AMERICAN-MARSH PUMPS

480 Series Vertical Turbine

Engineering Information

VERTICAL TURBINE PUMPS

AMERICAN-MARSH PUMPS

ENGINEERING DATA

TURBINE TERMINOLOGY

1. DATUM OR GRADE - The elevation of the surface from which the pump is supported.

2. STATIC WATER LEVEL - The vertical distance from grade to the water level when no water is being drawn from the well.

3. DRAWDOWN - The distance between the static water level and the water level when pumping at required capacity.

4. PUMPING WATER LEVEL - The vertical distance from grade to water level when pumping at required capacity. Pumping water level equals static Water Level plus Drawdown.

5. SETTING - The distance from grade to the top of the pump bowl assembly.

6. FIELD PUMPING HEAD - Lift below discharge plus head above discharge plus friction losses in discharge line. This is the head for which the customer is responsible and does not include any losses within the pump.

7 COLUMN FRICTION LOSS - Head loss in the pump due to friction in the column assembly. Friction loss is measured in feet and is dependent upon column and shaft size and setting. See Turbine Column Friction Loss Table in catalog.

8. TDH (LAB. HEAD) - Total head which the pump bowl assembly must deliver at the given capacity. TDH equals Field Pumping Head plus Column Friction Loss.

9 LABORATORY EFFICIENCY - The efficiency of the bowl unit only. This value is read directly from the performance curve.

10. LABORATORY HORSEPOWER - The horsepower required by the bowls only to deliver a given capacity against Laboratory Head.

$$\text{LAB. HP} = \frac{\text{Capacity x TDH}}{\text{3960 x Laboratory Efficiency}}$$

11. SHAFT FRICTION LOSS - The horsepower required to turn the lineshaft in the bearings. See Mechanical Friction in Turbine Pump Line Shafts in catalog.

12. FIELD HORSEPOWER OR BRAKE HORSEPOWER - Sum of laboratory horsepower plus shaft loss (and the driver thrust bearing loss under certain conditions.)

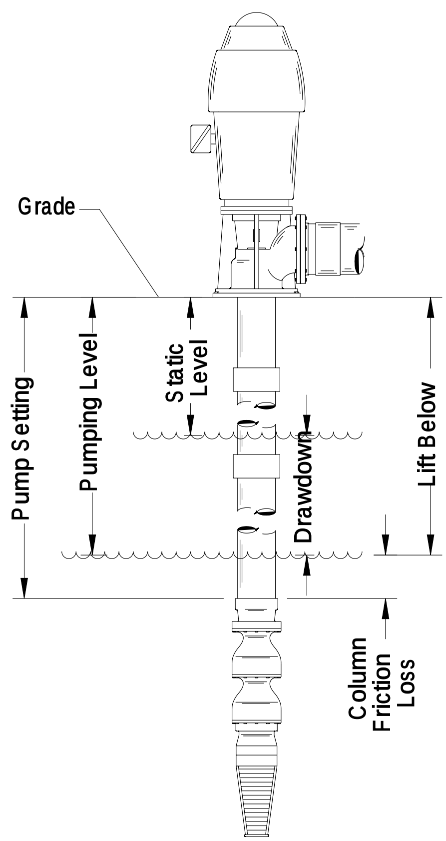
13. PUMP FIELD EFFICIENCY (WATER TO WATER) - The efficiency of the complete pump less the driver, with all losses between laboratory and field performance being taken into account.

$$\text{Field Efficiency} = \frac{\text{Capacity x Field Pumping Head}}{\text{3960 x Brake Horsepower}}$$

14. TOTAL PUMP THRUST - The sum of the weight of the shaft plus hydraulic thrust of the liquid being pumped. See Shaft Stretch Section in Catalog for shaft weight per foot. Performance curves give hydraulic "K" factor. Total thrust equals:

Shaft Wt. Per Foot x Setting in Feet + "K" x TDH

15. OVERALL EFFICIENCY (WIRE TO WATER) - The efficiency of the pump and motor complete. Overall efficiency = Pump Field Efficiency x Motor Efficiency.



NOT TO SCALE

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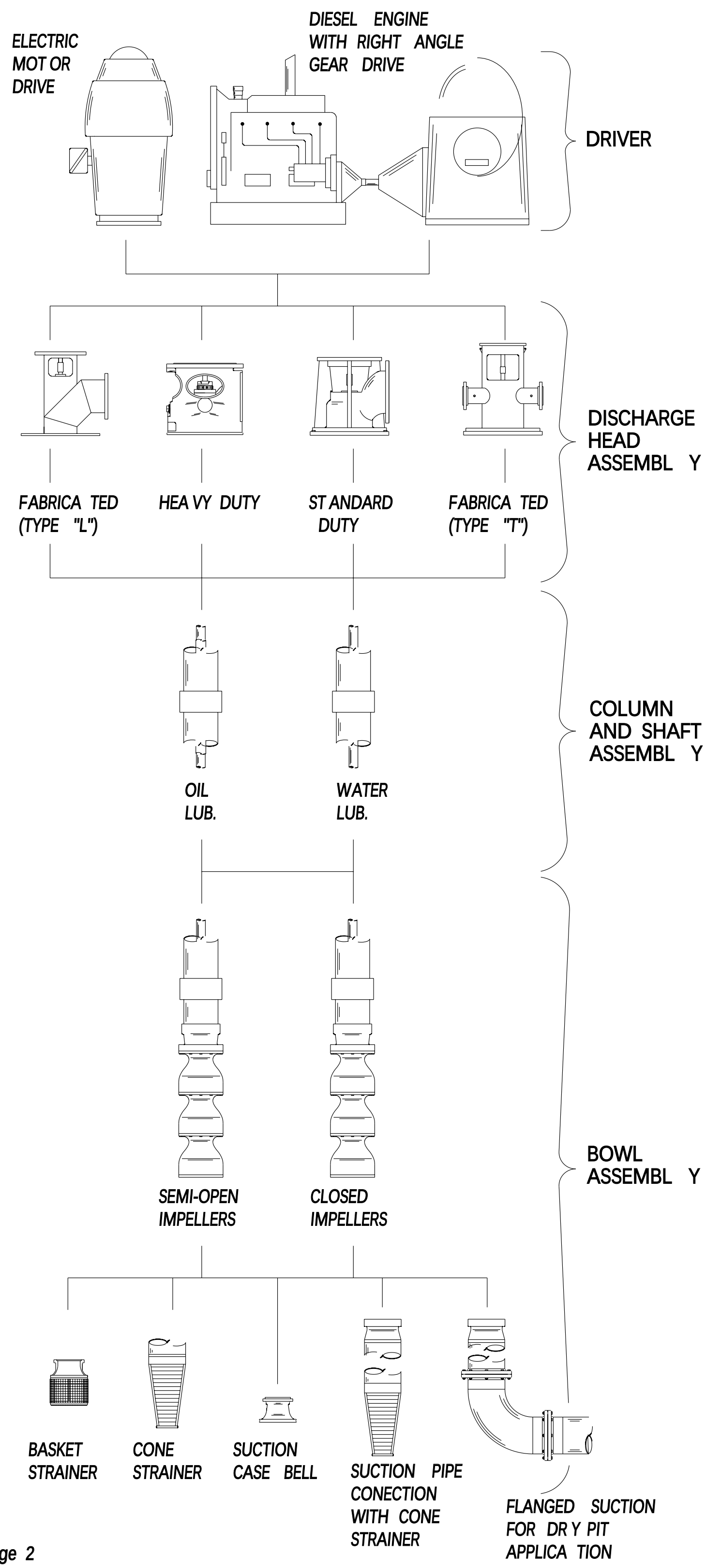
VERTICAL PUMP SELECTION GUIDE
WELL OR SUMP PUMP



WELL OR SUMP CONDITIONS

Pump application: ____________________
Inside diameter of surface casing or sump (in): ____________________
Total depth of well or sump (ft.): ____________________
Is casing stepped down?: ____ Where? (ft.): ____
Smaller (in.): ____ casing from (ft.): ____ to (ft.): ____
Screen set (ft.): ____ to (ft.): ____
and from (ft.): ____ to (ft.): ____
Sump water level: Min.: ____ Max.: ____
Well static water level (ft.): ____________________
Well pumping water level (ft.): ____________________

FLUID CONDITIONS

Fluid to be pumped: ____________________
Ph value: ____________________
Specific Gravity: ____ Temperature (°F): ____
Viscosity: ____________________
Foreign matter in fluid. (Describe): ____________________

PUMPING HYDRAULIC CONDITIONS

Capacity (USGPM): ____________________
Bowl head in feet (TDH): ____________________
Bowl setting (column length) (ft.): ____________________
Column lubrication required: (oil) or (water) .
Type impellers required: (semi-open) or (enclosed).
Bowl size: ____ No. of stages: ____
NOTE: Bowl head = elevation difference in feet between pumping liquid level in well or sump and pump discharge connection plus friction losses in column and discharge head plus system head requirements.

DRIVER

Size: ____ HP: ____ Type: (VHS) (VSS)
Speed of driver (RPM): ____ Ratio (if gear): ____
Non-reverse ratchet desired: ____________________
Current characteristics (ph): ____ (hz): ____ (volts): ____
Other driver characteristics: ____________________

SUCTION AND DISCHARGE COMPONENTS

Suction pipe size (in.): ____ Length (ft.): ____
Strainer (type): ____ (material): ____
Discharge flange size: ____ Column pipe size: ____
Discharge head connection: (standard) or (special)
(If special, describe fully): ____________________
Seal arrangement: Stretch Nipple Kit (oil lube): ____
Packing Housing Kit (water lube): ____
Mechanical seal: ____

VERTICAL PUMP SELECTION GUIDE CAN PUMP

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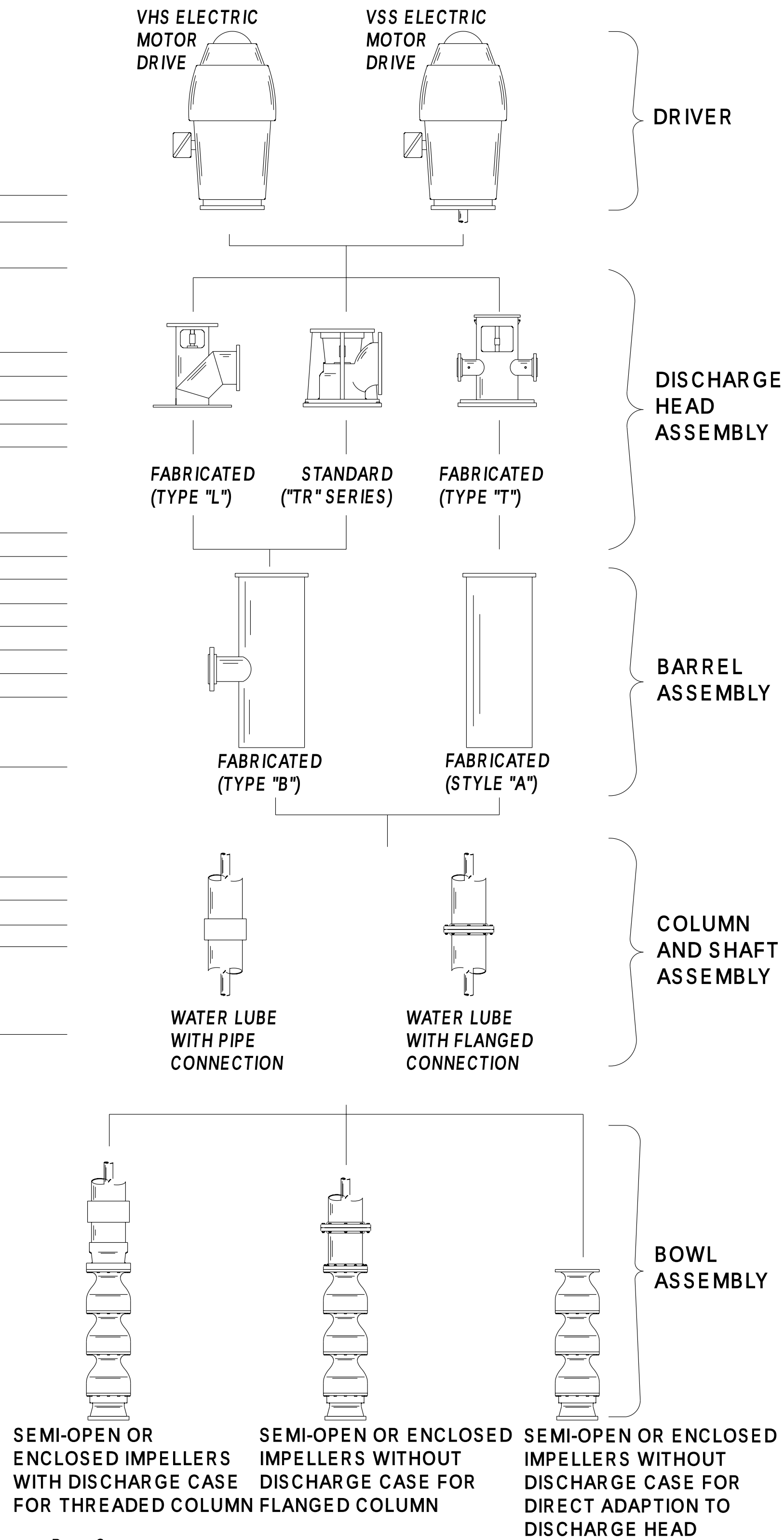

BARREL SIZING

Suspension pipe size (in): ____________________

Outside diameter of barrel (in): ____________________

Suction type: (Flanged) (Plain end) (Victaulic)

Total length of barrel (ft.): ____________________

FLUID CONDITIONS

Fluid to be pumped: ____________________

Ph value: ____________________

Specific Gravity: Temperature (°F): ____________________

Viscosity: ____________________

Foreign matter in fluid. (Describe): ____________________

PUMPING HYDRAULIC CONDITIONS

Discharge Pressure (ft.): ____________________

Suction Pressure (ft.): ____________________

Differential Pressure or Head (ft.): ____________________

Capacity (USGPM): ____________________

Pump differential head (ft.): ____________________

Column losses (ft.): ____________________

Suction barrel and discharge head losses (ft.): ____________________

Bowl head (differential head plus losses) (ft.): ____________________

Column lubrication: (product).

Type impellers required: (semi-open) or (enclosed) .

Bowl size: ________ No. of Stages: ____________________

DRIVER

Size: ______ HP: ______ Type: (VHS) (VSS)

Speed of driver (RPM): Ratio (if gear): ____________________

Non-reverse ratchet desired: ____________________

Current characteristics (ph): (hz): ____ (volts): ____ ____________________

Other driver characteristics: ____________________

SUCTION AND DISCHARGE COMPONENTS

Suction pipe size (in.): ____________________

Suction pipe type: (flanged) (plain end) (victaulic groove)

Discharge flange size (in.): ____ Rating: (150#) (300#)

Seal arrangement: (mechanical seal kit) (Packing Housing Kit)

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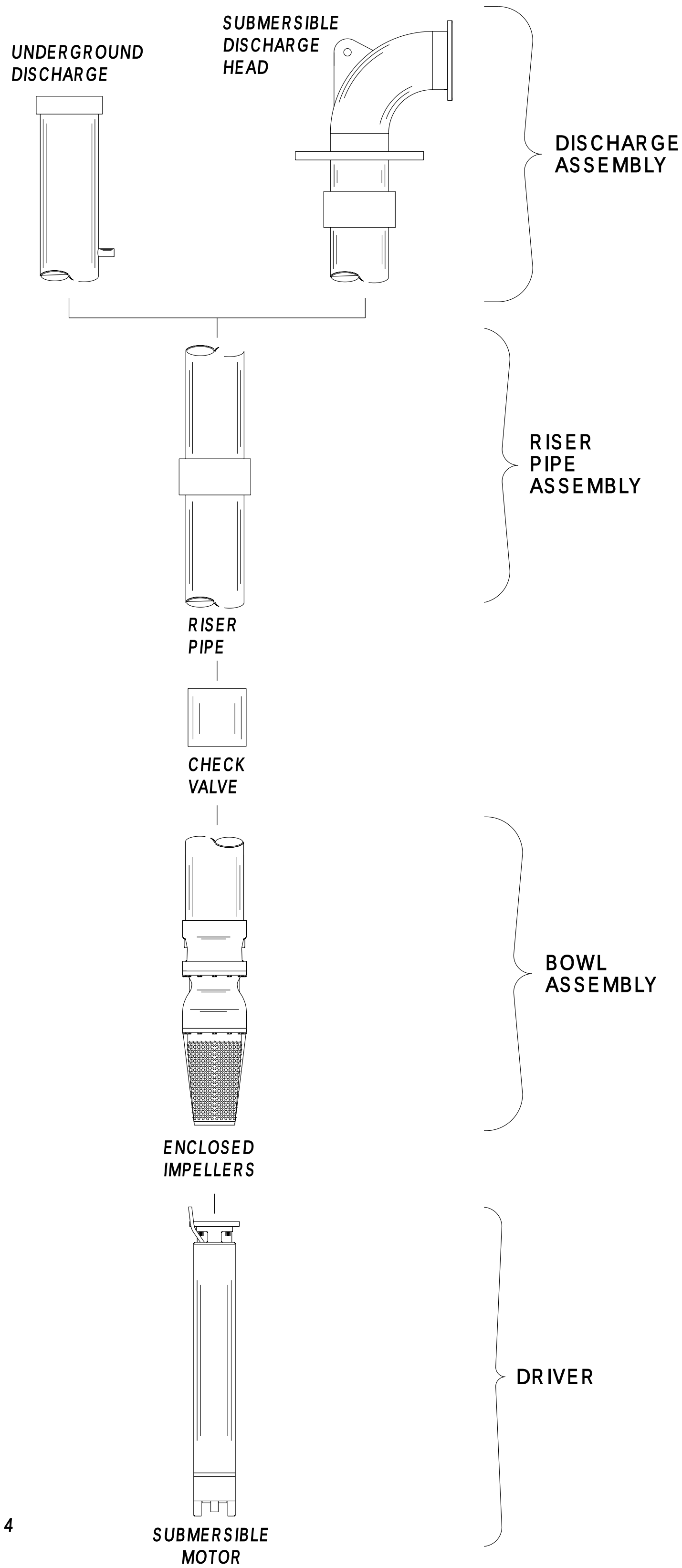
VERTICAL PUMP SELECTION GUIDE SUBMERSIBLE PUMP



WELL OR SUMP CONDITIONS

Inside diameter of surface casing or sump (in.): ________
Total depth of well or sump (ft.): ________
Is casing stepped down?: Where? (ft.): ________
Smaller (in.): casing from (ft.): ____to (ft.): ________
Screen set (ft.): to (ft.): ________
and from (ft.): to (ft.): ________
Sump water level (ft.): Min.: ________Max.: ________
Well static water level (ft.): ________
Well pumping water level (ft.): ________

FLUID CONDITIONS

Fluid to be pumped: ________
Ph value: ________
Specific Gravity: Temperature (°F): ________
Viscosity: ________
Foreign matter in fluid. (Describe): ________

PUMPING HYDRAULIC CONDITIONS

Capacity (USGPM): ________
Bowl head in feet (TDH): ________
Bowl setting (column length) (ft.): ________
Type impellers required: (Enclosed only offered on Submersibles. Contact factory for semi-open impeller application.)
Bowl size: No. of stages: ________
NOTE: Bowl head = elevation difference in feet between pumping level in well or sump and pump discharge connection plus friction losses in column and discharge head plus system head requirements.)

DRIVER

Motor Diameter: HP: ________
Speed of driver (RPM): Full load amps: ________
Current characteristics (ph/): (hz): (volts): ________
Min. velocity past motor for cooling (ft/sec): ________
Other driver characteristics: ________

SUCTION AND DISCHARGE COMPONENTS

Discharge flange type: ________
Riser pipe size: ________
Column check valve: (yes) (no) Size: ________
Electrical cable size:________Length: ________

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VERTICAL PUMP SELECTION GUIDE
LOW LIFT (Mixed Flow/Axial Flow) PUMP

WELL OR SUMP CONDITIONS

Inside diameter of surface casing or sump (in): ______
Total depth of well or sump (ft.): ______
Sump water level: Min.: Max.: ______

FLUID CONDITIONS

Fluid to be pumped: ______
If water, is it clear, raw, river, brackish?: ______
Specific Gravity: Temperature (°F): ______
Foreign matter in fluid. (Describe): ______
"Discharge Pressure" or head (ft.): ______

PUMPING HYDRAULIC CONDITIONS

Capacity (USGPM): ______
Continuous flow: Min.: Max.: ______
Bowl head in feet (TDH): ______
Bowl setting (column length) (ft.): ______
Column lubrication preferred: (oil) or (water) .
Bowl size: Type: (mixed flow) (axial flow)

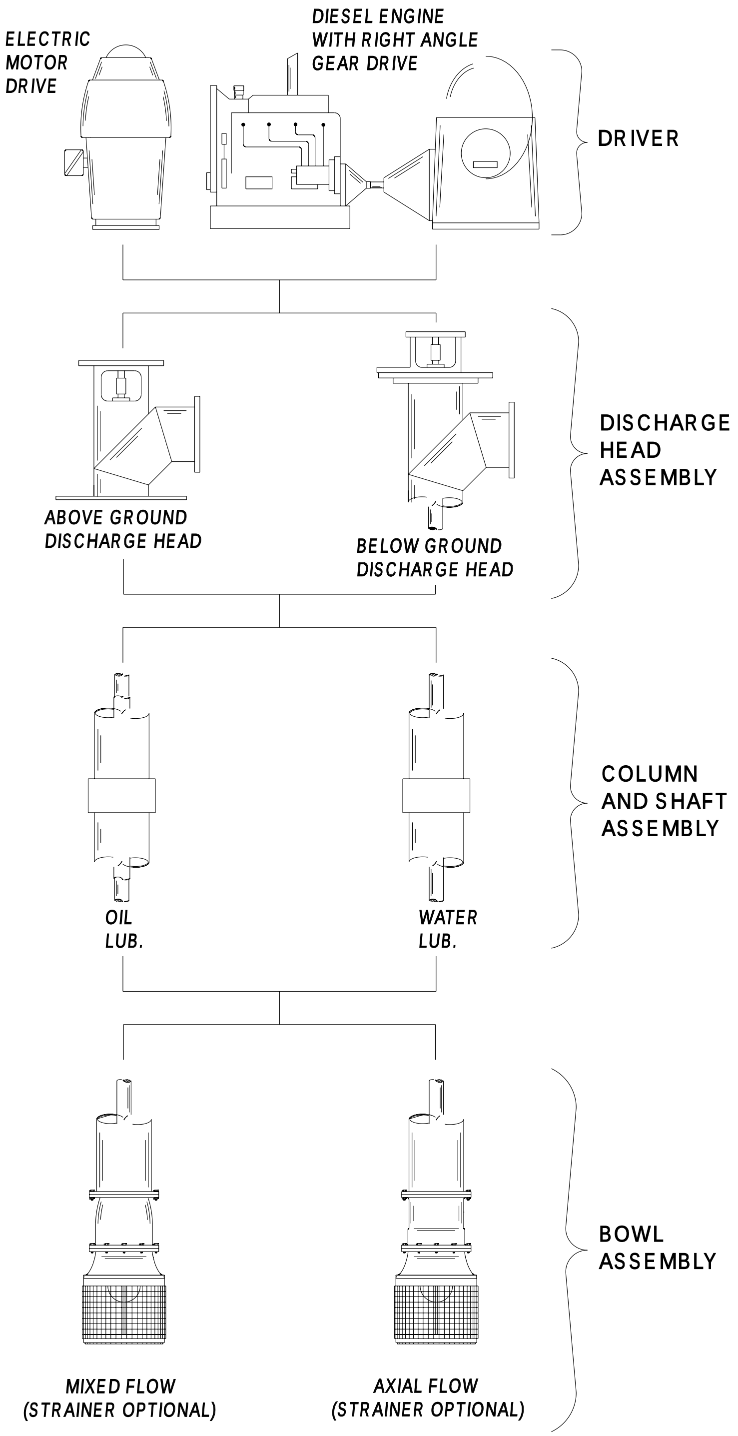
NOTE: Bowl head = elevation difference in feet between pumping liquid level in well or sump and pump discharge connection plus friction losses in column and discharge head plus system head requirements. Velocity head must be added by user to determine system total head.

DRIVER

Type: ______
Speed of driver (RPM): Ratio (if gear): ______
Non-reverse ratchet desired: ______
Current characteristics (ph): (hz): (volts): ______
Other driver characteristics: ______

SUCTION AND DISCHARGE COMPONENTS

Strainer (type): (material): ______
Discharge size: ______
Discharge type: (150# flange) (plain end) (victaulic groove)



INTAKE DESIGN

NOTE:

The following sump design recommendations are based on the Hydraulic Institute Standards 14th Edition 1983. These recommendations are not to be considered exact as there are many design considerations to evaluate in arriving at a properly designed sump which do not appear in this section.

GENERAL

The function of the intake is to supply an evenly distributed flow of water to the pump suction bell or suction case. An uneven flow is characterized by strong local currents, favors the formation of vortices, and under certain submergence conditions will result in the introduction of air into the pump with a resulting loss of pump performance, accompanied by noise, also uneven distribution can increase or decrease the power consumption with a change in total dynamic head. There can be vortices which do not appear on the surface, and these also may have adverse effects.

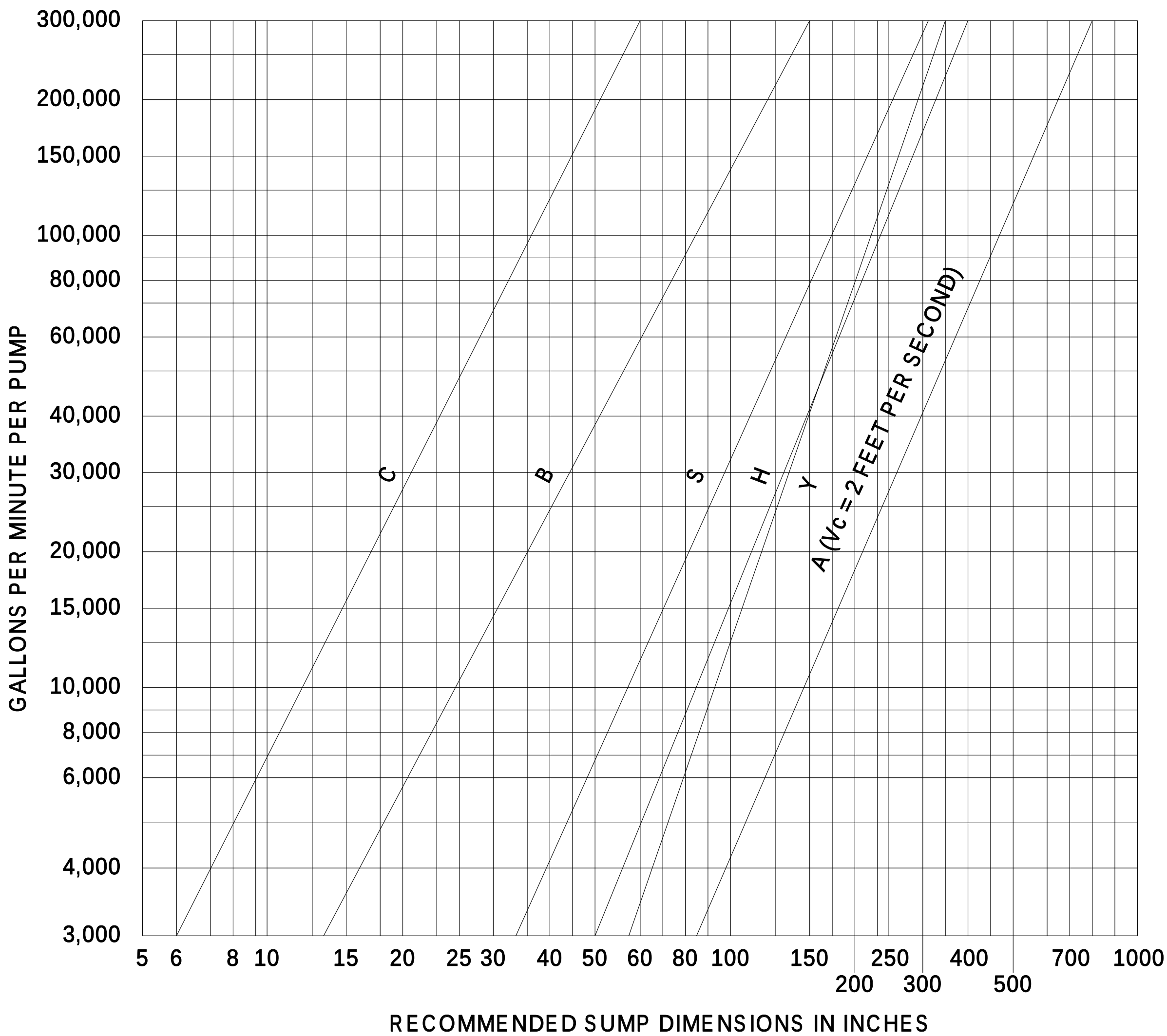
Uneven velocity distribution leads to rotation of portions of the mass of water about a centerline called vortex motion. This centerline may also be moving. Uneven distribution of flow is caused by the geometry of the intake and the manner in which water is introduced into the intake from the primary source.

Calculated low average velocity is not always a proper basis for judging the excellence of an intake. High local velocities in currents and in swirls may be present in intakes which have very low average velocity. Indeed, the uneven distribution which they represent occurs less in a higher velocity flow with sufficient turbulence to discourage the gradual built-up of a larger and larger vortex in any region. Numerous small surface eddies may be present without causing any trouble.

The ideal intake design is a direct channel going directly to the pump. Any turn or obstructions are detrimental since they may cause eddy currents and tend to initiate deep-cored vortices.

Water should not flow past one pump to reach another. If pumps must be placed in line of flow, it may be necessary to construct an open front cell around each pump or to put turning vanes under the pump to deflect the water upward. Streamlining should be used to reduce the trail of alternating vortices in the wake of the pump or of other obstructions in the stream flow.

