Final Report

MET 330

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2. Abstract

Continental AG is planning for a new manufacturing facility. As part of the new plant, there will be an automated machining line in which five machines will be supplied with coolant from the same reservoir. TRDoubleD Engineering has put together the following plan to supply the system with coolant. The system includes a 47,000 gallon main reservoir for storing the coolant, a 1,057 gallon feed tank to supply the machining area, and a 4,362 gallon tank to contain the fluid after use. The system is partially gravity feed and only requires three pumps and electric motors.

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5. Report Body

a. Job site location

Dayton, Ohio, United States

b. Specifications and design philosophy

- New coolant is delivered to the plant by railroad tank cars carrying 15,000 gal each. It has been decided that 3 tanker cars will be requester per delivery, totaling 45,000 gals.
- A holding tank for new coolant must be specified.
- The reservoir for the automated machining system must have a capacity of 1000 gal.
- The 1000-gal tank is normally emptied once per week. Emergency dumps are possible if the coolant becomes overly contaminated prior to the scheduled emptying.
- The dirty fluid is picked up by truck only once per month.
- A holding tank for the dirty fluid must be specified.
- The plant is being designed to operate two shifts per day, 7 days a week.
- Maintenance is normally performed during the third shift.
- The building is one-story high with a concrete floor.
- The floor level is at the same elevation as the railroad track.
- No storage tank can be inside the plant or under the floor except the 1000-gal reservoir that supplies the machining system.
- The roof top is 32 ft from the floor level and the roof can be designed to support a storage
- tank.
- The building is to be located in Dayton, Ohio, where the outside temperature may range from -20˚F to +105 ˚F.
- The frost line is 30 in below the surface.
- You are not asked to design the system to supply the machines.
- The basic coolant storage and delivery system is to have the functional design sketched in the block diagram in Figure 2.
- If pumps are required, only SULZER pumps have to be selected.

C. **Sources**

- *Endmemo*. EndMemo. (n.d.). Retrieved November 22, 2021, from http://www.endmemo.com/sconvert/galus_minft3_s.php.
- https://www.engineeringtoolbox.com/wrought-steel-pipe-bursting-pressure-d_1123.html
- https://www.gbrx.com/media/1466/tank29000.pdf
- *Hanger support spacing - rod sizes horizontal pipes*. Engineering ToolBox. (n.d.). Retrieved November 29, 2021, from https://www.engineeringtoolbox.com/piping-support-d_362.html.
- https://www.journal-news.com/news/local/wind-gusts-mph-gust-recorded-high-winds-move-through-area/6WwUMeWK2GmirlzqRcCULJ/
- Maus, P. (2021, February 2). *Wind load calculator*. How Much Force Does the Wind Produce? Retrieved November 21, 2021, from https://www.omnicalculator.com/physics/wind-load.
- Mott, R., Untener, J.A, “Applied Fluid Mechanics”, 7th edition, Pearson Education, Inc (2015)
- *https://www.omnicalculator.com/physics/wind-load*
- *Pipe Flange: Carbon Steel, blind flange, 5 in pipe size, raised face blind flange*. Grainger. (n.d.). Retrieved November 21, 2021, from https://www.grainger.com/product/GRAINGER-APPROVED-Pipe-Flange-Carbon-Steel-30WH03.
- Project Document
- Sulzer Catalog
- *Wrought steel pipe - bursting pressures*. Engineering ToolBox. (n.d.). Retrieved November 21, 2021, from https://www.engineeringtoolbox.com/wrought-steel-pipe-bursting-pressure-d_1123.html.

d. Materials and Specifications

i. Establish the pipe and tank material use

- Schedule 40 carbon steel pipes will be used (ASTM A106).
- Tanks will also be carbon steel (ASTM A106).
- Seams are electric resistance welded.

ii. Fluid characteristics

- The coolant is a solution of water and soluble oil with a specific gravity of 0.94 and a freezing point of 0 °F. Its corrosiveness is approximately the same as that of water.
- Viscosity and vapor pressure of the coolant are 1.50 times that of water at any temperature.

e. Preliminary drawings and sketches

i. Plot Plan

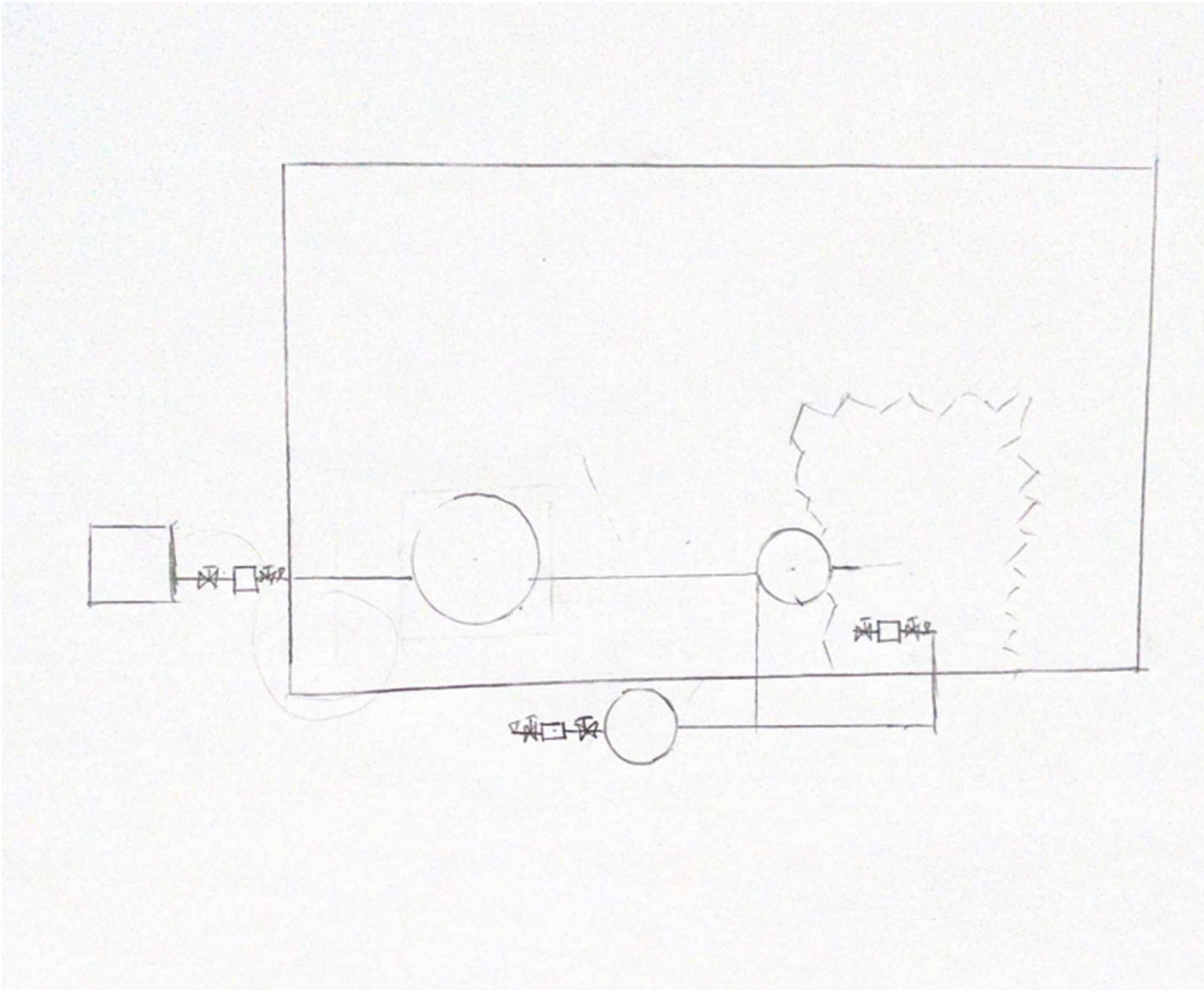


Figure 1: Plot Plan View of System Layout

ii. Elevations

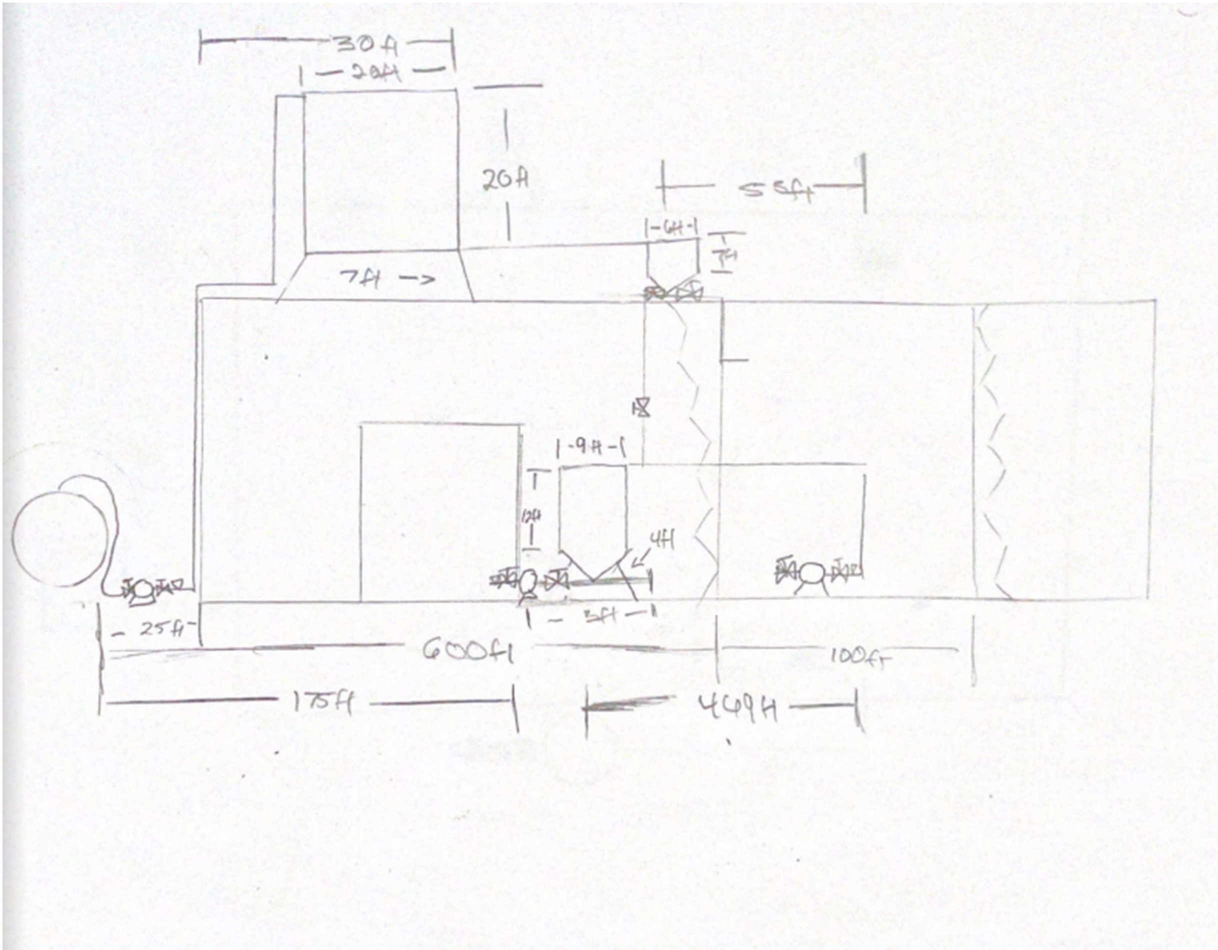


Figure 2: Elevations View of System Layout

f. Design Calculations

i. Tank Specifications

1. Specify the size and location of all storage tanks.

Purpose:

Layout the piping system and determine the tank sizes.

Sources:

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

Design Considerations:

1. New coolant is delivered to the plant by railroad tank cars carrying 15,000 gal each. A holding tank for new coolant must be specified.
2. The reservoir for the automated machining system must have a capacity of 1000 gal.
3. The 1000-gal tank is normally emptied once per week. Emergency dumps are possible if the coolant becomes overly contaminated prior to the scheduled emptying.
4. The dirty fluid is picked up by truck only once per month.
5. A holding tank for the dirty fluid must be specified.
6. The plant is being designed to operate two shifts per day, 7 days a week.
7. Maintenance is normally performed during the third shift.
8. The building is one-story high with a concrete floor.
9. The floor level is at the same elevation as the railroad track.
10. No storage tank can be inside the plant or under the floor except the 1000-gal reservoir that supplies the machining system.
11. The roof top is 32 ft from the floor level and the roof can be designed to support a storage tank.
12. The building is to be located in Dayton, Ohio, where the outside temperature may range from -20˚F to +105˚F.
13. The frost line is 30 in below the surface.
14. You are not asked to design the system to supply the machines.

Drawings and Diagrams

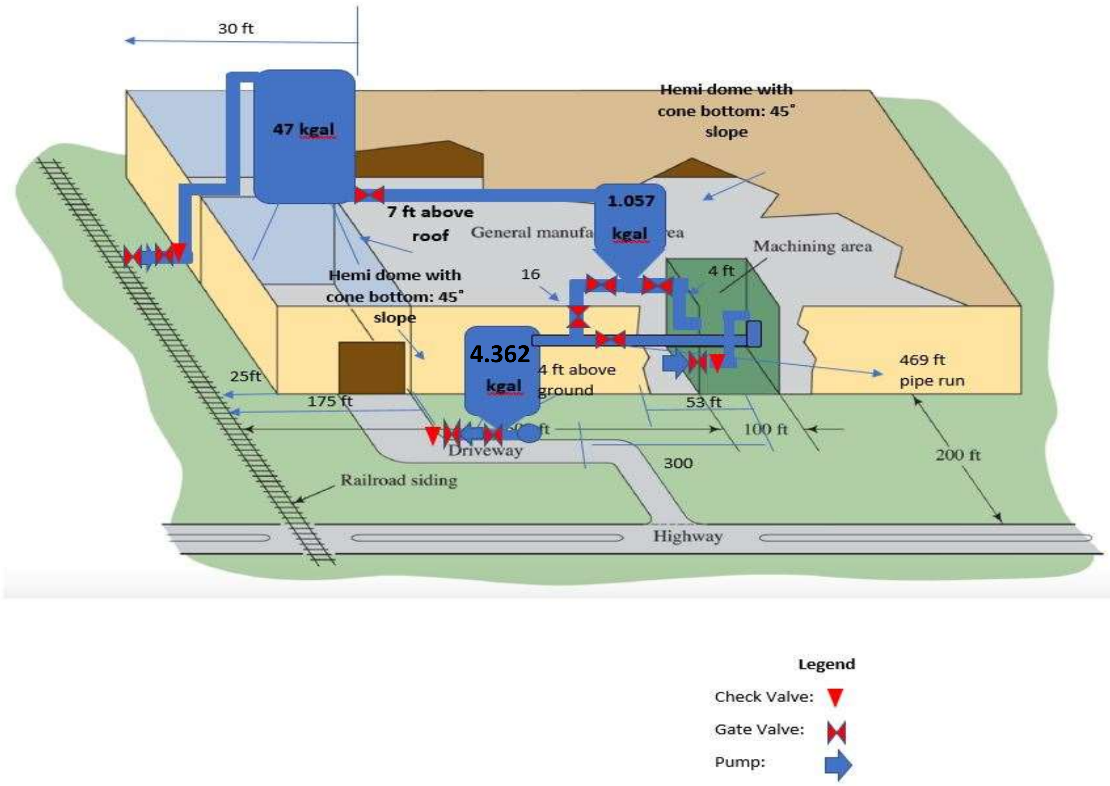


Figure 3: System Diagram

Materials:
- Coolant
- 3 Tanks: Carbon Steel ASTM A106, Electric Resistance Welded
    - 47000 Gal
    - 2115 Gal
    - 1057 Gal

Data and Variables:
- Distances in figure
- $\gamma = 58.656\ lb/ft^3$

Procedure:
1. An initial delivery of 45000 gallons of coolant was decided on.
2. A diameter of 20 ft was chosen for the large holding tank and the height was calculated based on the volume and diameter.
3. The same process to determine the size of the tanks was used for the two smaller tanks, with the addition of a 45˚ cone bottom to aid with emptying.
4. The tanks were laid out in a way to take advantage of gravity and minimize pump usage.
5. **Dimensions were added based on given dimensions, tank diameters, and approximate location**

Calculations:

Large Tank Size:

$$V = \frac{45000\ gal}{7.481} = 6015.625\ ft^3$$

$D_{inner} = 20\ ft$

$V = Ah \qquad A = \pi r^2$

$= \pi\, 10^2$

$= 314.195\ ft^2$

$6015.625\ ft^3 = 314.195\ ft^2\,(h) = 19.148\ ft$ of fluid

1000 gallon Tank: $D_{inner} = 6\ ft$

$$V = \frac{1000\ gal}{7.481} = 133.67\ ft^3 \qquad A = \pi r^2 = 28.27\ ft^2$$

45° cone bottom: $V \tfrac{1}{3}(h)(\pi r^2) = 27.99\ ft^3$

$V = 133.67\ ft^3 - 27.99\ ft^3 = 105.68\ ft^3$

$105.68\ ft^3 = 28.27 h = 3.73\ ft + 3\ ft = 6.73\ ft$

Dirty Tank:

$D = 9\,ft$

$V = \dfrac{4000\,gal}{7.481} = 534.69\,ft^3$

$42°$ cone bottom $= V = \frac{1}{3} h \pi r^2 = \frac{1}{3}(5\,ft)(\pi\, 4.5\,ft^2) = 106.03\,ft^3$

$V = 534.69\,ft^3 - 106.03\,ft^3 = 428.66\,ft^3$

$A = \pi r^2 = \pi\, 4.5\,ft^2 = 63.62\,ft^2$

$\dfrac{428.66\,ft^3}{63.62\,ft^2} = h_{cyl} = 6.73\,ft + 4.5\,ft = 11.74\,ft \cong 12\,ft$

$\dfrac{V}{63.62} = 7.5$

$V = 477.15\,ft^3 + 106.03\,ft^3 = 583.18 \times 7.481$

$V = 4362\,gal$

Summary:

The large tank must be 20 ft in diameter with a height of 20 ft. Rounding from the calculated height of 19.148 ft to 20 ft allows the piping to fill the tank from above the surface of the fluid when nearing the full 45000 gallons.

The dirty tank must be 9 ft in diameter with a height of 12 ft. Rounding from the calculated height of 11.74 ft to 12 ft allows the piping to fill the tank from above the surface of the fluid when nearing the full 4000 gallons.

The coolant reservoir tank must be 6 ft in diameter with a height of 7 ft. Rounding from the calculated height of 6.73 ft to 7 ft allows the piping to fill the tank from above the surface of the fluid when nearing the full 1000 gallons.

Analysis:

Rounding the tank heights to allow for room above the fluid results in tank volumes of 47000 gallons, 4362 gallons, and 1057 gallons. Due to the arrangement of the tanks, only 3 pumps will be required in the entire system.

2. Select tank material and specify wall thickness of storage tanks.

Purpose:

Chose the material the tanks will be made of and determine the required thickness of the tanks.

Sources:

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

Design Considerations:

- The reservoir for the automated machining system must have a capacity of 1000 gal.
- Constant Properties
- Incompressible Fluids
- Assuming pressure is highest at the bottom of the tank

Drawings and Diagrams:

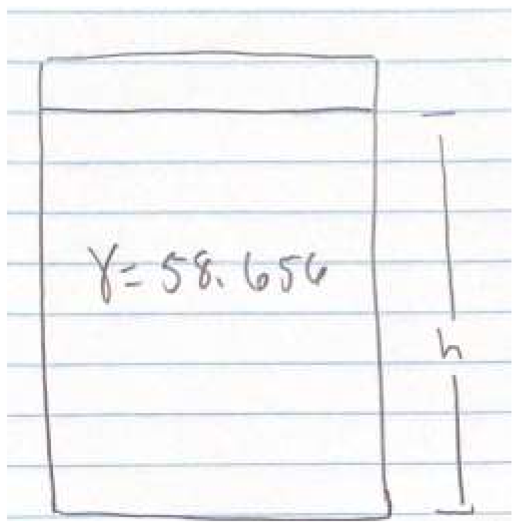


Figure 4: Determining Pressure at Bottom of a Tank

Data and Variables:

- Distances in figure
- $\gamma = 58.656\ lb/ft^3$
- S= 20000 psi
- E = 0.85
- Y= 0.4

Materials:

- Coolant
- 3 Tanks: Carbon Steel ASTM A106, Electric Resistance Welded
  - 47000 Gal
  - 2115 Gal
  - 1057 Gal

Procedure:

1. Gamma height equation was used to determine the pressure at the bottom of each tank.
2. The pressure was then plugged into equation 11.9 from the textbook to determine the wall thickness of the tank.

Calculations:

Large tank! $P = \gamma h$

$$P = 58.456\ \mathrm{lb/ft^3}\,(19.1489) = \frac{1122.55\ \mathrm{psf}}{144} = 7.795\ \mathrm{psi}$$

$$t = \frac{PD}{2(SE)+PY} = \frac{(7.795\,\mathrm{psi})(240\,\mathrm{in})}{2(20000\,\mathrm{psi} \times 0.85) + (7.795\,\mathrm{psi} \times 0.4)}$$

$t = 0.55\,\mathrm{in}$

$t_{min} = t + 0.08\,\mathrm{in} = 0.55\,\mathrm{in} + 0.08\,\mathrm{in} = 0.135\,\mathrm{in}$

$$t_{nom} = \frac{t_{min}}{0.875} = \frac{0.135\,\mathrm{in}}{0.875} = 0.154\,\mathrm{in}$$

Dirty tank: $P = 4.67\,\mathrm{psi}$ $t = 0.0099\,\mathrm{in}$ $t_{min} = 0.089\,\mathrm{in}$

$t_{nom} = 0.103\,\mathrm{in}$

Resivior: $P = 2.74\,\mathrm{psi}$ $t = 0.0059\,\mathrm{in}$ $t_{min} = 0.085\,\mathrm{in}$

$t_{nom} = 0.098\,\mathrm{in}$

Summary:

The nominal thickness for the large tank is 0.154 in
The nominal thickness for the dirty tank is 0.103 in
The nominal thickness for the large tank is 0.098 in

Analysis:

Applying a 2.5 factor of safety, the large tank thickness becomes 0.385 in. The dirty tank thickness becomes 0.258 in. The reservoir becomes 0.245 in.

3. Provide a future additional connection to drain ONE OF THE TANKS. Design the blind flange required to hold the pressure for such connection (size, thickness, etc.). This should include the number of bolts and nuts and the size of them.

Purpose: Design a tank to allow for a future connection by designing a blind flange, include bolt calculations.

Sources:

- Mott, R., Untener, J.A, "Applied Fluid Mechanics", 7th edition, Pearson Education, Inc (2015)
- *Pipe Flange: Carbon Steel, blind flange, 5 in pipe size, raised face blind flange*. Grainger. (n.d.). Retrieved November 21, 2021, from https://www.grainger.com/product/GRAINGER-APPROVED-Pipe-Flange-Carbon-Steel-30WH03.
- Project Document
- *Wrought steel pipe - bursting pressures*. Engineering ToolBox. (n.d.). Retrieved November 21, 2021, from https://www.engineeringtoolbox.com/wrought-steel-pipe-bursting-pressure-d_1123.html.

Design Considerations:

- Pressure is equal to pressure at bottom of the tank
- Bolt holes designed for ¾"

Drawings and Diagrams:

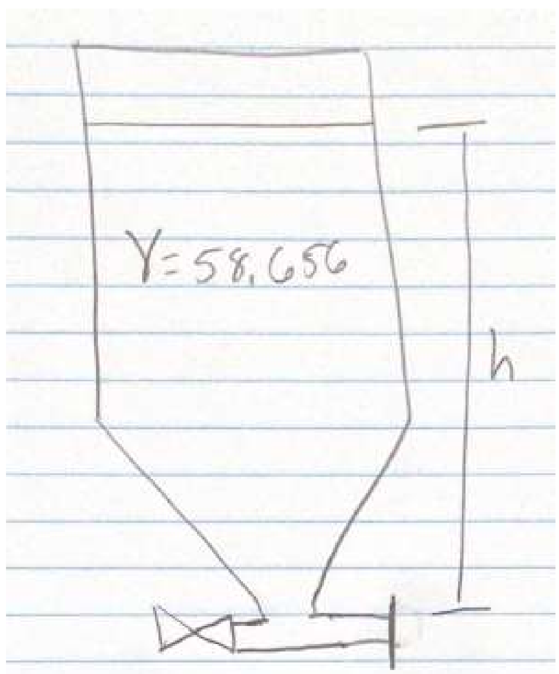


Figure 5: Future Connection

Data and Variables:

- Distances in figure
- $\gamma = 58.656\ lb/ft^3$
- S= 20000 psi
- E = 0.85
- Y= 0.4

Materials:

- Coolant
- 1 Tank: Carbon Steel ASTM A106, Electric Resistance Welded
  - 4362 Gal

Procedure:

1. Use pressure determined in task 2 to find the thickness of the blind flange using equation 11.9 from the textbook.
2. Calculate amount of bolts and their sizes.

