Pump Station Design Guidelines – Second Edition

Jensen Engineered Systems
825 Steneri Way
Sparks, NV 89431
For design assistance call (855)468-5600
JensenEngineeredSystems.com

TABLE OF CONTENTS

INTRODUCTION 3

PURPOSE OF THIS GUIDE 3

OVERVIEW OF A TYPICAL JES SUBMERSIBLE LIFT STATION 3

DESIGN PROCESS 3

BASIC PUMP SELECTION 5

THE SYSTEM CURVE 5

STATIC LOSSES 5

FRICTION LOSSES 6

TOTAL DYNAMIC HEAD 9

HOW PUMPS WORK 13

OVERVIEW OF A SUBMERSIBLE PUMP 13

BASIC IMPELLER THEORY 14

THE CASING 14

THE INLET 14

IMPELLER TYPES 15

MOTORS 17

MECHANICAL SEALS 19

PUMP CURVES 22

STEEPNESS OF PUMP CURVE 24

INTERACTION OF THE SYSTEM CURVE WITH THE PUMP CURVE 24

NET POSITVE SUCTION HEAD 25

INTRODUCTION TO WET WELL DESIGN 26

MINIMUM STORAGE VOLUME 27

SIZE OF WELL 28

HATCH SIZING 28

DIAMETER OF WELL 31

CONTROL ELEVATIONS 32

H_X – MINIMUM SUBMERGENCE 33

H_{MIN} – MINIMUM STORAGE 34

H_{LAG} – LAG STORAGE 34

H_{RES} – RESERVOIR STORAGE 34

INTRODUCTION

PURPOSE OF THIS GUIDE

The intent of this manual is to guide the engineering professional through a typical design of a Jensen Engineered Systems (JES) packaged lift station. These are the same steps and procedures followed by our engineers when we are designing a submersible lift station. JES can always provide you with a project specific design – free of charge. Simply contact one of our design professionals toll free at (855) 468-5600. For those who would like insight into our basis of design, or want to be more hands on in the design process, we hope this manual is of help.

OVERVIEW OF A TYPICAL JES SUBMERSIBLE LIFT STATION

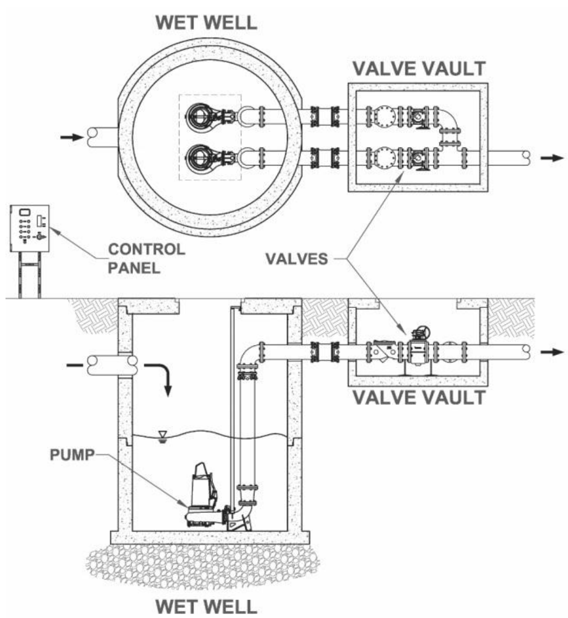


Figure 1

A typical submersible lift station by JES includes a wet well, dual submersible pumps, valves and an electronic pump control system. In smaller stations, the valves will often be installed in the wet well to save infrastructure costs. On larger systems, it is recommended that a separate valve vault be specified to provide easy access in the event maintenance is necessary.

DESIGN PROCESS

The typical design process starts with understanding what type of water needs to be pumped, and the volume of that water. Most JES submersible lift station applications are intended for either stormwater or wastewater. This is an important step, not only because of the obvious differences between the fluids, but also because of some not-so obvious implications which will be discussed later.

Determining the volume of the fluid can be as simple as identifying the fixture unit count in a residential home, or as complicated as preparing a detailed hydrology report for a 50-acre commercial site. The intent of this manual is not to detail these very broad subjects, but to point out the necessity of properly determining flowrates, as well as provide a good understanding of the occurrence of those flows and how they relate to a pump station.

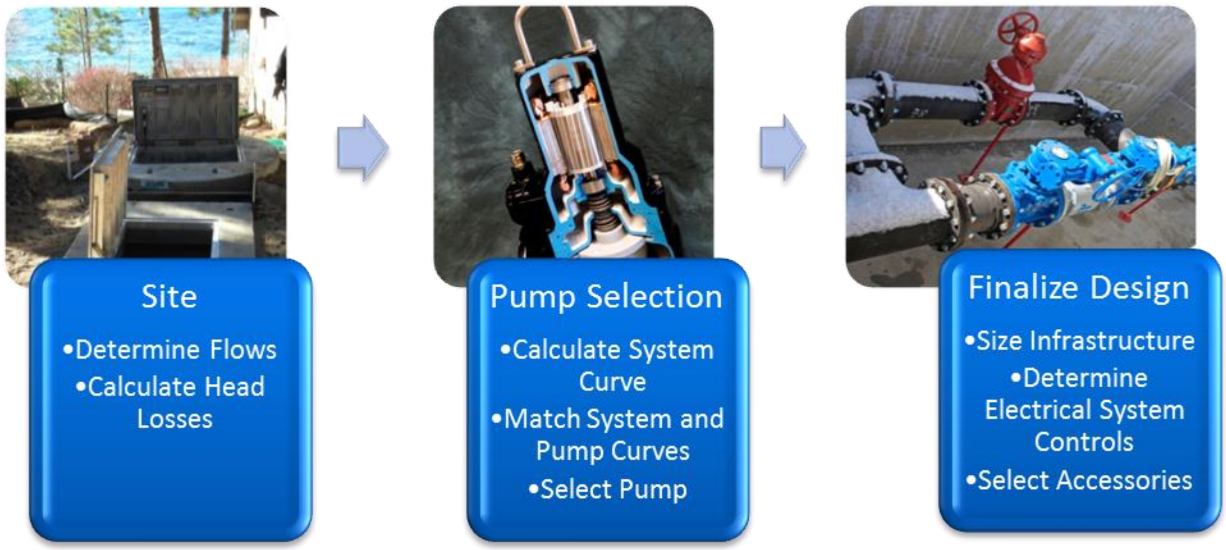


Figure 2

The next steps in design are site considerations. How far does the lift station need to pump? How high does the liquid need to rise? These questions, when coupled with the flowrate, are eventually going to determine the size of pump the system will require. Additional site considerations, such as whether or not the lift station is located in a roadway, will play into various other design criteria. The better the understanding of the site, the easier the lift station will be to design.

Once flow has been determined and site considerations have been taken into account, the system curve can be developed. The system curve is matched with various pump curves in an iterative process to determine which pump will best match the demands of the project. Once the pump is selected, all the additional components, such as the wet well, valve vault, valve and pipes, control system, etc., can be sized.

BASIC PUMP SELECTION

THE SYSTEM CURVE

The most important part of any pump selection is first determining the system curve. This means, at the very least, the flowrate and head that will be required of the pump must be identified. Often this is the first mistake made in the selection process.

In many potable water booster stations, the flowrate is determined by a downstream demand. In a typical JES application, the purpose of the lift station is to simply move water from one location to another. Therefore, the flow is typically governed by the inflow to the station, and not an outflow demand. Once the flowrate into the station is found, the amount of head required by the pump can be determined by calculating the system losses in the piping network. Rule of thumb estimates and outright guesses of the friction losses will lead to poorly sized equipment that will have a poor efficiency and reliability.

Calculation of the system losses at several different flowrates will yield a system curve. System curves represent a loss of energy in systems with a variation in the flowrate. Or, stated differently, the amount of energy the pump must generate to operate at a given flowrate. System losses come in two forms that are outlined below.

STATIC LOSSES

Static losses are due to differences in either elevation or pressure between the inlet and the discharge. In some booster pump systems, the discharge could be in a pressurized holding tank. So tank pressure must be taken into consideration during design. In a typical JES lift station, the force main discharges to gravity in either a manhole or other holding structure.

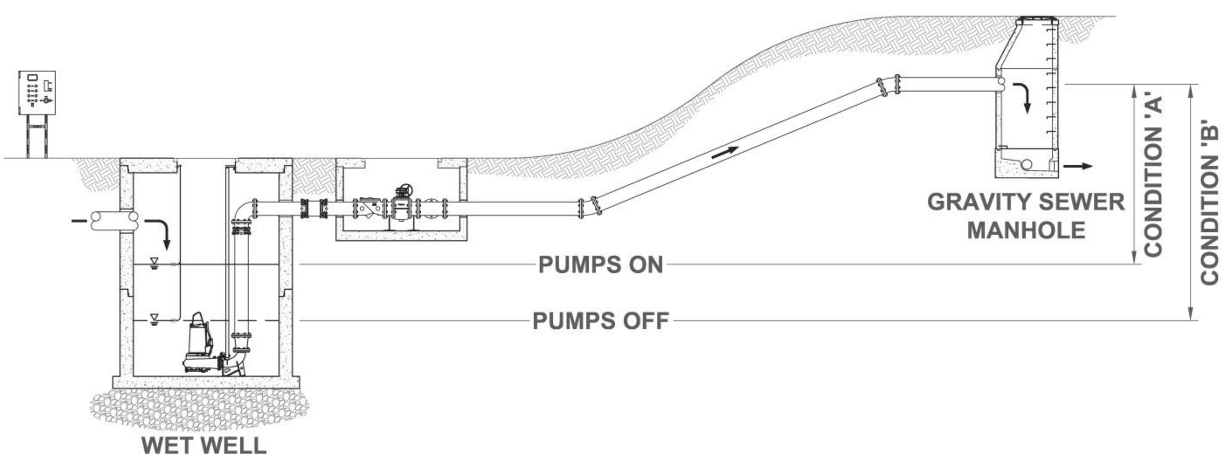


Figure 3

In figure 3, the static head (in Feet) is the difference between the discharge elevation of the force main and the water level of the wet well. It is important to note the difference between condition 'A' and condition 'B'. While the pumps are in the off position, the water level in the wet well rises. When the water level reaches a pre-determined storage elevation, the pumps turn on and the surface elevation draws down until it reaches the pumps off position. Determining these elevations will be addressed later, but for now it is important to understand that the static head is not a fixed value, but rather a floating value dependent upon the water surface elevation in the wet well. Many designers consider the system curve to be a fixed and unchanging representation of the pumping system. However, because the system curve is based upon the static head, which is a fluctuating variable, the system curve itself can fluctuate. Therefore, the system curve is not a single point set – It is a range of curves. This will become more apparent when developing a system curve is discussed later in this manual.

FRICTION LOSSES

The second part of losses in the pumping system are the dynamic losses. Dynamic losses are dependent upon the flowrate through the system and primarily attributed to the system friction. This includes friction of the liquid flowing through the pipe and fittings, as well as the friction internal to the fluid itself. The friction begins at the level of the total static head and increases with flowrate.

The subject of friction losses in a piping network is vast and complex. Example: in designing a water booster station for a residential neighborhood, a detailed pipe analysis of the network, including its physical characteristics and flow demands, would be necessary. Rather than delve deeper is determining these losses in this manual, the following discussion assumes a very simple, single force main, gravity discharge lift station. For projects requiring a more complicated distribution system, contact Jensen Engineered Systems. We utilize advanced computer modeling software to detail the piping network.