FIGURE 1 SUMP DIMENSIONS VERSUS FLOW



AMERICAN-MARSH PUMPS

INTAKE DESIGN

The amount of submergence for a successful operation will depend greatly on the approaches to the intake and the size of the pump. While specific design is generally beyond the scope of the pump manufacture's responsibility, he may comment while the intake layout is still preliminary if he is provided with the necessary intake drawings reflecting the physical limitations of the site.

SUMP DIMENSIONS

Figures 1, 2 & 3 provide general sump dimension information for single and simple multiple pump arrangements. These guidelines cover pumps in the 3,000 to 300,000 GPM range. Additional information for smaller capacity pumps can be found at the end of this discussion. All of the dimensions in Figures 1, 2 & 3 are based on the rated capacity of the pump at design head. If the pump is to operate for significant periods at a higher capacity, the higher capacity should be used as the basis for determining dimensions.

Dimension C (distance from lip of suction bell to sump floor) is an average value based on the analysis of many pumps. The manufacturer should be consulted prior to determining a final value

Dimension B (distance between pump centerline and back wall) is a suggested maximum dimension. If the position of the back wall is dictated by other factors, it may be necessary to install a "false" wall for proper intake design.

Dimension S (minimum sump width for a single pump) can be increased, but if it is to be made smaller the manufacturer should be consulted or a model constructed to determine the adequacy of the design.

Dimension H (normal low water level) takes into account friction losses through the inlet screen and approach channel. The pump should be operated only momentarily or infrequently when the sump water level falls below this level.

NOTE: Minimum submergence (provided with bowl performance data) is normally specified as "Dimension H minus Dimension C".

Dimensions Y and A are recommended minimum values. These dimensions can be as large as desired but should be restricted to the limitations indicated on the curve. If the design does not include a screen, dimension A should be considerably longer. The screen or gate widths should not be substantially less than S, and heights should not be less than H. If the main stream velocity is more than two feet per second, it may be necessary to construct straightening vanes in the approach channel, increase dimension A, conduct a sump model test of the installation, or work out some combination of these factors.

On multiple pump installations the information presented in Figures 1, 2 & 3 applies, with the addition of the following factors:

Figure 4 (low velocity and straight line flow to all units) represents the recommended style of pit for multiple pump installations. Velocities in the pump area should approximate one foot per second, although with careful design velocities of two feet per second or higher may produce satisfactory results. Not recommended would be any design feature that would introduce eddying (such as an abrupt change in size of inlet pipe to sump or inlet on one side.)

FIGURE 2 SUMP DIMENSIONS VERSUS FLOW

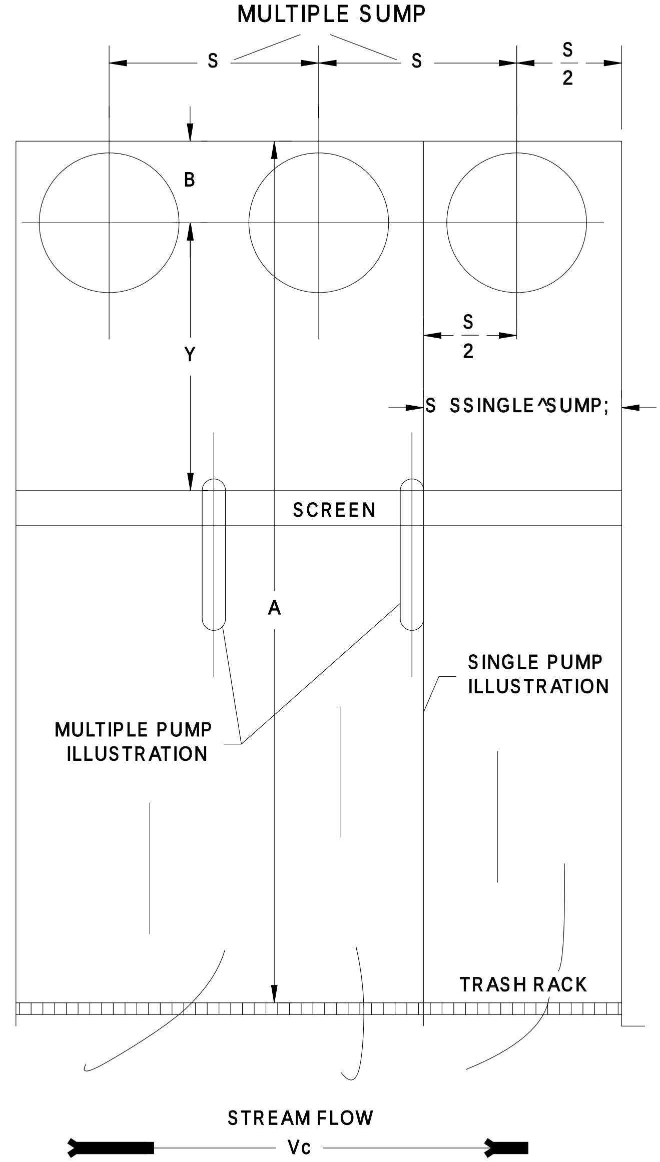
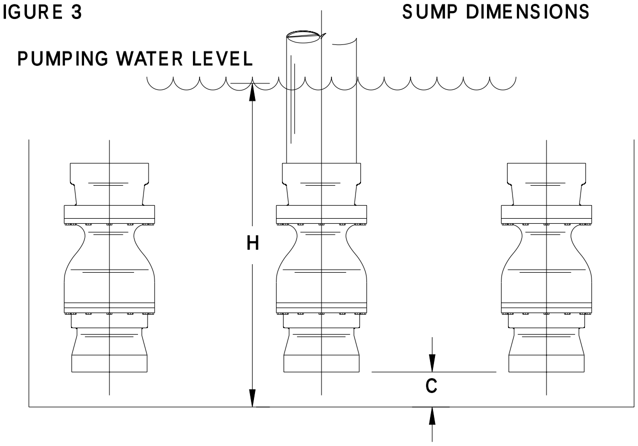
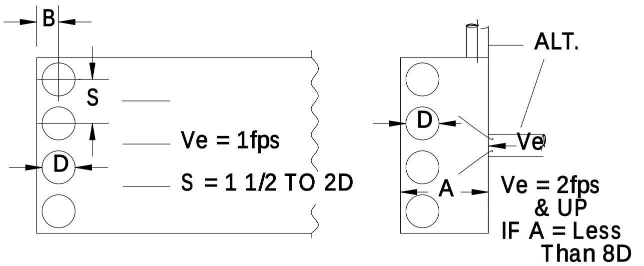


FIGURE 3 SUMP DIMENSIONS



INTAKE DESIGN

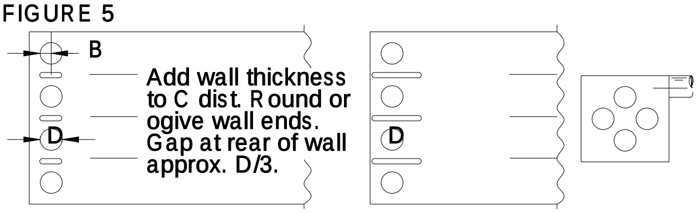
FIGURE 4



RECOMMENDED NOT RECOMMENDED

Figure 5 (multiple pumps in same sump) indicates best operation without separating walls. If all pumps are in operation at the same time, the use of separating walls will improve operation. If walls are to be used for structural purposes, and pumps will operate intermittently, flow space should be left behind each wall from the pit floor up to at least the minimum water level. The wall should not extend upstream beyond the rim of the suction bell. Pumps should NOT be placed around the edge of a sump either with or without dividing walls.

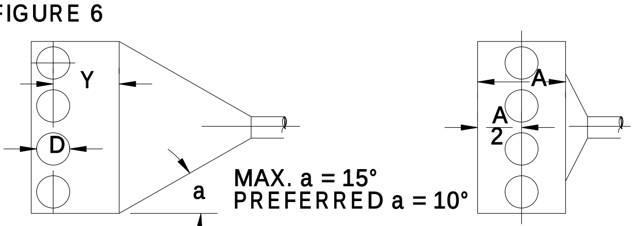
FIGURE 5



RECOMMENDED NOT RECOMMENDED

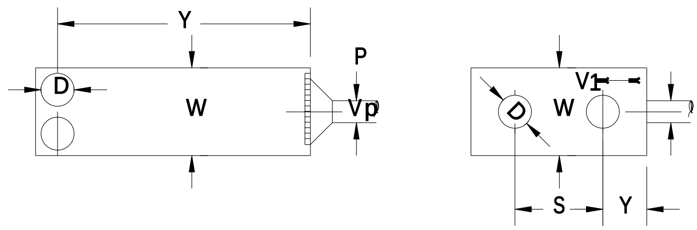
Figure 6 correctly shows a small pipe emptying into a large pump pit with a gradually increasing taper section. Abrupt changes in size are not desirable. The angle should be as large as possible, preferably not less than 45 degrees. Pit velocities should be kept at less than one foot per second.

FIGURE 6



RECOMMENDED NOT RECOMMENDED

Figure 7 demonstrates how an abrupt change from inlet pipe to pit con be accommodated. Pit velocities must be kept below one foot per second and the length must equal or exceed the values shown. As ratio W/P increases, the inlet velocity at P may be increased up to an allowed maximum of eight feet per second at W/P = 10. In line pumps are not recommended unless the ratio of pit to pump size is quite large, and pumps are separated by a generous margin longitudinally.



Baffles, grating or strainer should be introduced across inlet channel at beginning of maximum width section.

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

RECOMMENDED

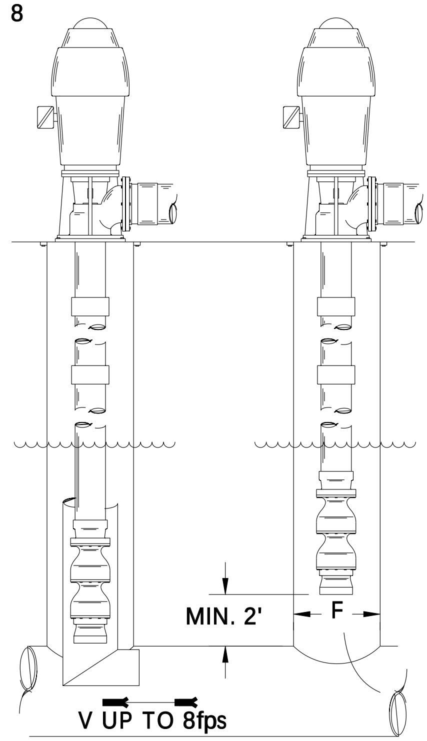
NOT Recommended Unless:

W = 5D or more, or
V1= 0.2fps or less AND
Y = Same as chart to left
S = is greater than 4D

NOT RECOMMENDED

Figure 8 illustrates installation in a tunnel or pipe line. The intake design must incorporate an inlet "ell" (suitable for flows up to 8 feet per second) or suction bell with the pump located at least two pipe diameters above the top of the tunnel. The tunnel must be free of air or it may be necessary to lower the scoop.

FIGURE 8



RECOMMENDED

NOTES:
1. The dimension D is generally the diameter of the suction bell measured at the inlet. This dimension may vary depending upon pump design. Contact factory for specific dimensions.
2. Figures apply to sumps for clear liquid. Contact factory for fluid/solids mixtures.

INTAKE DESIGN

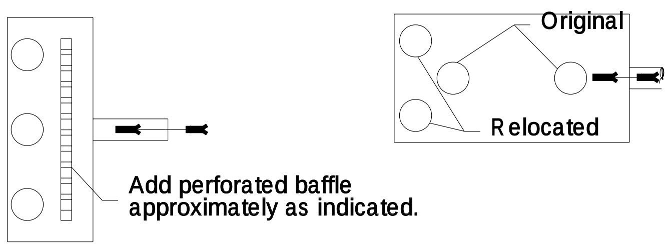
CORRECTION OF EXISTING SUMPS

Vortexing in pump suction pits is harmful to pit structures and the pumps themselves. While it is possible to eliminate sump problems in the design phases, it is much more difficult and often expensive to solve problems in existing sumps or modified sumps Sump model tests are recommended to test the effectiveness of proposed changes before construction begins. Figures 9 through 19 illustrate typical sump problems with possible solutions.

In Figure 9 the inlet velocity should be reduced by spreading the inflow over a larger area or by adding baffling to change its direction and speed. The baffles may be floor mounted extending above the minimum water level or ceiling mounted extending close to the floor.

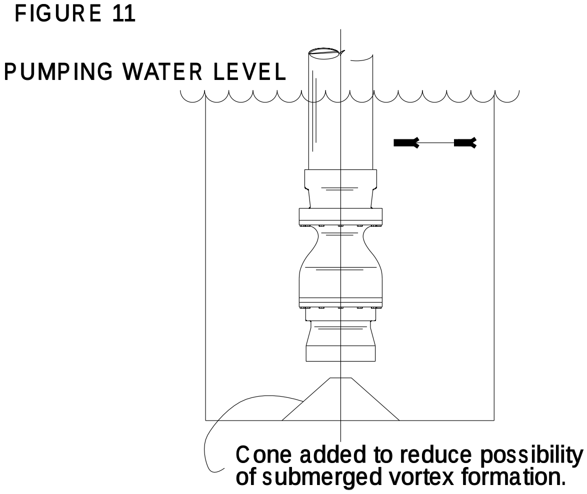
In Figure 10 the location of the pumps should be changed in relation to the direction of inflow.

FIGURE 9



In Figure 11 a cone is added to reduce the possibility of submerged vortex formation.

FIGURE 11



In Figures 12 and 13 the "no-flow" bays are modified by either breaking them open at the back and rounding the edges (Figure 12) or by removing them altogether (figure 13).

FIGURE 12

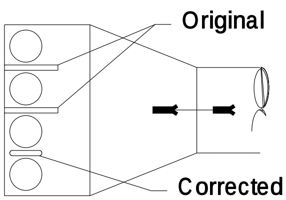
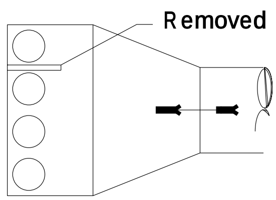


FIGURE 13



In Figure 14 sharp corners at gates, screens, etc., are eliminated to allow a smooth flow.

In Figure 15 velocity and vortexing is reduced by adding a bell extension suction plate and splitter to the suction bell. The splitter is attached in line with the flow.

FIGURE 15

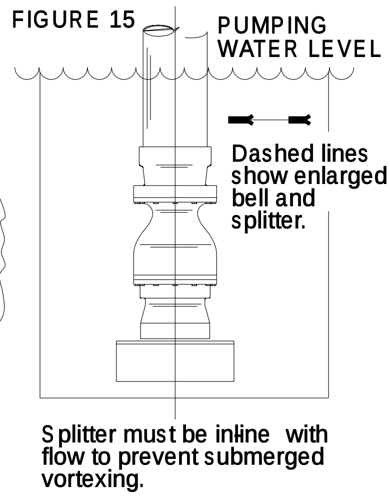


FIGURE 14

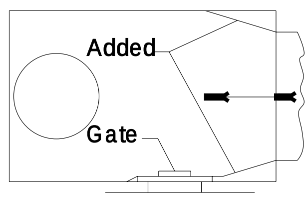
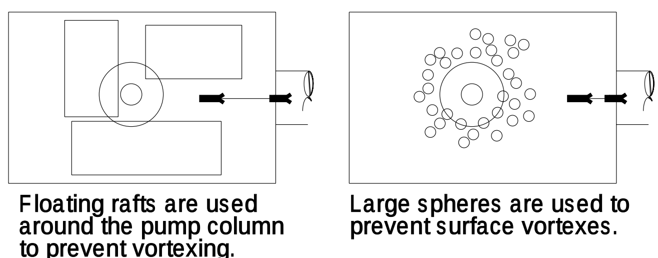


Figure 16 shows the use of floating rafts around the pump column to prevent surface vortices.

Figure 17 shows the use of floating large spheres to prevent surface vortices.

FIGURE 16



In Figure 18 the clearance between the back wall and the pump was reduced to improve the velocity pattern to the pump and to reduce the possibility of vortex formation.

In Figure 19 the direction of inlet flow was changed gradually by means of parallel turning vanes.

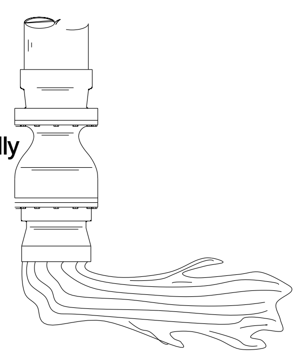
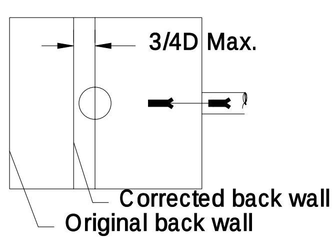
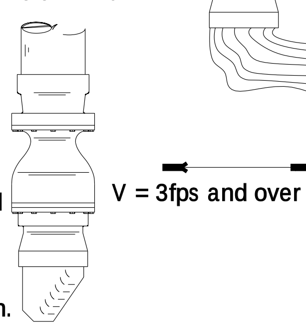


FIGURE 18



The velocity pattern to the pump can often be improved to reduce the possibility of vortex formation.

FIGURE 19



MAXIMUM BEARING SPACING ON OPEN LINESHAFT PUMPS

SPEED (rpm)		SHAFT DIAMETER (INCHES)						
		3/4	1	1 3/16	1 1/2	1 11/16	1 15/16	2 3/16
880	R			120	120	120	120	120
	SB			60	84	90	96	100
1200	R	120	120	120	120	120	120	120
	SB	60	60	60	72	78	84	88
1460	R	120	120	120	120	120	120	120
	SB	42	48	54	66	66	72	76
1760	R	120	120	120	120	120	120	120
	SB	42	48	48	60	60	66	70
2900	R	60	60	60	60	60	60	60
	SB	30	36	36	42	48	48	52
3600	R	60	60	60	60	60	60	60
	SB	30	30	36	36	42	48	48

Note: R - Standard Rubber Bearing
SB - Solid Bearing (brass, carbon, graphite, teflon, etc.)

The above chart is to aid in the selection of lineshaft bearing spacing for pump column on vertical turbine pumps.

These recommendations are based on manufacture standards and first critical frequency shaft calculation.

Solid type bearings tend to have a slight running noise.

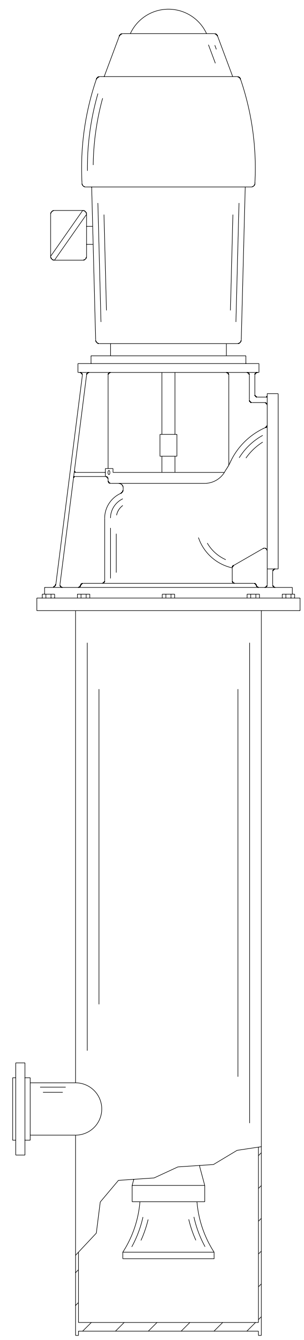
SUCTION BARREL SELECTION

Recommended capacity for a suction barrel can be determined by calculation of the fluid velocity. Recommended flow past the largest portion of the pump should not exceed 5 ft/sec. (V.)

TO CALCULATE:

$$V = \frac{GPM \times Q}{(D_1)^2 - (d_2)^2}$$

where: GPM - amount of flow required
- Q - constant 0.4085 (cubic feet per sec)
- D_1 - I.D. of barrel in inches
- d_2 - O.D. of bowl in inches



EXAMPLE:

12 3/4" steel suction barrel with a wall thickness of 0.375 would have an I.D. of 12"

10MC bowl with an O.D. of 9 1/2"

Flow design of 500 GPM

TO CALCULATE:

$$V = \frac{GPM \times Q}{(D_1)^2 - (d_2)^2}$$

$$V = \frac{500 \times 0.4085}{(12)^2 - (9\ 1/2)^2} =$$

$$\frac{500 \times 0.4085}{144 - 90.25} =$$

$$\frac{204.25}{53.75} = 3.8 \text{ ft/sec.}$$

This indicates that this bowl will work in this size barrel.

1. NPSH should always be considered when placing any pump into a suction can.

2. The pump suction should be located (2) barrel diameters below barrel inlet or (2) suction diameters above the barrel inlet, never allow the suction inlet of the pump to be set in the area of the suction barrel inlet.

3. Suction inlet size should have an inlet velocity of equal to or less than 5 ft/sec.

4. If the suction of the barrel is below the suction inlet of the pump, the fluid velocity can be disregarded in barrel selections.

5. If there is sufficient NPSH and efficiency is not important, it is permissible to exceed 5 ft/sec. - the maximum recommended ever would be 9 ft/sec.

A SIMPLE METHOD OF SURVEYING A DEEP WELL

FIGURE NO. 1

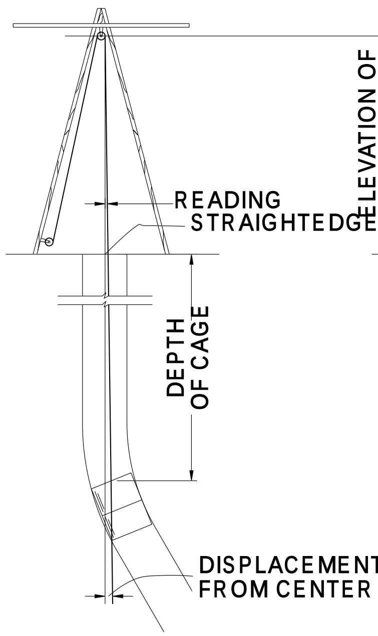


FIGURE NO. 2

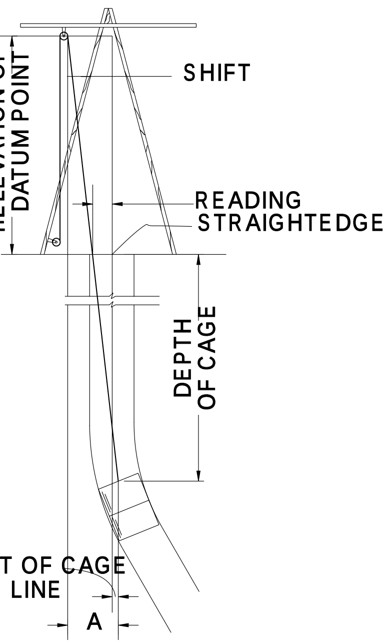


FIGURE NO. 3

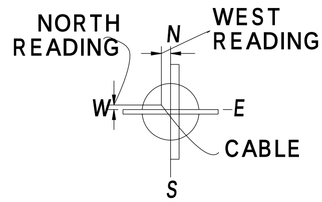
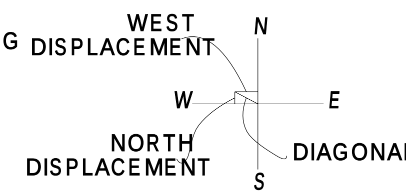


FIGURE NO. 4



(1) PURPOSE OF SURVEYING A WELL - It is advantageous to know whether or not a pump will fit in a well and operate normally. Since the pump column can be curved, within limits, without being detrimental to pump operation, a well is surveyed to find out in what directions and how sharply it curves throughout its length.

(2) EQUIPMENT NECESSARY - The equipment to be used is listed below.

(a) A reel of small steel cable long enough to reach to the desired depth.

(b) A cage about two feet long with a diameter about 1/4" smaller than the I.D. of the well casing.

(c) A small pulley for the steel cable, having a frame which can be bolted to a board.

(3) EQUIPMENT SET-UP - Attach the pulley to a board thick enough to carry the weight of the cage and line without bending. Support the board horizontally at least ten feet above the top of the well casing. The board and supports should be arranged so that the location of the pulley can be shifted at least two feet in any horizontal direction by sliding the board. A derrick is the most convenient support, but if there is no derrick a tripod or other structure must be constructed.

Run the cable through the pulley and attach the end to the center of the cage. Hang the cage slightly above the well casing and move the pulley until the axis of the cage is in line with the center of the well.

Next, establish with a straightedge four marks on the well casing or floor which can be used to determine two horizontal lines at right angles to each other, passing as near to the center of the well as the cable will permit without the cable being deflected from its center position. To simplify the procedure, one of these lines should be parallel to the board carrying the pulley. Usually the lines are laid off as North-South and East-West lines.

(4) CAGE POSITION READINGS - Lower the cage ten feet at a time and measure deflections in the North, South, East or West directions from the center, as shown in Figure No. 3, at each ten-foot interval. The vertical distance of the center of the pulley above the level of the place where deflection measurements are made must also be measured. The center of the pulley is called the datum point.

Figure No. 1 shows an elevation of a well with a cage in the well, and Figure No. 3 shows a plan view of the top of the well with a straightedge in its two positions. In making the measurements indicated, it must be remembered that the straightedge was located by the side of the cable instead of the center. Since the cable was in the center of the well, the straightedge does not lie exactly on a diameter and deflections must always be measured on the same sides of the cable that determined the North-South and East-West lines.

(5) SHIFTING THE DATUM POINT - The cable must not be touching the side of the well casing when readings are taken. By refering to Figure No. 1 it is evident that if the well is crooked the cable can touch the well casing when the cage is lowered past a bend or spiral. When the cable touches the casing, any readings beyond this point are valueless. It is necessary to shift the datum point so that the cable will not touch the casing. If the crook is above the water level, this condition can be observed. If it is below the water level, it may be suspected if the location of the cable remains constant as the cage is lowered. It occasionally happens, however, that the well is actually slanting or curving so that it will give a constant reading for a while and, since shifting the pulley is apt to introduce errors unless great care is used, it is best to be sure that the cable is touching before any shifts are made. If the well is so crooked that the cable touches at a second point after being shifted, it is impossible to survey it beyond this point by using this method.

If the data is worked up and plotted, it is possible to see whether or not any of the above difficulties are present. Usually, a shift in one direction will be sufficient, but sometimes it is necessary to shift in the other direction also. The pulley should be shifted as far as the surveyor deems necessary and for every shift it is important that the position (vertically and in both directions horizontally) of the datum point be accurately determined. Figure No. 2 shows a well being surveyed with the datum point shifted.

(6) PLOTTING THE CAGE POSITIONS - After the data has been calculated it should be plotted on graph paper. The deflections for the true view are determined graphically by drawing the diagonals of parallelograms produced by plotting the displacement readings as shown in Figure No. 4. Cross-section paper with one-inch squares, divided into ten smaller squares to the inch, is convenient for laying out the well. If the depth is laid out with ten feet to the inch and the cage displacement with ten inches to the inch, it usually results in a satisfactory representation of the well. The horizontal scale should be large enough to show the defects in the well clearly, but at the same time it must be

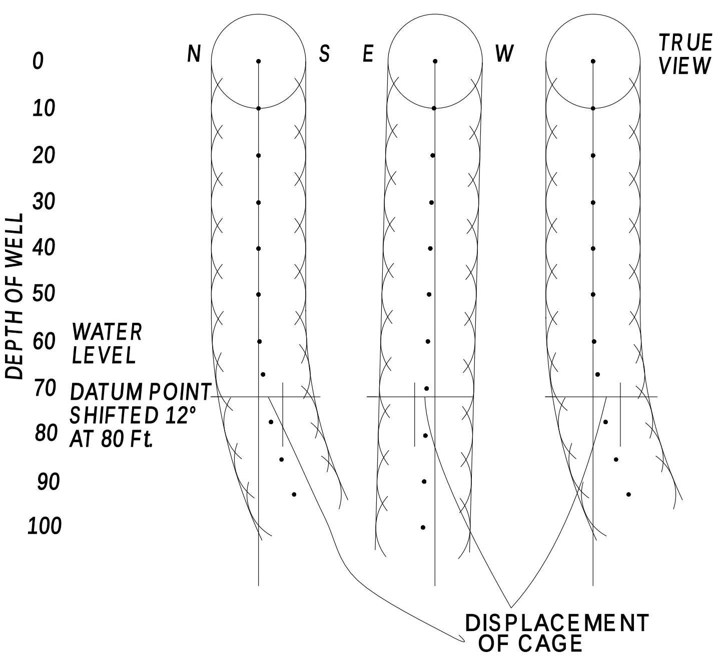
SAMPLE DATA, CALCULATIONS AND PLOT 18" WELL

recognized that unless the well diameter is plotted to exactly the same scale as the side deflection, the well plot will be misleading and of practically no value.

On the plot a straightedge can be used to represent the cable and to indicate whether or not the cable was touching the casing at any point.

If it is determined from the plot that the cable probably did not touch the well casing at any of the readings, during the survey, it is an indication that the readings were accurate. In order to determine what size of column pipe and bowl diameter can be safely operated in the well, construct from cardboard a cross-section of the column and bowl unit to the same scale as used in plotting the survey. If the model of column and bowl unit can be placed in the well without binding on the well casing, when inserted on the plot of the true view of the survey, it is a good indication that the actual pump will do likewise.

CENTER LINES INDICATE CENTER OF THE WELL AT ITS SURFACE



Elevation of datum point 20 ft. Water level 60 ft.
Datum point shifted 12" North at 80 ft.

