Full Pipeline System Design for Continental Ag. Manufacturing Plant

AN ENGINEERING DESIGN PROJECT

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Purpose

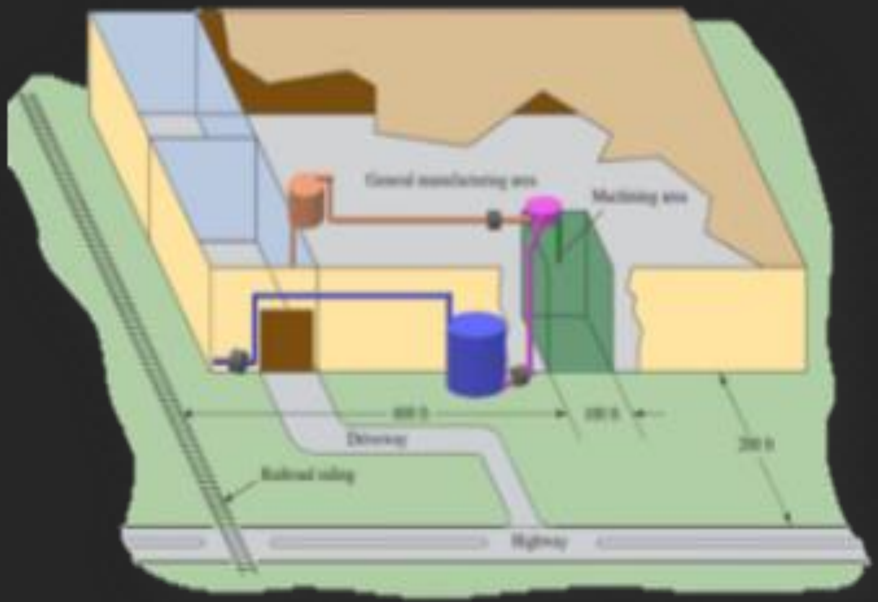
We were tasked with designing a full pipeline system for a manufacturing plant named Continental Ag. in Dayton, Ohio.

This involved developing several main sub-systems: one to supply the plant's machines with coolant, one to transfer coolant to a storage facility, and one transfer the coolant to a truck for transporting and processing.

This was a semester-long project that demonstrates the concepts learned in class and in our assignments.

The full pipeline system was designed from scratch based on set constraints provided by the instructor.

Note: Slides 4 through 14 demonstrate some of the hand calculations I completed for this project.



Project Tasks

There were 21 required tasks that we had to complete that pertained to the following:

- Storage tanks (size, location, thickness, wind load, and weight)
- Energy losses in systems
- Valves, elbows, fittings in systems and piping
- Pumps and their requirements (Mechanical/Electrical)
- Instrumentation for measuring flow
- A Bill of Materials
- Formatting and following a writing rubric for final report

Storage Tanks
Dimensions & Capacities

Our team decided to use Norwesco brand tanks that are made of High-Density Polyethylene (HDPE). The information for three tanks that were used can be seen in the tables to the right.

This type of tank was chosen for the following reasons:

- Cost effective
- Longevity
- Corrosion resistance, no rust will form within
- Lightweight yet strong

The tanks come in a multitude of shapes and sizes and are all vented to atmospheric pressure.

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

Table 1: Tank Size and Location

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

Table 2: Vertical Storage Tank Volumes

Storage Tanks
Wind Load and Weight

We were required to state the wind load and weight of our tanks.

For the wind load, I used the density of air at -20F (Coldest temp. in Dayton, Ohio), and a wind velocity of 12mph (which was found on Accuweather for the day I did the calculation).

For the weight, I considered all tanks to be at maximum capacity. I decided to do this because my weight calculation is for the civil engineers' use foundations and supports. Under normal operating load, the tanks will not be at max capacity, however, if one were to be overfilled, I would want the structure to be able to support the load of a max-filled tank.