There are many different methods for determining friction losses in a pipe. These include The Darcy-Weisbach Equation, along with the Hazen Williams Equation. We will be using the Hazen Williams Equation because of its relative ease for a simple force main. There are some limitations to the Hazen Williams Equation which will be discussed. To explore this subject in further detail, or learn more about other methods, there are some very good current publications: *Pumping Station Design (Revised Third Edition)* by Jones, Sanks, Tchobanoglous, and Bosserman, published by Butterworth-Heinemann, is thought by many to be the most in-depth resource for pump station design. Another publication worth reviewing is *Hydrology and Hydraulic Systems (Second Edition)* by Gupta, published by Waveland Press, Inc.

There are some advantages of using the Hazen Williams Equation. Not only is it simple and easy to use, its use is also required by many regulatory agencies. The Hazen Williams Equation is somewhat constraining because there must be turbulent flow within the pipe, and the fluid type must be water that is at, or near, room temperature. Additionally, the fluid velocity must be between 3 to 9 ft/sec. This last constraint actually lends itself quite well to wastewater lift design because if the wastewater velocity is below 3 ft/sec., there will not be enough energy to scour the pipe of solids. Conversely, water flowing above 9 ft/sec. can scour the pipe material and damage the force main. The final constraint is that the Hazen-Williams Equation should not be used on force mains larger than 60” in diameter.

HAZEN WILLIAM EQUATION

The base form of the Hazen William Equation is as follows:

$$v = 1.318CR^{0.63}S^{0.54}$$

Equation 1

v = velocity (ft/sec)
C = Hazen Williams friction coefficient
R = hydraulic radius (feet)
S = headloss (feet/foot)

HAZEN WILLIAM DESIGN COEFFICIENTS (C)		
PIPE MATERIAL	C_{WATER}	$C_{WASTEWATER}$
DI – unlined	80-120	80-110
DI – cement lined	100-140	100-130
Steel – unlined	110-130	110-130
Steel – cement lined	120-145	120-140
PVC	135-150	130-145
CPP	130-140	120-130

Table 1 - Source: Hydraulic Design Handbook by Mays

The conventional form of the equation has been re-arranged as follows:

$$h_f = (L)10.5\left(\frac{Q}{C}\right)^{1.85} D^{-4.87}$$

Equation 2

h_f = headloss due to friction (feet/feet)
L = Force main length (ft)
Q = flow (gpm)
D = pipe diameter (in)

MINOR LOSSES

As discussed earlier, when water flows through a straight pipe there are energy losses due to the internal friction of the fluid, as well as the friction between the water and the pipe wall. Additional energy losses are encountered as the fluid passes through bends, constrictions, valves, etc. These are referred to as minor losses. The two typical methods of calculating minor losses are K value estimates, and the equivalent length. In this example, the K value estimate method is used which assigns coefficients to various fittings and valves. The following values can be used to estimate minor losses. These values should be verified against specific manufacturers' recommendations.

Entrance	Bellmouth	0.005
	Rounded	0.25
	Sharp-Edged	0.5
	Projecting	0.8
Exits		1.0
90° Bend		0.25
45° Bend		0.18
Tee, line flow		0.30
Tee, branch flow		0.75
Cross, line flow		0.50
Cross, branch flow		0.75
Wye, 45°		0.50

Ball		0.04
Check Valves	Ball	0.9-1.7
	Rubber flapper (v < ft/s)	2.0
	Rubber flapper (v > ft/s)	1.1
	Swing	0.6-2.2
Gate	Double Disc	0.1-0.2
	Resilient seat	0.3
Knife Gate	Metal seat	0.2
	Resilient seat	0.3
Eccentric Plug	Rectangular (80%) opening	1.0
	Full bore opening	0.5

Table 2 – K Values, Source: Pumping Station Design

To calculate the headloss due to minor losses, the following equation can be used:

$$h_m = \Sigma K \frac{v^2}{2g}$$

Equation 3

$$h_m = Minor\ headloss\ (ft)$$

$$K = K\ coefficient\ (unitless)$$

$$v = velocity\ of\ fluid\ in\ pipe\ (ft/s) = \frac{Flow\ (cf/s)}{Cross\ sectional\ area\ of\ pipe\ (sf)}$$

$$g = gravity\ = 32.2\ ft/s$$

TOTAL DYNAMIC HEAD

The total dynamic head (TDH) of a submersible lift station is essentially the total energy loss experienced by the fluid between the pump and the outlet of the station. Another way to think of TDH is the head the pump must overcome to move the fluid to its destination. Consider the following example:

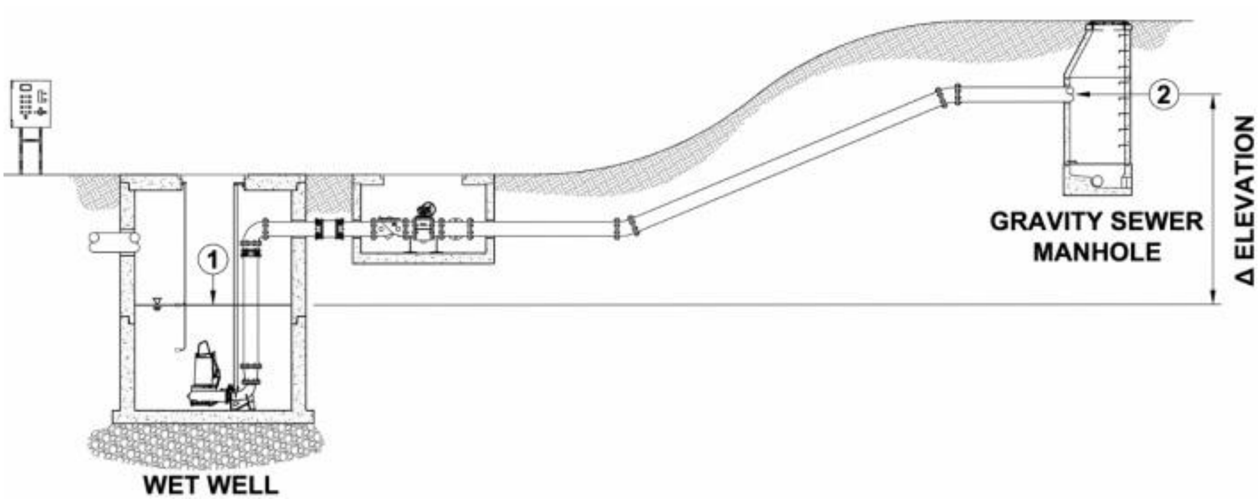


Figure 4

The TDH which the pump must overcome is the summation of the elevation head, headloss due to friction, and the minor headlosses.

$$TDH = \Delta H + h_f + h_m$$

Equation 4

$$TDH = Total\ Dynamic\ Head\ (ft)$$

$$\Delta H = Static\ head = elev_2 - elev_1\ (ft)$$

Plugging equations 2 and 3 into equation 4 results in:

$$TDH = \Delta H + (L)10.5\left(\frac{Q}{C}\right)^{1.85} D^{-4.87} + \Sigma K\,\frac{v^2}{2g}$$

Equation 5

Equation 5 demonstrates that the only variable which is not constant in a submersible lift station system is the flow. By calculating multiple TDH values based on incrementing flow values, and plotting these on a graph of TDH (ft) vs flow (gpm), the system curve is determined. This is demonstrated in the following example:

Example 1:

A submersible lift station is needed for a wastewater application. The gravity sewer manhole to which the lift station is discharging is located 110’ away from the wet well. The invert elevation of the force main at the discharge is 250.0’. The finished grade at the wet well is 244.0’, and the gravity invert to the wet well is at 240.0’. The pump’s off elevation is 236.0’, and the lead pump on elevation is 238.0’. The force main will be schedule 80 PVC and will be 3” in diameter. There will be a 90° bend leaving the wet well, a check valve, a plug valve, and two 45° bends in the force main.

Solution:

- Begin by identifying all of the minor loss coefficients and summing those. Refer to Table 2 for the coefficient values.

Minor Loss	K	# Fittings	Sum K
Entrance	0.25	1	0.25
90 Elbow	0.25	1	0.25
Check Valve	2.2	1	2.2
Plug Valve	1	1	1.0
45 deg	0.18	2	0.36
Exit	1	1	1.0
		SK=	5.1

- By referring to Table 1, a Hazen-Williams Equation coefficient (C value) of 135-145 can be identified. The lower limit of 135 determines the first curve.
- To determine the static head, the maximum elevation difference the pump will need to overcome must be identified. This will be when the water surface elevation in the wet well is at its lowest, or just above pump’s off elevation. To be conservative, use the pump’s off elevation. Therefore:

$$\Delta H = \text{discharge elevation} - \text{pumps off elevation} = 250.0' - 236.0' = 14.0'$$

- The final step before calculating the actual curve is to determine the cross sectional area of the pipe in ft^2. This will be used to determine the velocity at each given flow.

For a 3” Dia. pipe, the Area = $0.05 ft^2$.

- The system curve can now be calculated. The easiest way to do this is by using a spreadsheet with increasing flowrates, starting at zero, which will yield their associated TDH values.

Flow	Flow	Area	Velocity	K	h_m	C	FM Length	h_f	ΔH	TDH
gpm	cfs	(sf)	fps		ft		ft	ft	ft	ft
0	0	0.05	0	5.1	0	135	110	0	14	14
20	0.04	0.05	0.9	5.1	0.07	135	110	0.2	14	14.2
40	0.09	0.05	1.8	5.1	0.26	135	110	0.6	14	14.8
60	0.13	0.05	2.7	5.1	0.59	135	110	1.2	14	15.8
80	0.18	0.05	3.6	5.1	1.05	135	110	2.1	14	17.1
100	0.22	0.05	4.5	5.1	1.63	135	110	3.1	14	18.8
120	0.27	0.05	5.4	5.1	2.35	135	110	4.4	14	20.8
140	0.31	0.05	6.4	5.1	3.2	135	110	5.9	14	23.1
160	0.36	0.05	7.3	5.1	4.18	135	110	7.5	14	25.7

- The system curve can be seen by plotting the total dynamic head in feet versus the flow in gallons per minute.

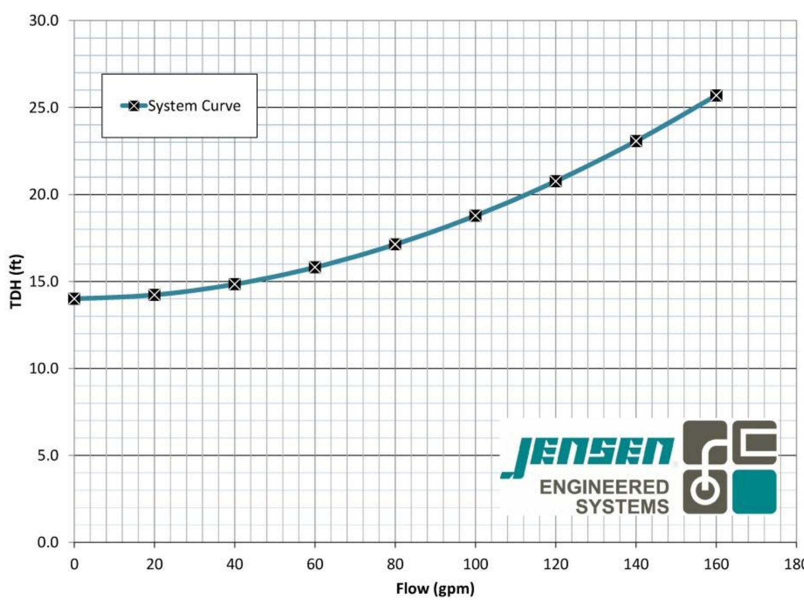


Figure 5

- It is important to think of the system curve as a general area where the system curve could potentially lie. For example; the C value for a brand new pipe versus that of a ten-year-old pipe with corrosion and debris build up could be very different. Also, static head fluctuates through the pump cycle and will shift the system curve up as the water level in the wet well is drawn down. The following curve shows the upper limit with a static head of 14’ and a C value of 135, and the lower limit with a static head of 12’ and a C value of 145.