Depth Cage Feet	*Readings of Deflections of Cable Inches*				*Displacement of Cage Inches*			
	N	S	E	W	N	S	E	W
10	0.10				0.15			
20	0.15		0.10		0.30		0.20	
30	0.20		0.20		0.50		0.50	
40	0.10		0.30		0.30		0.90	
50			0.35				1.25	
60		0.50	0.37			0.20	1.48	
70		1.10	0.40			6.75	1.80	
80	8.00		0.40			8.00	2.00	
90	7.80		0.42			11.10	2.30	
100	7.50		0.44			15.00	2.64	

FORMULA

No. 1 *Displacement of cage from center line =*

$$\frac{\text{Reading} \times (\text{depth of cage} + \text{El. of Datum})}{\text{El. of Datum}}$$

Sample Calculation at 10 ft. depth:

$$\text{Cage displacement} = \frac{0.10 \times (10 + 20)}{20} = .15"$$

No. 2 *Displacement of cage from center line when datum point is shifted =*

$$^{*}(\text{shift} = \text{reading}) \times \frac{(\text{Cage Depth} + \text{El. of Datum})}{\text{El. of Datum}} - \text{shift}$$

**Add the reading to the shift when shift and reading are on opposite sides of center line: Subtract when shift and reading are on same side of center line.*

Sample Calculation at 80 ft. depth:

Cage diplacement =

$$(12 - 8) \times \frac{(80 + 20)}{20} - 12 = 8"$$

Since distance A (see Figure No. 2) is greater than the shift, it is evident that the cage displacement is south from the center line.

Formula No. 1 is used in all EAST calculations in this case, but if there is a shift in the East-West direction, Formula No. 2 would have to be used in the same manner as for the North-South direction.

PRELUBRICATION RECOMMENDATION FOR WATER LUBRICATED OPEN SHAFT TURBINE PUMPS

AMERICAN-MARSH PUMPS

Open line shaft pumps utilize neoprene line shaft bearings which must be kept wet when the unit is operating. After the pump liquid fills the column pipe, the bearings are kept lubricated by this liquid. At start-up and shut-down, however, certain precautions must be taken to provide lubrication to these bearings.

START-UP: Under normal conditions if the static water level is 30 feet of less, prelubrication is not required since the bearings will hold enough moisture to provide initial lubrication. On deeper settings, however, it is necessary that the bearings receive prelubrication as outlined below.

SHUT-DOWN: Non-Reverse Ratchet mechanisms mounted in the driver are recommended on units with settings of 50 feet or more. The Non-Reverse Ratchet prevents reverse rotation due to backflow, thus eliminating post lubrication requirements. If Non-reverse ratchets are not employed, post lubrication must be provided using methods similar to those outlined below.

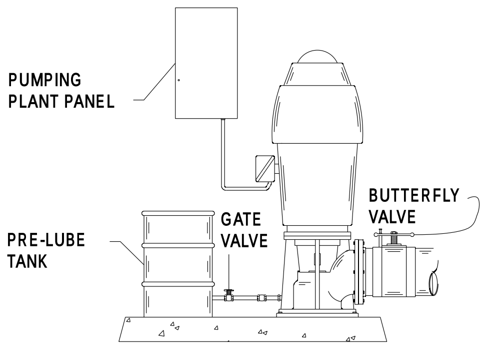


FIGURE 1

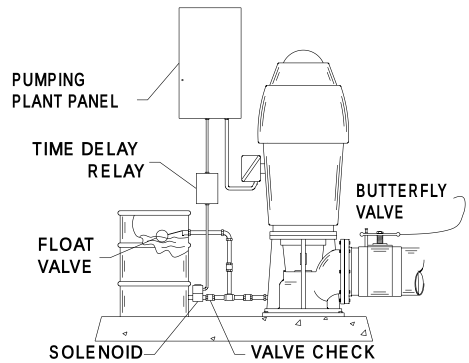


FIGURE 2

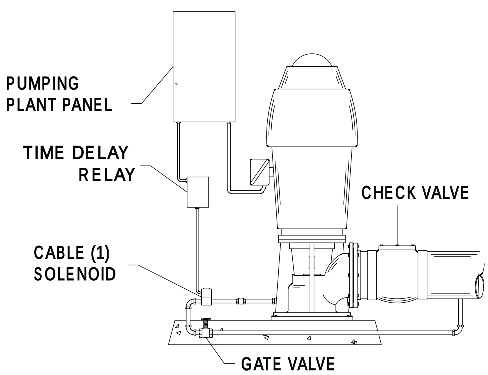


FIGURE 3

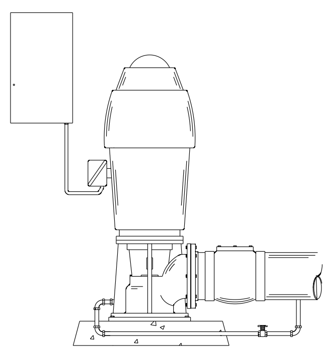


FIGURE 4

SOLENOID VALVE AND FITTINGS

PRESSURE ON SOLENOID VALVE	OUTER COLUMN SIZE (INCHES)		
	5" SMALLER	6" AND 8"	10" AND LARGER
	SOLENOID VALVE AND FITTING SIZE INCHES		
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"

TIME DELAY RELAY

STATIC WATER LEVEL (FEET)	TIME DELAY (MINUTES)
0'-30'	1/2 min.
31'-70'	1 min.
71'-150'	1-1/2 min.
151'-250'	2-1/2 min.
251'-350'	3-1/2 min.
351'-450'	

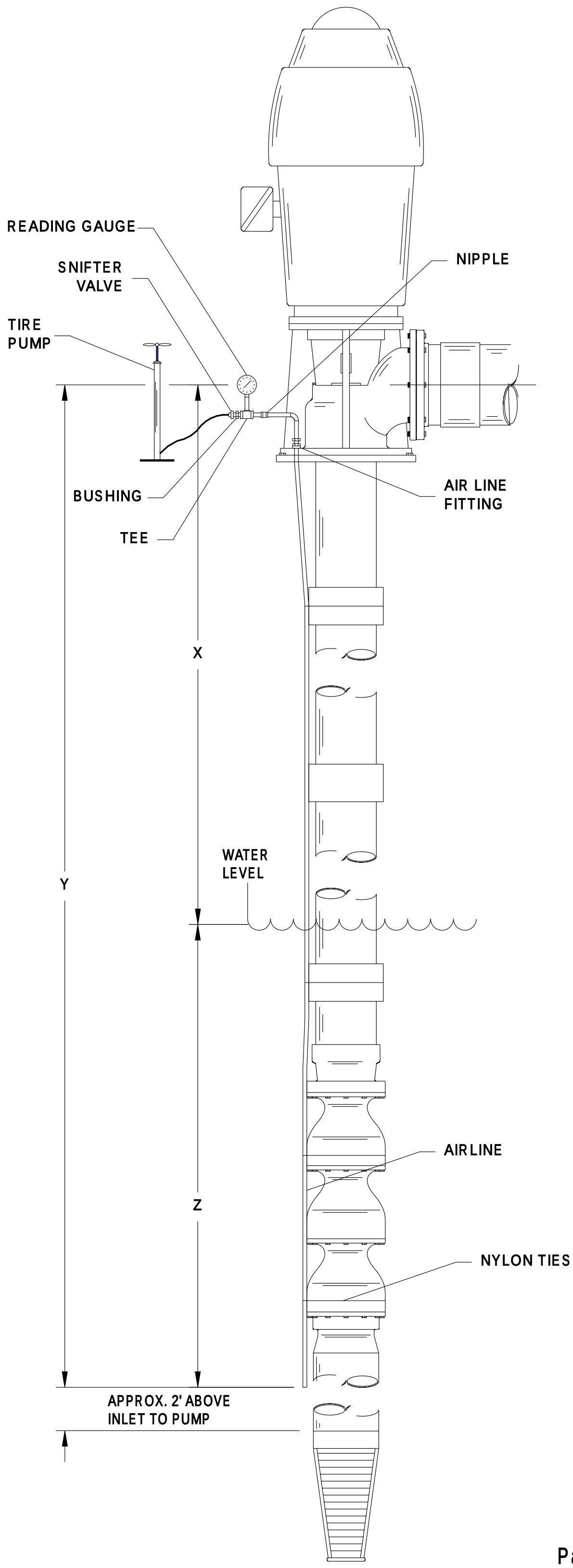
TANK & FITTING SIZES

OUTER COLUMN SIZE	SIZE					
	FITTINGS 1"	TANK 50 GAL.	FITTINGS 1-1/2"	TANK 100 GAL.	FITTINGS 2"	TANK 200 GAL.
	STATIC WATER LEVEL (FEET)					
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'	

NOTES:
TIME DELAY SETTING BASED ON PROPER SOLENOID SELECTION

0-30 FT. SETTING NO PRE-LUBRICATION REQUIRED.

WATER LEVEL TESTING



There are two commonly used methods to determine the water level in wells - airline and gauge, or an electric sounder.

AIRLINE METHOD:

The airline method can use a standard pressure gauge, indirect reading depth gauge, or direct reading depth gauge.

INSTALLATION: The airline is installed so that the lower end is approx. 2' from the inlet of the pump - for reliable readings the airline should extend 20' below low water level if possible. All airline joints must be air tight for proper operation. The upper end of the airline is connected to a gauge and snifter valve. Exact vertical length of the airline must be recorded at installation, by noting on the face of the gauge. Use as many nylon ties as necessary to firmly attach the airline to the column and bowl assembly without collapsing the airline.

METHOD OF OPERATION: A tire pump is used to expel all water from the airline, when this point is reached the gauge reading will remain constant. The maximum maintained pressure is equal to the height of water above the end of the airline (DIM Z).

INDIRECT READING DEPTH GAUGE (FIXED DIAL): Pump up airline until maximum pressure (all water is expelled from airline) is reached, reading on gauge will be distance "Z". Water level (below surface) is obtained by subtracting "Z" from "Y" (W.L. = Y - Z).

DIRECT READING DEPTH GAUGE (MOVABLE DIAL): Set the movable gauge dial so that the length of airline (Y) is at the pin stop (gauge pointer position at 0 pressure). Pump airline to maximum pressure, gauge will read water level (Y - Z) direct.

PRESSURE GAUGE: A pressure gauge can be used by converting PSI to feet of water as follows:

Feet of Water = PSI x 2.31

Operation would be identical to indirect reading gauge.

ELECTRIC SOUNDER METHOD

The electric sounder consists essentially of a battery, a spool of well insulated waterproof wire and a millivolt meter. One terminal of the battery is connected to the pump head and the other through the potentiometer to one end of the spool of wire. The other end of the wire from the spool must be protected so that it will not close the circuit, if it should bump against the pump in being lowered into the well, but at the same time so arranged that the circuit will be closed when the end of the wire contacts the water in the well. The wire from spool, then, is lowered into the well until; the needle of the potentiometer deflects, indicating that the water level has been reached and the contact closed. The wire is then properly marked, pulled from the well and measured with a steel tape to determine the water level. (It is possible to calibrate the spool of wire so that it is direct reading.)

X = Depth to water in feet (this is unknown).
Y = Length of air line in feet (this was measured at installation).
Z = Water pressure on airline, in feet head of water. Standard gauge reads in lb./sq. in. Multiply reading by 2.31 to convert to feet of water. Altitude type gauge reads directly in feet of water.

EXAMPLE:

X = Y - Z
Y = 100 ft.
Z = 15 lb./sq. in.
= 15 x 2.31 = 34.65 ft.

X = ?
X = 100 - 34.65
= 65.35 ft.

*If the effective length of the airline is not known, it may be determined by measuring the actual water level. When the pump is idle by some other method, and comparing it to the maximum reading obtained on the direct reading gauge or adding to the maximum reading of the indirect gauge.

AMERICAN-MARSH PUMPS

FORMULA FOR CHANGING PUMP SPEED CHANGES IN PERFORMANCE

Pump curves are generally based on standard motor speeds. For performance of pumps at speeds other than those published, it is necessary to calculate new Capacity, Head, and BHP.

The following "Affinity Laws" are used in speed variation calculations:

1. The capacity of a turbine pump varies in direct proportion to the speed.
2. The head of a turbine pump varies in proportion to the square of the speed.
3. The horsepower varies in proportion to the cube of the speed.

In general, it is good engineering practice not to increase the speed of a turbine pump designed for 1760 RPM to more than 2200 RPM. At higher RPM harmonic or shaft vibration may occur causing excessive wear in the pump.

FORMULAS:

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

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

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

Q = Quantity in GPM
H = Head in Feet
BHP = Brake Horsepower
N_1 = Speed in RPM (published)
N_2 = Speed in RPM (required)

NOTE: DO NOT EXCEED MINIMUM TRIM DIAMETER INDICATED ON STANDARD CATALOG CURVE.

EXAMPLE

An impeller operating at 1760 RPM is rated to deliver 900 GPM at 76 Feet of Head, and requires 20.3 BHP to drive it. What will the effect of changing the speed to 1460 RPM?

The direct ratio of speeds will be:	$\frac{1460}{1760}$ = 0.82955
1460 RPM CAPACITY will be:	0.82955 x 900 = 746.59 GPM
The square of the ratio will be:	0.82955 x 0.82955 = 0.68815
1460 RPM HEAD will be;	0.68815 x 76 = 52.30 Feet of Head
The cube of the ratio will be:	0.82955 x 0.82955 x 0.82955 = 0.57085
1460 RPM BRAKE HORSEPOWER will be:	0.57085 x 20.3 = 11.59 BHP

The efficiency is still another factor to be considered, but is not seriously altered for small changes in speed. If the above pump had an efficiency of 85% at 900 GPM when operating at 1760 RPM, the efficiency would still be approximately 85% at 746.59 GPM when operating at 1460 RPM.

MULTIPLIERS FOR PUMP PERFORMANCE AT VARIOUS SPEEDS USING 1760 RPM AS REFERENCE SPEED

RPM	GPM	HEAD	HP
1400	0.7955	0.6327	0.5033
1450	0.8239	0.6788	0.5592
1500	0.8523	0.7264	0.6191
1550	0.8807	0.7756	0.6831
1600	0.9091	0.8264	0.7513
1650	0.9375	0.8789	0.8240
1700	0.9659	0.9330	0.9012
1760	REFER TO PERFORMANCE CURVE		
1800	1.0227	1.0460	1.0697
1850	1.0511	1.1049	1.1614
1900	1.0795	1.1654	1.2581
1950	1.1080	1.2276	1.3601
2000	1.1364	1.2913	1.4674
2050	1.1648	1.3567	1.5802
2100	1.1932	1.4237	1.6987
2150	1.2216	1.4923	1.8230
2200	1.2500	1.5625	1.9531
2250	1.2784	1.6343	2.0893
2300	1.3068	1.7078	2.2317
2350	1.3352	1.7828	2.3805
2400	1.3636	1.8595	2.5357
2450	1.3920	1.9378	2.6975

RPM	GPM	HEAD	HP
2500	1.4205	2.0177	2.8660
2550	1.4489	2.0992	3.0415
2600	1.4773	2.1823	3.2239
2650	1.5057	2.2671	3.4135
2700	1.5341	2.3534	3.6104
2750	1.5625	2.4414	3.8147
2800	1.5909	2.5310	4.0266
2850	1.6193	2.6222	4.2462
2900	1.6477	2.7150	4.4736
2950	1.6761	2.8094	4.7090
3000	1.7045	2.9055	4.9525
3050	1.7330	3.0031	5.2043
3100	1.7614	3.1024	5.4645
3150	1.7898	3.2033	5.7332
3200	1.8182	3.3058	6.0105
3250	1.8466	3.4099	6.2967
3300	1.8750	3.5156	6.5918
3350	1.9034	3.6230	6.8960
3400	1.9318	3.7319	7.2094
3450	1.9602	3.8425	7.5322
3500	1.9886	3.9547	7.8644
3520	2.000	4.000	8.000

IMPELLER DIAMETER TRIM PROCEDURE
CHANGES IN PERFORMANCE

The effect of changing the outer diameter is to decrease the peripheral speed of the impeller which has exactly the same effect as reducing the rotative speed without altering the diameter. The effect is to change the Head generated in proportion to the square of the speed, or the square of the diameter, according to the fundamental formula:

$V^2 = 2GH$
G = Gravity (32.17 feet per second)
H = Head (in feet)

When the peripheral speed is changed, however, the velocity of the water flowing through the impeller is changed in direct proportion. Since this changes the quantity of water delivered, both changes must be considered when trimming an impeller.

There is still a third factor to be considered. Assuming there is no major change in the speed (with the quantity in direct proportion to the diameter, and the head in proportion to the square) the work done (or power required) will be as the product of the two which is proportional to the cube of the diameter.

EXAMPLE:
An impeller of 9.313 inch outside diameter (of vanes) is rated to deliver 900 GPM at 76 Feet of Head, and requires a driver of 20.3 BHP. (Data taken form a published curve.) What will be the effect of changing the diameter to 8.750 inches?

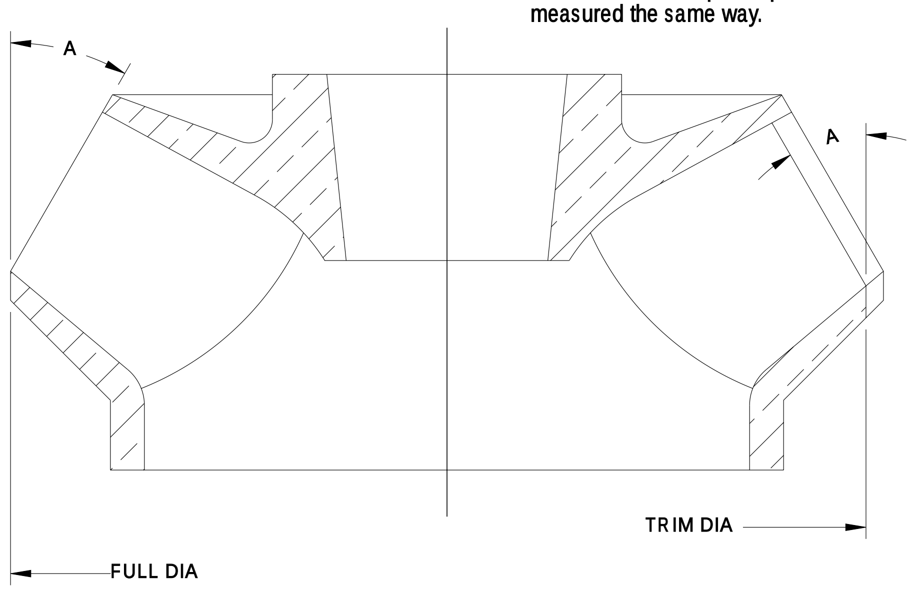
SOLUTION:
The direct ratio of diameters will be: 8.750 - 9.313 = 0.93955
So the new Quantity will be: 0.93955 x 900 = 845.59 GPM
The square of the ratio will be: 0.93955 x 0.93955 = 0.88275
So the new Head will be: 0.88275 x 76 = 67.09 Feet of Head
The cube of the ration will be: 0.93955 x 0.93955 x 0.93955 = 0.82938
So the new power will be: 0.82938 x 20.3 = 16.84 BHP

The efficiency is still another factor to be considered. It is not seriously altered for small changes in diameter. Refer to bowl performance curves for actual change in efficiency.

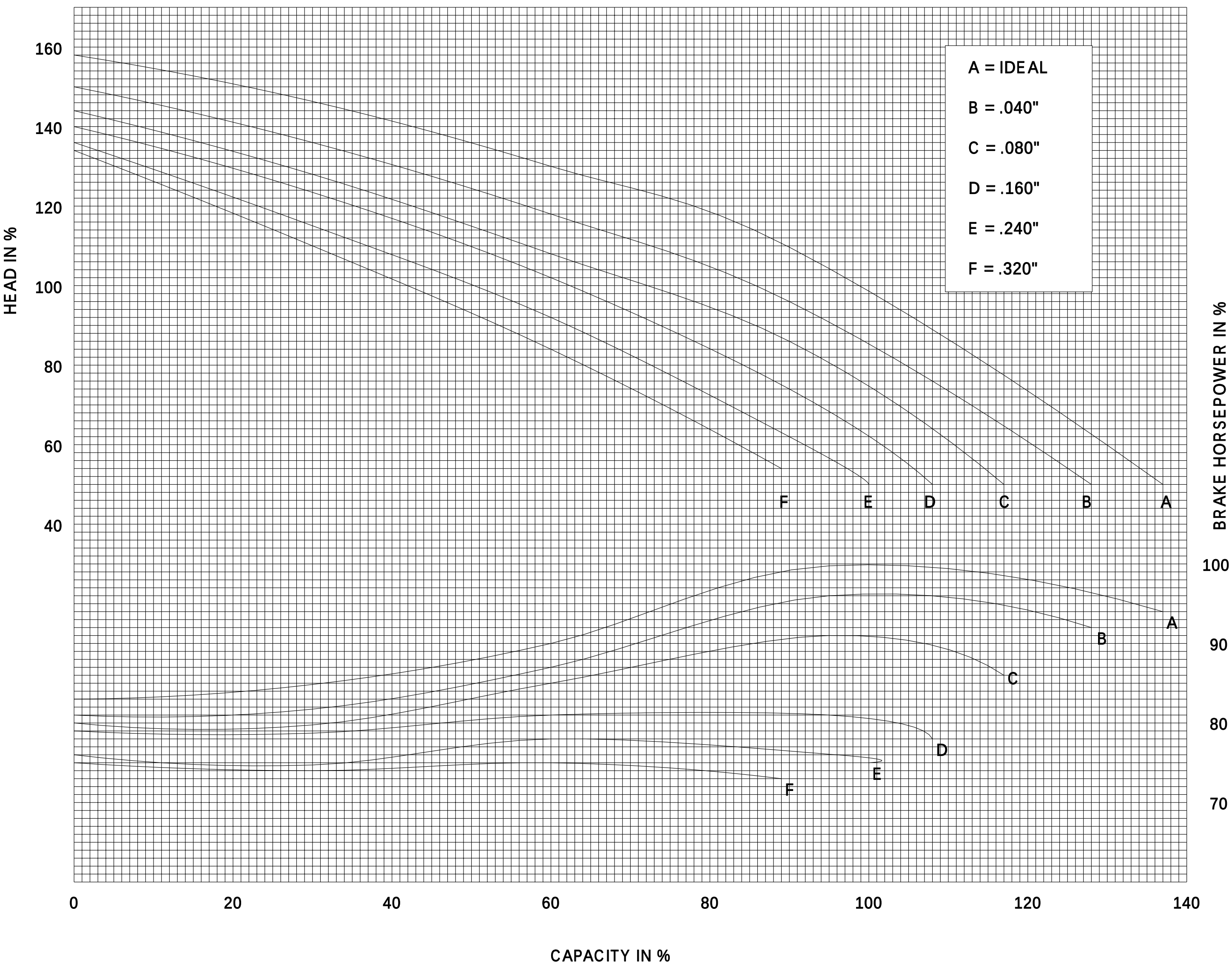
Since a pump is made of several bowls with impellers, it is only necessary to figure one impeller. The new head is multiplied by the number of stages for the whole pump. The quantity and efficiency, however, will be as calculated for the single impeller, as all will perform the same in series.

This procedure applies in the same way to open and enclosed impellers.

*All impellers require lower shroud removal for all trim diameters. Semi-open impeller trim diameters are measured the same way.



EFFECTS OF RAISING SEMI-OPEN IMPELLERS



The above chart indicates the approximate effect of raising semi-open impellers from their ideal (A) operating position. Raising the impellers increases the clearance between impeller and bowl seat and reduces the performance accordingly. The chart is general and will not be exactly correct for any particular pump model since each model will react differently. 100% head and capacity are to be taken as the head and capacity of the pump at peak efficiency. - EXAMPLE: If a particular pump delivers 250 GPM at 50' head at peak efficiency when the impellers are properly adjusted, raising the impellers 0.080" would reduce the capacity to approximately 181 GPM (72 1/2% of 250 GPM) while maintaining the 50' head - or conversely, the pump would deliver 250 GPM at 37 1/2' head (75% of 50'.) The horsepower would be about 91 1/2% of the previous horsepower.

VELOCITY HEAD

The Velocity Head (Head due to Velocity) of moving water at a given velocity is the equivalent Head through which it would have to fall to acquire the same velocity, or the Head necessary to accelerate water. The Velocity head must always be considered when accurate testing is required, but normally is such a negligible amount that its factor is a small value when figuring total Head conditions. It is important to consider Velocity Head when your Total Head values are low, which occurs in Axial Flow pumps or when Suction Lift valves are high, such as in centrifical pump applications.

The formula for Velocity Head is:

$hv = \frac{v^2}{2g}$ where "g" is the acceleration due to gravity (32.174 feet per second)

By knowing the head we can transpose the formula to read:

$v = \sqrt{2gh}$ this obtains the velocity.

Velocity can also be obtained by applying the values to these formulas.

$hv = 0.0155v^2$ where $v = \frac{0.4085 \times GPM}{D^2}$ or $v = \frac{0.321 \times GPM}{Area}$

$hv = \frac{0.00259\,(GPM)^2}{D^4}$

where hv = velocity head

v = velocity in feet/second

GPM = Gallons per minute

D = I.D. of column pipe

d = O.D. of shaft or oil tube

A (area) = 0.7854 x (D+d)(D-d)

<u>Example</u>

Calculations Based on: GPM = 1219.35
D = 10" (Pipe O.D. = 10.750, Wall thickness 0.330)
d = 1.6875

$hv = \frac{0.00259 \times GPM^2}{D^4}$

$hv = \frac{0.00259 \times 1219.35^2}{10^4}$

$hv = \frac{0.00259 \times 1486814.422}{10{,}000}$

$hv = \frac{3850.849354}{10{,}000} = 0.38508$

or

$hv = 0.0155v^2$ where $v = \left(\frac{0.321 \times GPM}{AREA}\right)^2$

$hv = 0.0155v^2$ where $v = \left(\frac{0.321 \times 1219.35}{77.72353}\right)^2$

$hv = 0.0155v^2$ where $v = \left(\frac{391.41135}{77.72353}\right)^2$

$hv = 0.0155\ (5.035944)^2 = 0.39309$

Notes:
to calculate velocity in column pipe with lineshaft or oil tubing the following formula is used to determine the area.

SPECIFIC SPEED

Specific speed is the speed in RPM at which a given impeller would operate if reduced proportionally in size so as to deliver a capacity of one GPM at one foot of head.

TO CALCULATE:

$$Ns = \frac{R/MIN \times (GAL/MIN)^{.5}}{FT^{.75}} \quad \text{or} \quad Ns = \frac{R/MIN \times \sqrt{GAL/MIN}}{FT^{.75}}$$

WHERE:

R/MIN = Pump rotational speed in revolutions per minute.

GAL/MIN = Pump discharge in U.S. gallons per minute

FT = Total bowl head per stage in feet of liquid pumped

FOR METRIC:

$$Ns = \frac{R/MIN \times (M^3/H)^{.5}}{M^{.75}}$$

WHERE:

M^3/H = Pump discharge in cubic meters per hour

M = Total bowl head per stage in meters of liquid pumped

1. The capacity and head should be selected at the best efficiency point of the largest diameter impeller used in the pump.
2. Specific Speed (Ns) is always calculated for a single stage.
3. Specific Speed (Ns) of any given pump is the same at all rotative speeds.
4. Low specific speed (Ns) indicates the pump design is for low capacity and high head.
5. High specific speed (Ns) indicates the pump design is for high capacity and low head.

