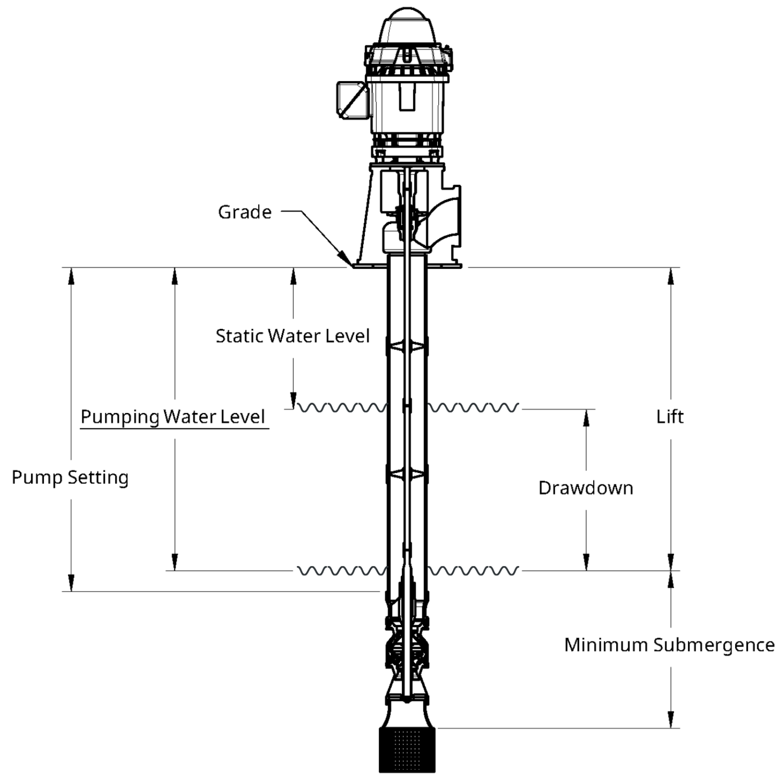
Terminology

1. **Grade** - The elevation of the surface supporting the pump.
2. **Static Water Level** - The vertical distance between grade and the water level in the well when the pump is OFF.
3. **Drawdown** - The vertical distance the water is lowered in the well during pumping.
4. **Pumping Water Level** - The vertical distance between grade and the water level in the well when the pump is ON.
5. **Pump Setting** - The vertical distance between grade and the top of the pump assembly
6. **Lift** - The vertical distance from the pumping water level to the discharge level (this may be higher than grade).
7. **Minimum Submergence** - The lowest acceptable water level in the well for pump operation.
8. **Column Friction Loss** - Losses incurred by the flow of water through the pump column assembly. See Column Friction Loss Table for approximate values.



9. **TDH (Total Dynamic Head)** - The total of the following: vertical elevation from the pumping water level to the discharge point plus all losses in the column and discharge piping.
10. **Lab Efficiency** - Efficiency of the bowl assembly only. This value can be found on the pump performance curve.
11. **Lab Horsepower** - The horsepower required by the bowl assembly as measured during laboratory testing.

$$Lab\ HP = \frac{Lab\ TDH\ x\ Capacity}{3960\ x\ Lab\ Efficiency}$$

12. **Shaft Friction Loss** - The horsepower required to overcome the friction in the lineshaft bearings.
13. **Field Horsepower or Brake Horsepower** - The sum of Lab Horsepower plus Shaft Friction Loss plus any losses in the driver thrust bearing.
14. **Pump Field Efficiency** - The efficiency of the entire pump, less the driver.

$$Pump\ Field\ Efficiency = \frac{Field\ TDH\ x\ Capacity}{3960\ x\ Brake\ Horsepower}$$

15. **Overall Efficiency (Wire to Water)** - The efficiency of the pump and motor. Equal to Pump Field Efficiency x Motor Efficiency.
16. **Total Pump Thrust** - The sum of the shaft weight and the hydraulic thrust created by the impellers moving liquid.

Table of Contents

Sump Design

Note: The following sump design guidelines are recommendations and not to be considered exact. There are many different design aspects and considerations that are not included in this guide.

General Design Considerations

- The goal of proper sump design is to achieve an *evenly distributed flow* to the pump intake.
- Uneven flow can be characterized by the formation of visible and invisible vortices along with uneven velocity in the sump.
  - Vortices can cause premature wear on the pump and motor by constantly increasing or decreasing power consumption. This is a result in fluctuating TDH caused by the uneven velocity of the vortex.
  - Vortices may be invisible to the naked eye, or strong enough to reach the surface. If a vortex reaches the surface, it is possible that it will draw air down into the pump, causing cavitation, reduced performance, and premature wear.
  - Uneven velocity may occur in localized regions of a sump, even if the general sump velocity is low. For this reason, low sump velocity is not necessarily an indication of good sump design. In fact, higher velocities tend to discourage uneven localized velocities.
- The best intake is a direct channel that carries water directly to the pump suction. Any additional geometry or flow obstructions may create eddy currents and form vortices.
  - Water should never flow past one pump's suction to reach another pump's suction

Sump Dimensions

Use Table 1 on the next page to find the dimensions for the following figures.

Note:

- Dimension C is an average value and the manufacturer should be consulted for the final dimension
- Dimension B is a suggested maximum dimension. If the back wall distance cannot be achieved, it may be necessary to install a "false wall" behind the pump
- Dimension S can be increased but should not be reduced without a manufacturer consultation
- Dimension H is the normal low water level. The pump should only momentarily be operated when the sump level falls below this point.
  - Minimum submergence is typically equal to Dimension H minus Dimension C
- Dimension Y and Dimension A are recommended minimum values. These can be as large as desired up to the limit shown in Figure 3. If there is no screen, dimension A should be substantially longer. The width of the screen and rack should not be much less than Dimension S and their heights should not be less than Dimension H.
- If the velocity in the channel is greater than 2ft/s, it is recommended to do one or both of the following: install straightening vanes or increase dimension A. It may be necessary to conduct a test of the sump in order to determine what is required.
- Figure 2 shows the ideal installation with straight line flow directly to each pump. Optional separating walls can be installed to optimize efficiency if multiple pumps will run at the same time. Velocity in the channel should be between 1 and 2 ft/s.

Figure 1: Sump Dimensions

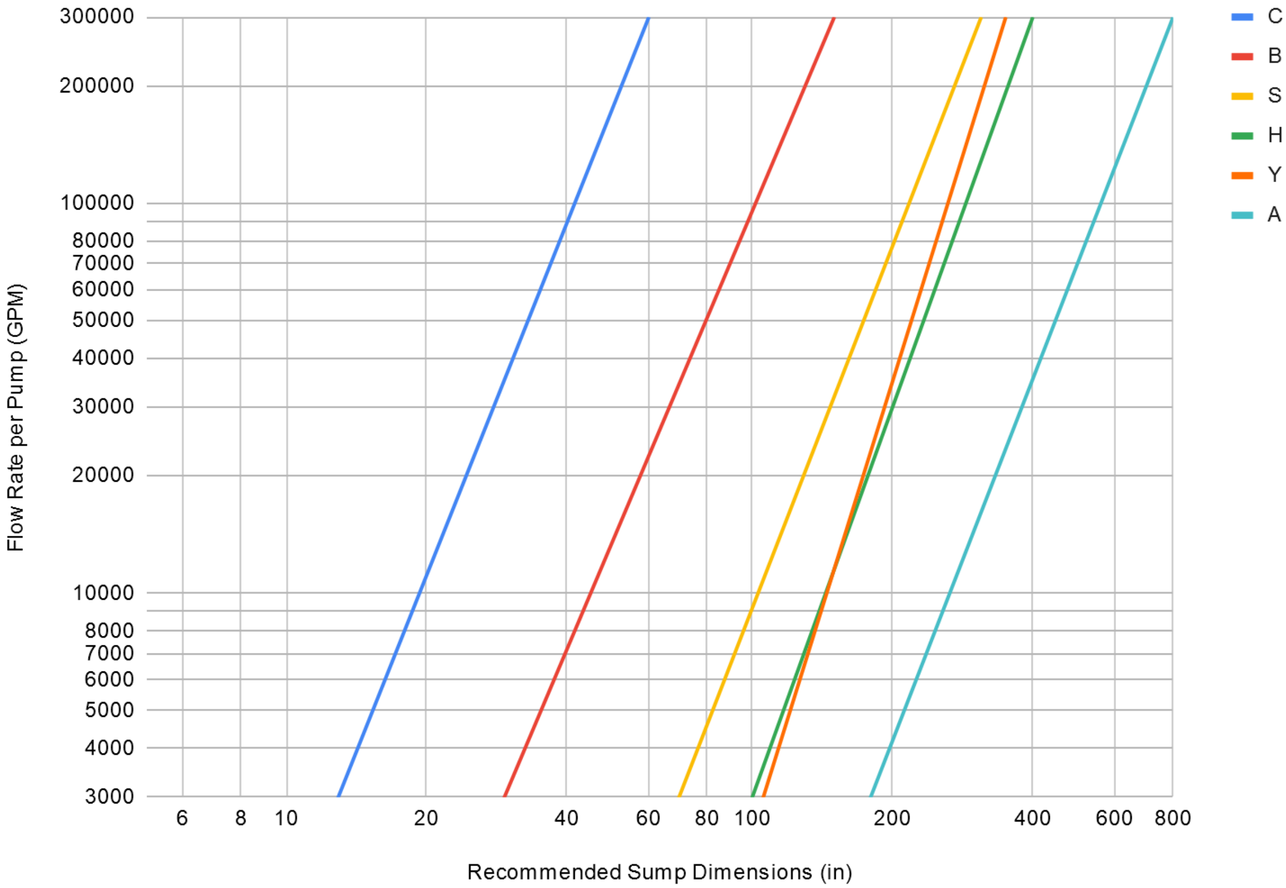


Figure 2: Multiple Sump

(Separating walls should not extend beyond the suction bell.)

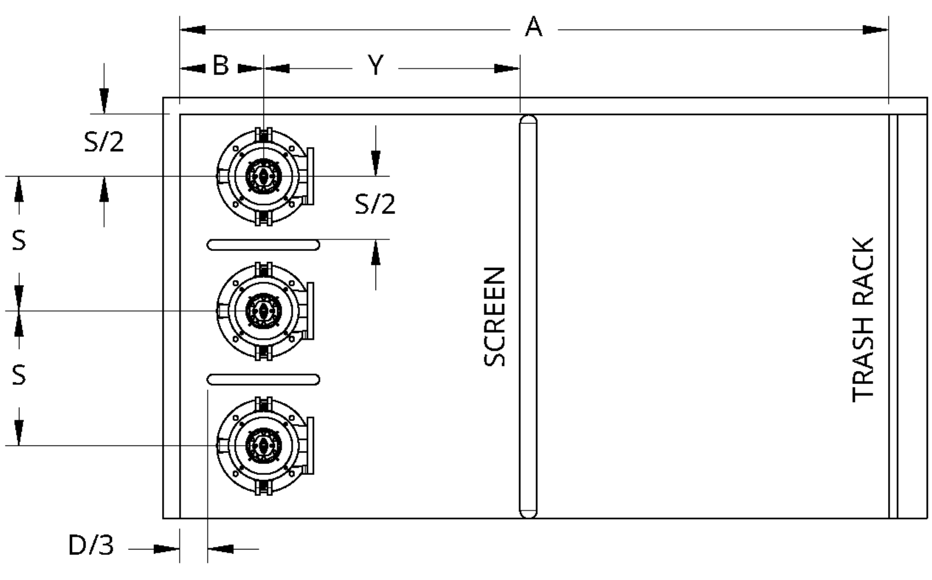


Figure 3: Minimum Submergence and Offset

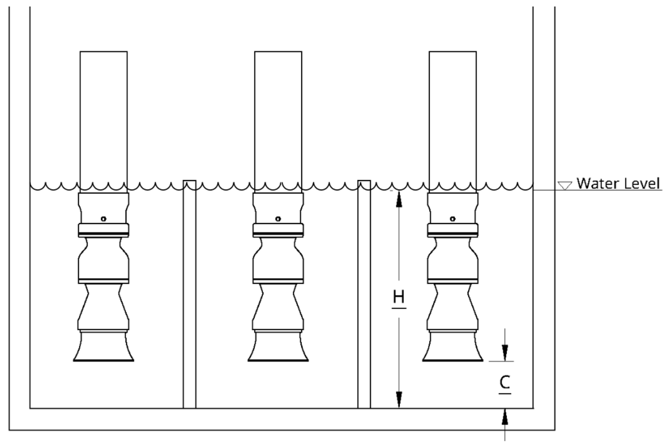


Figure 4: Minimum Submergence for Vortex Suppression

- Values in the chart below are in regards to vortex suppression, *not* NPSH required. Submergence to meet NPSH requirements may be higher than these values.
- Values shown below are measured from the intake or suction bell of the pump to the water level, or H - C as shown in Figure 3.