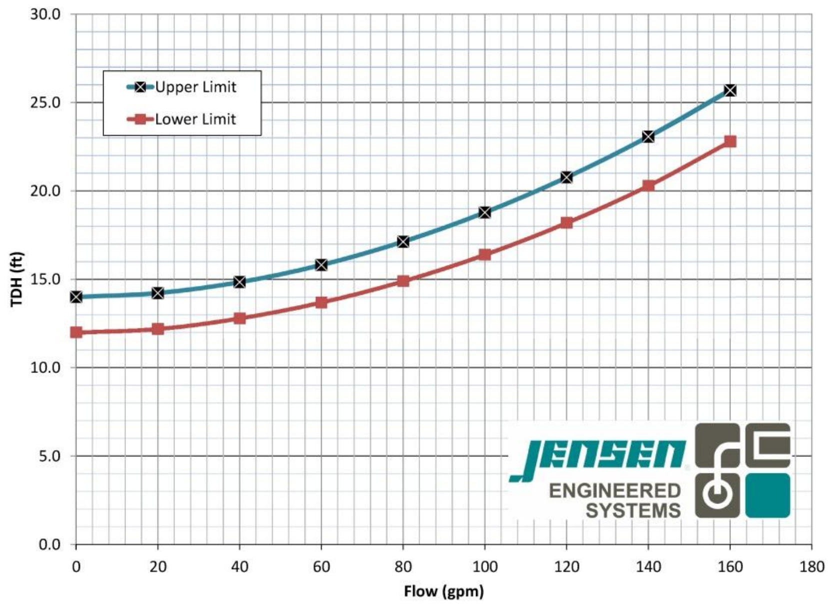
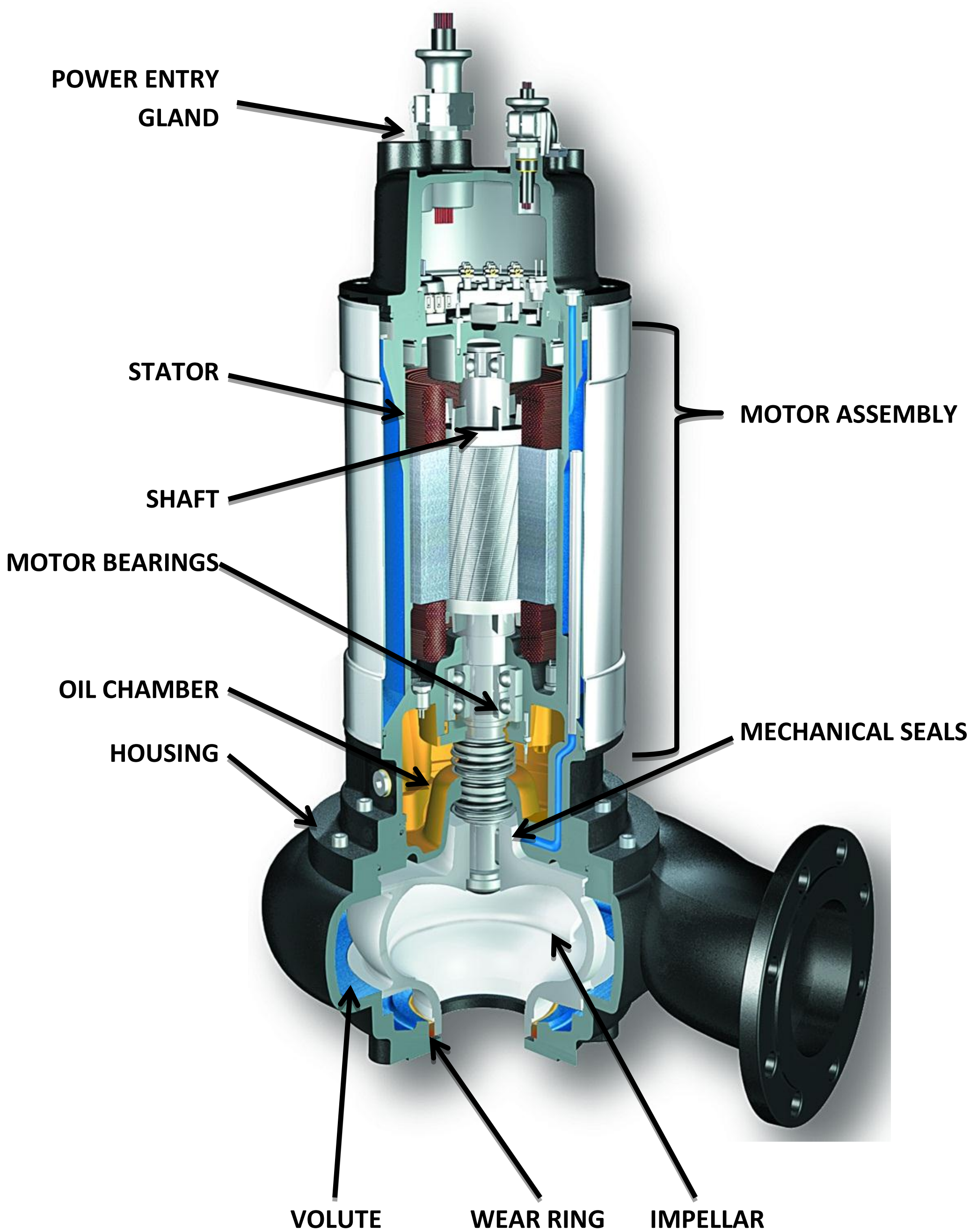


Figure 6

OVERVIEW OF A SUBMERSIBLE PUMP

Figure 7 - Courtesy of HOMA Pumps



BASIC IMPELLER THEORY

The Impeller is the heart of the pump and the only part that adds energy to the liquid. Simply put, energy is added by accelerating the liquid from the smaller radius at the impeller inlet to a larger radius at the impeller exit. The amount of energy input into the fluid can be amplified by increasing the outside diameter of the impeller, or increasing the speed at which it operates.

THE CASING

The energy added by the spinning impeller exits as a high speed fluid, which is generally not very useful for process applications. Pump output usually requires higher pressure, not higher speed. To convert from higher speed to higher pressure, the flow must be diffused (speed reduced) converting high velocity energy into pressure and energy. See Bernoulli's Equation below.

$$\frac{P_1 * 2.31}{sg} + \frac{V_1^{\,2}}{2g} + Z_1 = \frac{P_2 * 2.31}{sg} + \frac{V_2^{\,2}}{2g} + Z_2$$

Equation 6

P = Pressure (psi)
sg = Specific gravity (unitless)
V = Velocity of the fluid (ft/s)
G = Acceleration due to gravity (32.16 ft/sec^2)
Z = Elevation of the centerline of the liquid path
Subscripts:
1 = Upstream condition
2 = Downstream condition

In the pump, the elevation change from point 1 (exiting the impeller in this example) to point 2 (centerline of the volute channel) is generally small, and is considered negligible in most cases. Therefore, only the change in pressure and velocity is left to be considered. In order for the two sides of the equation to balance, decrease in velocity from point 1 to point 2 must have a corresponding increase in pressure from point 1 to point 2. Bernoulli's Equation is a simplified representation of this process. Technically, it only applies to flows along a streamline and neglects friction, but it is sufficient to understand the basic principle.

THE INLET

The following discussion concerning inlets is less relevant in submersible lift station design because there is no inlet piping before the impeller. However, these considerations should be taken into account when designing a dry pit lift station.

The job of the inlet is to convey the liquid from the inlet pipe to the impeller entrance in a fashion that imposes minimal loss and creates the most uniform velocity profile at the impeller entrance. Therefore, the ideal inlet geometry is a straight pipe entrance with a slight taper from the pipe flange to the impeller eye. The taper somewhat increases the velocity and tends to stabilize the fluid streamlines prior to the impeller.

All curved inlets cause at least a minor penalty and, in some cases, a major efficiency penalty. This includes straight inlets with an elbow attached close to the suction flange. It is preferred to have at least 5 diameters of straight pipe (the same size as the inlet flange) leading up to the pump inlet to prevent non-uniform velocity profile at the impeller eye. With submersible pumps, the inlet configuration is planned by the manufacturer based on the mounting of the pump. Additional features may need to be included in a tank mount to optimize the operation of the pump.

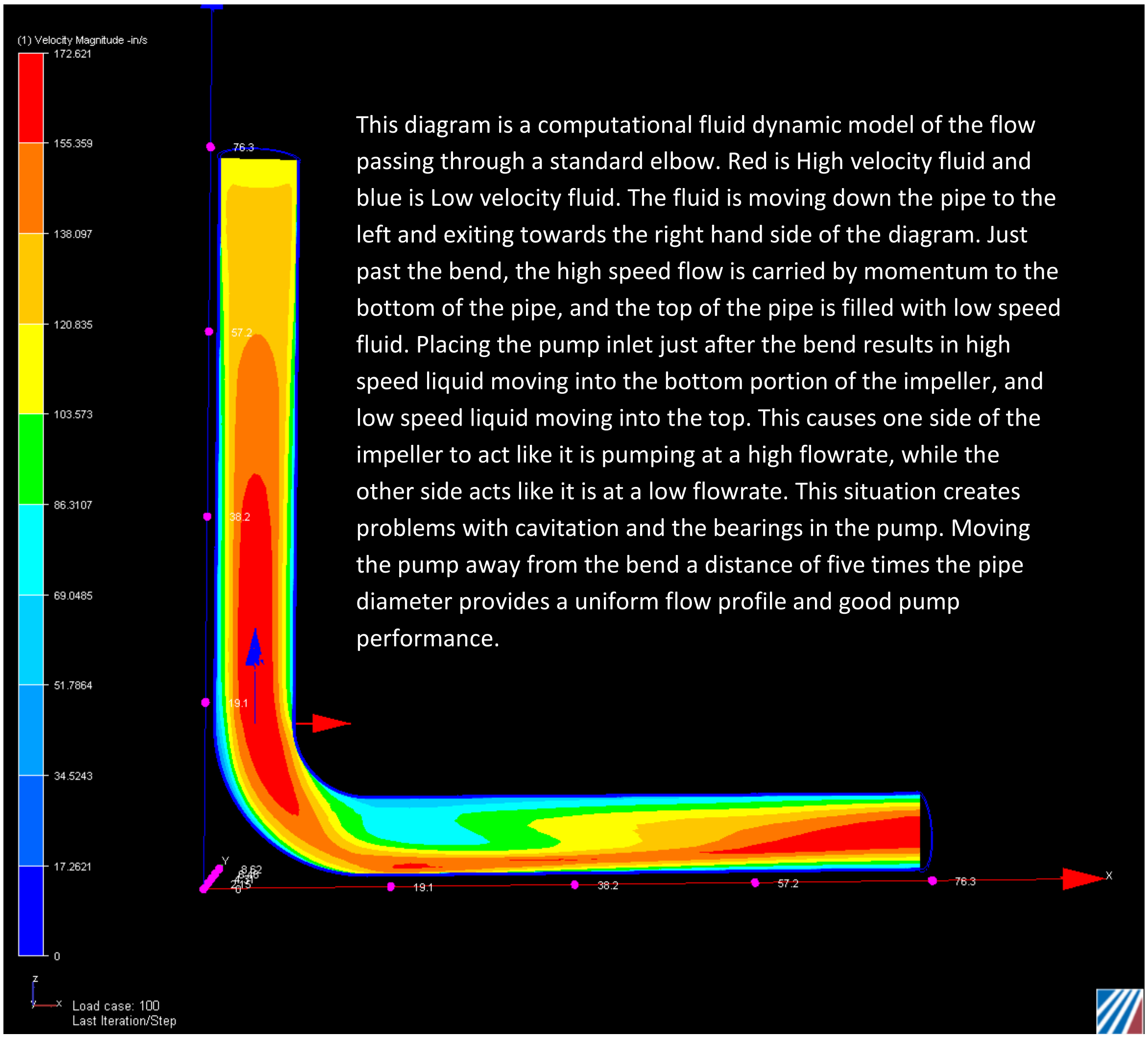


Figure 8 - Effect of inlet bend on pump performance.

