from dirty tank to truck, $L_{Old\text{-}Truck}$ = 2 feet

Total Length of Pipe, ΣL_{Pipe} = 1187.75 feet

Procedure

Using the diagrams above, the number of bends can be determined to find the number of elbows. For each system, there will need to be a valve at the end of the pipe nearest the outlet to prevent fluid from leaving the pipe when the system is not being filled or emptied. This allows for the number of valves to be determined.

Calculations

For the number of elbows and valves,

Number of elbows = number of bends in pipe

Number of valves = number of exits for the pipe

It can be determined from the diagrams that

Number of elbows = 7

Number of valves = 2

Summary

There must be 7 elbows and 2 valves for the pipe system.

Materials

- 1.5-inch Schedule 40 Steel Pipe

Analysis

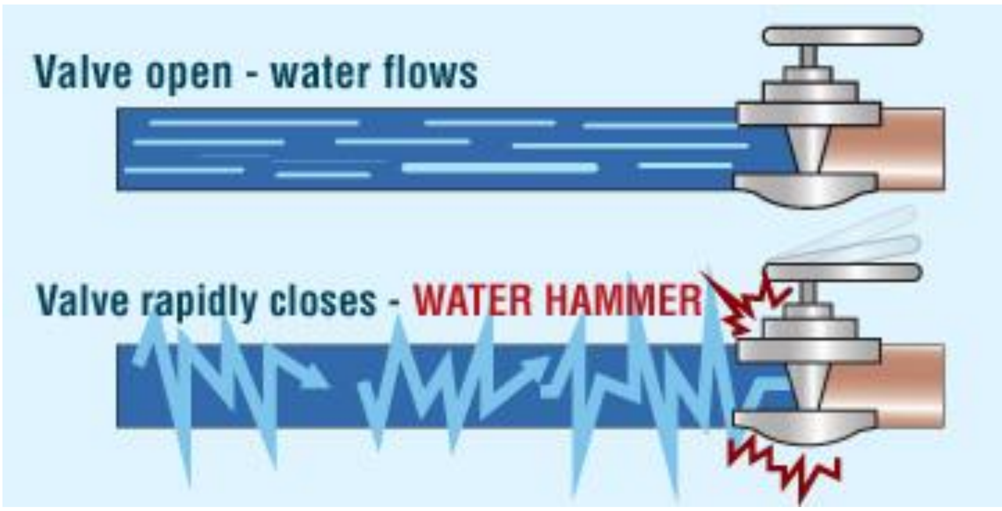
Once the layout of the pipe has been designed, the number of elbows and valves is simple to calculate and find. The fittings should be the same size as the pipe, which is 1.5 in for all elbows and valves. The elbows and valves should also be the same material as the pipe to eliminate the possibility of galvanization between fittings. This design could be modified in the future in the event of maintenance or even possible expansion. It is important to note that modifications to this system, in particular, addition of fittings, would incur greater energy losses, which may require modifications or even replacement of the currently selected pump.

Water Hammer

Purpose

The purpose of this calculation is to determine the pipeline will not be at risk for damage from impulsive forces due to sudden valve closure

Drawings & Diagrams



Sources

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

"Water Hammer - The Causes and Effects." Hays Fluid Controls | Blog, 30 June 2015, www.haysfluidcontrols.com/blog/water-hammer-the-causes-and-effects/.

Design Considerations

- Incompressible fluid
- No air in system
- Valves closed instantaneously
- System in steady-state before valve closes

Data & Variables

Fluid & Pipe Properties	Yield Strength, Sy	36000	psi	350	MPa
	Wall thickness, t	0.145	in	0.0037	m
	Outer diameter, do	1.9	in	0.0483	m
	Bulk modulus of coolant, Eo	316000	psi	2179	MPa
	Elastic modulus, E	29000000	psi	200	GPa
	Density of coolant, ro	0.012664	slug/in^3	6.526702	kg/m^3

- Velocity, V = 9.14 ft/s
- Operating pressure, P_{op} = 17.14 psi

Procedure

The yield pressure is calculated using Barlow's equation to determine the pressure at which the system can be subject to before bending. After this pressure is obtained, the water hammer coefficient (C) is calculated, along with the change in pressure (ΔP) and maximum pressure (P_{max}). The maximum pressure is then compared to the yield pressure to determine if shock arrestor devices should be incorporated into the design.

Calculations

$$Yield\ Pressure, P_y = \frac{2 * S_y * t}{d_0} = \frac{2 * \ 36000psi * 0.145in}{1.9in} = 5497.74\ psi$$

$$Water\ Hammer\ Coefficient, C = \frac{\sqrt{\frac{E_0}{r_o}}}{\sqrt{1 + \frac{E_0 * D}{E * t}}} = \frac{\sqrt{\frac{316000psi}{\frac{0.012664slug}{in^3}}}}{\sqrt{1 + \frac{316000psi * 1.9in}{29000000psi * 0.145in}}} = 4672.81\ psi$$

$$Change\ in\ Pressure, \Delta P = \ \rho * C * V = 0.012664slug/in^3 * 4672.81psi * 9.14ft/s = 45.07\ psi$$

$$Maximum\ Pressure, P_{max} = \Delta P + P_{op} = 45.07psi + 17.14psi = 62.21\ psi$$

$$P_{max} < P_y$$

Summary

After confirming the maximum pressure does not exceed the yield pressure, it was determined that we do not need to add a shock arrestor device.

Materials

- Coolant

Analysis

It was determined that a shock arrestor was not needed for this system, since the yield pressure far exceeds the maximum operating pressure. Factors that influence water hammer include, but are not limited to: operating pressure, pump sizes, velocity, pipe material, coolant composition, and pipe diameter. If any of these major factors change, this calculation will need to be re-evaluated to determine if a shock arrestor is necessary.

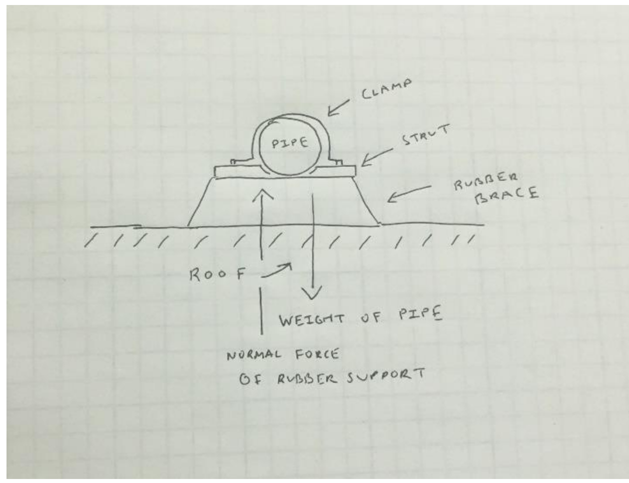
Pipeline Support Information

Purpose

The purpose of researching pipeline supports it specify the type of supports, distance or span between each support, and the forces applied to each support.

Drawings and Diagrams

Pipe Support



Free-Body Diagram

Sources

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

Hibbler, R. C., "Engineering Mechanics: Statics," 14th edition Pearson Education, Inc., (2016)

Design Considerations

The supports are tested and designed for a steady state, or that the building is not moving.

Data and Variables

Weight of Pipe, W_{pipe} = 2.72 lbs/ft or 0.227 lb/in

Weight of Coolant W_{Cool}= 0.9 lbs/ft or 0.075 lb/in

Length of Span, L

Modulus of Elasticity, E = 225 GPa or 32,630,000 psi

Deflection, Δ = 0.015 in.

Moment of Inertia, I = 0.3099 in.^{4}

Factor of Safety, FoS = 1.25

Procedure

The type of support is specified based on the position of the pipe. The deflection equation for a simply supported beam is used for the calculations for the span between each support. Using the equation for deflection, an iteration process is done to calculate the minimum span length.

Calculations

The basic equation for a simply supported beam:

$$Deflection, \Delta = \frac{5WL^4}{384EI}$$

Applying a factor of safety of 5/4 or 1.25:

$$Deflection, \Delta = \frac{5WL^4}{384EI} \times \frac{5}{4} = \frac{25WL^4}{1536EI}$$

Because the length is dependent of the weight per length of the span, an iteration process is required. Solving the equation for length:

$$L = \sqrt[4]{\frac{1536EI\Delta}{25W}} = \sqrt[4]{\frac{1536(3.25 \times 10^7\ psi)(0.3099\ in^4)(0.015\ in.)}{25(43.44\frac{lb}{in})}} = 144\ inches = 12\ feet$$

Modulus of Elasticity		Iteration Step	Length of Span (ft.)	Length of Span (in.)	Weight of Pipe (lb/in)	Weight of Water (lb/in)	Total Weight (lb/in)	Actual Length of Span (in.)	Percent Error
225	Gpa	1	5	60	13.6	4.5	18.1	113.9632754	0.899387923
3.26E+07	psi	2	6	72	16.32	5.4	21.72	119.2779686	0.656638453
		3	7	84	19.04	6.3	25.34	123.9643854	0.475766493
Moment of Inertia		4	8	96	21.76	7.2	28.96	128.1725186	0.335130402
0.3099	in^4	5	9	108	24.48	8.1	32.58	132.0027713	0.222247882
		6	10	120	27.2	9	36.2	135.525938	0.129382816
Requested Deflection		7	11	132	29.92	9.9	39.82	138.7939683	0.051469457
0.015	inches	8	11.5	138	31.28	10.35	41.63	140.3449796	0.016992606
		9	11.75	141	31.96	10.575	42.535	141.1015837	0.000720452
Weight of Pipe		10	12	144	32.64	10.8	43.44	141.846209	-0.014956882
2.72	lb/ft								
0.226667	lb/in								
Weight of Water									
0.9	lb/ft								
0.075	lb/in								
Safety Factor									
1.25									

Summary

Using the iterative process, the distance between spans was found to be approximately 11.75 feet or 141 inches.

Materials

- Small rubber roof pipe support
- Galvanized steel strut and clamp assembly

Analysis

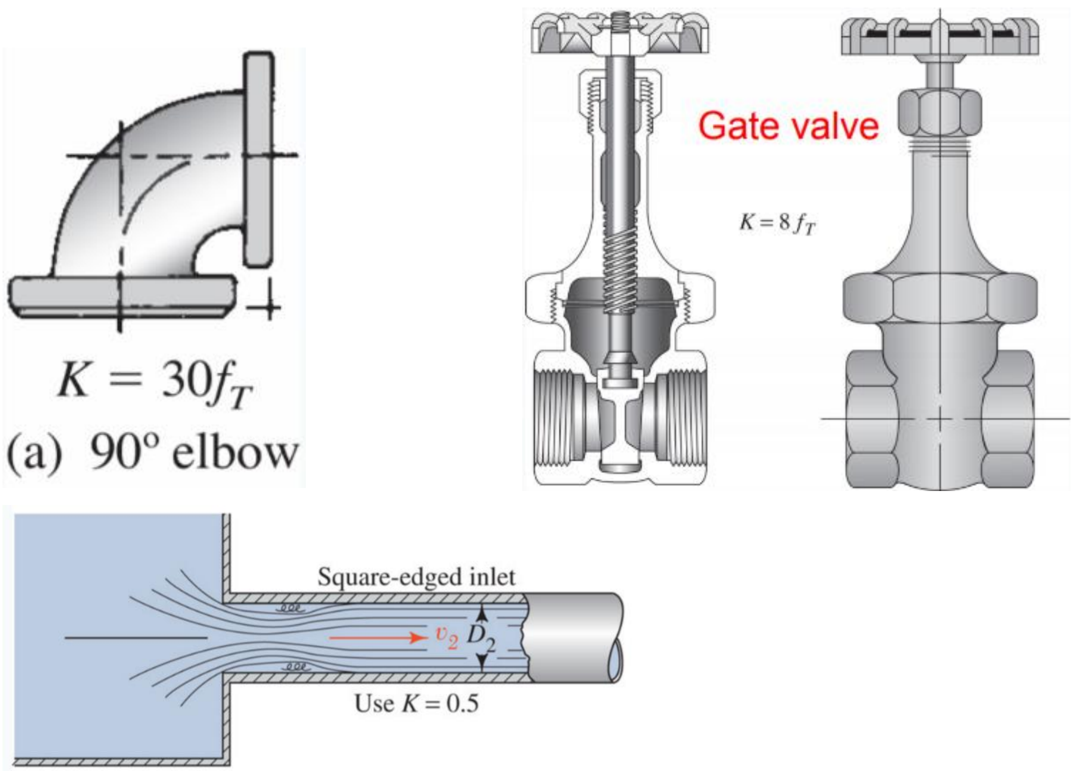
The pipe system that was tested was the longest system in the design, the system connecting the clean coolant tank to the reservoir tank. It is important to note that an iterative process was chosen as the length of the span varies directly with the weight of the pipe as the weight is dependent on the span length. In addition, a factor of safety of 1.25 was applied to the system in the event of buildup of metal shavings, sediment, or other debris in the piping. As a result, a standard strut-style rooftop pipe support would be sufficient for carrying the load of the pipe filled with coolant. The span between supports should be recomputed if specific weight, pipe diameter, or pipe material changes.

Energy losses

Purpose

The purpose of energy loss calculations is to determine the losses in the system for evaluation towards pump requirements.

Drawings and Diagrams



Losses: Elbow, Entrance, and Valve

Sources

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

Design Considerations

- Incompressible fluids
- Isothermal process
- Average temperature of 50°F
- Entrance losses
- Friction losses

- Elbow losses
- Losses between filler inlet and new tank are negligible
- Losses between drain outlet and old tank are negligible

Data and Variables

Relative roughness, D/ε = 0.000046m

Gravitational constant, g = 9.81m/s^2

Average velocity, v = 2.78 m/s

Procedure

To calculate losses, Reynold's number is calculated for the system. This result defines which equations to use to find a friction factor for the system. Once the friction factor is calculated, it can be applied to each type of fitting and valve in the system to calculate component losses. In addition to this, the losses due to pipe friction are also computed. From these calculations, we are able to determine the loss of each subsystem.