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

Table 5: Weight and Wind Loads for Storage Tanks

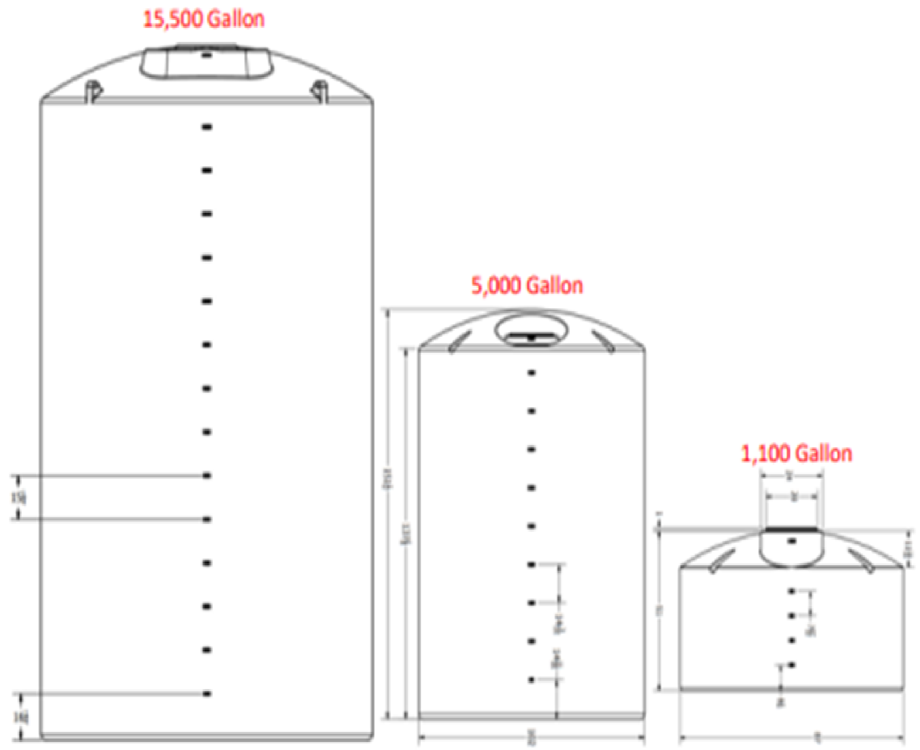
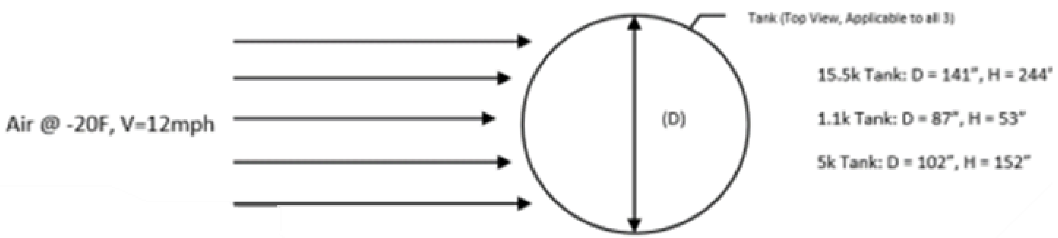


Figure VI-9: Tank Dimensions for Wind Load Calculations



Storage Tanks
Wind Load Calculation

For the wind load calculations, I used the formula for drag force. But before I could do this, I had to determine the drag coefficient for the shape of the tank, the projected area of the tank, and the density of the air at -20F which required interpolation because it wasn't in the Appendices of the textbook.

An example of my work on calculating the wind load on the storage tanks can be found to the right, where I found the wind load on the 15,500-gal storage tank to be 116.221 lbs. for a 12mph wind at a temperature of -20F.

Wind load on 15,500-gallon tank:

$C_D = 1.12 \text{ for semitubular cylinders}$

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

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

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

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

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

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

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

$x = -28.8889°C$	$y = \rho_{AIR} @ -28.8889°C$
$x_1 = -30°C$	$y_1 = 1.452\ kg/m^3$
$x_2 = -20°C$	$y_2 = 1.394\ kg/m^3$

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

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

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

Using the drag force equation, plug in all variables

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

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

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

Storage Tanks

Weight Calculation

For the weight calculations, I used the dry weight of the tanks provided by the manufacturer and added this to the weight of the coolant if it were at maximum capacity.

To find the weight of the coolant, I simply used the formula for volume of a cylinder, found the density of the coolant, and solved for the weight of the coolant using the mass equation (Mass = Density x Volume).