EXAMPLE

WHERE: *R/MIM = 1780*

GAL/MIN = 200

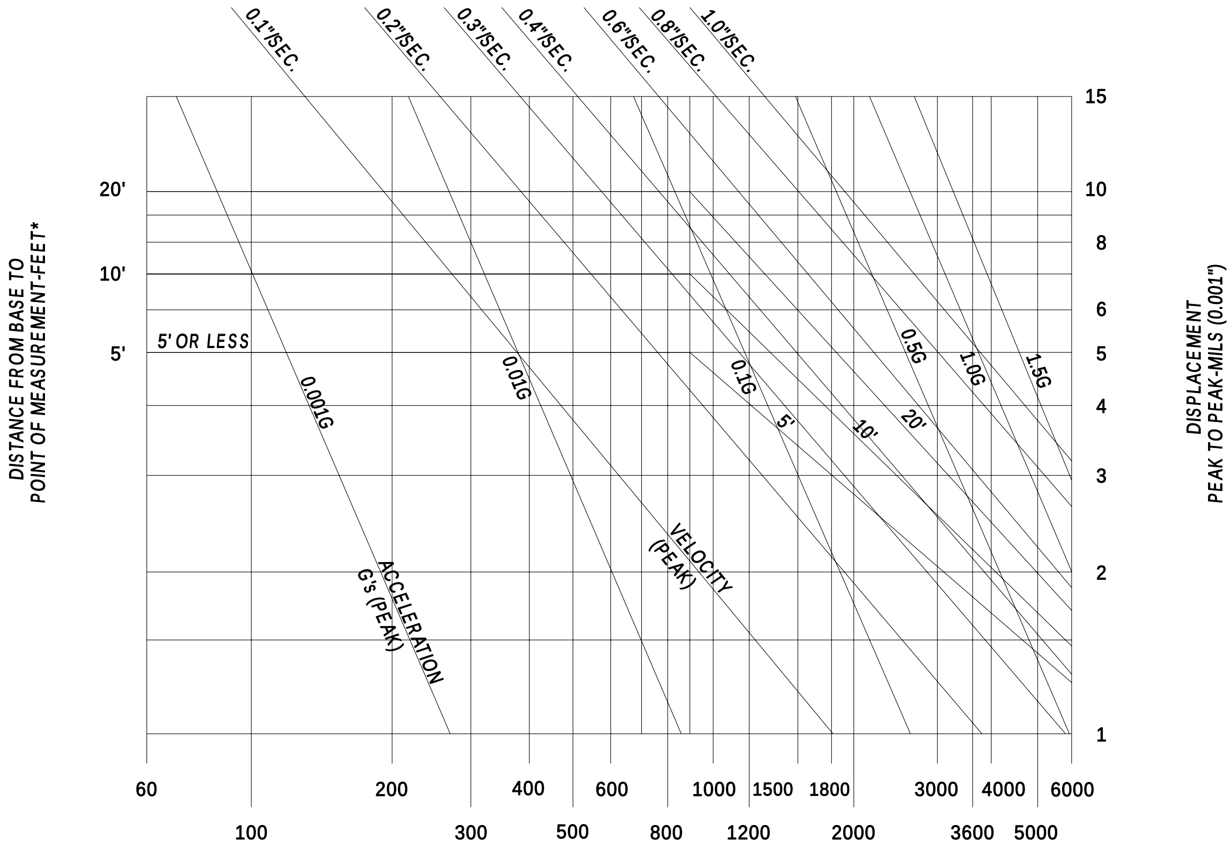
FT = 14.375

$$Ns = \frac{R/MIN \times (GAL/MIN)^{.5}}{FT^{.75}} \qquad Ns = \frac{R/MIN \times \sqrt{GAL/MIN}}{FT^{.75}}$$

$$Ns = \frac{1780 \times (200)^{.5}}{14.375^{.75}} \quad \text{or} \quad Ns = \frac{1780 \times \sqrt{200}}{14.375^{.75}}$$

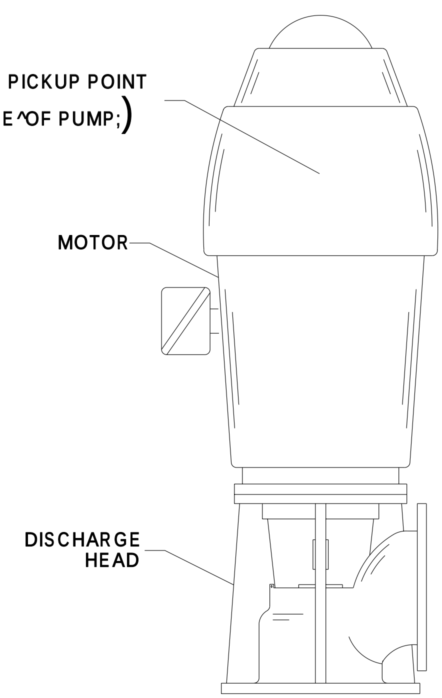
$$Ns = \frac{25173.00141}{7.38254} = 3409.80 \qquad Ns = \frac{25173.00141}{7.38254} = 3409.80$$

MOTOR VIBRATION LIMITS



**Measure Vibration at top motor bearing*

ACCEPTABLE FIELD VIBRATION LIMITS FOR VERTICAL PUMPS (NON-RIGID STRUCTURES) - Based on Hydraulic Institute Standards, 14th Edition, 1983



The chart and illustration represented is the recommended acceptable vibration limits for vertical pumps. Vibrations in excess of the chart values may be acceptable if they show no continued increase over long periods of time and there is no other indication of damage, such as an increase in bearing clearance or noise level. Conversion formulas for vibration readings are as follows:

$D = 1.910 \times 10^{4} \frac{v}{f}$

$v = 3.696 \times 10^{3} \frac{a}{f}$

$a = 2.704 \times 10^{-4} vf$

where,

D = Displacement, peak to peak, in mils (0.001 inches)

v = Velocity, peak, in inches per second

a = Acceleration, peak, in G's

f = Frequency in cycles per minute

TORQUE REGUIRED TO ACCELERATE A REVOLVING BODY

Torque (lb. ft.) = $\dfrac{WR^2\,N}{307 \times t}$

W = weight in lbs. (weight of impeller + weight of taper lock)

R = radius of gyration in feet

N = change in RPM

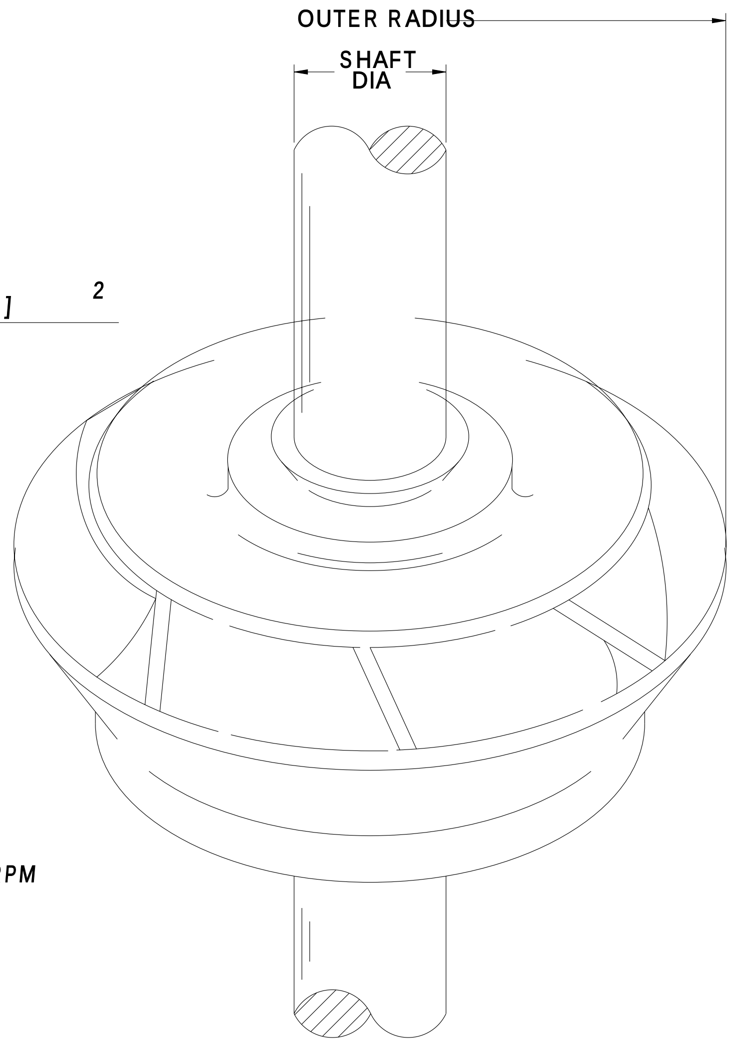
t = time of acceleration in seconds

Inertia of a cylinder about its axis

$$WR^2 = \frac{\text{Weight (lbs.)} \times [\text{radius (ft.)}]^2}{2}$$

Inertia of an impeller about its axis

$$WR^2 = \frac{\text{Weight (lbs.)} \times [(\text{outer radius in ft.})^2 - (\text{shaft diameter})]^2}{2}$$



CONVERTING LINEAR TO ROTATIONAL INERTIA

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

W = Weight in pounds (lbs.)

V = Linear Velocity in feet per min. (fpm) = 0.262 x Dia. (in.) x RPM

N = Motor speed in RPM when load is moving at velocity V

EQUIVALENT WR^2 FOR BELTED OR GEARED LOADS

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

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

N load = Full Speed of Load (RPM)

N motor = Full Speed of Motor (RPM)

THRUST

There are two types of thrust in a vertical turbine. It's very important to understand these types and their effects on performance and failures of pumps and drivers.

In a vertical turbine the water or liquid is picked up by a vaned impeller and discharged thru a number of equally spaced diffusion vanes. Sometimes, if this vane spacing is not equal or the inlet angles are not equal, you will have a small amount of unbalance radial thrust due to the hydraulic forces.

Axial thrust is created from the pressure difference of the impeller and is normally in the downward position. However, the most important is the upward force coming form the velocity of liquid entering the impeller eye. The liquid flow enters axially and is turned by the vanes, leaving the impeller in a semi-radial direction. The angle of exit depends on the specific speed. The upward dynamic force in the low-specific speed impeller is significant enough to overcome the effect of pressure difference, therefore causing a net upthrust, which will start at 120% of BEP. This is not unusual for enclosed impellers.

In a normal operating pump, the upthrust is usually momentary and much smaller than the downthrust. Because the predominant force is downward, driver manufactures design their units (whether electric motors or right angle gear drives) for continuos downthrust operation. Most drivers are designed to handle 30% momentary upthrust loads. Sometimes, when a pump is run at a very high capacity, the upthrust can be greater than the downthrust, especially on close coupled turbines.

In a typical deep well turbine, the static weight of the driveshaft serves as sufficient static head to offset most upthrust situations. Close-coupled barrel pumps should be started against a closed valve or system head which eliminates the momentary upthrust problem.

When a pump must fill a long pipeline, the initial startup head is low and it may run in upthrust until the system head develops. And in some applications the NPSHR may exceed the NPSHA allowing it to run in upthrust. A throttling valve or temporary orifice plate is a good solution. This is especially important in a submersible pump.

Some of the problems caused by continuous operation in upthrust are:

1. Mechanical seal failure caused when the shaft moves upward in an excess amount. This changes the adjustment between the stationary face and the rotating face.

2. Bent lineshaft caused by compression loads which cause rapid bearing wear and vibration.

3. Impellers can rub or bump the top of the bowls hub causing wear on the bowl and impellers. Sometimes enough pressure is applied to loosen the taperlock and allow the impeller to spin freely on the shaft.

4. Damage to thrust bearings in the drivers.

THRUST LOADS

(The total downthrust produced is the sum of the hydraulic thrust plus the static thrust (dead weight) of the shaft and impellers.

SHAFT WEIGHTS AND AREAS										
DIA. (IN's)	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.	1.50	2.67	3.77	4.17	6.01	7.60	10.02	12.78	13.52	15.87
AREA	0.44	0.78	1.11	1.23	1.77	2.24	2.95	3.76	3.97	4.67

TOTAL THRUST FORMULA

TOTAL THRUST = (K x H x SG) + (W x S) + (Imp. Weight x No. of Stages)

K = Thrust Factor for Pump

H = Bowl Head (Total Head + Column Friction Loss) in Feet

W = Weight of Shaft in Pounds

S = Setting (Total Column Length) in Feet

SG = Specific Gravity of Liquid

EXAMPLE: *The total thrust for a 12KCA-4 stage pump with a 1 1/2" line shaft with a total head of 304 feet and a setting of 250 feet, would be calculated as follows:*

TOTAL THRUST = (K x H x SG) + (W x S) + (Imp. Weight x No. of Stages)

K = 6.50 H = 306 W = 6.01 S = 250 SG = 1.00

TOTAL THRUST = (6.50 x 306 x 1) + (6.01 x 250) + (16 x 4)

= 1989 + 1502.5 + 64 = 3555.5

NOTES:

The driver selected must have thrust capacity greater than the total thrust value.
Thrust factors (K) and impeller weights can be found on the performance curve page for the specific model.

FRICTION H.P. LOSS PER 100 FEET

SHAFT SIZE	REVOLUTIONS PER MINUTE						
	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

NPSHR CALCULATIONS

NPSH

(NET POSITIVE SUCTION HEAD) Is the total suction head in feet of the liquid being pumped (absolute at the pump centerline or impeller eye) less the absolute vapor pressure (in feet) of the liquid being pumped. It must always have a positive value and can be calculated by the following equations: To help explain the conditions two expressions will be used: the first expression is basis for when the suction lift-liquid supply level is below the pump centerline or impeller eye; the second expression is basis for positive suction, (flooded), where the liquid supply level is above the pump centerline or impeller eye.

For Suction Lift:

$$NPSH = h_a - h_{vpa} - h_{st} - h_{fs}$$

For Positive (flooded) Suction:

$$NPSH = h_a - h_{vpa} + h_{st} - h_{fs}$$

NPSHR

(NET POSITIVE SUCTION HEAD REQUIRED) Is the amount of suction head, over vapor pressure, required to prevent more than 3% loss in total head to the first stage of the pump at a specific capacity. NPSHR is obtained by laboratory testing, (closed loop system). In a closed-loop test facility you can obtain deareated water, which can not be accomplished in a open pit installation. Dereation occurs in a closed-loop system when the static pressure is below atmospheric pressure and it increases at higher water temperature. Therefore, it is necessary to run comparable NPSHR test at the same water temperature. Also a closed-loop system can be used in special requirement situations when pumping liquids other than water.

*Calculations with regard to NPSHR as follows:

$$NPSHR = h_a - h_{vpa} + (Z_i - Z_e)$$

$$Z_s = NPSHR - \frac{2.31}{S.G.} Pa + \frac{2.31}{S.G.} Pvpa - Z_{st} - \frac{V_s^2}{2g} + h_3 + h_4$$

NPSHA

(NET POSITIVE SUCTION HEAD AVAILABLE) Is the total suction head in feet (meters) of liquid absolute, determined at the first stage impeller datum, less the absolute vapor pressure of the liquid in feet (meters).

*Calculations with regard to NPSHA as follows:

$$NPSHA = h_a - h_{vpa} + Z_s \qquad OR \qquad NPSHA = \frac{2.31}{S.G.}(Pa - Pvpa) + Z_s$$

SYMBOLS AND DEFINITIONS

Z_i (Required submergence) = $NPSHR - h_a + h_{vpa} + Z_s$

h_a = absolute pressure (in feet of the liquid being pumped) on the surface of the liquid supply level (this will be barometric pressure if suction is from an open tank or sump; or the absolute pressure existing in a closed tank such as a condenser hotwell or deareator).

h_{vpa} = Vapor pressure of pumped liquid, ft (m) absolute, at pumping temperature.

h_{st} = Static height in feet that the liquid supply level is above or below the pump centerline or impeller eye.

h_{fs} = All suction line losses (in feet) including entrance losses and friction losses through pipe, valves and fittings, ect.

Z_s = Water depth over impeller eye or the vertical distance from the suction pipe centerline to the eye of the bottom impeller, ft (m).

Z_e = the vertical distance from the suction inlet to the impeller centerline.

Z_{st} = the vertical distance from the pumping water level to the discharge centerline.

P_s = Suction pressure, lb/in^2 (KPa), above atmospheric pressure. This may be positive or negative.

h_3 = Friction loss between pressure tap connection and suction flange.

h_4 = Summation of friction and shock losses in suction elbow and barrel.

P_a = Atmospheric pressure in lb/in^2 (KPa) absolute.

$Pvpa$ = Vapor pressure of water in lb/in^2 (KPa) absolute at pumping temperature.

S.G. = Specific gravity of liquid at pumping temperature.

$V_s^2/2g$ = the velocity head in suction pipe at point of pressure tap or piezometer connection.

g = 32.16 feet per second (acceleration of gravity).

NOTES:

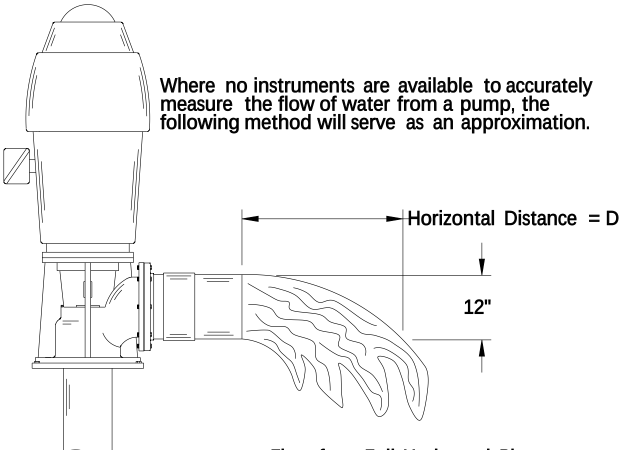
The typical pump curve depicts NPSHR or NPSH with the NPSHR rising as the capacity rises.

ATMOSPHERIC PRESSURE

ALTITUDE vs. ATMOSPHERIC PRESSURE			ALTITUDE OF SOME MAJOR CITIES	
ALTITUDE	PSIA	FT. OF WATER	CITY	APPROX. ALTITUDE
0	14.7	34.0	ALBUQUERQUE	5,200
500	14.4	33.3	ATLANTA	1,100
1,000	14.2	32.8	AMARILLO	370
1,500	13.9	32.1	CALGARY	3,440
2,000	13.7	31.5	CHEYENNE	6,100
2,500	13.4	31.0	CHICAGO	600
3,000	13.2	30.4	CINCINNATI	550
3,500	12.9	29.8	CLEVELAND	700
4,000	12.7	29.2	DENVER	5,270
4,500	12.4	28.8	DETROIT	580
5,000	12.2	28.2	EDMONTON	2,200
5,500	12.0	27.6	FORT WORTH	700
6,000	11.8	27.2	IDAHO FALLS	4,700
6,500	11.5	26.7	KANSAS CITY	800
7,000	11.3	26.2	MINNEAPOLIS	900
7,500	11.1	25.7	MONTREAL	100
8,000	10.9	25.2	NASHVILLE	500
8,500	10.7	24.7	OMAHA	1,000
9,000	10.5	24.3	OTTAWA	290
9,500	10.3	23.8	PHOENIX	1,100
10,000	10.1	23.4	PITTSBURGH	800
10,500	9.9	22.9	REGINA	1,900
11,000	9.7	22.4	ROSWELL	3,570
11,500	9.5	22.0	SALT LAKE CITY	4,250
12,000	9.3	21.6	SPOKANE	1,900
12,500	9.1	21.1	TORONTO	350
15,000	8.3	19.2	TULSA	800
			WINNIPEG	760

ESTIMATING FLOW FROM HORIZONTAL OR INCLINED PIPES

Where no instruments are available to accurately measure the flow of water from a pump, the following method will serve as an approximation.

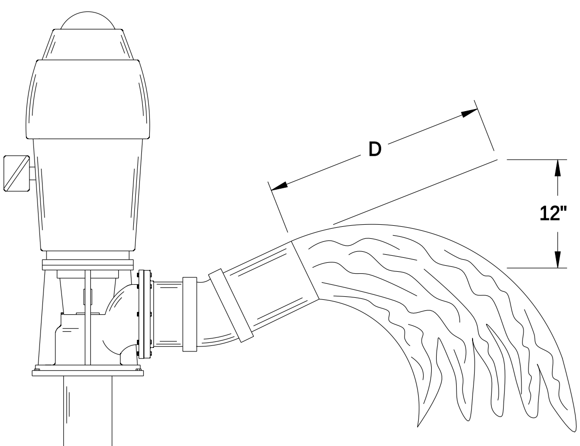


Flow from Full Horizontal Pipe

Using an ordinary rule of carpenters square, measure the horizontal distance from the end of the discharge pipe to a point exactly 12 inches above the falling stream of water. The discharge pipe must be level and running full of water when the reading is taken. Multiply this distance (in inches) by the cross sectional area of the pipe in square inches and the answer will be the approximate capacity in gallons per minute.

By checking this method of estimation using accurate flow meters it has been found a correction factor of 1.015 should be applied.

Flow (GPM) = A x D x 1.105
Where: A = Area of pipe in square inches
D = Horizontal distance in inches
1.015 = Correction Factor

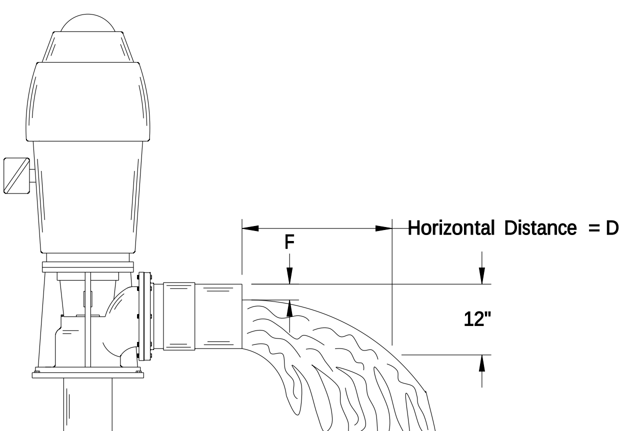


Flow from Inclined Pipe

Where: A = 78.54 (10" pipe)
D = 20"

EXAMPLE: A x D x 1.105
78.54 x 20 x 1.105
GPM = 1736

PARTIALLY FILLED PIPES



Flow From Horizontal Pipe (Partially Filled)

Flow (GPM) = A x D x 1.039 x F
Where: A = Area of pipe in square inches
D = Horizontal distance in inches
F = Effective area factor shown below
Area of pipe equals inside Dia. x 0.7854 ^{2}

RATIO F/D = R %	EFF. AREA FACTOR F	RATIO F/D = R %	EFF. AREA FACTOR F
5	0.981	55	0.436
10	0.948	60	0.373
15	0.905	65	0.312
20	0.858	70	0.253
25	0.805	75	0.195
30	0.747	80	0.142
35	0.688	85	0.095
40	0.627	90	0.052
45	0.564	95	0.019
50	0.500	100	0.000

EXAMPLE: D = 20 Inches - Pipe inside diameter = 10 inches -
F = 2 1/2 inches
A = 10 x 10 x 0.7854 = 78.54 square inches
R = 2 1/2 - 10 = 25%
F = 0.805
Flow = 78.54 x 20 x 1.039 x 0.805 = 1314 GPM

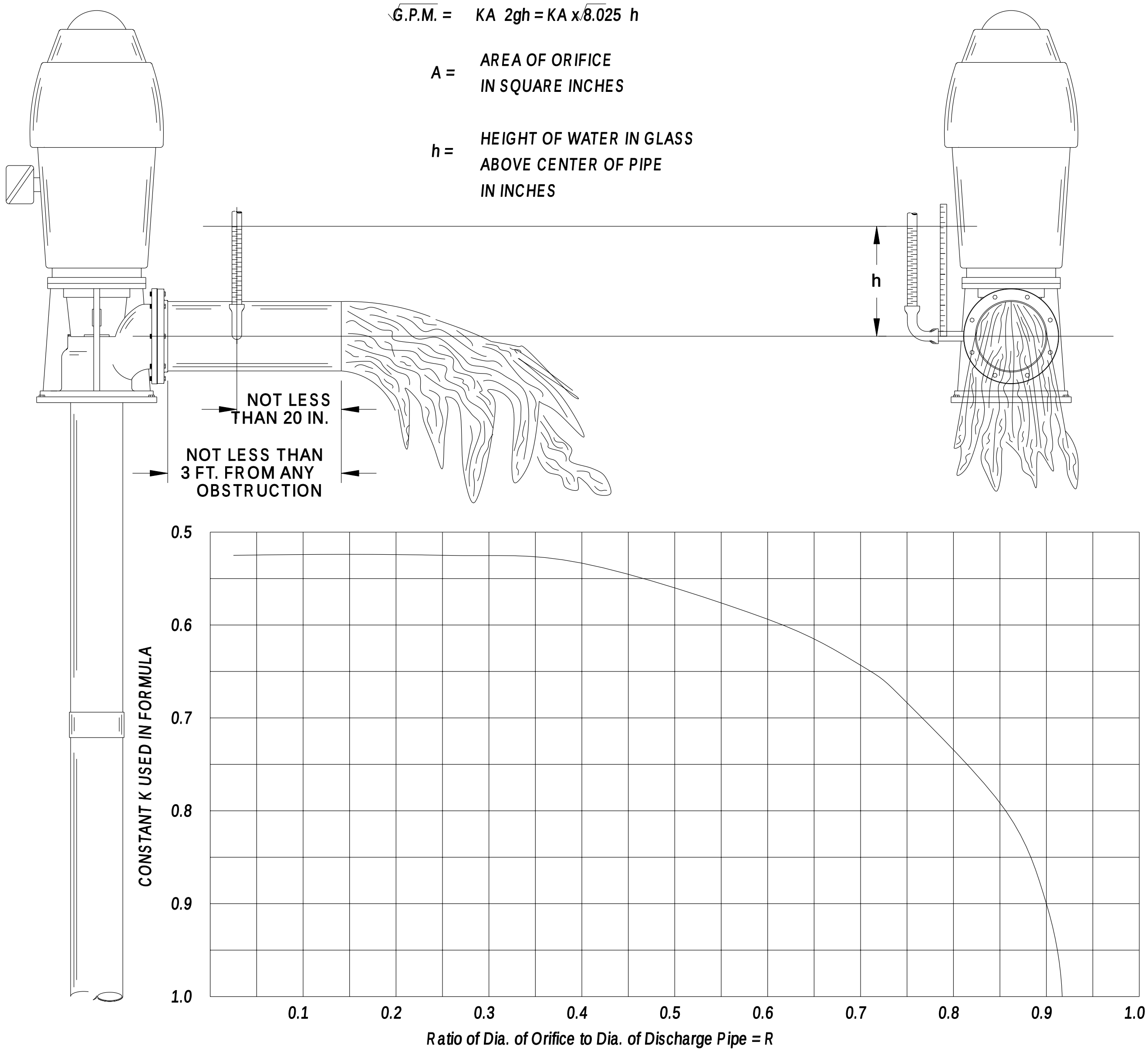
ORIFICE METHOD OF MEASURING WATER

$$R = \frac{\text{DIA. OF ORIFICE IN INCHES}}{\text{DIA. OF DISCHARGE PIPE}}$$

$$G.P.M. = KA\sqrt{2gh} = KA \times 8.025\sqrt{h}$$

A = AREA OF ORIFICE IN SQUARE INCHES

h = HEIGHT OF WATER IN GLASS ABOVE CENTER OF PIPE IN INCHES



The orifice method is a simple way to measure the flow of water from a pipe discharging horizontally into open air without bends or obstruction in the last three feet. The sketch shows the arrangement.

A plate is clamped over the end of the pipe with the circular orifice located at the exact center of the pipe. The size of the orifice should preferably be from one-half to three-quarter the size of the pipe, but must be selected of such size that it will be running full of water.

At a point on the side of the discharge pipe not less than 20 inches back from the orifice a hole should be drilled and tapped for an 1/8" or 1/4" pipe. A short piece of pipe is screwed into this hole until the inner end is just flush with the inner wall of the discharge pipe. A piece of rubber hose is slipped over this pipe and also over one end of a piece of glass tubing supported in a vertical position.

The diameter of the orifice divided by the inside pipe diameter gives the ratio "R" and from the curve the proper value of "K" is found. In the formula "A" is figured as the area of the orifice in square inches, "G" equals 32.2, and "H" is read as the height of the water column in the glass tube above the center of the pipe in inches. Gallons per minute "G.P.M." can then be figured. This method is simpler than a weir and fully as accurate.