Calculations

$$Reynolds\ Number, Nr = \frac{vD}{v_k} = 2.78\frac{m}{s} * \frac{0.0409m}{\frac{0.000002685m^2}{s}} = 43389.83$$

$$Friction\ factor, f = \frac{0.25}{\left(\log\left(\frac{1}{3.7\left(\frac{D}{\varepsilon}\right)} + \frac{5.74}{Nr^{0.9}}\right)\right)^2} = 0.02475$$

$$Elbow\ Loss_{1-2} = K_{Elbow} * f * N_{Elbow} = 30 * 0.02475 * 3 = 2.228m = 7.310ft$$

$$Elbow\ Loss_{2-3} = K_{Elbow} * f * N_{Elbow} = 30 * 0.02475 * 4 = 2.971m = 9.747ft$$

$$Valve\ Loss = K_{Valve} * f * N_{Valve} = 8 * 0.02475 = 0.198m = 0.650ft$$

$$Entrance\ Loss = K_{Entrance} * \frac{v^2}{2g} = 0.5 * \frac{2.78^2\left(\frac{m}{s}\right)}{2 * 9.81} = 0.196m = 0.643ft$$

$$Pipe\ Friction_{Train-New} = f * \frac{L}{D} * \frac{v^2}{2g} = 0.02475 * \frac{5.47m}{0.041m} * \frac{2.78^2\left(\frac{m}{s}\right)}{2*9.81\left(\frac{m}{s^2}\right)} = 1.31m = 4.29ft$$

$$Pipe\ Friction_{New-Res} = f * \frac{L}{D} * \frac{v^2}{2g} = 0.02475 * \frac{192.63m}{0.041m} * \frac{2.78^2\left(\frac{m}{s}\right)}{2*9.81\left(\frac{m}{s^2}\right)} = 46.08m = 151.181ft$$

$$Pipe\ Friction_{Res-Old} = f * \frac{L}{D} * \frac{v^2}{2g} = 0.02475 * \frac{165.74m}{0.041m} * \frac{2.78^2\left(\frac{m}{s}\right)}{2*9.81\left(\frac{m}{s^2}\right)} = 40.67m = 133.432ft$$

$$Total\ Loss_{Train-New} = Pipe\ Friction + Elbow\ Loss_{1-2} + Valve\ Loss + Entrance\ Loss$$
$$= 1.31m + 1.49m + 0.2m + 0.2m = 3.19m = 10.48ft$$

$$Total\ Loss_{New-Res} = Pipe\ Friction + Elbow\ Loss_{1-2} + Valve\ Loss + Entrance\ Loss$$
$$= 46.08m + 2.228m + 0.198m + 0.196m = 48.70m = 159.79ft$$

$$Total\ Loss_{Res-Old} = Pipe\ Friction + Elbow\ Loss_{1-2} + Valve\ Loss + Entrance\ Loss$$
$$= 40.67m + 2.970m + 0.198m + 0.196m = 44.03m = 144.47ft$$

Summary

The calculations above are summarized in the following table by system.

	System 1	System 2	System 3	System 4	
Pipe Friction	1.31	46.08	40.67	0.10	m
Elbows	1.49	2.23	2.97	0.00	m
Valves	0.20	0.20	0.20	0.20	m
Entrances	0.20	0.20	0.20	0.20	m
Total Loss Per system	3.19	48.70	44.03	0.50	m
	10.48	159.79	144.47	1.64	ft

Losses and Loss Types for each System

Materials

- Steel tube, 1.5in diameter
- 90-Degree elbows (7)
- Gate Valve (2)
- Coolant

Analysis

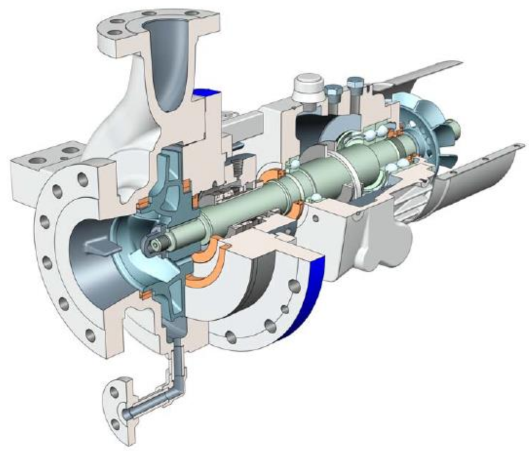
Upon calculating the Reynold's number for this system, we realized that the flow within the pipes will be turbulent, which results in higher losses. These losses will require more work done by the pump to transport the fluid. These losses are subject to change based on pipe material, diameter, fluid velocity, and length.

Pump selection

Purpose

To determine the minimum required power to transport fluid through the two subsystems.

Drawings and Diagrams



Centrifugal Pump with Cutaway

Sources

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

Design Considerations

- Incompressible fluids
- Isothermal process
- One pump for each subsystem

Data and Variables

- System 1 losses, $h_{L1\text{-}2}$ = 10.48 ft
- System 2 losses, $h_{L2\text{-}3}$ = 159.79 ft
- System 3 Losses, $h_{L3\text{-}4}$ = 144.47 ft
- Specific weight of coolant, γ = 58.656 lb/ft^3
- Flow rate, Q = .1292 ft^3/s

Procedure

The losses for the pump head will be calculated using Bernoulli's equation. Once these values are defined, the power for the pump can be computed.

Calculations

$$h_p + \frac{p_1}{\gamma} + \frac{v_1^2}{2g} + z_1 = \frac{p_2}{\gamma} + \frac{v_2^2}{2g} + z_2 + h_{l_{1\to 2}}$$

For system 1,

$$h_p + \frac{0}{\gamma} + \frac{\left(2.78\frac{m}{s}\right)^2}{2\left(9.81\frac{m}{s^2}\right)} + .9144\,m = \frac{0}{\gamma} + \frac{\left(2.78\frac{m}{s}\right)^2}{2\left(9.81\frac{m}{s^2}\right)} + 4.8768\,m + 3.19\,m$$

solving for pump head,

$$System\ 1\ Pump\ Head, h_{p1} = 4.8768\,m - .9144\,m + 3.19\,m = 7.156\,m$$

For system 2,

$$h_p + \frac{0}{\gamma} + \frac{\left(2.78\frac{m}{s}\right)^2}{2\left(9.81\frac{m}{s^2}\right)} + 4.8768\,m = \frac{0}{\gamma} + \frac{\left(2.78\frac{m}{s}\right)^2}{2\left(9.81\frac{m}{s^2}\right)} + 1.524\,m + \ 48.70\,m$$

$$System\ 2\ Pump\ Head, h_{p2} = 1.524\,m - 4.8768\,m + 48.70\,m \ = 46.265\,m$$

For system 3,

$$h_p + \frac{0}{\gamma} + \frac{\left(2.78\frac{m}{s}\right)^2}{2\left(9.81\frac{m}{s^2}\right)} + 1.524\,m = \frac{0}{\gamma} + \frac{\left(2.78\frac{m}{s}\right)^2}{2\left(9.81\frac{m}{s^2}\right)} + 6.7056\,m + \ 44.03\,m$$

$$System\ 3\ Pump\ Head, h_{p3} = \ 6.7.056\,m - 1.524\,m + 44.03\,m = 47.39\,m$$

For the pump power requirement,

Pump 1 Power, $P1 = \gamma * Q * h_{L1-2} = 58.656\left(\frac{lb}{ft^3}\right) * 0.13924\left(\frac{ft^3}{s}\right) * 23.48\,ft = 177.95\left(\frac{ft-lb}{s}\right) =$

$$0.32\,HP$$

$$Pump\ 2\ Power, P2 = \gamma * Q * h_{L1-2} = 58.656\left(\frac{lb}{ft^3}\right) * 0.13924\left(\frac{ft^3}{s}\right) * 151.79\ ft = 1438.494\left(\frac{ft-lb}{s}\right)$$

$$= 2.09\ HP$$

$$Pump\ 3\ Power, P3 = \gamma * Q * h_{L2-3} = 58.656\left(\frac{lb}{ft^3}\right) * 0.13924\left(\frac{ft^3}{s}\right) * 155.47 = 1178.41\left(\frac{ft-lb}{s}\right)$$

$$= 2.14\ HP$$

Summary

The pump head and pump power requirements can be summarized by the table below:

Pump Head -- System 1	23.48	ft	z2-z1+hl = hp
Pump Head -- System 2	151.79	ft	z2-z1+hl = hp
Pump Head -- System 3	155.47	ft	z2-z1+hl = hp
Pump 1 Power	177.95	ft-lb/sec	
	0.32	HP	
	0.24	kW	
Pump 2 Power	1150.56	ft-lb/sec	
	2.09	HP	
	1.56	kW	
Pump 3 Power	1178.41	ft-lb/sec	
	2.14	HP	
	1.60	kW	

Materials

- Pump
- Coolant

Analysis

Because it is impossible to purchase pump motors that output the exact power requirements as stated above, a standardized motor needs to be selected. For the first system, a .5 HP pump motor is required. For the second system, a 2.5 HP pump motor is required. For the third system, a 2.5 HP pump motor is also required.

Selection of Pump Type

Purpose

To select the pump type for the required systems that need to be pumped.

Drawings and Diagrams

Centrifugal and Positive-Displacement Pumps

Sources

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

Liu, Ben, and Ram Limjoco. "Comparison between the 2 Most Used Pump Types | All Pumps." All Pumps Blog, All Pumps, 12 Jan. 2018, www.allpumps.com.au/blog/2015/05/08/comparison-between-the-2-most-used-pump-types-centrifugal-pumps-vs-positive-displacement-pumps/.

Design Considerations

- Isothermal process
- Incompressible fluid
- Steady state
- One pump per subsystem

Data and Variables

- Pump 1 Head, h_{A1} = 23.48 ft
- Pump 2 Head, h_{A2} = 151.79 ft

- Pump 3 Head, h_{A3} = 155.47 ft
- System 1 Flow Rate, Q_1 = 52 gpm
- System 2 Flow Rate, Q_2 = 58 gallons/min
- System 3 Flow Rate, Q_3 = 57 gallons/min
- Reynolds Number for System 1, N_1 = 1760 rev/min
- Reynolds Number for System 2 and 3, $N_2 = N_3$ = 3520 rev/min
- D = 1.61 in

Procedure

To determine the type of pump needed for the systems, the flow rate, motor speed, diameter and pump head are needed. The pumps selected are kinetic pumps. To verify that the centrifugal pumps needed are correct, the type of flow needs to be checked. To do this, specific speed and specific diameter for each system need to be calculated. After calculating the specific speed and diameter, the numbers can be read off the chart in Figure 13.53.

Calculations

For system 1,

$$Specific\ Speed, N_S = \frac{N\sqrt{Q}}{H^{\frac{3}{4}}} = \frac{1760\frac{rev}{min} * \sqrt{52\frac{gal}{min}}}{(23.48\ ft)^{\frac{3}{4}}} = 1189.85\ (Unitless)$$

$$Specific\ Diameter, D_S = \frac{DH^{\frac{1}{4}}}{\sqrt{Q}} = \frac{1.61\ in * (23.48\ ft)^{\frac{1}{4}}}{\sqrt{52\frac{gal}{min}}} = 0.49\ (Unitless)$$

For system 2,

$$Specific\ Speed, N_S = \frac{3520\frac{rev}{min} * \sqrt{58\frac{gal}{min}}}{(151.79\ ft)^{\frac{3}{4}}} = 619.9\ (Unitless)$$

$$Specific\ Diameter, D_S = \frac{1.61\ in * (151.79\ ft)^{\frac{1}{4}}}{\sqrt{58\frac{gal}{min}}} = 0.74\ (Unitless)$$

For system 3,

$$Specific\ Speed, N_S = \frac{3520\frac{rev}{min} * \sqrt{57\frac{gal}{min}}}{(155.47\ ft)^{\frac{3}{4}}} = 603.6\ (Unitless)$$

$$N_S = 603.6$$

$$Specific\ Diameter, D_S = \frac{1.61\ in * (155.47\ ft)^{\frac{1}{4}}}{\sqrt{57\frac{gal}{min}}} = 0.75\ (Unitless)$$

Materials

- Pump
- Coolant

Summary

For all systems, each pump is in the radial flow region, according to the chart in Figure 13.54.

Analysis

The pumps chosen are kinetic pumps instead of positive displacement because the fluid has a relatively low viscosity and there is not a precise flow rate requirement for any system. The pumps are all centrifugal because each pump has a radial flow for the fluid. This can be seen from Figure 13.53. If the flow rates or diameter of the pipe are changed at a later date, these calculations will vary with those changes. These calculations will need to be repeated to ensure that the initial calculations are correct in determining the pump type for each subsystem.

Pump & System Curves

Purpose

The purpose of this analysis is to specify the characteristics of the chosen pumps, points of operation, and actual pump size.

Drawings and Diagrams

Sulzer Pump

Sources

- Mott, R., Untener, J., "Applied Fluid Mechanics," 7th edition Pearson Education, Inc., (2015)
- VanAire: Sulzer Pumps, Van Aire Inc, www.vanaireinc.com/daf/SulzerAPT222BPump.

Design Considerations

- Isothermal process
- Incompressible fluid
- Steady state

Data and Variables

- Pump 1 Head, h_{A1} = 23.48 ft
- Pump 2 Head, h_{A2} = 151.79 ft
- Pump 3 Head, h_{A3} = 155.47 ft
- System 1 Flow Rate, Q_1 = 52 gallons/min
- System 2 Flow Rate, Q_2 = 58 gallons/min
- System 3 Flow Rate, Q_3 = 58 gallons/min

Procedure

First, the pump must be selected from the given pump head and flow rate. After the pump is found from using Chart 1, the system curve needs to be drawn on the pump curve. For the selected pump, the system curve can be found by substituting different flow rates into Bernoulli's equation to find the different pump head at the specified flow rate. Using zero flow rate, half the original flow rate, and the original flow rate, the system curve can be drawn. The point of operation will lie on the system curve where the impeller diameter and system curve intersect. This procedure is repeated for the other two systems.