To the right is an example of my weight calculations. This one is for the 5000-gal dirty coolant storage tank.

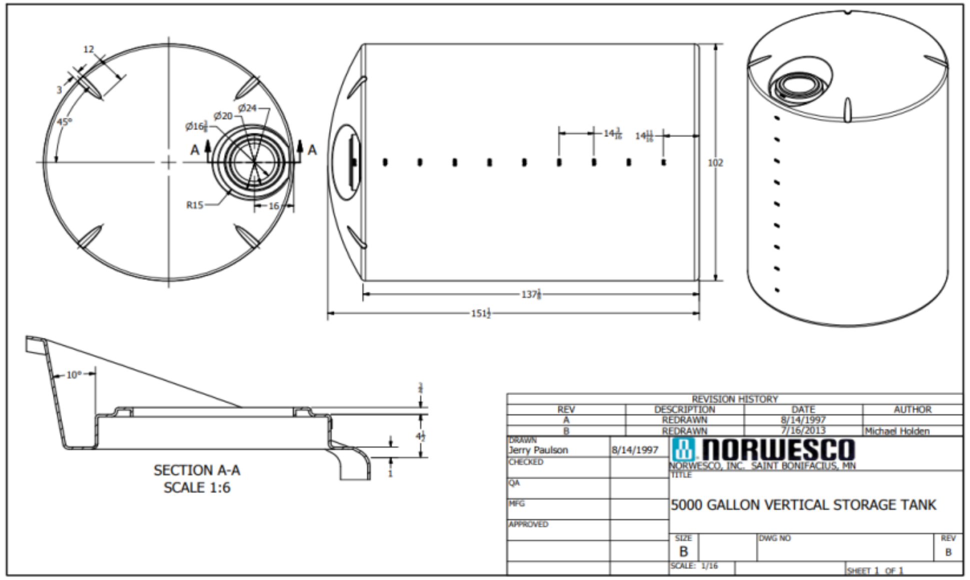


Figure VI-3: 5,000 Gallon Vertical Storage Tank

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

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

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

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

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

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

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

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

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

Instrumentation

Flow Nozzle & Mercury Manometer

For one task, the requirement was to pick an instrument to measure the flow of the pressure differential type then specify its dimensions, type of pressure gauge, and the range of pressures to measure.

For this, I chose a flow nozzle with a monometer and placed it in System 2, that way flow rate could be measured in the piping that supplies the 1100 gal. coolant storage tank above the machines. For all of out systems, the pipe chosen was 3” Schedule 40 Steel piping.