ORIFICE METHOD OF MEASURING WATER

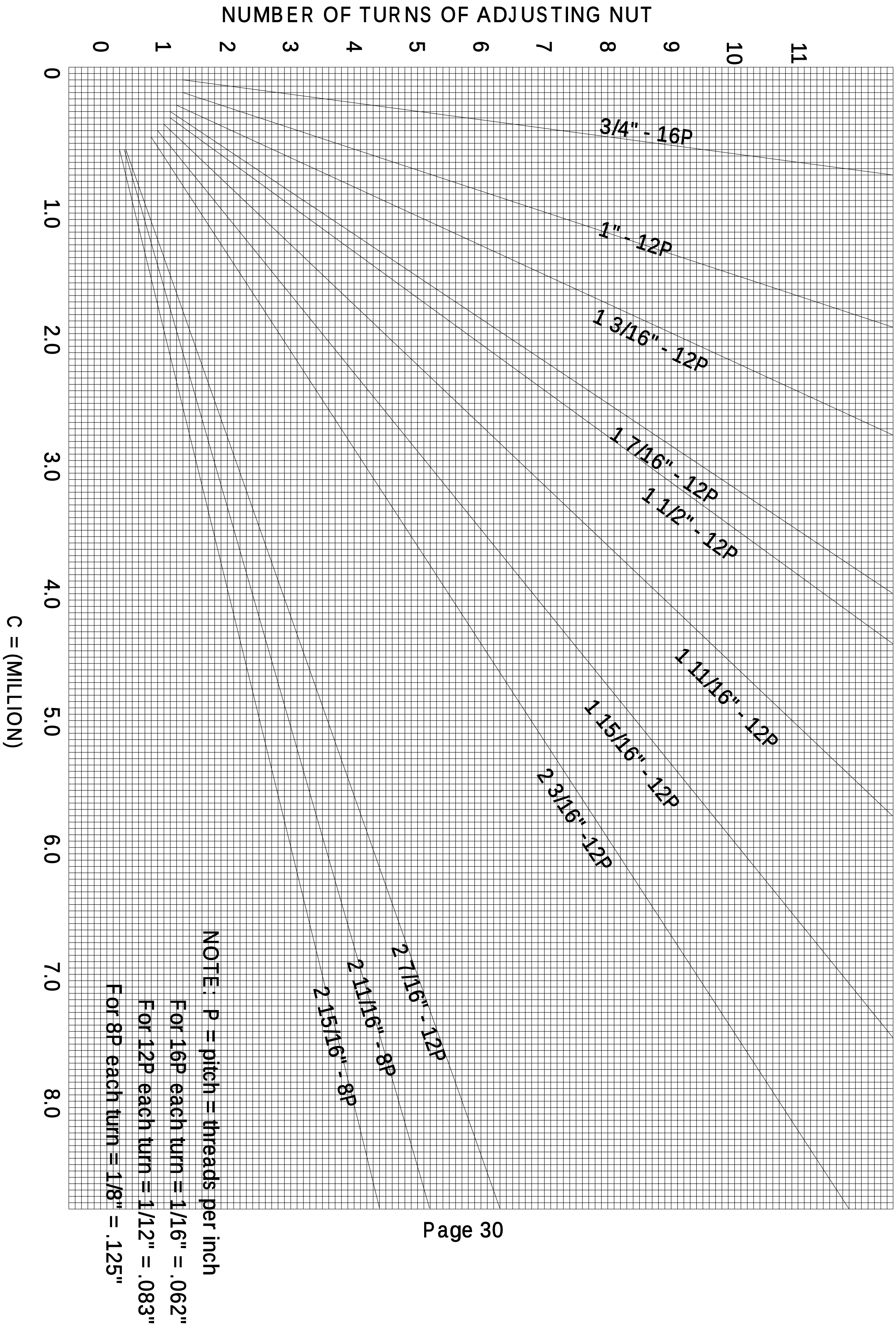
H. INCHES	√H	GALLONS PER MINUTE						H. INCHES	√H	GALLONS PER MINUTE					
		3" PIPE 1 3/4" ORIFICE	4" PIPE 2 1/2" ORIFICE	6" PIPE 3" ORIFICE	6" PIPE 4 3/4" ORIFICE	8" PIPE 6" ORIFICE	10" PIPE 8" ORIFICE			3" PIPE 1 3/4" ORIFICE	4" PIPE 2 1/2" ORIFICE	6" PIPE 3" ORIFICE	6" PIPE 4 3/4" ORIFICE	8" PIPE 6" ORIFICE	10" PIPE 8" ORIFICE
1	1.000	11.3	24.0	31.7	101.6	154.5	296.5	61	7.810	88.3	187.5	248.1	793.0	1207.0	2314.0
2	1.414	16.1	33.9	44.9	143.8	218.5	419.0	62	7.874	89.2	189.0	250.1	799.0	1217.0	2332.0
3	1.732	19.6	41.6	55.2	176.0	267.6	513.0	63	7.937	89.8	190.5	252.1	806.0	1227.0	2352.0
4	2.000	22.7	48.0	63.5	203.3	309.0	593.0	64	8.000	90.6	192.0	254.1	813.0	1237.0	2371.0
5	2.236	25.3	53.6	71.1	227.4	345.5	692.0	65	8.062	91.3	193.5	256.1	819.0	1247.0	2389.0
6	2.449	27.8	58.8	77.8	249.0	378.5	727.0	66	8.124	92.2	195.2	258.1	826.0	1256.0	2408.0
7	2.646	30.0	63.5	84.0	268.7	409.0	785.0	67	8.185	92.7	196.4	260.1	832.0	1265.0	2426.0
8	2.828	32.0	67.9	89.9	287.2	437.0	838.0	68	8.246	93.4	198.0	262.0	837.0	1274.0	2442.0
9	3.000	34.0	72.0	95.4	305.0	463.0	889.0	69	8.307	94.1	199.5	264.0	844.0	1284.0	2461.0
10	3.162	35.9	76.0	100.6	321.2	488.0	938.0	70	8.367	94.8	201.0	265.9	850.0	1293.0	2479.0
11	3.317	37.6	79.6	105.4	337.0	512.0	982.0	71	8.426	95.5	202.3	267.9	856.0	1303.0	2496.0
12	3.464	39.3	83.2	110.2	352.0	530.0	1027.0	72	8.485	96.1	203.9	269.9	862.0	1312.0	2518.0
13	3.606	40.8	86.6	114.6	367.0	557.0	1069.0	73	8.544	96.8	205.0	271.9	868.0	1321.0	2538.0
14	3.742	42.3	89.8	119.0	381.0	577.5	1108.0	74	8.602	97.5	206.6	273.8	874.0	1330.0	2550.0
15	3.873	43.9	93.0	123.1	394.0	598.0	1148.0	75	8.660	98.2	208.0	275.6	880.0	1339.0	2565.0
16	4.000	45.3	96.0	127.2	407.0	618.0	1186.0	76	8.718	98.8	209.3	277.2	886.0	1348.0	2583.0
17	4.123	46.7	98.9	131.0	419.0	637.0	1222.0	77	8.775	99.4	210.6	279.0	892.0	1356.0	2605.0
18	4.243	48.1	101.8	135.0	432.0	655.0	1258.0	78	8.832	100.1	211.9	280.8	898.0	1365.0	2619.0
19	4.359	49.3	104.7	138.5	442.0	673.0	1291.0	79	8.888	100.7	213.2	282.7	903.0	1374.0	2632.0
20	4.472	50.7	107.4	142.3	454.0	692.0	1327.0	80	8.944	101.3	214.6	284.0	908.0	1383.0	2653.0
21	4.583	52.0	110.0	145.8	466.0	708.0	1359.0	81	9.000	102.0	216.0	286.0	914.0	1392.0	2664.0
22	4.690	53.2	112.6	149.3	477.0	725.0	1391.0	82	9.055	102.6	217.3	287.8	920.0	1400.0	2682.0
23	4.796	54.3	115.0	152.5	487.0	740.0	1421.0	83	9.110	103.2	218.6	289.4	926.0	1408.0	2700.0
24	4.899	55.5	117.6	155.7	497.0	757.0	1452.0	84	9.165	103.8	219.9	291.0	932.0	1416.0	2713.0
25	5.000	56.7	120.0	159.0	508.0	772.0	1482.0	85	9.220	104.4	221.2	292.6	937.0	1424.0	2731.0
26	5.099	57.7	122.3	162.0	518.0	788.0	1512.0	86	9.274	105.2	222.6	294.6	942.0	1433.0	2748.0
27	5.196	58.8	124.8	165.2	528.0	802.0	1539.0	87	9.327	105.7	223.9	296.6	947.0	1441.0	2764.0
28	5.292	60.0	127.0	168.4	538.0	817.0	1568.0	88	9.381	106.3	225.2	298.2	953.0	1449.0	2780.0
29	5.385	61.1	129.2	171.3	547.0	833.0	1598.0	89	9.434	106.9	226.4	300.0	958.0	1457.0	2798.0
30	5.477	62.1	131.5	174.2	557.0	846.0	1623.0	90	9.487	107.5	227.6	301.6	964.0	1465.0	2812.0
31	5.568	63.0	133.8	177.0	566.0	860.0	1651.0	91	9.539	108.0	228.9	303.2	970.0	1473.0	2827.0
32	5.657	64.0	135.9	180.0	575.0	874.0	1675.0	92	9.592	108.7	230.2	304.8	975.0	1481.0	2845.0
33	5.745	65.1	137.9	182.7	584.0	887.0	1702.0	93	9.644	109.3	231.5	306.4	980.0	1489.0	2859.0
34	5.831	66.1	140.0	185.5	593.0	900.0	1729.0	94	9.695	109.8	232.7	308.0	985.0	1497.0	2875.0
35	5.916	67.0	141.8	188.0	602.0	913.0	1751.0	95	9.747	110.4	233.9	309.6	990.0	1505.0	2890.0
36	6.000	68.0	144.0	190.8	610.0	927.0	1779.0	96	9.798	111.0	235.2	311.2	995.0	1513.0	2905.0
37	6.083	68.9	146.0	193.5	617.0	940.0	1802.0	97	9.849	111.6	236.5	312.8	1000.0	1521.0	2920.0
38	6.164	69.8	148.0	196.0	626.0	952.0	1828.0	98	9.900	112.2	237.6	314.7	1006.0	1529.0	2936.0
39	6.245	70.7	149.9	198.5	634.0	964.0	1850.0	99	9.950	112.7	238.8	316.2	1011.0	1537.0	2950.0
40	6.325	71.7	151.9	201.0	643.0	977.0	1875.0	100	10.000	113.3	240.0	317.7	1017.0	1545.0	2965.0
41	6.403	72.5	153.8	203.5	651.0	989.0	1899.0	101	10.050	113.9	241.2	319.3	1022.0	1553.0	2980.0
42	6.481	73.4	155.7	206.0	659.0	1001.0	1921.0	102	10.100	114.4	242.4	321.0	1027.0	1560.0	2995.0
43	6.557	74.2	157.5	208.5	666.0	1013.0	1942.0	103	10.149	115.0	243.8	322.6	1032.0	1568.0	3010.0
44	6.633	75.0	159.2	211.0	674.0	1025.0	1965.0	104	10.198	115.5	244.8	324.0	1037.0	1573.0	3025.0
45	6.708	75.9	161.0	213.5	681.0	1037.0	1987.0	105	10.247	116.2	246.0	325.9	1042.0	1583.0	3039.0
46	6.782	76.8	162.8	215.7	689.0	1049.0	2010.0	106	10.296	116.6	247.2	327.0	1047.0	1591.0	3052.0
47	6.856	77.7	164.9	218.0	696.0	1060.0	2032.0	107	10.344	117.2	248.4	328.8	1052.0	1598.0	3068.0
48	6.928	78.5	166.4	220.0	704.0	1071.0	2050.0	108	10.392	117.7	249.4	330.0	1057.0	1606.0	3079.0
49	7.000	79.2	168.0	222.5	712.0	1082.0	2073.0	109	10.440	118.3	250.6	332.0	1062.0	1613.0	3095.0
50	7.071	80.0	169.9	224.7	718.0	1093.0	2095.0	110	10.488	118.8	251.7	333.6	1067.0	1620.0	3110.0
51	7.141	80.8	171.5	227.0	725.0	1104.0	2118.0	111	10.536	119.4	252.8	335.0	1071.0	1628.0	3128.0
52	7.211	81.6	173.1	229.0	732.0	1114.0	2135.0	112	10.583	119.9	254.0	336.5	1076.0	1635.0	3140.0
53	7.280	82.5	174.9	231.5	740.0	1125.0	2157.0	113	10.630	120.4	255.2	338.0	1081.0	1642.0	3152.0
54	7.349	83.2	176.3	233.5	747.0	1135.0	2178.0	114	10.677	121.0	256.2	339.5	1086.0	1650.0	3163.0
55	7.416	83.8	178.0	235.5	753.0	1145.0	2195.0	115	10.724	121.3	257.5	341.0	1090.0	1657.0	3179.0
56	7.483	84.8	179.5	238.0	761.0	1156.0	2219.0	116	10.770	122.0	258.5	342.5	1095.0	1664.0	3193.0
57	7.550	85.5	181.4	240.0	767.0	1167.0	2239.0	117	10.817	122.5	259.6	344.0	1100.0	1671.0	3210.0
58	7.616	86.2	182.8	242.0	773.0	1177.0	2253.0	118	10.863	123.1	260.7	345.5	1105.0	1678.0	3219.0
59	7.681	86.8	184.2	244.2	781.0	1187.0	2278.0	119	10.909	123.6	261.8	347.0	1109.0	1685.4	3235.0
60	7.746	87.7	186.0	246.2	787.0	1197.0	2297.0	120	10.955	124.1	262.9	348.3	1114.0	1693.0	3243.0

To arrive at capacities in Gallons per Minute when the reading of H is more than 120" use the following formulas:

G.P.M. 3" pipe 1 3/4" orifice = 11.329 x the square root of H.
G.P.M. 4" pipe 2 1/2" orifice = 24.00 x the square root of H.
G.P.M. 6" pipe 3" orifice = 31.77 x the square root of H.
G.P.M. 6" pipe 4 3/4" orifice = 101.69 x the square root of H.
G.P.M. 8" pipe 6" orifice = 154.50 x the square root of H.
G.P.M. 10" pipe 8" orifice = 296.50 x the square root of H.

AMERICAN-MARSH PUMPS

SHAFT ADJUSTMENT



SHAFT STRETCH THRUST LOAD

The turbine pump lineshaft stretches when the pump is in operation due to a downward pull or hydraulic thrust. The stretch due to this thrust may be determined and the impellers raised by that amount so that they will operate in the desired location with respect to the bowls when the pump is running.

Careful adjustment must be made, particularly with semi-open impellers where optimum performance is obtained with only a few thousands of an inch clearance between the bottom of the impeller and the bowl face. For side seal or combination seal impellers, the location is not critical, and they are usually set so there is adequate clearance to prevent rubbing.

The End Play (Lateral) - Thrust Constant table shows the thrust constant (K) for each bowl size representing the pounds thrust for each foot of pumping head.

Following is a curve sheet indicating the turns of the adjusting nut to equal the shaft stretch as a function of "C" for various shaft sizes.

"C" is the product of bowl thrust constant (K), the total pumping head and the shaft length or setting.

The following is an example of how the shaft stretch is determined:

	Bowls	*12MC*
	Shaft Size	*1 3/16"*
	Setting	*400'*
	Total Head	*180'*

"C" = Bowl thrust constant (K) x total pumping head x setting = 10.6 x 180 ft. x 400 ft. = 763,200.

From chart for "C" on back = 763,200 and 1 3/16" shaft, number of turns = 3.25.

The shaft should be raised 0.85 turns after the impellers just clear the bowl face to allow for the shaft stretch due to hydraulic thrust.

THRUST BEARING LOAD

The thrust-bearing load that the driver thrust bearing must carry can be determined by adding the weight of the lineshafting to the product of the thrust constant (K) and the total pumping head plus the weight of impellers.

The table below show the weight of the lineshafting for common sizes.

Example:	*Bowls*	*12MCB3*
	Total Head	*180'*
	Setting	*400'*
	Shaft Size	*1 3/16"*
	Motor 75 Hp	*1750 RPM*

Hydraulic Thrust = thrust constant (K) times total head (ft.).
Thrust constant from the table for 12MC = 10.6 Lbs. per foot.
Hydraulic Thrust = 10.6 Lbs./ft. x 180 ft. = 1,908 Lbs.
Shaft weight per foot for 1 1 3/16" shafting from Data Sheet E 210A = 3.8 Lbs. per foot.
Total shaft weight = 3.8 Lbs./ft. x 400' = 1520 Lbs.
Total Impeller weight = 3 x 15 = 45 Lbs.
Total load on thrust bearing equals hydraulic thrust plus shaft weight plus impeller wt. 1908 Lbs. + 1520 Lbs. + 45 Lbs. = 3,473 Lbs.

From the motor manufacture's specification, allowable thrust load for 75 Hp, 1,750 RPM motor is 4,800 Lbs.

Therefore the thrust is well within the permissible load for the standard heavy thrust motor. If the thrust load exceeded the motor rating, and extra heavy thrust bearing would be required.

See Reverse for Shaft Adjustment Chart.

Shaft Diameter	*Weight Per Foot (Lbs.)*
3/4"	*1.50*
1"	*2.70*
1 3/16"	*3.80*
1 7/16"	*6*
1 1/2"	*6*
1 11/16"	*7.60*
1 15/16"	*10.00*
2 3/16"	*13*
2 7/16"	*15.90*
2 11/16"	*19.30*
2 15/16"	*23*

AMERICAN-MARSH PUMPS

SHAFT SELECTION CHART

SHAFT DIAMETER (in.)	SPEED (rpm)	PUMP THRUST (LBS) 1,000		2,000		3,000		5,000		7,500		10,000		15,000		20,000		30,000	
		POWER RATING (H.P.)																	
		1045	SS	1045	SS	1045	SS	1045	SS	1045	SS	1045	SS	1045	SS	1045	SS	1045	SS
1	3450	114	140	114	140	113	139	111	139	107	134								
	2900	95	117	94	117	94	116	92	115	88	112								
	2200	72	89	72	89	71	88	70	87	67	85								
	1760	57	70	57	70	57	70	56	69	53	67								
	1460	47	58	47	58	47	58	46	57	44	55								
	1160	38	46	37	46	37	46	36	55	35	44								
	880	N/A	34	N/A	34	N/A	34	N/A	33	N/A	32								
1 3/16	3450	203	248	202	248	202	247	200	245	196	243	191	238						
	2900	168	208	168	208	167	207	166	205	162	204	158	200						
	2200	127	158	127	158	127	157	125	156	123	154	120	151						
	1760	102	124	101	124	101	124	100	123	98	122	96	120						
	1460	84	102	84	102	84	102	83	102	82	101	80	99						
	1160	67	82	67	82	67	82	66	81	65	80	63	79						
	880		61		60		60		60		59		58						
	770		48		47		47		47		46		46						
1 1/2	3450					433	530	432	528	429	526	425	523	413	514				
	2900					359	445	357	443	355	442	352	439	342	432				
	2200					272	337	271	336	269	335	267	333	260	327				
	1760					217	266	217	265	215	264	213	263	208	258				
	1460					180	220	180	219	179	218	177	218	172	214				
	1160					143	175	143	175	143	174	140	173	137	170				
	880					106	130	106	129	105	129	104	128	101	126				
	700						103		102		101		101		100				
1 11/16	2200						485		485		483		482		476				
	1760					318	388	317	388	316	387	314	386	309	381	302			
	1460					263	321	263	321	262	321	260	320	256	316	250			
	1160					209	256	209	256	208	255	207	254	204	251	199			
	880					155	190	155	189	154	189	153	188	151	186	147			
	700						151		150		150		149		147				
1 15/16	1760							493	603	492	602	490	601	486	597	480	592		578
	1460							409	500	408	499	407	498	403	495	398	491	384	479
	1160							325	397	324	397	323	396	320	394	316	390	305	381
	880							241	294	240	294	239	293	237	292	234	289	226	282
	700								233		233		233		232		229		224
2 3/16	1760							724	885	723	884	721	883	718	880	712	876	697	863
	1460							600	734	599	733	598	732	595	729	591	726	578	715
	1160							477	583	476	582	475	582	473	580	469	577	459	569
	880							353	432	353	432	352	431	350	430	348	428	340	422
	700							280	343	280	343	279	342	278	342	276	340	270	335
2 7/16	1760									951	1163	949	1162	946	1159	941	1155	927	1144
	1460									788	964	787	963	785	961	782	958	769	948
	1160									626	766	626	765	623	764	620	761	611	754
	880									464	568	464	567	462	566	460	560	453	558
	700									369	568	369	450	367	450	365	445	360	443
2 15/16	1760												2106		2103		2100		2091
	1460												1746		1744		1741		1734
	1160												1387		1385		1383		1377
	880												1053		1051		1050		1045
	700												837		836		835		831

*Determine pump thrust from the thrust constant and shaft stretch thrust load pages of this section.

AMERICAN-MARSH PUMPS

SHAFT ELONGATION

Inches per 100 Ft. of Shaft
USE OF THIS TABLE IS LIMITED TO 500 FT. SETTING. FOR DEEPER SETTING, CONSULT THE FACTORY

HYDRAULIC THRUST	SHAFT DIAMETER													
	3/4	1	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	0.030	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	0.020	0.015	0.012	0.010					
1600	0.150	0.084	0.060	0.038	0.030	0.022	0.018	0.014	0.012					
1800	0.169	0.095	0.067	0.042	0.033	0.025	0.020	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	0.090	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	0.030	0.025	0.020	0.017	0.015			
3200		0.169	0.119	0.075	0.059	0.045	0.035	0.028	0.023	0.020	0.017	0.014		
3600		0.190	0.135	0.085	0.067	0.051	0.040	0.032	0.026	0.022	0.019	0.016		
4000		0.211	0.150	0.094	0.074	0.056	0.044	0.036	0.029	0.025	0.021	0.018	0.016	
4400		0.240	0.164	0.103	0.081	0.062	0.048	0.039	0.032	0.027	0.024	0.020	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	0.020	0.018
5600			0.209	0.131	0.107	0.079	0.062	0.050	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	0.020
6500			0.243	0.153	0.120	0.091	0.071	0.058	0.047	0.040	0.034	0.029	0.025	0.022
7000			0.260	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	0.080	0.066	0.055	0.047	0.040	0.035	0.031
10,000				0.234	0.185	0.140	0.110	0.089	0.073	0.061	0.052	0.045	0.039	0.034
12,000				0.281	0.222	0.168	0.132	0.106	0.088	0.073	0.062	0.054	0.047	0.041
14,000					0.259	0.196	0.154	0.124	0.102	0.086	0.073	0.062	0.055	0.048
16,000					0.296	0.224	0.176	0.142	0.117	0.098	0.083	0.071	0.062	0.054
18,000						0.252	0.198	0.160	0.131	0.110	0.093	0.080	0.070	0.061
20,000						0.280	0.220	0.176	0.146	0.122	0.104	0.089	0.078	0.068
22,000							0.242	0.195	0.160	0.134	0.114	0.098	0.086	0.074
24,000							0.264	0.213	0.175	0.147	0.124	0.107	0.094	0.082
26,000							0.286	0.230	0.190	0.159	0.135	0.116	0.102	0.088
28,000								0.248	0.204	0.171	0.145	0.125	0.109	0.095
30,000								0.266	0.219	0.183	0.156	0.134	0.117	0.104
32,000								0.283	0.233	0.196	0.166	0.143	0.125	0.109
34,000									0.248	0.208	0.176	0.152	0.133	0.116
36,000									0.262	0.220	0.187	0.160	0.140	0.122
38,000									0.277	0.232	0.197	0.170	0.148	0.129
40,000									0.292	0.245	0.207	0.178	0.156	0.136

Downthrust due to the hydraulic thrust of the pump causes the shaft and column to stretch after the pump is in operation. Unless the impellers can be and are raised off the impeller fit in the bowls enough to allow for this stretch plus some running clearance, the impellers will rub, causing the pump to wear and increase the horsepower required. With the total hydraulic downthrust known and the Column Elongation determined from this Chart the total stretch of the column tube for the setting in question can be determined. To find the net elongation subtract the shaft elongation from column elongation.

$$e = \frac{L \times 12 \times H.T.}{E \times G.S.A.}$$

Where:
e = Elongation (in inches)
L = Shaft Length (feet)
E = Modulus of Elasticity (29,000,000)
H.T. = Hydraulic Thrust (pounds)
G.S.A. = Gross Shaft Area (sq. inches)

AMERICAN-MARSH PUMPS

COLUMN AND TUBE ELONGATION

Inches per 100 Ft. of Column (For open Line Shaft Column multiply values by 1.3)
USE OF THIS TABLE IS LIMITED TO 500 FT. SETTING. FOR DEEPER SETTING, CONSULT THE FACTORY

	COLUMN DIAMETER Standard pipe, nominal I.D. except as indicated by*								
HYDRAULIC THRUST	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	0.010	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	0.020	0.015	0.012	0.009	0.007			
2400	0.032	0.023	0.019	0.014	0.010	0.008	0.006		
2800	0.037	0.027	0.022	0.016	0.012	0.010	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	0.010	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	0.010	0.009
5200		0.051	0.040	0.030	0.023	0.018	0.014	0.011	0.010
5600		0.055	0.043	0.033	0.024	0.019	0.015	0.012	0.011
6000			0.046	0.035	0.026	0.020	0.016	0.013	0.011
6500			0.050	0.038	0.028	0.022	0.017	0.014	0.012
7000			0.054	0.041	0.030	0.024	0.018	0.015	0.013
7500			0.058	0.044	0.033	0.025	0.020	0.016	0.014
8000			0.062	0.047	0.035	0.027	0.021	0.017	0.015
9000				0.053	0.039	0.030	0.023	0.019	0.017
10,000				0.059	0.043	0.034	0.026	0.021	0.019
12,000				0.070	0.052	0.041	0.031	0.025	0.023
14,000				0.082	0.061	0.048	0.036	0.029	0.026
16,000				0.094	0.070	0.054	0.042	0.034	0.030
18,000					0.078	0.061	0.047	0.038	0.034
20,000					0.087	0.068	0.052	0.042	0.037
22,000					0.096	0.075	0.057	0.046	0.041
24,000					0.104	0.082	0.063	0.050	0.045
26,000					0.113	0.088	0.068	0.055	0.049
28,000						0.095	0.073	0.059	0.052
30,000						0.102	0.078	0.063	0.056
32,000						0.109	0.083	0.067	0.060
34,000						0.115	0.089	0.071	0.064
36,000						0.122	0.094	0.076	0.068
38,000						0.129	0.099	0.080	0.071
40,000						0.136	0.104	0.084	0.075

Downthrust due to the hydraulic thrust of the pump causes the shaft and column to stretch after the pump is in operation. Unless the impellers can be and are raised off the impeller fit in the bowls enough to allow for this stretch plus some running clearance, the impellers will rub, causing the pump to wear and increase the horsepower required. With the total hydraulic downthrust known and the Column Elongation determined from this Chart the total stretch of the column tube for the setting in question can be determined. To find the net elongation subtract the shaft elongation from column elongation.

$$e = \frac{L \times 12 \times H.T.}{E \times G.S.A.}$$

e = Elongation (in inches)
L = Shaft Length (feet)
E = Modulus of Elasticity (29,000,000)
H.T. = Hydraulic Thrust (pounds)
G.S.A. = Gross Shaft Area (sq. inches)

THRUST BEARING HORSEPOWER

The published efficiencies of drivers do not include any external thrust on rotor. The thrust load as a unit operating loss must be added to the brake horsepower along with mechanical friction in BHP to arrive at the actual pump brake horsepower requirements of a pump (field BHP).

FORMULA:

Convert total thrust to loss in H.P. by the following formula:

$$\text{Thrust Bearing H.P.} = 0.0075 \times \frac{RPM}{100} \times \frac{THRUST}{1000}$$

Example: *Total Thrust of 4457.3*

$$\text{Thrust Bearing H.P.} = 0.0075 \times \frac{1770}{100} \times \frac{4457.3}{1000}$$

$$= 0.6 \text{ Thrust Bearing H.P. Loss.}$$

THRUST BEARING LOSSES

TOTAL THRUST IN POUNDS	THRUST BEARING LOSS RPM 3500	1770	1170	880
1000	0.262	0.133	0.088	0.066
2000	0.525	0.268	0.175	0.132
3000	0.790	0.400	0.263	0.198
4000	1.05	0.532	0.350	0.264
5000	1.32	0.665	0.438	0.330
6000	1.58	0.796	0.525	0.396
7000	1.84	0.930	0.615	0.460
8000	2.10	1.06	0.700	0.528
9000	2.36	1.20	0.79	0.593
10,000	2.62	1.33	0.88	0.66
15,000	3.95	1.98	1.40	0.99
20,000	5.25	2.68	1.75	1.32
25,000		3.32	2.20	1.65
30,000		4.00	2.63	1.98
35,000		4.65	3.07	2.30
40,000		5.32	3.50	2.64
45,000		5.98	3.95	2.97
50,000			4.38	3.30

NOTE:
THE ABOVE EXPRESSION IS BASED ON ANGULAR CONTACT ANTI-FRICTION BEARINGS ONLY.

AMERICAN-MARSH PUMPS

POWER CONSUMPTION AND COST

POWER CONSUMPTION OF ELECTRIC MOTORS

There are two methods commonly used to check the power consumption of an electric motor.

The first of these requires the use of an ammeter and voltmeter. The following formula is then used utilizing the instrument readings:

$$\text{Kilowatts} = \frac{I \times E \times P.F. \times C}{1000}$$

where I = amperes (meter reading)

E = volts (meter reading)

P.F. = Power Factor (See manufacturer's published operating characteristics for vertical motors.)

C = 1 for single phase current
= 2 for two phase, four wire control
= 1.73 for three phase current

The second method commonly used to determine power consumption utilizes the watt-hour meter in the power line. The exact time for a given number of revolutions of the meter disc is measured with a stopwatch and the following formula used:

$$\text{Kilowatts} = 3.6 \times K \times M \times R/t$$

where K = Disc constant, representing watt-hours per revolution. This factor is found on the meter nameplate or painted on the disc.

M = Product of current transformer ratio and potential transformer ratio. (When either transformer is not used the equivalent ratio is 1.)

R = Total revolutions of watt-hour meter disc.

t = Time for total revolutions of disc in seconds.

COST OF PUMPING USING AN ELECTRIC MOTOR

The cost of operating a vertical turbine pump may be determined by several different methods.

1. If the cost of operation per hour is desired, the power consumption as determined by use of the methods previously described may be used:

Cost/hour of operation = KW's consumed x cost per KWh.

2. The cost of operation may be estimated by determining the input horsepower and converting it to kilowatts:

Cost/hour of operation = 1 HP x 0.746 x cost per KWh.

3. A somewhat less accurate estimate may be made by using the following formula:

$$\text{Cost/hr. of operation} = \frac{\text{GPM} \times \text{Tot. Hd.} \times 0.746 \times \text{Cost/KWh}}{3960 \times \text{Pump Eff.} \times \text{Motor Eff.}}$$

4. It is often desirable to express the cost of operating a pump in terms of "cost per 1000 gallons". To do this the above figures of cost per hour of operation may be used with the rated capacity of the pump as follows:

$$\text{Cost per 1000 Gallons} = \frac{\text{Cost per Hour}}{1000}$$

5. For convenience the following table may be used to estimate power consumption and cost of operation when the overall efficiencies are known. The table gives power consumed pumping 1000 GPM at one foot total head at various overall pump efficiencies.

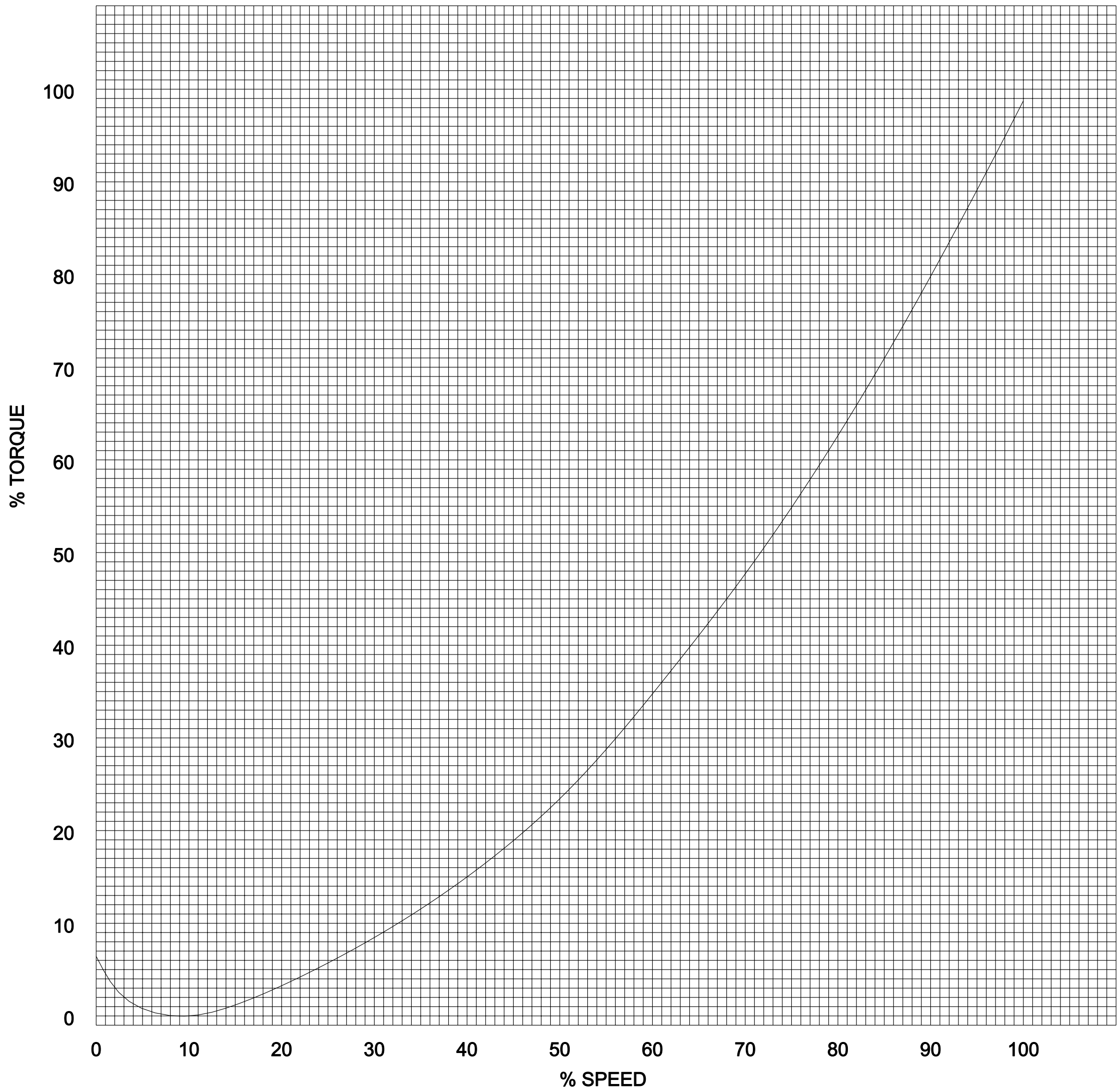
OVERALL EFFICIENCY PUMP UNIT	KILOWATTS PER 1000 GALLONS AT ONE FOOT TDH	OVERALL EFFICIENCY PUMP UNIT	KILOWATTS PER 1000 GALLONS AT ONE FOOT TDH
32	0.00980	62	0.00506
33	0.00951	63	0.00498
34	0.00922	64	0.00490
35	0.00896	65	0.00482
36	0.00871	66	0.00475
37	0.00848	67	0.00468
38	0.00826	68	0.00461
39	0.00804	69	0.00454
40	0.00784	70	0.00448
41	0.00765	71	0.00442
42	0.00747	72	0.00435
43	0.00730	73	0.00430
44	0.00713	74	0.00424
45	0.00697	75	0.00418
46	0.00682	76	0.00413
47	0.00667	77	0.00407
48	0.00653	78	0.00402
49	0.00640	79	0.00397
50	0.00627	80	0.00392
51	0.00615	81	0.00387
52	0.00603	82	0.00382
53	0.00592	83	0.00378
54	0.00581	84	0.00373
55	0.00570	85	0.00369
56	0.00560	86	0.00365
57	0.00550	87	0.00360
58	0.00541	88	0.00356
59	0.00532	89	0.00352
60	0.00523	90	0.00348
61	0.00514		

Overall efficiency as indicated is the input-output efficiency including all losses in the pump unit, pumping 1000 gallons of clear water one foot of total head. Therefore, in determining the kilowatts per 1000 gallon pumped, it is only necessary to multiply the factor corresponding to the overall efficiency by the number of feet head at which the total dynamic head has been calculated.