Calculations

For initial flow rate,

		System 1	System 2	System 3	System 4	Sum	
Pipe Lengths	length of pipe (ft)	18	632	557.75	5	1212.75	ft
	length of pipe (m)	5.486	192.634	170.002	1.524	369.6462	m
	number of elbows	2	3	4	0	9	Elbows
	number of gate valves	1	1	1	1	4	Valves
	number of entrances	1	1	1	1	4	Entrances

Characteristics (1.5" Pipe)	diameter of pipe	0.0409	m
	avg velocity	2.78	m/s
	friction factor	0.02	unitless
	gravitational constant	9.81	m/s^2
	kinematic viscosity	2.63E-06	m^2/s
	reynolds number	43389.83	unitless
	Roughness	4.60E-05	m
	D/e	889.13	unitless
Characteristics (2.0" Pipe)	diameter of pipe	0.0525	m
	avg velocity	1.69	m/s
	friction factor	0.02	unitless
	gravitational constant	9.81	m/s^2
	kinematic viscosity	2.63E-06	m^2/s
	reynolds number	33756.70	unitless
	Roughness	4.60E-05	m
	D/e	1141.30	unitless

		System 1	System 2	System 3	System 4	
Minor Losses	Pipe Friction	1.31	46.08	40.67	0.10	m
	Elbows	1.49	2.23	2.97	0.00	m
	Valves	0.20	0.20	0.20	0.20	m
	Entrances	0.20	0.20	0.20	0.20	m
	Total Loss Per system	3.19	48.70	44.03	0.50	m
		10.48	159.79	144.47	1.64	ft

For half of the flow rate,

	System 1	System 2	System 3	System 4	Sum	
length of pipe (ft)	18	632	557.75	5	1212.75	ft
length of pipe (m)	5.486	192.634	170.002	1.524	369.6462	m
number of elbows	2	3	4	0	9	Elbows
number of gate valves	1	1	1	1	4	Valves
number of entrances	1	1	1	1	4	Entrances

diameter of pipe	0.0409	m
avg velocity	1.39	m/s
friction factor	0.02	unitless
gravitational constant	9.81	m/s^2
kinematic viscosity	2.63E-06	m^2/s
reynolds number	21694.92	unitless
Roughness	4.60E-05	m
D/e	889.13	unitless
diameter of pipe	0.0525	m
avg velocity	1.69	m/s
friction factor	0.02	unitless
gravitational constant	9.81	m/s^2
kinematic viscosity	2.63E-06	m^2/s
reynolds number	33756.70	unitless
Roughness	4.60E-05	m
D/e	1141.30	unitless

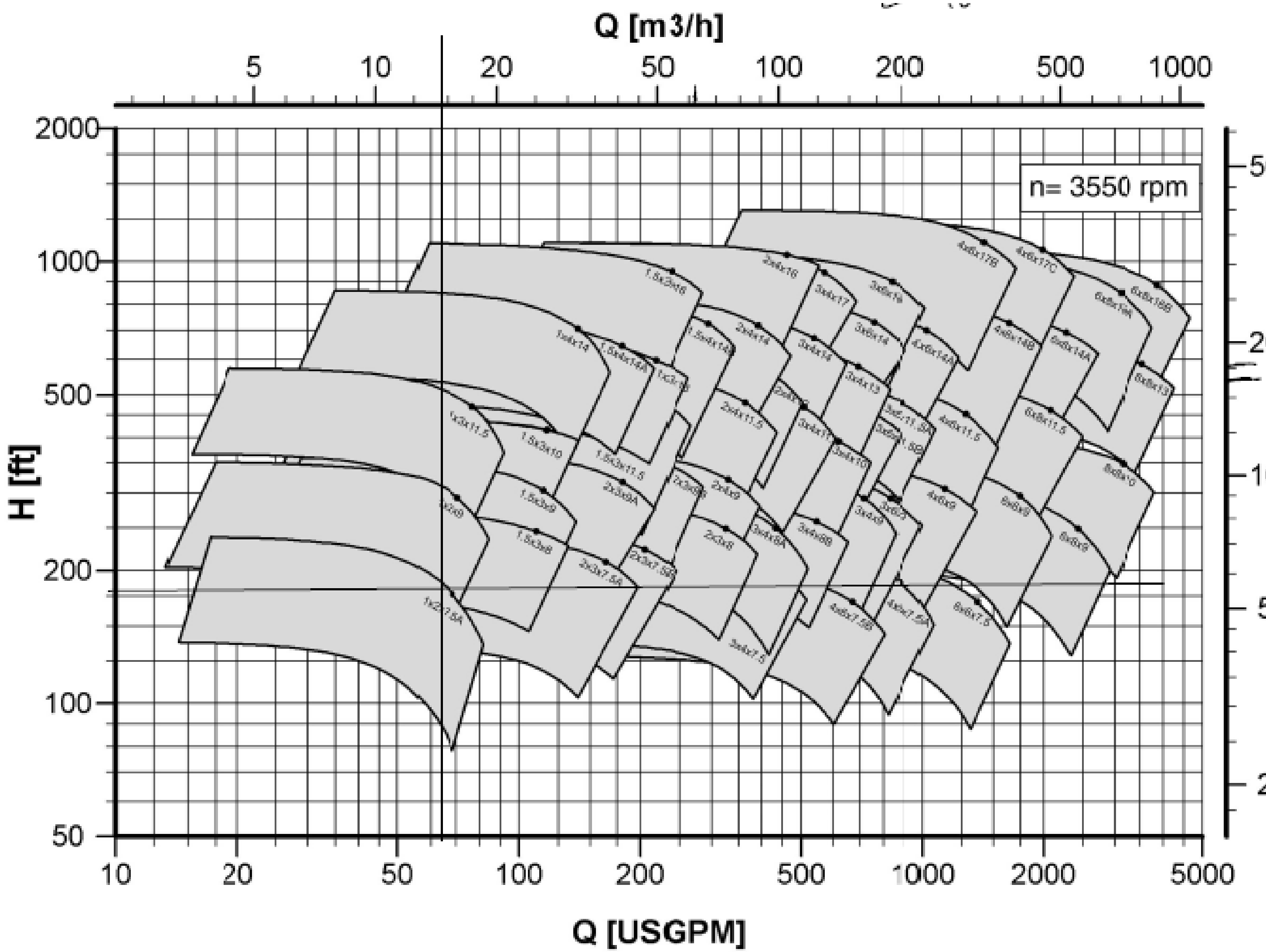
	System 1	System 2	System 3	System 4	
Pipe Friction	0.33	11.52	10.17	0.10	m
Elbows	1.49	2.23	2.97	0.00	m
Valves	0.20	0.20	0.20	0.20	m
Entrances	0.05	0.05	0.05	0.05	m
Total Loss Per system	2.06	14.00	13.38	0.35	m
	6.76	45.92	43.91	1.15	ft

For no flow rate,

	System 1	System 2	System 3	System 4	Sum	
length of pipe (ft)	18	632	557.75	5	1212.75	ft
length of pipe (m)	5.486	192.634	170.002	1.524	369.6462	m
number of elbows	2	3	4	0	9	Elbows
number of gate valves	1	1	1	1	4	Valves
number of entrances	1	1	1	1	4	Entrances

diameter of pipe	0.0409	m
avg velocity	0.00	m/s
friction factor	0.02	unitless
gravitational constant	9.81	m/s^2
kinematic viscosity	2.63E-06	m^2/s
reynolds number	0.00	unitless
Roughness	4.60E-05	m
D/e	889.13	unitless
diameter of pipe	0.0525	m
avg velocity	1.69	m/s
friction factor	0.02	unitless
gravitational constant	9.81	m/s^2
kinematic viscosity	2.63E-06	m^2/s
reynolds number	33756.70	unitless
Roughness	4.60E-05	m
D/e	1141.30	unitless

	System 1	System 2	System 3	System 4	
Pipe Friction	0.00	0.00	0.00	0.10	m
Elbows	1.49	2.23	2.97	0.00	m
Valves	0.20	0.20	0.20	0.20	m
Entrances	0.00	0.00	0.00	0.00	m
Total Loss Per system	1.68	2.43	3.17	0.30	m
	5.52	7.96	10.39	0.99	ft



SULZER
CURVES - TYPE OHH
Range of Performances – 60 Hertz 3550 rpm

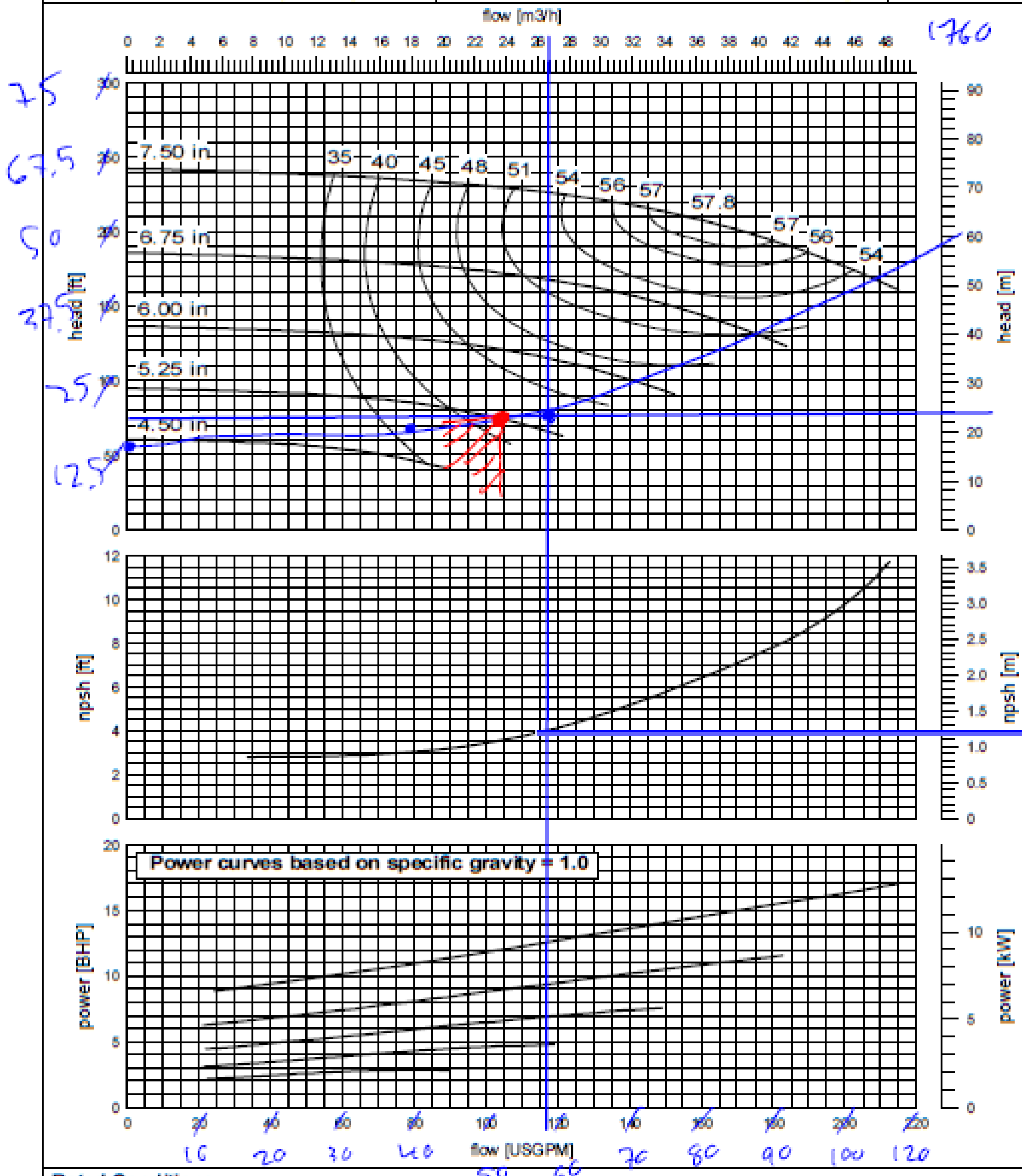
Series **2.00**	Page
CURVES	60hz
12 February 2002	**E0D**

2 x 3 x 7.5A -1 OHH

SULZER

Series 2.00 / 60Hz E002

Curve No	OHH 52-1-1-03	NSS	11000 (213)	Speed
Efficiency Basis	API Std. clearances	NS	847 (16)	~~3520~~ rpm
Max Solid	.79 in (20 mm)	Rotation	CCW viewed from coupling	1760



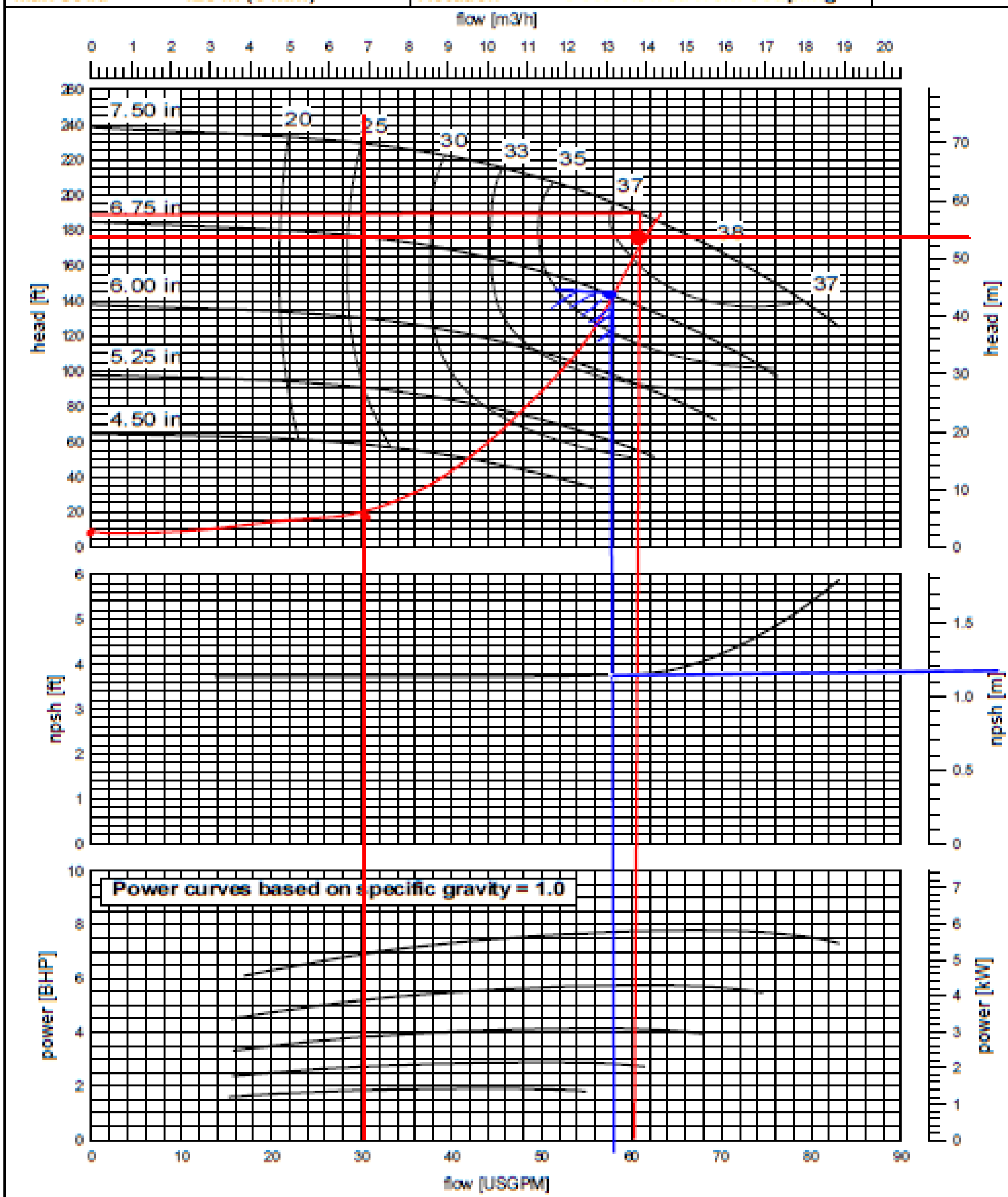
Rated Conditions		
Project	Item	
H =	Q =	P =
Calculated Efficiency =		NPSH$_{3\%}$ =