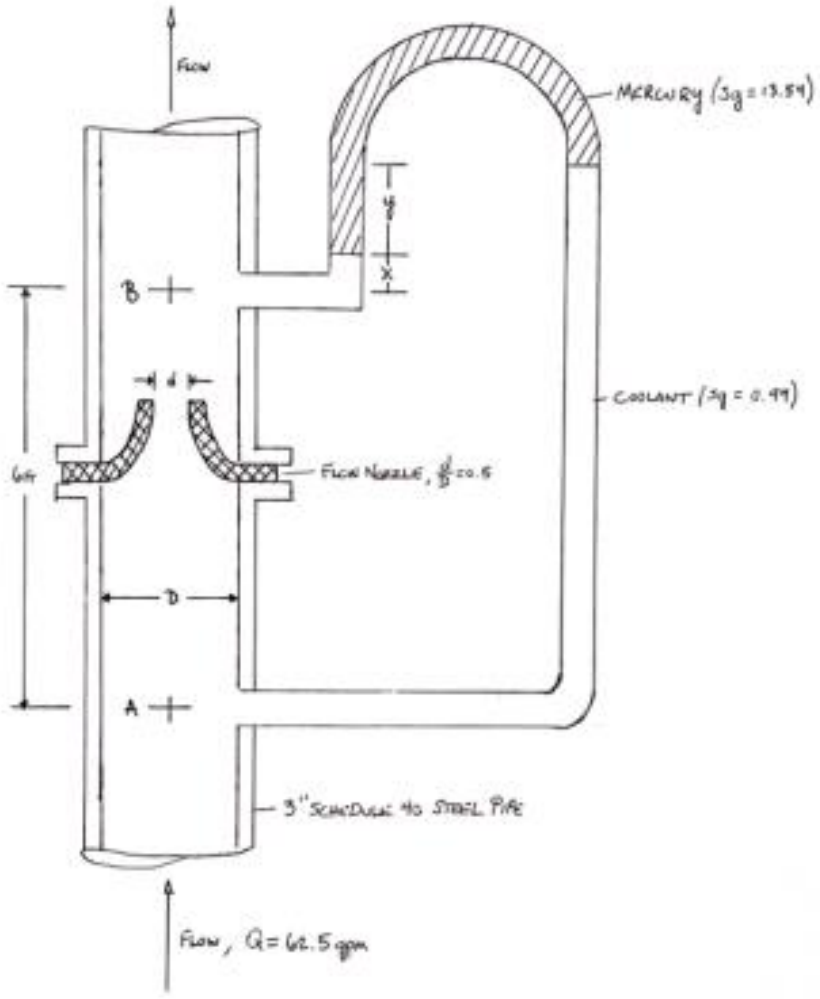


Figure VI-39: Flow Nozzle with Manometer

Instrumentation

Flow Nozzle & Mercury Manometer

I first solved for the pressure drop across the flow nozzle by finding Reynold's number and then manipulating the equation for a flow meter. Then, I plugged in my known values and determined the pressure drop to be 0.7342psi.

To determine the length of the manometer and deflection of gage fluid, I applied the manometric equation to Figure VI-39 and determined that the LHS of the manometer should be minimum if 0.3131 ft in length for the calculated pressure drop. The RHS of the manometer would have to be 6.3131 ft.

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

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

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

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

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

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

Solving for pressure drop across the flow nozzle,

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

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

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

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

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

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

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

$$y = 0.3131\ ft + 6\ ft$$

$$\boxed{y = 6.3131\ ft}$$

Piping Supports

System 1: Forces & Deflection

For this task, we were required to provide the types of supports, distance between supports, and the forces on the supports for one of our systems.

To do this, I separated the system into sections, solved for the sum of the relevant reaction forces (vertical & horizontal), and the weight of the piping itself, then determined deflection in the piping using the formula for deflection of a simply supported beam under a uniform distributed load. The goal was to ensure the maximum deflection during normal operating conditions would be less than 1% of the overall pipe diameter (3.5”).