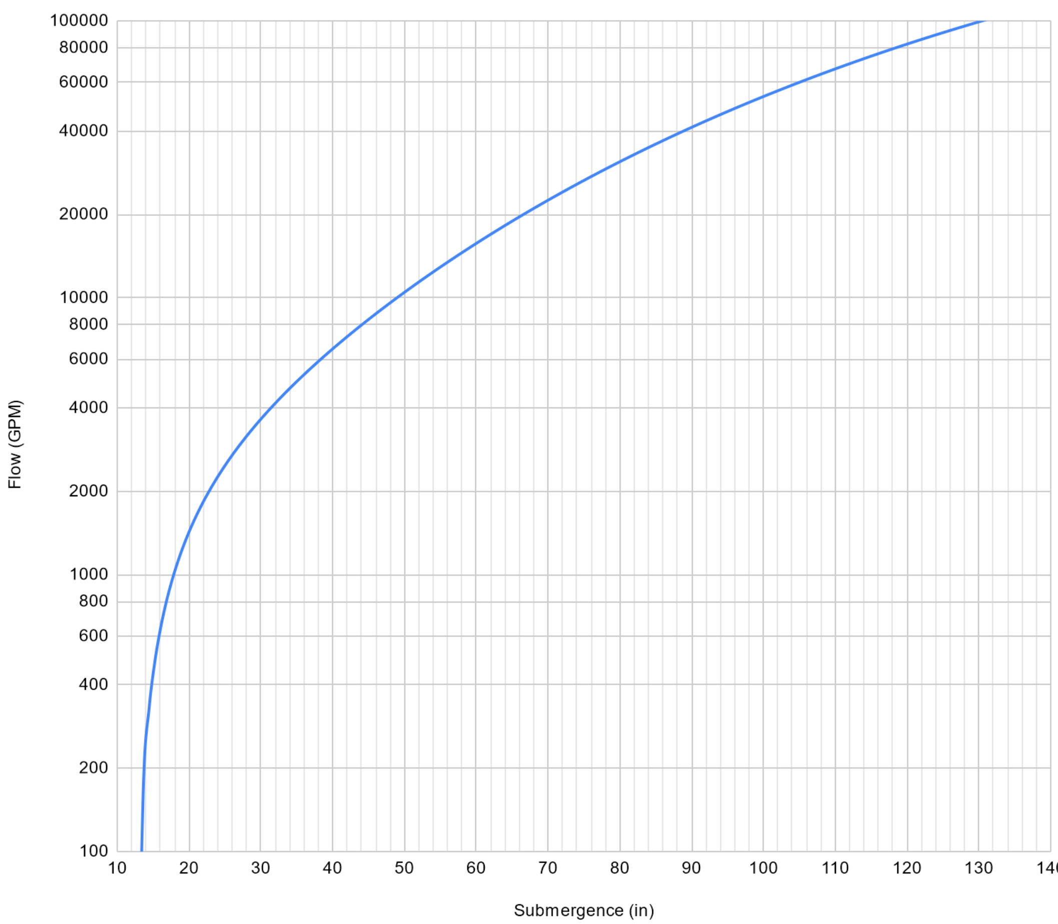
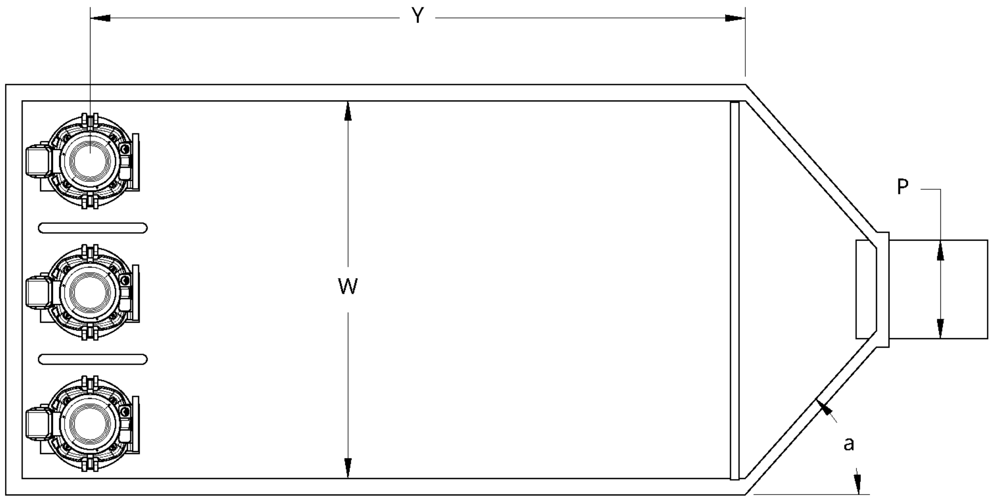


Figure 5: Pipe Fed Sump

- For an abrupt change in diameter from inlet pipe to sump, follow the layout below. Use the table to find the allowable channel velocity and Dimension Y based on the ratio of Dimension W to Dimension P.
- It is recommended that baffles, grating, or some type of strainer be used where the maximum channel width (W) first occurs.

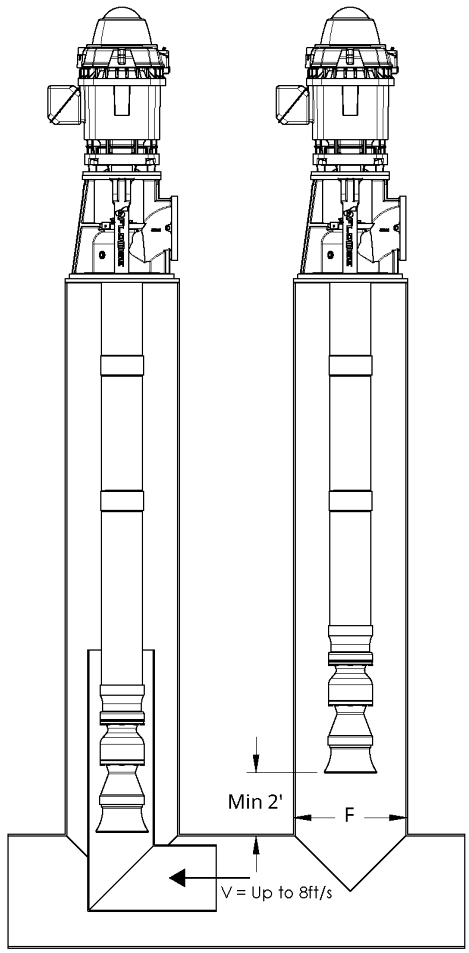


W/P	1.0	1.5	2.5	4.0	10.0
Y	3D	5D	8D	10D	15D
Velocity	1	2	4	6	8

"a" max = 15°
Recommended = 10°

Figure 6: Pipe Installation

- For pipeline installation as shown below in Figure 5, the design must either have an intake elbow as shown on the left, or the suction bell must be a minimum of 2 pipe diameters (or 2ft) above the top of the tunnel.
- The max velocity for the discharge elbow is 8ft/s as shown.



Correction of Existing Sumps

- If water must flow past one pump to get to another pump, rearrange the pump layout to match that shown in Figure 2.
- To aid in the prevention of vortices, add a cone underneath the pump suction as shown in Figure 6.
- Ensure that there is space between the back of the separating walls and the sump wall as shown in Figure 2. This dimension should be equal to D/3. Also ensure that the separating walls do not extend past the suction bell on either side.
- Ensure that Dimension B is a maximum of 3/4D. Install a false wall if necessary to achieve this dimension.
- Use floating rafts or spheres to break up surface vortices.
- Eliminate corners, sharp edges, or objects that are causing turbulence. Install smooth transitions where flow enters the sump as shown in Figure 7.

Figure 7: Suction Cone

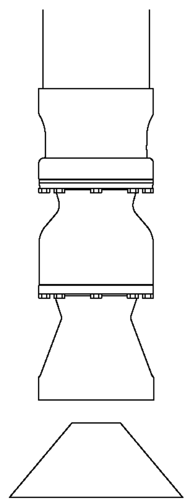
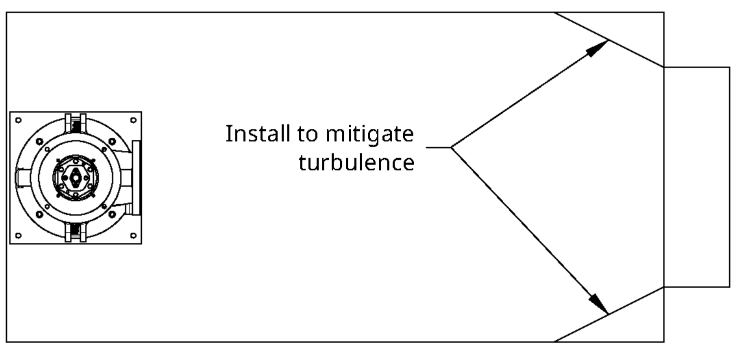


Figure 8: Sump Corner Adjustments



Bearing Spacing

Table 1: Maximum Bearing Spacing for Open Lineshaft Pumps

- Use Table 1 to find the appropriate lineshaft bearing spacing for open lineshaft pump column pipe.

Speed (RPM)	Bearing Material	Shaft Diameter (in.)						
		3/4	1	1-3/16	1-1/2	1-11/16	1-15/16	2-3/16
880	Rubber			120	120	120	120	120
	Solid			60	84	90	96	100
1200	Rubber	120	120	120	120	120	120	120
	Solid	60	60	60	72	78	84	88
1460	Rubber	120	120	120	120	120	120	120
	Solid	42	48	54	66	66	72	76
1760	Rubber	120	120	120	120	120	120	120
	Solid	42	48	48	60	60	66	70
2900	Rubber	60	60	60	60	60	60	60
	Solid	30	36	36	42	48	48	52
3600	Rubber	60	60	60	60	60	60	60
	Solid	30	30	36	36	42	48	48

Note:

- Solid bearing materials are brass, carbon, graphite, teflon, etc.

Suction Barrel Sizing

- Suction barrel capacity is limited by the fluid velocity. Maximum fluid velocity in the suction barrel should not exceed 5ft/sec.
- Use the following equation to determine the velocity in the suction barrel.

$$V = \frac{GPM \ x \ C}{D_1^{\ 2} - D_2^{\ 2}}$$

where:
GPM = maximum design flow (gallons per minute)
C = constant = 0.4085 (cubic ft/s)
D_1 = inner diameter of suction barrel (inches)
D_2 = outside diameter of bowl (inches)

<u>Example</u>

GPM = 1000
D_1 = 16 in. (suction barrel ID)
D_2 = 11.5 in. (bowl OD)

$$V = \frac{1000 \ x \ 0.4085}{16^2 - 11.5^2} = \frac{408.5}{256 - 132.25} = \frac{408.5}{123.75} = 3.3 \ ft/s$$

3.3 ft/s is less than 5ft/s, so a 16in suction barrel is acceptable.

Note:

- NPSHA may be affected by suction barrel installations and should be calculated during design
- Pump suction inlet should be qty (2) barrel diameters below or above the suction barrel inlet. The pump suction should never be in the area near the barrel inlet.
- The suction barrel inlet shall also have a maximum velocity of 5ft/s

Table 2: Suction Barrel Selection Chart
The chart below can be used as a reference guide for suction barrel sizing and is based on a fluid velocity of 5ft/s.

Bowl Diameter (in)	Barrel Diameter (in)									
	8	10	12	14	16	18	20	24	30	36
	Flow (GPM)									
5	448	892								
6	337	767								
7		603	1120	1486						
8		450	765	1372						
9			753	1115	1761	2490				
10			315	922	1620	2407				
11				625	1272	2001	2849			
12				382	1080	1867	2767			
13-14					450	1237	2137	4230		
15						877	1777	3870	7717	
16							1395	3487	7335	
18								2655	6502	

Prelubrication of Water Lubricated Pumps

For an open lineshaft pump with neoprene rubber shaft bearings, it may be necessary to have a prelubrication system installed on the pump. This system provides moisture to the bearings while the liquid level rises or falls in the column pipe at pump startup and shutdown.

Lubrication at Startup

- If the static water level is less than 30ft, a prelubrication system is usually not necessary at startup.
- If the static water level is greater than 30ft, refer to Figures and Table to design the prelubrication system.

Lubrication at Shutdown

- If a non-reverse ratchet (NRR) is installed in the driver, prelubrication at shutdown is not required.
- If there is not an NRR, refer to Figures and Table to design the prelubrication system.

Table 3: Prelubrication Solenoid Valve Size

- One method for prelubrication is to use a solenoid valve attached to a pressurized water source. Use Table 2 to determine the proper solenoid valve and fittings based on the pump column size and available pressure at the solenoid valve.