Issued: MSC / 08.28.2002

1 x 2 x 7.5 -1 OHH

SULZER
Series 2.00 / 60Hz E001

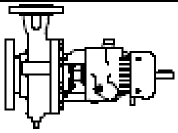
Curve No	OHH 53-1-1-02	NSS	10190 (197)	Speed 3520 rpm
Efficiency Basis	API Std. clearances	NS	560 (11)	
Max Solid	.20 in (5 mm)	Rotation	CCW viewed from coupling	



Rated Conditions

Project	Item	
H =	Q =	P =
Calculated Efficiency =		$NPSH_{3\%}$ =

Issued: MSC /05.13.2002

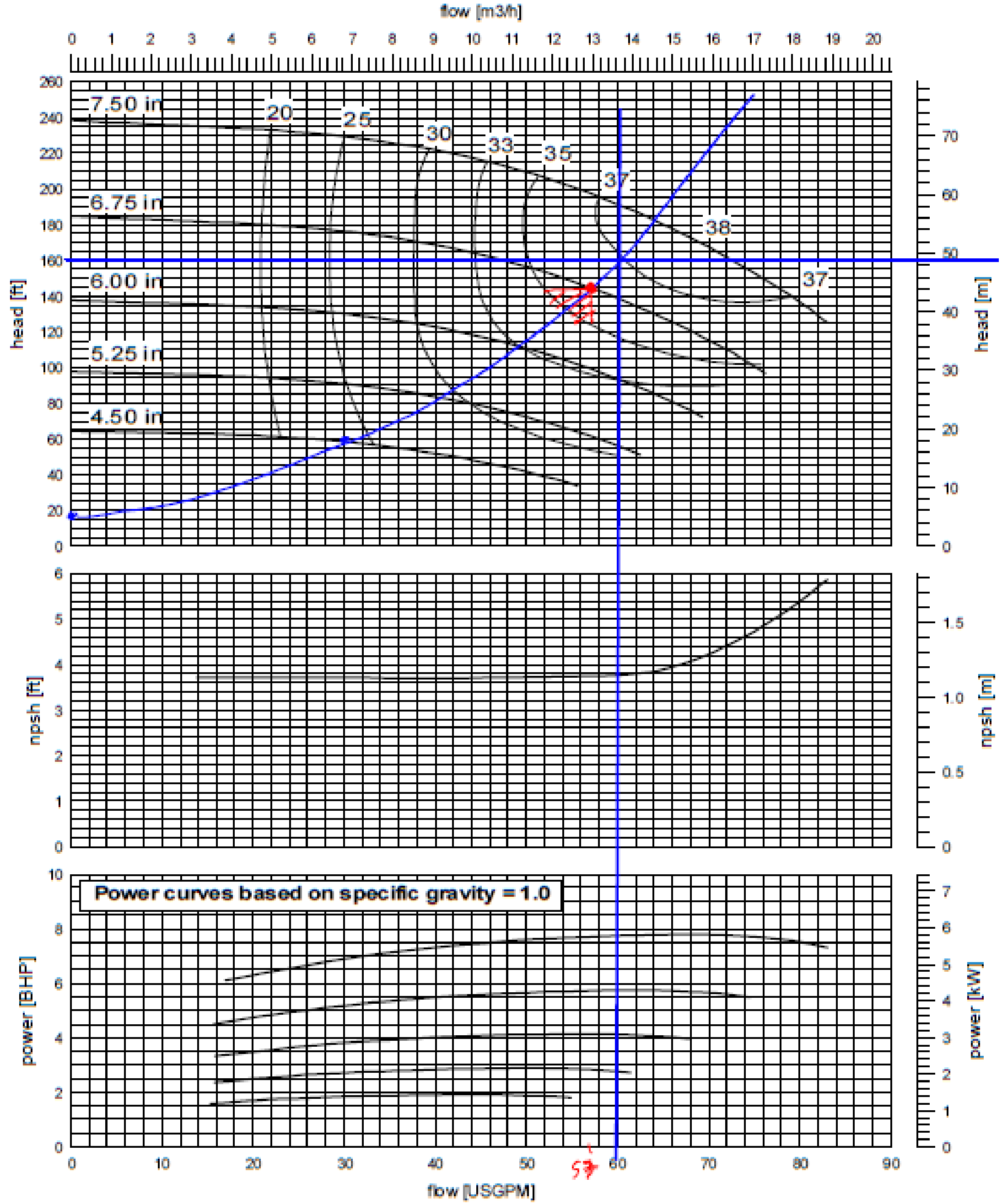


1 x 2 x 7.5 -1 OHH

SULZER

Series 2.00 / 60Hz E001

Curve No	OHH 53-1-1-02	NSS	10190 (197)	Speed
Efficiency Basis	API Std. clearances	NS	560 (11)	3520
Max Solid	.20 in (5 mm)	Rotation	CCW viewed from coupling	rpm



Rated Conditions		
Project	Item	
H =	Q =	P =
Calculated Efficiency =		$NPSH_{3\%}$ =
		Issued: MSC /05.13.2002

Materials

- Pump
- Coolant

Summary

For the first system, a 2 x 3 x 7.5A pump is used with a 5.25 in diameter impeller running at 1760 rpm and 52 gallons per minute. For the second system, a 1 x 2 x 7.5 pump is used with a 6.75 in diameter impeller running at 3520 rpm and 58 gallons per minute. For the third system, a 1 x 2 x 7.5 pump is used with a 6.75 in diameter impeller running at 3520 rpm and 57 gallons per minute.

Analysis

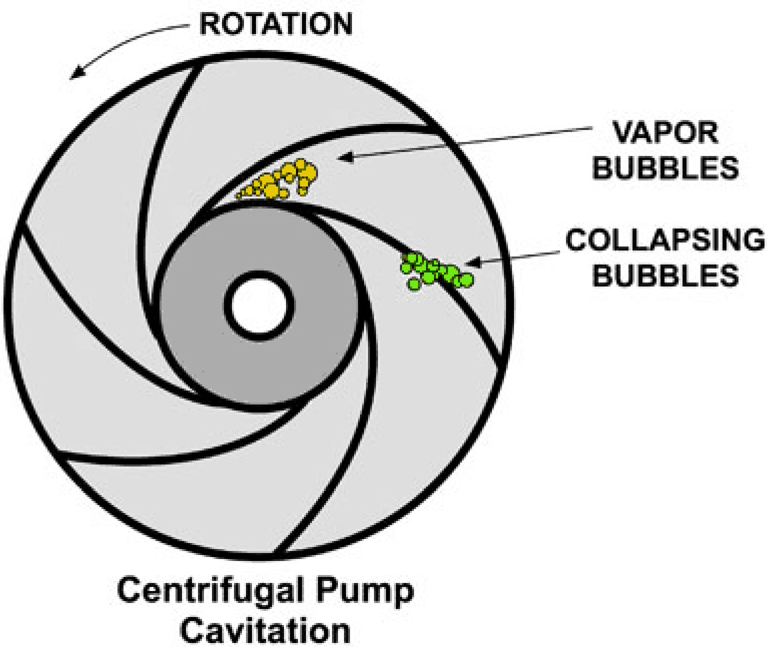
For each system, the point of operation was chosen to maximize the upgradability of pump. A larger diameter impeller can be chosen for each of the systems to help increase flow rate. The operation points also take into consideration that as the pump ages, the efficiency will decrease with time, unless on the right-hand side of the pump curve. The points of operation were selected to maximize the efficiency for the life of the pump. These pump curves were derived from the specific system created in this report. If any changes are made to the pipe layout or the flow rate of the tank, the calculations for pump head will change, ensuring a change in the system curve. These calculations will need to be repeated if any change to the system occurs beyond the received date of the report.

Cavitation

Purpose

The purpose of testing for cavitation is to determine whether the given systems create excessive amounts of air bubbles that could damage pump impellers.

Drawings and Diagrams



Sources

- Mott, R., Untener, J., "Applied Fluid Mechanics," 7th edition Pearson Education, Inc., (2015)
- "Cavitation in Centrifugal Pumps." Nuclear Power, Nuclear Power for Everybody, www.nuclear-power.net/nuclear-engineering/fluid-dynamics/centrifugal-pumps/cavitation/.

Design Considerations

- Isothermal process
- Incompressible fluid
- Steady state

Data and Variables

- Pressure at start of the system, P_1 = 101.3 kPa
- Gamma of water, γ = 9.2214 kN/m^3
- Change in elevation of the system, Δz = 10 ft = 3.048 m
- Pressure at the valve, P_V = .9158 kPa

Procedure

For each pumped system, the NPSH available needs to be calculated. To do this, the energy losses for each system from the selected points needs to be calculated. Once this is calculated, the values for pressure, specific weight, change in height and vapor pressure can be substituted into the equation. This will give a value for the NPSH available to each system. Then the graph for NPSH from the pump curve for each system will need to be checked to confirm that the available NPSH is greater than the NPSH from the pump curve.

Calculations

For system 1,

$$Pump\ 1\ Head, h_{A\ 1-S} = \left(\Sigma K + \frac{fL}{D}\right)\frac{V^2}{2g} = \left(K_{entrance} + \frac{fL}{D}\right)\frac{V^2}{2g} = 0.27912\ m$$

Substituting known values into the NPSH formula,

$$NPSH_{avail} = \frac{P_1}{\gamma} + \Delta z - h_{L\ 1-S} - \frac{P_v}{\gamma} = \frac{101.3\ kPa}{9.2214\frac{kN}{m^3}} + 3.048\ m - .27912\ m - \frac{.91575\ kPa}{9.2214\frac{kN}{m^3}}$$
$$= 13.65\ m$$

For the second system,

$$Pump\ 2\ Head, h_{A\ 1-S} = \left(\Sigma K + \frac{fL}{D}\right)\frac{V^2}{2g} = \left(K_{entrance} + \frac{fL}{D}\right)\frac{V^2}{2g} = 0.345041\ m$$

Substituting into the NPSH available formula,

$$NPSH_{avail} = \frac{P_1}{\gamma} + \Delta z - h_{L\,1-S} - \frac{P_v}{\gamma} = \frac{101.3\ kPa}{9.2214\,\frac{kN}{m^3}} + 4.877\ m - .345041\ m - \frac{.91575\ kPa}{9.2214\,\frac{kN}{m^3}}$$

$$= 15.42\ m$$

For the third system,

$$h_{A\,1-S} = \left(\Sigma K + \frac{fL}{D}\right)\frac{V^2}{2g} = \left(K_{entrance} + \frac{fL}{D}\right)\frac{V^2}{2g} = 0.33589\ m$$

Substituting known values into the NPSH formula,

$$NPSH_{avail} = \frac{P_1}{\gamma} + \Delta z - h_{L\,1-S} - \frac{P_v}{\gamma} = \frac{101.3\ kPa}{9.2214\,\frac{kN}{m^3}} + 1.524\ m - .33589\ m - \frac{.91575\ kPa}{9.2214\,\frac{kN}{m^3}}$$

$$= 12.074\ m$$

Materials

- Pump
- Coolant

Summary

After checking the NPSH available against the graphs given from the pump curves, the NPSH available is greater than the chart value.

Analysis

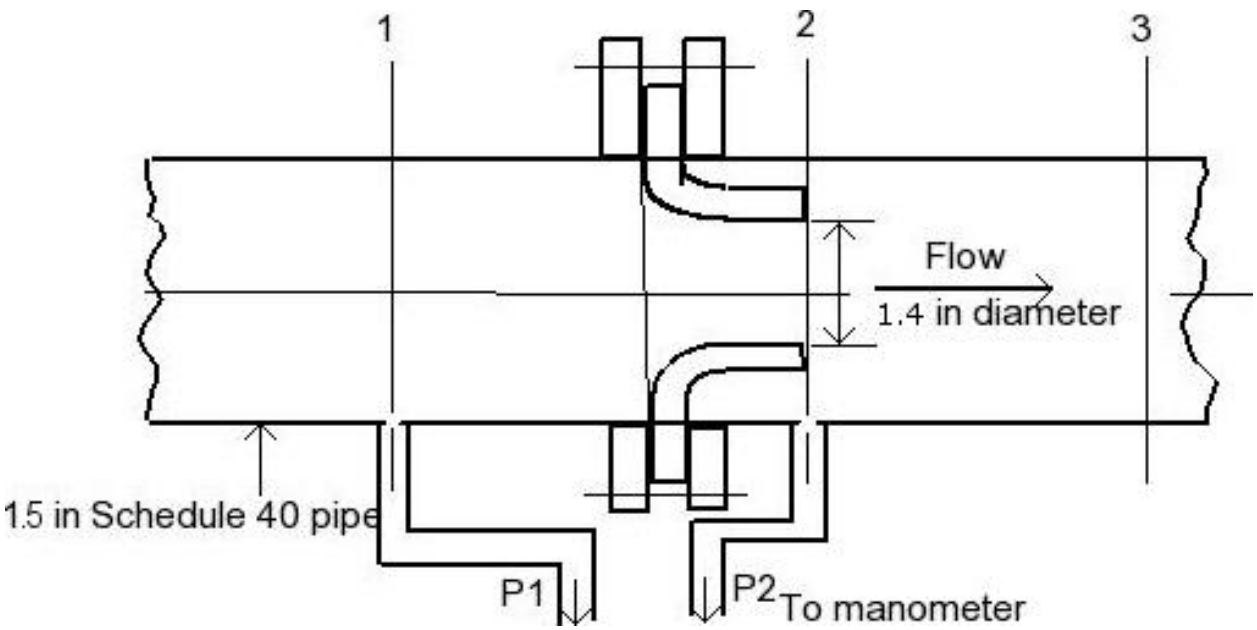
Since none of the NPSH available values were less than the NPSH from the pump curves, the systems will not cavitate. The values are also very large, meaning that even with changes in the system, the pumps will not be likely to cavitate. The calculations for NPSH are dependent entirely on the pumps and the pump head derived from previous calculations. If the pump head, desired flow rate or pipe layout is changed, these calculations for NPSH will be invalid. New computations for the changed system would be critical to ensure the safety of the system and prevent cavitation.

Instrumentation

Purpose

The purpose of instrumentation is to pick the correct instruments for measuring flow rate and pressure differential in one of the systems.