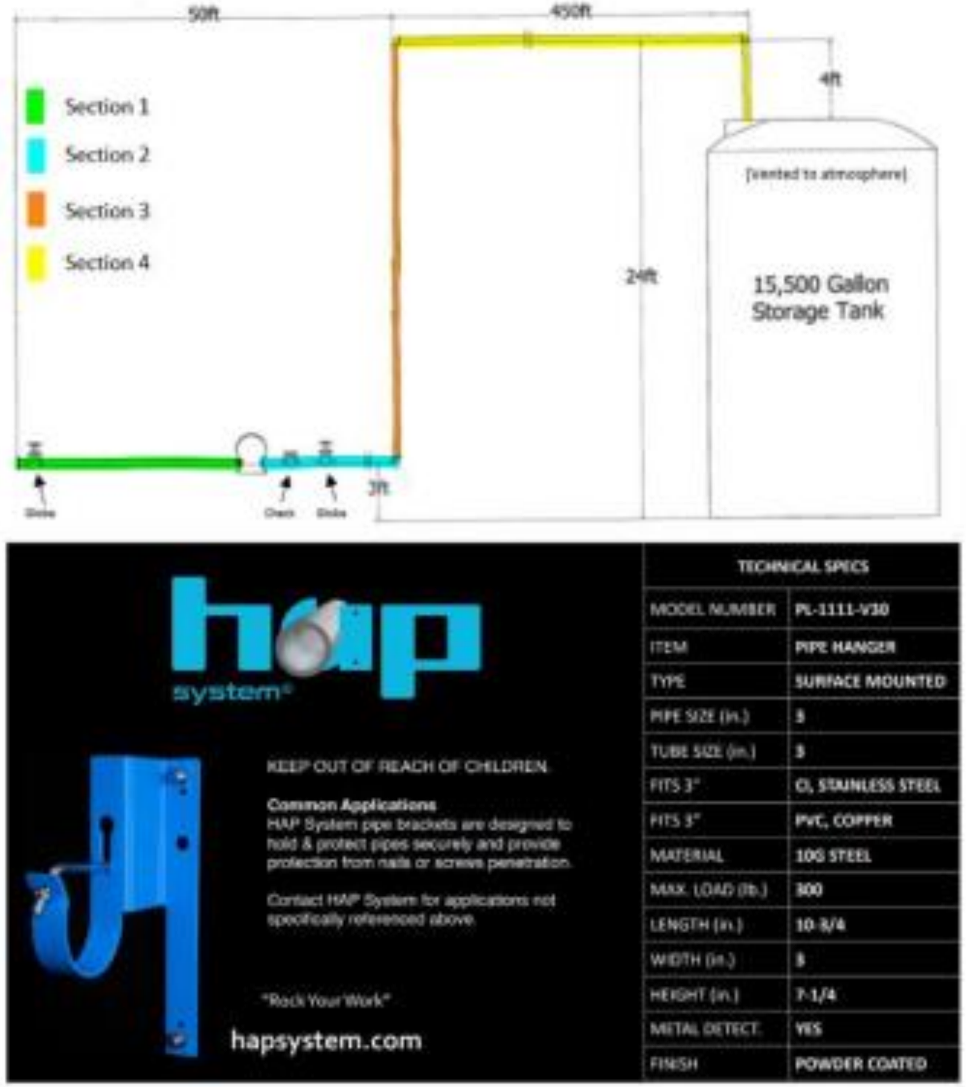


Figure VI-26: Pipeline Support Diagram

Piping Supports

System 1: Forces

The main reason for splitting the system into sections was to make my calculation simpler because I was getting confused by the number of forces and was making mistakes.

Also, there is a pump in the system which is mounted to a foundation on the ground. Therefore, the system must be treated as independent sections for hanger supports.

The forces in Section 4 were the greatest due to the length of the piping being much greater than the others.

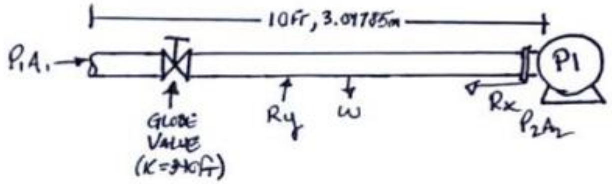


Figure VI-27: System 1 Pipe Support

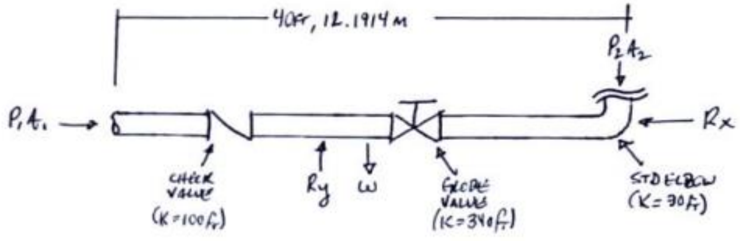


Figure VI-28: System 2 Pipe Support

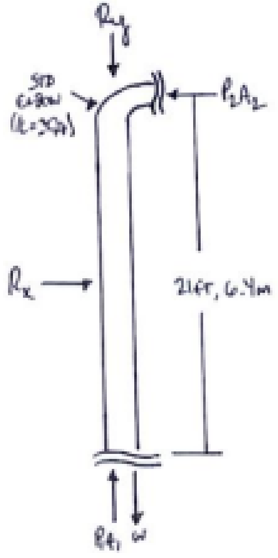


Figure VI-29: System 3 Pipe Support

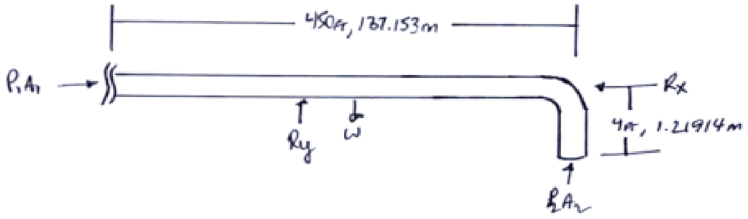


Figure VI-30: System 4 Pipe Support

Piping Supports

System 1: Forces

Here is an example of how the forces were calculated for one of the sections.