Pressure at Solenoid Valve	Column Size		
	5" or less	6" and 8"	10" and larger
1-10 PSI	1-1/4"	1-1/2"	2-1/2"
11-75 PSI	1"	1-1/4"	2"
76-150 PSI	3/4"	1"	1-1/2"

Table 4: Prelubrication Time Delay Relay

- Prelubrication is only required for a certain amount of time during startup. Use Table 3 to determine the required time delay setting for the solenoid valve.

Static Water Level (ft)	Time Delay (minutes)
0-30	0.5
31-70	1
71-150	1.5
151-250	2.5
251-350	3.5
351-450	

Table 5: Tank sizing

- If using a tank as the water source for prelubrication, use Table 4 to determine the proper tank size based on the pump column size and the static water level.

Column Size	Fittings	Tank Size (gal)	Fittings	Tank Size (gal)	Fittings	Tank Size (gal)
	1"	50	1-1/2"	100	2"	200
	Static Water Level					
2-1/2" to 4"	30'-300'		300'-400'			
4-1/2" to 6"	30'-200'		200'-400'			
8" to 10"	30'-125'		125'-300'		300'-400'	
12"	30'-70'		70'-200'		200'-400'	
14"	30'-50'		50'-150'		150'-300'	

Water Level Testing

Using an Airline

- Install an airline that extends from the surface to 2' above the inlet of the pump. Securely attach the airline to the pump assembly but take care not to crimp the airline. Measure and record the exact vertical length of the airline during installation.
- Attach a depth gauge or pressure gauge and snifter valve to the airline at the surface.
- Connect a tire pump to the snifter valve and expel all the water in the airline (if using a gauge with a movable dial - set the dial to 0 first).
- If you used an indirect depth gauge with a fixed dial: The reading on the dial after the water is expelled will be the level of water above the bottom of the airline (Dimension Z).
- If you used a direct depth gauge with a movable dial: Set the dial to 0 prior to expelling the water from the line. Use the tire pump to expel the water from the airline. The reading on the dial will now be the static water level (Dimension X)
- If you used a pressure gauge: The reading on the dial after the water is expelled will need converted from PSI to feet.

  Feet of water = 2.31 x PSI

  After converting to feet, this measurement is equal to the level of water above the bottom of the airline (Dimension Z).

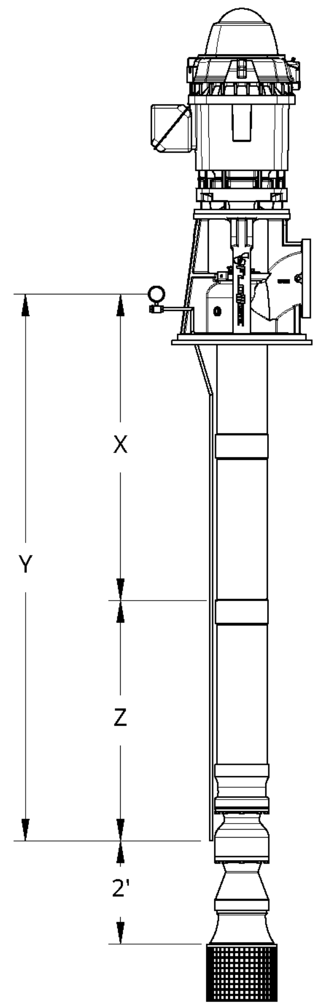


Figure 9: Water Level

Using an Electric Sounder

An electric sounder can also be used to measure water level in the well. The basic operation of a sounder is such:

- One terminal of a battery is connected directly to the pump discharge head.
- The other terminal of the battery is connected through a potentiometer to a spool of wire.
- The wire is lowered into the well until it reaches water. At this point, the circuit will close, the potentiometer needle deflects, and the length of wire is recorded as the water level depth.
- Care must be taken to ensure that the exposed wire does not contact the pump assembly while being lowered into the well. This would also close the circuit and provide a false reading of water level.

Affinity Laws - Changing Speed

In order to calculate the performance of a pump at speeds not shown on the manufacturer's published curves, one can use Affinity Laws.

Affinity Laws can be stated as such:

The flow is directly proportional to the speed

- $Q_2 = Q_1 \frac{N_2}{N_1}$

The head is proportional to the square of the speed

- $H_2 = H_1 (\frac{N_2}{N_1})^2$

The horsepower is proportional to the cube of the speed

- $BHP_2 = BHP_1 (\frac{N_2}{N_1})^3$

where:

Q = Flow (GPM)

H = Head (ft)

BHP = Brake Horsepower

N_1 = Original Speed in RPM (from known data)

N_2 = New Speed in RPM

Example:

A pump produces 1000 GPM at 37' TDH when operating at 1760 RPM and requires 12 BHP

What flow and head will it produce at 1400 RPM and what is the new BHP required?

Step 1: Flow

- $Q_2 = Q_1 \frac{N_2}{N_1}$
- $Q_2 = 1000 \frac{1400}{1760}$
- $Q_2 = 795\ GPM$

Step 2: Head

- $H_2 = H_1(\frac{N_2}{N_1})^2$
- $H_2 = 37(\frac{1400}{1760})^2$
- $H_2 = 23.4'\, TDH$

Step 3: BHP

- $BHP_2 = BHP_1(\frac{N_2}{N_1})^3$
- $BHP_2 = 12(\frac{1400}{1760})^3$
- $BHP_2 = 6\, BHP$

Note: It is not recommended to operate a turbine pump beyond 2200 RPM due to vibration and harmonics.

Similarly,

Trimming an impeller is another way to alter the performance of a pump. Since trimming the impeller changes the peripheral speed in the same way as reducing the impeller rotational speed, we can use a very similar set of formulas for calculating performance from a trimmed impeller.

- $Q_2 = Q_1 \frac{Imp_2}{Imp_1}$
- $H\ = H_1(\frac{Imp_2}{Imp_1})^2$
- $BHP_2 = BHP_1(\frac{Imp_2}{Imp_1})^3$

where:

Q = Flow (GPM)

Imp_1 = Original impeller trim (in)

Imp_2 = New impeller trim (in)

Table 6: Affinity Law Multipliers

Affinity Law Multipliers for Various Speeds Using 1760 RPM as Reference Speed								
RPM	GPM	Head	BHP		RPM	GPM	Head	BHP
1400	0.7955	0.6327	0.5033		2500	1.4205	2.0177	2.8660
1450	0.8239	0.6788	0.5592		2550	1.4489	2.0992	3.0415
1500	0.8523	0.7264	0.6191		2600	1.4773	2.1823	3.2239
1550	0.8807	0.7756	0.6831		2650	1.5057	2.2671	3.4135
1600	0.9091	0.8264	0.7513		2700	1.5341	2.3534	3.6104
1650	0.9375	0.8789	0.8240		2750	1.5625	2.4414	3.8147
1700	0.9659	0.9330	0.9012		2800	1.5909	2.5310	4.0266
1760					2850	1.6193	2.6222	4.2462
1800	1.0227	1.0460	1.0697		2900	1.6477	2.7150	4.4736
1850	1.0511	1.1049	1.1614		2950	1.6761	2.8094	4.7090
1900	1.0795	1.1654	1.2581		3000	1.7045	2.9055	4.9525
1950	1.1080	1.2276	1.3601		3050	1.7330	3.0031	5.2043
2000	1.1364	1.2913	1.4674		3100	1.7614	3.1024	5.4645
2050	1.1648	1.3567	1.5802		3150	1.7898	3.2033	5.7332
2100	1.1932	1.4237	1.6987		3200	1.8182	3.3058	6.0105
2150	1.2216	1.4923	1.8230		3250	1.8466	3.4099	6.2967
2200	1.2500	1.5625	1.9531		3300	1.8750	3.5156	6.5918
2250	1.2784	1.6343	2.0893		3350	1.9034	3.6230	6.8960
2300	1.3068	1.7078	2.2317		3400	1.9318	3.7319	7.2094
2350	1.3352	1.7828	2.3805		3450	1.9602	3.8425	7.5322
2400	1.3636	1.8595	2.5357		3500	1.9886	3.9547	7.8644
2450	1.3920	1.9378	2.6975		3520	2.0000	4.0000	8.0000

Specific Speed

Specific Speed is a useful measurement to compare impeller designs and understand their output. The specific speed of an impeller is the speed in RPM that the impeller would need to rotate if reduced in size to the point that it produces one GPM at one ft of head.

The calculation for specific speed is given below:

$$N_S = \frac{RPM\ x\ \sqrt{Flow_{GPM}}}{H_{ft}^{0.75}} \quad \text{OR} \quad N_S = \frac{RPM\ x\ \sqrt{Flow_{m3/hr}}}{H_m^{0.75}}$$

Where:

N_S = Specific speed

RPM = pump rotational speed in revolutions per minute

$Flow_{GPM}$ = pump flow in gallons per minute

$Flow_{m3/hr}$ = pump flow in cubic meters per hour

H_{ft} = head in feet

H_m = head in meters

Note:

1. Flow and head should be chosen at the best efficiency point of the max diameter shown on the pump performance curve.
2. Specific speed is always the value for a single impeller, not multiple stages.
3. The specific speed of a pump will be the same value at all rotational speeds.
4. Low specific speed is an indication that the pump is designed for low GPM and high head.
5. High specific speed is an indication that the pump is designed for high GPM and low head.

Example:

RPM = 1770

$Flow_{GPM}$ = 975

H_{ft} = 38

$$N_S = \frac{RPM\ x\ \sqrt{Flow_{GPM}}}{H_{ft}^{0.75}}$$

$$N_S = \frac{1770\ x\ \sqrt{975}}{38^{0.75}}$$

$$N_S = 3611.09$$

Required Torque

$$torque\ (lb.ft) = \frac{WR^2N}{307\,t}$$

where:

W = weight of impeller plus taper lock (lbs)

R = radius of gyration (ft)

N = change in RPM

t = time of acceleration (s)

Convert Linear Inertia to Rotational Inertia

$$Equivalent\ WR^2 = \frac{W}{39.48}\left(\frac{V}{N}\right)^2$$

where:

W = weight in lbs

V = linear velocity in ft/min = 0.262 x Dia(in.) x RPM

N = motor speed(RPM) when load is moving at velocity V

Equivalent WR^2 for Belted or Geared Loads

$$Equivalent\ (at\ Motor\ Shaft)\ WR^2 = WR^2_{(load)}\left(\frac{N_{load}}{N_{motor}}\right)^2$$

$$WR^2 = \frac{Actual\ Calculated}{WR^2{}_{of\ load}}$$

$$N_{load} = Full\ Speed\ of\ Load\ (RPM)$$

$$N_{motor} = Full\ Speed\ of\ Motor\ (RPM)$$

Thrust

There are two types of thrust to understand in a vertical turbine pump.

- Downthrust is the force created from moving the liquid upward in the pump.
- Upthrust is the force created by the velocity of liquid entering the impeller.

For an impeller with a low specific speed (low flow, high head), the upthrust can sometimes be larger than the downthrust. This creates a lift on the impellers, shafting, and ultimately the driver.

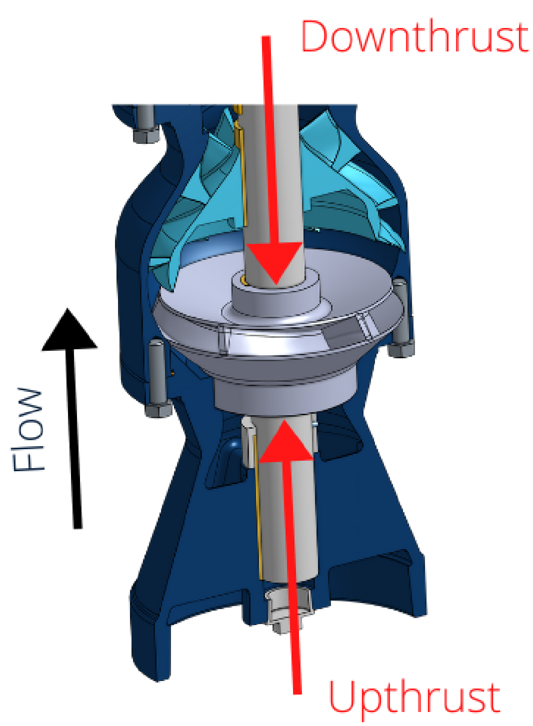
In most instances, the drive is designed to handle 30% momentary upthrust loads. Upthrust will typically occur at 120% of BEP. The force of the upthrust on deep well pumps is typically negated by the weight of the shafting and impellers.

For short pumps that may experience consistent upthrust, it is recommended that the pump be started against a closed valve until the system has developed sufficient pressure.

Note that the following issues can arrive from a pump continuously operating with an upthrust greater than the downthrust.

- Seal failure
- Bent lineshafts
- Impeller rub inside of the bowls
- Driver thrust bearing damage

Figure 10: Thrust



The total downthrust produced is the sum of the hydraulic thrust plus the static thrust, or dead weight, of the shaft and impellers. Use Table 6 to calculate the weight of the shafting.