Drawings and Diagrams



Sources

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

Design Considerations

- Incompressible fluids
- Steady state
- Isothermal process
- Temperature is at 32 °F
- Mercury manometer used to find the pressure difference across the nozzle
- Q_{max} is at maximum flow rate of pump
- Q_{min} is at operating flow rate of pump

Data and Variables

- Diameter of the flow nozzle, D_{nozzle} = 1.4 inches = 0.1667 ft
- Diameter of the pipe, D_{pipe} = 1.5-inch nominal = 0.1342 feet
- Area of the flow nozzle, A_{nozzle} = 1.069 x 10^-2 ft^2
- Area of the pipe, A_{pipe} = 1.414 x 10^-2 ft^2
- Temperature, T = 32 °F
- Minimum flow rate, Q_{min} = 42 gallons/minute = 0.0936 ft^3/s
- Maximum flow rate, Q_{max} = 82 gallons/minute = 0.1827 ft^3/s
- Gamma of water, γ_{water} = 58.656 lb/ft^3
- Gamma of mercury, $\gamma_{mercury}$ = 58 lb/ft^3
- Pipe to nozzle ratio, β = 1.4/1.5 = 0.93
- Kinematic viscosity, $V_{kinematic}$ = 1.7766 x 10^-5 ft^2/s

Procedure

A nozzle size of 1.4 inches was picked to not greatly disturb the flow of the 1.5-inch nominal pipe. The maximum velocity was found using the maximum flow rate possible for the pump selected. The minimum velocity was chosen using the operating flow rate of the pump in the system. From this, maximum and minimum discharge coefficients were found. From this point on, both the minimum velocity and discharge coefficient and maximum velocity and discharge coefficient were used to find the minimum height on the scale and the maximum height on the manometer scale respectively.

Calculations

With the maximum and minimum flow rates of the pump, the respective velocities are calculated. The minimum velocity is found:

$$Minimum\ velocity, V_{min} = \frac{Q_{min}}{A_{pipe}} = \frac{0.936\ \frac{ft^3}{s}}{0.01414\ ft^2} = 6.62 ft/s$$

Followed by the maximum velocity:

$$Maximum\ velocity, V_{max} = \frac{Q_{max}}{A_{pipe}} = \frac{0.183\ \frac{ft^3}{s}}{0.01414\ ft^2} = 12.94 ft/s$$

A calculation for Reynolds Number is then done:

$$Minimum\ Reynolds\ Number, Re_{min} = \frac{V_{min}D}{V_{kinematic}} = \frac{6.62\ \frac{ft}{s} \times 0.1342\ ft}{(1.89\ \times\ 10^{-5}\ \frac{ft^2}{s})(0.94)} = 50005.9$$

$$Maximum\ Reynolds\ Number, Re_{max} = \frac{V_{max}D}{V_{kinematic}} = \frac{12.94\ \frac{ft}{s} \times 0.1342\ ft}{(1.89\ \times\ 10^{-5}\ \frac{ft^2}{s})(0.94)} = 97745.6$$

$$Minimum\ flow\ nozzle\ coefficient, C_{min}\ =\ 0.9975 - 6.53\sqrt{\frac{\beta}{Re_{min}}} = 0.969$$

$$Maximum\ flow\ nozzle\ coefficient, C_{max}\ =\ 0.9975 - 6.53\sqrt{\frac{\beta}{Re_{max}}} = 0.977$$

To find the maximum and minimum values of the manometer heights, C_{min} and C_{max} are plugged into the follow equations:

$$Minimum\ velocity, V_{min} = C_{min}\sqrt{\frac{2gh_{min}[\frac{\gamma_{mercury}}{\gamma_{water}} - 1]}{(\frac{A_{pipe}}{A_{nozzle}})^2 - 1}}$$

$$Maximum\ velocity, V_{max} = C_{max}\sqrt{\frac{2gh_{max}[\frac{\gamma_{mercury}}{\gamma_{water}} - 1]}{(\frac{A_{pipe}}{A_{nozzle}})^2 - 1}}$$

Using the iterative process to solve for “h”, the maximum and minimum values for our manometer scale are found to be:

Manometer Minumum Height				Manometer Maximum Height			
Hmin	Vmin	RHS	Error	Hmax	Vmax	RHS	Error
1	6.62	34.0689485	414.64%	0.2	12.94	15.3286931	18.46%
0.5	6.62	24.0773822	263.71%	0.19	12.94	14.9384311	15.44%
0.1	6.62	10.7211006	61.95%	0.18	12.94	14.5376963	12.35%
0.05	6.62	7.53954289	13.89%	0.17	12.94	14.1255975	9.16%
0.02	6.62	4.68895851	-29.17%	0.1	12.94	10.8096133	-16.46%
0.03	6.62	5.79705013	-12.43%	0.15	12.94	13.2630421	2.50%
0.04	6.62	6.72497202	1.59%	0.155	12.94	13.4838549	4.20%
0.041	6.62	6.81081452	2.88%	0.145	12.94	13.0384904	0.76%
0.0405	6.62	6.76802937	2.24%	0.144	12.94	12.9931144	0.41%
0.039	6.62	6.63801949	0.27%	0.1425	12.94	12.9247516	-0.12%
0.0391	6.62	6.64676593	0.40%	0.143	12.94	12.9475793	0.06%
Hmin	0.468 in			Hmax	1.716 in		

$$Minimum\ height, h_{min} = 0.468\ inches$$

$$Maximum\ height, h_{max} = 1.742\ inches$$

<u>Summary</u>

The correct diameter dimension for the flow nozzle was found to be 1.4 inches. As a result, the proper scale was found to have a minimum value of 0.468 inches and a maximum value of 1.74 inches.

<u>Materials</u>

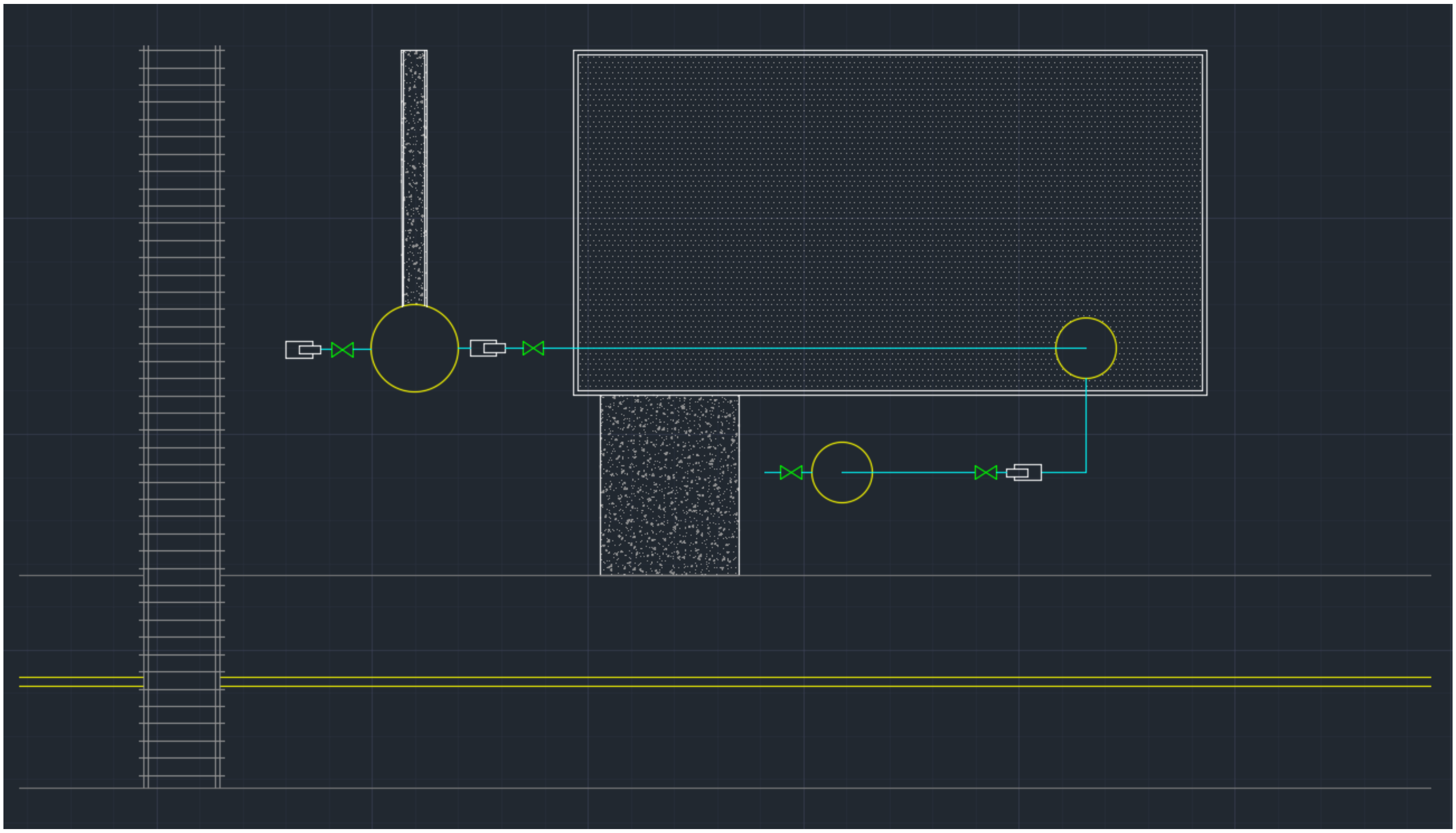
- Flow nozzle with a manometer filled with mercury for scale
- Bourdon pressure gauge with a scale from 0-2 bar (0-30psi)

Analysis

The system picked for flow measurements was the one linking the coolant reservoir to the dirty coolant tank. This system was selected so that plant workers could monitor the pipe flow rate and pressure difference for buildup of sediments, machining chips, and other debris. Should there be a sudden drop or increase in pressure, that would be a good warning to plant managers that there is a blockage in the pipe or a malfunction with the pump. A flow nozzle was selected over an orifice plate because the orifice plate would have incurred a greater frictional energy loss over a flow nozzle. A Venturi tube would have had the smallest energy loss, but the price and complexity of those meters was deemed outweigh the benefits. In addition, a Bourdon-style pressure gauge with a range from zero to 30 psi should be added to the system to test pressure at a particular point. A design change in instrumentation, whether that be a switch to a Venturi tube or orifice plate, would affect the energy losses of the system going from the coolant reservoir to the pump for the dirty coolant tank.