Calculations:

1.5 in pipe DN 40 $OD_f = 5$ in × by 25.4 = 127 mm

$$S_c = D_c \sqrt{\frac{0.125 P}{\sigma_s}} + C$$

S_c = Thickness σ_s - standard = 240 MPa = 240,000 kPa

D_c = Diameter

P = pressure on head = 7.795 psi = 53.746525 × by 6.895

$$S_c = .127 \sqrt{\frac{0.125\,(53.746525)}{240}} + C$$

$S_c = 0.6719366069 + C$

required thickness = 0.67 mm

Minimum commercially available thickness = 12.7 mm

Blind flange Plate = 5 in diameter, 0.5 in thick

5 in Blind flange Grainger.com 4 bolt holes 7/8 in th.

Item # 30WH03

ASME B16.5

$L = 2(.25 + .75 + .5 + .0125) + .0125$

$= 1.4425$ L minimum

Bolt Diameter 3/4 in

Bolt length minimum = 1.44 in

Nut 3/4 in

$L = 2(s + n + h + rf) + g$

s = 1/3 Diameter h = flange thickness g - gasket

n = bolt diameter rf = height of raised face

Summary:
The thickness of the blind flange is calculated as 0.092 in.

Analysis:
Applying the 2.5 factor of safety, the thickness becomes 0.25 in.

4. Specify wind load and weight of storage tanks for our civil engineer colleagues.

<u>Purpose:</u> Find the wind load and weight of the tanks to help design the required supports
<u>Sources:</u>

- Mott, R., Untener, J.A, "Applied Fluid Mechanics", 7th edition, Pearson Education, Inc (2015)
- https://www.journal-news.com/news/local/wind-gusts-mph-gust-recorded-high-winds-move-through-area/6WwUMeWK2GmirlzqRcCULJ/
- *https://www.omnicalculator.com/physics/wind-load*
- Project Document

<u>Design Considerations:</u>

- Wind load pressure using max wind recorded for each tank
- Weight pressure for each tank, assuming full liquid volume

<u>Drawings and Diagrams:</u>

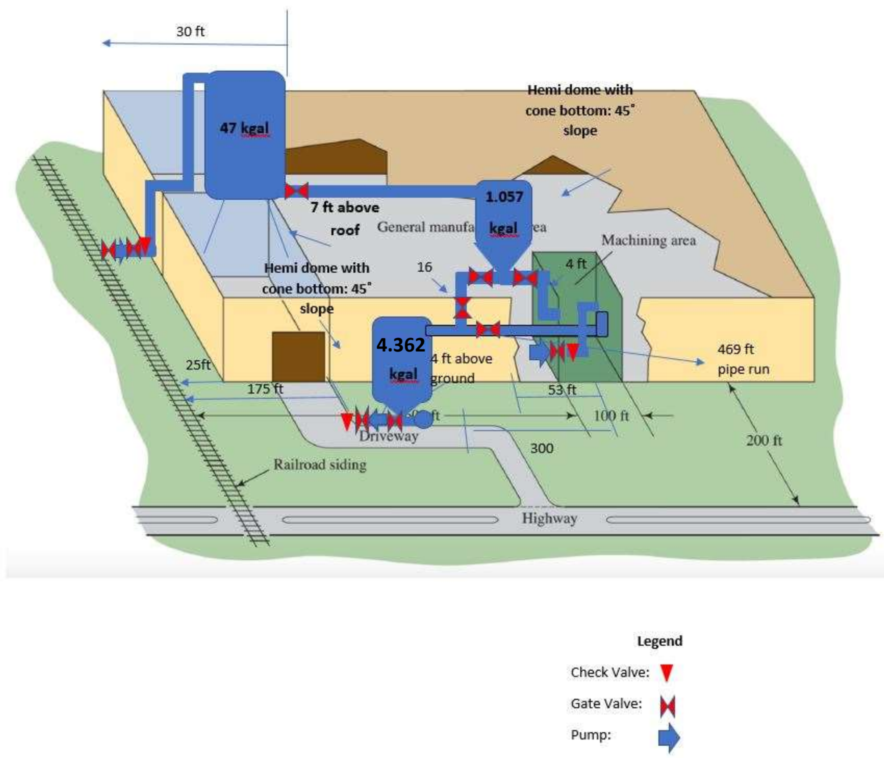


Figure 3: System Diagram

Data and Variables:

- Max recorded wind speed in location: Approx. 70 mph = 102.7 ft/s
- Average Air Pressure = 14.18 psi = 2041.92psf
- large tank, v= 45.000 gal
- medium tank, v= 4,000 gal
- small tank, v= 1,000 gal
- sg = 0.94
- density of steel = 480.709 lb/ft^3
- density of water = 1.94 slug/ft^3
- kinematic velocity of air @ 60 F v= 1.58E-4 ft^2/s

Materials:

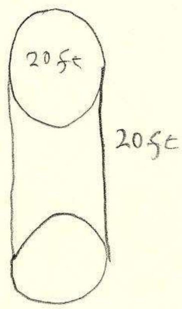
- Coolant
- Tank: Carbon Steel ASTM A106, Electric Resistance Welded
    - 45000 Gal
    - 4000 Gal
    - 1000 Gal

Procedure:

1. Find area of tank
2. Multiply area of tank by thickness of steel for tank wall to find total surface area of tank
3. Use surface area and density to find weight of empty tank
4. Find mass of fluid
5. Find interior volume of tank
6. Use data from 4 and 5 to find weight of fluid
7. Combine weight of tank with weight of fluid for total weight
8. Find area of tank
9. Multiply area of tank by thickness of steel for tank wall to find total surface area of tank
10. Use surface area and density to find weight of empty tank
11. Find mass of fluid
12. Find interior volume of tank
13. Use data from 11 and 12 to find weight of fluid
14. Combine weight of tank with weight of fluid for total weight
15. Find area of tank
16. Multiply area of tank by thickness of steel for tank wall to find total surface area of tank
17. Use surface area and density to find weight of empty tank
18. Find mass of fluid
19. Find interior volume of tank
20. Use data from 18 and 19 to find weight of fluid
21. Combine weight of tank with weight of fluid for total weight
22. Find Reynold's Number for local air at maximum recorded wind speed
23. Find Coefficient of Drag from table in figure 17.6 in textbook
24. Calculate wind load for tank
25. Find Reynold's Number for local air at maximum recorded wind speed
26. Find Coefficient of Drag from table in figure 17.6 in textbook
27. Calculate wind load for tank
28. Find Reynold's Number for local air at maximum recorded wind speed
29. Find Coefficient of Drag from table in figure 17.6 in textbook
30. Calculate wind load for tank

Calculations:

Wind Load + Weight



45,000 gal

sg = 0.94

0.385 in Steal = ,03208333

$A = (2\pi r)h$

r = 10 h = 20 A = 1256.637061 * ,032 = 40.31710567

$\rho = 7700\ kg/m^3 \div$ by 16.018 = 480.709 2021 lb/ft^3

40.317 * 480,709 = 19,380.8037

Sg to Mass

$0.94\left(1.94\ \frac{slug}{ft^3}\right) = 1.8236\ \frac{slug}{ft^3} = 58.71992\ \frac{lb}{ft^3}$

Int. Vol

$A = (2\pi r)h$ = 180085.8187 in^2

r = 119.8075 in h = 239.23 r = 9.983958335 h = 19.93583334

$V = \pi r^2 h$ = 6242.948993 ft^3

Weight = Vol * mass

= 6242.948993 (58.71992) = 366584.4654

Total

19,380.8037 + 366584.4654 = 385966.2691 lb

Wind load = area x pressure x cd

$R_e = \frac{VD}{\nu}$

ν_{air} @ 60°F = 1.58×10^{-4} ft²/s

Max Wind Speed in Dayton, Ohio ≈ 70 mph = 102.6669 ft/s

Pressure at Dayton, Ohio (avg) = 14.18 psi ≈ 2041.92 psf

$R_e = \frac{102.6669(20)}{1.58 \times 10^{-4}} = 12995810.13$

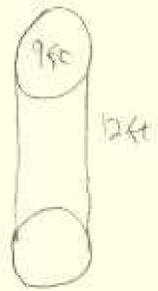
Extrapolating from fig. 17.6

$C_d = 1.24$

Area = 1256.637061 ft²

Wind load

= 1256.637 (2041.92) 1.24 = 3181780.911 **lbf**



4,200 gal

$sg = 0.94$

0.103 in Steel = 0.00858333 ft

$A = (2\pi r)h$

$r = 4.5$ ft $h = 12$ ft $A = 339.2920066$ ft^2 $\times .0085$

$V = A * wall = 2.912255259$ ft^3 $= 2.192255259$

$\rho = 7700$ kg/m^3 $\div 16.018 = 480.7092021$ lb/ft^3

$2.912 * 480.709 = 1399.947902$ lb

Sg to mass

$0.94(1.94 \frac{slug}{ft^3}) = 1.8236 \frac{slug}{ft^3} = 58.71992$ lb/ft^3

Int Vol

$A = (2\pi r)h = 338.4835097$ ft^2

$r = 4.495708335$ $h = 11.98283334$

$V = \pi r^2 h = 760.8615679$ ft^3

Weight = Vol * mass

$= 760.86 * 58.71992 = 44677.7304$ lb

Total Weight

$1399.947802 + 44677.7304 = 46077.6783$ lb

$V_{windmax} = 102.6609$ ft/s

$\nu_{air} @ 60°F = 1.58 \times 10^{-4}$ ft^2/s

$Re = \frac{VD}{\nu} = \frac{(102.6)(9)}{1.58 \times 10^{-4}} = 5847772.785$

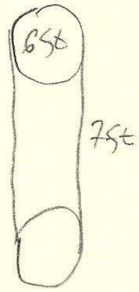
Extrapolate from fig 12.6 $C_d = 1.24$

Wind load = Area * P * Cd

P @ Dayton Ohio = 14.9 psi = 2041.92 psf

$= 339.292 * 2041.92 * 1.24$

$= 859080.843$ **lbf**



1,000 gal

$Sg = 0.94$

0.098 in Steel = .008166667

$A = (2\pi r)h = 131.94689(.00816) = 1.077566285$

$r = 3$ ft $h = 7$ ft

$\rho = 7700\ kg/m^3 \div 16.018 = 480.7092021\ lb/ft^3$

$W_{steel} = 1.077(480.709) = 517.9960289$

Sg to mass

$0.94(1.94\ \frac{slug}{ft^3}) = 1.8236\ \frac{slug}{ft^3} = 58.71992\ \frac{lb}{ft^3}$

It Vol

$V = \pi r^2 h = 196.9217629\ ft^3$

$r = 2.995916667$ $h = 6.983666667$

Weight = Vol (mass)

$= 196.921(58.71992) = 11563.20668$

Total

$11563.2 + 517.996 = 12081.20271$ lb

$V_{wind\,max} = 102.6609$ ft/s

$131.946(2041.92)1.24 = 334084.9703$ **lbf**

ν air @ $60°F = 1.58 \times 10^{-4}\ ft^2/s$

$R_e = \frac{102.66(7)}{1.58 \times 10^{-4}} = 4548267.722$

$C_d = 1.24$

Summary:
The weight of the 45,000 gal tank is 385,966 lb
The wind load on said tank is 3,181,780 lbf
The weight of the 4,000 gal tank is 46,077.68 lb
The wind load on said tank is 859,000 lbf
The weight of the 1,000 gal tank is 11,563 lb
The wind load on said tank is 334,084 lbf

Analysis:
The data for wind load and weight for each tank is listed above for proper support computation by the civil engineers on this project.

5. Consider that one of your storage tanks could fail. To repair it the company decided to drain it using a pumped system with same flow rate you estimated but this time the fluid will go into an open channel that will take the fluid to a location far from the plant (you have to select such location). You are now also in charge of the design of such open channel system to dump the fluid to that location. The design should include the path of the channel and its cross section.

<u>Purpose:</u>
To design an open channel that can take the fluid to a storage area should the holding tank fail

<u>Sources:</u>

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

<u>Design Considerations:</u>

- Shape of channel
- Depth of channel
- Length of channel
- Width of channel
- Slope of channel

<u>Drawings and Diagrams:</u>

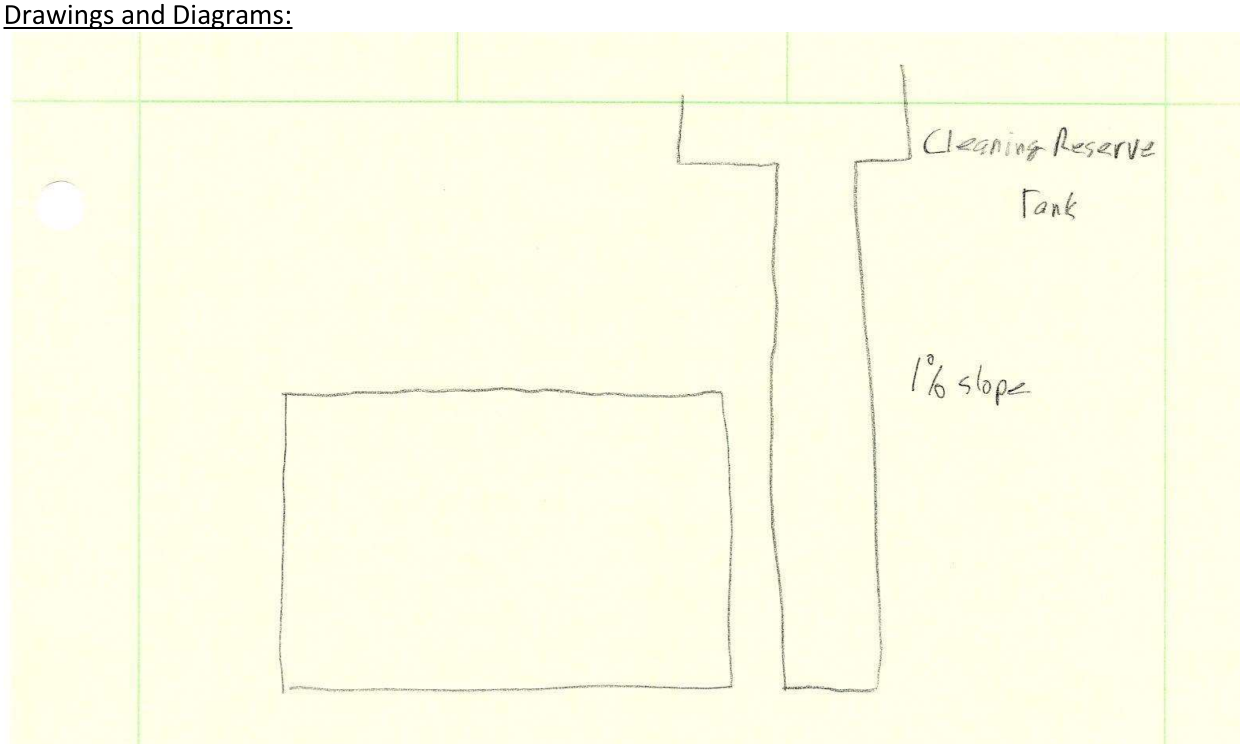


Figure 6: Open Channel

Data and Variables:

- Rectangular Channel
- 1% Slope
- Hydraulic Radius = 1.5
- Wetted Perimeter = 12
- roughness, n = 0.017
- Flow Rate = 9.6 ft/s
- Depth, y = 3 ft
- Width, b = 6 ft
- Area, A = 18 ft^2

Materials:

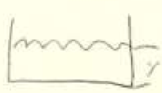
- Coolant
- Cement
- Concrete

Procedure:

1. Choose location for storage tank
2. Choose shape for channel
3. Choose slope for channel flow
4. Design channel based on fluid characteristics and required flow
5. Choose depth and width
6. Calculate flow rate and velocity to verify that it meets needs

Calculations:

9.6 ft/s   formed, unfinished Concrete $n = 0.017$



$b = 2y = T$

Rectangular Trench $A = 2y^2$

$y = 3 \quad b = 6$

$A = 18 \text{ ft}^2$

$Q = Av = 653.7384432 \text{ ft}^3/\text{s}$

$v = \frac{1.49}{n} R^{2/3} S^{1/2} = \frac{1.49}{.017} 1.5^{2/3} 0.1^{1/2} = 36.31680243 \text{ ft/s}$

$R = y/2 = 1.5$

$WP = 4y = 12$

$S = 0.1$

Summary:
The channel is rectangular, with a flow velocity of 36 ft per second.

Analysis:
The channel is more than capable of handling the flow rate of 9.6 ft/s that the pipe design calls for.

ii. Flow rate

4. Estimate the time required to fill and empty all tanks (you are supposed to fix them).
Specify the desired flow rate to fill and empty all tanks.

<u>Purpose:</u> Use the chosen fill times to determine the flow rates and the time to fill and empty each tank.

<u>Sources:</u>

- *Endmemo*. EndMemo. (n.d.). Retrieved November 22, 2021, from http://www.endmemo.com/sconvert/galus_minft3_s.php.
- Mott, R., Untener, J.A, "Applied Fluid Mechanics", 7th edition, Pearson Education, Inc (2015)
- Project Document

<u>Design Considerations:</u>

- Constant Properties
- Incompressible Fluids
- 8-hour work shifts
- Critical velocity of 9.6 ft/s

<u>Drawings and Diagrams:</u>
N/A

<u>Materials:</u>

- Coolant
- 3 Tanks: Carbon Steel ASTM A106, Electric Resistance Welded
  - 47000 Gal
  - 4362 Gal
  - 1057 Gal

<u>Data and Variables:</u>

- Large tank is 47 Kgal
- Reservoir is 1.057 Kgal
- Dirty tank is 4.362 Kgal

<u>Procedure:</u>

1. Use the chosen fill times and the tank volumes to determine the flow rate for each tank.
2. Use the flow rate and desired times to calculate time to empty and fill the tanks.

Calculations:

$$\frac{V}{T} = Q = \frac{47000\text{gal}}{180\text{ mins}} = 261.11\text{GPM}$$

$$\frac{V}{T} = Q = \frac{1057\text{ gal}}{20\text{ mins}} = 52.85\text{ GPM}$$

$$\frac{V}{T} = Q = \frac{4362\text{ gal}}{124} = 35.17\text{GPM}$$

Summary:

Table 1: Fill Times									
Tank Sizes		Desired Fill Times		Calculated Fill Flow Rate		Desired Empty Times		Calculated Empty Flow Rate	
47	Kgal	180	Mins	261.11	GPM	20	Mins	52.85	GPM
1.057	Kgal	60	Mins	17.61	GPM	20	Mins	52.85	GPM
4.362	Kgal	124	Mins	35.17	GPM	31	Mins	141	GPM

It will take 180 mins to fill the large tank with 45000 gallons of coolant and 52.85 minutes to empty 1000 gallons into the reservoir tank.

It will take 124 minutes to fill the dirty tank with 4000 gallons of coolant. It will take 52.85 mins to empty it.

It will take 60 minutes to drain the reservoir of 1000 gallons of coolant.

Analysis

All the tanks are equipped with gate valves so their flow rates can be altered depending on need. The flow rates are assumed the gate valve is 100% open.

iii. Piping Size

5. Specify the layout of the piping system, the material type and sizes of all pipes, and the lengths required. Please note that if choosing to have a system driven by gravity, the pipe calculations are different to the case of pumped systems. Please also remember that for a pumped system, the pipe size is chosen with the critical velocity criteria and the desired flow rate.

Purpose:

Determine pipe length, size, and types for the system.

Sources:

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

Design Considerations:

- Constant Properties
- Incompressible Fluids
- Critical velocity of 9.6 ft/s

Drawings and Diagrams:

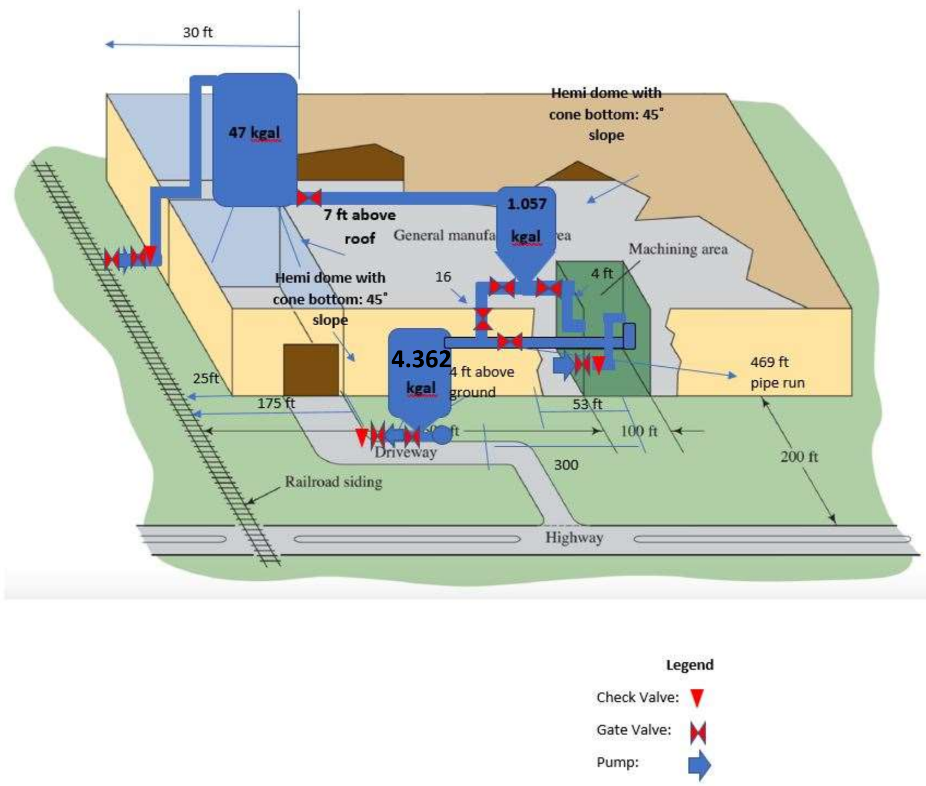


Figure 3: System Diagram

Data and Variables:

- Large tank: 261.11 GPM to fill 52.85 GPM to empty
- Dirty tank: 35.17 GPM to fill and 141 GPM to empty
- Feed tank: 17.61 GPM to fill and 52.85 GPM to empty

Materials:

- Coolant
- 3 Tanks: Carbon Steel ASTM A106, Electric Resistance Welded
  - 47000 Gal
  - 4362 Gal
  - 1057 Gal

Procedure:

1. Use chosen flow rate and critical velocity to determine the flow area of the pipes in each section using the Q= AV equation.