IMPELLER TYPES

Selecting the proper type of impeller can have a significant bearing on the application's ultimate success. High efficiency is desired, along with the need for good reliability, and minimal ongoing maintenance. All of these considerations come into play when selecting the right impeller style.

OPEN IMPELLERS

Figure 9

Open impellers do not have a front or a rear shroud, so it allows debris that might foul the impeller to be dragged along and rubbed against the front and rear stationary wear plates, thus grinding down the particulate to a small enough size to pass through the impeller. This works well with soft particulates, but generally causes too much abrasion on both the impeller and the wear plates if the compound of the particulate is harder than that of the impeller.

Another disadvantage of this open style is the need for the impeller vanes to be fairly thick. They must have the mechanical strength to support themselves under the stress of pumping the liquid. This added thickness results in a decrease in the flow area. Additionally, leakage in the impeller is caused by the clearances at the front and rear of the blade (where the hub and shrouds would be on a closed impeller). This leakage is very dependent on the clearances between the impeller and the wear plates. As the pump wears over time, these clearances become larger, further increasing the leakage losses, degrading the pumps efficiency and, in many cases, flow and head levels. An advantage of open impellers is that they develop almost no axial hydraulic thrust loads due to the lack of shrouds. Without cores they are also easy to manufacture – making them less expensive.

CLOSED IMPELLERS

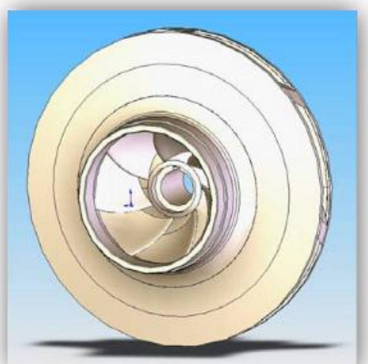


Figure 10

Closed impellers (also called enclosed impellers) have shroud and hub surfaces attached. The surfaces have several advantages. They eliminate the leakage losses across the vanes. They provide strength and stability allowing the vane thickness to be reduced, which increases the flow area through the impeller. The two shrouds also provide an axial thrust surface from which the pressure differential can be balanced. The obvious disadvantage of closed impellers is that any debris entering the vanes that is too large to pass through the impeller becomes stuck and must be removed by hand. This cleaning process, often referred to as de-ragging in the wastewater industry, requires time consuming and costly disassembly of the pump.

SEMI-OPEN IMPELLERS

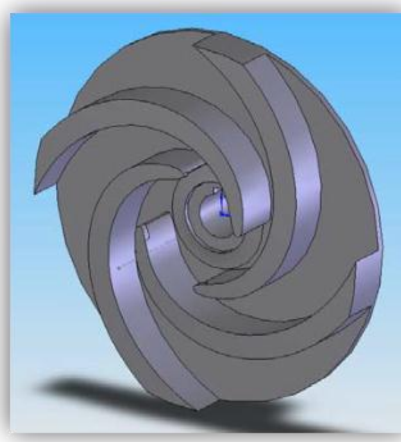


Figure 11

Semi-open impellers have only one shroud on either the front or the back. They have some of the advantages of each of the other styles, and their own set of drawbacks. Since fluid has only one leakage path over the blade, leakage losses are reduced making them more efficient than fully open impeller designs. Having one face of the impeller open allows particulate to pass that would clog many closed impellers. Their major disadvantage is the fact that they have only one shroud that fluid pressure builds upon. The differential pressure across the impeller can cause extreme axial thrust loads putting excessive stress on the bearings, or requiring thrust balancing techniques that increase leakage losses, or power consumption lowering the pumps overall efficiency

VORTEX IMPELLERS

Vortex impellers are a subset of the semi-open impeller style. Instead of trying to minimize the impeller front clearance, the impeller is intentionally mounted towards the rear of the casing cavity allowing a large gap between the rotating vanes and the front of the stationary casing. Often this gap is the size of the discharge port. The concept is that the spinning impeller creates a forced vortex out in front of the impeller. A low pressure core forms at the inlet, and speed and pressure increase outward radially until liquid is thrown out the discharge. The advantage is that particulate can pass through the pump without having to physically pass through the impeller allowing the pump to pass problem liquids with a larger amount of particulate, or fibrous material, without binding or clogging.

NON-CLOG IMPELLERS

Often in the sewage or wastewater industry, impellers are designed with a minimal number of vanes to allow particulate to pass without fouling the impeller. Some designs have only one vane that wraps around the impeller. More commonly, two and three vane designs are used to improve performance while still allowing the passage of solids. A common criteria for this style is that it can pass a 3-inch diameter solid without fouling the pump. To meet this standard, a three inch marble must be able to be rolled through the impeller passage without becoming stuck. Many of the new, higher efficiency multi-vane impeller designs can no longer pass a 3-inch solid. For applications where this is a requirement, the manufacturer should be contacted to assure that the pump can actually pass a 3-ince spherical solid. Generally, non-clog impellers are of the closed design allowing them to maintain clearances in abrasive environments without the need of continuously replacing wear plates.

MOTORS

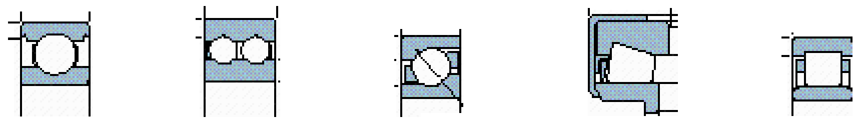
When selecting an electric motor, it is important to select one that can develop enough horsepower to drive the pump. Motors are often selected to be non-overloading at the end of curve. This means that even at the maximum power requirement for the pump, the motor is large enough to drive the pump without overloading. There are several important factors that come into play for proper selection of the motor in submersible pumps.

CABLE CONNECTIONS

If the motor is submerged in liquid, the power cable must enter the motor housing at a junction box located below the liquid level. This is a prime location for a leak. Some manufacturers believe that this should be a rigid, permanent connection with built-in strain relief. This style connection often has a packing gland around the entrance of the junction box and, occasionally, a secondary seal to prevent leakage. Other manufacturers see an advantage to having a quick disconnect on the cable allowing the pump to be replaced without the need to re-cable the unit to the control panel. If pump changes are frequent due to the need to de-rag the pump or other operational problems, it may be advantageous to have a pump with a quick disconnect since it allows the unit to be changed without the need for an electrician.

BEARINGS

There are generally two bearings of which to be concerned in submersible pump designs. The first is the upper bearing designed to support the rotor (pump impeller, shaft, and motor rotor) in the radial direction. This bearing moves axially via a slip fit to the housing. This design allows for thermal growth in the rotor as it heats during operation. The second is the lower bearing which is usually responsible for supporting the rotor in both radial and axial loading. It is fixed in place allowing it to transfer the axial loads from the pump into the motor frame.



(A) (B) (C) (D) (E)

For pure radial loading in pumps, a single row deep groove ball bearing (A) typically works well. These bearings are inexpensive and have more than enough radial load capability for common pump applications. Occasionally double row deep groove bearings (B) and roller bearings (E) are used when very high radial loading is expected, however, these are often more expensive, and are generally found on large horizontal pumps.

For the combination of radial and axial loading, the single row, deep groove bearing can be used, but the axial load capability is somewhat limited. The double row, deep groove bearing supports much higher axial loading (often 1.7 to 2 times the capability). When very high axial loads are expected, angular contact (C) and tapered roller (D) bearings can be used. These bearings have 3 to 5 times the load carrying ability, but can only do so in one direction.

Specifications often include mean time between failures (MTBF) or mean time between repairs (MTBR) at very high numbers (60K, 80K, 100K hours). This leads the manufacturers to install larger bearings to meet the very high design life requirements. Bearings running in very lightly loaded conditions often cause the rolling elements to skid instead of roll. Because more oil is not being drawn in by the rolling action of the bearing, the skidding action causes the lubricant film between the rotating element and the raceway to dissipate, eventually leading to metal-to-metal contact. This condition leads to spalling of the bearing and ultimate failure. The most common cause of bearing failure is actually improper lubrication either from contamination of the lubricant, or poor preventative maintenance practices.

OIL-FILLED VS AIR-FILLED MOTORS

Oil-filled motors offer several benefits. Due to a much higher thermal transfer capacity of oil as compared to air (approximately 7X) oil-filled motors tend to run cooler. The oil also provides continuous lubricant for the bearings and the windings. Some manufacturers claim that the vibration, or start up torque pulses, of the windings causes the insulation to wear subsequently leading to shorts within the motors. Oil-filled motors are designed to lubricate the windings and prevent degradation from chaffing during start-up. There are also studies in support of the claim that oil-filled motors prevent moisture from getting into the hydroscopic insulation on the windings, a benefit since the insulation tends to breakdown more quickly in moist environments.

Air-filled motors have a lower amount of drag loss as compared to an oil-filled motor. Typical estimates range from 1% to 2% less loss. Air-filled motors work best in applications where the liquids are always cool and provide plenty of heat dissipation. If heat dissipation might be an issue, oil-filled motors have the advantage over air-filled.

In the case of submersible pumps in dry pit applications, heat dissipation is a major concern. Many manufacturers will require that the motor stator be jacketed to help remove the motor heat. The jacket circulates liquid over the outside of the motor helping to dissipate the heat. These jackets typically come in two forms; product-cooled and self-contained. A product-cooled jacket is a traditional design that cools the motor by passing some of, or the entire product being pumped, through the jacket. As the pumped product circulates over the motor, the heat is transferred to the liquid, which is then discharged through a port in the jacket. The product-cooled design is not recommended in applications where large particulate may cause plugging of the jacket ports and lead to a motor failure. The self-contained option uses a separate cooling loop to pass clean liquid over the motor and generally does not suffer from the plugging problem.

MECHANICAL SEALS

The job of the mechanical seal is to prevent the liquid being pumped from leaking into the bearing and motor housing. This is accomplished by rotating one extremely flat seal face in very close proximity to a stationary face of approximately equal flatness. The faces are lapped to within 2-4 helium light bands of being perfectly flat. In the space shown as the fluid wedge in the diagram below, a small amount of liquid is wicked through the faces and is vaporized by the heat generated by their rotation. There must be liquid in the seal faces to cool and lubricate them. Seal faces fail quickly when they run dry – sometimes within fractions of a second. In a typical single seal pump, pumped liquid passes through the faces and vaporizes as it enters that atmosphere. Some small submersible pumps use a single seal to protect the motor from the pumped liquid.