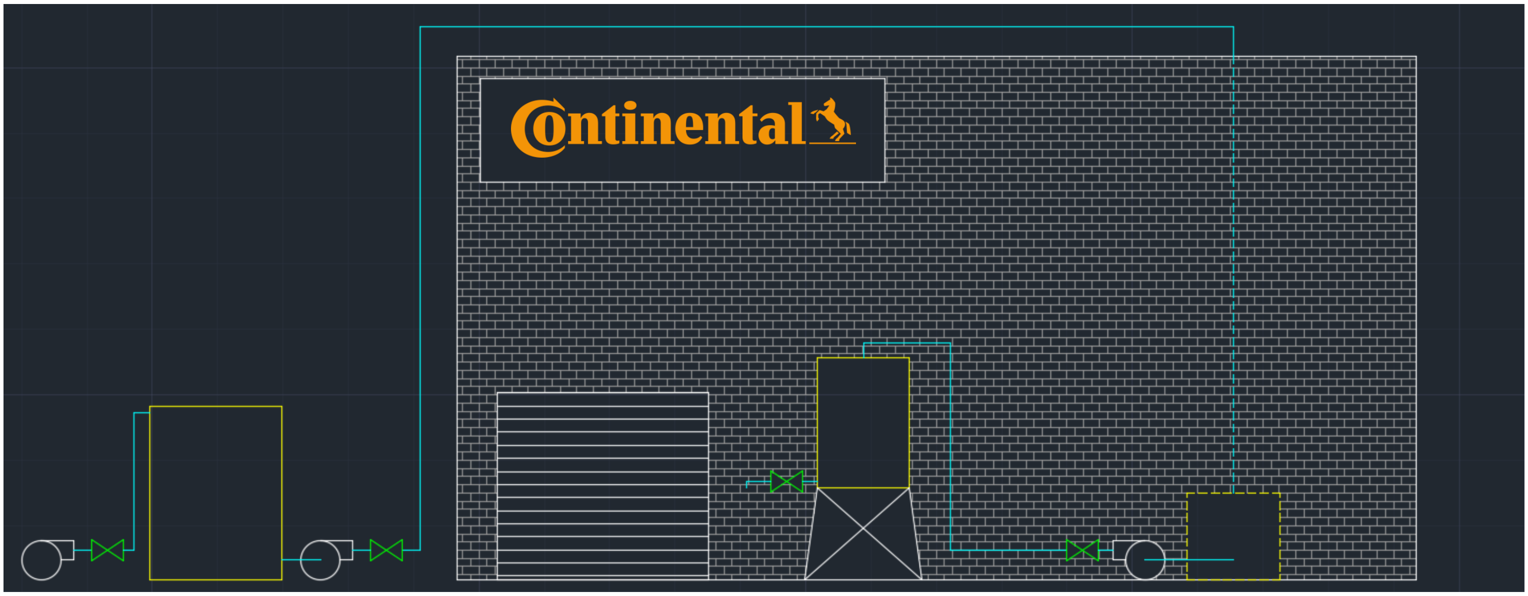
Final drawings

A. Plot Plan



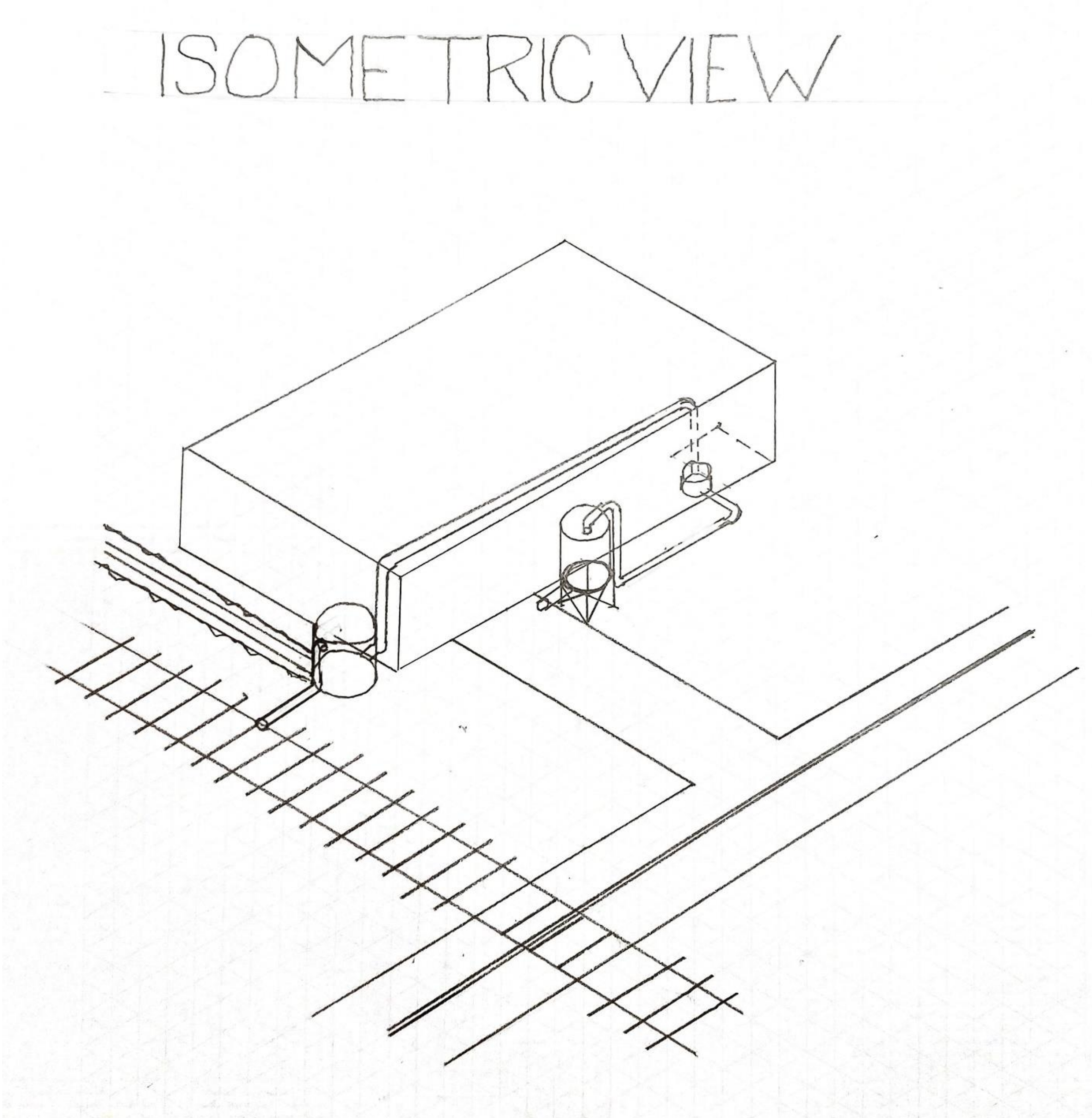
Plot Plan with Road and Railway

B. Elevations View



Elevations View with Tanks and Pipelines

C. Isometric View



Isometric View with Tanks, Pipelines, Road, and Railway

Bill of Materials and Equipment

Bill of Materials		
1 1/2 in Sch 40 steel pipe	**1207.75**	ft
2 in Sch 40 steel pipe	**5**	ft
14 ft D x 16 ft H HDPE tank	**3**	each
10.5 ft D x 8 ft H HDPE tank	**1**	each
5 ft D x 6 ft H HDPE tank	**1**	each
Steel elbows	**9**	each
Steel gate valves	**4**	each
rooftop pipe supports	**48**	each
ground level supports	**45**	each
1 x 2 x 7.5 Sulzer pump	**2**	each
2 x 3 x 7.5 Sulzer pump	**1**	each
Pipe heater	**1**	each
pipe support stands	**48**	each
pump motors	**3**	each
pressure gage	**1**	each
manometer	**1**	each
flow nozzle	**1**	each

Final Remarks

The design for the coolant delivery pipe system is very complex. There are several factors that contribute to the design of the pipe systems such as flow rate, the number of fittings, valves and gages, the number of pumps and materials used for construction. The system is designed for a flow rate of 58 gallons per minute to fill the large storage tank in an allotted time that would not take more than one working shift to complete. The tanks and piping were designed around this flow rate and design constraint. The pipe layout is designed to cover the least amount of space on the ground and minimize the distance between the tanks. This design helps minimize the cost for pipe materials and fittings. There are three pumped systems required for this design. The pumps for each system were systematically selected to adequately move coolant from each tank and transfer it to the corresponding secondary tank. The pumps are also designed to handle expansion in the future and can handle greater or less flow rate. They are also designed to maximize the efficiency for the lifetime of the pump.

Appendix

TABLE 8.2 Pipe roughness—design values

Material	Roughness ε (m)	Roughness ε (ft)
Glass	Smooth	Smooth
Plastic	3.0×10^{-7}	1.0×10^{-6}
Drawn tubing; copper, brass, steel	1.5×10^{-6}	5.0×10^{-6}
Steel, commercial or welded	4.6×10^{-5}	1.5×10^{-4}
Galvanized iron	1.5×10^{-4}	5.0×10^{-4}
Ductile iron—coated	1.2×10^{-4}	4.0×10^{-4}
Ductile iron—uncoated	2.4×10^{-4}	8.0×10^{-4}
Concrete, well made	1.2×10^{-4}	4.0×10^{-4}
Riveted steel	1.8×10^{-3}	6.0×10^{-3}

Table 8.2 - Pipe roughness chart

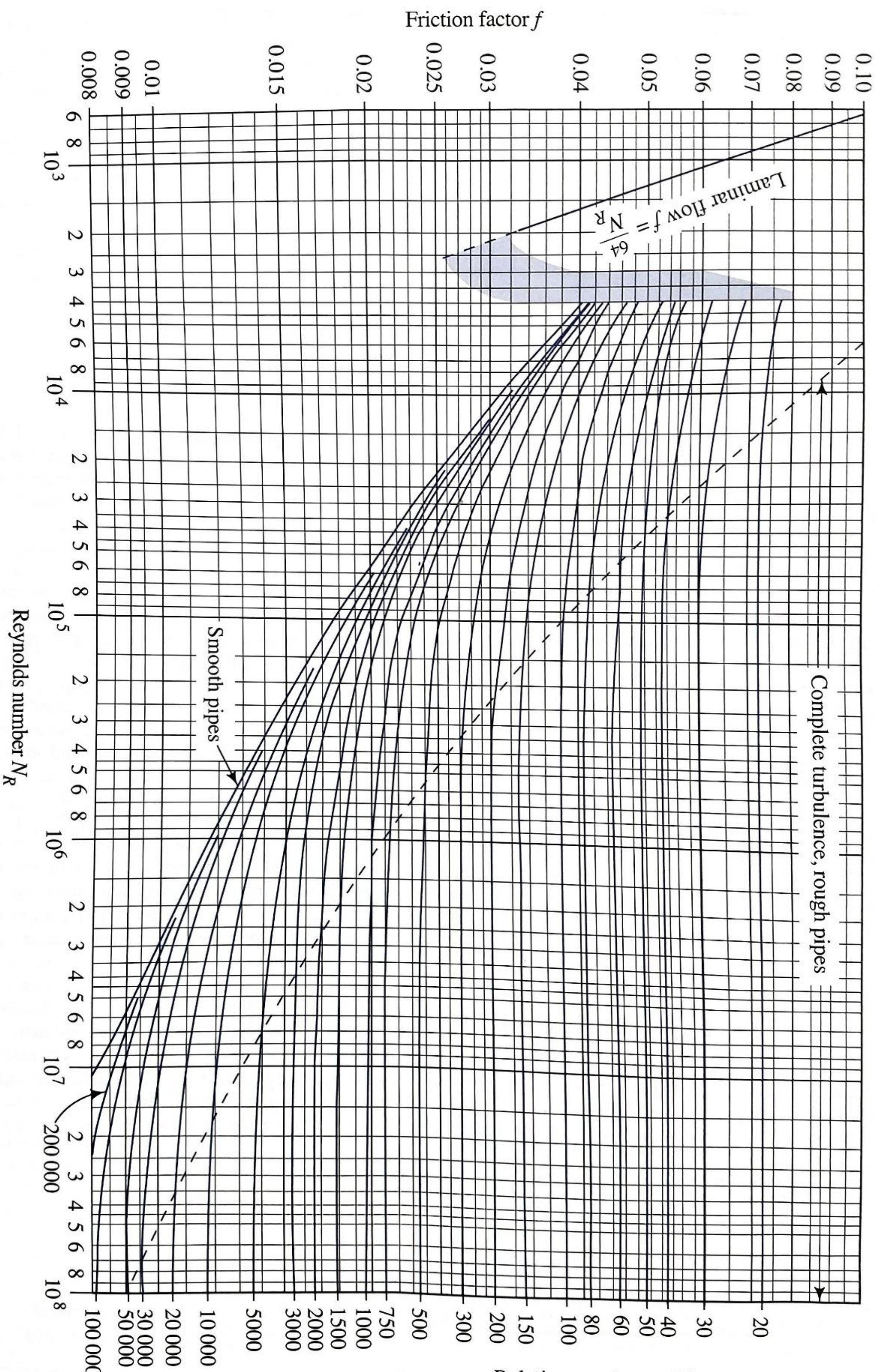
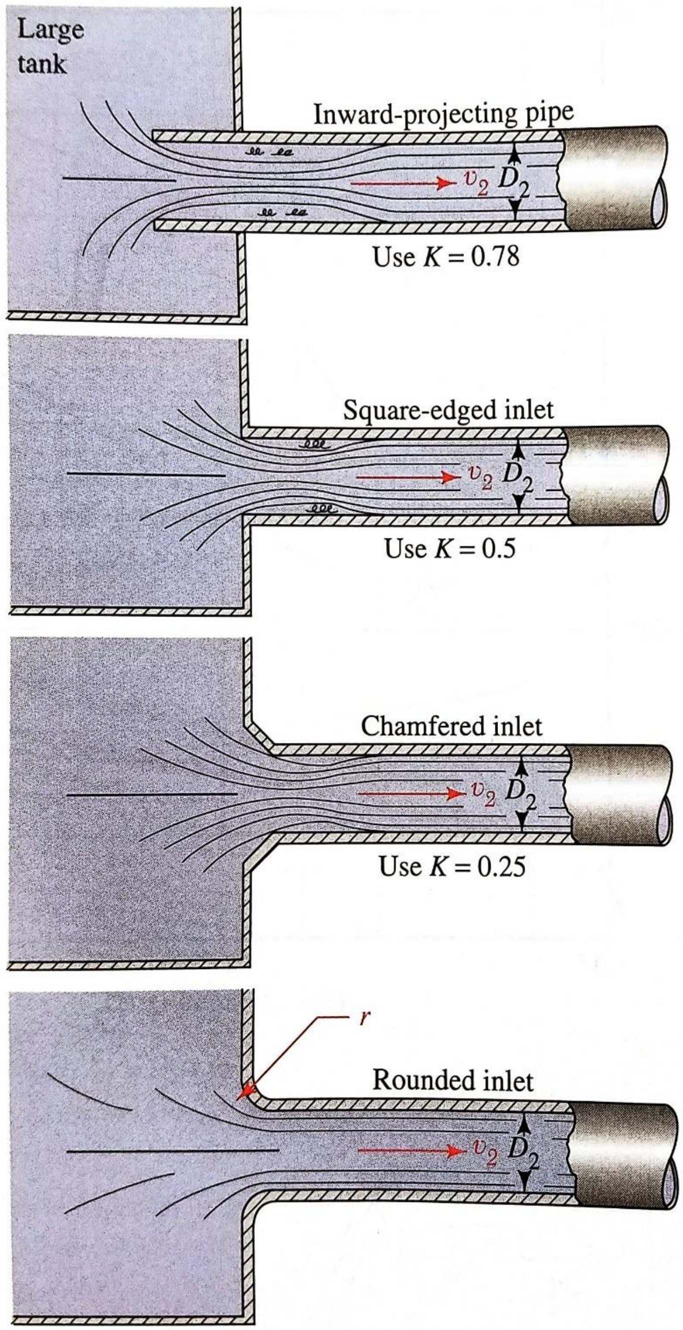


FIGURE 8.7 Moody's diagram. (*Source:* Pao, R.H.E. *Fluid Mechanics*, p. 284. Copyright 9c. 1961. Reprinted by permission of the author.)

Moody Diagram



r/D_2	K
0	0.50
0.02	0.28
0.04	0.24
0.06	0.15
0.10	0.09
>0.15	0.04 (Well-rounded)

Entrance losses chart

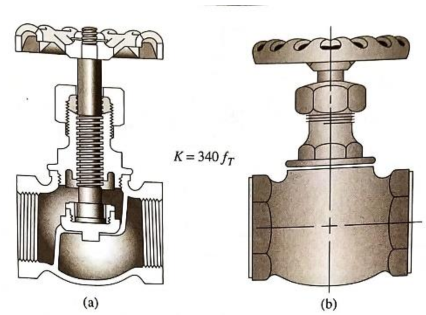


FIGURE 10.15 Globe valve. (Reprinted with permission from "Flow of Fluids Through Valves, Fittings and Pipe, Technical Paper 410" 2009. Crane Co. All Rights Reserved)

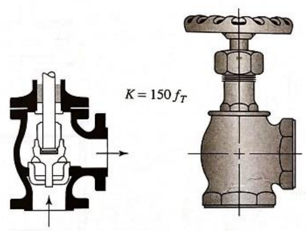


FIGURE 10.16 Angle valve. (Reprinted with permission from "Flow of Fluids Through Valves, Fittings and Pipe, Technical Paper 410" 2009. Crane Co. All Rights Reserved)

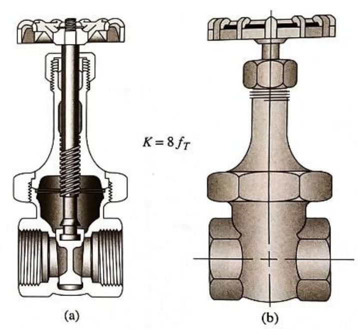


FIGURE 10.17 Gate valve. (Reprinted with permission from "Flow of Fluids Through Valves, Fittings and Pipe, Technical Paper 410" 2009. Crane Co. All Rights Reserved)

Gate valves diagram

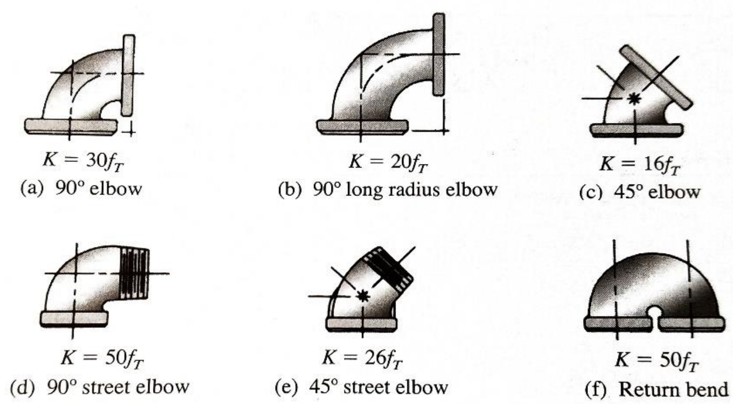


FIGURE 10.23 Pipe elbows. (Reprinted with permission from "Flow of Fluids Through Valves, Fittings and Pipe, Technical Paper 410" 2009 Crane Co. All Rights Reserved)