System 4:

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

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

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

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

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

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

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

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

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

Piping Supports
System 1: Deflection

To calculate deflection, the section forces were totaled and added to the total weight of the section piping (3” SCH 40 Steel).

Then, the Deflection formula was applied to each section based on weight per length, total length, Young’s modulus of material, and the Moment of Inertia.

The maximum deflection at 12ft between hangers was less

TOTAL FORCES OF SECTIONS (WEIGHT + FORCES)

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

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

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

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

$= 1486.92\text{ lbs}$

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

$= 3903.7\text{ lbs}$

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

DEFLECTION OF PIPING

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

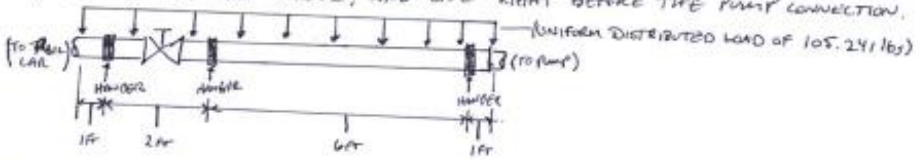
w = FORCE APPLIED PER UNIT LENGTH
L = LENGTH BETWEEN SUPPORTS
E = YOUNGS MODULUS
I = MOMENT OF INERTIA

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

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

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

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



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

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

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

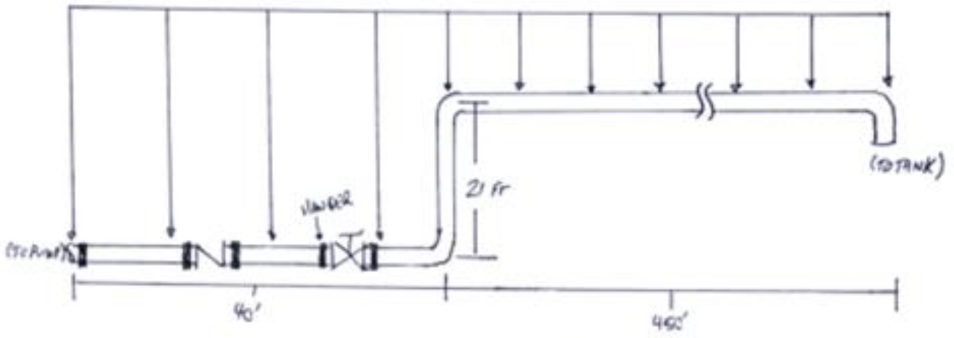
THEREFORE, THE SUPPORT LOCATIONS ARE ADEQUATE.

Piping Supports
System 1: Deflection

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

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

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



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

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

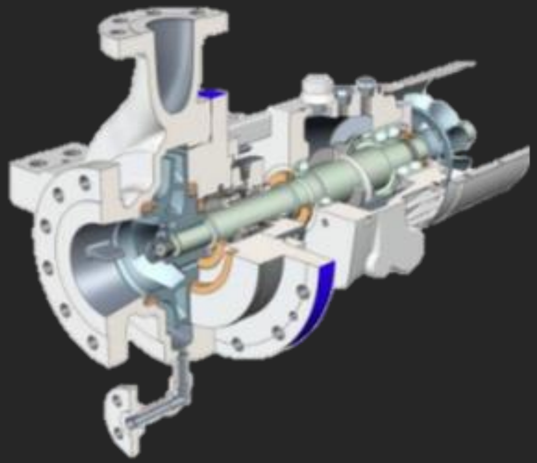
THE SUPPORT HANGER LOCATIONS CAN BE SPACED 12 FT APART OTHER THAN AT VALVES, CONNECTIONS, AND ELBOWS AND WILL ADEQUATELY SUPPORT THE SYSTEM FORCES.

Pump Selection

Calculations from other tasks were used to determine the required pumps for the system. Our system is designed so that the same pump type can be used for each sub-system.