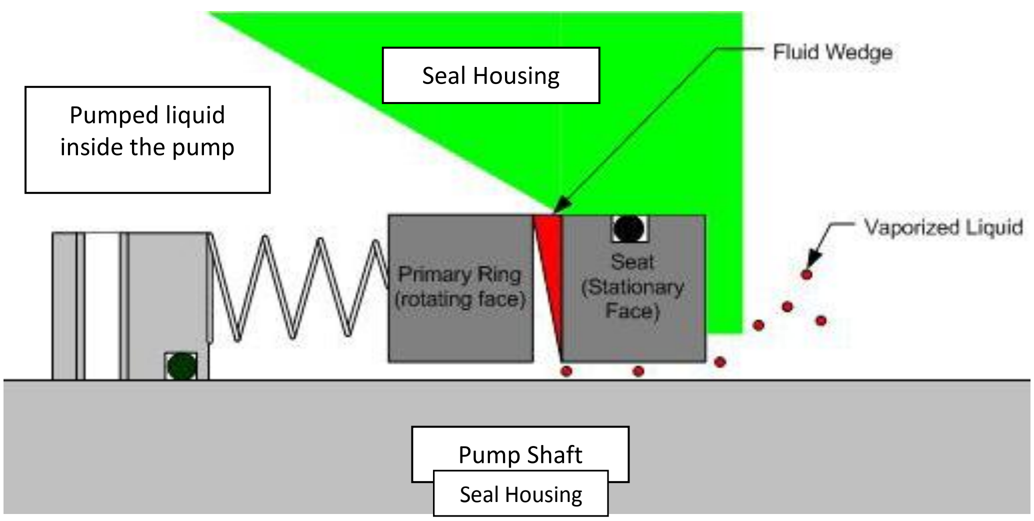


Figure 12 - Typical single seal in a process pump

The typical seal arrangement is a dual seal in either a double or tandem arrangement in most submersible pumps.

The dual seal offers the protection of two seals to prevent the pumped product from getting into the bearings and motor. The tandem arrangement has both seals facing the same direction. The bottom seal is in the pumped liquid (A). As the pressure increases in the pumped liquid, it forces the seal faces closer together thus reducing the amount of liquid that passes through the faces. The oil in the seal chamber (B) is intended to be the lubricant for this lower seal. The upper seal is fed with oil in the motor or bearing housing.

With a double seal, the two seals are positioned back to back. This has the advantage that both seals are operating in the clean oil environment. The disadvantage is that a pressure spike in the liquid on the pump side of the seal can cause the seal faces to force open and product can be introduced into the seal cavity. This reduces or destroys the lubricity of the oil in the seal chamber, eventually leading to bearing failure.

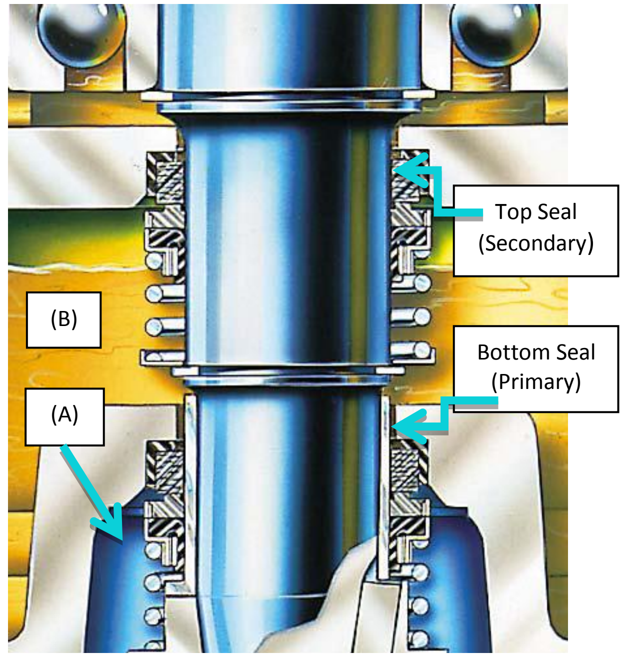


Figure 13 - Typical tandem dual seal in a submersible pump, diagram courtesy of Hydromatic pump

SEAL FACE MATERIALS

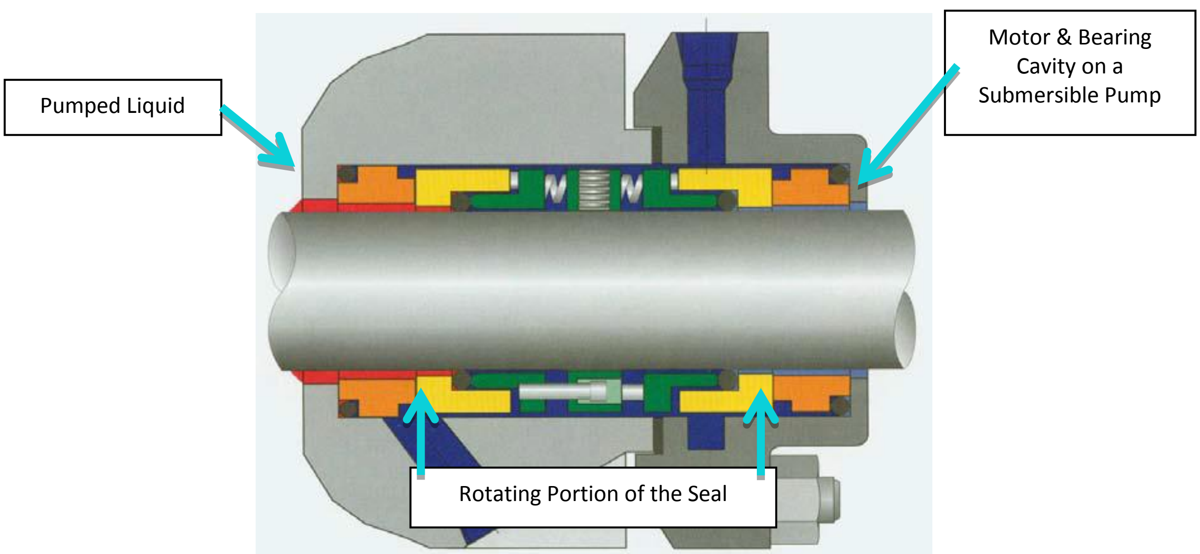


Figure 14 - Typical double seal arrangement from a process, pump, diagram courtesy of Flowserve Corporation

Seal faces can be made from a variety of materials. The most common materials are carbon (F), ceramic (C), tungsten carbide (WC), and silicon carbide (SiC). Carbon is a good seal material because it is somewhat self-lubricating and is fairly inexpensive. Its nemesis is abrasives. Carbon is such a soft material that it is easily scratched. Scratches can provide leak paths leading to seal failure. Ceramic is often paired with carbon. It is harder than carbon but, generally, not harder than common abrasives. It too is easily scratched leading to failures in abrasive environments. Ceramic does not have the mechanical strength of Tungsten and Silicon Carbide (it flexes under pressure). It is susceptible to thermal shock (quick temperature change) causing it to shatter. Its primary attribute is that it is inexpensive, and therefore very popular. Tungsten Carbide is an extremely hard material that has very good mechanical properties coupled with excellent corrosion resistance. The material does extremely well in abrasive conditions. Silicon Carbide is slightly harder than Tungsten Carbide. It has excellent corrosion resistance and very good mechanical properties. Silicon Carbide is often paired with Tungsten Carbide as a combination of faces in abrasive situations.

In most wastewater applications, it is desirable to have the primary seal made from harder materials such as SiC vs SiC, or SiC vs WC. The upper seal should only be pumping oil, so a carbon ceramic seal would be a good choice. Some companies offer both seals with hard faces, but unless there is a particular reason, the use of F vs C can reduce the amount of heat generated by the seal and decrease the cost of the pump.

MOISTURE SENSORS

So what if the mechanical seals fail? How would you know? On typical process equipment, the puddle under the pump is your first warning. With submersible equipment, there is no such visual indicator. Most manufacturers of submersible equipment have at least one moisture detection device in the pump. These devices are typically time-tested technologies that indicate when there is moisture in a portion of the pump where there should be none. The location of these sensors is important. Some manufacturers mount the sensor in the cavity between the primary and the secondary seal. If the primary seal leaks, the secondary seal prevents the liquid from getting to the bearings and motor where it can cause damage. Since the pump does not need to be shut down immediately, most Manufacturers using this method additionally state that the pump repair can then be scheduled for a convenient time. Other manufactures locate the sensor in the motor cavity. This location can be problematic because liquid has already reached the bearings and motor when the sensor indicates there is a problem, thus requiring an immediate shutdown of the unit to prevent further damage.

PUMP CURVES

Reading and understanding centrifugal pump curves is the key to proper pump selection. There are four important curves shown on the standard performance curve from the manufacturer. They are listed below and shown on the manufacturers curve (Figure 15).

1. Head
2. Efficiency
3. Power
4. Net positive suction head required (NPSHR)

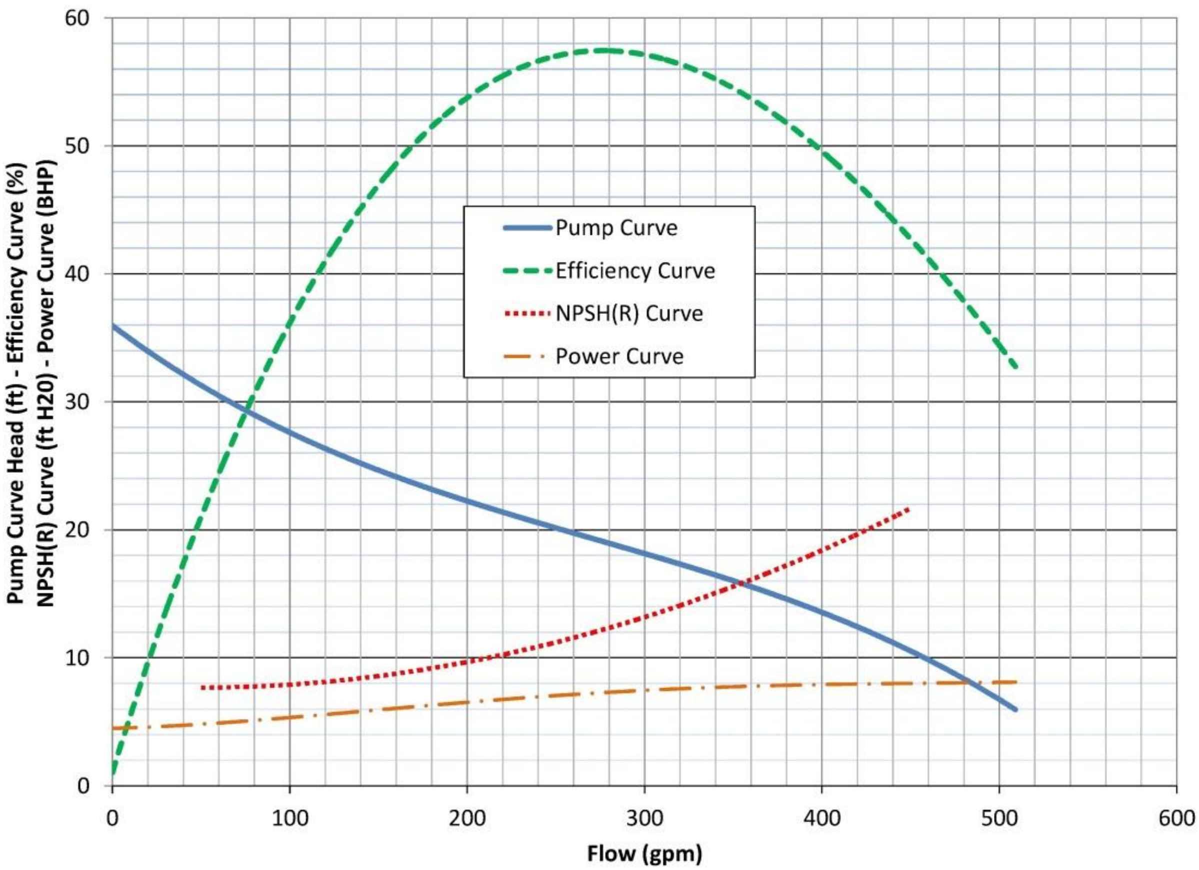


Figure 15 - Single line pump curve

Various manufacturers and industries display the information on their curves with slight variations. One very common variation is shown below (Figure 16). Instead of an efficiency curve, the efficiency is broken into several iso-efficiency lines with each line representing a constant efficiency. It is read much like a topographic map, with the iso-efficiency lines corresponding to elevation lines on the map.