EXAMPLE: Assume an overall efficiency of 65% and a total head of 200 feet.

Kilowatts per 1000 gallons = 0.00482 x 200 = 0.964

PUMP SPEED TORQUE CURVE AND FORMULA



The driver must be capable of supplying more torque at each successive speed from zero to full load RPM than what is required by the pump in order to reach rated speed. This means that the Speed - Torque curve of the driver must not intersect the pump torque curve anywhere on the curve before 100% speed is reached.

$$\text{Torque (ft.-lbs.)} = \frac{(5250)\ (\text{BHP})}{\text{RPM}}$$

Torque varies as the square of the speed; therefore, to obtain torque at:
3/4 speed - multiply full speed torque by 0.563
1/2 speed - multiply full speed torque by 0.250
1/4 speed - multiply full load speed torque by 0.063
1/8 speed - multiply full load speed torque by 0.016

NOTE: If the pump starts against a closed valve, use the shut off BHP for torque calculation.

*Based on Hydraulic Institute Standards.

CORROSION

Webster defines it as the "action or process or effect of corroding" and corroding is defined as "eating away by degrees by chemical action". Pertaining to pumps, we should think of corrosion as any action which is detrimental to
its performance. Thus, a pump's performance can be destroyed by corrosion, abrasion or plugging action of solids, or a combination of the above.

The usual is to think in terms of only one of these conditions being present and affecting the pump's performance. In the case of metallic corrosion where this is true, the solution is one of selection of materials in contact with the fluid which will corrode very slowly. Vertical pumps present additional considerations in that because of costs, further refinement between rotating and non-rotating, wetted parts may require examination. Many coatings have been developed and we will not attempt to advise on the proper selection or application in this writing.

The area where we notice the greatest difficulties are those where corrosion and abrasion are present. All metals gain their projection against corrosive agents by a thin skin of corrosion. If this skin is wiped clean, the metal will recorrode, forming a new skin. Thus, if minor amounts of abrasion are found in the corrosive fluid, it is possible for these to cause this wiping action. This will occur whenever the pumped fluid abrasive content is harder than the skin corrosion or coating when it is used. Thus, soft abrasives can be present in large quantities without causing excessive wear. This condition is further activated by the velocity within the pump. By slowing the pump speed or oversizing the pump for the design conditions, the internal velocity is lowered. This will reduce the abrasive wear and can be considered on some corrosion-abrasion problems.

This brings us to the second combination which can cause difficulties. A vertical pump utilizes sleeve bearings inside a bearing retainer for proper shaft support. The straightness of this shafting is the secret of long vertical pump life. Each shaft runs with a thin film of fluid around it. By centrifugal force, this film is kept even in all bearings, thus assuring lubrication and balance in the pump. Anything which upsets this balance can cause difficulties. Thus, abrasion can wipe away bearing surfaces causing the bearing to become elliptical. The shaft then tends to destroy all bearings by its own out-of balance action and the pump fails. This same difficulty can develop in chemical fluids which tend to precipitate out. Then this solid material falling into the pump's path in large quantities causes an imbalance within the pump. This will destroy it over a period of time even when the material has a lower brinell number than the pump components.

The lack of lubricating qualities within the fluid can cause the same difficulties in the bearings. This is overcome by injection of a fluid with good lubricating qualities at each bearing journal. This injected fluid must be compatible with the fluid being pumped even though only small quantities are used. When abrasives are present in corrosive fluids, this system can be utilized by taking the pumped fluid through a small centrifugal separator, building the pressure with a small pump and lubricating the packing nut and the enclosed line shaft. This pressure should be approximately 12/15 PSI above the maximum design pressure. The shaft should be in an enclosed tube or with fluid injected through the spider to the line shaft bearing. By bringing a flushing line down to the tail bearing and rifle drilling the shaft, bearing holes can be bored at each of the bowl bearings. This will insure that clean fluid is being used for lubrication at each bearing. This flow will be away from the bearing, thus keeping all abrasives out in the main fluid flow passage.

One of the most difficult corrosive fluids to handle and understand is sea water. This is because of several variables which can alter the effects of this fluid upon different metals being employed. The first consideration is temperature. All corrosive fluids are more active as the temperature is elevated. Therefore, where cast iron might be used successfully at 30°F, the story changes at 90°F. The other chemicals in sea water can cause difficulty if their presence is not known. Around oil docks, drilling, etc., sulfides might be present. Even though found in small quantities, this can cause the fluid to be much more corrosive than just the primary fluid. The other consideration is the quantity of sand present. Offshore installations are subject to tides and wave action. This can cause the sand content to continually change and therefore, present a difficult system to analyze. The electolitic action of dissimilar metals in the presence of the sea water must also be taken into account.

The best corrosive defense is your own experience on any given unit. Thus, good records are a necessity. Each pump should be checked for vibration and amperage periodically. This information, along with the shut-off head, should be noted in the permanent record. Any changes should be cause for investigation. Any repairs should be noted with complete description of parts used, materials and conditions of the parts being replaced. With this type of record, it is possible to ascertain improvements in performance and be aware of what materials or action brought them about. Without this information it is not possible to be certain that the solution is correct for a particular application and expensive parts can be lost. Corrosion is a never-ceasing battle to extend the life of equipment to acceptable economic limits, and good records are essential for proper determination. A qualified J-Line sales engineer stands ready to assist you in this endeavor at all times - give us a call.

AMERICAN-MARSH PUMPS

CONVERSION TABLES

UNITS OF PRESSURE AND HEAD

UNITS	LBS. PER SQUARE INCH	FEET OF WATER	METERS OF WATER	INCHES OF MERCURY	ATMOS-PHERES	KILOGRAMS PER SQ. C.M.
1 LB. PER SQ. INCH	1.	2.31	0.704	2.04	0.0681	0.0703
1 FT. OF WATER	0.433	1.	0.305	0.882	0.02947	0.0305
1 METER OF WATER	1.421	3.28	1.	2.89	0.0967	0.1
1 INCH OF MERCURY	0.491	1.134	0.3456	1.	0.0334	0.0345
1 ATMOSPHERE (AT SEA LEVEL)	14.70	33.93	10.34	29.92	1.	1.033
1 KILOGRAM PER SQ. C.M.	14.22	32.8	10.	28.96	0.968	1.

Equivalent units are based on density of fresh water from 32° to 62°F.
Equivalent units are based on density of mercury from 32° to 62° F. - sufficient accuracy.
Each 1000 ft. of ascent decreases pressure about 1/2 lb./sq. in.

UNITS OF VOLUME AND WEIGHT

UNITS	U.S. GALLONS	IMPERIAL GALLONS	CUBIC INCHES	CUBIC FEET	ACRE FEET	* POUNDS	CUBIC METERS	LITERS
1 U.S. GALLON	1.	0.833	231.	0.1337	0.00000307	8.35	0.003785	3.785
1 IMPERIAL GALLON	1.201	1.	277.4	0.1605	0.00000369	10.02	0.004546	4.546
1 CUBIC INCH	0.00433	0.00360	1.	0.000579		0.0361		0.0164
1 CUBIC FOOT	7.48	6.23	1728.	1.	0.0000230	62.4	0.02832	28.32
1 ACRE - FOOT	325.850	271.335		43.560	1.		1233.5	
1 POUND	0.120	0.0998	27.7	0.0160		1.		0.454
1 CUBIC METER	264.2	220.	61.023	35.314	0.000811	2205.	1.	1000.
1 LITER	0.2642	0.220	0.061023	0.0353		2.205		1.

*WTS. Shown based on maximum density of fresh water at 39° Fahrenheit.

UNITS OF AREA

UNITS	SQUARE INCH	SQUARE FEET	SQUARE YARD	ACRES	SQUARE MILES	SQUARE CENTI-METER	SQUARE METERS	HECTARES
1 SQUARE INCH	1.	0.00694	0.00077			6.452		
1 SQUARE FOOT	144.	1.	0.111			929.	0.0929	
1 SQUARE YARD	1296.	9	1.	0.000207		8361.	0.836	
1 ACRE		43,560.	4840.	1.	0.00156		4049.	0.405
1 SQUARE MILE		27,878,400.	3,097,600.	640.	1.		2,593,388.16	258.
1 SQUARE CENTIMETER	0.155	0.001076				1.	0.0001	
1 SQUARE METER	1549.	10.76	1.196	0.000247		10.000	1.	0.0001
1 HECTARE		107,639.	11,960.	2.471	0.00386	99,996,631.	10,000.	1.

CONVERSION TABLES

UNITS OF FLOW

UNITS	U.S. GALLONS PER MINUTE	MILLION U.S. GALLONS PER DAY	CUBIC FEET PER SECOND	CUBIC METERS PER HOUR	LITERS PER SECOND	MINER'S INCH*		
						I	II	III
1 U.S. GALLON PER MINUTE	1.	0.001440	0.00223	0.2270	0.0631	0.0891	0.1114	0.0856
1 MILLION U.S. GALLONS PER DAY	694.5	1.	1.547	157.73	43.8	61.9	77.4	59.4
1 CU. FOOT PER SEC.	448.8	0.646	1.	101.9	28.32	40.	50.	38.4
1 CU. METER PER HOUR	4.403	0.00634	0.00981	1.	0.2778			
1 LITER PER SEC.	15.85	0.0228	0.0353	3.60	1.			
1 MINER'S INCH* I	11.22	0.01618	0.0250					
1 MINER'S INCH* II	8.98	0.01294	0.0200					
1 MINER'S INCH* III	11.69	0.01682	0.0260					
1 MINER'S INCH* IV	12.58	0.01811	0.0280					

*The Miner's Inch varies in definition in different areas. State laws have set the value as follow:
I. 40 M.I. = 1 CFS. in Arizona, California, Montana and Oregon
II. 50 M.I. = 1 CFS. in Idaho, Nebraska, Nevada, New Mexico, Utah, Kansas, S. Dakota, N. Dakota
III. 38.4 M.I. = 1 CFS. in Colorado
IV. 35.7 M.I. = 1 CFS. in British Columbia
Usual practice in Southern California, 50 M.I. = 1 CFS = 448.8 GPM

UNITS OF POWER

UNITS	HORSE-POWER	FT.-LBS. PER MINUTE	WATTS	KILOWATTS	METRIC HORSE-POWER	B.T.U. PER MINUTE
1 HORSEPOWER	1.	33,000.	746.	0.746	1.014	42.4
1 FT.-LB. PER MINUTE	0.0000303	1.	0.0226	0.0000226	0.0000307	0.001285
1 WATT	0.001340	44.2	1.	0.001	0.001360	0.0568
1 KILOWATT	1.341	44,250.	1000.	1.	1.360	56.8
1 METRIC HORSEPOWER	0.986	32,550.	736.	0.736	1.	41.8
1 BTU PER MINUTE	0.0236	778.4	17.6	0.0176	0.0239	1.

UNITS OF LENGTH

1 Inch = 0.0833 ft. = 0.0278 yd. = 25.4 millimeters = 2.54 centimeters
1 Foot = 12 inches = 0.333 yd. = 30.48 centimeters = 0.3048 meters
1 Yard = 36 inches = 3 feet = 91.44 centimeters = 0.9144 meters
1 Mile = 5280 ft. = 1760 yds. = 1.61 kilometer = 1609 meters
1 Meter = 3.281 ft. = 39.37 in. = 0.000622 miles = 0.001 kilometers
1 Kilometer = 1000 meters = 1093.61 yds. = 0.62137 miles = 3281 feet

AMERICAN-MARSH PUMPS

TURBINE COLUMN FRICTION LOSS TABLE

COLUMN FRICTION LOSS (IN FEET) PER 100 FEET OF COLUMN

COL. SIZE	3"	4"			5"			6"		
SHAFT SIZE	3/4"	3/4"	1"	1 3/16"	3/4"	1"	1 3/16"	1"	1 3/16"	1 1/2"
TUBE SIZE	1 1/4"	1 1/4"	1 1/2"	2"	1 1/4"	1 1/2"	2"	1 1/2"	2"	2 1/2"
G.P.M.										
25	1.8									
50	4.6	0.65	0.86	1.6						
75	9.0	1.3	1.7	3.3						
100	14.0	2.2	2.8	5.3	0.54	0.65	0.94			
125		3.2	4.2	7.8	0.81	0.96	1.4			
150		4.4	5.8	10.6	1.1	1.3	1.9			
175		5.8	7.5	13.8	1.5	1.7	2.5			
200		7.3	9.4	17.1	1.8	2.2	3.1	0.73	0.96	1.4
225		9.0	12.0	21.1	2.3	2.7	3.9	0.90	1.2	1.7
250		10.9	14.0		2.7	3.3	4.7	1.1	1.4	2.0
275		13.0	16.8		3.3	3.9	5.6	1.3	1.7	2.4
300		15.2	19.2		3.8	4.5	6.4	1.5	2.0	2.8
325		19.8			4.4	5.2	7.4	1.7	2.3	3.2
350					5.0	6.0	8.4	2.0	2.6	3.6
375					5.6	6.7	9.5	2.2	2.9	4.1
400					6.3	7.5	10.6	2.5	3.3	4.6
450					7.8	9.3	13.1	3.1	4.1	5.7
500					9.2	11.2	15.7	3.7	5.0	6.9
550					11.0	13.2	18.6	4.4	5.8	8.1
600					12.9	15.5		5.2	6.8	9.5
650					14.8	20.3		6.0	7.9	11.0
700					16.8			6.9	9.1	12.5
750					19.0			7.9	10.3	14.1
800								8.8	11.5	15.7
850								9.9	12.8	17.7
900								11.0	14.3	19.5
950								12.1	15.8	21.5

COL. SIZE	8"			10"			12"				COL. SIZE
SHAFT SIZE	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
TUBE SIZE	2"	2 1/2"	3"	2"	2 1/2"	3'	2"	2 1/2"	3"	3 1/2"	TUBE SIZE
G.P.M.											G.P.M.
25											25
50											50
75											75
100											100
125											125
150											150
175											175
200											200
225											225
250											250
275											275
300											300
325											325
350											350
375											375
400	0.61	0.74	1.0								400
450	0.77	0.91	1.3								450
500	0.93	1.1	1.5								500
550	1.1	1.3	1.8								550
600	1.3	1.5	2.1								600
650	1.5	1.8	2.5								650
700	1.7	2.0	2.8								700
750	1.9	2.3	3.2								750
800	2.2	2.6	3.6	0.57	0.65	0.77					800
850	2.4	2.9	4.0	0.63	0.72	0.86					850
900	2.7	3.2	4.5	0.70	0.80	0.96					900
950	2.9	3.5	4.9	0.77	0.88	1.1					950
1000	3.2	3.9	5.4	0.85	0.97	1.2	0.34	0.38	0.44	0.50	1000
1200	4.5	5.4	7.6	1.2	1.4	1.6	0.47	0.54	0.62	0.71	1200
1400	6.0	7.2	10.0	1.6	1.8	2.2	0.62	0.71	0.82	0.94	1400
1600	7.6	9.1	13.0	2.0	2.3	2.8	0.80	0.90	1.1	1.2	1600
1800	9.4	11.0	15.7	2.5	2.8	3.4	0.99	1.1	1.3	1.5	1800
2000	11.0	13.0	19.2	3.0	3.5	4.2	1.2	1.4	1.6	1.8	2000
2200	13.2	16.5	22.9	3.6	4.1	5.0	1.4	1.6	1.9	2.1	2200
2400	15.5	19.3		4.2	4.9	5.8	1.7	1.9	2.2	2.5	2400
2600	17.9	22.4		4.9	5.6	6.8	1.9	2.2	2.5	2.9	2600
2800	20.5			5.6	6.4	7.8	2.2	2.5	2.8	3.3	2800
3000				6.4	7.4	8.8	2.5	2.9	3.3	3.8	3000
3200				7.1	8.1	9.9	2.8	3.2	3.7	4.3	3200
3400				7.9	9.0	11.1	3.2	3.6	4.2	4.8	3400
3600				8.8	10.0	12.4	3.5	4.0	4.7	5.3	3600
3800				9.8	11.1	13.7	3.9	4.4	5.1	5.9	3800
4000				10.7	12.2	15.0	4.3	4.9	5.6	6.4	4000
4200				11.8	13.4	16.4	4.7	5.3	6.2	7.1	4200
4400				12.9	14.6	17.9	5.1	5.8	6.7	7.7	4400
4600				13.9	15.8	19.3	5.6	6.3	7.4	8.4	4600
4800				15.0	17.2	21.0	6.0	6.8	7.9	9.0	4800

The following table will serve as a guide in figuring column friction losses to allow for condition of pipe:

Condition of Pipe Inside	Approx. Age of Pipe in Ordinary Use for Cold Clear Water	Use Friction Loss Figures from table multiplied by the following factor
Very Smooth	New	1.00
Fairly Smooth	1 - 5 years	1.51
Rough	6 or more years	2.35

AMERICAN-MARSH PUMPS

FRICTION LOSSES IN STEEL PIPES-3 THROUGH 36 INCH

3 INCH

3" 3.00 I.D.

U.S. GALLONS PER MIN.	VELOCITY FT. PER SECOND	HEAD LOSS FT. HEAD/ 100 FT.
40	1.82	0.91
60	2.72	1.94
80	3.63	3.30
100	4.54	4.98
120	5.45	6.98
140	6.35	9.28
160	7.26	11.9
180	8.16	14.8
200	9.08	18.0
250	11.3	27.1
300	13.6	38.0
350	15.9	50.5
400	18.2	64.7
450	20.5	80.5
500	22.7	97.8

4 INCH

4" 4.026 I.D.

U.S. GALLONS PER MIN.	VELOCITY FT. PER SECOND	FRICTION LOSS FT. HEAD/100 FT. PIPE	
		STEEL C=100	PVC C=150
50	1.26	0.33	0.16
75	1.89	0.70	0.34
100	2.52	1.19	0.58
125	3.15	1.80	0.87
150	3.78	2.53	1.22
200	5.05	4.29	2.08
250	6.30	6.45	3.12
300	7.57	9.09	4.41
350	8.82	12.0	5.82
400	10.1	15.5	7.52
500	12.6	23.4	11.3
600	15.1	32.8	15.9
700	17.6	43.6	21.1
800	20.2	55.8	27.0
900	22.7	69.3	33.6

5 INCH

5" 5.00 I.D.

U.S. GALLONS PER MIN.	VELOCITY FT. PER SECOND	HEAD LOSS FT. HEAD/ 100 FT.
75	1.22	0.24
100	1.63	0.42
125	2.04	0.63
150	2.45	0.88
175	2.86	1.17
200	3.27	1.50
250	4.09	2.27
300	4.90	3.17
400	6.54	5.39
500	8.17	8.15
600	9.80	11.7
700	11.4	15.2
800	13.1	19.4
1000	16.3	29.4
1200	19.6	41.1

6 INCH

6" 6.065 I.D.

U.S. GALLONS PER MIN.	VELOCITY FT. PER SECOND	FRICTION LOSS FT. HEAD/100 FT. PIPE	
		STEEL C=100	PVC C=150
100	1.11	0.16	0.07
150	1.67	0.34	0.16
200	2.22	0.58	0.28
250	2.77	0.89	0.42
300	3.33	1.24	0.58
400	4.44	2.11	0.99
500	5.55	3.19	1.50
600	6.66	4.46	2.10
700	7.78	5.93	2.79
800	8.90	7.60	3.60
900	10.0	9.44	4.46
1000	11.1	11.5	5.44
1200	13.3	16.1	7.62
1400	15.6	21.4	10.1
1600	17.8	27.4	13.0

8 INCH

8" STD. WT. STEEL 7.981" I.D.

FLOW GALLONS PER MIN.	VELOCITY FT. PER SECOND	VELOCITY HEAD FT.	HEAD FT. PER 100 FT.
130	0.83	0.01	0.069
140	0.90	0.01	0.079
150	0.96	0.01	0.091
160	1.03	0.02	0.102
170	1.09	0.02	0.114
180	1.15	0.02	0.126
190	1.22	0.02	0.140
200	1.28	0.03	0.154
220	1.41	0.03	0.183
240	1.54	0.04	0.215
260	1.67	0.04	0.250
280	1.80	0.05	0.286
300	1.92	0.06	0.325
350	2.24	0.08	0.433
400	2.57	0.10	0.554
450	2.88	0.13	0.689
500	3.20	0.16	0.838
550	3.52	0.19	0.999
600	3.85	0.23	1.170
650	4.17	0.27	1.360
700	4.49	0.31	1.560
750	4.81	0.36	1.770
800	5.13	0.41	1.990
850	5.45	0.46	2.230
900	5.77	0.52	2.480
950	6.09	0.58	2.740
1000	6.41	0.64	3.020
1100	7.05	0.77	3.600
1200	7.69	0.92	4.230
1300	8.33	1.08	4.900
1400	8.97	1.25	5.620
1500	9.61	1.44	6.390
1600	10.30	1.70	7.200
1800	11.50	2.10	8.950
2000	12.80	2.50	10.900
2200	14.10	3.10	13.000
2400	15.40	3.70	15.200
2600	16.70	4.30	17.700
2800	18.00	5.00	20.300
3000	19.20	5.70	23.000
3500	22.40	7.80	30.600
4000	25.60	10.20	39.200
4500	28.80	12.90	48.800
5000	32.00	15.90	59.300
5500	35.30	19.40	70.700

10 INCH

10" STD. WT. STEEL 10.02" I.D.

FLOW GALLONS PER MIN.	VELOCITY FT. PER SECOND	VELOCITY HEAD FT.	HEAD FT. PER 100 FT.
180	0.73	0.01	0.042
200	0.81	0.01	0.051
220	0.89	0.01	0.061
240	0.98	0.01	0.071
260	1.06	0.02	0.083
280	1.14	0.02	0.095
300	1.22	0.02	0.108
350	1.42	0.03	0.143
400	1.63	0.04	0.183
450	1.83	0.05	0.228
500	2.04	0.06	0.277
550	2.24	0.08	0.330
600	2.44	0.09	0.388
650	2.64	0.11	0.450
700	2.85	0.13	0.516
800	3.25	0.16	0.660
900	3.66	0.21	0.821
1000	4.07	0.26	0.998
1100	4.48	0.31	1.190
1200	4.89	0.37	1.400
1300	5.30	0.44	1.620
1400	5.70	0.50	1.860
1500	6.10	0.58	2.110
1600	6.51	0.66	2.380
1700	6.92	0.74	2.660
1800	7.32	0.83	2.960
1900	7.73	0.93	3.270
2000	8.14	1.03	3.600
2200	8.95	1.24	4.290
2400	9.76	1.48	5.040
2600	10.60	1.70	5.840
2800	11.40	2.00	6.700
3000	12.20	2.30	7.610
3200	13.00	2.70	8.580
3400	13.80	3.00	9.600
3600	14.60	3.30	10.700
3800	15.50	3.70	11.800
4000	16.30	4.10	13.000
4500	18.30	5.20	16.100
5000	20.30	6.40	19.600
5500	22.40	7.80	23.400
6000	24.40	9.30	27.500
6500	26.40	10.8	31.800
7000	28.50	12.6	36.500
7500	30.50	14.5	41.500

12 INCH

12" STD. WT. STEEL 12.000" I.D.

FLOW GALLONS PER MIN.	VELOCITY FT. PER SECOND	VELOCITY HEAD FT.	HEAD FT. PER 100 FT.
200	0.57	0.01	0.021
250	0.71	0.01	0.032
300	0.85	0.01	0.045
350	0.99	0.02	0.059
400	1.14	0.02	0.076
450	1.28	0.03	0.095
500	1.42	0.03	0.115
550	1.56	0.04	0.137
600	1.70	0.05	0.161
700	1.99	0.06	0.214
800	2.27	0.08	0.275
900	2.56	0.10	0.341
1000	2.84	0.13	0.415
1100	3.12	0.15	0.495
1200	3.41	0.18	0.581
1300	3.69	0.21	0.674
1400	3.98	0.25	0.773
1500	4.26	0.28	0.878
1600	4.55	0.32	0.990
1800	5.11	0.41	1.230
2000	5.68	0.50	1.500
2200	6.25	0.61	1.780
2400	6.81	0.72	2.100
2600	7.38	0.85	2.430
2800	7.95	0.98	2.780
3000	8.52	1.13	3.170
3500	9.95	1.54	4.210
4000	11.4	2.00	5.390
4500	12.8	2.50	6.700
5000	14.2	3.10	8.150
5500	15.6	3.80	9.720
6000	17.0	4.50	11.400
6500	18.4	5.30	13.200
7000	19.9	6.20	15.200
7500	21.3	7.10	17.300
8000	22.7	8.00	19.400
8500	24.2	9.10	21.700
9000	25.6	10.2	24.200
9500	27.0	11.3	26.700
10000	28.4	12.5	29.400
11000	31.2	15.1	35.000
12000	34.1	18.1	41.200
13000	36.9	21.2	47.700
14000	39.8	24.6	54.700
15000	42.6	28.2	62.200

14 INCH

14" STD. WT. STEEL 13.25" I.D.

FLOW GALLONS PER MIN.	VELOCITY FT. PER SECOND	VELOCITY HEAD FT.	HEAD FT. PER 100 FT.
300	0.70	0.01	0.028
400	0.93	0.01	0.047
500	1.16	0.02	0.071
600	1.40	0.03	0.100
700	1.63	0.04	0.132
800	1.86	0.05	0.170
900	2.09	0.07	0.211
1000	2.33	0.08	0.256
1100	2.56	0.10	0.306
1200	2.79	0.12	0.359
1300	3.02	0.14	0.416
1400	3.26	0.17	0.477
1500	3.49	0.19	0.542
1600	3.72	0.22	0.611
1700	3.95	0.24	0.684
1800	4.19	0.27	0.760
1900	4.42	0.30	0.840
2000	4.65	0.34	0.924
2500	5.81	0.52	1.400
3000	6.98	0.76	1.960
3500	8.15	1.03	2.600
4000	9.31	1.35	3.320
4500	10.5	1.70	4.130
5000	11.6	2.10	5.030
6000	14.0	3.00	7.050
7000	16.3	4.10	9.380
8000	18.6	5.40	12.000
9000	20.9	6.80	14.900
10000	23.3	8.40	18.100
11000	25.6	10.2	21.600
12000	27.9	12.1	25.400
13000	30.2	14.2	29.500
14000	32.6	16.5	33.800
15000	34.9	18.9	38.400
16000	37.2	21.5	43.300
17000	39.5	24.2	48.400
18000	41.9	27.3	53.800
20000	46.5	33.6	65.400
22000	51.2	40.7	78.000
24000	55.8	48.4	91.600

FRICTION LOSSES IN STEEL PIPES-3 THROUGH 36 INCH (CONT.)

16 INCH

16" STD. WT. STEEL 16.000" I.D.

FLOW GALLONS PER MIN.	VELOCITY FT. PER SECOND	VELOCITY HEAD FT.	HEAD FT. PER 100 FT.
500	0.88	0.01	0.036
600	1.05	0.02	0.050
700	1.23	0.02	0.067
800	1.41	0.03	0.086
900	1.58	0.04	0.106
1000	1.76	0.05	0.129
1200	2.11	0.07	0.181
1400	2.46	0.09	0.241
1600	2.81	0.12	0.308
1800	3.16	0.16	0.383
2000	3.51	0.19	0.466
2500	4.39	0.30	0.704
3000	5.27	0.43	0.987
3500	6.15	0.59	1.310
4000	7.03	0.77	1.680
4500	7.91	0.97	2.090
5000	8.79	1.20	2.540
6000	10.5	1.70	3.560
7000	12.3	2.40	4.730
8000	14.1	3.10	6.060
9000	15.8	3.90	7.530
10000	17.6	4.80	9.150
11000	19.3	5.80	10.900
12000	21.1	6.90	12.800
13000	22.8	8.10	14.900
14000	24.6	9.40	17.100
15000	26.3	10.7	19.200
16000	28.1	12.3	21.800
18000	31.6	15.5	27.100
20000	35.1	19.1	33.000
22000	38.7	23.3	39.300
24000	42.2	27.7	46.200
25000	43.9	30.0	49.900
26000	45.7	32.6	53.600
28000	49.2	37.6	61.500
30000	52.7	43.2	69.800
32000	56.2	49.1	78.700
34000	59.8	55.6	88.300
36000	63.3	62.3	97.600
38000	66.8	69.3	108.000

18 INCH

18" STD. WT. STEEL 17.18" I.D.