2. Find nearest pipe flow area on Appendix F of textbook (round up).

Calculations:

$$Q = VA \;\rightarrow\; \frac{Q}{V} = A$$

$$\frac{(261.11\,\text{GPM} \times 0.0022280093)}{9.6\,\text{ft/s}} = 0.0606\,\text{ft}^2$$

Summary:

Table 2: Pipe Size									
System	**Flow rate**		**Velocity**		**Pipe Area**		**Nominal Pipe Size from App. F**	
Train to Large Tank (Discharge)	261.11	GPM	9.6	ft/s	0.0606	ft^2	3.5	in
Large tank to Feed Tank (Discharge)	52.85	GPM	9.6	ft/s	0.012266	ft^2	1.5	in
Feed tank to Machine Area (Suction)	52.85	GPM	9.6	ft/s	0.012266	ft^2	1.5	in
Machine Pump to Dirty Tank (Discharge)	105.70	GPM	9.6	ft/s	0.024531	ft^2	2	in
Feed tank to Dirty Tank (Discharge)	52.85	GPM	9.6	ft/s	0.012266	ft^2	1.5	in
Dirty Tank to Trucks (Discharge)	141	GPM	9.6	ft/s	0.032724	ft^2	2.5	in

- The length of pipe from the railroad up to the large tank will be a horizontal length of 10 ft and vertical length of 52 feet. The pipe will be 3 ½ in nominal diameter.
- The length of pipe from the large tank to the reservoir will be 524 ft horizontally with no vertical distance. This pipe will have a 1.5 in nominal diameter.
- The pipe length from the reservoir to the machining area will be 4 ft vertically. This pipe will have a 1.5 in nominal diameter.
- The pipe length from the reservoir to the emergency dirty tank dump will be 6 ft. This pipe will have a 1.5 in nominal diameter.
- The pipe from the machining area pump to the dirty reservoir will have a height of 16 ft and a horizontal length of 469 ft. This pipe will have a 2 in nominal diameter.
- The pipe from the dirty tank to the pump that feeds the trucks will be 3 ft long and have a 2.5 in nominal diameter.

Analysis:
These dimensions assume an industry standard critical velocity of 9.6 ft/s. Carbon Steel ASTM A106 will be used for the pipe to aid in welding of similar metals.

6. Specify the number, types, material, and size of all valves, elbows, and fittings. Please note that if choosing to have a system driven by gravity, the pipe calculations are different to the case of pumped systems. Please also remember that for a pumped system, the pipe size is chosen with the critical velocity criteria and the desired flow rate. (to final report section 5.f.iii)

   Purpose: Create a parts list of required valves, elbows, and fittings.

   Sources:

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

   Design Considerations:

   - Constant Properties
   - Incompressible Fluids
   - Critical velocity of 9.6 ft/s

Drawings and Diagrams:

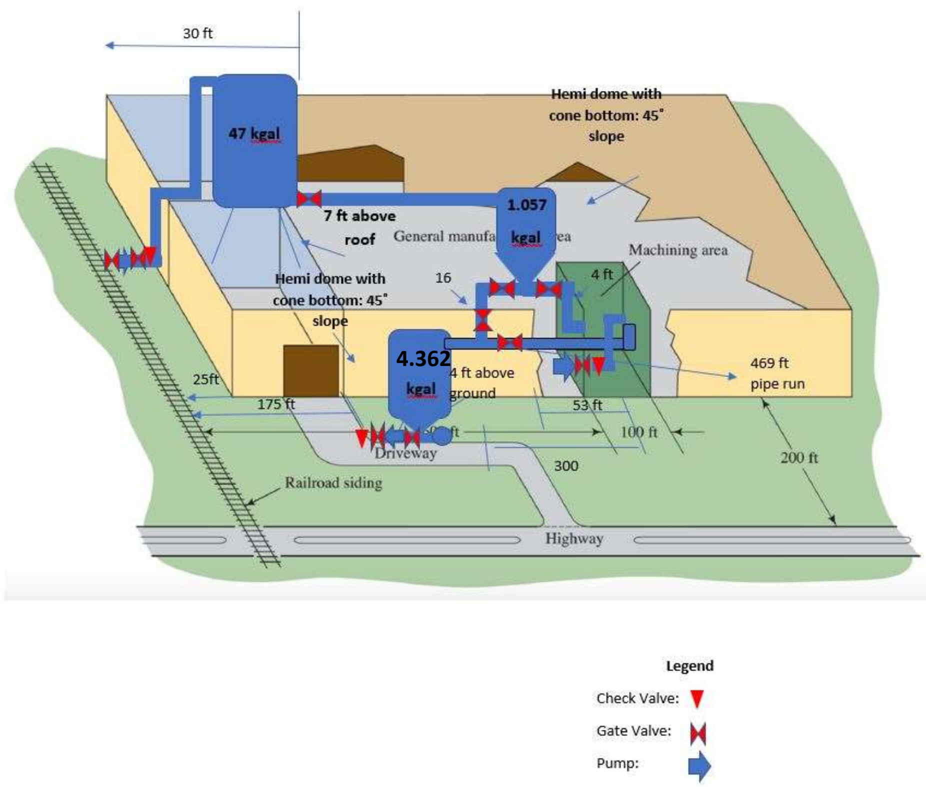


Figure 3: System Diagram

Data and Variables:

- N/A

Materials:

- Coolant
- Carbon Steel ASTM A106

Procedure:

1. Use piping layout from diagram to list required parts.

Calculations:

Sizes are based on pipe sizes calculated in task 5.

Summary:

- Train car to large tank
  - 2 Gate valves for 3.5 in pipe
  - 1 check vale for 3.5 in pipe
  - 4 Short Radius Elbows for 3 ½ in pipe
  - 20 ft hose 3.5 in
- Large tank to feed tank
  - 1 Gate valve for 1.5 in pipe
- Feed tank to machine area
  - 1 Standard Tee for 1.5 in pipe
  - 2 Short Radius Elbows for 1.5 in pipe
  - 1 Gate valve for 1.5 in pipe
- Feed tank to dirty tank
  - 1 Standard Tee for 1.5 in pipe (same tee used in previous system)
  - 1 Short Radius Elbow for 1.5 in pipe
  - 2 Gate valve for 1.5 in pipe
  - 1 Expander (1.5 in to 2 in)
  - 1 standard tee for 2 in pipe
- Machine pump to dirty tank
  - 4 Short Radius Elbows for 2 in pipe
  - 2 Gate valves for 2 in pipe
  - 1 check valve for 2 in pipe
  - 1 standard tee for 2 in pipe (same tee used in previous system)
- Dirty tank to Trucks
  - 1 Standard Tee for 2.5 in pipe
  - 2 Gate valves for 2.5 in pipe
  - 1 check valve for 2.5 in pipe
  - 20 ft hose 2.5 in

Analysis:

All material will match material of piping indicated in task 5.

9. Specify pipe wall thickness (schedule). You need for this the maximum operating pressure of the system. In a pumped system this pressure is typically at the exit of the pump.

Purpose:
Determine pumps required to run the system

Sources:

- https://www.gbrx.com/media/1466/tank29000.pdf
- Mott, R., Untener, J.A, "Applied Fluid Mechanics", 7th edition, Pearson Education, Inc (2015)
- Project Document

Design Considerations:

- Refer to task 1 design considerations.
- Velocity in the pipes is 9.6 ft/s (industry standard for flow rate of 100 GPM)
- Tanks are at atmospheric pressure

Drawings and Diagrams:

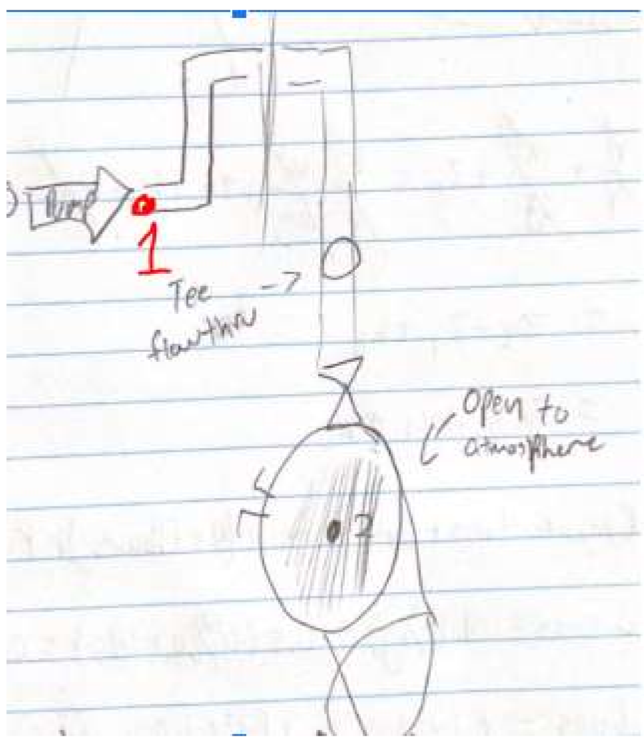


Figure 7: Pump from machinery area to dirty tank

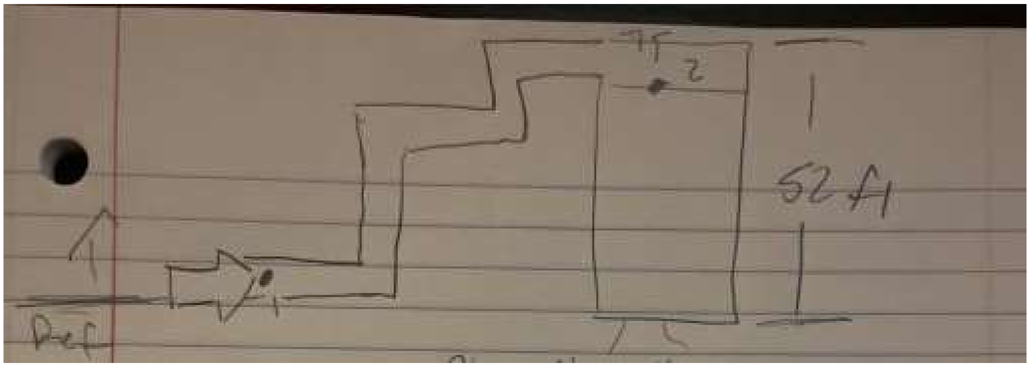


Figure 8: Pressure for pump in train to large tank exit

Data and Variables:
- γ=58.656 lb/ft^3
- ρ= 1.8236 slug/ft^3
- Length of 3 ½ in pipe= 84 ft
- Length of 2 in pipe= 485 ft
- S= 20000 psi
- E = 0.85
- Y= 0.4

Materials:
- Coolant
- Carbon Steel ASTM A106

Procedure:
1. Determine maximum operating pressure.
2. Plug pressure in to equation 11.9 from the textbook to solve for thickness, minimum thickness, and nominal thickness
3. Use the tables in the appendix for steel pipe dimensions to determine which schedule lines up with the nominal wall thickness

Calculations:

Pressure in machinery area pump exit:

$$\frac{P_A}{\gamma} + \cancel{\frac{V_A^2}{2g}} + \cancel{z_A} = \cancel{\frac{P_B}{\gamma}} + \cancel{\frac{V_B^2}{2g}} + z_B + h_L + h_a$$

$$P_A = \gamma(z_B + h_L + h_a)$$

$$\left(58.656\ lb/ft^3(16ft + 96.025ft + 245.285ft)\right)/144$$

$$= 145.54\ psi$$

Pressure for pump in train to large tank exit:

$$P_a = \frac{58.656\ \frac{lb}{ft^3}(42ft + 6.77ft + 58.31\ ft)}{144} = 43.619\ psi$$

Pipe thickness example calculation:

$$t = \frac{PD}{2(SE + PY)} = \frac{(43.62 psi)(4 in)}{2(20000 psi \times 0.85 + 43.619(0.4))}$$

$$t = 0.00512\ in$$

$$t_{min} = t + 0.08\ in = 0.00512\ in + 0.08\ in$$

$$t_{min} = 0.08512\ in$$

$$t_{nom} = \frac{t_{min}}{0.875} = \frac{0.08512\ in}{0.875}$$

$$t_{nom} = 0.097\ in$$

Thickness of pipes in each system:

Thickness of pipe from big tank to 1kg	
t	0.0002599
tmin	0.0802599
tnom	0.0917256
p	4.6520833
D	1.9
S	20000
E	0.85
Y	0.4

Thickness of pipe from train to big tank	
t	0.005126386
tmin	0.085126386
tnom	0.097287299
p	43.61900558
D	4
S	20000
E	0.85
Y	0.4

Thickness of pipe from feed tank to machine area	
t	0.0001532
tmin	0.0801532
tnom	0.0916036
p	2.7413533
D	1.9
S	20000
E	0.85
Y	0.4

Thickness of pipe from machine pump to dirty tank	
t	0.010131872
tmin	0.090131872
tnom	0.103007854
p	145.5424617
D	2.375
S	20000
E	0.85
Y	0.4

Thickness of pipe from feed tank to dirty tank	
t	0.0001532
tmin	0.0801532
tnom	0.0916036
p	2.7413533
D	1.9
S	20000
E	0.85
Y	0.4

Thickness of pipe from dirty tank to trucks	
t	0.000394887
tmin	0.080394887
tnom	0.091879871
p	4.670484028
D	2.875
S	20000
E	0.85
Y	0.4

Summary:

The maximum operating pressure in the piping from the train to the large tank is 43.62 psi.
The maximum operating pressure in the piping from the large tank to the feed tank is 4.65 psi.
The maximum operating pressure in the piping from the feed tank to the machine area is 2.74 psi.
The maximum operating pressure in the piping from the machine area to the dirty tank is 145.54 psi.
The maximum operating pressure in the piping from the feed tank to the dirty tank is 2.74 psi.
The maximum operating pressure in the piping from the dirty tanks to the trucks is 4.67 psi.
The nominal wall thickness for the piping from the train to the large tank is 0.0973 in.
The nominal wall thickness for the piping from the large tank to the feed tank is 0.0917 in.
The nominal wall thickness for the piping from the feed tank to the machine area 0.0916 in.
The nominal wall thickness for the piping from the machine area to the dirty tank is 0.103 in.
The nominal wall thickness for the piping from the feed tank to the dirty tank is 0.0916 in.
The nominal wall thickness for the piping from the dirty tanks to the trucks is 0.0918 in.

Analysis:

Because all wall thicknesses are well below the listed wall thicknesses for the respective pipe sizes in the schedule 40 table, schedule 40 pipe will be used for the entire system.

10. Check your design for water hammer problems. Check if the pipe you selected can hold such over-pressure, if not, propose the use of a water-hammer arrestor by specifying the pressure that it will handle.

Purpose:

To find if the pipes in the design can withstand the effects of water hammer

Sources:

- https://www.engineeringtoolbox.com/wrought-steel-pipe-bursting-pressure-d_1123.html
- Mott, R., Untener, J.A, "Applied Fluid Mechanics", 7th edition, Pearson Education, Inc (2015)
- Project Document

Design Considerations:

- Refer to step 9 pipe design information

Data and Variables:

- Longest pipe: 524 ft
- Schedule 40 steel pipe

Materials:

- 2 in steel pipe
- 3 ½ in steel pipe

Procedure:

1. Determine a speed for valve closing
2. Find pressure caused by said valve closing speed in longest pipe

Calculations:

Find pressure change in pipe due to sudden valve closure

Water Hammer

$\Delta p = .070\, \Delta v\, \ell / \Delta t$ Longest Pipe 524 ft

$.070\,(9.6\text{ ft/s})(524)/(0.1) = 3521.28$ psi Closing time 0.1s

Sch 40 pipe

Bursting Pressure 2 in 5145 psi

3 ½ in 5610 psi

Summary:

The maximum pressure in the pipes from the water hammer phenomenon is 3521 psi.

The burst pressure in the smaller 2 in pipe is 5185 psi, while the burst pressure in the larger 3 ½ in pipe is 5610 psi.

Analysis:

Using a sudden closure speed of 0.1 seconds, the water hammer phenomenon reached a value of 3521 psi, well below the bursting pressures of either pipe being used for this design, therefore, no arrestor is required.

iv. Provide pipeline support info

13. For one particular pipe system, decide the type of supports and determine the force acting upon each support. Your work here includes the distance between supports so the pipe does bend much. Our civil engineer colleagues need this.

Purpose:

Determine the amount of hanger required to support one stretch of pipe. Also determine the type of hanger and distance the hangers are apart.

Sources:

- *Hanger support spacing - rod sizes horizontal pipes*. Engineering ToolBox. (n.d.). Retrieved November 29, 2021, from https://www.engineeringtoolbox.com/piping-support-d_362.html.
- Mott, R., Untener, J.A, "Applied Fluid Mechanics", 7th edition, Pearson Education, Inc (2015)
- Project Document

Design Considerations:

- Refer to task 1 design considerations.
- Velocity in the pipes is 9.6 ft/s
- Tanks are at atmospheric pressure

Drawings and Diagrams:

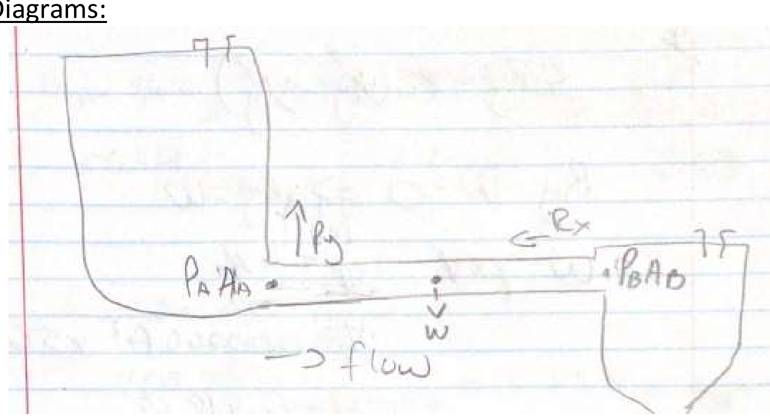


Figure 9: Piping Support FBD

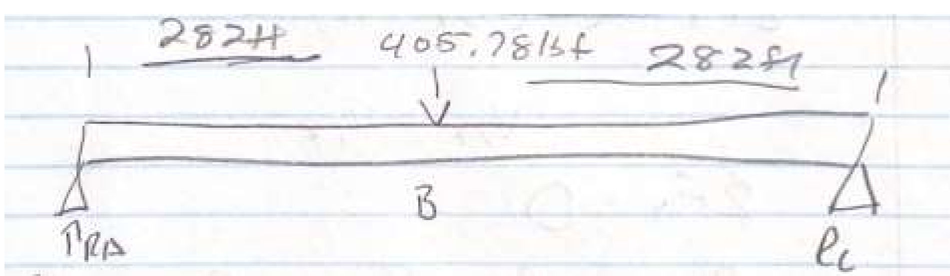


Figure 10: Piping Simply Supported Reaction Forces FBD

Data & Variables:

- ρ= 1.8236 slug/ft[3]
- Length of 1 ½ in pipe= 564 ft
- V =9.6 ft/s
- Q= 52.85 GPM
- P_a= 1122.66 psf
- A= 0.012266 ft^2
- γ=58.656 lb/ft^3

Materials:

- Coolant
- Carbon Steel ASTM A106 piping

Procedure:

1. Determine reaction forces in the X and Y directions
2. Use pressure in bottom of large tank found in task 9
3. Use Bernoulli's to solve for pressure at inlet of the feed tank.
4. Use chart found in sources to determine the spacing for 1.5 in schedule 40 steel pipe
5. Divide Ry by number of hangers

Calculations:

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

X

$$\Sigma F_x = \rho Q(V_{2x} - V_{1x})$$

$$V_{2x} = V_{1x}$$

$$\Sigma F_x = 0$$

$$P_A A_A - R_x - P_B A_B = 0$$

$$R_x = (P_A - P_B)A$$

$P_A = 1122.55$ psf
$P_B = 1026.634$ psf

$$R_x = (1122.55 \text{ psf} - 1026.634 \text{ psf})\,0.012266 \text{ ft}^2$$

$$\boxed{R_x = 1.176 \text{ lbf}}$$

y

$\Sigma F_y = \rho Q(V_{2y} - V_{1y})$

$R_y - W = 0 \Rightarrow R_y = W$

$W = \gamma \times \forall \quad \forall = A \times L$

$\forall = 0.012266\ ft^2 \times 564\ ft$

$\forall = 6.918\ ft^3$

$W = 58.656\ lb/ft^3 \times 6.918\ ft^3 = 405.78\ lbs$

$\boxed{R_y = 405.78\ lbf}$

$\circlearrowleft M_A =$

$R_L \times (564\ ft) = 405.78\ lbf\ (282\ ft)$

$R_L = 202.89\ lbf$

$\circlearrowleft M_L = R_A \times (564\ ft) = 405.78\ lbf\ (282\ ft)$

$R_A = 202.89\ lbf$

Hanger spacing every 9 ft, $564\ ft / 9\ ft = 62.667$ hangers ≈ 63

$564\ ft / 63\ hangers = 8.95\ ft$ apart

force on each hanger:

$405.78\ lbs / 63\ hangers = 6.44\ lbs$ per hanger.

Summary:

Per the hanger support spacing chart noted in the sources, the pipe needs a support every 9 feet. With this distance there is not a whole number of hangers. The calculated 62.67 hangers were rounded up to 63 hangers. This number of hangers required the hangers to be placed about 8.95 ft apart from each other. Each hanger has 6.44 lbf acting on it from the pipe.

Analysis:

Rounding up to 63 hangers is safer than rounding down to 62 hangers because it provides extra support. The chart in the sources recommends 3/8” steel rods to be used to support the pipe.

v. Energy Losses

7. Develop the hydraulic analysis of all parts of the system; this includes energy losses due to friction and minor losses. You should list the energy losses per section(s) of each of the coolant sub-systems. For this task, you are allowed to use software but one of the coolant sub-systems MUST be calculated by hand and compare against the software results.

Purpose:
Calculate all the minor losses in system.