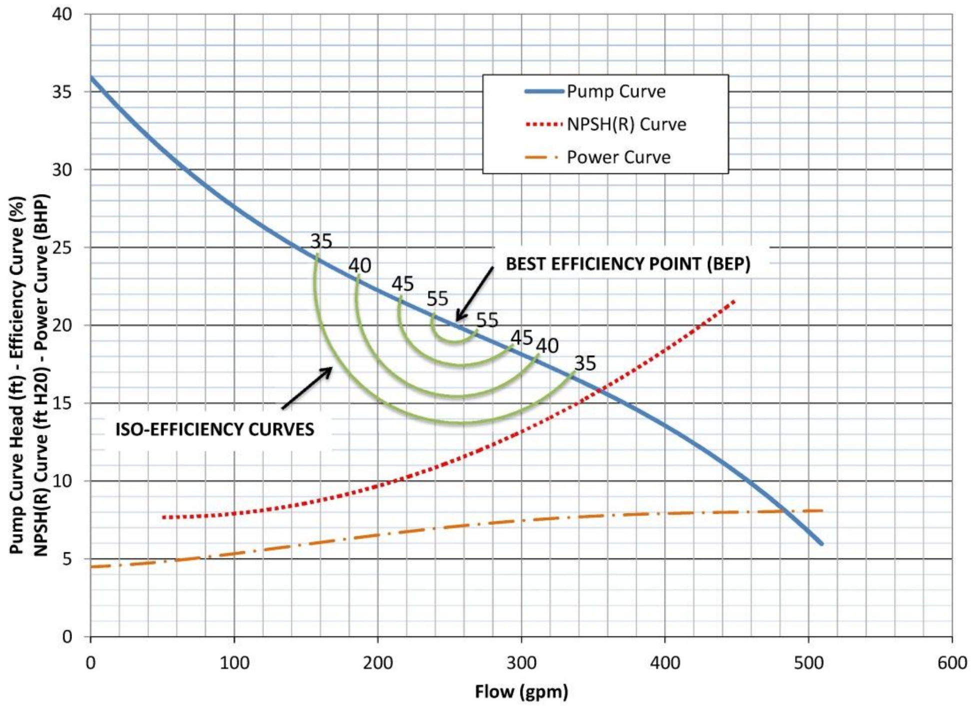


Figure 16 - Single line curve w/ Iso-Efficiency curves

Another common curve variation is to show multiple impeller diameters on the same curve (Figure 17). As the impeller diameter decreases in size, the performance is reduced. This allows the pump performance to be modified to meet specific application requirements. Additionally, a reduction in diameter reduces the pump power requirement. A reduction of only 10% in the impeller diameter can result in a 27% reduction in power requirements.

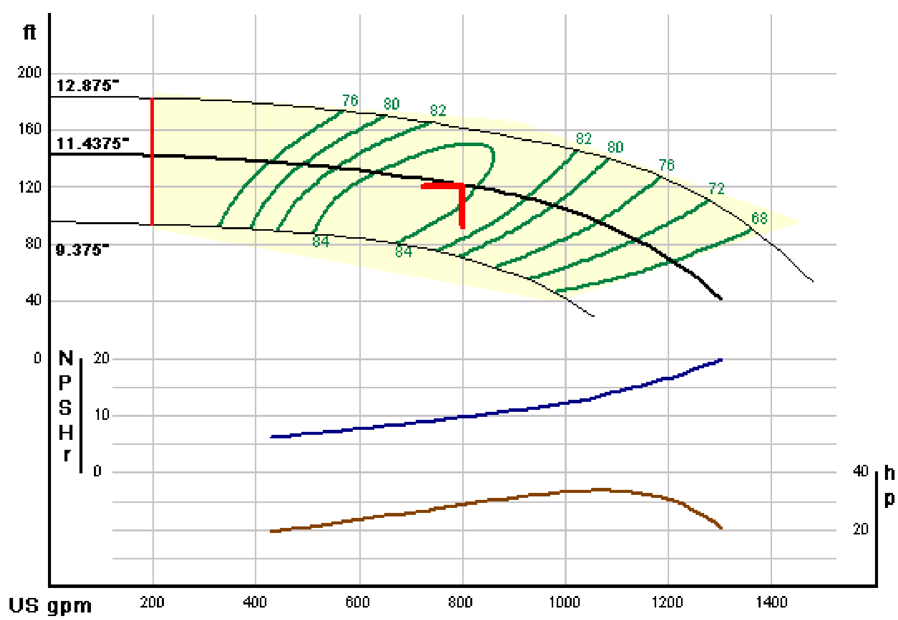


Figure 17 - Multi-trim curve

STEEPNESS OF PUMP CURVE

Pump head curves with very flat head flow characteristics can make the pump difficult to control. Small changes in system resistance can create large changes in flowrate. The pump difficulty arises in situations that have a flat system loss curve. The problem is exacerbated when variable-frequency drives are used to control the pump operating point.

INTERACTION OF THE SYSTEM CURVE WITH THE PUMP CURVE

Pumps operate where the pump curve meets the system curve. Ideally, pumps should be sized to run as closely as possible to its best efficiency flowrate. This not only makes the pump more efficient, but also improves its reliability. Correct sizing requires that both pump curves be fairly accurate. Minor variances of the manufacturer's tolerances may affect the pumps performance, but all curves have a tolerance of approximately ±3%. System curves have a much wider range of inaccuracy due to variations in pipe and fitting friction losses between various manufacturers. The note at the bottom of the *Cameron Hydraulic Data* book of pipe tables advises that a 15% to 20% increase in loss should be used above the loss levels shown in their tables. This inaccuracy, and the rising cost of power, make it imperative that larger pumps be field tested to determine actual flowrate. Excessive flow causes excessive friction thus driving up power consumption and operating costs. Field testing allows the system calculations to be confirmed, and the pump to be modified, to meet the actual system conditions.

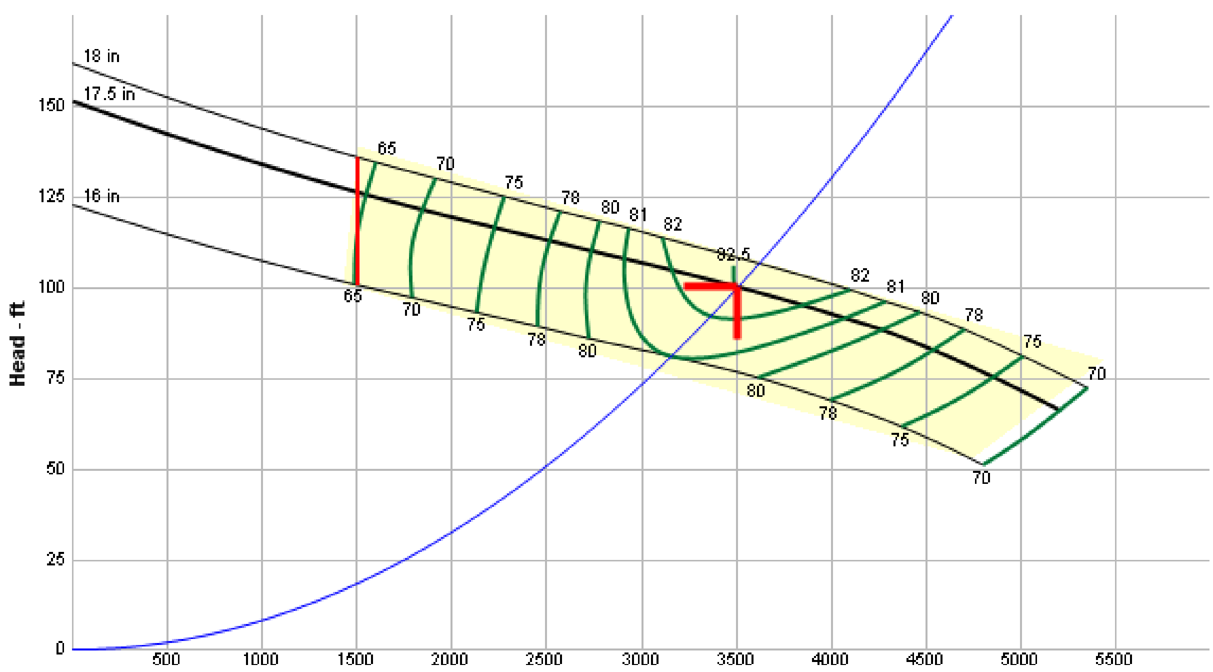


Figure 18 - Pump curve and system curve

NET POSITVE SUCTION HEAD

While Net Positive Suction Head (NPHA) analysis is not a concern with submersible pump design, when designing a dry pit, a NPSH analysis is critical. The following discussion demonstrates why NPHA analysis is not necessary in submersible pump design.

There are two forms of NPSH. Net Positive Suction Head Required (NPSHR) is provided by the manufacturer, and net positive suction head available (NPSHA) is the amount of energy available at the inlet of the pump in relation to the system layout. NPSHA is calculated using the formula below:

$$NPSHA = h_{atm} \pm Z_s - h_{vp} - h_f$$

Equation 7

h_{atm} = Atmospheric pressure at the surface of the liquid (ft)
Z_s = Suction Static Head (ft)
h_{vp} = The liquids vapor pressure at the pumped temperature (ft)
h_f = The friction losses in the pipe and fittings from the suction tank to the pump inlet (ft)

NPSHR is provided on the manufacturers curve. The most important thing to know about NPSH is that the NPSHA must be greater than the NPSHR. Typically, a factor of safety of 1.3 is used. Thus:

$$NPSH\ Margin = \frac{NPSHA}{NPSHR} \geq 1.3$$

Equation 8

The purpose of a net positive suction head analysis is to ensure that the impeller of the pump is submerged with liquid. For example, in a dry pit design the water is stored in a wet well, and the pump is stored in a separate structure and is not submerged. If the layout was such that, at some point, the water level in the wet well dropped low enough that it was not being forced into the pump impeller, the pump would begin to cavitate.