FLOW GALLONS PER MIN.	VELOCITY FT. PER SECOND	VELOCITY HEAD FT.	HEAD FT. PER 100 FT.
500	0.69	0.01	0.020
600	0.83	0.01	0.028
700	0.97	0.01	0.037
800	1.11	0.02	0.048
900	1.25	0.02	0.060
1000	1.38	0.03	0.072
1200	1.66	0.04	0.101
1400	1.94	0.06	0.135
1600	2.21	0.08	0.173
1800	2.49	0.10	0.215
2000	2.77	0.12	0.261
2500	3.46	0.19	0.394
3000	4.15	0.27	0.553
3500	4.85	0.37	0.735
4000	5.54	0.48	0.941
4500	6.23	0.60	1.170
5000	6.92	0.74	1.420
6000	8.31	1.10	1.990
7000	9.70	1.50	2.650
8000	11.1	1.90	3.390
9000	12.5	2.40	4.220
10000	13.8	3.00	5.120
12000	16.6	4.30	7.180
14000	19.4	5.80	9.550
16000	22.1	7.60	12.200
18000	24.9	9.60	15.200
20000	27.7	11.9	18.500
22000	30.3	14.3	22.000
24000	33.2	17.1	25.900
26000	36.0	20.1	30.000
28000	38.8	23.4	34.400
30000	41.5	26.8	39.100
32000	44.3	30.5	44.100
34000	47.1	34.5	49.400
36000	49.9	38.7	54.800
38000	52.6	43.0	60.600
40000	55.4	47.1	66.600
42000	58.2	52.6	72.900
44000	61.0	57.8	79.400
46000	63.7	63.0	86.300

20 INCH

20" STD. WT. STEEL 19.18" I.D.

FLOW GALLONS PER MIN.	VELOCITY FT. PER SECOND	VELOCITY HEAD FT.	HEAD FT. PER 100 FT.
800	0.89	0.01	0.028
1000	1.11	0.02	0.042
1200	1.33	0.03	0.059
1400	1.55	0.04	0.079
1500	1.78	0.05	0.101
1800	2.00	0.06	0.126
2000	2.22	0.08	0.153
2500	2.78	0.12	0.231
3000	3.33	0.17	0.323
3500	3.89	0.24	0.430
4000	4.45	0.31	0.551
5000	5.55	0.48	0.832
6000	6.67	0.69	1.170
7000	7.78	0.94	1.550
8000	8.89	1.20	1.980
10000	11.1	1.90	3.000
12000	13.3	2.70	4.200
14000	15.5	3.70	5.590
15000	16.7	4.30	6.350
16000	17.8	4.90	7.150
18000	20.0	6.20	8.900
20000	22.2	7.70	10.800
22000	24.4	9.30	12.900
24000	26.7	11.1	15.100
25000	27.8	12.0	16.300
26000	28.9	13.0	17.600
28000	31.1	15.0	20.100
30000	33.3	17.2	22.900
32000	35.6	19.7	25.800
34000	37.8	22.2	28.900
35000	38.9	23.5	30.400
36000	40.0	24.9	32.100
38000	42.2	27.7	35.400
40000	44.5	30.8	39.000
45000	50.0	38.9	48.500
50000	55.5	47.9	58.900
55000	61.1	58.0	70.300
60000	66.7	69.0	82.300
65000	72.2	81.0	95.700
70000	77.8	94.0	110.000

24 INCH

24" STD. WT. STEEL 24" I.D.

FLOW GALLONS PER MIN.	VELOCITY FT. PER SECOND	VELOCITY HEAD FT.	HEAD FT. PER 100 FT.
350	0.252	0.00	0.002
700	0.495	0.00	0.007
1000	0.712	0.01	0.014
1400	0.999	0.01	0.026
1700	1.210	0.02	0.038
2000	1.420	0.03	0.051
2400	1.700	0.045	0.071
2700	1.920	0.060	0.089
3100	2.210	0.080	0.114
3400	2.410	0.090	0.136
3800	2.700	0.113	0.167
4200	2.990	0.130	0.200
4500	3.200	0.160	0.226
4800	3.410	0.180	0.255
5200	3.690	0.210	0.298
5500	3.910	0.240	0.329
5900	4.190	0.270	0.377
6200	4.410	0.320	0.413
6500	4.620	0.330	0.450
6900	4.900	0.370	0.502
7600	5.400	0.500	0.603
8300	5.900	0.540	0.713
9000	6.400	0.640	0.820
9700	6.900	0.740	0.950
10000	7.110	0.790	1.000
11000	7.820	0.950	1.190
12000	8.550	1.140	1.400
12500	8.860	1.220	1.520
13000	9.250	1.340	1.630
14000	9.950	1.540	1.860
15000	10.630	1.770	2.110
16000	11.380	2.020	2.420
18000	12.800	2.570	2.980
19000	13.500	2.850	3.280
20000	14.200	3.160	3.610

30 INCH

30" STD. WT. STEEL 30" I.D.

FLOW GALLONS PER MIN.	VELOCITY FT. PER SECOND	VELOCITY HEAD FT.	HEAD FT. PER 100 FT.
700	0.322	0.00	0.002
1000	0.450	0.00	0.004
1300	0.590	0.01	0.008
1700	0.780	0.01	0.013
2000	0.910	0.01	0.017
2400	1.095	0.02	0.023
2700	1.230	0.02	0.030
3100	1.430	0.03	0.039
3400	1.550	0.04	0.046
3800	1.740	0.05	0.057
4100	1.870	0.06	0.065
4500	2.050	0.07	0.077
4800	2.190	0.08	0.087
5200	2.370	0.09	0.101
5500	2.510	0.10	0.112
5900	2.690	0.11	0.127
6200	2.830	0.12	0.139
6900	3.120	0.15	0.170
7600	3.470	0.18	0.203
8300	3.790	0.22	0.240
9000	4.100	0.26	0.278
9700	4.420	0.306	0.319
10000	4.560	0.325	0.337
11000	5.010	0.392	0.401
12000	5.470	0.47	0.473
12500	5.700	0.51	0.510
13000	5.940	0.55	0.550
14000	6.400	0.64	0.630
15000	6.850	0.73	0.710
16000	7.300	0.83	0.810
18000	8.200	1.05	1.000
19000	8.670	1.17	1.110
20000	9.120	1.30	1.220
24000	10.090	1.89	1.730
28000	12.750	2.46	2.270

36 INCH

36" STD. WT. STEEL 36" I.D.

FLOW GALLONS PER MIN.	VELOCITY FT. PER SECOND	VELOCITY HEAD FT.	HEAD FT. PER 100 FT.
1400	0.44	0.00	0.004
1700	0.53	0.00	0.005
2000	0.63	0.01	0.007
2400	0.75	0.01	0.010
2800	0.88	0.01	0.015
3400	1.07	0.02	0.019
4000	1.26	0.02	0.026
4800	1.51	0.04	0.036
5600	1.76	0.05	0.048
6200	1.95	0.06	0.057
7000	2.20	0.07	0.072
7600	2.39	0.09	0.084
8300	2.61	0.10	0.098
9000	2.83	0.12	0.114
9700	3.05	0.14	0.131
10000	3.14	0.16	0.139
11000	3.46	0.19	0.164
12000	3.78	0.22	0.193
13000	4.09	0.26	0.226
14000	4.40	0.30	0.260
15000	4.71	0.34	0.294
16000	5.03	0.39	0.330
18000	5.66	0.50	0.412
19000	5.98	0.56	0.454
20000	6.30	0.61	0.504
21000	6.60	0.67	0.544
22000	6.92	0.74	0.590
23000	7.24	0.81	0.640
24000	7.55	0.88	0.695
26000	8.18	1.04	0.806
28000	8.80	1.20	0.935
30000	9.44	1.38	1.065
34000	10.70	1.77	1.340
38000	11.95	2.20	1.650
42000	13.20	2.70	1.990

AMERICAN-MARSH PUMPS

FRICTION LOSSES IN STEEL PIPES-3 THROUGH 30 INCH

FRICTION LOSSES AS EQUIVALENT LENGTHS OF PIPE - FEET

Type of fitting and application	Nominal Size of Pipe and Fitting 3"	4"	5"	6"	8"	10"	12"	14"	16"	20"	24"	30"
ELBOWS -												
90° Standard Elbow - or Run of Tee reduced by 1/2	7.7	10	13	15	20	25	30	35	40	50	61	76
90° Long Radius - or Run of Standard Tee	5.2	6.8	8.5	10	14	17	20	24	27	34	40	50
45° Standard	3.6	4.7	5.9	7.1	9.4	12	14	17	19	24	28	35
Standard Tee through Side Outlet	16	20	26	31	40	51	61	71	81	101	121	151
Gate Valve (fully open)	1.6	2.1	2.7	3.2	4.3	5.3	6.4	7.5	8.5	11	13	16
Swing Check Valve (fully open)	20	26	33	39	52	65	77	90	104	129	155	193
Ordinary Entrance	4.5	6.0	7.3	9.0	12	15	17	20	22	28	35	43

FITTINGS - Friction losses expressed as equivalent lengths of pipe (feet)

Type of Fitting	Material	1"	1 1/4"	1 1/2"	2"	2 1/2"	3"	3 1/2"	4"	5"	6"	8"	10"	12"
90° Standard Elbow, or run of tee reduced to half size	Steel	3	4	4	5	6	8	-	10	13	15	20	25	30
	Plastic	6	7	8	9	10	14	-	17	-	25	-	-	-
	Copper	3	4	4	5	6	8	9	10	13	15	20	-	-
45° Elbow	Steel	1.3	1.7	2.1	2.6	3	3.6	-	4.7	5.9	7.1	9.4	12	14
	Plastic	2.5	3	4	5	6	7	-	9	-	13	13	-	-
	Copper	1.3	1.7	2.1	2.6	3	3.6	4.1	4.7	5.9	7.1	9.4	-	-
Standard Tee with flow through branch	Steel	6	8	9	11	14	16	-	20	26	31	40	51	61
	Plastic	9	12	13	17	20	23	-	29	-	45	-	-	-
	Copper	6	8	9	11	14	16	18	20	26	31	40	-	-
90° long radius elbow, or run of standard tee	Steel	1.7	2.3	2.8	3.6	4.2	5.2	-	6.8	8.5	10	14	17	20
	Plastic	3	4	5	7	8	10	-	12	-	17	-	-	-
	Copper	1.7	2.3	2.8	3.6	4.2	5.2	6.1	6.8	8.5	10	14	-	-
Adapter-slip/solder fitting to thread Insert coupling	Steel	3	3	3	3	3	3	-	3	-	3	-	-	-
	Plastic	1	1	1	1	1	1	1	1	1	1	1	-	-
	Copper	3	3	3	3	3	3	-	3	-	3	-	-	-
Gate Valve (fully open)		0.60	0.80	0.95	1.15	1.4	1.6	1.85	2.1	2.7	3.2	4.3	5.3	6.4
Swing Check Valve		7	9	11	13	16	20	23	26	33	39	52	67	77
Ordinary entrance		1.5	2.0	2.4	3.0	3.7	4.5	5.2	6.0	7.3	9.0	12	15	17

AMERICAN-MARSH PUMPS

CAST DISCHARGE HEAD FRICTION LOSS CHART

CAST DISCHARGE HEAD FRICTION LOSS CHART

Discharge Size	Capacity in Gallons Per Minute										
	100	200	300	400	500	600	700	800	1000	1200	1400
6"	0.016	0.062	0.140	0.248	0.388	0.558	0.760	0.922	1.550		
8"				0.078	0.125	0.176	0.226	0.312	0.488	0.703	0.958
	Capacity in Gallons Per Minute										
	500	1000	1500	2000	2500	3000	3500	4000	4500	5000	
10"	0.048	0.192	0.433	0.769	1.202	1.731					
12"		0.091	0.204	0.363	0.565	0.861	1.110	1.145			
14"				0.201	0.314	0.452	0.615	0.803	1.017	1.255	

VERTICAL TURBINE FAB DISCHARGE HEAD FRICTION LOSS CHART

Discharge Size	Capacity in Gallons Per Minute										
	40	80	100	140	180	200	300	400	500	600	700
2"	0.517	2.069	3.233								
3"		0.419	0.655	1.283	2.122	2.620					
4"				0.534	0.675	0.834	1.875	3.333			
6"						0.220	0.355	0.630	0.990	1.419	1.932
	Capacity in Gallons Per Minute										
	800	1000	1200	1400	1600	1800	2000	2500	3000	4000	5000
8"	0.812	1.216	1.827	2.487	3.248						
10"		0.545	0.786	1.069	1.396	1.767	2.182	3.409			
12"				0.488	0.638	0.807	0.996	1.557	2.242		
14"						0.523	0.646	1.009	1.453	2.584	
16"								0.554	0.798	1.419	2.217

MECHANICAL FRICTION IN TURBINE PUMP LINE SHAFTS (HORSEPOWER/100 FEET SHAFT LENGTH)								
SHAFT DIA. (inches)	B.H.P. AT R.P.M.							
	3450	2900	2200	1760	1460	1160	880	700
3/4			0.38	0.30	0.25	0.22		
1	1.04	0.87	0.65	0.52	0.45	0.35		
1 3/16	1.44	1.20	0.90	0.72	0.60	0.44		
1 1/2	2.30	1.92	1.44	1.15	0.95	0.74	0.56	
1 11/16				1.40	1.20	0.92	0.70	
1 15/16				1.80	1.50	1.20	0.90	0.72
2 3/16				2.30	1.90	1.50	1.15	0.92
2 7/16				2.85	2.40	1.85	1.40	1.13

SHAFT WEIGHTS (WT./FT. - LBS.)		
SHAFT DIAMETER	ENCLOSED *(1)	OPEN
3/4	1.50	1.30
1	2.60	2.30
1 3/16	3.80	3.30
1 11/16	6.00	5.30
1 15/16	7.60	6.30
2 3/16	10.00	8.80
2 7/16	12.80	11.20

*NOTE: (1) Oil lubricated shaft does not displace liquid above the pumping water level and therefore has a greater net weight.

HARDNESS OF MATERIAL

Hardness is one of the physical properties generally considered to be an important factor in the selection of the best material for given service. Numerous methods have been developed to describe the degree of hardness of a material. Brinell, Rockwell, Vickers and Shore are the most widely used scales and can be compared by use of the conversion tables on the following pages. As you will note, various scales must be used with the Brinell and Rockwell methods in order to cover the full range of hardness. Only the more commonly used scales are listed in the table. A hard material (high hardness number) is likely to have a high tensile strength and be somewhat brittle. The tensile strength of steel can be roughly approximated by multiplying the Brinell hardness by 500.

The approximate range of Brinell hardness for a number of the metals commonly used are:

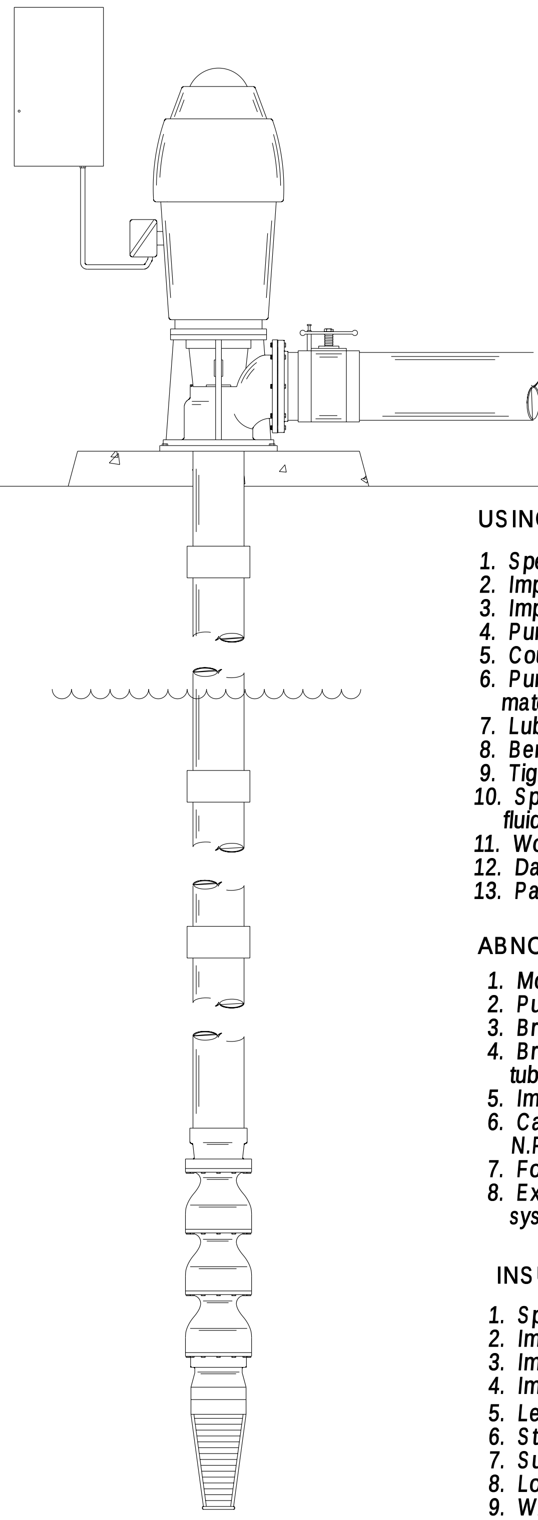
(Typical)

METAL	TENSILE STRENGTH (PSI)	BRINELL HARDNESS RANGE
CAST IRON	*30,000*	*190 - 210*
	40,000	*210 - 250*
	50,000	*230 - 280*
410 - 416 STAINLESS STEEL	*75,000 - 135,000*	*155 - 290*
DUCTILE IRON	*60,000*	*132*
1045 STEEL	*80,000 - 120,000*	*160 - 240*
316 STAINLESS STEEL	*80,000 - 90,000*	*150 - 190*
304 STAINLESS STEEL	*85,000 - 125,000*	*212 - 277*
MONEL 400	*84,000 - 120,000*	*160 - 225*
K MONEL	*135,000 - 185,000*	*255 - 370*
17-4 PH STAINLESS STEEL	*145,000 - 200,000*	*311 - 420*
ALUMINUM BRONZE	*80,000 - 110,000*	*150 - 250*
SAE 40 BRONZE	*26,000 - 42,000*	*60 - 80*
NICKEL ALUMINUM BRONZE	*85,000*	*159*

The hardness and tensile strength of many materials can be changed considerably by heat-treating or cold-working. An example of the latter is cold-drawn shafting which has harder and stronger metal near the surface of the shaft than in the center.

It is difficult to obtain accurate hardness measurements of thin sections of materials such as coatings and overlays applied over a softer material. The actual hardness is not of great importance in this instance as the wear characteristics of a material are more likely to be related to its "toughness" or the mating surface than to the hardness. This is not to say that materials with low hardness ratings have excellent wear resistance.

TROUBLE SHOOTING
OPERATING SYMPTOMS



INSUFFICIENT PRESSURE

1. Speed too slow (check voltage.)
2. Impeller trimmed incorrectly.
3. Impeller loose.
4. Impeller plugged.
5. Wear rings worn.
6. Entrained air in pump.
7. Leaking joints or bowl casings.
8. Wrong rotation.
9. Incorrect impeller adjustment.

NO LIQUID DELIVERED

1. Pump suction broken (water level) below inlet.)
2. Suction valve closed.
3. Impeller plugged.
4. Strainer clogged.
5. Wrong rotation.
6. Shaft broken or unscrewed.
7. Impeller loose.
8. Barrel or discharge not vented.
9. Driver inoperative.

VIBRATION

1. Motor imbalance-electrical.
2. Motor bearing not properly seated or worn.
3. Motor drive coupling out of balance or alignment.
4. Misalignment of pump, casings, discharge head column, or bowls.
5. Discharge head misaligned by improper mounting or pipe strain.
6. Bent shafting.
7. Worn pump bearings.
8. Clogged impeller or foreign material in pump.
9. Improper impeller adjustment.
10. Vortex problems in sump.
11. Reasonance-system frequency at or near pump speed.
12. Cavitation.
13. Impeller out of balance.

USING TOO MUCH POWER

1. Speed too high.
2. Improper impeller adjustment.
3. Improper impeller trim.
4. Pump out of alignment.
5. Coupling out of alignment.
6. Pumping sand, silt, or foreign material.
7. Lubricating oil too heavy.
8. Bent shaft.
9. Tight bearing or packing.
10. Specific gravity or viscosity of fluid higher than design.
11. Worn pump.
12. Damaged pump.
13. Partial freezing of pump liquid.

ABNORMAL NOISE

1. Motor noise.
2. Pump bearing running dry.
3. Broken column bearing retainers.
4. Broken shaft or shaft enclosing tube.
5. Impellers dragging on bowl case.
6. Cavitation due to insufficient N.P.S.H.A. and/or submergence.
7. Foreign material in pump.
8. Excessive fluid velocity in pipe system.

INSUFFICIENT CAPACITY

1. Speed too slow.
2. Impeller trimmed incorrectly.
3. Impeller loose.
4. Impeller or bowl partially plugged.
5. Leaking joints.
6. Strainer or suction pipe clogged.
7. Suction valve throttled.
8. Low water level.
9. Wrong rotation.
10. Insufficient submergence.
11. Insufficient N.P.S.H.A.
12. Incorrect impeller adjustment.
13. Worn pump.
14. System pressure higher than design.

BEARINGS

TROUBLE SOURCE	PROBABLE CAUSE	REMEDY
Premature bearing wear	Abrasive action	Consider converting to fresh water flushing on all bearings or pressure grease or oil lubrication.
Bearing seized or galling on shaft	Running dry without lubrication.	Check lubrication, look for plugged suction or evidence of flashing.
Bearing failure or bearing seized	High temperature failure.	Check pump manufacturer for bearing temperature limits.
Excessive shaft wear under rubber bearings	Rubber bearings will swell in hydro-carbon, H_2S & High temperature.	Change bearing material.
Uneven wear on bearings, uniform wear on shaft.	Pump non-rotating parts misaligned.	Check mounting & discharge pipe connection, dirt between column joints. Correct misalignment, replace bearings & repair or replace shaft.
Uniform wear on bearing and shaft	Abrasive action.	Replace parts, consider changing material or means of lubrication.
Uniform wear on bearings, uneven wear on shaft	1. Shaft runnout caused by bent shafts, shafts not butted in couplings, dirt or grease between shafts. 2. Shaft ends not properly faced.	1. Straighten shaft or replace, clean & assemble correctly. 2. Face parallel & concentric.

SHAFT AND COUPLINGS

Bent shaft	Mishandling in transit or assembly.	Check straightness. Correct to 0.0005/ft. total runout or replace.
Shaft coupling unscrewed.	Pump started in reverse rotation.	Shafts may be bent; check shafts & couplings. Correct rotation.
Shaft coupling elongated. (neck down)	1. Motor started while the pump was running in reverse. 2. Corrosion. 3. Pipe wrench fatigue on reused couplings. 4. Power being applied to shafts that are not butted in coupling.	1. Look for faulty check valve. Could also be momentary power failure or improper starting timers. 2. Replace couplings. 3. Replace couplings. 4. Check for galling on shaft ends.
Broken shaft or coupling.	1. Can be caused by same reasons listed for coupling elongation. 2. Can also be caused by bearings seized due to lack of lubrication. 3. Foreign material locking impellers or galling wear rings. 4. Metal fatigue due to vibrations. 5. Improper impeller adjustment or continuous upthrust conditions, causing impeller to drag.	1. Same as above. 2. Same as above for bearing seizure. 3. Add strainers or screens. 4. Check alignment of pump components to eliminate vibration. 5. See Sections on Impeller Adjustment and Upthrusting.

INNER COLUMN

Water in inner column	1. Bypass ports plugged. 2. Badly worn bypass seal or bearings. 3. Tubing joint leaking. 4. Crack or hole in tubing.	1. Remove cause. 2. Replace worn parts. 3. Ensure tubing joint face is clean and butted squarely. 4. Replace section affected.

COMPONENT PROBLEM SOLVING

IMPELLERS

TROUBLE SOURCE	PROBABLE CAUSE	REMEDY
Wear on exit vanes and shrouds	*Abrasive action*	*Replace impeller if excessive. Consider coating or upgrading material.*
Pitting on entrance vanes of impeller	*Cavitation:*	*Correct condition or upgrade material to extend life.*
Pitting on impellers and bowl casting.	*Corrosion/Erosion.*	*Investigate cost of different materials vs. frequency of replacements.*
Wear on impeller skirts and/or bowl seal ring area.	*1. Abrasive action or excess wear allowing impeller skirts to function as bearing journal.* *2. Impellers set to high.*	*1. Install new bearings and wear rings. Upgrade material if abrasive action.* *2. Re-ring & adjust impellers correctly.*
Impeller loose on shaft (extremely rare occurrence)	*1. Repeated shock load by surge in suction or discharge line. (Can loosen first or last stage impellers.)* *2. Foreign material jamming impeller. (May break shaft or trip overloads before impeller becomes loose.)* *3. Differential expansion due to temperature.* *4. Parts improperly machined and/or assembled.* *5. Torsion loading on submersible pumps.*	*1. Re-fit impellers. If collet mounted, consider changing to key mounting.* *2. Remove cause of jamming.* *3. If collet mounted, consider changing to key mounted. Avoid sudden thermal shock.* *4. Correct parts if necessary and re-fit.* *5. Add keyway to collet mounting.*

PACKING HOUSING

Packing Housing overheating	*1. Improper packing procedure.* *2. Packing too tight.* *3. Insufficient lubrication.* *4. Incorrect type of packing.*	*1. Repack correctly.* *2. Release gland pressure.* *3. Repack correctly.* *4. Repack with correct grade for service.*
Packing wears prematurely	*1. Improper packing procedure.* *2. Insufficient lubrication.* *3. Shaft or sleeve scored.* *4. Incorrect type of packing.* *5. Abrasives in liquid.*	*1. Repack correctly.* *2. Repack correctly.* *3. Remachine or replace scored parts.* *4. Repack with correct grade for service.* *5. Remove source of abrasives.*
Excessive leakage	*1. Improper packing procedure.* *2. Incorrect type of defective packing.* *3. Worn shaft or sleeve.*	*1. Repack correctly.* *2. Repack with correct grade for service.* *3. Remachine or replace scored parts.*

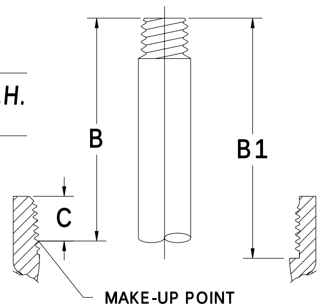
BOWLS

Wear on bowl vanes	*Abrasive action*	*Coat bowls, upgrade materials, or rubber line.*

AMERICAN-MARSH PUMPS

WATER LUBE BOWL REPLACEMENT

SHAFT:
DIA. ____________
THREADS/IN. ________ L.H.
THREAD LENGTH. ______



TAPER COLUMN
SHAFT PROJECTION (B) ____________ IN.
COLUMN MAKE-UP (C) ____________ IN.

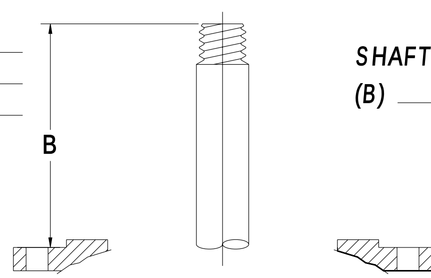
BUTT COLUMN
SHAFT PROJECTION (B1) ____________ IN.

TAPER OUTER COLUMN
COLUMN SIZE ________ I.D. ☐ O.D. ☐
THREADS/IN. ____________
TAPER: 3/16 ☐ 3/8 ☐ 3/4 ☐

BUTT JOINT THREADS
COLUMN SIZE ________ I.D. ☐ O.D. ☐
THREADS/IN. ____________

FLANGED COLUMN

SHAFT:
DIA. ____________
THREADS/IN. ________
THREAD LENGTH. ______



SHAFT PROJECTION
(B) ____________ IN.

PURCHASER ____________________

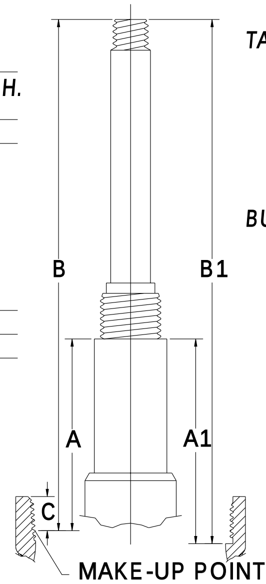
MODEL ____________________

P.O. NUMBER ____________________

AMERICAN-MARSH PUMPS

OIL LUBE BOWL REPLACEMENT

SHAFT:
- DIA. ____________
- THREADS/IN. ________ L.H.
- THREAD LENGTH. ________
- COUPLING O.D. ________



TAPER COLUMN
- TUBE PROJECTION (A) ____________ IN.
- SHAFT PROJECTION (B) ____________ IN.
- COLUMN MAKE-UP (C) ____________ IN.

BUTT COLUMN
- TUBE PROJECTION (A1) ____________ IN.
- SHAFT PROJECTION (B1) ____________ IN.