Table 7: Shaft Weight/Area

Shaft Weight and Area										
Dia (in.)	3/4	1	1-3/16	1-1/4	1-1/2	1-11/16	1-15/16	2-3/16	2-1/4	2-7/16
Lbs/ft	1.5	2.67	3.77	4.17	6.01	7.6	10.02	12.78	13.52	15.87
Area (sq.in)	0.44	0.78	1.11	1.23	1.77	2.24	2.95	3.76	3.97	4.67

Total Thrust

$$Total\ Thrust\ =\ (K\ x\ H\ x\ SG)\ +\ (W\ x\ S)\ +\ (Impeller\ Weight\ x\ \#Stages)$$

where:

K =Thrust factor

H = Bowl head (total head + column friction loss) (ft)

W = Weight of shaft (lbs)

S = Total column length (ft)

SG = Specific gravity of fluid

Example:

K = 6.25 for given 3-stage pump with 1-½ lineshaft, total head of 205ft, and 250ft of column

K = 6.25

H = 207ft

W = 6.01lb/ft

S = 250ft

SG = 1 (water)

$$Total\ Thrust\ =\ (6.25\ x\ 207\ x\ 1)\ +\ (6.01\ x\ 250)\ +\ (16\ x\ 3)\ =\ 2,844.3\ lbs$$

Note:

- Thrust factors (K) and impeller weights are unique to each pump model and should be found with the performance data.
- The driver must have thrust capacity greater than the total thrust calculated.

Table 8: HP Loss Due to Mechanical Friction per 100ft of Column Pipe

Shaft Size (in)	RPM						
	3600	2900	1800	1500	1200	1000	900
3/4	0.60	0.52	0.32	0.26	0.20	0.17	0.15
1	1.10	0.88	0.55	0.44	0.35	0.29	0.26
1-3/16	1.45	1.30	0.75	0.61	0.48	0.40	0.36
1-1/4		1.33	0.79	0.67	0.52	0.44	0.39
1-1/2		1.90	1.20	0.96	0.75	0.60	0.55
1-11/16		2.36	1.40	1.20	0.94	0.78	0.70
1-15/16			1.90	1.60	1.20	1.00	0.90
2-3/16			2.30	2.00	1.50	1.30	1.15
2-1/4			2.50	2.07	1.60	1.41	1.26
2-7/16			2.90	2.40	1.90	1.60	1.40

Net Positive Suction Head

There are two terms for Net Positive Suction Head, NPSHa and NPSHr.

Net Positive Suction Head Available

NPSHa is the Net Positive Suction Head Available to the pump. This value is the total suction head less the absolute vapor pressure of the liquid.

For suction lift applications, NPSHa can be described as such:

$$NPSHa = h_{abs} - h_{vp} - h_{static} - h_{losses}$$

For flooded or pressurized suction applications, NPSHa can be described as such:

$$NPSHa = h_{abs} - h_{vp} + h_{static} - h_{losses}$$

where:

h_{abs} = absolute pressure (ft) on the liquid supply (atmospheric pressure if open tank or sump or absolute pressure in closed tank)

h_{vp} = vapor pressure of liquid (ft)

h_{static} = static height the pumped liquid is above or below the lowest pump impeller (ft)

h_{losses} = all losses on the suction side of the pump such entrance/exit and friction losses through pipe, valves, and fittings (ft)

Note:

The two different equations are to prevent NPSHa from ever being negative.

Net Positive Suction Head Required

Net Positive Suction Head Required (NPSHr) is the amount of suction head, less vapor pressure, that is required to prevent more than 3% of losses in total head of the first pump stage.

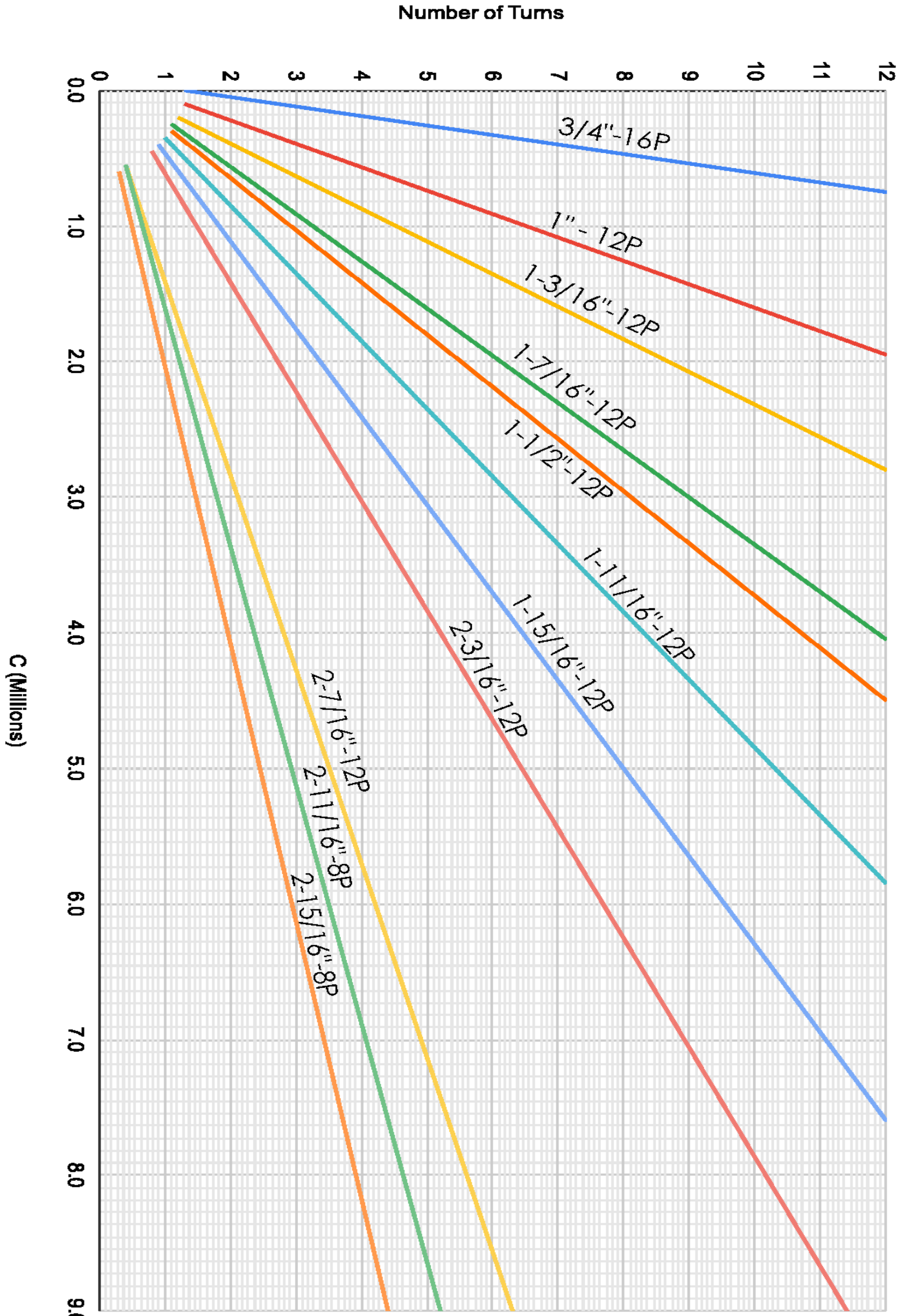
- Ensuring that NPSHr is less than NPSHa is very important in preventing air from coming out of solution. When entrained air is pulled from solution in a pump, it reduces performance, causes wear due to instability and vibration, and can cause cavitation.
- NPSHr is calculated under lab conditions at the manufacturer and should be found in the performance data.
- It is generally recommended that NPSHa exceed NPSHr by 2-3ft.

Table 9: Atmospheric Pressure at Altitude

Altitude	Atmospheric Pressure (PSI)	Atmospheric Pressure (ft of water)
0	14.7	34.0
500	14.4	33.3
1000	14.2	32.8
1500	13.9	32.1
2000	13.7	31.6
2500	13.4	31.0
3000	13.2	30.5
3500	12.9	29.8
4000	12.7	29.3
4500	12.4	28.6
5000	12.2	28.2
5500	12.0	27.7
6000	11.8	27.3
6500	11.5	26.6
7000	11.3	26.1
7500	11.1	25.6
8000	10.9	25.2
8500	10.7	24.7
9000	10.5	24.3
9500	10.3	23.8
10000	10.1	23.3
10500	9.9	22.9
11000	9.7	22.4
11500	9.5	21.9
12000	9.3	21.5
12500	9.1	21.0
15000	8.3	19.2

Shaft Adjustment

Figure 11: Number of Adjustment Nut Turns



Shaft Stretch

As a vertical turbine pump moves water upward in the column pipe, a downward force (downthrust) is exerted on the impeller and shafting. This force can stretch the shafting, especially on pumps with deep settings. The stretching of the shaft moves the impeller downward in the bowl.

To compensate for the shaft stretch, it is necessary to lift the impellers a certain distance off of the bottom of the bowls prior to pump startup.

This is achieved by turning the adjustment nut a certain number of turns.

To find out how many turns the adjustment nut requires, the first step is to determine the Thrust Constant (K) for the particular pump model in question. This can be found in the thrust constant table.

Once K is known, we can calculate a value of C and use that value in Figure X.

$$C \;=\; K\,x\,total\;head\;x\;setting$$

Example:

FloWise 14LC

77' Total Dynamic Head

500' Setting

1-11/16" lineshaft

K = 13lbs/ft

$$C \;=\; K\,x\,total\;head\;x\;setting$$

$$C = 13\;x\;77\;x\;500 = 500,500$$

Using a C of 0.5 on the 1-11/16" line in Figure 11, we see that 1.1 turns are required.

Shaft Selection Chart

Table 10: Shaft Selection

Shaft Diameter (in)	Speed (RPM)	Pump Thrust (lbs)								
		1000	2000	5000	7500	10000	15000	20000	25000	30000
		Power Rating (HP)								
1	3550	120	119	116	110					
	1770	60	59	58	55					
	1180	40	40	38	37					
1-3/16	3550	213	212	209	205					
	1770	106	106	104	102					
	1180	71	71	70	68					
1-1/2	3550	435	435	433	429	424	410			
	1770	217	217	216	214	212	205			
	1180	145	145	144	143	141	136			
	880	108	108	107	106	105	102			
1-11/16	3550	639	639	637	631	615	570	500		
	1770	319	319	318	314	307	284	249		
	1180	213	212	212	210	205	190	166		
	880	158	158	158	156	153	141	124		
1-15/16	1770	498	498	497	496	494	489	456	419	369
	1180	332	332	332	331	329	326	304	279	246
	880	248	248	247	247	246	243	226	208	184
2-3/16	1770	634	633	631	626	620	602	576	541	494
	1180	423	422	420	417	413	401	384	361	330
	880	315	315	313	311	308	299	286	269	246
2-7/16	1770	1037	1037	1036	1035	1029	1016	996	970	938
	1180	691	691	691	690	686	677	664	647	625
	880	515	515	515	515	512	505	495	482	466
2-11/16	1770	1358	1358	1358	1357	1356	1352	1347	1340	1332
	1180	906	906	905	905	904	901	898	893	888
	880	675	675	675	675	674	672	670	666	662
2-15/16	1770	1803	1803	1802	1802	1800	1797	1793	1787	1779
	1180	1202	1202	1202	1201	1200	1198	1195	1191	1186
	880	896	896	896	896	895	894	891	888	885
3-3/16	1770	2336	2336	2335	2334	2333	2330	2326	2321	2314
	1180	1557	1557	1557	1556	1555	1553	1551	1547	1543
	880	1161	1161	1161	1161	1160	1159	1156	1154	1150
3-7/16	1770	2740	2740	2738	2736	2732	2722	2708	2690	2667
	1180	1827	1827	1826	1824	1822	1815	1805	1793	1778
	880	1362	1362	1362	1360	1358	1353	1346	1337	1326

- Use a 0.75 multiplier for keyed shafts.
- For bowl shafts, use only the hydraulic thrust load.
- For lineshaft, use the total thrust.
- Hydraulic thrust = "K" x TDH
- Total thrust = Hydraulic thrust + Lineshaft weight

Material	Multiplier
316SS	0.88
416SS	1.18
17-4PH	1.59
K-MONEL	1.65

Shaft Elongation

Shaft elongation occurs from the downthrust of a pump plus the weight of the shafting and impellers.