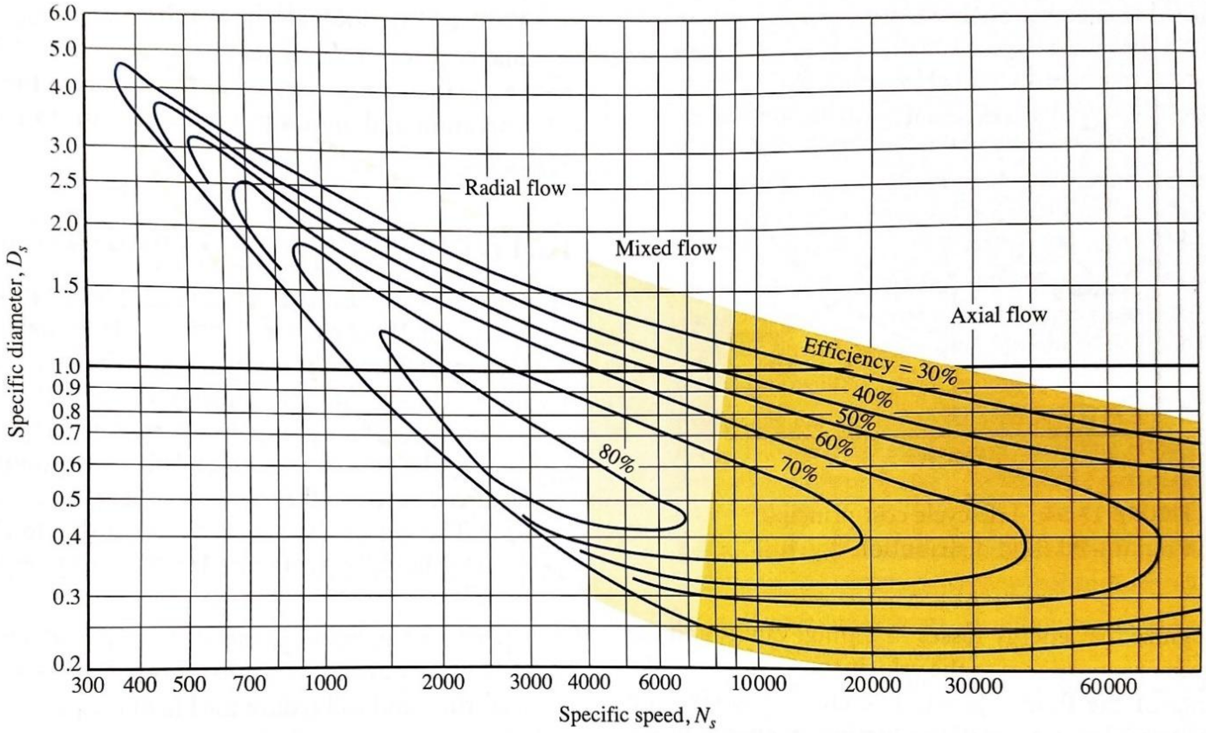
K values for elbows chart

TABLE 13.2 Vapor pressure and vapor pressure head of water

Temperature °C	Vapor Pressure kPa (abs)	Specific Weight (kN/m^3)	Vapor Pressure Head (m)
0	0.6105	9.806	0.06226
5	0.8722	9.807	0.08894
10	1.228	9.804	0.1253
20	2.338	9.789	0.2388
30	4.243	9.765	0.4345
40	7.376	9.731	0.7580
50	12.33	9.690	1.272
60	19.92	9.642	2.066
70	31.16	9.589	3.250
80	47.34	9.530	4.967
90	70.10	9.467	7.405
100	101.3	9.399	10.78

Temperature °F	Vapor Pressure (psia)	Specific Weight (lb/ft^3)	Vapor Pressure Head (ft)
32	0.08854	62.42	0.2043
40	0.1217	62.43	0.2807
50	0.1781	62.41	0.4109
60	0.2563	62.37	0.5917
70	0.3631	62.30	0.8393
80	0.5069	62.22	1.173
90	0.6979	62.11	1.618
100	0.9493	62.00	2.205
120	1.692	61.71	3.948
140	2.888	61.38	6.775
160	4.736	61.00	11.18
180	7.507	61.58	17.55
200	11.52	60.12	27.59
212	14.69	59.83	35.36

Table 13.2 – vapor pressure and vapor pressure head table



$$N_s = \frac{N\sqrt{Q}}{H^{3/4}} \qquad D_s = \frac{DH^{1/4}}{\sqrt{Q}}$$

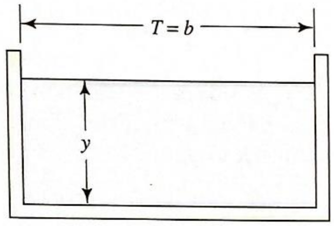
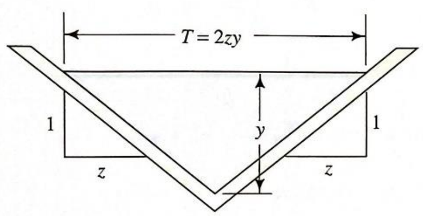
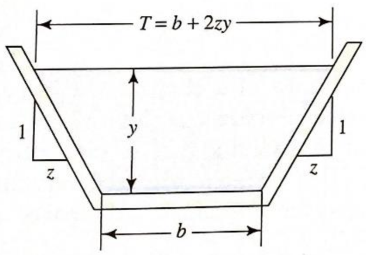
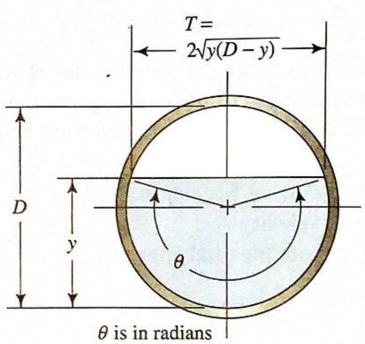
N = Rev/min H = Head, ft

Q = Flow, U.S. gpm D = Diameter, in

FIGURE 13.53 Specific speed versus specific diameter for centrifugal pumps—An aid to pump selection. (Excerpted by special permission from *Chemical Engineering*, April 3, 1978. Copyright © 1978 by McGraw-Hill, Inc., New York)

Figure 13.53 – specific speed vs. specific diameter for centrifugal pumps graph Table 14.2 –

TABLE 14.2 Geometry of open-channel sections

Section	Area A	Wetted Perimeter WP	Hydraulic Radius R
Rectangle 	by	$b + 2y$	$\dfrac{by}{b + 2y}$
Triangle 	zy^2	$2y\sqrt{1 + z^2}$	$\dfrac{zy}{2\sqrt{1 + z^2}}$
Trapezoid 	$(b + zy)y$	$b + 2y\sqrt{1 + z^2}$	$\dfrac{(b + zy)y}{b + 2y\sqrt{1 + z^2}}$
Circle 	$\dfrac{(\theta - \sin\theta)\, D^2}{8}$	$\theta D/2$	$\left[\dfrac{(\theta - \sin\theta)}{\theta}\right]\dfrac{D}{4}$

Note: θ must be in radians.
For $y < D/2$, $\theta = \pi - 2\sin^{-1}[1 - (2y/D)]$
For $y > D/2$, $\theta = \pi + 2\sin^{-1}[(2y/D) - 1]$

Geometry of open channel systems

TABLE 17.1 Typical drag coefficients

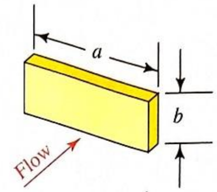
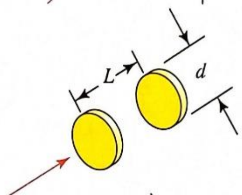
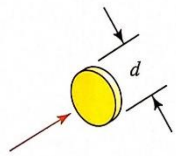
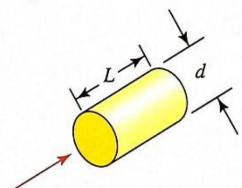
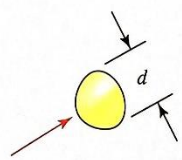
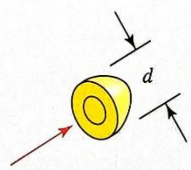
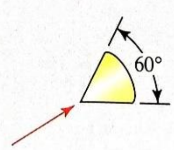
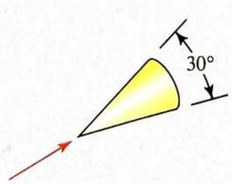
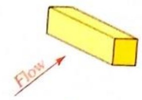
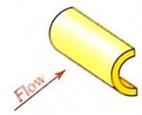
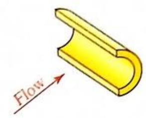
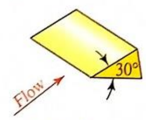
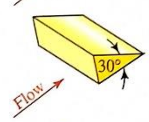
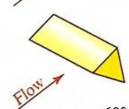
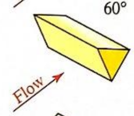
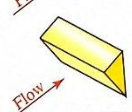
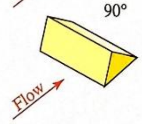
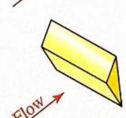
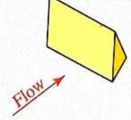
Shape of body	Orientation		C_D
Rectangular plate Flow is perpendicular to the flat front face.		a/b	
		1	1.16
		4	1.17
		8	1.23
		12.5	1.34
		25	1.57
		50	1.76
		∞	2.00
Tandem disks L = spacing d = diameter		L/d	
		1	0.93
		1.5	0.78
		2	1.04
		3	1.52
One circular disk			1.11
Cylinder L = length d = diameter		L/d	
		1	0.91
		2	0.85
		4	0.87
		7	0.99
Hemispherical cup, open back			0.41
Hemispherical cup, open front			1.35
Cone, closed base			0.51
			0.34

Table 17.1 – Typical drag coefficients

TABLE 17.1 Typical drag coefficients (*continued*)

Shape of body	Orientation	C_D
Square cylinder Flow is perpendicular to the flat front face.		1.60
Semitubular cylinders		1.12
		2.30
Triangular cylinders Flow is perpendicular to the flat front face - or in line with the angled front on the axis of the shape.		1.05
		1.85
		1.39
	60° 	2.20
		1.60
	90° 	2.15
		1.75
	120° 	2.05

Note: Reynolds numbers are typically from 10^4 to 10^5 and are based on the length of the body parallel to the flow direction, except for the semitubular cylinders, for which the characteristic length is the diameter.

Source: Data adapted from Avallone, Eugene A., and Theodore Baumeister III, eds. 1987. *Marks' Standard Handbook for Mechanical Engineers*, 9th ed. New York: McGraw-Hill, Table 4; and Lindsey, W. F. 1938. *Drag of Cylinders of Simple Shapes* (Report No. 619). National Advisory Committee for Aeronautics.

Table 17.1 continued – common drag coefficients

TABLE A.1 SI units [101 kPa (abs)]				
Temperature (°C)	Specific Weight γ (kN/m³)	Density ρ (kg/m³)	Dynamic Viscosity η (Pa·s)	Kinematic Viscosity ν (m²/s)
0	9.81	1000	1.75×10^{-3}	1.75×10^{-6}
5	9.81	1000	1.52×10^{-3}	1.52×10^{-6}
10	9.81	1000	1.30×10^{-3}	1.30×10^{-6}
15	9.81	1000	1.15×10^{-3}	1.15×10^{-6}
20	9.79	998	1.02×10^{-3}	1.02×10^{-6}
25	9.78	997	8.91×10^{-4}	8.94×10^{-7}
30	9.77	996	8.00×10^{-4}	8.03×10^{-7}
35	9.75	994	7.18×10^{-4}	7.22×10^{-7}
40	9.73	992	6.51×10^{-4}	6.56×10^{-7}
45	9.71	990	5.94×10^{-4}	6.00×10^{-7}
50	9.69	988	5.41×10^{-4}	5.48×10^{-7}
55	9.67	986	4.98×10^{-4}	5.05×10^{-7}
60	9.65	984	4.60×10^{-4}	4.67×10^{-7}
65	9.62	981	4.31×10^{-4}	4.39×10^{-7}
70	9.59	978	4.02×10^{-4}	4.11×10^{-7}
75	9.56	975	3.73×10^{-4}	3.83×10^{-7}
80	9.53	971	3.50×10^{-4}	3.60×10^{-7}
85	9.50	968	3.30×10^{-4}	3.41×10^{-7}
90	9.47	965	3.11×10^{-4}	3.22×10^{-7}
95	9.44	962	2.92×10^{-4}	3.04×10^{-7}
100	9.40	958	2.82×10^{-4}	2.94×10^{-7}

Table A.1 – properties of water, SI units

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-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}

Table A.2 – Properties of water, US customary units

TABLE B.1 SI units [101 kPa (abs) and 25°C]

	Specific Gravity sg	Specific Weight γ (kN/m^3)	Density ρ (kg/m^3)	Dynamic Viscosity η ($Pa\cdot s$)	Kinematic Viscosity ν (m^2/s)
Acetone	0.787	7.72	787	3.16×10^{-4}	4.02×10^{-7}
Alcohol, ethyl	0.787	7.72	787	1.00×10^{-3}	1.27×10^{-6}
Alcohol, methyl	0.789	7.74	789	5.60×10^{-4}	7.10×10^{-7}
Alcohol, propyl	0.802	7.87	802	1.92×10^{-3}	2.39×10^{-6}
Aqua ammonia (25%)	0.910	8.93	910	—	—
Benzene	0.876	8.59	876	6.03×10^{-4}	6.88×10^{-7}
Carbon tetrachloride	1.590	15.60	1 590	9.10×10^{-4}	5.72×10^{-7}
Castor oil	0.960	9.42	960	6.51×10^{-1}	6.78×10^{-4}
Ethylene glycol	1.100	10.79	1 100	1.62×10^{-2}	1.47×10^{-5}
Gasoline	0.68	6.67	680	2.87×10^{-4}	4.22×10^{-7}
Glycerin	1.258	12.34	1 258	9.60×10^{-1}	7.63×10^{-4}
Kerosene	0.823	8.07	823	1.64×10^{-3}	1.99×10^{-6}
Linseed oil	0.930	9.12	930	3.31×10^{-2}	3.56×10^{-5}
Mercury	13.54	132.8	13 540	1.53×10^{-3}	1.13×10^{-7}
Propane	0.495	4.86	495	1.10×10^{-4}	2.22×10^{-7}
Seawater	1.030	10.10	1 030	1.03×10^{-3}	1.00×10^{-6}
Turpentine	0.870	8.53	870	1.37×10^{-3}	1.57×10^{-6}
Fuel oil, medium	0.852	8.36	852	2.99×10^{-3}	3.51×10^{-6}
Fuel oil, heavy	0.906	8.89	906	1.07×10^{-1}	1.18×10^{-4}
Approximate data for selected natural and biological fluids. Values vary significantly with composition.					
Olive oil at 68°F (20°C)	0.92	9.03	920	0.085	9.24×10^{-5}
Honey at 70°F (21°C)	1.42	13.93	1420	10.0	7.04×10^{-3}
Ketchup at 70°F (21°C)	1.48	14.52	1480	50.0	3.38×10^{-2}
Peanut butter at 70°F (21°C)	1.30	12.75	1300	250	1.92×10^{-1}
Blood at 50°F (10°C)	1.06	10.20	1060	0.01	9.43×10^{-6}
Blood at 98.6°F (37°C)	1.06	10.20	1060	3.5×10^{-3}	3.30×10^{-6}