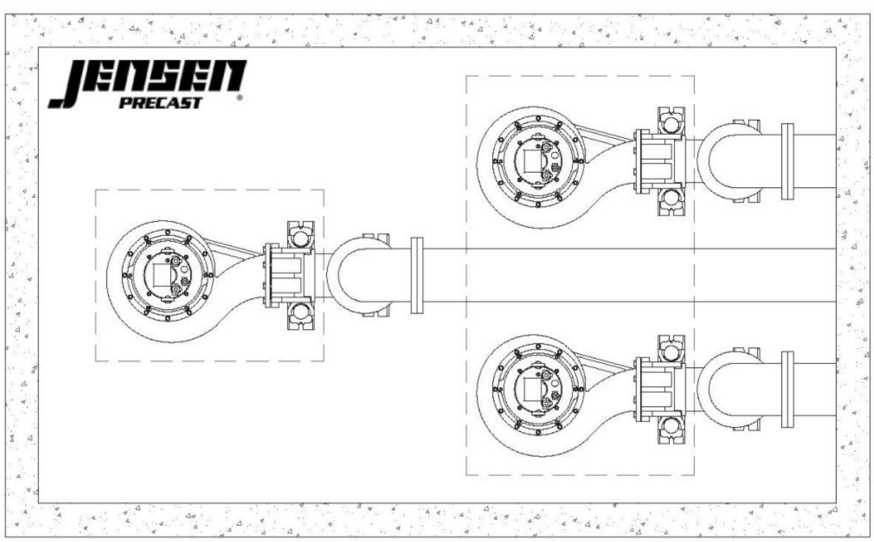
In a submersible pump station with proper design of the control elevations, the pump is always submerged and forcing the fluid into the impeller thus eliminating this concern.

INTRODUCTION TO WET WELL DESIGN

Once the system curve has been created and a pump performance has been properly matched to that system curve (refer to Pumps section of this manual to determine pump performance), the wet well can be sized.

Most Jensen Engineered Systems packaged lift stations use round manholes for the wet well. There are many reasons for this including reduced material costs, a smaller footprint, and a round structure's strength properties.

However, in some instances a rectangular vault is recommended to use for a wet well. An example would be a high flow system using large pumps that would not fit inside a 10' diameter manhole. Other than the surface area calculation, the design process for a rectangular well is the same as a round well. For the rest of this discussion, assume a round wet well.



MINIMUM STORAGE VOLUME

The following discussion concerning wet well sizing is based upon the assumption of a "low" flow system of less than 3000 gpm. Systems with flows larger than 3000 gpm require additional considerations.

The first design criteria to identify are the total system inflow rate and the flow at which the pumps will discharge. As discussed in the pump design section, the discharge flow can be found at the intersection of the system curve and the pump performance curve. These flows will be identified as:

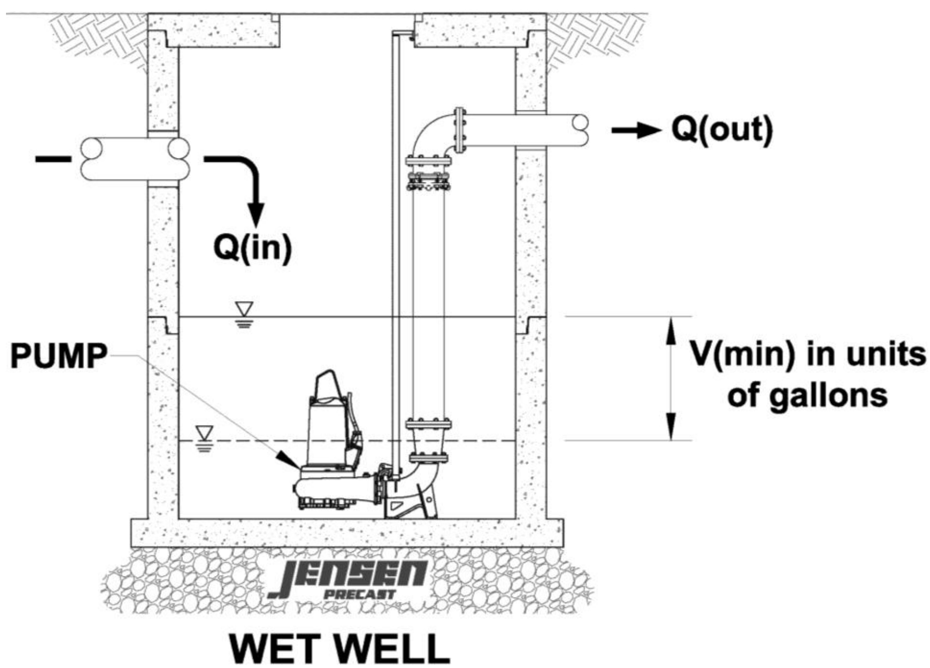
Q_{IN} = Inflow rate into wet well (gpm)

Q_{OUT} = Discharge flow rate out of wet well (gpm)

The intent is to determine the minimum storage volume the wet well needs to hold between pump starts. Typically the recommended minimum time between pump starts should be eight to ten minutes, or roughly six starts per hour. However, this can vary from manufacturer to manufacturer so check with the particular pump maker. Also, verify the minimum run time of the pumps with the manufacturer. These values will be represented as follows:

T_{MIN} = Minimum cycle time between pump starts (minutes)

V_{MIN} = Minimum storage volume of wet well to hold/gather fluid during pump off (gallons)



V(min) can be determined by starting with the following equation which relates the inflow, storage volume, and outflow to T(min):

$$T_{MIN} = \frac{V_{MIN}}{Q_{IN}} + \frac{V_{MIN}}{Q_{OUT} - Q_{IN}}$$

Hydraulic Institute Intake Design – 1998 Equation B.1

Assuming the flows entering the wet well have been properly estimated, and an appropriate pump has been selected for the demand, the worst case scenario is that the inflow is twice the rate as the outflow. Or:

$$Q_{IN} = Q_{OUT} / 2$$

Equation V.1

The result of plugging this into the Q_{IN} component of equation 1 and rearranging for V_{MIN} is:

$$T_{MIN} = \frac{V_{MIN}}{Q_{IN}} + \frac{V_{MIN}}{Q_{OUT} - Q_{IN}} = \frac{4 * V_{MIN}}{Q_{OUT}}$$

Equation V.2

Therefore:

$$V_{MIN} = \frac{T_{MIN} * Q_{OUT}}{4}$$

Equation V.3

SIZE OF WELL

So, the minimum volume of the wet well needs to handle all inflow rates has been determined. Now the wet well shape and size needs to be selected. This is an iterative process. A good place to start is by sizing the minimum hatch dimensions, as this sets the minimum well diameter.

HATCH SIZING

First, determine the horizontal dimensions of the pumps by referring to the technical information sheets for the selected pump. Some important dimensions are shown in the example on the following page.

JENSEN ENGINEERED SYSTEMS

HOMA PUMPEN MIT SYSTEM

Dimensions

AK636-320/26F/C

Wet well installation with auto coupling kit (F-motor, 320...340)
Dimensions in inch, letters see table

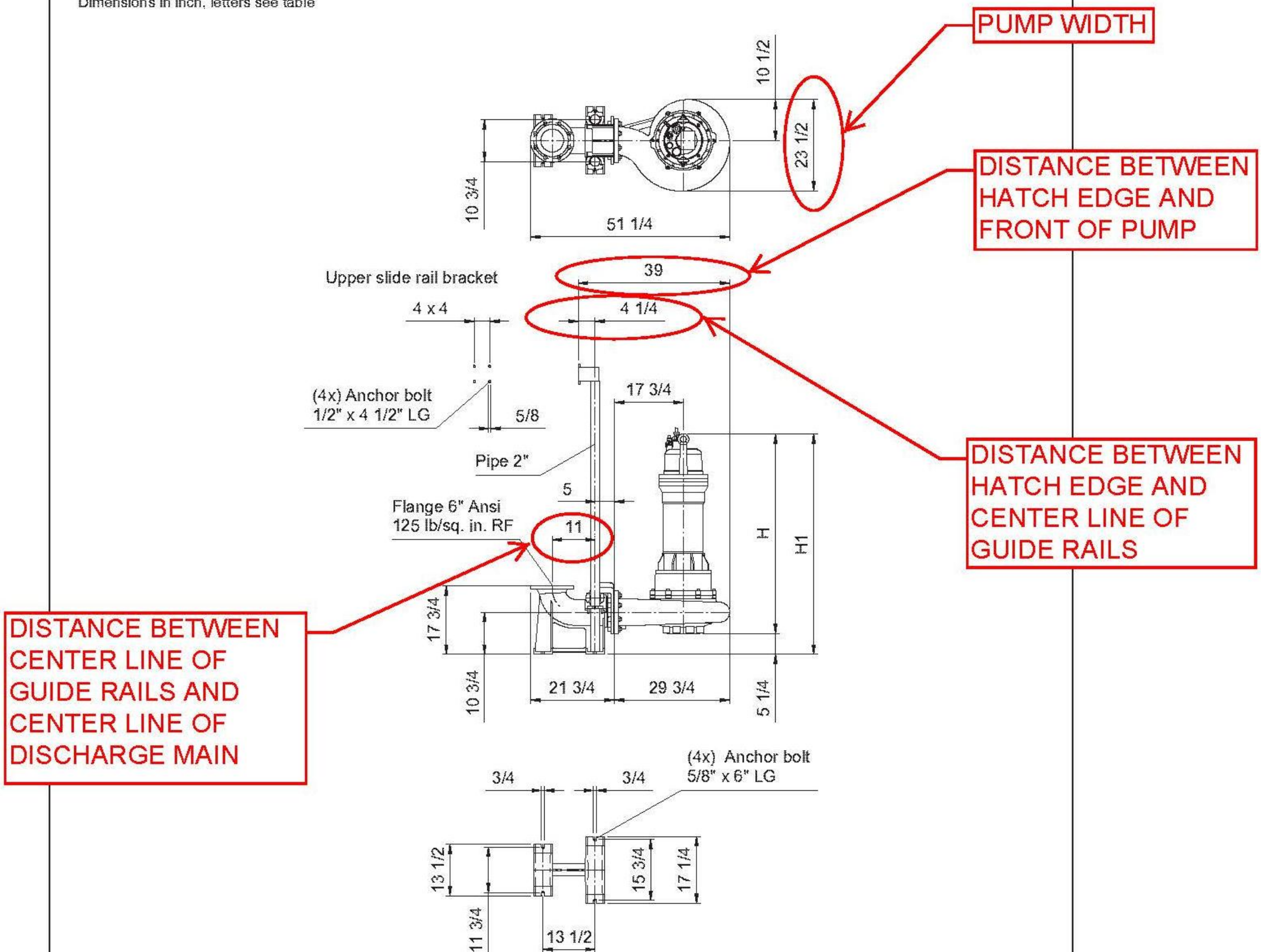


Table Dimensions **(inch)**

H	$46^1/_2$
H1	52

2.0 - 29.01.2009 (Build 168)

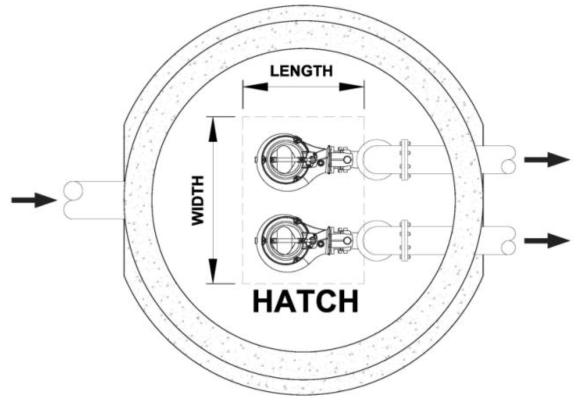
Project	Project no.:	Created by:	Page: 3	Date: 2011-02-10

Example of Homa Technical Dimensions Sheet

The minimum "length" of the hatch is the distance between the point where the guide rail bracket mounts to the hatch edge and the front of the pump. This is typically shown on the pump technical sheet.