It is expressed as follows:

$$e = \frac{L \, x \, 12 \, x \, Thrust}{E \, x \, GSA}$$

where:

e = shaft elongation (in)

L - shaft length (ft)

E = modulus of elasticity (29,000,000)

Thrust = hydraulic thrust (lbs)

GSA = gross shaft area (in^2)

Table 11: Shaft Elongation per 100ft

Hydraulic Thrust	Shaft Diameter													
	3/4	12/31	1-3/16	1-1/2	1-11/16	1-15/16	2-3/16	2-7/16	2-11/16	2-15/16	3-3/16	3-7/16	3-11/16	3-15/16
500	0.047	0.026	0.018	0.012	0.009	0.007								
600	0.056	0.032	0.022	0.014	0.011	0.008	0.006							
800	0.075	0.042	0.03	0.019	0.015	0.011	0.009							
1000	0.094	0.053	0.037	0.024	0.019	0.014	0.011	0.009						
1200	0.112	0.063	0.045	0.028	0.022	0.017	0.013	0.011						
1400	0.131	0.074	0.052	0.033	0.026	0.02	0.015	0.012	0.01					
1600	0.15	0.084	0.06	0.038	0.03	0.022	0.018	0.014	0.012					
1800	0.169	0.095	0.067	0.042	0.033	0.025	0.02	0.016	0.013	0.011				
2000	0.187	0.105	0.075	0.047	0.037	0.028	0.022	0.018	0.015	0.012				
2400	0.225	0.127	0.09	0.056	0.044	0.034	0.026	0.021	0.018	0.015	0.012			
2800	0.262	0.148	0.105	0.066	0.052	0.039	0.03	0.025	0.02	0.017	0.015			
3200		0.169	0.119	0.075	0.059	0.045	0.035	0.028	0.023	0.02	0.017	0.014		
3600		0.19	0.135	0.085	0.067	0.051	0.04	0.032	0.026	0.022	0.019	0.016		
4000		0.211	0.15	0.094	0.074	0.056	0.044	0.036	0.029	0.025	0.021	0.018	0.016	
4400		0.24	0.164	0.103	0.081	0.062	0.048	0.039	0.032	0.027	0.024	0.02	0.017	
4800		0.253	0.179	0.113	0.089	0.067	0.053	0.043	0.035	0.029	0.025	0.021	0.019	0.016
5200		0.274	0.194	0.122	0.096	0.073	0.057	0.046	0.038	0.032	0.027	0.023	0.02	0.018
5600			0.209	0.131	0.107	0.079	0.062	0.05	0.041	0.034	0.029	0.025	0.022	0.019
6000			0.224	0.141	0.111	0.084	0.066	0.053	0.044	0.037	0.031	0.027	0.023	0.02
6500			0.243	0.153	0.12	0.091	0.071	0.058	0.047	0.04	0.034	0.029	0.025	0.022
7000			0.26	0.164	0.129	0.098	0.077	0.062	0.051	0.043	0.036	0.031	0.027	0.024
7500				0.176	0.139	0.105	0.082	0.067	0.055	0.046	0.039	0.033	0.029	0.026
8000				0.188	0.148	0.112	0.088	0.071	0.058	0.049	0.042	0.036	0.031	0.027
9000				0.211	0.167	0.126	0.098	0.08	0.066	0.055	0.047	0.04	0.035	0.031
10000				0.234	0.185	0.14	0.11	0.089	0.073	0.061	0.052	0.045	0.039	0.034
12000				0.281	0.222	0.168	0.132	0.106	0.088	0.073	0.062	0.054	0.047	0.041
14000					0.259	0.196	0.154	0.124	0.102	0.086	0.073	0.062	0.055	0.048
16000					0.296	0.224	0.176	0.142	0.117	0.098	0.083	0.071	0.062	0.054
18000						0.252	0.198	0.16	0.131	0.11	0.093	0.08	0.07	0.061
20000						0.28	0.22	0.176	0.146	0.122	0.104	0.089	0.078	0.068
22000							0.242	0.195	0.16	0.134	0.114	0.098	0.086	0.074
24000							0.264	0.213	0.175	0.147	0.124	0.107	0.094	0.082
26000							0.286	0.23	0.19	0.159	0.135	0.116	0.102	0.088
28000								0.248	0.204	0.171	0.145	0.125	0.109	0.095
30000								0.266	0.219	0.183	0.156	0.134	0.117	0.104
32000								0.283	0.233	0.196	0.166	0.143	0.125	0.109
34000									0.248	0.208	0.176	0.152	0.133	0.116
36000									0.262	0.22	0.187	0.16	0.14	0.122
38000									0.277	0.232	0.197	0.17	0.148	0.129
40000									0.292	0.245	0.207	0.178	0.156	0.136

Column and Tube Elongation

Table 12: Column and Tube Elongation per 100ft

Hydraulic Thrust	Column Diameter								
	3"	4"	5"	6"	8"	10"	12"	14"	16"
500	0.007	0.005	0.004	0.003					
600	0.008	0.006	0.005	0.004					
800	0.011	0.008	0.006	0.005					
1000	0.013	0.01	0.008	0.006	0.004				
1200	0.016	0.012	0.009	0.007	0.005				
1400	0.019	0.014	0.011	0.008	0.006				
1600	0.021	0.016	0.012	0.009	0.007	0.005			
1800	0.024	0.018	0.014	0.011	0.008	0.006			
2000	0.027	0.02	0.015	0.012	0.009	0.007			
2400	0.032	0.023	0.019	0.014	0.01	0.008	0.006		
2800	0.037	0.027	0.022	0.016	0.012	0.01	0.007		
3200	0.043	0.031	0.025	0.019	0.014	0.011	0.008		
3600	0.048	0.035	0.028	0.021	0.016	0.012	0.009	0.008	
4000		0.039	0.031	0.023	0.017	0.014	0.01	0.008	
4400		0.043	0.034	0.026	0.019	0.015	0.011	0.009	
4800		0.047	0.037	0.028	0.021	0.016	0.013	0.01	0.009
5200		0.051	0.04	0.03	0.023	0.018	0.014	0.011	0.01
5600		0.055	0.043	0.033	0.024	0.019	0.015	0.012	0.011
6000			0.046	0.035	0.026	0.02	0.016	0.013	0.011
6500			0.05	0.038	0.028	0.022	0.017	0.014	0.012
7000			0.054	0.041	0.03	0.024	0.018	0.015	0.013
7500			0.058	0.044	0.033	0.025	0.02	0.016	0.014
8000			0.062	0.047	0.035	0.027	0.021	0.017	0.015
9000				0.053	0.039	0.03	0.023	0.019	0.017
10000				0.059	0.043	0.034	0.026	0.021	0.019
12000				0.07	0.052	0.041	0.031	0.025	0.023
14000				0.082	0.061	0.048	0.036	0.029	0.026
16000				0.094	0.07	0.054	0.042	0.034	0.03
18000					0.078	0.061	0.047	0.038	0.034
20000					0.087	0.068	0.052	0.042	0.037
22000					0.096	0.075	0.057	0.046	0.041
24000					0.104	0.082	0.063	0.05	0.045
26000					0.113	0.088	0.068	0.055	0.049
28000						0.095	0.073	0.059	0.052
30000						0.102	0.078	0.063	0.056
32000						0.109	0.083	0.067	0.06
34000						0.115	0.089	0.071	0.064
36000						0.122	0.094	0.076	0.068
38000						0.129	0.099	0.08	0.071
40000						0.136	0.104	0.084	0.075

Thrust Bearing Horsepower Loss

Losses from external thrust loads on the rotor must be added to mechanical friction in order to get the total pump brake horsepower requirement.

Thrust loss in HP can be calculated as follows:

$$Thrust\ Bearing\ HP\ Loss\ =\ 0.0075\ x\ \frac{RPM}{100}\ x\ \frac{Thrust}{1000}$$

For example:

If total thrust = 3676 lbs at 1770 RPM

$$Thrust\ Bearing\ HP\ Loss\ =\ 0.0075\ x\ \frac{1770}{100}\ x\ \frac{3676}{1000}\ =\ 0.49\ HP\ Loss$$

Table 12 shows the approximate Thrust Bearing HP Loss at given thrust values and speeds.

Table 13: Thrust Bearing HP Loss

(Assuming angular contact anti-friction bearings)

Total Thrust (lbs)	RPM			
	3500	1770	1170	880
1000	0.262	0.133	0.088	0.066
2000	0.525	0.268	0.175	0.132
3000	0.79	0.4	0.263	0.198
4000	1.05	0.532	0.35	0.264
5000	1.32	0.665	0.438	0.33
6000	1.58	0.796	0.525	0.396
7000	1.84	0.93	0.615	0.46
8000	2.1	1.06	0.7	0.528
9000	2.36	1.2	0.79	0.593
10000	2.62	1.33	0.88	0.66
15000	3.95	1.98	1.4	0.99
20000	5.25	2.68	1.75	1.32
25000		3.32	2.2	1.65
30000		4	2.63	1.98
35000		4.65	3.07	2.3
40000		5.32	3.5	2.64
45000		5.98	3.95	2.97
50000			4.38	3.3

Power Consumption

There are two primary methods of measuring power consumption.

The first method uses an ammeter and voltmeter. Using the values obtained from these meters, you can solve the equation below to calculate power consumption in kilowatts.

$$Kilowatts = \frac{I \; x \; E \; x \; P.F. \; x \; C}{1000}$$

where:

I = Amperes

E = Volts

P.F. = Power Factor (see motor manufacturer's published motor operating characteristics)

C = 1 (single phase)

OR

C = 2 (two phase, four wire)

OR

C = 1.73 (three phase)

The second method to calculate power consumption uses the watt-hour meter in the power line.

If you measure the revolutions of the meter disc over a set period of time, you can use the equation below to find power consumption in kilowatts.

$$Kilowatts = 3.6 \; x \; K \; x \; M \; x \; R/t$$

where:

K = Disc constant (Represents watt-hrs/rev and can be found on the meter nameplate or disc)

M = Product of current transformer ratio and potential transformer ratio (If either of the transformers are not used, the equivalent ratio is 1)

R = Number of revolutions of the watt-hr meter disc

t = Length of measurement in seconds

Energy Cost of Pumping with an Electric Motor

It is often beneficial to understand the energy costs of a pumping system.

If you have already calculated the power consumption of the motor in the previous section, then you can quickly figure the cost per hour as follows:

$$Energy\ Cost/hr\ of\ pumping\ =\ KW\ consumed\ x\ Cost\ per\ Kilowatt\ Hour$$

To estimate energy cost when measured power consumption values are not available you can use the following two methods:

Either

$$Energy\ Cost/hr\ of\ pumping\ =1\ HP\ x\ 0.746\ x\ Cost\ per\ Kilowatt\ Hour$$

OR

$$Energy\ Cost/hr\ of\ pumping\ =\frac{GPM\ x\ Total\ Head\ x\ 0.746\ x\ Cost\ per\ Kilowatt\ Hour}{3960\ x\ Pump\ Efficiency\ x\ Motor\ Efficiency}$$

If you want to convert Energy Cost/hr of pumping to Energy Cost/1000 Gallons:

$$Energy\ Cost/1000\ Gallons\ =\frac{Energy\ Cost/hr\ of\ Pumping}{GPM}\ x\ 16.667$$

Table 13 shows the approximate Kilowatt-hrs per 1000 Gallons at 1 ft TDH with respect to overall pump efficiency. You can use this table to quickly get approximate values of Energy Cost/1000 Gallons as follows:

Example:

Assume 84% overall pump efficiency (including all losses in the pump unit), 175ft TDH, and $0.11/KW-hr

Per Table 13 and an 84% overall efficiency, we get 0.00373 Kilowatt-hrs/1000 Gallons at 1 ft TDH

Kilowatt-hrs/1000 Gallons = 0.00373 x TDH = 0.00373 x 175 = 0.6528

Energy Cost/1000 Gallons = 0.6528 * Cost per Kilowatt Hour = 0.6528 * 0.11 = $0.0718

Table 14: Approximate KW-hr / 1000GPM

Overall Efficiency	Kilowatt-hrs per 1000 Gallons at 1ft TDH		Overall Efficiency	Kilowatt-hrs per 1000 Gallons at 1ft TDH
32	0.00981		62	0.00506
33	0.00951		63	0.00498
34	0.00923		64	0.00491
35	0.00897		65	0.00483
36	0.00872		66	0.00476
37	0.00849		67	0.00469
38	0.00826		68	0.00462
39	0.00805		69	0.00455
40	0.00785		70	0.00449
41	0.00766		71	0.00442
42	0.00748		72	0.00436
43	0.00730		73	0.00430
44	0.00714		74	0.00424
45	0.00698		75	0.00419
46	0.00683		76	0.00413
47	0.00668		77	0.00408
48	0.00654		78	0.00403
49	0.00641		79	0.00397
50	0.00628		80	0.00392
51	0.00616		81	0.00388
52	0.00604		82	0.00383
53	0.00592		83	0.00378
54	0.00581		84	0.00374
55	0.00571		85	0.00369
56	0.00561		86	0.00365
57	0.00551		87	0.00361
58	0.00541		88	0.00357
59	0.00532		89	0.00353
60	0.00523		90	0.00349
61	0.00515		91	0.00345