Table B.1 – properties of common substances, SI units

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)
1/8	6	0.405	10.3	0.068	1.73	0.269	0.0224	6.8	0.000 394	3.660×10^{-5}
1/4	8	0.540	13.7	0.088	2.24	0.364	0.0303	9.2	0.000 723	6.717×10^{-5}
3/8	10	0.675	17.1	0.091	2.31	0.493	0.0411	12.5	0.001 33	1.236×10^{-4}
1/2	15	0.840	21.3	0.109	2.77	0.622	0.0518	15.8	0.002 11	1.960×10^{-4}
3/4	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}
1 1/4	32	1.660	42.2	0.140	3.56	1.380	0.1150	35.1	0.010 39	9.653×10^{-4}
1 1/2	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}
2 1/2	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}
3 1/2	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

Table F.1 – properties of Schedule 40 steel pipe

TABLE K.1 Conversion factors

Mass Standard SI unit: kilogram (kg). Equivalent unit: $N{\cdot}s^2/m$.

$$\frac{14.59\ kg}{slug} \quad \frac{32.174\ lb_m}{slug} \quad \frac{2.205\ lb_m}{kg} \quad \frac{453.6\ grams}{lb_m} \quad \frac{2000\ lb_m}{ton_m} \quad \frac{1000\ kg}{metric\ ton_m}$$

Force Standard SI unit: Newton (N). Equivalent unit: $kg{\cdot}m/s^2$.

$$\frac{4.448\ N}{lb_f} \quad \frac{10^5\ dynes}{N} \quad \frac{4.448 \times 10^5\ dynes}{lb_f} \quad \frac{224.8\ lb_f}{kN}$$

Length

$$\frac{3.281\ ft}{m} \quad \frac{39.37\ in}{m} \quad \frac{12\ in}{ft} \quad \frac{1.609\ km}{mi} \quad \frac{5280\ ft}{mi} \quad \frac{6076\ ft}{nautical\ mile}$$

Area

$$\frac{144\ in^2}{ft^2} \quad \frac{10.76\ ft^2}{m^2} \quad \frac{645.2\ mm^2}{in^2} \quad \frac{10^6\ mm^2}{m^2} \quad \frac{43\,560\ ft^2}{acre} \quad \frac{10^4\ m^2}{hectare}$$

Volume

$$\frac{1728\ in^3}{ft^3} \quad \frac{231\ in^3}{gal} \quad \frac{7.48\ gal}{ft^3} \quad \frac{264.2\ gal}{m^3} \quad \frac{3.785\ L}{gal} \quad \frac{35.31\ ft^3}{m^3}$$

$$\frac{28.32\ L}{ft^3} \quad \frac{1000\ L}{m^3} \quad \frac{61.02\ in^3}{L} \quad \frac{1000\ cm^3}{L} \quad \frac{1.201\ U.S.\ gal}{Imperial\ gallon}$$

Volume Flow Rate

$$\frac{449\ gal/min}{ft^3/s} \quad \frac{35.31\ ft^3/s}{m^3/s} \quad \frac{15\,850\ gal/min}{m^3/s} \quad \frac{3.785\ L/min}{gal/min}$$

$$\frac{60\,000\ L/min}{m^3/s} \quad \frac{2119\ ft^3/min}{m^3/s} \quad \frac{16.67\ L/min}{m^3/h} \quad \frac{101.9\ m^3/h}{ft^3/s}$$

***Density* (mass/unit volume)**

$$\frac{515.4\ kg/m^3}{slug/ft^3} \quad \frac{1000\ kg/m^3}{gram/cm^3} \quad \frac{32.17\ lb_m/ft^3}{slug/ft^3} \quad \frac{16.018\ kg/m^3}{lb_m/ft^3}$$

***Specific Weight* (weight/unit volume)**

$$\frac{157.1\ N/m^3}{lb_f/ft^3} \quad \frac{1728\ lb/ft^3}{lb/in^3}$$

Pressure Standard SI unit: pascal (Pa). Equivalent units: N/m^2 or $kg/m{\cdot}s^2$.

$$\frac{144\ lb/ft^2}{lb/in^2} \quad \frac{47.88\ Pa}{lb/ft^2} \quad \frac{6895\ Pa}{lb/in^2} \quad \frac{1\ Pa}{N/m^2} \quad \frac{100\ kPa}{bar} \quad \frac{14.50\ lb/in^2}{bar}$$

$$\frac{27.68\ inH_2O}{lb/in^2} \quad \frac{249.1\ Pa}{inH_2O} \quad \frac{2.036\ inHg}{lb/in^2} \quad \frac{3386\ Pa}{inHg} \quad \frac{133.3\ Pa}{mmHg} \quad \frac{51.71\ mmHg}{lb/in^2}$$

$$\frac{14.696\ lb/in^2}{Std.\ atmosphere} \quad \frac{101.325\ kPa}{Std.\ atmosphere} \quad \frac{29.92\ inHg}{Std.\ atmosphere} \quad \frac{760.1\ mmHg}{Std.\ atmosphere}$$

Table K.1 – conversion factors

TABLE K.1 Conversion factors (*continued*)

Note: Conversion factors based on the height of a column of liquid (e.g., inH_2O and mmHg) are based on a standard gravitational field ($g = 9.806\ 65\ m/s^2$), a density of water equal to $1000\ kg/m^3$, and a density of mercury equal to $13\ 595.1\ kg/m^3$, sometimes called ***conventional values*** for a temperature at or near 0°C. Actual measurements with such fluids may vary because of differences in local gravity and temperature.

Energy Standard SI unit: joule (J). Equivalent units: N·m or $kg{\cdot}m^2/s^2$.

$\frac{1.356\ J}{lb\text{-}ft}$ $\frac{1.0\ J}{N{\cdot}m}$ $\frac{8.85\ lb\text{-}in}{J}$ $\frac{1.055\ kJ}{Btu}$ $\frac{3.600\ kJ}{W{\cdot}h}$ $\frac{778.17\ ft\text{-}lb}{Btu}$

Power Standard SI unit: watt (W). Equivalent unit: J/s or N·m/s.

$\frac{745.7\ W}{hp}$ $\frac{1.0\ W}{N{\cdot}m/s}$ $\frac{550\ lb\text{-}ft/s}{hp}$ $\frac{1.356\ W}{lb\text{-}ft/s}$ $\frac{3.412\ Btu/hr}{W}$ $\frac{1.341\ hp}{kW}$

Dynamic Viscosity Standard SI unit: Pa·s or $N{\cdot}s/m^2$ (cP = centipoise)

$\frac{47.88\ Pa{\cdot}s}{lb\text{-}s/ft^2}$ $\frac{10\ poise}{Pa{\cdot}s}$ $\frac{1000\ cP}{Pa{\cdot}s}$ $\frac{100\ cP}{poise}$ $\frac{1\ cP}{1\ mPa{\cdot}s}$

Kinematic Viscosity Standard SI unit: m^2/s (cSt = centistoke)

$\frac{10.764\ ft^2/s}{m^2/s}$ $\frac{10^4\ stoke}{m^2/s}$ $\frac{10^6\ cSt}{m^2/s}$ $\frac{100\ cSt}{stoke}$ $\frac{1\ cSt}{1\ mm^2/s}$ $\frac{10^6\ mm^2/s}{m^2/s}$

Refer to Section 2.6.5 for conversions involving Saybolt Universal seconds.

General Approach to Application of Conversion Factors. Arrange the conversion factor from the table in such a manner that when multiplied by the given quantity, the original units cancel out, leaving the desired units.

Example 1 Convert $0.24\ m^3/s$ to the units of gal/min:

$$(0.24\ m^3/s)\frac{15\ 850\ gal/min}{m^3/s} = 3804\ gal/min$$

Example 2 Convert 150 gal/min to the units of m^3/s:

$$(150\ gal/min)\frac{1\ m^3/s}{15\ 850\ gal/min} = 9.46 \times 10^{-3}\ m^3/s$$

Temperature Conversions (Refer to Section 1.6)

Given the Fahrenheit temperature T_F in °F, the Celsius temperature T_C in °C is

$$T_C = (T_F - 32)/1.8$$

Given the temperature T_C in °C, the Fahrenheit temperature T_F in °F is

$$T_F = 1.8T_C + 32$$

Given the temperature T_C in °C, the absolute temperature T_K in K (kelvin) is

$$T_K = T_C + 273.15$$

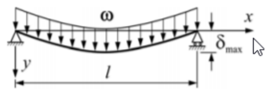
Given the temperature T_F in °F, the absolute temperature T_R in °R (degrees Rankine) is

$$T_R = T_F + 459.67$$

Given the temperature T_F in °F, the absolute temperature T_K in K is

$$T_K = (T_F + 459.67)/1.8 = T_R/1.8$$

Table K.1 continued – more conversion factors

8. Beam Simply Supported at Ends – Uniformly distributed load ω (N/m)			
	$\theta_1 = \theta_2 = \dfrac{\omega l^3}{24EI}$	$y = \dfrac{\omega x}{24EI}\left(l^3 - 2lx^2 + x^3\right)$	$\delta_{max} = \dfrac{5\omega l^4}{384EI}$

Deflection for a simply-supported beam

Robert’s Reflection:

This project has been an eye-opening experience. As someone who has not worked in the industry yet, this is a good introduction as to how projects will be structured and what will be asked of an engineer in the industry. This class and project have been one of the most informative classes I’ve ever taken in my college career. Not only are the ideas and material useful, they are used constantly in the industry to make designs and perform necessary operations in the field. Much of the material I learned will be most applicable to creating other pipe systems and fluid carrying systems for an engineering firm. Most firms would want to know the description of this project. It can only be described as a full design of a pipe system and implementation of the ideas and learning material that were taught in fluid dynamics. Applying everything that I learned in the class and seeing how it actually applies in the field is the most interesting and satisfying aspect of this project. I feel that I contributed an equal amount of the work in this project. As the planner, it was my job to keep the group on task and to set a schedule for the completion of the project. We decided to split the work up evenly between each member to sufficiently spread the workload between each member. Being able to remain productive and talk out solutions was my most valuable trait that I brought to this task. There were many times that the project was hitting a stalemate between the three of us. However, being able to talk it out and explain each of our perspectives was incredibly helpful in solving the problems that arose during the project. One of my weaknesses is the inability to properly organize my thoughts and ideas in an orderly fashion. To address this, much of the documentation and calculations were performed digitally which helped keep the group organized and the information clear and concise. This project definitely helps anyone working on it to fully understand the principals of fluid dynamics and system design. However, in real life, technical drawings are incredibly detailed and not limited to an 8.5”x11” sheet of paper and are usually drawn fully to scale with even the most minute details drawn out and fully detailed in the final report. As this was slightly out of the scope of this project, there were not high-level drawings of this nature. Only basic CAD drawings were needed to convey the design. It would be interesting to have other detail-oriented design drawings that were mandatory to practice the technical drawing aspect of engineering. If I were to start this class over knowing what I know now, I would mostly stay the same in my thought process in approaching the class. This included staying on schedule, completing the homework and completing more than the bare minimum of problems. I would have probably reviewed the lecture notes and other documents more frequently than I did this semester, but there was not much more I felt needed to be done to receive a good grade in the course.

Khanh’s Reflection:

The project taught me much more than concepts about fluid dynamics. This project taught me how to manage my time properly, how to speak to a client, and how to describe an engineering problem to someone in authority who may have lesser knowledge about the subject matter. I will use what I learn here in the motorsports industry when talking to team managers and immediate superiors. One must be clear, concise, and quick when explaining a problem in a field as fast-paced as motorsports. In an interview setting, I would describe my contribution to the project as a lead researcher. I did most of the analysis of each system and did a thorough job looking up previous engineering projects to maximize the efficiency of my own development. My strengths in work ethic and diligence really helped me cover things that others may overlook, mainly ideas “beyond theory”. Instead of theorizing over the project, I kept my mind open to how the “real world” would interact with the system. As a result, a major weakness of mine would be time management. Being “perfect” is very time consuming, and I need to learn when good enough is good enough. The technical strength of the project is the fact that the researching and calculations are extremely thorough and correct for an ideal-world situation. On the flip side, that means the greatest technical weakness of the project is the disconnect between the ideal-world and the real-world. Often, the “correct” item for the project may not necessarily be the most efficient or cheapest way to solve an engineering problem. For example, a flow nozzle may be the more efficient instrumentation choice, but the cheaper cost of orifice plates in the real-world make it a more desirable option despite its disadvantages in energy losses. If I could start the class over, I would tell myself to get ahead and stay ahead. Finishing a semester long project while halfway through finals week was a very difficult way to end the semester.

Nick’s Reflection

I believe what I have learned from this project is important for my professional career because it exposed me to real-world situations and different methods of how to complex problems. These concepts can be used in any industry, as it applies to the products as well as the manufacturing processes and machinery used to construct them. My role in this project was the leader, and it was my responsibility to make accurate calculations and make sure all of our numbers seemed realistic. Additionally, I had to make sure this project and our problem-solving skills progressed and developed in a timely manner. My inclination for math and my advanced computer skills were instrumental in completing this project. Many hours were spent modifying equations, calculating data, and iterating, which would prove to be a very difficult and frustrating experience for someone who does not posses these qualities. One thing that prevented me from working as efficiently as I could is when I was solving complex problems (such as energy losses) and getting frustrated because my numbers seemed unrealistic. This was a valuable experience for me because it taught me how to better manage my time when I was faced with a difficult problem. My solution for this was to take a quick break to help clear my head, and then return to work on something else in the project. This forced me to take time to think about different problem-solving strategies I could apply to the more difficult problem when I returned to it later. My technical strengths seen in this project are CAD drawing, document organization, algebraic manipulation, and calculators made in Microsoft Excel. Some weaknesses of mine include modifying system curves and calculating pump properties and requirements. If I took this class again, I would be sure to do the extra homework problems in order to help retain and practice the information we learn in class. There have been many occasions where I have had to go and watch a lecture several times because I had not had sufficient practice to know how to solve more complex problem. Printouts of equation sheets and commonly used tables would be very convenient to have on-hand so I did not have to log in to blackboard and find the lecture it was in to be able to find an equation I need.