Sources:

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

Drawings and Diagrams:

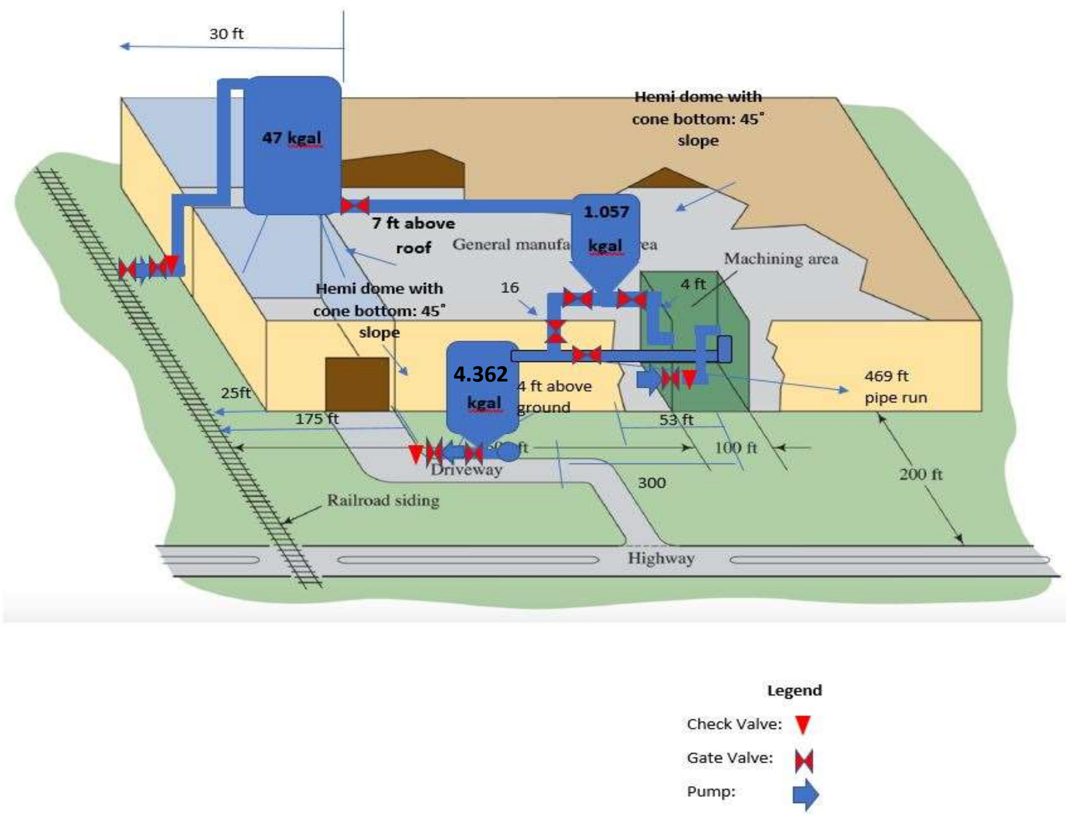


Figure 3: System Diagram

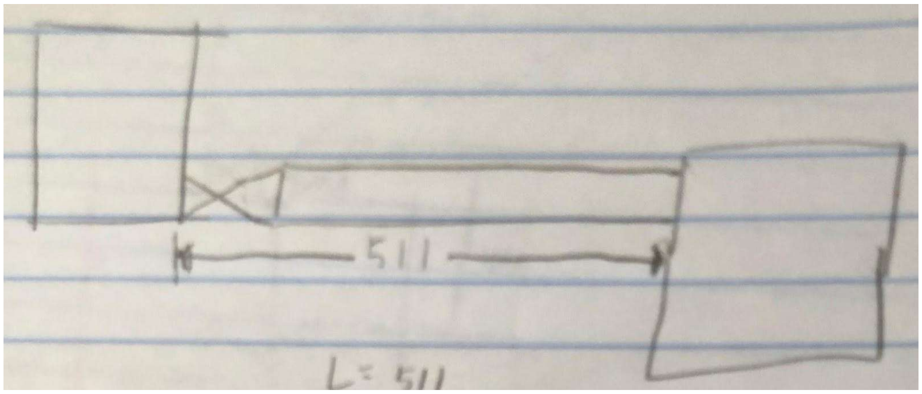


Figure 11: Large Tank to Reservoir Tank

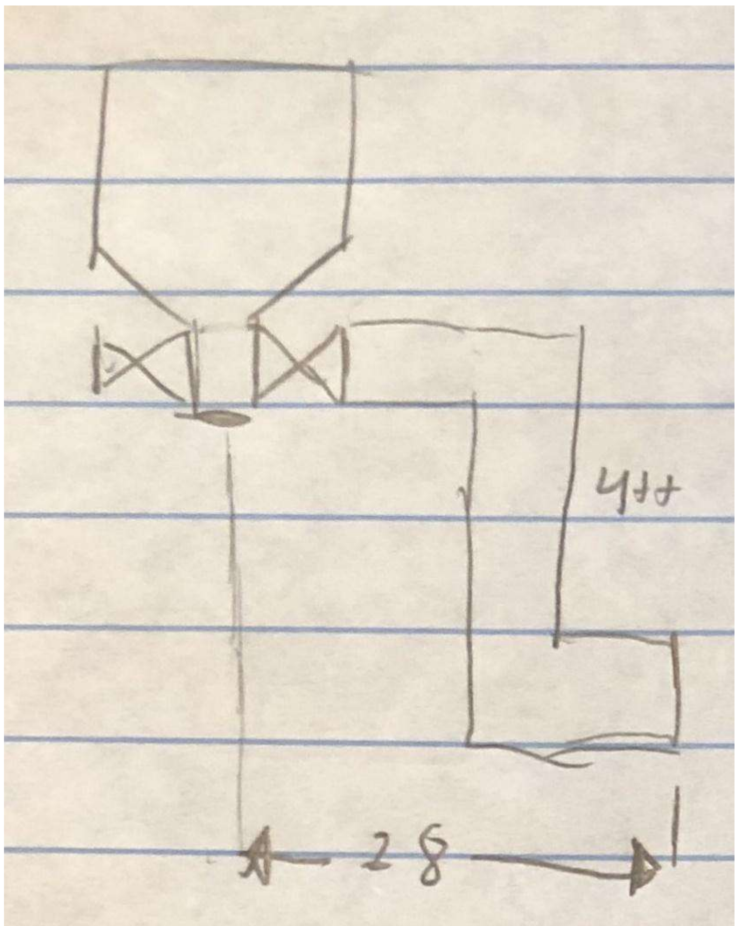


Figure 12: Reservoir Tank to Equipment

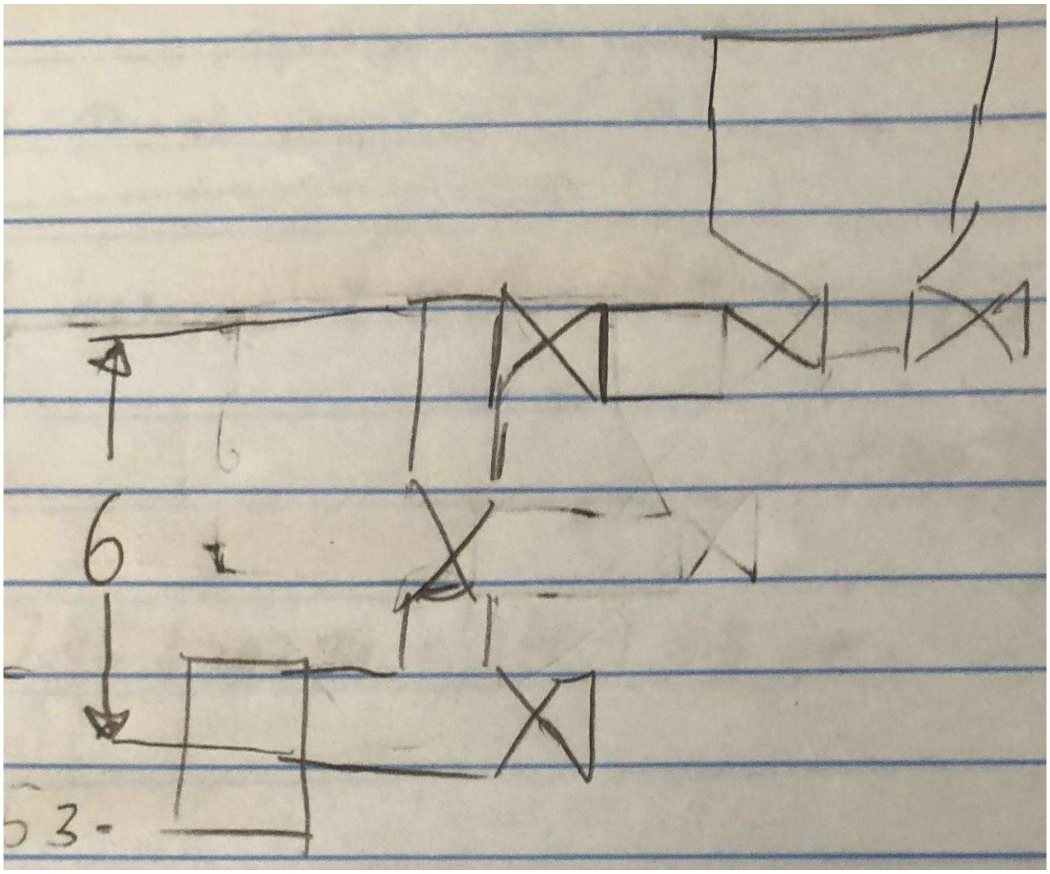


Figure 13: Reservoir Tank to Dirty Tank

Data and Variables:

- γ=58.656 lb/ft3
- V1=V2=0 ft/s
- Vp= 9.6 ft/s
- g= 32.2 ft/s2
- Di of 2 in pipe= 0.1723 ft
- ρ= 1.8236 slug/ft3
- Length of 3 ½ in pipe= 84 ft
- Length of 2 in pipe= 485 ft

Materials:

- Coolant
- Carbon Steel ASTM A106

Procedure:

Large Tank to Reservoir Tank:

1. Set points at the surface of each tank to assume zero velocity.
2. Solve Bernoulli's for pump head
3. Solve for entrance loss, exit loss, elbow loss, and pipe loss.
4. Add together total losses

Reservoir Tank to Equipment:

1. Set points at the surface of each tank to assume zero velocity.
2. Solve Bernoulli's for pump head
3. Solve for entrance loss, exit loss, elbow loss, and pipe loss.
4. Add together total losses

Reservoir Tank to Dirty Tank:

1. Set points at the surface of each tank to assume zero velocity.
2. Solve Bernoulli's for pump head
3. Solve for entrance loss, exit loss, elbow loss, and pipe loss.
4. Add together total losses

Total loss minus losses in pump sections (8):

1. Add together total losses

Calculations:

$$h_a = \cancel{\frac{P_1}{\gamma_1}} + \cancel{\frac{V_1^2}{2g}} + \cancel{z_1} = \cancel{\frac{P_2}{\gamma_1}} + \cancel{\frac{V_2^2}{2g}} + \cancel{z_2} + h_L$$

$$h_a = h_L$$

h_L = entrance Loss + exit Loss + valve Loss + friction Loss + Elbow loss

$$\text{Entrance Loss} = K\left(\frac{V_1^2}{2g}\right) = .5\left(\frac{9.6^2}{64.4}\right) = 0.72\ \text{ft} = EL$$

$$\text{Exit Loss} = k\left(\frac{V_2^2}{2g}\right)\ 1\left(\frac{0^2}{64.4}\right) = 0\ \text{ft} = ExL$$

$$\text{Gate Valve Loss} = k\left(\frac{V_p^2}{2g}\right) = .44\left(\frac{9.6^2}{64.4}\right) = 0.64\ \text{ft} = GL$$

$$\text{Elbow loss} = K\left(\frac{V_p^2}{2g}\right) = .57\left(\frac{9.6^2}{64.4}\right) = 0.8\ \text{ft} = EbL$$

$$\text{T. Lose} = k\left(\frac{V_p^2}{2g}\right)\ 1.1\left(\frac{9.6^2}{64.4}\right)\ 1.59\ \text{ft} = TL$$

Pipe Loss:

$$N_R = \frac{VD\rho}{\eta} = \frac{9.6(.172)(1.82)}{2.6\times10^{-3}}$$

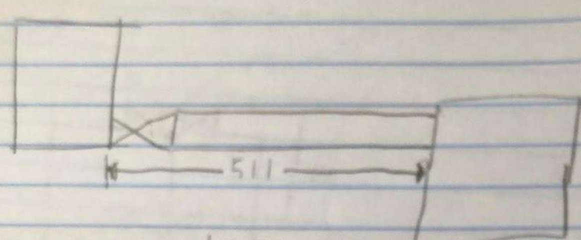
$$N_R = 1144.09 < 2000 = Lam$$

$$h_L = \frac{32\eta L V}{\gamma D^2} = \frac{32(2.6\times10^{-3})L(9.6)}{58.65(0.1723)}$$

L left becouse it is an independent Variable

$$h_L = \frac{0.8064L}{1.7217} = 0.4685L$$

Section 1



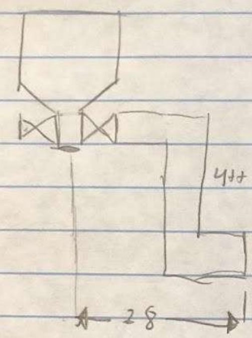
$L = 511$

$$h_L = 0.4685(511) = 239.4$$

$$h_{L\,total} = EL + GL + h_L + ExL$$

$$h_{L\,total} = 0.72 + 0.64 + 239.4 + 0 = 240.76\ ft$$

Section 2



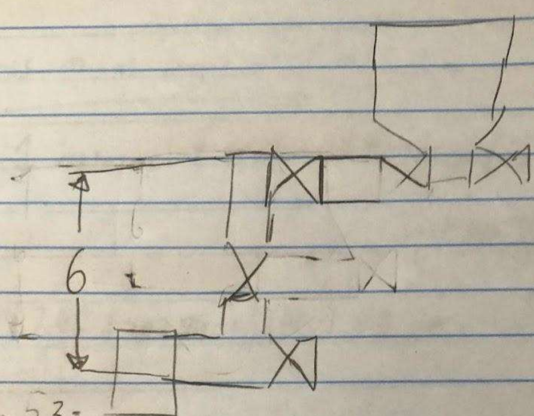
$L = 4 + 28 = 32\text{ ft}$

$hL = 0.4685(32) = 14.99$

$hL_{TOTAL} = EL + GL + 2EbL + hL + EXL$

$hL_{TOTAL} = .72 + .64 + 2(.8) + 14.99 + 0 = 17.95\text{ ft}$

Section 3



L 485 − 53 −

$L = 485 - 53 + 6 = 422$

$hL = 0.4685(422) = 197.7$

$hL_{TOTAL} = EL + 3GL + EbL + TL + EXL + hL$

$= .72 + 3(.64) + .8 + 1.59 + 0 + 197.7 = ?$

202.74 ft

Total Loss minus pipe sections =

section 1 hLtotal + section 2 hLtotal + section3 hLtotal

240.76 + 17.95 + 202.74 = 461.45 ft

Table 3: Energy Losses

System	Nr	D/e	f	ft	hl_elb	hl_gv	hl_exp	hl_tees (branch flow)	hl_pipes	hl_exit	hl_ent	hl_cv	hl_t	Pump Head (ft)	Pressure (psi)
Train to Large Tank (Discharge)	1185185	23333.33	0.0123	0.0103	1.17	0.12	0	0.00	4.17	1.43	0.00	1.47	8.36	52.86	42.05
Train to Large Tank (suction)	1185185	23333.33	0.0123	0.0103	1.17	0.12	0	0.00	1.21	0.00	0.00	0.00	2.50		
Large tank to Feed Tank (Discharge)	507936.5	10000.00	0.0144	0.0120	0.00	0.14	0	0.00	93.74	1.43	0.72	0.00	96.03		
Feed tank to Machine Area (Suction)	507936.5	10000.00	0.0144	0.0120	0.69	0.14	0	1.03	0.66	0.00	0.00	0.00	2.51		
Machine Pump to Dirty Tank (Suction)	677248.7	13333.33	0.0137	0.0114	0.32	0.13	0	0.00	0.35	0.00	0.00	0.00	0.81		
Machine Pump to Dirty Tank (Discharge)	677248.7	13333.33	0.0137	0.0114	0.97	0.39	0	0.32	56.89	1.43	0.00	1.62	61.63	79.87	64.15
Feed tank to Dirty Tank (Discharge)	507936.5	10000.00	0.0144	0.0120	0.34	0.27	0.544	2.06	30.59	1.43	0.00	0.00	35.24		
Dirty Tank to Trucks (Discharge)	846560.8	16666.67	0.0131	0.0109	0.00	0.12	0	0.00	1.80	0.00	0.00	1.56	3.48	14.81	13.97
Dirty Tank to Trucks (Suction)	846560.8	16666.67	0.0131	0.0109	0.00	0.12	0	0.94	0.27	0.00	0.00	0.00	1.33		

Summary:

The system loses the most energy in sections with long stretches of pipe. Due to this the system losses 96,026 ft between the large tank and the feed tank, 59.75 ft between the machine pump and the dirty tank, and 35.24 ft between the feed tank and the dirty tank. The other losses are very minimal.

Analysis:

Most losses are from Friction in the pipes but that also being said with complexity of this system these losses are negatable.

vi. Pump Selection

8. How many pumps do you need? What are the requirements (this is, you have to provide pump head and flow rate) of each pump? For this task, you are allowed to use software but one of the coolant sub-systems MUST be calculated by hand and compare against the software results.

<u>Purpose:</u>
Determine pumps required to run the system

<u>Sources:</u>

- https://www.gbrx.com/media/1466/tank29000.pdf
- Mott, R., Untener, J.A, "Applied Fluid Mechanics", 7th edition, Pearson Education, Inc (2015)
- Project Document

<u>Design Considerations:</u>

- Constant Properties
- Incompressible Fluids
- Standard tanker car is 8 ft in diameter and will be approximately 2 feet above the ground.
- Standard tanker cars are vented to atmosphere during unloading
- Assume atmospheric pressure at machinery outlet
- Standard critical velocity of 9.6 ft/s assumed at machinery outlet

<u>Drawings and Diagrams:</u>

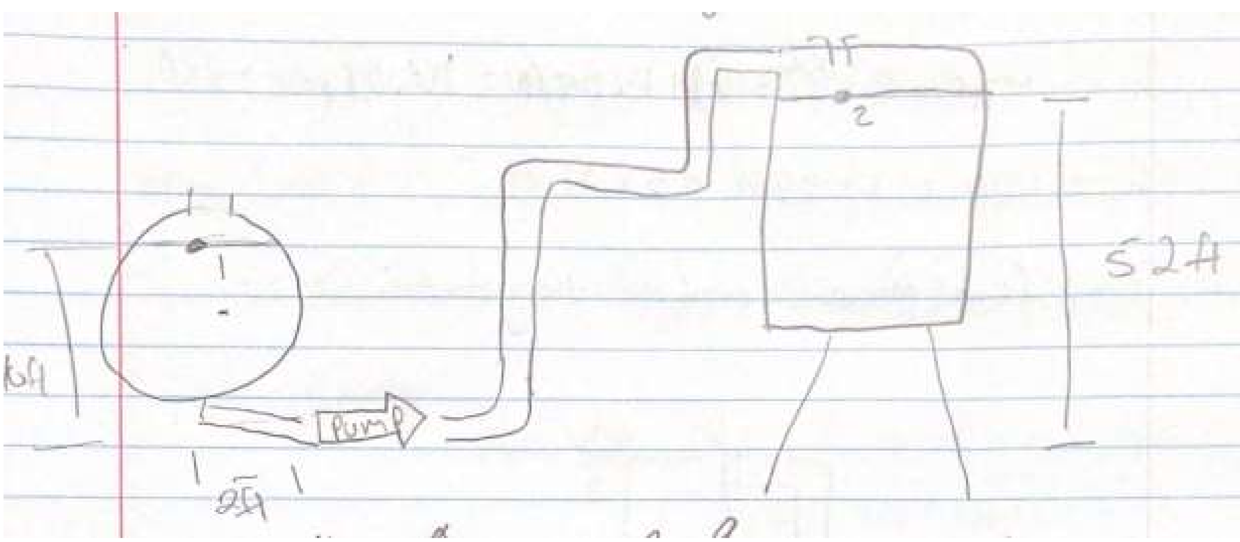


Figure 14: Pump from Train to Large Tank

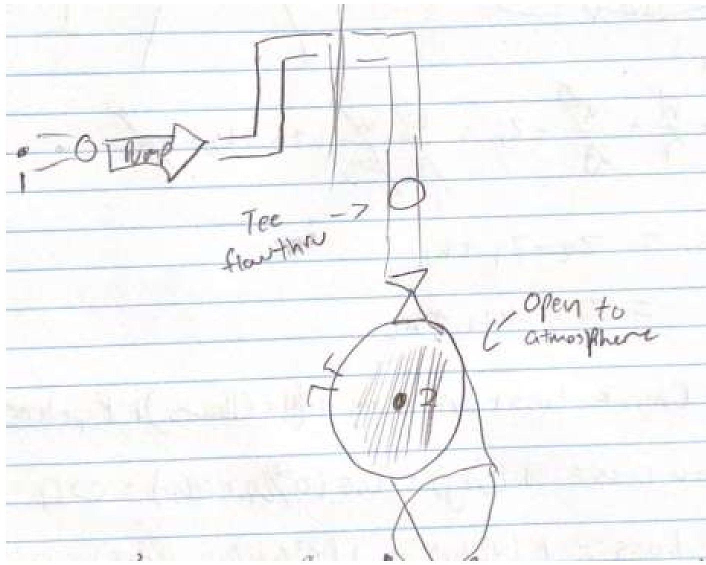


Figure 15: Pump from Machinery Area to Dirty Tank

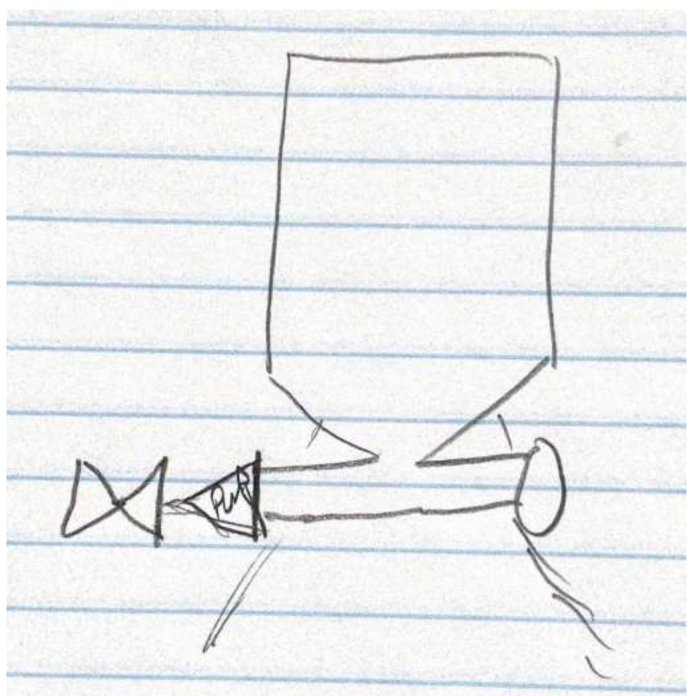


Figure 16: Pump from Dirty Tank to Trucks

Data and Variables:

- γ=58.656 lb/ft^3
- V_1=V_2=0 ft/s
- V_p= 9.6 ft/s
- g= 32.2 ft/s^2
- D_i of 3 ½ in pipe= 0.2975 ft
- D_i of 2 in pipe= 0.1723 ft
- ρ= 1.8236 slug/ft^3
- Length of 3 ½ in pipe= 84 ft
- Length of 2 in pipe= 485 ft

Materials:

- Coolant
- Carbon Steel ASTM A106

Procedure:

Pump from train to large tank:

1. Set points at the surface of each tank to assume zero velocity.
2. Solve Bernoulli’s for pump head
3. Solve for entrance loss, exit loss, elbow loss, and pipe loss.
4. Add together total losses and plug in to Bernoulli’s to solve for pump head.

Pump from machining area to dirty tank:

1. Set points at the outlet of machining area and surface of dirty tank
2. Solve Bernoulli’s for pump head
3. Solve for exit loss, elbow loss, tee loss, and pipe loss.
4. Add together total losses and plug in to Bernoulli’s to solve for pump head.

Calculations:

$$h_a + \cancel{\frac{P_1}{\gamma}} + \cancel{\frac{v_1^2}{2g}} + z_1 = \cancel{\frac{P_2}{\gamma}} + \cancel{\frac{v_2^2}{2g}} + z_2 + h_L + \cancel{h_R}$$

$$h_a = z_2 - z_1 + h_L$$

$$= 52ft - 10ft + h_L$$

h_L = Entrance Loss + Exit Loss + (4x Elbows) + Pipeloss

Entrance Loss = $k(v_3^2/2g) = 0.5(0^2 ft/s / 64.4 ft/s^2) = 0 ft$

Exit Loss = $K(v_1^2/2g) = 1(0^2 ft/s / 64.4 ft/s^2) = 0 ft$

Pipe Loss: $N_R = \frac{vD\rho}{\eta} = \frac{(9.6 ft/s)(0.2975 ft)(1.85 slugs/ft^3)}{(2.625 \times 10^{-3} lb\text{-}s/ft^2)}$

$N_R = 1994.08 < 2000$, Laminar flow

$f_L = \frac{64}{N_R} = \frac{64}{1984.08} = 0.032$

$h_L = \frac{32\eta L v_p}{\gamma D^2} = \frac{32(2.625 \times 10^{-3} lb\text{-}s/ft^2)(84 ft)(9.6 ft/s)}{(59.656 lb/ft^3 \times (0.2975 ft)^2)}$ —>

Elbowloss $= k(v_p^2/2g) = 0.57(9.6 ft/s^2 / 64.4 ft/s^2) = 0.816 ft$

Pipeloss = 13.05 ft

$h_{L\,total} = 0 + 0 + 4(0.816)ft + 13.05 ft = 16.3 ft$

$h_a = 42 ft + 19.99 ft = 58.31 ft.$

Pump from Machine area to dirty tank:

$$h_a + \cancel{\frac{P_1}{\gamma}} + \frac{V_1^2}{2g} + \cancel{Z_1} = \cancel{\frac{P_2}{\gamma}} + \cancel{\frac{V_2^2}{2g}} + Z_2 + h_L + \cancel{h_R}$$

$$h_a = -\frac{V_1^2}{2g} + Z_2 + h_L$$

$$= \frac{-9.6^2 \text{ ft/s}}{2(32.2 \text{ ft/s}^2)} + 16 \text{ ft} + h_L$$

h_L = Elbow loss + Tee loss + Pipe loss + Exit loss + Gate Valve loss

Exit Loss $= 1.0\,(v_1^2/2g) = 1\,(9.6^2 \text{ ft/s} / 64.4 \text{ ft/s}^2) = 1.43 \text{ ft}$

Elbow loss $= K\,(v_p^2/2g) = 0.57\,(9.6^2 \text{ ft/s} / 64.4 \text{ ft/s}^2) = 0.816 \text{ ft}$

Tee loss (flow thru) $= k\,(v_p^2/2g) = 1.11\,(9.6^2 \text{ ft/s} / 64.4 \text{ ft/s}^2) = 1.59 \text{ ft}$

$$h_L = h_{L\,els} + h_{L\,gv} + h_{L\,tees} + h_{L\,pipes} + h_{L\,exit} + h_{L\,cv} = 62.44 \text{ ft}$$

$$h_{L\,pipes} = f\left(\frac{L}{D}\right)\left(\frac{V^2}{2g}\right) = 0.0137\left(\frac{485.4}{0.167 \text{ ft}}\right)\left(\frac{92.16 \text{ ft/s}}{64.4 \text{ ft/s}}\right)$$

$$= 56.98$$

$$h_a = \frac{9.6^2 \text{ ft/s}}{2(32.2 \text{ ft/s})} + 16 \text{ ft} + 62.44$$

Summary:

Flow rate for the pump from train to large tank was determined to be 261.11 GPM and a pump head of 52.86 ft was calculated. Flow rate for the pump from the machinery area to the dirty tank was determined to be 105.7 GPM and a pump head of 79.87 ft was calculated. Flow rate from the dirty tank to the trucks was determined to be 141 GPM and a pump head of 14.81ft was calculated.