Column Friction Loss

Table 15 - Column Friction Loss

Column Size (in)	3	4			5			6			8			10			12				Column Size (in)
Shaft Size (in)	3/4	3/4	1	1 3/16	3/4	1	1 3/16	1	1 3/16	1 1/2	1 3/16	1 1/2	1 15/16	1 3/16	1 1/2	1 15/16	1 3/16	1 1/2	1 15/16	2 3/16	Shaft Size (in)
Tube Size (in)	1 1/4	1 1/4	1 1/2	2	1 1/4	1 1/2	2	1 1/2	2	2 1/2	2	2 1/2	3	2	2 1/2	3	2	2 1/2	3	3 1/2	Tube Size (in)
GPM	Column Friction Loss (ft) Per 100 Feet of Column																				GPM
25	1.80																				25
50	4.60	0.65	0.86	1.60																	50
75	9.00	1.30	1.70	3.30																	75
100	14.00	2.20	2.80	5.30	0.54	0.65	0.94														100
125		3.20	4.20	7.80	0.81	0.96	1.40														125
150		4.40	5.80	10.60	1.10	1.30	1.90														150
175		5.80	7.50	13.80	1.50	1.70	2.50														175
200		7.30	9.40	17.10	1.80	2.20	3.10	0.73	0.96	1.40											200
225		9.00	12.00	21.10	2.30	2.70	3.90	0.90	1.20	1.70											225
250		10.90	14.00		2.70	3.30	4.70	1.10	1.40	2.00											250
275		13.00	16.80		3.30	3.90	5.60	1.30	1.70	2.40											275
300		15.20	19.20		3.80	4.50	6.40	1.50	2.00	2.80											300
325		19.80			4.40	5.20	7.40	1.70	2.30	3.20											325
350					5.00	6.00	8.40	2.00	2.60	3.60											350
375					5.60	6.70	9.50	2.20	2.90	4.10											375
400					6.30	7.50	10.60	2.50	3.30	4.60	0.61	0.74	1.00								400
450					7.80	9.30	13.10	3.10	4.10	5.70	0.77	0.91	1.30								450
500					9.20	11.20	15.70	3.70	5.00	6.90	0.93	1.10	1.50								500
550					11.0	13.20	18.60	4.40	5.80	8.10	1.10	1.30	1.80								550
600					12.9	15.50		5.20	6.80	9.50	1.30	1.50	2.10								600
650					14.8	20.30		6.00	7.90	11.00	1.50	1.80	2.50								650
700					16.8			6.90	9.10	12.50	1.70	2.00	2.80								700
750					19.0			7.90	10.30	14.10	1.90	2.30	3.20								750
800								8.80	11.50	15.70	2.20	2.60	3.60	0.57	0.65	0.77					800
850								9.90	12.80	17.70	2.40	2.90	4.00	0.63	0.72	0.86					850
900								11.00	14.30	19.50	2.70	3.20	4.50	0.70	0.80	0.96					900
950								12.10	15.80	21.50	2.90	3.50	4.90	0.77	0.88	1.10					950

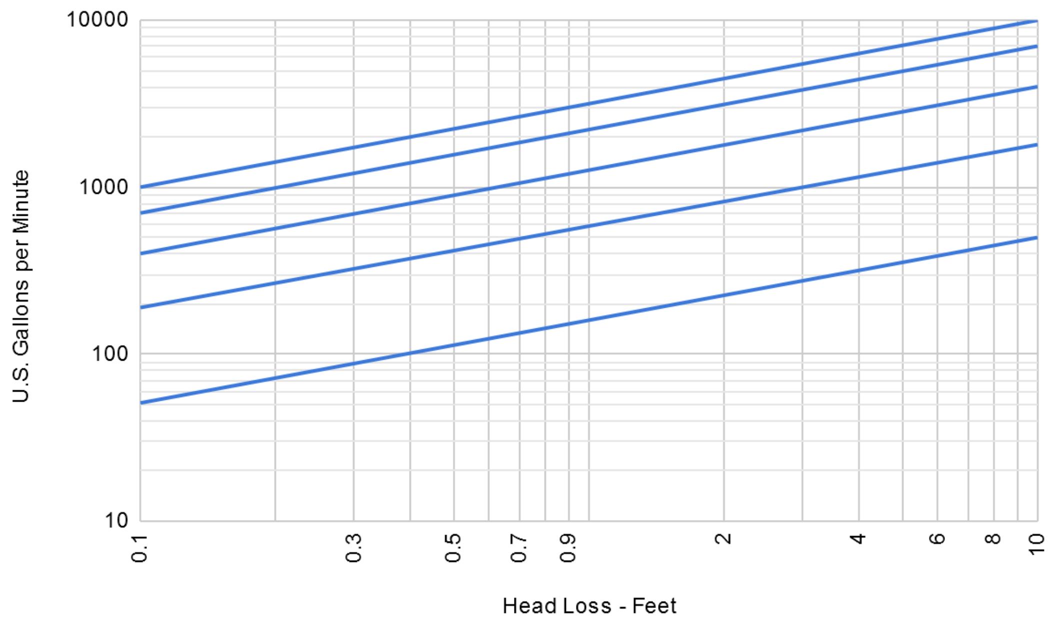
Mutlipliers for Pipe Condition		
Condition Inside	Approx Age	Multi-plier
Very Smooth	New	1.00
Fairly Smooth	1-5 Years	1.51
Rough	> 6 years	2.35

Column Size (in)	3	4			5			6			8			10			12				Column Size (in)
Shaft Size (in)	3/4	3/4	1	1 3/16	3/4	1	1 3/16	1	1 3/16	1 1/2	1 3/16	1 1/2	1 15/16	1 3/16	1 1/2	1 15/16	1 3/16	1 1/2	1 15/16	2 3/16	Shaft Size (in)
Tube Size (in)	1 1/4	1 1/4	1 1/2	2	1 1/4	1 1/2	2	1 1/2	2	2 1/2	2	2 1/2	3	2	2 1/2	3	2	2 1/2	3	3 1/2	Tube Size (in)
GPM	Column Friction Loss (ft) Per 100 Feet of Column																				GPM
1600											7.60	9.10	13.00	2.00	2.30	2.80	0.80	0.90	1.10	1.20	1600
1800											9.40	11.00	15.70	2.50	2.80	3.40	0.99	1.10	1.30	1.50	1800
2000											11.00	13.00	19.20	3.00	3.50	4.20	1.20	1.40	1.60	1.80	2000
2200											13.20	16.50	22.90	3.60	4.10	5.00	1.40	1.60	1.90	2.10	2200
2400											15.50	19.30		4.20	4.90	5.80	1.70	1.90	2.20	2.50	2400
2600											17.90	22.40		4.90	5.60	6.80	1.90	2.20	2.50	2.90	2600
2800											20.50			5.60	6.40	7.80	2.20	2.50	2.80	3.30	2800
3000														6.40	7.40	8.80	2.50	2.90	3.30	3.80	3000
3200														7.10	8.10	9.90	2.80	3.20	3.70	4.30	3200
3400														7.90	9.00	11.10	3.20	3.60	4.20	4.80	3400
3600														8.80	10.00	12.40	3.50	4.00	4.70	5.30	3600
3800														9.80	11.10	13.70	3.90	4.40	5.10	5.90	3800
4000														10.70	12.20	15.00	4.30	4.90	5.60	6.40	4000
4200														11.80	13.40	16.40	4.70	5.30	6.20	7.10	4200
4400														12.90	14.60	17.90	5.10	5.80	6.70	7.70	4400
4600														13.90	15.80	19.30	5.60	6.30	7.40	8.40	4600
4800														15.00	17.20	21.00	6.00	6.80	7.90	9.00	4800

Mutlipliers for Pipe Condition		
Condition Inside	Approx Age	Multi-plier
Very Smooth	New	1.00
Fairly Smooth	1-5 Years	1.51
Rough	> 6 years	2.35

Cast Discharge Head Friction Loss

Figure 12 - Cast Discharge Head Friction Loss



Fabricated Discharge Head Friction Loss

Figure 13 - Fabricated Discharge Head Friction Loss

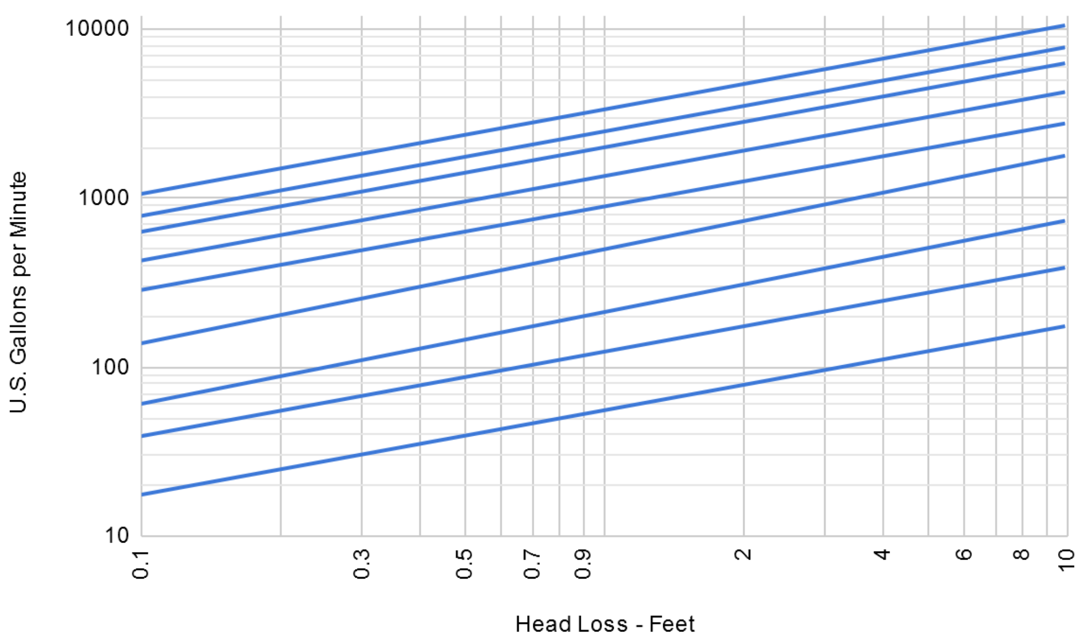


Table 16: Mechanical Friction in Turbine Pump Line Shafts

Mechanical Friction in Turbine Pump Line Shafts (HP/100ft)								
Shaft Dia (in)	RPM							
	3450	2900	2200	1760	1460	1160	880	700
3/4			0.38	0.3	0.25	0.22		
1	1.04	0.87	0.65	0.52	0.45	0.35		
1-3/16	1.44	1.2	0.9	0.72	0.6	0.44		
1-1/2	2.3	1.92	1.44	1.15	0.95	0.74	0.56	
1-11/16				1.4	1.2	0.92	0.7	
1-15/16				1.8	1.5	1.2	0.9	0.72
2-3/16				2.3	1.9	1.5	1.15	0.92
2-7/16				2.85	2.4	1.85	1.4	1.13

Shaft Weights

Table 17: Shaft Weight

Shaft Weights (lb/ft)		
Shaft Diameter	Enclosed	Open
3/4	1.5	1.3
1	2.6	2.3
1-3/16	3.8	3.3
1-11/16	6	5.3
1-15/16	7.6	6.3
2-3/16	10	8.8
2-7/16	12.8	11.2