Based on the charts from the Sulzer pump manual and our predetermined system requirements, we decided on three 2x3x7.5 pumps with a 6-inch impeller.

The motor required to power the selected pumps was determined using the manual, a 2-HP Motor with a NEMA Frame Size of 324T-405T.



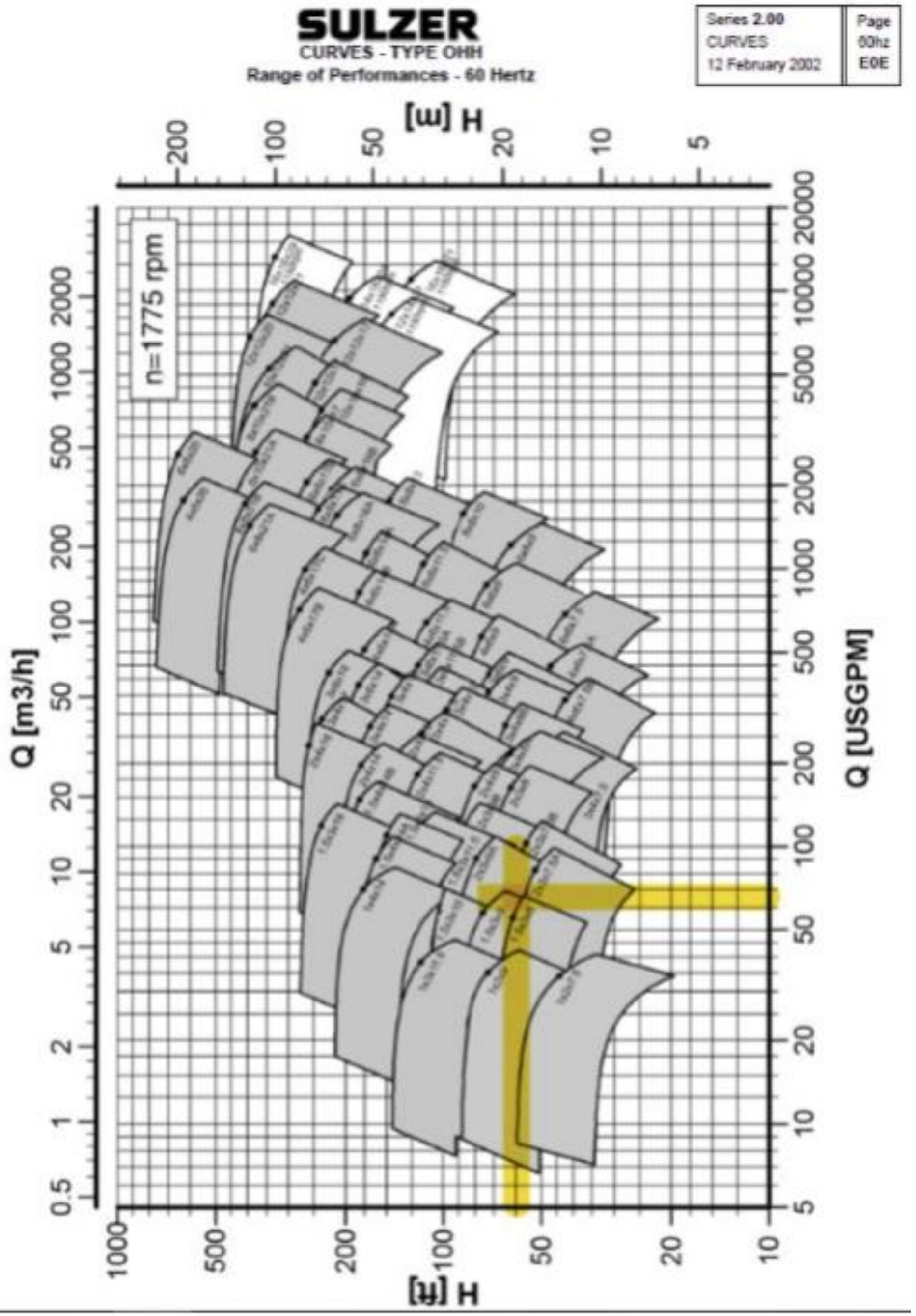


Figure 38: Pump Impeller Design

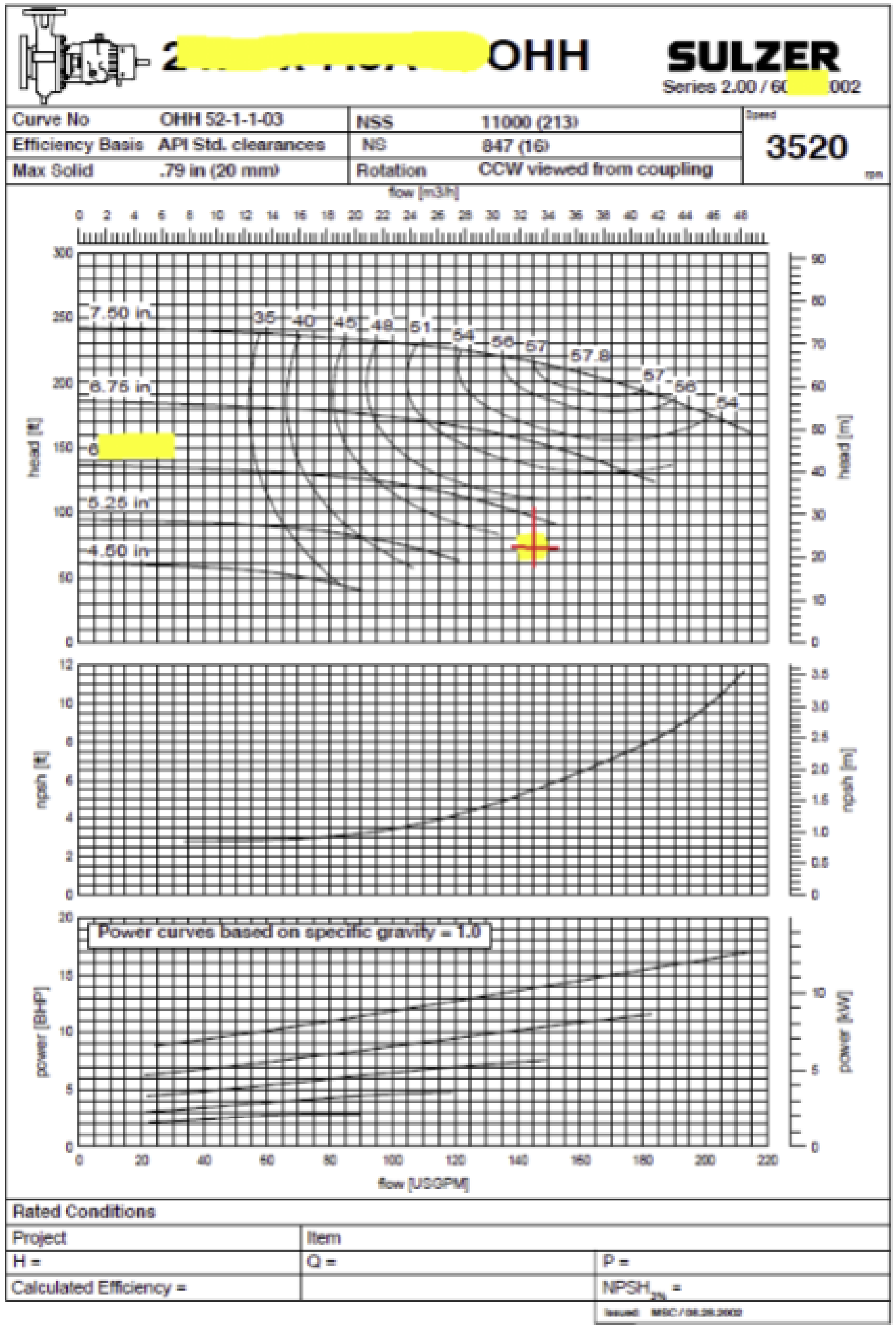


Figure 37: Sulzer 1x2x7.5 Pump

This presentation does not capture the design project in its entirety, or all calculations performed. For more information regarding the design, important figures and sketches, system energy losses, the emergency tank drainage system, materials used, and much more, please see the full report below.

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