Analysis:

The pump from the machining area to the dirty tank requires more power because it has a significantly longer distance for the fluid to travel.

15. Specify the number of pumps, their types, flow capacities, head requirements, and power required. Why did you choose the pump you chose? Argue why you need a kinetic pump (instead of a positive displacement) and prove that the radial-pump is the type of kinetic pump you need.

Purpose: Outline the number of pumps and their types, flow capacity, head requirement, and power required. Include an explanation for the specified pumps.

Sources:

- Mott, R., Untener, J.A, "Applied Fluid Mechanics", 7th edition, Pearson Education, Inc (2015)
- Project Document
- Sulzer catalog

Design Considerations:

- Constant Properties
- Incompressible Fluids
- Standard tanker car is 8 ft in diameter and will be approximately 2 feet above the ground.
- Standard tanker cars are vented to atmosphere during unloading
- Assume atmospheric pressure at machinery outlet
- Standard critical velocity of 9.6 ft/s assumed at machinery outlet

Drawings & Diagrams:

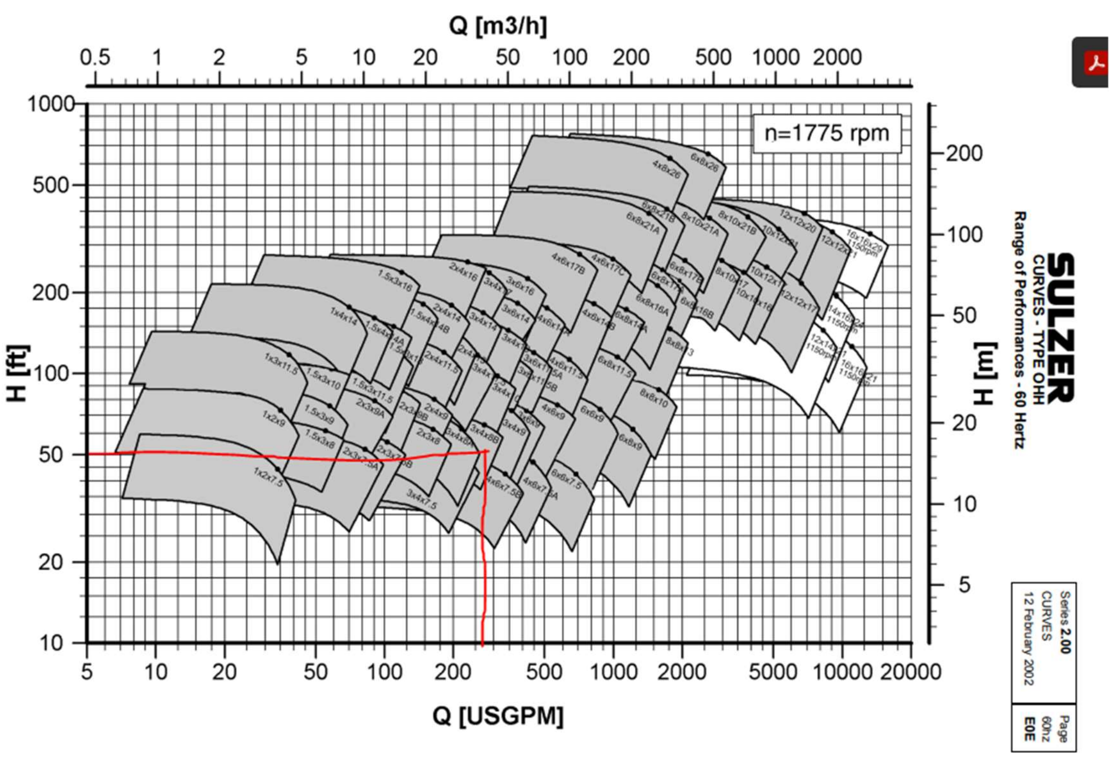


Figure 17: Pump Selection for Pump 1

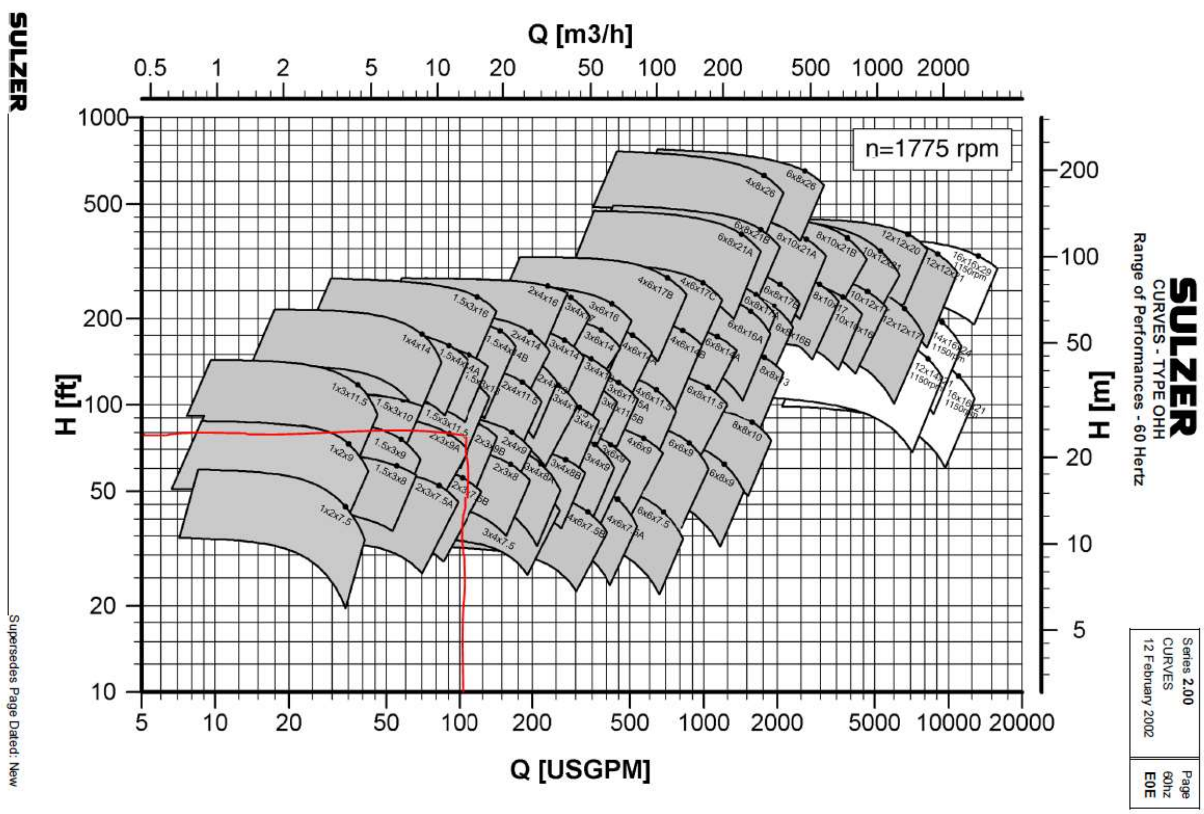


Figure 18: Pump Selection for Pump 2

Data & Variables:

- Q (pump 1): 261.11 GPM
- Q (pump 2): 105.70 GPM
- Q (pump 3): 141 GPM
- Head (pump 1): 52.86 ft
- Head (pump 2): 79.87 ft
- Head (pump 3):14.81 ft

Materials:

- Coolant
- Carbon Steel ASTM A106

Procedure:

1. Use pump head and flow rate to find the desired pump in the Sulzer catalog
2. The smallest pump that included the flow rate needed was chosen for the 3rd pump because it did not align with any pumps in the pump selection graph in the catalog.
3. Use affinity laws to read charts made for higher speed pumps.
4. Use the name of the pump to determine the other specifications of the pump listed in the catalog
5. Use the pump speed from the Sulzer catalog in combination with the flow rate and pump head to determine the specific speed.
6. Use chart provided in lecture notes to verify specific speed is under 4000 rpm.

Calculations:

$$Q_1 = \frac{Q_2}{2}$$

$$h_{a1} = \frac{h_{a2}}{4}$$

Pump 1:

$$N_s = \frac{N\sqrt{Q}}{H^{3/4}}$$

$$\frac{1775\ \text{rpm}\left(\sqrt{261.11}\ \text{gpm}\right)}{52.84\ \text{ft}^{3/4}}$$

$$N_s = 1463.059$$

Pump 2:

$$N_s = \frac{1775\ \text{rpm}\left(\sqrt{105.7}\ \text{gpm}\right)}{79.96\ \text{ft}^{3/4}} = 683.05$$

Pump 3:

$$N_s = \frac{1775\ \text{rpm}\left(\sqrt{141}\ \text{gpm}\right)}{14.814\ \text{ft}^{3/4}}$$

$$N_s = 2791.345$$

The pump head was calculated in task 8 and the flow rate was calculated in step 5. No further calculations.

Summary:

The table below outlines the pump head, flow, type, and number of pumps. The kinetic pump is chosen because it is cheaper and smaller than the positive displacement pumps and allows for better control over the flow rate. Additionally, the viscosity of the fluid is close to that of water, so the positive displacement pump is not necessary. The pumps were chosen from the provided Sulzer catalog based on the calculated flow rates and pump heads from tasks 5 and 8. The pumps are radial because their specific speed is under 4000 rpm, this was verified on the specific speed chart in Appendix B.

Table 4: Pumps

Pump I.D	Pump Location	Pump Type	Flow (GPM)	Pump Head (ft)	Pump Power (HP)
1	Train to Large Tank	Kinetric Pump, Double Impreller, Radial	261.11	52.86	47
2	Machine Pump to Dirty Tank	Kinetric Pump, Single Impreller, Radial	105.7	79.06	55
3	Dirty Tank to Trucks	Kinetric Pump, Single Impreller, Radial	141	14.54	18

Analysis:

Pump 3 is the smallest pump because it has the shortest distance to travel and encounters few losses.

16. Specify the characteristics of the chosen pumps, point of operation, and actual pump size and weight. Some of the information is for our civil engineer colleagues. VERY IMPORTANT: Include pump curves with the system curve, and point of operation. For this task, please print out the pages in the catalog where you got the information.

Purpose:

Outline the pump characteristics, point of operation, and actual pump size and weight.

Sources:

- Mott, R., Untener, J.A, "Applied Fluid Mechanics", 7th edition, Pearson Education, Inc (2015)
- Project Document
- Sulzer catalog

Design Considerations:

- Constant Properties
- Incompressible Fluids
- Standard tanker car is 8 ft in diameter and will be approximately 2 feet above the ground.
- Standard tanker cars are vented to atmosphere during unloading
- Assume atmospheric pressure at machinery outlet
- Standard critical velocity of 9.6 ft/s assumed at machinery outlet

Drawings & Diagrams:

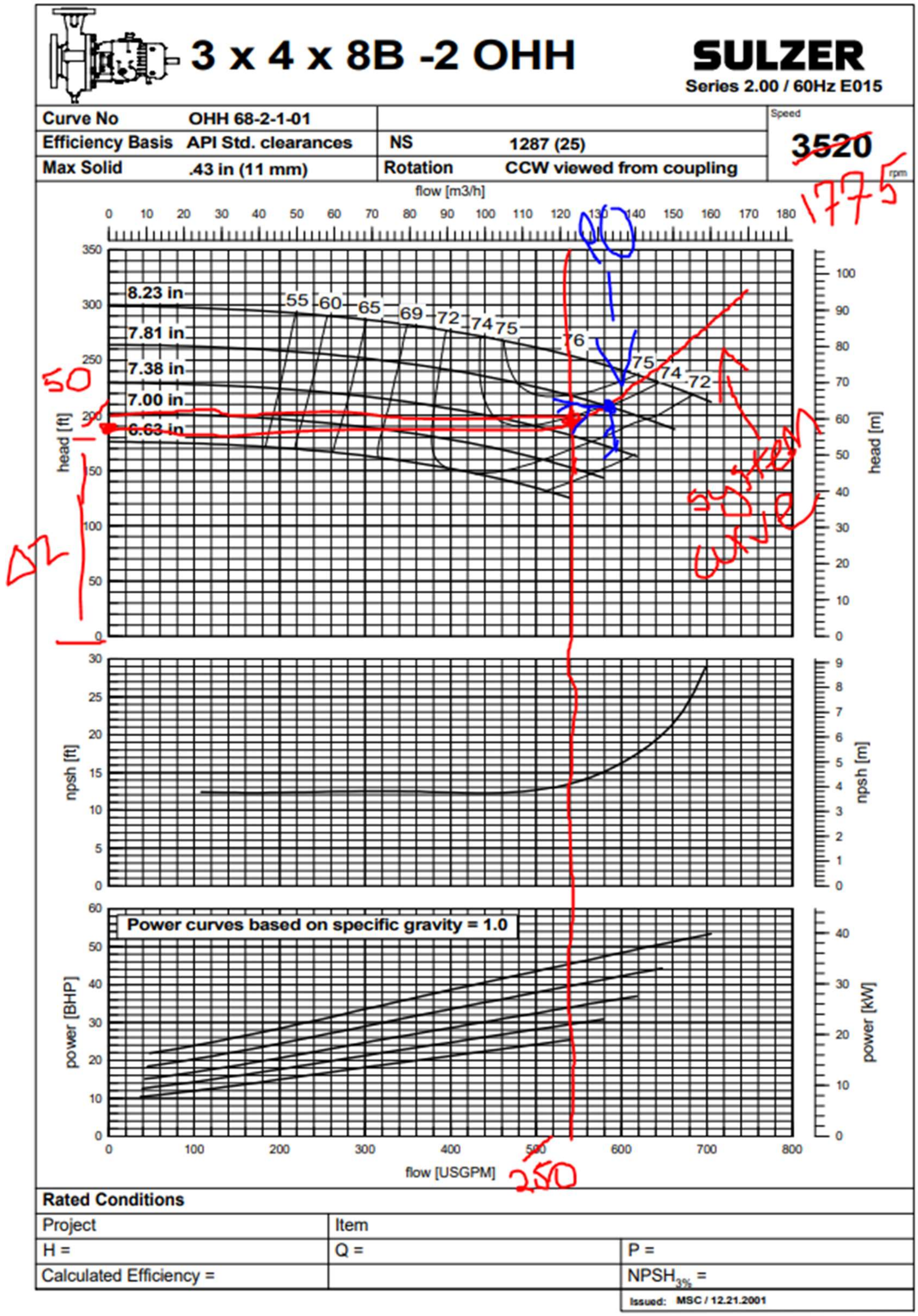


Figure 19: Pump 1 Performance Curve

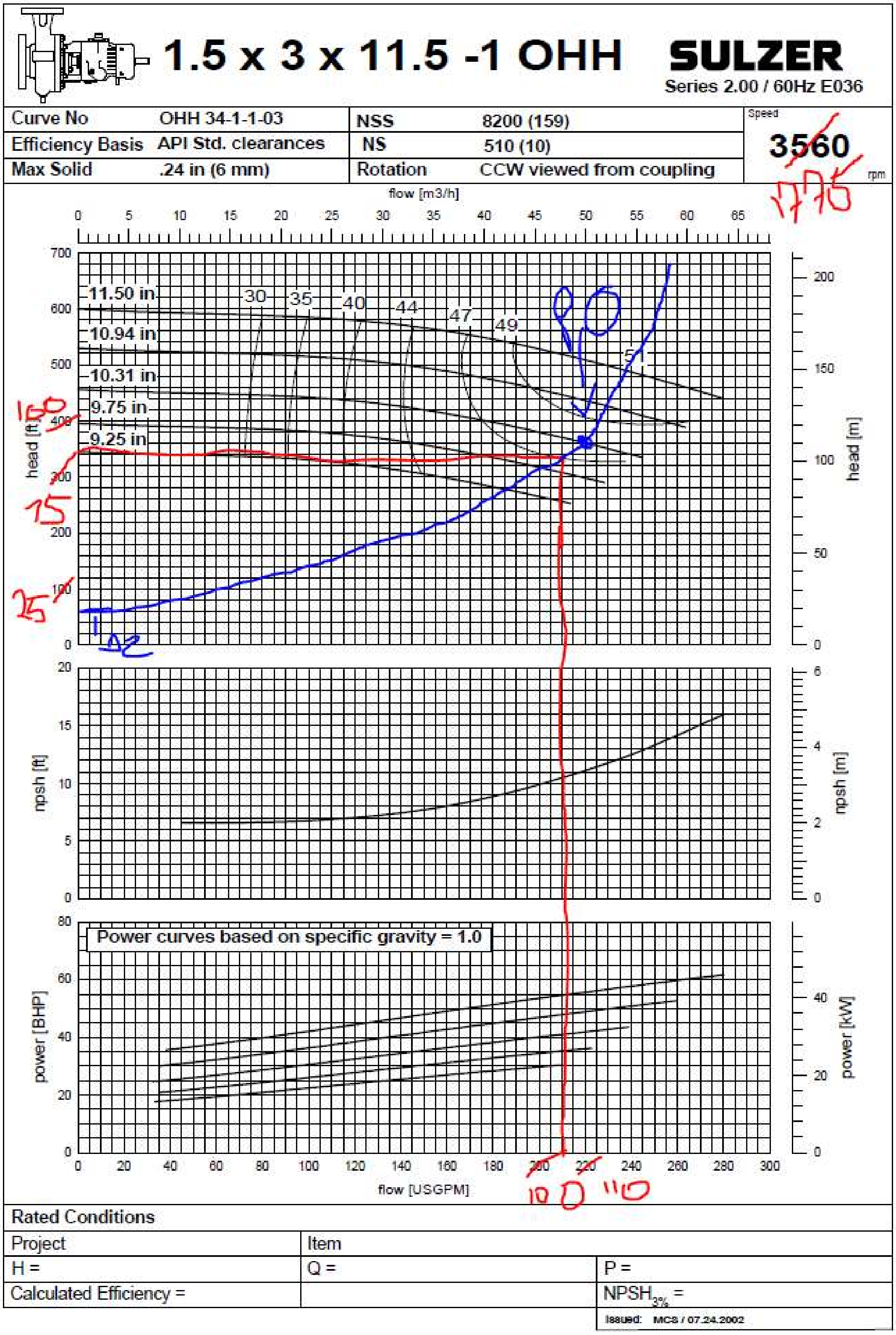


Figure 20: Pump 2 Performance Curve

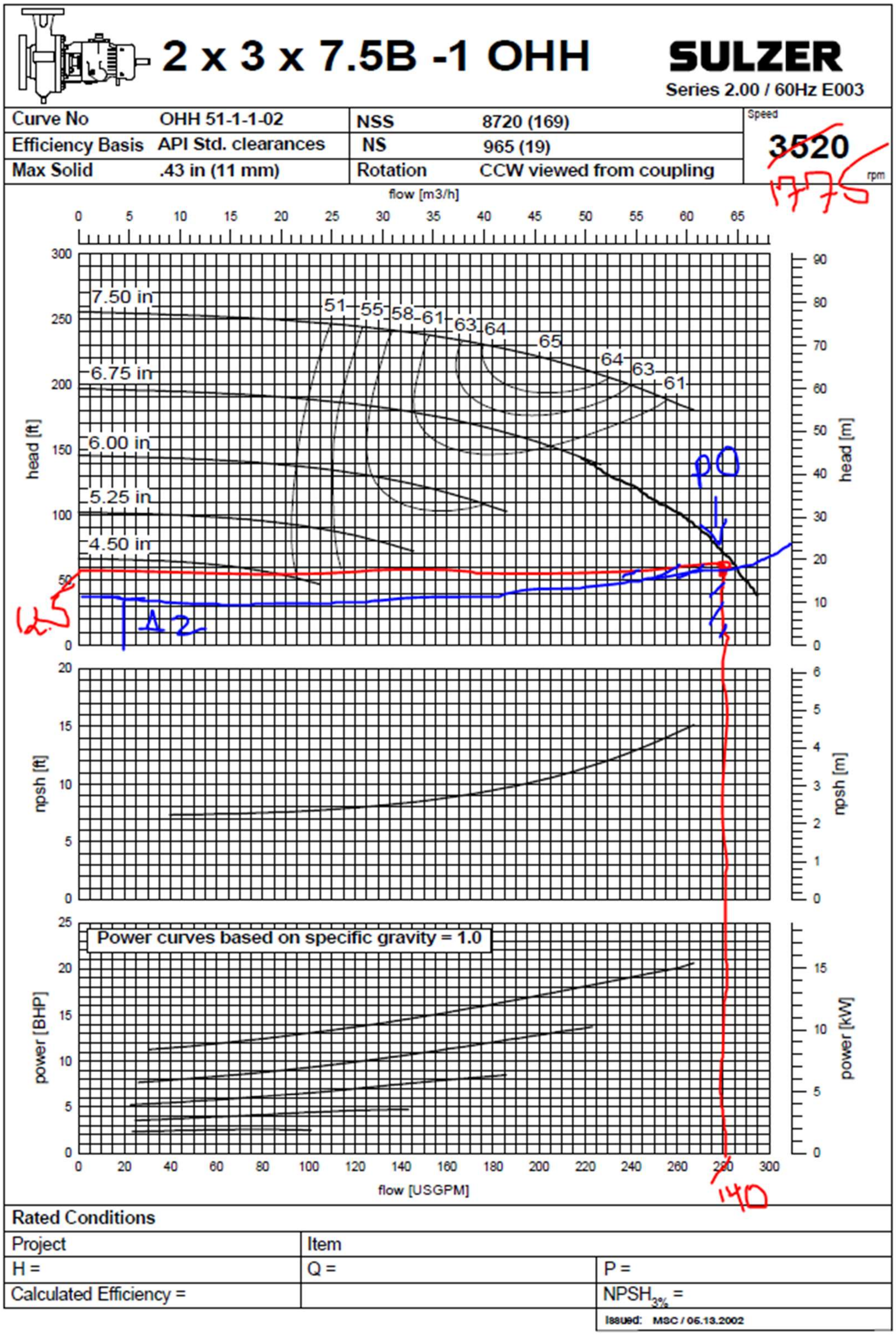


Figure 21: Pump 3 Performance Curve

Calculations:

$$Q_1 = \frac{Q_2}{2}$$

$$h_{a1} = \frac{h_{a2}}{4}$$

Summary:

Table 5: Pump Characteristics						
Pump I.D	**Pump Location**	**Pump Size**	**Impeller size (in)**	**Weight (lbs)**	**Speed (RPM)**	**Efficiency**
1	Train to Large Tank	3X4X8B	7.81	370	1775	74%
2	Machine Pump to Dirty Tank	1.5X3X11.5-1	10.31	522	1775	51%
3	Dirty Tank to Trucks	2X3X7.5B-1	6.75	331	1775	51%

The point of operation for each pump is indicated in blue in its respective figure.

Analysis:

The pump 1 is the most efficient pump because it has the highest flow rate. Additionally, each pump will was chosen to use the larger impeller and use a valve to throttle down to the previously calculated flow rates.

17. Specify electrical motor requirement for our pump for our electrical engineering colleagues. Recall that we specify the power of the electrical motor as about 1.10 times the power required by the pump.

Purpose: Determine electrical power needed for each pump

Sources:

- Mott, R., Untener, J.A, "Applied Fluid Mechanics", 7th edition, Pearson Education, Inc (2015)
- Project Document
- Sulzer catalog

Design Considerations:

- Constant Properties
- Incompressible Fluids
- Electrical power is 1.10 times the pump power.