Turbine Mechanical Data

Table 18 - Turbine Mechanical Data

Model	Standard Lateral	Max. Lateral	Allowable Sphere (in)	Net Eye Area (in²)	No. of Vanes	Impeller Weight (lbs)		K Factor	
						Closed	Open	Closed	Open
5I	0.50	0.56	0.15	2.11	5	1.2	N/A	1.30	N/A
5K	0.50	0.56	0.15	2.11	9	1.2	N/A	1.30	N/A
5L	0.31	0.31	0.22	2.95	5	1.8	N/A	1.40	N/A
5H	0.31	0.31	0.22	2.95	8	1.8	N/A	1.40	N/A
5W	0.38		0.43	5.03	5		N/A		N/A
5Y	0.38		0.43	5.03	8		N/A		N/A
6I	0.438	0.63	0.19	3.70	6	1.9	N/A	2.24	N/A
6K	0.375	0.63	0.19	3.70	6	1.9	N/A	2.24	N/A
6L	0.38	0.625	0.22	4.12	5	2.1	N/A	2.10	N/A
6H	0.38	0.625	0.22	4.12	8	2.1	N/A	2.10	N/A
6W	0.44	0.625	0.50	7.15	4	1.1	N/A	5.60	N/A
6Y	0.44	0.625	0.50	7.15	7	1.1	N/A	5.60	N/A
7L	0.50	0.50	0.43	6.44	5	3.9	2.18	3.50	3.80
7H	0.50	0.50	0.43	6.44	8	3.9	2.29	3.50	3.80
7W	0.38	0.75	0.83	10.11	4	4.3	N/A	4.50	N/A
7Y	0.38	0.75	0.83	10.11	7	4.5	N/A	4.56	N/A
8I	0.438	0.438	0.25	3.66	6	4.7	2.30	2.98	3.52
8K	0.438	0.438	0.25	4.61	6	4.5	2.69	2.98	3.34
8L	0.50	0.56	0.43	8.51	5	5.0	2.86	4.00	5.30
8H	0.50	0.56	0.43	8.51	8	4.9	2.89	4.00	5.30
8Q	0.56	1.75	0.46	13.81	5	5.0	3.00	7.90	9.90
8R	0.56	1.75	0.46	13.81	7	4.8	2.95	7.90	9.90
8W	0.56	0.88	0.46	14.58	7	5.4	3.92	7.90	9.00
9L	0.88	1.25	0.56	10.93	5	6.6	3.5	4.90	6.00
9H	0.88	1.25	0.56	10.93	8	6.7	3.7	4.90	6.00
9W	0.75	2	1	17.08	4	8.1	5.3	9.00	10.50
9Y	0.75	1.875	0.68	17.08	7	13.7	11.0	9.00	10.50
10I	0.63	0.75	0.45	8.64	5	7.3	4.1	4.60	6.50
10K	0.63	0.75	0.45	8.64	8	7.3	4.2	4.65	6.50
10L	0.75	1	0.68	13.09	5	8.1	4.3	7.00	9.50
10M	0.75	1	0.68	13.09	6	8.2	4.5	7.00	9.50
10H	0.75	1	0.68	13.09	8	8.6	4.9	7.00	9.50
10W	0.88	1.13	0.87	19.41	6	7.9	4.9	10.30	11.20
10Y	0.75	1.13	0.87	19.41	6	8.1	4.9	10.30	11.40
10Z	0.50	0.88	1.43	26.70	6	9.2	5.3	11.40	13.50
11L	0.75	0.88	0.68	15.71	5	10.9	5.8	7.10	9.10
11M	0.75	0.88	0.68	15.71	7	11.0	6.0	7.00	9.10
11H	0.75	0.88	0.68	15.71	8	10.9	6.0	6.80	9.10

Model	Standard Lateral	Max. Lateral	Allowable Sphere (in)	Net Eye Area (in^2)	No. of Vanes	Impeller Weight (lbs)		K Factor	
						Closed	Open	Closed	Open
11R	1.5	1.5	0.81	16.83	7	10.3	6.0	5.10	5.13
11LXL	2	2	0.68	15.70	5	12.3	N/A	10.30	N/A
11MXL	2	2	0.68	15.70	7	13.6	N/A	10.30	N/A
11HXL	2	2	0.68	15.70	8	12.5	N/A	11.40	N/A
12D	0.625	0.875	0.63	12.69	5	14.1	7.9	5.13	7.50
12E	0.625	0.875	0.50	12.69	8	14.6	8.0	6.60	9.50
12I	0.63	1	0.62	18.92	5	12.9	7.1	6.75	8.20
12K	0.63	1	0.62	18.92	8	14.4	7.6	6.50	7.75
12L	1.00	1.75	0.73	18.19	5	15.2	8.8	7.50	10.00
12M	1.00	1.75	0.73	18.19	7	15.2	8.9	7.40	10.00
12H	1.00	1.75	0.73	18.19	8	14.7	8.4	7.50	10.00
12R	0.75	1.50	0.75	32.39	6	10.5	6.4	16.50	19.00
12W	0.88	2.00	1.375	30.22	6	13.3	8.6	18.20	20.80
12X	0.75	2.00	1.375	30.22	6	13.5	9.6	16.20	17.40
12Z	0.90	1.25	0.67	38.33	7	19.8	10.9	14.00	20.00
13M	0.88	2.13	0.75	21.00	8	14.2	N/A	7.90	N/A
13YXL	2	3.25	0.91	30.83	8	25.4	N/A	20.30	N/A
14L	1.00	2.00	0.98	30.23	5	23.3	14.2	13.00	16.20
14M	1.00	2.00	0.98	30.23	7	23.6	13.8	13.00	16.20
14H	1.00	2.00	0.98	30.23	8	23.5	14.0	13.00	16.20
14LXL	2.00	4.00	0.98	30.22	5	26.8	N/A	13.00	N/A
14MXL	2.00	2.25	0.98	30.22	6	26.8	N/A	13.00	N/A
14HXL	2.00	2.25	0.98	30.22	8	27.4	N/A	13.00	N/A
14W	1.00	2.25	1.18	35.06	7	36.4	13.8	16.00	24.00
14Y			0.92	39.34	8	36.6	N/A	45.00	N/A
14YXL	2.25	4.00	0.92	39.34	8	36.6	N/A	45.00	N/A
15W	1.75	2.75	1.44	67.12	6	31.5	20.8	30.00	45.00
16M	0.75	2.25	0.72	40.37	7	62.0	N/A		N/A
18M	0.90	2.51	1.00	48.54	7	53.4	N/A		N/A

Submersible Motor Cooling Flow Rate

$$V = \frac{GPM\ x\ 0.408}{(W_{ID})^2 - (M_{OD})^2}$$

where:

V = Velocity

GPM = Flow rate in Gallons Per Minute

W_{ID} = Well casing Inside Diameter

M_{OD} = Motor Outside Diameter

At the maximum motor operating temperature of 86°F, the minimum Velocity of flow past the motor is:

0.25 ft/s for a 4" motor diameter

0.50ft/s for a 6" and larger motor diameter

- If the Velocity of flow past the motor is less than the values shown above, then the motor must be installed in a flow sleeve.
- If the temperature of the water is greater than 86°F, the flow rate past the motor should not be less than 3.0 ft/s.

The Horsepower required for a submersible motor increases when water temperature is above 86°F. This can be calculated using a Heat Factor multiplier, as shown below.

Table 19 - HP Required for Submersible Motor if Water Temp is above 86°F

Maximum Water Temperature	< 5HP	5-30HP	> 30HP
140°F	1.25	1.62	2.00
131°F	1.11	1.32	1.62
122°F	1.00	1.14	1.32
113°F	1.00	1.00	1.14
104°F	1.00	1.00	1.00
95°F	1.00	1.00	1.00

To calculate horsepower required when the water temperature is above 86°F, use the following equation and insert the Heat Factor multiplier.

$$HP_{Required} = P_{HP}\ x\ HF$$

where:

$HP_{Required}$ = Horsepower Required

P_{HP} = Pump Horsepower

HF = Heat Factor multiplier

Cable Selection for Single and Three Phase Motors

Table 20 - Cable Selection for Single Phase Motor

Single Phase, 60Hz (Service Entrance to Motor) - Values are Maximum Length in Feet														
Two or Three Wire Cable														
		AWG Copper Wire Size												
	HP	14	12	10	8	6	4	3	2	1	0	00	000	0000
230V Single Phase	1/2	130	210	340	540	840	1300	1610	1960	2390	2910	3540	4210	5060
	3/4	100	160	250	390	620	960	1190	1460	1780	2160	2630	3140	3770
	1	250	400	630	990	1540	2380	2960	3610	4410	5360	6520		
	1-1/2	190	310	480	770	1200	1870	2320	2850	3500	4280	5240		
	2	150	250	390	620	970	1530	1910	2360	2930	3620	4480		
	3	120*	190	300	470	750	1190	1490	1850	2320	2890	3610		
	5	0	0	180*	280	450	710	890	1110	1390	1740	2170	2680	
	7-1/2	0	0	0	200*	310	490	610	750	930	1140	1410	1720	
	10	0	0	0	0	250*	390	490	600	750	930	1160	1430	1760
	15	0	0	0	0	170*	270*	340	430	530	660	820	1020	1260

- Lengths without an asterisk meet the U.S. National Electrical Code ampacity for either individual conductors or jacketed 60°C cable.
- Length marked *with* an asterisk meet the NEC ampacity only for individual conductor 60°C cable in free air or water, not in conduit. If cable rated other than 60°C is used, lengths remain unchanged, but the minimum size acceptable for each rating must be based on the NEC table column for that temperature cable.
- Flat molded cable is considered jacketed cable.
- Maximum lengths shown maintain motor voltage at 95% of service entrance voltage, running at maximum nameplate amperes. If service entrance voltage will be at least motor nameplate voltage under normal load conditions, 50% additional length is permissible for all sizes.
- This table is based on copper wire. If aluminum wire is to be used, it must be two sizes larger.
- The portion of total cable length which is between the supply and single phase control box with line contactor should not exceed 25% of the maximum allowable length to ensure reliable contactor operation. Single phase control boxes without line contactors may be connected at any point in the total cable length.
- Lengths represent a 5% voltage drop. If 3% is required, multiply by 0.6 for maximum length.

Table 21 - Cable Selection for Three Phase Motor

Three Phase, 60Hz (Service Entrance to Motor) - Values are Maximum Length in Feet														
Three Wire Cable														
		AWG Copper Wire Size												
	HP	14	12	10	8	6	4	3	2	1	0	00	000	0000
230V Three Phase	1-1/2	420	670	1060	1670	2610	4050	5030	6160	7530	9170	0	0	0
	2	320	510	810	1280	2010	3130	3890	4770	5860	7170	8780	0	0
	3	240	390	620	990	1540	2400	2980	3660	4480	5470	6690	8020	9680
	5	140	230	370	590	920	1430	1790	2190	2690	3290	4030	4850	5870
	7-1/2	0	160*	260	420	650	1020	1270	1560	1920	2340	2870	3440	4160
	10	0	0	190*	310	490	760	950	1170	1440	1760	2160	2610	3160
	15	0	0	0	210*	330	520	650	800	980	1200	1470	1780	2150
	20	0	0	0	0	250*	400	500	610	760	930	1140	1380	1680
	25	0	0	0	0	0	320*	400	500	610	750	920	1120	1360
	30	0	0	0	0	0	260*	330*	410*	510	620	760	930	1130
460V Three Phase	1-1/2	1700	2710	4270	6730									
	2	1300	2070	3270	5150	8050								
	3	1000	1600	2520	3970	6200								
	5	590	950	1500	2360	3700	5750							
	7-1/2	420	680	1070	1690	2640	4100	5100	6260	7680				
	10	310	500	790	1250	1960	3050	3800	4680	5750	7050			
	15		340*	540	850	1340	2090	2600	3200	3930	4810	5900	7110	
	20			410*	650	1030	1610	2000	2470	3040	3730	4580	5530	
	25				530*	830	1300	1620	1990	2450	3010	3700	4470	5430
	30				430*	680	1070	1330	1640	2030	2490	3060	3700	4500
	40					500*	790	980	1210	1490	1830	2250	2710	3290
	50						640*	800	980	1210	1480	1810	2190	2650
	60						540*	670*	830*	1020	1250	150	1850	2240
	75								680*	840*	1030	1260	1520	1850
	100									620*	760*	940*	1130	1380
	125											740*	890*	1000*
	150												760	920*
	175													810*
	200													

Cable Splicing

First select correct cable based on motor rating and length required. Then follow the steps below

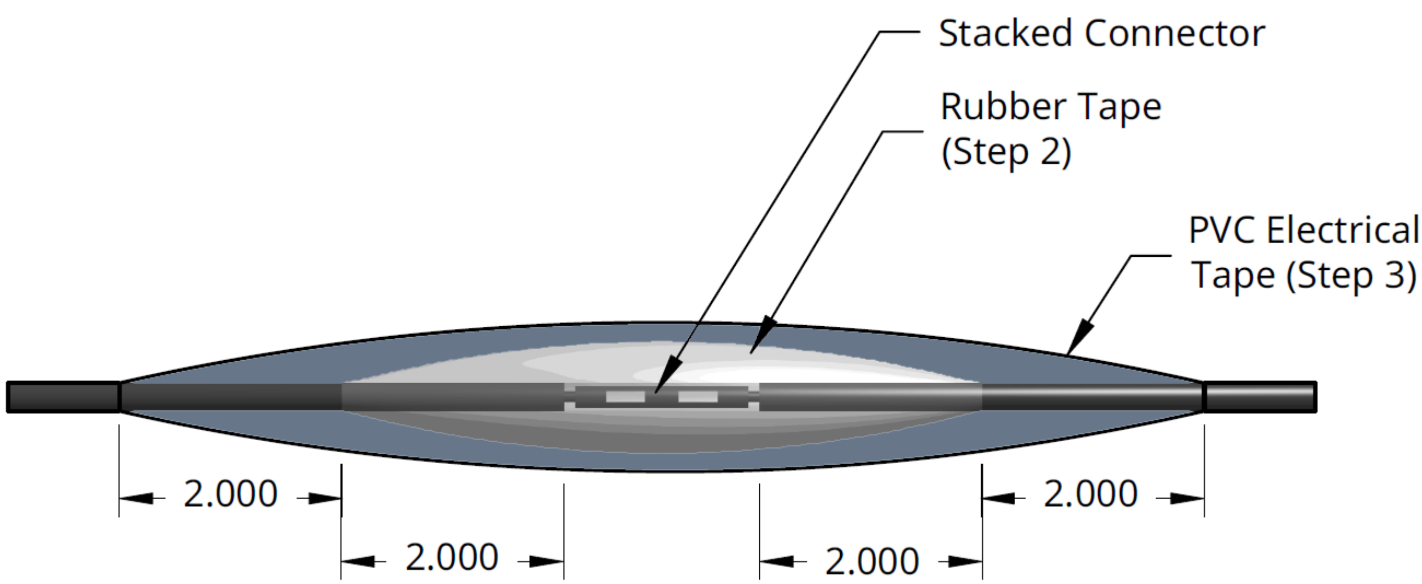
600V Tape Splicing

1. Strip individual conductor insulation only as far as necessary to provide room for a stake type connector. Tubular connectors of the staked type are preferred. If the connector O.D. is not as large as the cable insulation, build-up with rubber electrical tape.
2. Tape the individual joints with rubber electrical tape, using two layers: the first extending two inches beyond each end of the conductor insulation end, the second layer two inches beyond the ends of the first layer. Wrap tightly, eliminating air spaces as much as possible.
3. Tape over the rubber electrical tape with #33 Scotch electrical tape or equivalent, using two layers as in step "B" and making each layer overlap the end of the preceding layer by at least two inches.