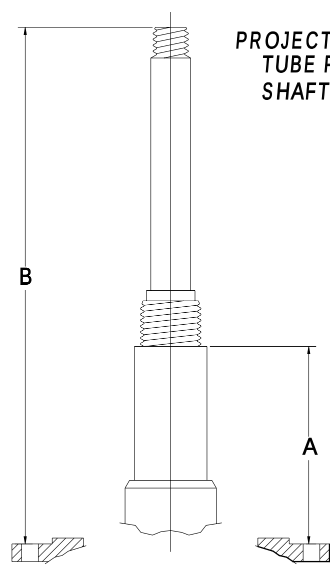
LINESHAFT BEARING:
- O.D. ____________
- THREADS/IN. ____________
- THREAD LENGTH. ________
- RH ☐
- LH ☐

TAPER OUTER COLUMN
COLUMN SIZE________ I.D. ☐ O.D.☐
THREADS/IN.____________
TAPER: 3/16 ☐ 3/8 ☐ 3/4 ☐

BUTT JOINT THREADS
COLUMN SIZE________ I.D. ☐ O.D.☐
THREADS/IN.____________

FLANGED COLUMN

SHAFT:
- DIA. ____________
- THREADS/IN. ____________
- THREAD LENGTH. ________
- COUPLING O.D. ________



PROJECTION
- TUBE PROJECTION (A) ____________ IN.
- SHAFT PROJECTION (B) ____________ IN.

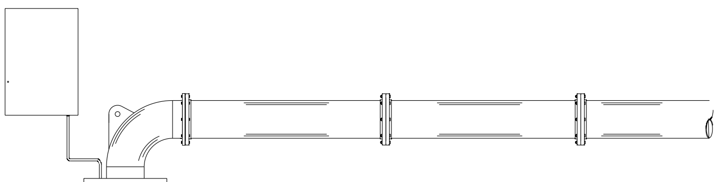
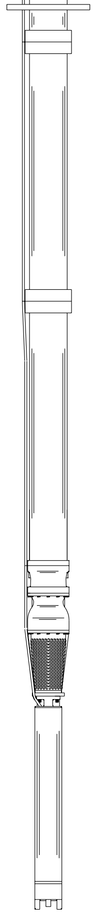
LINESHAFT BEARING:
- O.D. ____________
- THREADS/IN. ____________
- THREAD LENGTH. ________
- RH ☐
- LH ☐

PURCHASER____________

MODEL____________

P.O. NUMBER ____________

CAPACITY OF PIPE FOR WATER PER FOOT

DIAMETER OF PIPE (INCHES)	NUMBER OF GALLONS	WEIGHT (POUNDS)
2"	0.16	1.335
4"	0.65	5.421
6"	1.50	12.510
8"	2.60	21.684
10"	4.10	34.194
12"	5.90	49.206
14"	8.00	66.72
16"	10.50	87.57
18"	13.20	110.08
20"	16.30	135.94
22"	19.80	165.13
24"	23.50	195.99
26"	27.60	230.18
28"	32.00	266.88
30"	36.70	306.07
36"	52.90	441.18

NOTES:

A gallon of water (U.S. Standard) contains 231 cubic inches and weights approximately 8 1/3 lbs.
To find the pressure in pounds per square inch of a column of water, multiply the height of the column in feet by 0.434.
Doubling the diameter of a pipe increases its capacity four times.
The weight of water (in pounds) in any length of pipe is obtained by multiplying the length in feet by the square of the diameter and by 0.34.
The capacity of a pipe in gallons is equal to the square of the diameter in inches times the length times 0.0034.

CABLE SELECTION FOR SINGLE AND THREE PHASE MOTORS

Single Phase Cable, 60 HZ (Service Entrance to Motor - Maximum Length in Feet)														
Motor Rating		Maximum Feet of AWG Copper Wire Size												
Volts	HP	14	12	10	8	6	4	3	2	1	0	00	000	0000
Two or Three Wire Cable, 60 HZ														
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 the asterisk* meet the U. S. National Electrical Code ampacity for either individual conductors or jacketed 60° C cable.

Lengths marked* 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° cable 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. Example: If the table calls for #12 copper wire, #10 aluminum wire would be required.

The portion of the total cable length which is between the supply and single phase control box with line contractor should not exceed 25% of the maximum allowable, 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 feet.

Three Phase Cable, 60 HZ (Service Entrance to Motor - Maximum Length in Feet)														
Motor Rating		AWG Copper Wire Size												
Volts	HP	14	12	10	8	6	4	3	2	1	0	00	000	0000
230V 60 HZ Three Phase Three Wire	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
460 V 60 HZ Three Phase Three Wire	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	0	340*	540	850	1340	2090	2600	3200	3930	4810	5900	7110	
	20	0	0	410*	650	1030	1610	2000	2470	3040	3730	4580	5530	
	25	0	0	0	530*	830	1300	1620	1990	2450	3010	3700	4470	5430
	30	0	0	0	430*	680	1070	1330	1640	2030	2490	3060	3700	4500
	40	0	0	0	0	500*	790	980	1210	1490	1830	2250	2710	3290
	50	0	0	0	0	0	640*	800	980	1210	1480	1810	2190	2650
	60	0	0	0	0	0	540*	670*	830*	1020	1250	150	1850	2240
	75	0	0	0	0	0	0	0	680*	840*	1030	1260	1520	1850
	100	0	0	0	0	0	0	0	0	620*	760*	940*	1130	1380
	125	0	0	0	0	0	0	0	0	0	0	740*	890*	1000*
	150	0	0	0	0	0	0	0	0	0	0	0	760	920*
	175	0	0	0	0	0	0	0	0	0	0	0	0	810*
	200	0	0	0	0	0	0	0	0	0	0	0	0	0

SUBMERSIBLE MOTOR FLOW RATE

To calculate the flow required to keep a submersible motor cool, use the following formula:

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

WHERE: V_F = VELOCITY FLOW
GPM = GALLONS PER MINUTE
W_{ID} = WELL CASING INSIDE DIAMETER
M_{OD} = MOTOR OUTSIDE DIAMETER

At a maximum temperature of 86°F (30°C) the minimum Velocity Flow past motor would be:

0.25 ft/sec (7.62 CM/sec) - 4" diameter motor

0.50 ft/sec (15.24 CM/sec) - 6" diameter and larger motor)

If the flow past the motor is less than the minimum velocity, the motor needs to be installed in a flow sleeve.

When the water temperature is greater than 86°F (30°C) the flow rate past the motor should not be less than 3.0 ft/sec.

To calculate Horsepower required at temperature greater than 86°F (30°C):

$$H_{PR} = P_{HP} \times HF$$

WHERE: H_{PR} = HORSEPOWER REQUIRED
P_{HP} = PUMP HORSEPOWER AT DESIGN
HF = HEAT FACTOR MULTIPLIER AT 3 ft/sec FLOW

Heat Factor Multiplier at 3 ft/sec flow

MAXIMUM WATER TEMPERATURE	up to 5 H.P.	up to 30 H.P.	over 30 H.P.
140°F (60°C)	1.25	1.62	2.00
131°F (55°C)	1.11	1.32	1.62
122°F (50°C)	1.00	1.14	1.32
113°F (45°C)	1.00	1.00	1.14
104°F (40°C)	1.00	1.00	1.00
95°F (35°C)	1.00	1.00	1.00

NOTE: (The above chart is for can type submersible motor, for water proof wire type consult factory.)

CABLE SPLICING

Select first correct cable copper cross-section based on motor rating and length required.

600V TAPE SPLICING

A) Strip individual conductor of insulation only as for 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.

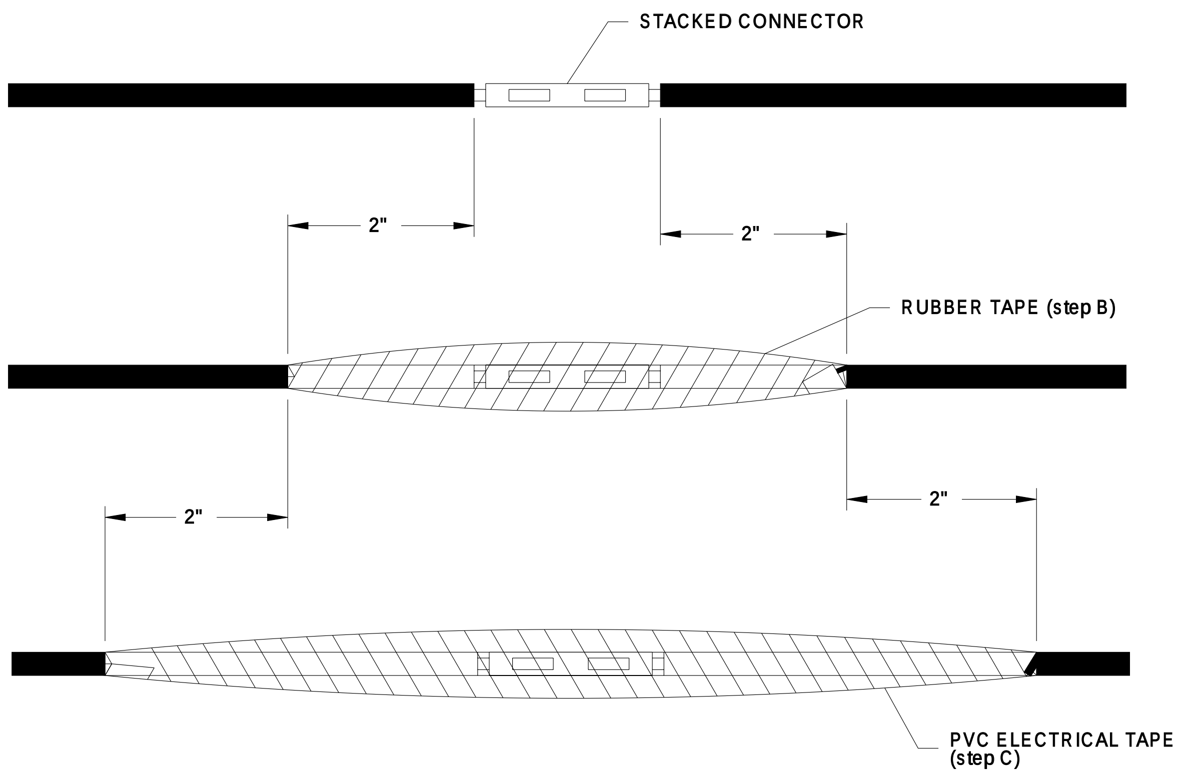
B) Tape the individual joints with rubber electrical tape, using two layers: the first extending two inches beyond each end of the conductor insulation end, the second layer two inches beyond the ends of the first layer. Wrap tightly, eliminating air spaces as much as possible.

C) Tape over the rubber electrical tape with #33 Scotch electrical tape, (Minnesota Mining Co.) 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.

NOTES:

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.



ATTENTION

The key factor of success is to follow exactly the manufacturer's application instructions for the splicing kits. The basis for a good splice is the connection of the copper wires for which we recommend the use of crimping connectors available for almost any common wire size.

Special attention should be given that the crimp connectors match the wire sizes used in order to get an electrically low resistance joint.

ATTENTION

In case of splicing cables of a six-lead motor for Y start, be sure that the extension cables continue with the same lead colors and phase designation as the original motor cables. This will ease up above ground connections to the Y panel or an external connection for DOL start.

ATTENTION

Loose wire connections can cause the burn out of the splice and short circuit, leading to possible motor failure.

INSULATION RESISTANCE READINGS

NORMAL OHM AND MEGOHM VALUES BETWEEN ALL LEADS AND GROUND

Insulation resistance varies little with rating. Motors of all H.P., voltage, and phase rating have similar values of insulation.

CONDITION OF MOTOR AND LEADS	OHM VALUE	MEGOHM VALUE
A new motor (without drop cable)	20,000,000 (or more)	>20.0
A used motor which can be reinstalled in the well	10,000,000 (or more)	>10.0

MOTOR IN WELL. Ohm readings are for drop cable plus motor.	OHM VALUE	MEGOHM VALUE
A new motor in the well.	2,000,000 (or more)	>2.0
A motor in the well in reasonably good condition	500,000-2,000,000	0.5-2.0
A motor which may have been damaged by lightning or which may have damaged leads. Do not pull the pump for this reason.	20,000-500,000	0.02-0.5
A motor which definitely has been damaged or with damaged cable. The pump should be pulled and repairs made to the cable or the motor replaced. The motor will not fail for this reason alone, but it will probably not operate for long.	10,000-20,000	0.01-0.02
A motor which has failed or with completely destroyed cable insulation. The pump must be pulled and the cable repaired or the motor replaced.	less than 10,000	0-0.01

AMERICAN-MARSH PUMPS

DEALING WITH THREE PHASE UNBALANCE

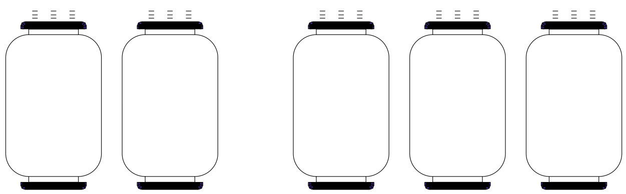
Current unbalance, particularly in rural areas with heavy single-phase loads, can cause premature motor failure - a result of reduced starting and breakdown torque, excessive and uneven heating, and excessive vibration.

It is important that the electrical load to the submersible motor be reasonable balanced. It is also important that the installation be made properly. Here's how to make a proper electrical installation and what to do if you can't.

STEP 1 PRE-INSTALLATION

Prior to installation, the power company should be notified of the motor data, plus other loads that are on the transformer bank.

You should know if the service provided is a true three-phase, three transformer system or a two transformer system. This can be determined by counting the transformers if the service is in, or by questioning the power company if service is not yet in. Here is an open-delta or wye system (on the left), with a true three-phase, three transformer system (on the right).



OPEN DELTA OR WYE TRUE THREE PHASE

Make sure that the transformer rating, in KVA, is adequate for the motor load by referring to this chart which references the KVA requirement by horsepower. Note that the minimum KVA rating, at the right on the chart, refers to "each" transformer used. This chart, incidentally, covers only the motor KVA requirements and does not make allowances for other loads.

SUBMERSIBLE 3" DIA MOTOR H.P. RATING	TOTAL EFFECTIVE KVA REQUIRED	SMALLEST KVA RATING "EACH TRANSFORMER"	
		OPEN WYE OR DELTA 2 TRANSFORMERS	WYE OR DELTA 3 TRANSFORMERS
1 1/2	3	2	1
2	4	2	1 1/2
3	5	3	2
5	7 1/2	5	3
7 1/2	12	7 1/2	5
10	15	10	5
15	20	15	7 1/2
20	27	15	10
25	32	20	10
30	40	25	15
40	53	30	20
50	65	37 1/2	25

STEP 2 SYSTEM REQUIREMENTS

Select the proper size magnetic starter or pump panel for the H.P. rating of the motor.

Use ambient compensated extra quick trip heaters on all three legs. The recommended heater size for the make of panel is provided with the motor.

Manufacturer's recommended cable size must be followed from the transformer to the pump panel and from the panel to the motor, based on H.P. and voltage rating. Make sure that the length of cable in each case is no longer than the manufacturer's recommendation for that size of cable.

If the pump is to operate properly on a system with two transformers, it is good practice to use both the next higher H.P. motor rating and the nest larger cable size. Taking these steps gives a greater margin of safety and provides more tolerance to current unbalance.

STEP 3 START UP

While the pump is still above ground check the insulation resistance of the motor to make sure that it is at least two million or more ohms (two megohms).

After the drop cable has been spliced to the motor leads and the pump is installed in the well, check the insulation resistance again to determine if your splice and cable are good.

If insulation resistance is still one million ohms or more, voltage can be applied to the motor.

Whatever you do, remember "SAFETY FIRST!" Before working inside the pump panel or magnetic starter, always disconnect the line to the panel or starter. Be sure it is off. Double-check this with a voltmeter.

Check for correct rotation of the motor by running it first in one direction and then the other. Use a discharge valve and pressure gage. The rotation that gives the highest pressure is always the correct one.

Rotation can be changed by switching any two of the three motor leads at the pump panel.

Now that correct rotation is established, the amount of current unbalance between legs should be calculated.

CURRENT UNBALANCE BETWEEN LEGS SHOULD NOT EXCEED 5% OF THE AVERAGE.

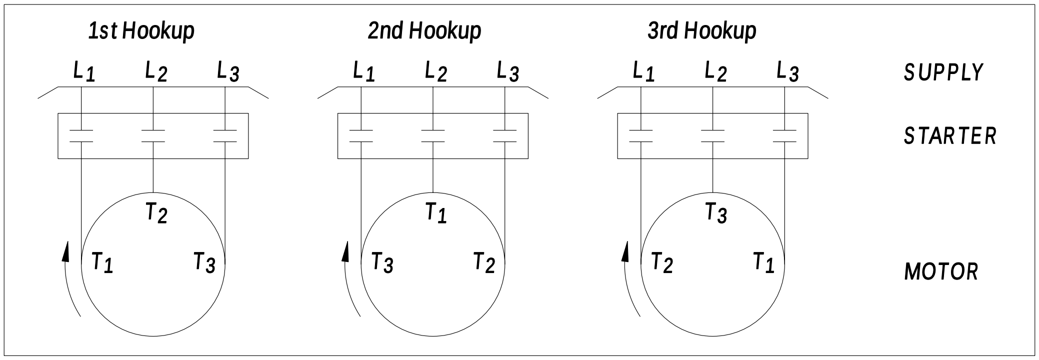
The percent of current unbalance is defined and calculated as follows:

$$\text{Percent current unbalance} = \frac{\text{maximum current difference from average current}}{\text{average current}} \times 100$$

Current readings in amps should be checked on each leg using the three possible hookups shown in the illustration. The best hookup is the one that has the lowest percentage of unbalance.

To prevent changing motor rotation when taking these readings, the motor leads should be rolled across the starter terminals by always moving them in the same direction as shown in the illustration. (On next page).

DEALING WITH THREE PHASE UNBALANCE



Use simple arithmetic to calculate the percentage of current unbalance for all three hookups. Here is an example of current readings at maximum pump loads on each leg of a three-wire hookup. To start with, add up all three readings for hookup number 1.

Hookup 1	Hookup 2	Hookup 3
T_1= 51 Amps	T_3= 50 Amps	T_2= 50 Amps
T_2= 46 Amps	T_1= 48 Amps	T_3= 49 Amps
T_3= 53 Amps	T_2= 52 Amps	T_1= 51 Amps

Total for each of the three hookups are as follows:

T_1= 51 Amps

T_2= 46 Amps

T_3= 53 Amps

TOTAL 150 Amps

Divide the total by three to obtain the average.

150 Amps - 3 = 50 Amps

Calculate the greatest amp difference from the average.

50 Amps
-46 Amps
4 Amps

Divide this difference by the average to obtain the percentage of unbalance.

4.00 Amps - 50 = 0.08 or 8%

In this case, the current unbalance for hookup number 1 is 8%.

If you use the same method of calculation, the maximum current unbalance for hookup number 2 is 4% and for hookup number 3 is 2%.

Next, compare the percentage of unbalance for all three hookups. The first hookup exceeds 5% current unbalance and should not be used. The second and third hookups did not exceed 5%, so either would be satisfactory. However, the third hokup had the lowest percentage of current unmbalance and it should be used, since the motor will then be operating at maximum efficiency and reliability.

By observing where the highest current reading is for each leg on the various hookups, you can now determine if the unbalance is caused by the power source or the electric motor. For example, note in the example that the highest leg was always on the same leg, L_3; this indicates that most of the unbalance was from the power source.

If the high current were on a different leg each time the motor leads were changed, you would know that the motor or a poor connection caused most of the unbalance.

Since loads on a transformer bank may vary during the day, readings should be taken at least twice - once during the day in what would be considered the normal load period and once in the evening during the usual peak load period. The installation should then be connected for the lowest percentage of current unbalance (again, not to exceed 5%.)

Should you continue to have problems with balance or if you encounter heater trip problems. contact your power company for help.

So you can fully explain the situation to the power use engineer, or to your pump supplier, keep a complete log of your pump installation.

You should keep a record of:

Well location and owner
Depth to water
Horsepower
Motor make
Size of control panel
Heaters used
Cable sizes and lengths
Current and voltage readings, by leg

With this type of log, you will be on solid ground when you discuss the current unbalance problem with your power company. You can show what you did and why. This will help pinpoint the problem.

PROBLEM: MOTOR DOES NOT START

CAUSE OF TROUBLE	CHECKING PROCEDURE	REMEDY
No power or incorrect voltage.	Using voltmeter check the line terminals. Voltage must be ±10% of rated voltage.	Contact power company if voltage is incorrect.
Fuses blown or circuit breakers tripped.	Check fuses for recommended size and check for loose, dirty or corroded connections in fuse receptacle. Check for tripped circuit breaker. Look for damage from heat, lightning or surge.	Replace with proper fuse or reset circuit breaker.
Defective pressure switch.	Check voltage at contact points. Improper contact of switch points can cause voltage less than line voltage.	Replace pressure switch or clean points.
Control panel malfunction.	For detailed procedure, see panel manufacturers instructions.	
Defective wiring.	Check for loose or corroded connections. Check the motor lead terminals with a voltmeter for power.	Correct faulty wiring or connections.
Bound pump.	Locked rotor conditions can result from misalignment between pump and motor or sand bound pump. Amp readings 3 to 6 times higher than normal will be indicated. Well may be crooked.	Sand bound pumps can sometimes be corrected by temporarily reversing any two leads in the control panel. If pump does not rotate freely, it must be pulled.
Defective cable or motor.	Check insulation resistance.	Repair or replace.

PROBLEM: MOTOR STARTS TOO OFTEN

CAUSE OF TROUBLE	CHECKING PROCEDURE	REMEDY
Pressure switch.	Check setting on pressure switch and examine for defects.	Reset limit or replace switch.
Check valve, stuck open.	Damaged or defective check valve will not hold pressure.	Replace if defective.
Waterlogged tank (air supply)	Check air volume control or snifter valve for proper operation.	Clean or replace. Drain and recharge tank.
Leak in system.	Check system for leaks.	Replace damaged pipes or repair leaks.

PROBLEM: MOTOR RUNS BUT OVERLOAD PROTECTOR TRIPS

CAUSE OF TROUBLE	CHECKING PROCEDURE	REMEDY
Incorrect voltage.	Using voltmeter check the line terminals. Voltage must be ±10% of rated voltage.	Contact power company if voltage is incorrect.
Overheated protectors.	Direct sunlight or other heat source can make control box hot causing protectors to trip. The box must not be too hot to touch.	Shade box, provide ventilation or move box away from heat source.
Control panel malfunction.	For detailed procedure, see panel manufacturers instructions.	
Defective cable or motor.	Check insulation resistance.	Repair or replace.
Worn pump or motor.	Check running current.	Repair or replace pump and/or motor.
Water system malfunction.	Check for waterlogged tank or defective pressure switch.	Repair or replace.
Pump load too great.	Incorrect pump for motor. Operating too far out on curve. Rotation backwards.	

PROBLEM: MOTOR RUNS CONTINUOUSLY

CAUSE OF TROUBLE	CHECKING PROCEDURE	REMEDY
Pressure switch.	Switch points may be "welded" in closed position. Pressure switch may be set too high.	Clean points or replace switch, or readjust setting.
Low level well.	Pump may exceed well capacity. Shut off pump, wait for well to recover. Check static and draw-down levels from well head.	Throttle back pump output or reset pump at a lower level. Do not lower if sand may clog pump.
Leak in system.	Check system for leaks.	Replace damaged pipes or repair leaks.
Worn Pump.	Symptoms of worn pump are similar to those of drop pipe leak or low water level in well. Reduce pressure switch setting, if pump shuts off, worn parts may be at fault. Sand is usually present in tank.	Pull pump and replace worn impellers, casing or other close fitting parts.
Loose or broken motor shaft.	No or little water will be delivered if coupling between motor and pump shaft is loose or if a jammed pump has caused the motor shaft to shear off.	Check for damaged shafts if coupling is loose and replace worn or defective units.
Pump screen blocked.	Restricted flow may indicate a clogged intake screen on pump. Pump may be installed in mud or sand.	Clean screen and reset at less depth. It may be necessary to clean well.
Check valve stuck closed.	No water will be delivered if check valve is in closed position.	Replace if defective.
Control panel malfunction.	See panel manufacturers instructions.	
Pump performance	Pump selection incorrect for system.	Recalculate Requirements.

PROBLEM: NO WATER DELIVERED

CAUSE OF TROUBLE	CHECKING PROCEDURE	REMEDY
Pump not turning.	Check for broken pump/motor shaft or coupling. Motor is not running.	Repair or Replace.
Discharge line restricted.	Check valve could be installed backwards or could be stuck closed.	Replace if defective.
Inlet restricted.	Screen could be plugged, well may be collapsed or the water level may be too low.	Repair or change well locations.
Wrong rotation.	Rotation of three phase motor.	Repair or Replace.
Wrong pump selection.	Recalculate requirements.	Replace.

PROBLEM: LOW WATER DELIVERY

CAUSE OF TROUBLE	CHECKING PROCEDURE	REMEDY
Pump speed or selection.	Rotation could be backwards. Check for low voltage or low phase. Wrong speed, pump vs. motor.	Recalculate requirements.
Restricted discharge.	Line clogged, corroded or ruptured. Check valve stuck partially closed.	Repair or Replace.
Restricted inlet.	Partially clogged inlet screen. Well partially collapsed. Water level too low in well. Plugged impeller.	Repair or change well locations.
Mechanical problem.	Pump worn or loose impeller.	Repair or Replace.

PROBLEM: ELECTRIC SHOCK

CAUSE OF TROUBLE	CHECKING PROCEDURE	REMEDY
Grounding	Improper (or absence of) ground on electrical system, controls or motor.	Repair or Replace.
Wiring	Check for improper wiring, grounded motor or cable.	Repair or Replace.
Controls	Defective or burned out component.	Repair or Replace.
Moisture	Wet controls or wiring.	Repair or Replace.

CAUTION: Electric shock from contact with any pumping system part is never safe to ignore! Find and correct the cause!

AMMETER ANALYSIS OF MOTOR PROBLEMS

OPERATION AND AMMETER READINGS	CONDITION INDICATED	WHAT TO LOOK FOR: 1 PHASE MOTOR	WHAT TO LOOK FOR: 3 PHASE MOTOR
Motor won't start. Ammeter reads zero on all lines.	Power is not connected to motor.	Dead power line. Blown fuses. Tripped overload. Defective pressure switch. Defective control box. Separated motor leads.	Same Same Same Same Same Same
Motor won't start. Ammeter reads high on two lines, zero on other line.	Power is connected to only part of motor windings.	Defective control box. Control box not mounted vertically. Very low voltage. Damaged cables or splice.	Same One blown fuse One lead in power supply Same
Motor won't start. Ammeter reads several times normal on all leads.	Power is connected to motor but something prevents starting.	Tight motor bearings. Tight pump bearings. Damaged cables or splice. Very low voltage.	Same Same Same Same
Single phase motor runs but ammeter reads high, especially on red lead. Overload may trip.	Controls are not switching out start capacitor.	Control box not mounted vertically. Defective relay. Very low voltage. Motor connected to wrong phase or voltage.	Not applicable
Motor runs but ammeter reads high on some or all leads. Overload may trip.	Motor is overloaded or voltage is incorrect.	Faulty motor. Faulty pump. Low or high voltage. Incorrect pump for motor.	Same Same Same Same

AMERICAN-MARSH PUMPS

END PLAY (LATERAL) -THRUST CONSTANT

CLOSED IMPELLER BOWLS IMPELLER SEAL - BOTTOM FACE AND SKIRT TO MATCHING BOWL SECTION				SEMI-OPEN IMPELLER SEAL - BOTTOM OF VANES TO MATCHING BOWL SECTION		
BOWL MODEL	END PLAY STANDARD (inches)	END PLAY *SPECIAL (inches)	THRUST CONSTANT K (Lbs./Ft. Head)	BOWL MODEL	END PLAY SPECIAL (inches)	THRUST CONSTANT K (Lbs./Ft. Head)
6JC	3/8	N/A	1.56	M4	3/8	1.50
6LC				H4	1/4	
6MC	1/2		2.24			
6HC						
6XC	5/8		2.83			
6WC	3/8		4.13			
6YC	1/4		4.10			
8JC	7/16	23/32	2.98	8JS	7/16	3.52
8LC				8LS		3.34
8KC		11/16	3.93	8KS		4.42
8MC				8MS		4.28
8EHC	3/8	5/8	5.40	8EHS	3/8	5.40
8WC	1/2	5/8	6.20			
8YC	3/8	5/8	8.00	8YS	3/8	8.00
10JC	1/2	3/4	3.98			
10KC			4.20			
10LC	5/8	7/8	6.60	10LS	5/8	7.50
10MC				10MS		
10HC		1	8.10	10HS		9.20
10WC	7/8	1 1/4	10.30	10WS	7/8	11.20
10YC	3/4	1 1/8		10YS	3/4	11.40
10ZC	1/2	7/8	13.60	10ZS	1/2	13.50
12JC	5/8	7/8	6.60			
12IC	5/8	7/8	6.75	12IS	5/8	8.20
12KC	5/8	7/8	6.50	12KS	1/2	7.75
12LC	3/4	1 3/8	10.60	12LS	7/8	12.50
12MC		1 1/4		12MS	3/4	
12HC			16.50	12HS	3/4	19.00
12XC	7/8	1 1/4	18.20	12XS	7/8	20.80
12WC	3/4	1 3/16	16.20	12WS	3/4	17.40
12YC	7/8	1 1/2	14.20	12YS	7/8	16.50
13M	15/16	1 3/16	8.00			
14LC	7/8	1 3/8	17.20	14LS	7/8	19.70
14MC			21.80	14MS		23.40
14HC				14HS		25.20
14XC				14XS		23.40
14WC			24.80	14WS		26.20
14YC	1/2	5/8	24.00			
14ZC	1/2	5/8	24.70			
15KC	11/16	1 1/8	28.00	15KS	11/16	30.00
18LC	1/2	3/4	21.75			
18MC	1/2	3/4	21.75			
18HC	1/2	3/4	23.50			
20MC	3/4	1	29.40			
20HC	5/8	1	58.00			

*Special End Play may reduce performance.
NOTE: CONSULT FACTORY if additional end play is required.