Drawings & Diagrams:

N/A

Data & Variables:

- Pump 1 power = 47 HP
- Pump 2 power = 55 HP
- Pump 3 power = 18 HP

Materials:

- Coolant
- Carbon Steel ASTM A106
- Sulzer Pumps: 3X4X8B, 1.5X3X11.5-1, 2X3X7.5B-1

Procedure:

Multiply each pump power from task 15 by 1.10

Calculations:

$$Pump\ Power\ X\ 1.10\ =\ Electrical\ Power\ Requirement$$

Pump 1 power

$$47\ HP\ X\ 1.10\ =\ 51.7\ HP$$

Pump 2 power

$$55\ HP\ X\ 1.10\ =\ 60.5\ HP$$

Pump 3 power

$$18\ HP\ X\ 1.10\ =\ 19.8\ HP$$

Summary:

The motor for pump 1 will need to be a minimum of 51.7 HP. The motor for pump 2 will need to be a minimum of 60.5 HP. The motor for pump 3 will need to be a minimum of 19.8 HP.

Analysis:

The electrical motor size varies proportionally to the pump size. The electrical motor powers will more than likely not be sold to the exact calculations, the numbers presented in the calculations are the minimum requirements to power the pumps.

18. Evaluate the NPSH available for your design, and demonstrate that your pump has an acceptable NPSH required. Specify the installation requirements for the pumps, including the complete suction line system.

Purpose:
Determine NPSH required to ensure the available NPSH is higher

Sources:

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

Design Considerations:

- Pump Head
- Pipe and valve losses

Drawings & Diagrams:

N/A

Data & Variables:

- h suction pump
- h s
- h f
- h vacuum pump

Procedure:

1. Solve for NPSH available
2. Compare to NPSH required
3. If available is greater than required, the pumps are valid

Calculations:

$NPSH_A \quad h_{sp} \pm h_s - h_f - h_{vp}$

$h_{sp} = \frac{P}{\gamma} \quad \gamma = 58.71992\ lb/ft^3$

$P = 14.295\ psi \quad (atm @ 738') = 2058.48\ psf$

$h_{sp} = \frac{2058\ \cancel{lb}/ft^2}{58.71992\ \cancel{lb}/ft^3} = 35.05590607\ ft$

$h_{s_1} = 8\ ft$

$h_{s_2} = 0\ ft$

$h_{s_3} = 16\ ft$

$h_{f_1} = 2.5\ ft$

$h_{f_2} = 0.807\ ft$

$h_{f_3} = 1.330\ ft$

$h_{vp} = 0.5917 \quad (Water @ 60°)$

$Pump_1 \quad h_{sp} - h_s - h_f - h_{vp} = 35.06 - 8 - 2.5 - 0.5917 = 23.9683\ ft$

$Pump_2 \quad h_{sp} + h_s - h_f - h_{vp} = 35.06 + 0 - 0.807 - 0.5917 = 33.6613\ ft$

$Pump_3 \quad h_{sp} + h_s - h_f - h_{vp} = 35.06 + 16 - 1.330 - 0.5917 = 49.1383$

$NPSH_{R_1} = 13\ ft$

$NPSH_{R_2} = 11.5\ ft$

$NPSH_{R_3} = 17\ ft$

$NPSH_{A_1}\ 23.97\ ft > NPSH_{R_1}\ 13\ ft$

$NPSH_{A_2}\ 33.7\ ft > NPSH_{R_2}\ 11.5\ ft$

$NPSH_{A_3}\ 49.1\ ft > NPSH_{R_3}\ 17\ ft$

Pumps are Valid

Summary:

The data calculated shows that for each pump, the NPSH available is greater than the NPSH required (found in the above pump data sheets) for each pump. Each pump would require valves both before and after in order to isolate the pump for repair. All three pumps would also require swing gate check valves to stop reverse flow of the fluid. Pump one would have to be mounted above the tank car top height, thus putting it above the reservoir it is pumping from, with a flexible suction line in order to pump the fluid from the tanker cars up to the storage tank. Pump two and pump three would both be under the reservoir tank, with gravity fed suction lines in order to pump the fluid to the next reservoir in the system as required.

Analysis:

Since all three pumps have a NPSH available greater than the required value, all three pumps are tenable for this project.

vii. Instrumentation Selection

14. Just for one of your pipeline systems, select the required instruments. This is, pick an instrument to measure the flow of the pressure differential type (specify its dimensions) and pressure gauges (specify range of pressures to measure).

Purpose:

Determine measuring device for one part of system

Sources:

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

Design Considerations:

- Constant Properties
- Incompressible Fluids
- Pressure Variations
- Flow Rate

Drawings & Diagrams:

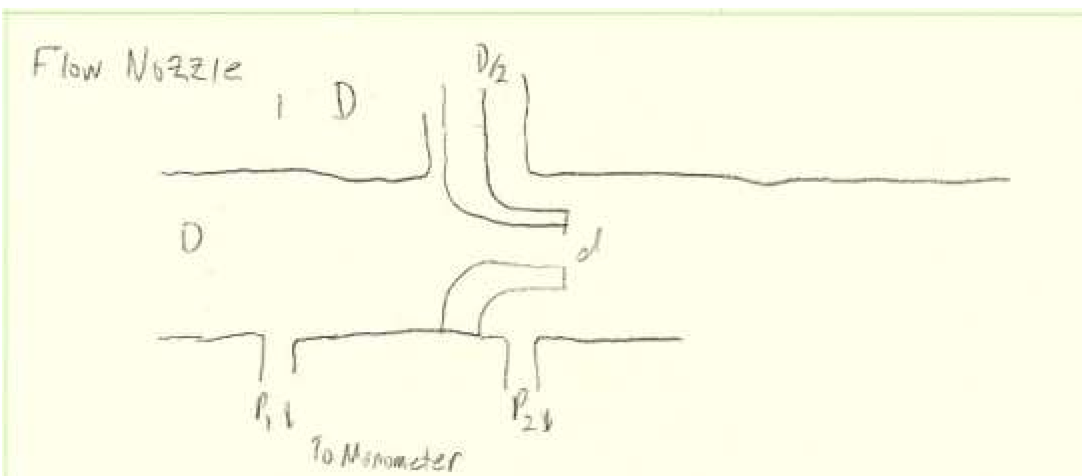


Figure 22: Flow Nozzle

Data & Variables:

- Diameter of main pipe: Schedule 40 1.5 in steel pipe, ID = 1.610 in
- Low Pressure value: 0 psi
- High Pressure value (at Water Hammer): 5610 psi

Materials:

- Coolant
- Carbon Steel ASTM A106

Procedure:

4. Use D diameter to find distance from flow nozzle entrance to manometer pipe exits
5. Use the formula for $\boldsymbol{\beta}$ to calculate d diameter

Calculations:

$$\Delta P = P_{max} - P_{min} = 3600 - 0 = 3600\ psi$$

$$D = 1.610\ in$$

$$D/2 = .805\ in$$

$$\beta = d/D$$

$$\text{assume } \beta = .50$$

$$.50 = d/1.610$$

$$.50(1.610) = d$$

$$d = .805\ in$$

$$R_e = \frac{vD}{\nu} = \frac{9.6\ ft/s\ \ .2058\ ft}{1.815E-5} = 108852.8926$$

$$\nu = \nu_{water@60}(1.5) = 1.21E-5\ (1.5)\ ft^2/s = 1.815E-5$$

$$C = 0.9975 - 6.53\sqrt{B/R_e}$$

$$0.9975 - 6.53\sqrt{.5/108852.8926} = .9835048294$$

Summary:

The data calculated shows that a flow nozzle placed in the long pipe from the large tank to the medium tank would need to have an inner diameter (d) of 0.805 in, with the manometer pipes being 1.61 inches in front of the nozzle entrance, and 0.805 inches beyond the nozzle entrance. This would allow the manometer to give pressure readings on the flow in the pipe. This pressure difference is 0-3600 psi, allowing for zero flow at the low value, and water hammer phenomena at the high value.

Analysis:

A manometer connected to a flow nozzle would provide the required data on flow rate through the pipe running from the large tank to the medium tank.

6. Final Drawings

a. Plot Plan

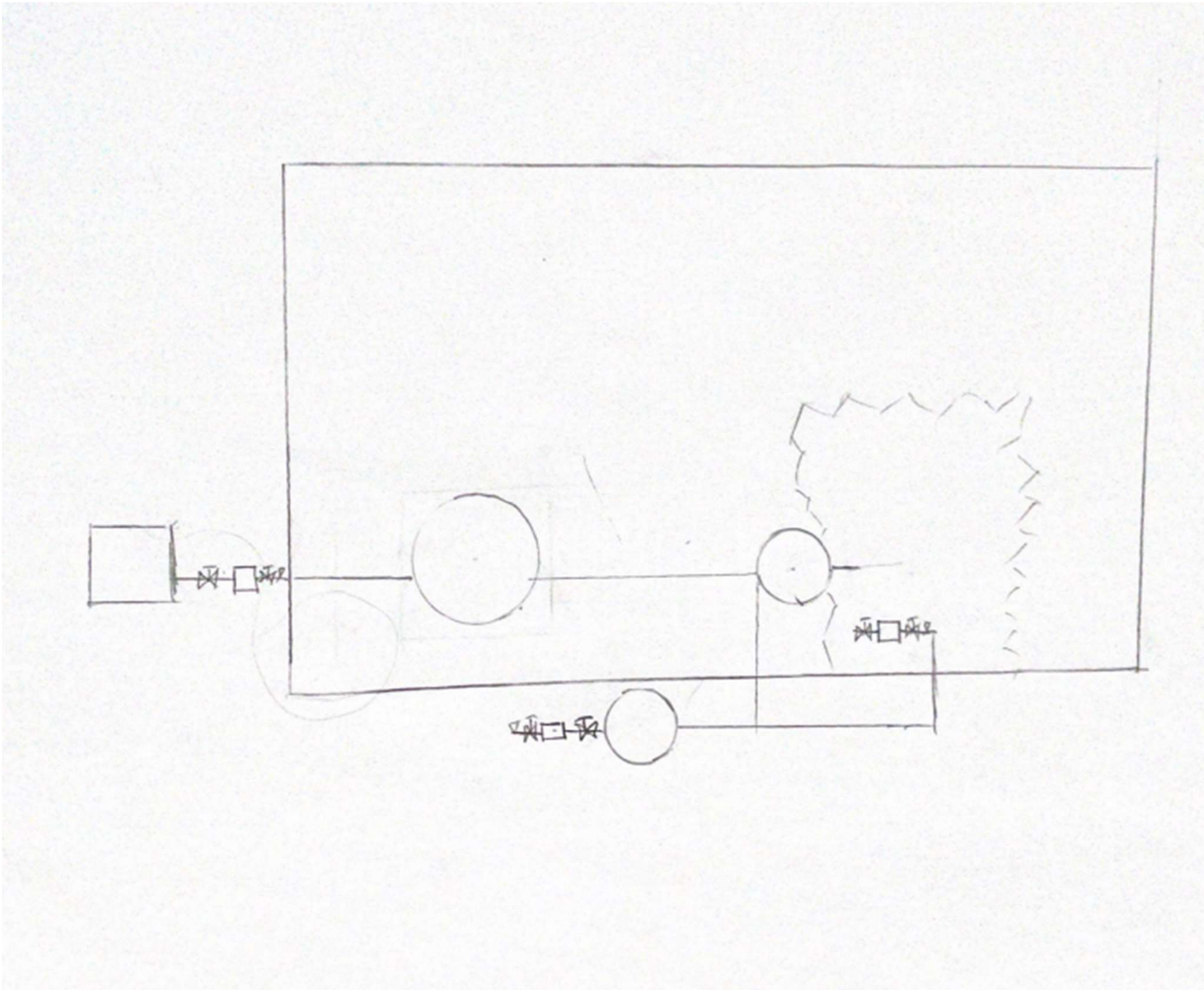


Figure 1: Plot Plan View of System Layout

b. Elevations View

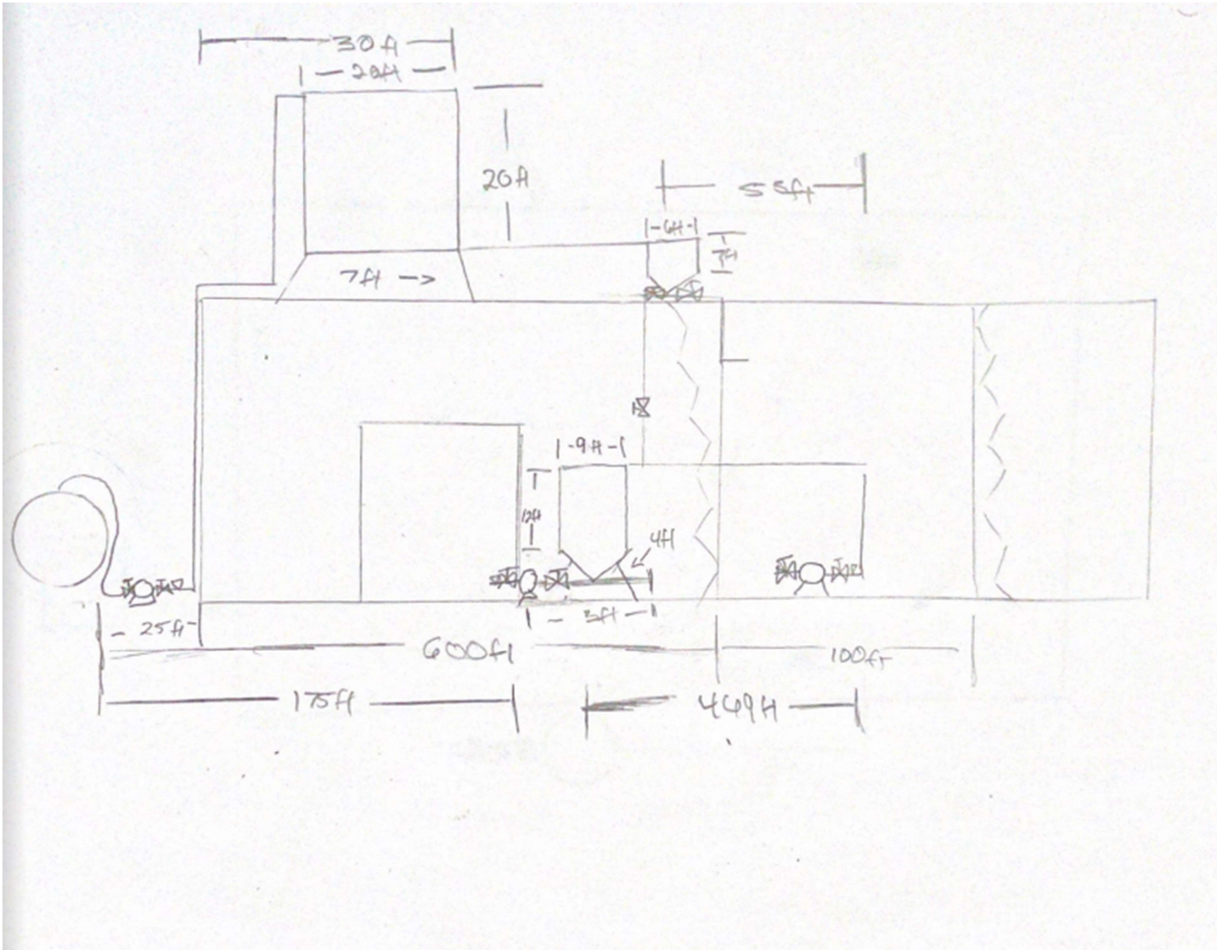
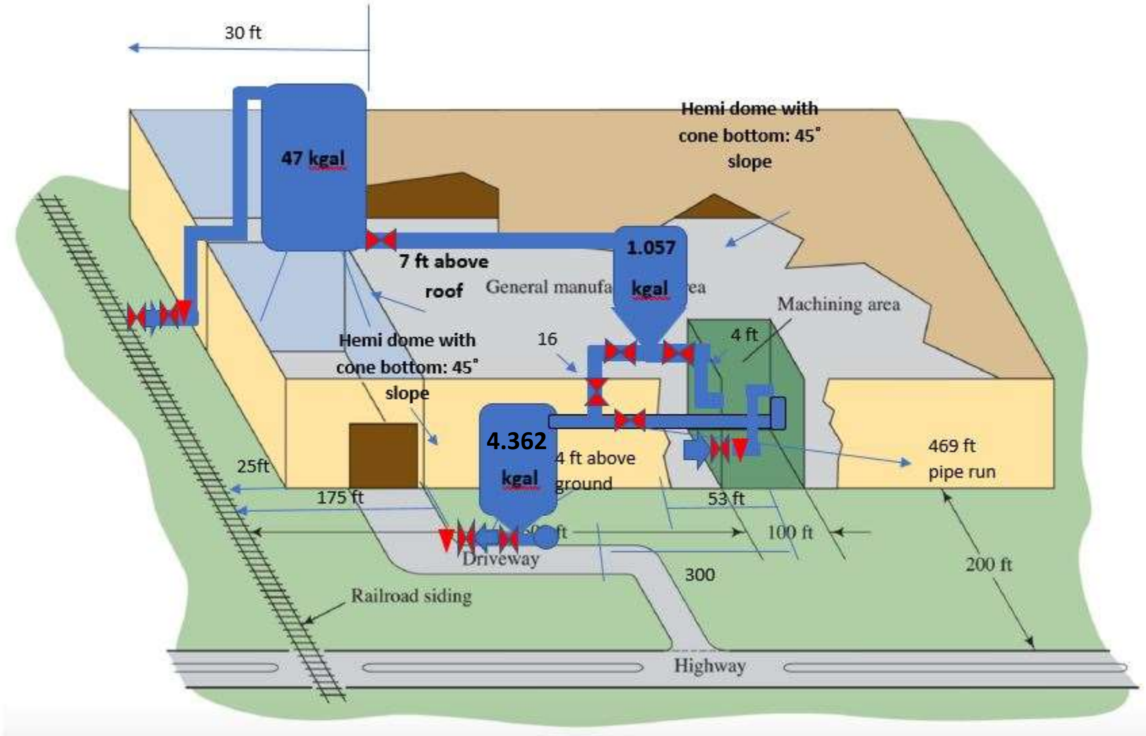


Figure 2: Elevations View of System Layout

c. Isometrics



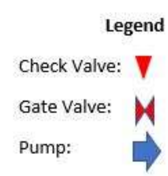


Figure 3: Isometric View of System Layout

7. Bill of Materials and Equipment List

Bill of Materials			
Item	**Item**	**Description**	**Qty.**
1	Pump 3X4X8B	Kinetic Pump, Double Impeller, Radial	1
2	Pump 1.5X3X11.5-1	Kinetic Pump, Single Impeller, Radial	1
3	Pump 2X3X7.5B-1	Kinetic Pump, Single Impeller, Radial	1
4	Electric Motor	Minimum power requirement of 51.7 HP	1
5	Electric Motor	Minimum power requirement of 37.4 HP	1
6	Electric Motor	Minimum power requirement of 19.8 HP	1
7	Revisor Tank	20 ft X 20ft hemi dome tank Carbon Steel ASTM A106, Electric Resistance Welded	1
8	Feed Tank	6 ft X 7 ft hemi dome tank w/ 45-degree cone bottom Carbon Steel ASTM A106, Electric Resistance Welded	1
9	Dirty tank	6 ft X 12 ft hemi dome tank w/ 45-degree cone bottom Carbon Steel ASTM A106, Electric Resistance Welded	1
10	Gate Valve	Gate valve for 3.5 in pipe	2
11	Gate Valve	Gate valve for 2.5 in pipe	2
12	Gate Valve	Gate valve for 2 in pipe	3
13	Gate Valve	Gate valve for 1.5 in pipe	4
14	Check Valve	Check valve for 3.5 in pipe	1
15	Check Valve	Check valve for 2.5 in pipe	1
16	Check Valve	Check valve for 2 in pipe	1
17	Steel Pipe	Schedule 40 3.5 in	69 ft
18	Steel Pipe	Schedule 40 2.5 in	3 ft
19	Steel Pipe	Schedule 40 2 in	488 ft
20	Steel Pipe	Schedule 40 1.5 in	756 ft
21	Hose	3.5 in	20 ft
22	Hose	2.5 in	20 ft
23	Elbow	90-degree elbow Schedule 40 3.5 in	8
24	Elbow	90-degree elbow Schedule 40 2 in	4
25	Elbow	90-degree elbow Schedule 40 1.5 in	3
26	Tee	Schedule 40 1.5 in	2
27	Tee	Schedule 40 2.5 in	1
28	Blind Flange	0.25 in thick Carbon Steel ASTM A106	1
29	Bolts	3/4 in Carbon Steel ASTM A106	4
30	Nuts	3/4 in Carbon Steel ASTM A107	4
31	Flow Nozzle	0.805 in ID	1
32	Manometer Pipe	N/A	2

8. Final Remarks

The system is designed to hold 45,000 gallons of coolant in the main reservoir which is enough to supply the machine area for 11 months. The main reservoir then feeds the feed tank which directly distributes the coolant to the machine area. Once the fluid is used, it is pumped out of the machine area into a holding tank for dirty fluid that is positioned next to the driveway. The positioning of the dirty fluid holding tank allows for quick pickup when the fluid is carried off in tanker trucks. The feed tank also has an emergency dump line that flows directly to the dirty holding tank in case the fluid in the feed tank gets contaminated. The dirty hold tank has a blind flange which can be used for future expansion if needed. The system is cost effective because it uses very direct paths for the pipelines and is partially gravity fed. With the help of gravity, the system only requires three pumps and electric motors. Additionally, because the main reservoir is so large, it will save money on shipping fees because the fluid will not have to be delivered at such a high frequency.

9. Appendix

Appendix A Sulzer Catalog Charts

<u>Pump 1:</u>

SULZER

TECHNICAL DATA - TYPE OHH

Mechanical Dimensions & Ratings – US Standard Units

Series **2.00** TECHNICAL DATA September - 02	Page US **B18**

OHH TECHNICAL DATA	US	1.5x3x8-1	2x3x8-1	3x4x8A-1	3x4x8B-1	3x4x8B-2	1x2x9-1	1.5x3x9-1
Max/Min Impeller Dia. (in)		8.2/6.5	8.2/6.5	8.2/6.5	8.2/6.5	8.2/6.5	9.3/7.4	9.3/7.4
Volute Construction		Single	Single	Single	Double	Double	Single	Single
Max Operating Pressure (Psig)		740	740	740	740	740	740	740
Hydrostatic Test Pressure (Psig)	6	1110	1110	1110	1110	1110	1110	1110
Max Operating Temperature (°F)		Refer to series B11-B16 for max pump operating temperatures for each material class						
Size of Casing Drain Construction NPT		0.75	0.75	0.75	0.75	0.75	0.75	0.75
Bearing Housing No.		3	3	3	3	3	3	3
Shaft Dia. between bearings (in)		2.56	2.56	2.56	2.56	2.56	2.56	2.56
Span between bearings (in)		7.34	7.34	7.34	7.34	7.34	7.34	7.34
Span CL Rad Brg to CL Imp (in)		10.51	10.67	10.55	10.75	10.75	10.16	10.20
Shaft Dia. at Seal Chamber (in)		1.89	1.89	1.89	1.89	1.89	1.89	1.89
Shaft Dia. at coupling (in)		1.26	1.26	1.26	1.26	1.26	1.26	1.26
Typical API Baseplate #	7	1.5	1.5	1.5	2.0	2.0	1.5	1.5
Radial Bearing Number		6310	6310	6310	6310	6310	6310	6310
Thrust Bearing Number		7311	7311	7311	7311	7311	7311	7311
Pump Weight (lb)		336	336	361	370	370	320	334
Typical API Baseplate weight (lb)	7	802	802	802	932	932	802	802
Minimum Case Thickness (in)		Refer to series B40-B41						
Max. Dia. Spherical Solids (in)	3	0.28	0.37	0.37	0.43	0.43	0.20	0.20
Wear Ring Diameter - Eye (in)		4.13	4.53	4.53	5.31	5.31	4.13	4.53
Wear Ring Diameter - Hub (in)		4.13	4.53	4.53	5.31	5.31	4.13	4.53
Clearance Below 500°F - Eye (in)	1	0.015	0.016	0.016	0.017	0.017	0.015	0.016
Clearance Below 500°F - Hub (in)	1	0.015 (8)	0.016 (8)	0.016 (8)	0.017 (8)	0.017 (8)	0.015 (8)	0.016 (8)
Mass Moment of Inertia (lb-ft2) WR2	5	0.63	0.69	0.69	0.84	0.84	0.89	0.91
Shaft Stiffness Factor L3/D4	4	91	95	92	97	97	82	83
Critical Speed (Dry) (cpm)		9685	9035	9208	8251	8251	9806	9481
Temperature Limits (°F)		Refer to series B42 for recommended pump and bearing features required for elevated temperatures						

1) Above 260°C (500°F) API requires that clearances are increased by 0.005 in. Pump efficiency must be reduced to compensate for increased internal leakage. Also be sure to correct the power and efficiency for dual seals and high suction pressures.
2) ANSI class 300 RF, EN1092-1 PN64, PN40 flanges are available. Pump operating conditions are not to exceed flange design working pressure and/or temperature. Note: On C-6 and A-8 pumps, the use of higher rated flanges for auxiliary casing connections may be necessary to equal the casing MAWP
3) Maximum spherical diameter is 0.12 in. less than narrowest internal passage.
4) See page B44 for additional information and intended application regarding L3/D4 shaft stiffness evaluation
5) The values shown are for "Wet" WR2 (includes the pumpage contained in the impeller). The coupling WR2 value (to be supplied by coupling manufacturer) should be added to the impeller.
6) Auxiliary process piping shall be tested to the pressure of the casing. (Tubing or threaded piping Is Not hydrotested).
7) Typical Baseplate size and weights are an average provided to speed quoting activity and estimating. For an exact size and weight for the baseplate, use motor frame and pump information to pick a baseplate from pages D01 to D24.
8) These pumps are furnished without hub wear rings and throat bushing to pressurize the seal chamber above vapor pressure and extend seal life.