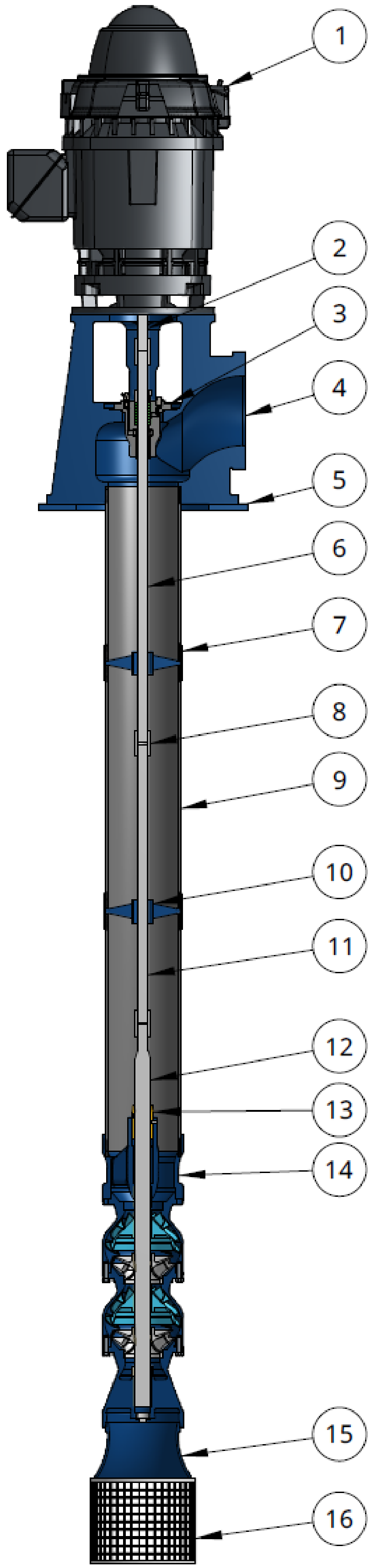
   Note:

   - In the case of a cable with three conductors encased in a single outer sheath, tape the individual conductors as described, staggering joints.
   - Total thickness of tape should be no less than the thickness of the conductor insulation.

Figure 14: Cable Splicing

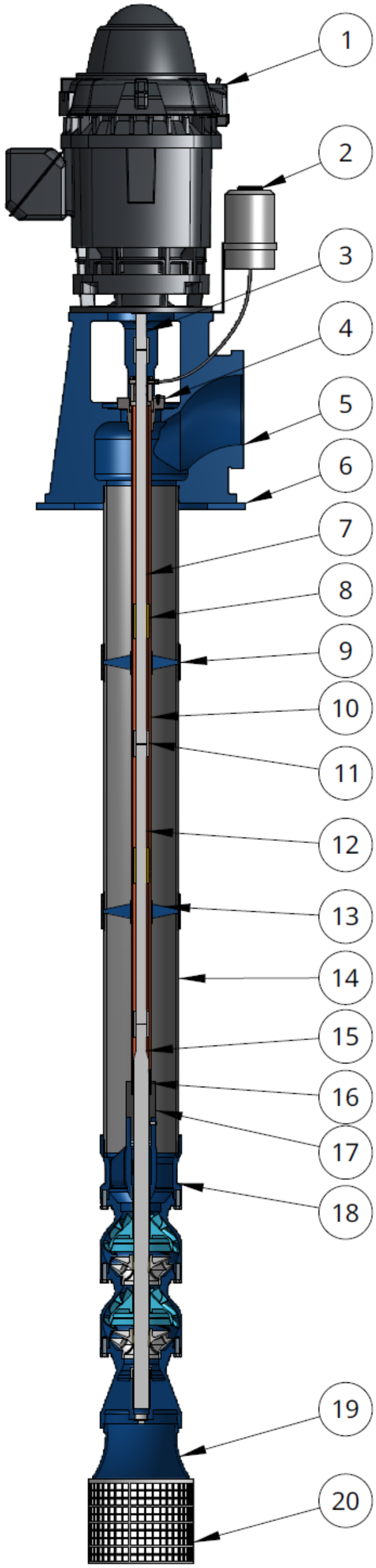


Typical Pump Assembly Layout – *Open Lineshaft*



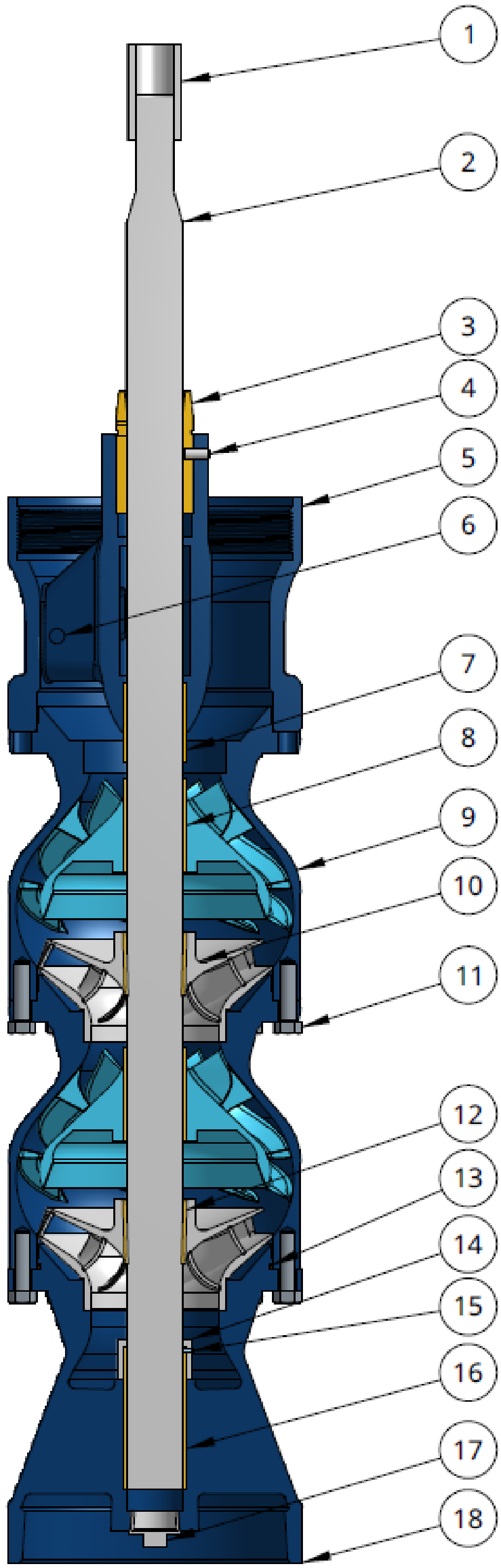
Item No.	Description
1	Vertical Hollowshaft Motor
2	Headshaft
3	Stuffing Box Assembly
4	Discharge Head
5	Foundation Plate
6	Lineshaft - Top
7	Column Coupling
8	Shaft Coupling
9	Column Pipe
10	Spider
11	Lineshaft - Intermediate
12	Bowl Shaft
13	Bearing - Discharge Case Upper
14	Bowl Assembly
15	Suction Bell
16	Strainer

Typical Pump Assembly Layout – *Enclosed Lineshaft*



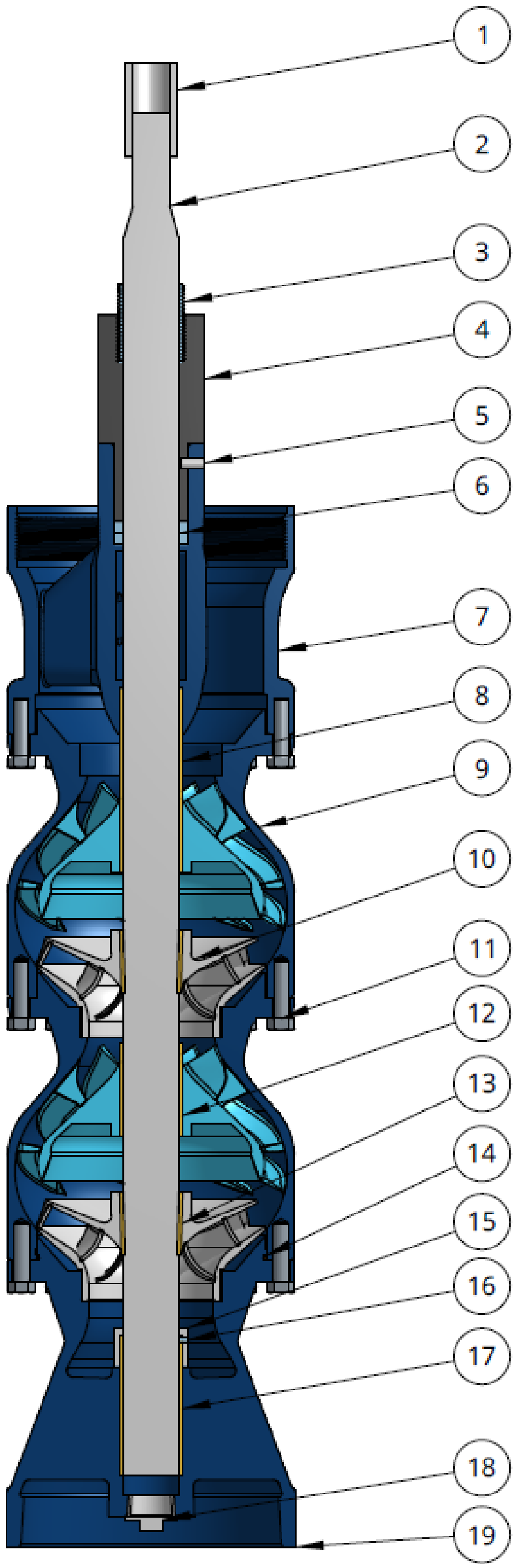
Item No.	Description
1	Vertical Hollowshaft Motor
2	Oil Reservoir
3	Headshaft
4	Oil Tensioner Assembly
5	Discharge Head
6	Foundation Plate
7	Lineshaft - Top
8	Lineshaft Bearing
9	Column Coupling
10	Oil Tube
11	Shaft Coupling
12	Lineshaft - Intermediate
13	Spider
14	Column Pipe
15	Bowl Shaft
16	Bearing - Lineshaft
17	Inner Column Adapter
18	Bowl Assembly
19	Suction Bell
20	Strainer

Typical Bowl Assembly Layout – *Open Lineshaft*



Item No.	Description
1	Lineshaft Coupling
2	Bowl Shaft
3	Bearing - Discharge Case Upper
4	Set Screw - Discharge Case
5	Discharge Case
6	Plug - Discharge Case
7	Bearing - Discharge Case Lower
8	Bearing - Bowl
9	Bowl
10	Impeller - Enclosed or Semi Open
11	Capscrew
12	Collet
13	O-Ring
14	Sand Cap
15	Set Screw - Sand Cap
16	Bearing - Suction Case
17	Plug - Suction Case
18	Suction Case

Typical Bowl Assembly Layout – *Enclosed Lineshaft*



Item No.	Description
1	Lineshaft Coupling
2	Bowl Shaft
3	Bearing - Lineshaft
4	Inner Column Adapter
5	Set Screw - Discharge Case
6	Lip Seal
7	Discharge Case
8	Bearing - Discharge Case
9	Bowl
10	Impeller - Enclosed or Semi Open
11	Capscrew
12	Bearing - Bowl
13	Collet
14	O-Ring
15	Sand Cap
16	Set Screw - Sand Cap
17	Bearing - Suction Case
18	Plug - Suction Case
19	Suction Case