To determine the minimum "width" of the hatch, use the following equation:

Hatch Width = (Number Of Pumps * Pump Width) + [(Number Of Pumps - 1) * Minimum Pump Spacing]



Each pump manufacturer typically has a recommended spacing between each pump. Consult the selected pump manufacturer to determine these dimensions. The purpose of this spacing is to prevent the pumps from competing with each other in cases where they are running at the same time. JES recommends that the minimum spacing between the outer edge of the pump to the wet well wall correlates to the pump spacing dimension. Below is an example of Homa Pumps' recommended spacing from outer edge of pump to outer edge:

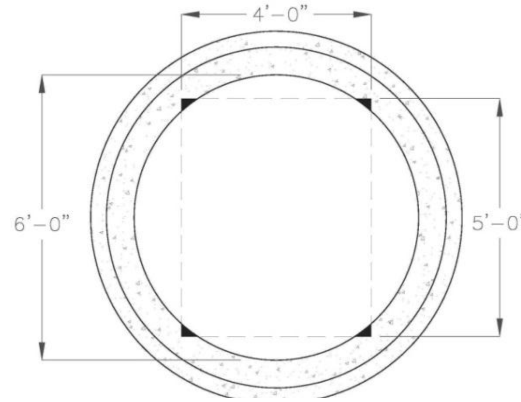
	Discharge Size			
Homa Model #	3”	4”	6”	8” & Larger
AK	Min N/A	Min 10”	Min13”	Please consult factory for layout information
AV	Min N/A	Min 10”	Min N/A	
AMX	Min 8”	Min 10”	Min 13”	

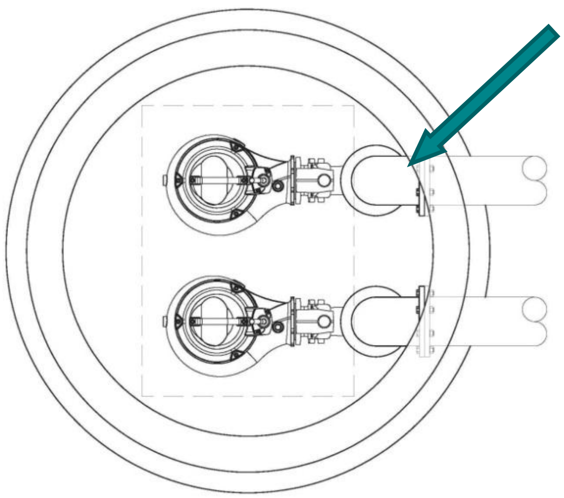
When specifying the hatch for the system, remember to address the following considerations:

1. Loading Criteria
    a. Pedestrian
    b. H-20 traffic loading
    c. Airport conditions
2. Material
    a. Steel (galvanized, painted, or powder coated)
    b. Aluminum (anodized optional)
3. Non-skid surfaces such as TraxPlate™, a product of Jensen MetalTech
4. Safety grates

DIAMETER OF WELL

As an example, assume a minimum hatch 4' long by 5' wide has been sized. It might seem logical that a 6' inside diameter manhole would work. This would be incorrect as it does not account for the circular shape of the manhole. For this reason it is important to always model the hatch and manhole together to ensure compatibility.



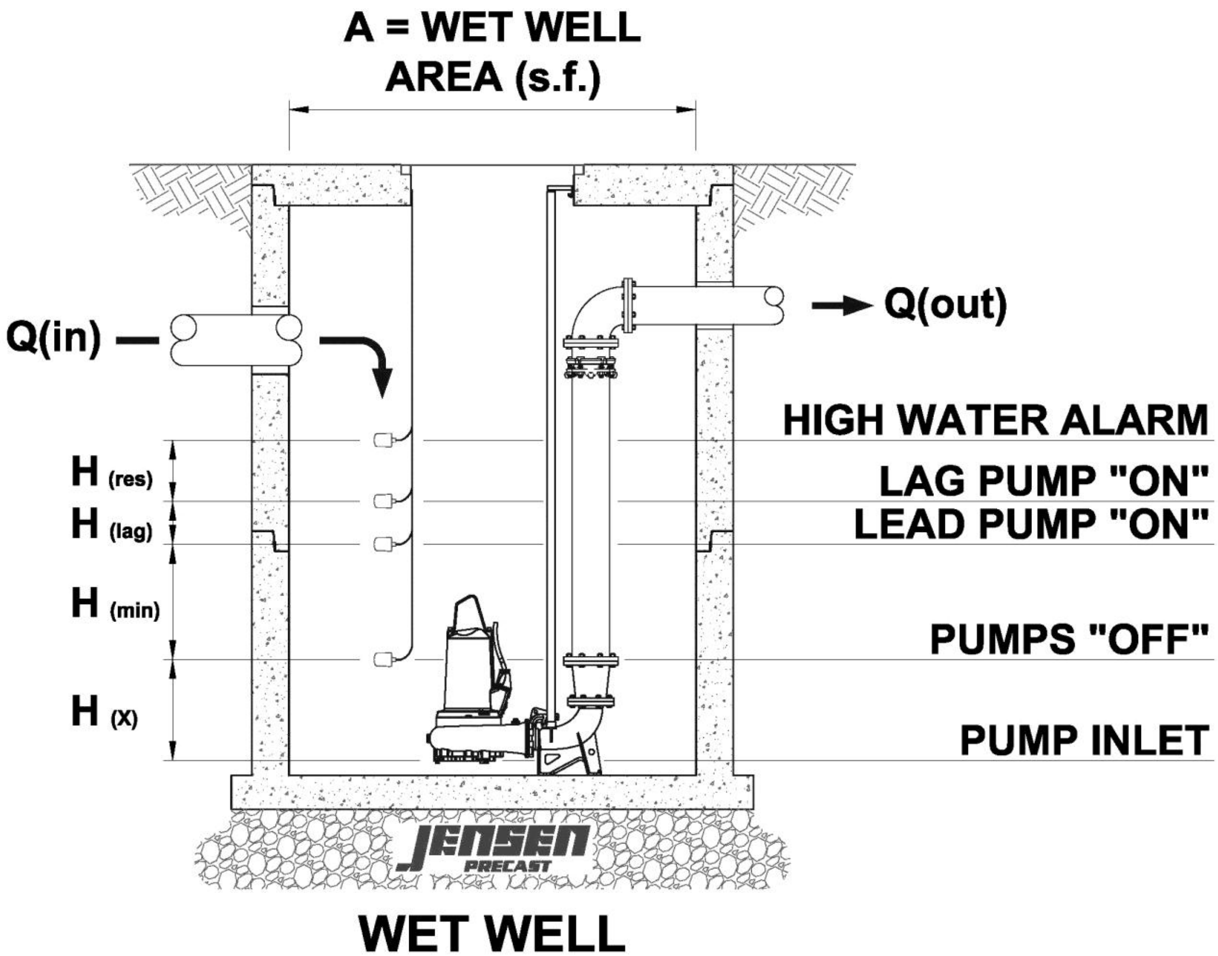


As a general rule, it is better to error with a larger manhole. One reason is the discharge piping. Many times the hatch is sized within the manhole correctly, but the discharge piping is forgotten. In the following example the hatch is correctly sized and the pumps are correctly spaced. However, with the existing flange of the 90 bend, the flange would not fit within the well so a larger manhole should be used. Also, keep in mind installation concerns such as making sure the contractor has enough working room to bolt the flanges on the 90 bends.

Some final considerations when sizing the wet well diameter are site location limitations such as:

- Are there restraints as to the depth the wet well can be, such as soil type or high groundwater?
- Site constraints that would force the well to have a small footprint?

CONTROL ELEVATIONS



Now that the minimum storage volume is known, as well as the size of the wet well, the system's control elevations can be determined. In a duplex pumping system there are 5 primary elevations of concern – the pump inlet, pumps "off", lead pump "on", lag pump "on", and high water alarm. The distances between these elevations are represented with the following variables:

H_X = Pump Inlet to the Pumps "Off" Elevation

H_{MIN} = Pumps "Off" to the Lead Pump "On" Elevation

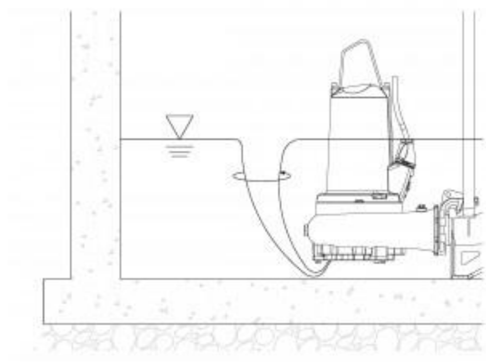
H_{LEAD} = Lead Pump "On" to the Lag Pump "On" Elevation

H_{RES} = Lag Pump "On" to the High Water Alarm Elevation

H_X – MINIMUM SUBMERGENCE

The purpose of minimum submergence is to prevent air from entering the pump. Lack of minimum submergence will cause what is known as a " pre-swirl" which can lead to a vortex.

To prevent a vortex, the following equation has been developed which is the minimum distance between the pump inlet and the water surface elevation. Sometimes, H(x) is referred to as S. Either way, it is the minimum submergence.



Example of a Vortex

$$H_X = D(1 + 2.3F_D)$$

Hecker, G.E., Ch 8, conclusions, "Swirling Flow Problems at Intakes," IAHR Hydraulic Structures Design Manual 1, 1987

Where F_D is the Hydraulic Froude Number (unit less) and is determined by:

$$F_D = \frac{V}{(gD^{0.5})}$$

Hydraulic Institute Intake Design – 1998 Equation (9.8.2.1-2)

Where:

D = Inlet Diameter (ft)
g = gravity (32.2 ft/s^2)
V = Velocity (ft/s) of fluid at the inlet and is determined by:

V = Q / A

Where:

Q = Pump Discharge Flow (cfs)
A = Area of inlet (ft^2)

H_{MIN} – MINIMUM STORAGE

H(min) is the distance between the pumps “off” and lead pump "on" elevations. It is determined by the following equation.

$$H_{MIN} = V_{MIN} / A$$

Where:

V_{MIN} = Minimum storage volume (gallons)

A = Cross sectional area of wet well (sq.ft.)

H_{LAG} – LAG STORAGE

Duplex submersible pump stations should be sized so that one pump will be able to handle peak flow events. During events where the inflow exceeds the predicted max flow, the second pump can be used to handle the additional flows. This is where H(lag) comes into play. It is an arbitrary factor of safety set by the engineer. Typically, in smaller flow stations of less than 200 gpm, an H(lag) of at least six inches is recommended. The larger the engineer makes H(lag), the more conservative the system, but material and construction costs will increase.

H_{RES} – RESERVOIR STORAGE

As with H(lag), H(res) is a factor of safety built into the submersible pump station. In the event the actual inflow far exceeds the max predicted inflow, or a pump fails, an alarm is triggered. This alarm signals station operators that there is a problem. For smaller flow stations (less than 200 gpm) an H(lag) of at least twelve inches is recommended. However, this should be a decision made by the engineer on a system by system basis.

We hope these guidelines have been helpful. For additional information on Pump Station and Wet Well Design, contact Jensen Engineered Systems at 855-468-5600, or visit www.JensenEngineeredSystems.com.