SULZER Supersedes Page Dated: 15 February 2002

SULZER

TECHNICAL DATA - TYPE OHH

Noise Calculations – US Customary Units

The following tabulation shows estimated noise level data for Type **OHH** pumps and is presented in either
a) "Sound Pressure Level" (dB) versus "Octave Center Frequency" (Hz).
b) "Weighted Sound Pressure Level" (dBA).

The Table lists values for each **OHH** pump size using:
- Full diameter impellers.
- Running at best efficiency capacity.
- Adequate NPSHa.
- Rated BHP at best efficiency with 1.0 SpGr.

Measurements were taken
- At three feet or one meter from pump or piping surfaces.
- At 60 Hertz design speed with the appropriate performance curve.

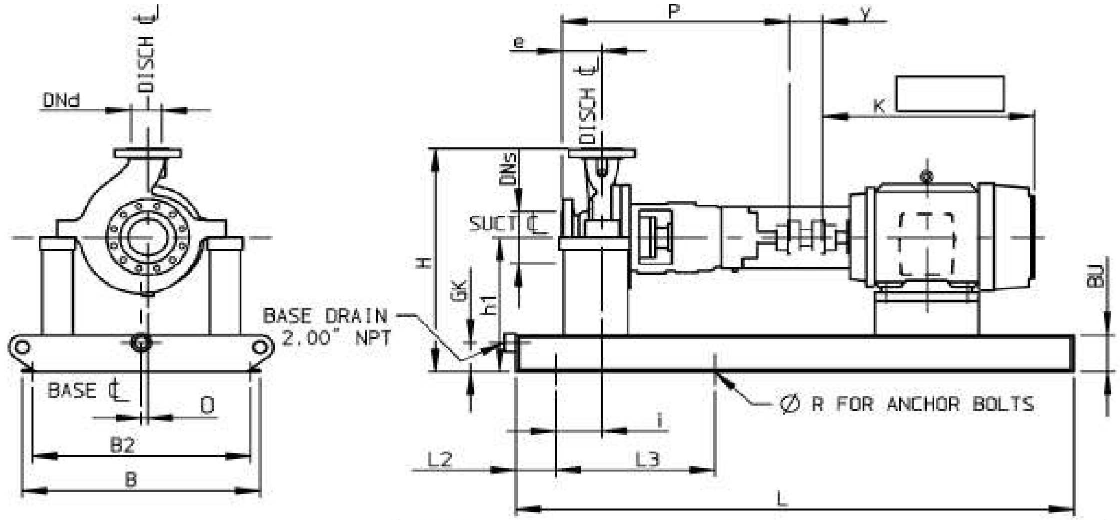
	Power	SPL	Octave Band Center Frequency in Hertz								
Basis OHH	HP	dBA	31.5	63	125	250	500	1000	2000	4000	8000
1x2x7.5-1	8	80.1	72	76	78	79	77	75	72	69	65
2x3x7.5A-1	15	82.2	72.1	78.1	78.1	81.1	79.1	77.1	74.1	71.1	67.1
2x3x7.5B-1	18	82.5	72.4	78.4	78.4	81.4	79.4	77.4	74.4	71.4	8.5
3x4x7.5-1	29	82.2	74.1	78.1	80.1	81.1	79.1	77.1	74.1	71.1	67.1
4x6x7.5B-1	37	81.5	73.4	77.4	79.4	80.4	78.4	76.4	73.4	70.4	66.4
4x6x7.5A-1	54	83.2	75.1	79.1	81.1	82.1	80.1	78.1	75.1	72.1	68.1
6x6x7.5-1	74	86	69.2	78.2	75.2	78.2	81.2	81.2	79.2	76.2	73.2
1.5x3x8-1	12	80.3	72.2	76.2	78.2	79.2	77.2	75.2	72.2	69.2	65.2
2x3x8-1	30	82.4	74.3	78.3	80.3	81.3	79.3	77.3	74.3	71.3	67.3
3x4x8A-1	37	81.6	73.5	77.5	79.5	80.5	78.5	76.5	73.5	70.5	66.5
3x4x8B-1	47	85.8	70.6	79.6	76.6	79.6	82.6	80.6	78.6	75.6	72.6
3x4x8B-2	47	85.8	70.6	79.6	76.6	79.6	82.6	80.6	78.6	75.6	72.6
1x2x9-1	17	81.8	73.6	77.6	79.6	80.6	78.6	76.6	73.6	70.6	66.6
1.5x3x9-1	19	82.9	72.8	78.8	78.8	81.8	798	77.8	74.8	71.8	67.8
2x3x9A-1	25	83.3	73.3	79.3	79.3	82.3	80.3	78.3	75.3	72.3	68.3
2x3x9B-1	34	83.8	73.7	79.7	79.7	82.7	80.7	78.7	75.7	72.7	68.7
2x4x9-1	42	83.2	75.1	79.1	81.1	82.1	80.1	78.1	75.1	72.1	68.1
3x4x10-1	83	85.7	75.8	79.8	81.8	82.8	82.8	80.8	77.8	74.8	70.8
3x4x10-2	83	85.7	75.8	79.8	81.8	82.8	82.8	80.8	77.8	74.8	70.8
3x4x9-1	71	84	75.8	79.8	81.8	82.8	80.8	78.8	75.8	72.8	68.8
3x4x9-2	71	84.3	76.2	80.2	82.2	83.2	81.2	79.2	76.2	73.2	69.2
3x6x9-1	82	84.2	76.1	80.1	82.1	83.1	81.1	79.1	73.1	73.1	69.1
3x6x9-2	82	84.5	76.4	80.4	82.4	83.4	81.4	79.4	76.4	73.4	69.4
4x6x9-1	109	85.4	75.5	79.5	81.5	82.5	82.5	80.5	77.5	74.5	70.5
4x6x9-2	109	85.4	75.5	79.5	81.5	82.5	82.5	80.5	77.5	74.5	70.5
6 x 6 x 9-1	104	85.9	76	80	82	83	83	81	78	75	71
6x8x9-1	188	87.4	70.7	79.7	76.7	79.7	82.7	82.7	80.7	77.7	74.7
8x8x10-1	330	88.5	71.8	80.8	77.8	80.8	83.8	83.8	81.8	78.8	75.8
8x8x10-2	330	88.5	71.8	80.8	77.8	80.8	83.8	83.8	81.8	78.8	75.8
1x3x11.5-1	32	82	73.9	77.9	79.9	80.9	78.9	76.9	73.9	70.9	66.9
1.5x3x13-1	68	85.5	77.4	81.4	83.4	84.4	82.4	80.4	77.4	74.4	70.4
1.5x3x11.5-1	55	85.2	75.1	81.1	81.1	84.1	82.1	80.1	77.1	74.1	70.1
2x4x11.5-1	77	84.8	76.6	80.6	82.6	83.6	81.6	79.6	76.6	73.6	69.6
2x4x13-1	106	85.9	77.7	81.7	83.7	84.7	82.7	70.7	77.7	74.7	70.7
3x4x11.5-1	90	86	75.9	81.9	81.9	84.9	82.9	80.9	77.9	74.9	70.9
3x4x11.5-2	90	85	76.9	80.9	82.9	83.9	81.9	79.9	76.9	73.9	69.9
3x4x13-1	153	87.2	77.3	81.3	83.3	84.3	84.3	82.3	79.3	76.3	72.3
3x4x13-2	153	87.4	77.3	83.3	83.3	86.3	84.3	82.3	79.3	76.3	72.3
3x6x11.5B-1	124	86.5	76.4	82.4	82.4	85.4	83.4	81.4	78.4	75.4	71.4
3x6x11.5B-2	124	85.5	77.4	81.4	83.4	84.4	82.4	80.4	77.4	74.4	70.4
3x6x11.5A-1	149	84.2	76.1	80.1	82.1	83.1	81.1	79.1	76.1	73.1	69.1
3x6x11.5A-2	149	85.8	77.7	81.7	83.7	84.7	82.7	80.7	77.7	74.7	70.7
4x6x11.5-1	187	87.1	77	83	83	86	84	82	79	76	72

SULZER

DIMENSIONAL DATA - TYPE OHH

General Arrangement Dimensions – End Suction – US Customary Units

Series 2.00	Page
DIMENSIONS	US
October - 02	D01



Certified by :_______________________
Date: _______________ Certified for Nozzle & Anchor Bolt locations only when signed above
Ref No. : _______________________
Project Name : _______________________
Project Item No.: _______________________
HH – Number of Base Hold Down Holes each side, equally spaced.
L3 – Distance between holes.
GH – # of Grout Holes

API No.	Base wt lbs	B	B2	BU	L	L2	L3	R	GK	HH	GH
1.5	800	30	27	6	72.5	6	30.25	1	4.75	3	4
2.0	930	30	27	6	84.5	6	24.16	1	4.75	4	4

Dimension K is derived from NEMA standards or motor supplier catalogue. The rotation is counterclockwise – H.I. looking from Drive End. All dimensions are in nominal inches and for guidance only. Certified Drawings will be issued for actual construction.

When Available 'TS" frame should be used to minimize coupling size, there is no price difference on drivers. **WP-XXX**

PUMP SIZE	PUMP wt lbs	DNs	DNd	O	B	e	P	h1	i	H	y	BASE REFERENCE (Electric Driver - NEMA Frame)
		in										
1x2x7.5-1	311	2	1	0	30	4.92	31.06	18	**5.63**	27.06	5	1.5 (143T-286T) (284TS-286TS) 2.0 (324T-405T) (324TS-405TS)
2x3x7.5A-1	332	3	2	0	30	4.92	31.18	18	**5.38**	27.06	5	
2x3x7.5B-1	331	3	2	0	30	4.92	31.18	18	**5.38**	27.06	5	
3x4x7.5-1	352	4	3	0	30	4.92	31.26	18	**5.36**	27.06	5	
4x6x7.5B-1	398	6	4	0	30	5.51	32.20	18	**5**	28.04	5	
4x6x7.5A-1	399	6	4	0	30	5.51	32.20	18	**5**	28.04	5	
6x6x7.5-1	522	6	6	1	30	8.07	35.67	18	**5.63**	30.60	7	2.0 (284T-365T) (284TS-365TS)
1.5x3x8-1	335	3	1.5	0	30	5.51	32.01	18	**5.65**	27.06	5	1.5 (143T-286T) (284TS-286TS) 2.0 (324T-405T) (324TS-405TS)
2x3x8-1	336	3	2	0	30	5.91	32.56	18	**5.63**	27.06	5	
3x4x8A-1	360	4	3	0	30	6.30	32.83	18	**5.38**	27.45	5	
3x4x8B-1	370	4	3	0	30	6.30	33.03	18	**5.63**	27.45	5	
3x4x8B-2	370	4	3	0	30	6.30	33.03	18	**5.63**	27.45	5	
1x2x9-1	320	2	1	0	30	5.51	31.65	18	**5.63**	27.06	5	
1.5x3x9-1	334	3	1.5	0	30	5.51	31.69	18	**5.63**	27.06	5	
2x3x9A-1	338	3	2	0	30	5.51	31.73	18	**5.5**	27.45	5	
2x3x9B-1	334	3	2	0	30	5.51	31.81	18	**5.63**	27.45	5	
2x4x9-1	360	4	2	0	30	5.51	31.77	18	**5.38**	28.04	5	
3x4x9-1	374	4	3	0	30	5.51	31.85	20	**7.38**	30.43	5	
3x4x9-2	374	4	3	0	30	5.51	31.85	20	**7.38**	30.43	5	
3x6x9-1	406	6	3	0.5	30	5.91	32.32	20	**7.25**	31.02	5	
3x6x9-2	406	6	3	0.5	30	5.91	32.32	20	**7.25**	31.02	5	

Pump 2:

SULZER

TECHNICAL DATA - TYPE OHH

Mechanical Dimensions & Ratings – US Standard Units

Series 2.00	Page
TECHNICAL DATA	US
September - 02	B21

OHH TECHNICAL DATA	US	8x8x10-1	8x8x10-2	1x3x11.5-1	1.5x3x11.5-1	2x4x11.5-1	3x4x11.5-1	3x4x11.5-2
Max/Min Impeller Dia. (in)		10.2/8.1	10.2/8.1	11.4/9.1	11.5/9.2	11.5/9.2	11.5/9.2	11.5/9.2
Volute Construction		Double	Double	Single	Single	Single	Single	Single
Max Operating Pressure (Psig)		740	740	740	740	740	740	740
Hydrostatic Test Pressure (Psig)	6	1110	1110	1110	1110	1110	1110	1110
Max Operating Temperature (°F)		Refer to series B11-B16 for max pump operating temperatures for each material class						
Size of Casing Drain Construction NPT		0.75	0.75	0.75	0.75	0.75	0.75	0.75
Bearing Housing No.		4.1	4.1	4.1	4.1	4.1	4.1	4.1
Shaft Dia. between bearings (in)		3.07	3.07	3.07	3.07	3.07	3.07	3.07
Span between bearings (in)		8.35	8.35	8.35	8.35	8.35	8.35	8.35
Span CL Rad Brg to CL Imp (in)		12.83	12.83	10.71	10.75	10.83	10.87	10.87
Shaft Dia. at Seal Chamber (in)		2.28	2.28	2.28	2.28	2.28	2.28	2.28
Shaft Dia. at coupling (in)		1.65	1.65	1.65	1.65	1.65	1.65	1.65
Typical API Baseplate #	7	4.0	4.0	3.5	3.5	3.5	3.5	3.5
Radial Bearing Number		6214	6214	6214	6214	6214	6214	6214
Thrust Bearing Number		7313	7313	7313	7313	7313	7313	7313
Pump Weight (lb)		798	798	514	523	541	549	549
Typical API Baseplate weight (lb)	7	1192	1192	1102	1102	1102	1102	1102
Minimum Case Thickness (in)		Refer to series B40-B41						
Max. Dia. Spherical Solids (in)	3	12	12	0.12	0.24	0.39	0.51	0.51
Wear Ring Diameter - Eye (in)		200	215	4.53	4.53	5.12	5.51	6.10
Wear Ring Diameter - Hub (in)		165	165	4.53	4.53	5.12	5.51	6.10
Clearance Below 500°F - Eye (in)	1	0.50	0.50	0.016	0.016	0.017	0.017	0.018
Clearance Below 500°F - Hub (in)	1	0.45	0.45	0.016	0.016	0.017	0.017	0.018
Mass Moment of Inertia (lb-ft2) WR2	5	3.65	3.65	2.28	2.28	2.63	2.46	2.46
Shaft Stiffness Factor L3/D4	4	78	78	45	46	47	47	47
Critical Speed (Dry) (cpm)		5753	5753	11697	11163	10376	10605	10605
Temperature Limits (°F)		Refer to series B42 for recommended pump and bearing features required for elevated temperatures						

1) Above 260°C (500°F) API requires that clearances are increased by 0.005 in. Pump efficiency must be reduced to compensate for increased internal leakage. Also be sure to correct the power and efficiency for dual seals and high suction pressures.
2) ANSI class 300 RF, EN1092-1 PN64, PN40 flanges are available. Pump operating conditions are not to exceed flange design working pressure and/or temperature. Note: On C-6 and A-8 pumps, the use of higher rated flanges for auxiliary casing connections may be necessary to equal the casing MAWP
3) Maximum spherical diameter is 0.12 in. less than narrowest internal passage.
4) See page B44 for additional information and intended application regarding L3/D4 shaft stiffness evaluation
5) The values shown are for "Wet" WR2 (includes the pumpage contained in the impeller). The coupling WR2 value (to be supplied by coupling manufacturer) should be added to the impeller.
6) Auxiliary process piping shall be tested to the pressure of the casing. (Tubing or threaded piping Is Not hydrotested).
7) Typical Baseplate size and weights are an average provided to speed quoting activity and estimating. For an exact size and weight for the baseplate, use motor frame and pump information to pick a baseplate from pages D01 to D24.
8) These pumps are furnished without hub wear rings and throat bushing to pressurize the seal chamber above vapor pressure and extend seal life.

SULZER

TECHNICAL DATA - TYPE OHH
Noise Calculations – US Customary Units

The following tabulation shows estimated noise level data for Type **OHH** pumps and is presented in either
a) "Sound Pressure Level" (dB) versus "Octave Center Frequency" (Hz).
b) "Weighted Sound Pressure Level" (dBA).

The Table lists values for each **OHH** pump size using:
- Full diameter impellers.
- Running at best efficiency capacity.
- Adequate NPSHa.
- Rated BHP at best efficiency with 1.0 SpGr.

Measurements were taken
- At three feet or one meter from pump or piping surfaces.
- At 60 Hertz design speed with the appropriate performance curve.

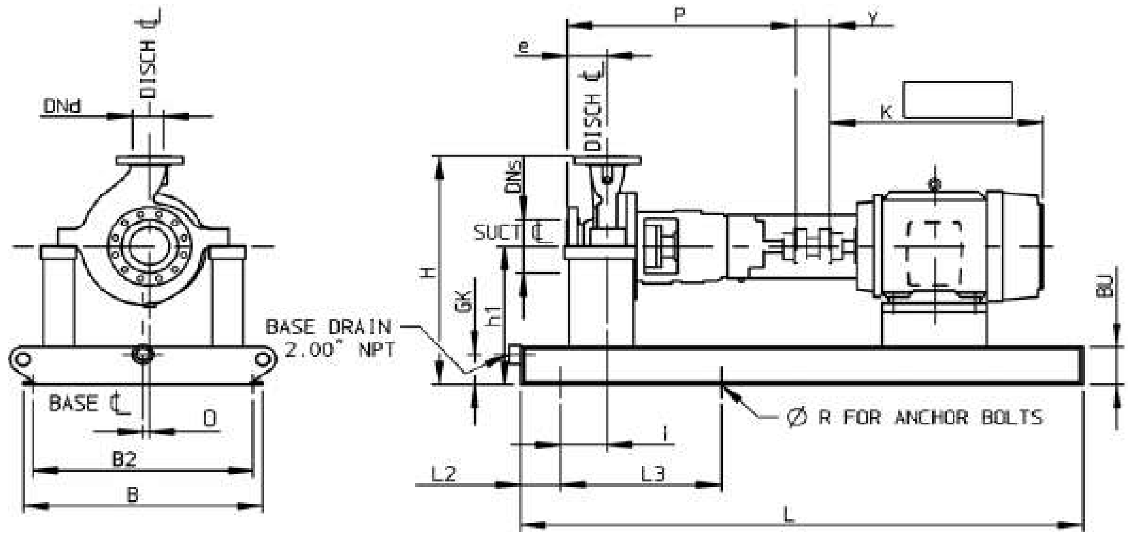
	Power	SPL	Octave Band Center Frequency in Hertz								
Basis OHH	HP	dBA	31.5	63	125	250	500	1000	2000	4000	8000
1x2x7.5-1	8	80.1	72	76	78	79	77	75	72	69	65
2x3x7.5A-1	15	82.2	72.1	78.1	78.1	81.1	79.1	77.1	74.1	71.1	67.1
2x3x7.5B-1	18	82.5	72.4	78.4	78.4	81.4	79.4	77.4	74.4	71.4	8.5
3x4x7.5-1	29	82.2	74.1	78.1	80.1	81.1	79.1	77.1	74.1	71.1	67.1
4x6x7.5B-1	37	81.5	73.4	77.4	79.4	80.4	78.4	76.4	73.4	70.4	66.4
4x6x7.5A-1	54	83.2	75.1	79.1	81.1	82.1	80.1	78.1	75.1	72.1	68.1
6x6x7.5-1	74	86	69.2	78.2	75.2	78.2	81.2	81.2	79.2	76.2	73.2
1.5x3x8-1	12	80.3	72.2	76.2	78.2	79.2	77.2	75.2	72.2	69.2	65.2
2x3x8-1	30	82.4	74.3	78.3	80.3	81.3	79.3	77.3	74.3	71.3	67.3
3x4x8A-1	37	81.6	73.5	77.5	79.5	80.5	78.5	76.5	73.5	70.5	66.5
3x4x8B-1	47	85.8	70.6	79.6	76.6	79.6	82.6	80.6	78.6	75.6	72.6
3x4x8B-2	47	85.8	70.6	79.6	76.6	79.6	82.6	80.6	78.6	75.6	72.6
1x2x9-1	17	81.8	73.6	77.6	79.6	80.6	78.6	76.6	73.6	70.6	66.6
1.5x3x9-1	19	82.9	72.8	78.8	78.8	81.8	798	77.8	74.8	71.8	67.8
2x3x9A-1	25	83.3	73.3	79.3	79.3	82.3	80.3	78.3	75.3	72.3	68.3
2x3x9B-1	34	83.8	73.7	79.7	79.7	82.7	80.7	78.7	75.7	72.7	68.7
2x4x9-1	42	83.2	75.1	79.1	81.1	82.1	80.1	78.1	75.1	72.1	68.1
3x4x10-1	83	85.7	75.8	79.8	81.8	82.8	82.8	80.8	77.8	74.8	70.8
3x4x10-2	83	85.7	75.8	79.8	81.8	82.8	82.8	80.8	77.8	74.8	70.8
3x4x9-1	71	84	75.8	79.8	81.8	82.8	80.8	78.8	75.8	72.8	68.8
3x4x9-2	71	84.3	76.2	80.2	82.2	83.2	81.2	79.2	76.2	73.2	69.2
3x6x9-1	82	84.2	76.1	80.1	82.1	83.1	81.1	79.1	73.1	73.1	69.1
3x6x9-2	82	84.5	76.4	80.4	82.4	83.4	81.4	79.4	76.4	73.4	69.4
4x6x9-1	109	85.4	75.5	79.5	81.5	82.5	82.5	80.5	77.5	74.5	70.5
4x6x9-2	109	85.4	75.5	79.5	81.5	82.5	82.5	80.5	77.5	74.5	70.5
6 x 6 x 9-1	104	85.9	76	80	82	83	83	81	78	75	71
6x8x9-1	188	87.4	70.7	79.7	76.7	79.7	82.7	82.7	80.7	77.7	74.7
8x8x10-1	330	88.5	71.8	80.8	77.8	80.8	83.8	83.8	81.8	78.8	75.8
8x8x10-2	330	88.5	71.8	80.8	77.8	80.8	83.8	83.8	81.8	78.8	75.8
1x3x11.5-1	32	82	73.9	77.9	79.9	80.9	78.9	76.9	73.9	70.9	66.9
1.5x3x13-1	68	85.5	77.4	81.4	83.4	84.4	82.4	80.4	77.4	74.4	70.4
1.5x3x11.5-1	55	85.2	75.1	81.1	81.1	84.1	82.1	80.1	77.1	74.1	70.1
2x4x11.5-1	77	84.8	76.6	80.6	82.6	83.6	81.6	79.6	76.6	73.6	69.6
2x4x13-1	106	85.9	77.7	81.7	83.7	84.7	82.7	70.7	77.7	74.7	70.7
3x4x11.5-1	90	86	75.9	81.9	81.9	84.9	82.9	80.9	77.9	74.9	70.9
3x4x11.5-2	90	85	76.9	80.9	82.9	83.9	81.9	79.9	76.9	73.9	69.9
3x4x13-1	153	87.2	77.3	81.3	83.3	84.3	84.3	82.3	79.3	76.3	72.3
3x4x13-2	153	87.4	77.3	83.3	83.3	86.3	84.3	82.3	79.3	76.3	72.3
3x6x11.5B-1	124	86.5	76.4	82.4	82.4	85.4	83.4	81.4	78.4	75.4	71.4
3x6x11.5B-2	124	85.5	77.4	81.4	83.4	84.4	82.4	80.4	77.4	74.4	70.4
3x6x11.5A-1	149	84.2	76.1	80.1	82.1	83.1	81.1	79.1	76.1	73.1	69.1
3x6x11.5A-2	149	85.8	77.7	81.7	83.7	84.7	82.7	80.7	77.7	74.7	70.7
4x6x11.5-1	187	87.1	77	83	83	86	84	82	79	76	72

SULZER

DIMENSIONAL DATA - TYPE OHH

General Arrangement Dimensions – End Suction – US Customary Units

Series **2.00**	Page
DIMENSIONS	US
October - 02	**D02**



Certified by :_________________________

Date: ______________ Certified for Nozzle & Anchor Bolt locations only when signed above

Ref No. : ________________________________

Project Name : ____________________________

Project Item No.: __________________________

HH -- Number of Base Hold Down Holes each side, equally spaced.

L3 -- Distance between holes. GH -- # of Grout Holes

API No.	Base wt lbs	B	B2	BU	L	L2	L3	R	GK	HH	GH
1.5	800	30	27	6	72.5	6	30.25	1	4.75	3	4
2.0	930	30	27	6	84.5	6	24.16	1	4.75	4	4
3.5	1100	36	33	6	84.5	6	24.16	1	4.75	4	4
4.0	1190	36	33	6	96.5	6	28.16	1	4.75	4	4
13	Non API Baseplate. Used when the standard API baseplates are too small to accommodate the required driver										

Dimension K is derived from NEMA standards or a motor supplier catalogue. The rotation is counterclockwise – H.I. looking from Drive End . All dimensions are nominal inches and for guidance only. Certified Drawings will be issued for actual construction. **When Available** "TS" frame should be used to minimize coupling size, there is no price difference on drivers.

WP-XXX

PUMP SIZE	PUMP wt lbs	DNs	DNd	O	B	e	P	h1	i	H	y	BASE REFERENCE (Electric Driver - NEMA Frame)
		in										
4x6x9-1	450	6	4	1.25	36	5.91	32.48	20	**7.13**	32.99	5	3.5 (143T-326T) (284 TS-365TS) 4.0 (364T-445T) (405TS-447TS)
4x6x9-2	450	6	4	1.25	36	5.91	32.48	20	**7.13**	32.99	5	
6 x 6 x 9-1	680	6	6	0	36	6.22	35.63	22	**7.13**	36.17	7	
6x8x9-1	768	8	6	1.25	36	8.07	38.03	22	**7.13**	36.96	7	
1.5x3x10-1	354	3	1.5	0	30	5.91	32.17	18	**7**	27.84	5	1.5 (143T-286T) (284TS-286TS) 2.0 (324T-405T) (324TS-405TS)
3x4x10-1	384	4	3	0	30	6.50	32.95	20	**7.38**	30.24	5	
3x4x10-2	384	4	3	0	30	6.50	32.95	20	**7.38**	30.24	5	
8x8x10-1	797	8	8	1.25	36	9.21	39.41	22	**7.13**	37.35	10	4.0 (364T) (105TS-445TS) 13 (447TS-Non NEMA Frame Sizes)
8x8x10-2	797	8	8	1.25	36	9.21	39.41	20	**7.13**	35.35	10	
1x3x11.5-1	514	3	1	1.25	36	5.91	33.98	20	**7.38**	30.43	5	3.5 (143T-326T) (284TS-365TS) 4.0 (364T-447T) (405TS-449TS)
1.5x3x11.5-1	522	3	1.5	0	36	5.91	34.02	20	**7.13**	31.02	5	
2x4x11.5-1	541	4	2	0	36	5.91	34.09	20	**7.25**	31.42	5	
3x4x11.5-1	549	4	3	0	36	5.91	34.13	20	**7.25**	31.42	5	
3x4x11.5-2	549	4	3	0	36	5.91	34.13	20	**7.25**	31.42	5	
3x6x11.5A-1	582	6	3	0	36	5.91	34.21	20	**7.25**	33.58	5	
3x6x11.5A-2	582	6	3	0	36	5.91	34.21	20	**7.25**	33.58	5	
3x6x11.5B-1	586	6	3	0	36	5.91	34.13	20	**7.25**	33.58	5	
3x6x11.5B-2	586	6	3	0	36	5.91	34.13	20	**7.25**	33.58	5	
4x6x11.5-1	626	6	4	1	36	5.91	34.29	22	**7.13**	36.37	5	
4x6x11.5-2	626	6	4	1	36	5.91	34.29	22	**7.13**	36.37	5	

Pump 3:

SULZER

TECHNICAL DATA - TYPE OHH

Mechanical Dimensions & Ratings – US Standard Units

Series 2.00 TECHNICAL DATA September - 02	Page US B17

OHH TECHNICAL DATA	US	1x2x7.5-1	2x3x7.5A-1	2x3x7.5B-1	3x4x7.5-1	4x6x7.5A-1	4x6x7.5B-1	6x6x7.5-1
Max/Min Impeller Dia. (in)		7.5/4.5	7.5/4.5	7.5/4.5	7.5/4.5	7.5/5.5	7.0/5.0	7.5/6.8
Volute Construction		Single	Single	Single	Single	Single	Single	Double
Max Operating Pressure (Psig)		740	740	740	740	740	740	740
Hydrostatic Test Pressure (Psig)	6	1110	1110	1110	1110	1110	1110	1110
Max Operating Temperature (°F)		Refer to series B11-B16 for max pump operating temperatures for each material class						
Size of Casing Drain Construction NPT		0.75	0.75	0.75	0.75	0.75	0.75	0.75
Bearing Housing No.		3	3	3	3	3	3	3
Shaft Dia. between bearings (in)		2.56	2.56	2.56	2.56	2.56	2.56	2.56
Span between bearings (in)		7.34	7.34	7.34	7.34	7.34	7.34	7.34
Span CL Rad Brg to CL Imp (in)		10.16	10.28	10.28	10.35	10.71	10.71	11.61
Shaft Dia. at Seal Chamber (in)		1.89	1.89	1.89	1.89	1.89	1.89	1.89
Shaft Dia. at coupling (in)		1.26	1.26	1.26	1.26	1.26	1.26	1.26
Typical API Baseplate #	7	1.5	1.5	1.5	1.5	2.0	1.5	2.0
Radial Bearing Number		6310	6310	6310	6310	6310	6310	6310
Thrust Bearing Number		7311	7311	7311	7311	7311	7311	7311
Pump Weight (lb)		312	332	332	352	399	399	522
Typical API Baseplate weight (lb)	7	802	802	802	802	932	802	932
Minimum Case Thickness (in)		Refer to series B40-B41						
Max. Dia. Spherical Solids (in)	3	0.20	0.79	0.43	0.51	0.54	0.43	0.31
Wear Ring Diameter - Eye (in)		4.13	4.53	4.53	5.12	6.10	6.10	6.50
Wear Ring Diameter - Hub (in)		4.13	4.53	4.53	5.12	5.12	5.12	5.12
Clearance Below 500°F - Eye (in)	1	0.015	0.016	0.016	0.017	0.018	0.018	0.018
Clearance Below 500°F - Hub (in)	1	0.015 (8)	0.016 (8)	0.016 (8)	0.017 (8)	0.017 (8)	0.017 (8)	0.017 (8)
Mass Moment of Inertia (lb-ft2) WR^2	5	0.48	0.52	0.53	0.64	0.83	0.77	1.22
Shaft Stiffness Factor L3/D4	4	82	85	85	87	96	96	123
Critical Speed (Dry) (cpm)		11141	10557	10647	9710	8458	8511	5869
Temperature Limits (°F)		Refer to series B42 for recommended pump and bearing features required for elevated temperatures						

1) Above 260°C (500°F) API requires that clearances are increased by 0.005 in. Pump efficiency must be reduced to compensate for increased internal leakage. Also be sure to correct the power and efficiency for dual seals and high suction pressures.
2) ANSI class 300 RF, EN1092-1 PN64, PN40 flanges are available. Pump operating conditions are not to exceed flange design working pressure and/or temperature. Note: On C-6 and A-8 pumps, the use of higher rated flanges for auxiliary casing connections may be necessary to equal the casing MAWP
3) Maximum spherical diameter is 0.12 in. less than narrowest internal passage.
4) See page B44 for additional information and intended application regarding L3/D4 shaft stiffness evaluation
5) The values shown are for "Wet" WR2 (includes the pumpage contained in the impeller). The coupling WR2 value (to be supplied by coupling manufacturer) should be added to the impeller.
6) Auxiliary process piping shall be tested to the pressure of the casing. (Tubing or threaded piping Is Not hydrotested).
7) Typical Baseplate size and weights are an average provided to speed quoting activity and estimating. For an exact size and weight for the baseplate, use motor frame and pump information to pick a baseplate from pages D01 to D24.
8) These pumps are furnished without hub wear rings and throat bushing to pressurize the seal chamber above vapor pressure and extend seal life.

SULZER

TECHNICAL DATA - TYPE OHH

Noise Calculations – US Customary Units

The following tabulation shows estimated noise level data for Type **OHH** pumps and is presented in either
a) "Sound Pressure Level" (dB) versus "Octave Center Frequency" (Hz).
b) "Weighted Sound Pressure Level" (dBA).

The Table lists values for each **OHH** pump size using:
- Full diameter impellers.
- Running at best efficiency capacity.
- Adequate NPSHa.
- Rated BHP at best efficiency with 1.0 SpGr.

Measurements were taken
- At three feet or one meter from pump or piping surfaces.
- At 60 Hertz design speed with the appropriate performance curve.

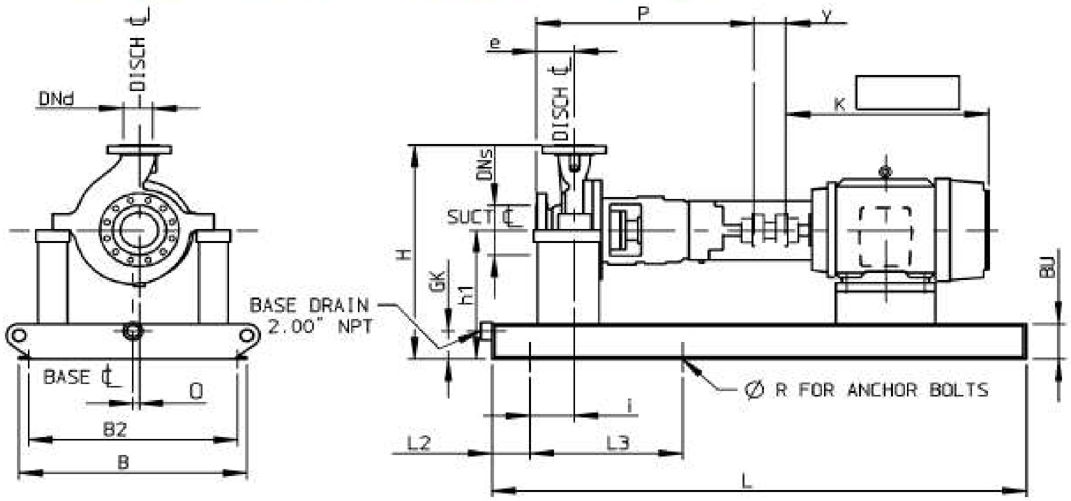
	Power	SPL	Octave Band Center Frequency in Hertz								
Basis OHH	HP	dBA	31.5	63	125	250	500	1000	2000	4000	8000
1x2x7.5-1	8	80.1	72	76	78	79	77	75	72	69	65
2x3x7.5A-1	15	82.2	72.1	78.1	78.1	81.1	79.1	77.1	74.1	71.1	67.1
2x3x7.5B-1	18	82.5	72.4	78.4	78.4	81.4	79.4	77.4	74.4	71.4	8.5
3x4x7.5-1	29	82.2	74.1	78.1	80.1	81.1	79.1	77.1	74.1	71.1	67.1
4x6x7.5B-1	37	81.5	73.4	77.4	79.4	80.4	78.4	76.4	73.4	70.4	66.4
4x6x7.5A-1	54	83.2	75.1	79.1	81.1	82.1	80.1	78.1	75.1	72.1	68.1
6x6x7.5-1	74	86	69.2	78.2	75.2	78.2	81.2	81.2	79.2	76.2	73.2
1.5x3x8-1	12	80.3	72.2	76.2	78.2	79.2	77.2	75.2	72.2	69.2	65.2
2x3x8-1	30	82.4	74.3	78.3	80.3	81.3	79.3	77.3	74.3	71.3	67.3
3x4x8A-1	37	81.6	73.5	77.5	79.5	80.5	78.5	76.5	73.5	70.5	66.5
3x4x8B-1	47	85.8	70.6	79.6	76.6	79.6	82.6	80.6	78.6	75.6	72.6
3x4x8B-2	47	85.8	70.6	79.6	76.6	79.6	82.6	80.6	78.6	75.6	72.6
1x2x9-1	17	81.8	73.6	77.6	79.6	80.6	78.6	76.6	73.6	70.6	66.6
1.5x3x9-1	19	82.9	72.8	78.8	78.8	81.8	798	77.8	74.8	71.8	67.8
2x3x9A-1	25	83.3	73.3	79.3	79.3	82.3	80.3	78.3	75.3	72.3	68.3
2x3x9B-1	34	83.8	73.7	79.7	79.7	82.7	80.7	78.7	75.7	72.7	68.7
2x4x9-1	42	83.2	75.1	79.1	81.1	82.1	80.1	78.1	75.1	72.1	68.1
3x4x10-1	83	85.7	75.8	79.8	81.8	82.8	82.8	80.8	77.8	74.8	70.8
3x4x10-2	83	85.7	75.8	79.8	81.8	82.8	82.8	80.8	77.8	74.8	70.8
3x4x9-1	71	84	75.8	79.8	81.8	82.8	80.8	78.8	75.8	72.8	68.8
3x4x9-2	71	84.3	76.2	80.2	82.2	83.2	81.2	79.2	76.2	73.2	69.2
3x6x9-1	82	84.2	76.1	80.1	82.1	83.1	81.1	79.1	73.1	73.1	69.1
3x6x9-2	82	84.5	76.4	80.4	82.4	83.4	81.4	79.4	76.4	73.4	69.4
4x6x9-1	109	85.4	75.5	79.5	81.5	82.5	82.5	80.5	77.5	74.5	70.5
4x6x9-2	109	85.4	75.5	79.5	81.5	82.5	82.5	80.5	77.5	74.5	70.5
6 x 6 x 9-1	104	85.9	76	80	82	83	83	81	78	75	71
6x8x9-1	188	87.4	70.7	79.7	76.7	79.7	82.7	82.7	80.7	77.7	74.7
8x8x10-1	330	88.5	71.8	80.8	77.8	80.8	83.8	83.8	81.8	78.8	75.8
8x8x10-2	330	88.5	71.8	80.8	77.8	80.8	83.8	83.8	81.8	78.8	75.8
1x3x11.5-1	32	82	73.9	77.9	79.9	80.9	78.9	76.9	73.9	70.9	66.9
1.5x3x13-1	68	85.5	77.4	81.4	83.4	84.4	82.4	80.4	77.4	74.4	70.4
1.5x3x11.5-1	55	85.2	75.1	81.1	81.1	84.1	82.1	80.1	77.1	74.1	70.1
2x4x11.5-1	77	84.8	76.6	80.6	82.6	83.6	81.6	79.6	76.6	73.6	69.6
2x4x13-1	106	85.9	77.7	81.7	83.7	84.7	82.7	70.7	77.7	74.7	70.7
3x4x11.5-1	90	86	75.9	81.9	81.9	84.9	82.9	80.9	77.9	74.9	70.9
3x4x11.5-2	90	85	76.9	80.9	82.9	83.9	81.9	79.9	76.9	73.9	69.9
3x4x13-1	153	87.2	77.3	81.3	83.3	84.3	84.3	82.3	79.3	76.3	72.3
3x4x13-2	153	87.4	77.3	83.3	83.3	86.3	84.3	82.3	79.3	76.3	72.3
3x6x11.5B-1	124	86.5	76.4	82.4	82.4	85.4	83.4	81.4	78.4	75.4	71.4
3x6x11.5B-2	124	85.5	77.4	81.4	83.4	84.4	82.4	80.4	77.4	74.4	70.4
3x6x11.5A-1	149	84.2	76.1	80.1	82.1	83.1	81.1	79.1	76.1	73.1	69.1
3x6x11.5A-2	149	85.8	77.7	81.7	83.7	84.7	82.7	80.7	77.7	74.7	70.7
4x6x11.5-1	187	87.1	77	83	83	86	84	82	79	76	72

SULZER

DIMENSIONAL DATA - TYPE OHH

General Arrangement Dimensions – End Suction – US Customary Units

Series 2.00	Page
DIMENSIONS	US
October - 02	D01



Certified by :______________________
Date: ______________ Certified for Nozzle & Anchor Bolt locations only when signed above
Ref No. : ________________________
Project Name : ______________________
Project Item No.: _____________________
HH -- Number of Base Hold Down Holes each side, equally spaced.
L3 -- Distance between holes.
GH -- # of Grout Holes

API No.	Base wt lbs	B	B2	BU	L	L2	L3	R	GK	HH	GH
1.5	800	30	27	6	72.5	6	30.25	1	4.75	3	4
2.0	930	30	27	6	84.5	6	24.16	1	4.75	4	4

Dimension K is derived from NEMA standards or motor supplier catalogue. The rotation is counterclockwise – H.I. looking from Drive End.
All dimensions are in nominal inches and for guidance only. Certified Drawings will be issued for actual construction.

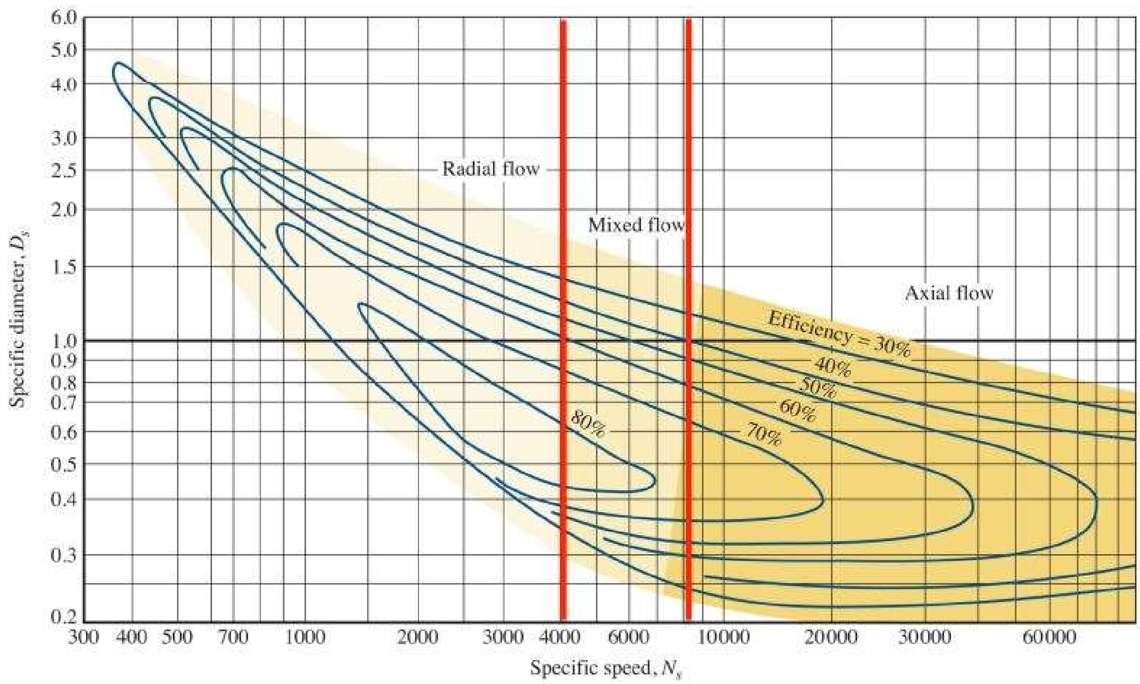
When Available 'TS" frame should be used to minimize coupling size, there is no price difference on drivers. **WP-XXX**

PUMP SIZE	PUMP wt lbs	DNs (in)	DNd (in)	O (in)	B (in)	e (in)	P (in)	h1 (in)	i (in)	H (in)	y (in)	BASE REFERENCE (Electric Driver - NEMA Frame)
1x2x7.5-1	311	2	1	0	30	4.92	31.06	18	**5.63**	27.06	5	1.5 (143T-286T) (284TS-286TS) 2.0 (324T-405T) (324TS-405TS)
2x3x7.5A-1	332	3	2	0	30	4.92	31.18	18	**5.38**	27.06	5	
2x3x7.5B-1	331	3	2	0	30	4.92	31.18	18	**5.38**	27.06	5	
3x4x7.5-1	352	4	3	0	30	4.92	31.26	18	**5.36**	27.06	5	
4x6x7.5B-1	398	6	4	0	30	5.51	32.20	18	**5**	28.04	5	
4x6x7.5A-1	399	6	4	0	30	5.51	32.20	18	**5**	28.04	5	
6x6x7.5-1	522	6	6	1	30	8.07	35.67	18	**5.63**	30.60	7	2.0 (284T-365T) (284TS-365TS)
1.5x3x8-1	335	3	1.5	0	30	5.51	32.01	18	**5.65**	27.06	5	1.5 (143T-286T) (284TS-286TS) 2.0 (324T-405T) (324TS-405TS)
2x3x8-1	336	3	2	0	30	5.91	32.56	18	**5.63**	27.06	5	
3x4x8A-1	360	4	3	0	30	6.30	32.83	18	**5.38**	27.45	5	
3x4x8B-1	370	4	3	0	30	6.30	33.03	18	**5.63**	27.45	5	
3x4x8B-2	370	4	3	0	30	6.30	33.03	18	**5.63**	27.45	5	
1x2x9-1	320	2	1	0	30	5.51	31.65	18	**5.63**	27.06	5	
1.5x3x9-1	334	3	1.5	0	30	5.51	31.69	18	**5.63**	27.06	5	
2x3x9A-1	338	3	2	0	30	5.51	31.73	18	**5.5**	27.45	5	
2x3x9B-1	334	3	2	0	30	5.51	31.81	18	**5.63**	27.45	5	
2x4x9-1	360	4	2	0	30	5.51	31.77	18	**5.38**	28.04	5	
3x4x9-1	374	4	3	0	30	5.51	31.85	20	**7.38**	30.43	5	
3x4x9-2	374	4	3	0	30	5.51	31.85	20	**7.38**	30.43	5	
3x6x9-1	406	6	3	0.5	30	5.91	32.32	20	**7.25**	31.02	5	
3x6x9-2	406	6	3	0.5	30	5.91	32.32	20	**7.25**	31.02	5	

Appendix B Specific Speed Graph

SPECIFIC SPEED

$$N_S = \frac{N\sqrt{Q}}{H^{3/4}}$$



$N_s = \frac{N\sqrt{Q}}{H^{3/4}}$ $D_s = \frac{DH^{1/4}}{\sqrt{Q}}$ N = Rev/min H = Head, ft
Q = Flow, U.S. gpm D = Diameter, in