## Engineering Data

I. Summary and Index to this Section
Careful planning and selecting of the correct pump and installation will result in more economy through longer, more efficient, and maintenance-free service. In this section you will find practical information, charts, and tables to assist you in your selection of the right DEMING pump. The data herein is not intended to be complete or sufficiently technical to cover every pumping problem. For any assistance, please refer to the factory.

For a correct pump selection the following factors should be taken into consideration:

A. Regarding the liquid being pumped - (if not clear water):
1. What is the liquid? (See Par. II)
2. What is the Specific Gravity? (See Par. III)
3. What is the Viscosity? (See Par. IV)
4. What is the Temperature of the liquid being pumped? (See Par. V)
5. Are there any solids in suspension? Largest diameter? (See Par. VI)

B. Regarding the installation:
1. Supply? - Shallow Well (Under 25' lift) or Deep Well (Over 25' lift)? Is the supply from a cistern, sump, lake, river, or other? Well Diameter?
Capacity of the well at pumping depth? Depth of the Well? Largest diameter? (See Par. VI)
2. Delivery required from the pump in gallons per minute?
3. Total Dynamic Head? (See Par. VIII)
4. Type of Installation?
   a) Domestic Water Supply
   b) Industrial Water Supply
   c) Irrigation
   d) Sump or Sewage
   e) Other
5. Power Available:
   a) Electric Current Characteristics: Voltage, phase, cycle, AC or DC?
   b) Diesel or Gasoline Engine? Direct Drive? Indirect Drive?

C. Other factors which may influence the proper selection of equipment:
1. Speed. (See Par. IX)
2. Suction Problems and Restrictions. (See Par. X)
3. Mechanical Seals and Stuffing Boxes. (See Par. XI)
4. NPSH - Net Positive Suction Head. (See Par. XII)

D. Reading, understanding and evaluating the Performance Curves. (See Par. XIII)
E. Electrical Starting Equipment. (See Par. XIV)
F. Centrifugal Pumps for the Hydro-Pneumatic Service. (See Par. XV)
G. Reference Tables and Charts. (See Par. XVI)

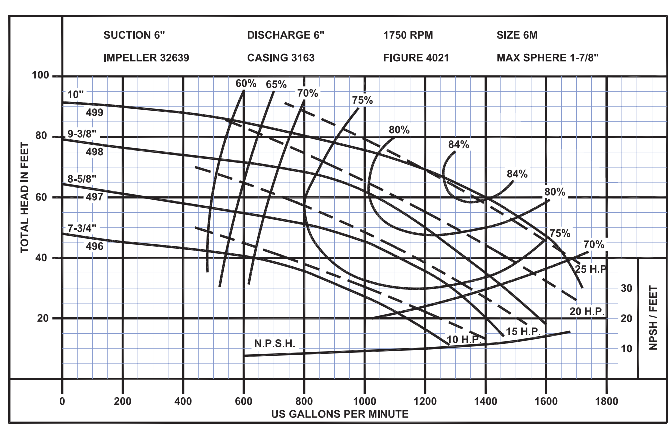
NOTE: In many standard installations involving clear water at normal temperatures several of the above factors may not be involved, much less present a problem. However, each of the points outlined above should be familiar to all Engineers, Technicians, and Salesmen who work with pumping equipment since so many pump applications must be engineered, with these factors in mind.

II. Liquid Handled and Pump Construction
The nature of the liquid being pumped will largely dictate the construction to be used in the pump itself. DEMING "Standard" Construction will in nearly every case be acceptable for clear water at normal temperature. Other liquids, depending on their corrosive characteristics may require liquid ends (that is, all parts of the pump coming into contact with the liquid being pumped) of other materials. Table No. 22 in Paragraph XVI lists some of the more common liquids with suggested construction in column No. 5.

NOTE: Occasionally it may be more economical to consider using a standard pump for a slightly corrosive liquid and replace the pump more often rather than use a very expensive alloy pump initially. This choice would largely depend upon experience in handling the given liquid and the availability of parts and service, as well as the comparative operating life between the standard unit and the alloy unit.

III. Specific Gravity Consideration in Pump Selection
Specific Gravity - is the weight of a given volume of liquid expressed in relation to the weight of an equal volume of clear water at 62°F is expressed as 1.0. Thus, a liquid having a specific gravity of 0.8 will be 20% lighter than water; whereas a liquid having a specific gravity of 1.2 will be 20% higher than water.
Specific Gravity affects pump selection in two ways:
A) Horsepower required and B) Pressure equivalent of the total dynamic head.

A. Horsepower - The horsepower required to operate a pump at a given rating (Capacity and Head) will vary in direct proportion to the Specific Gravity of the liquid being pumped. Referring to the curve below as an illustration:



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Using Horsepower Formula:

$$\frac{GPM \times TDH \times SG}{Efficiency \times 3960} = BHP$$

For 1400 GPM at 60' TDH

1. Water - Specific Gravity = 1.0

$$\frac{1400 \times 60 \times 1}{.84 \times 3960} = 25.4 \text{ BHP}$$

2. Jet Fuel - Specific Gravity = 0.7

$$\frac{1400 \times 60 \times 0.7}{.84 \times 3960} = 17.7 \text{ BHP}$$

3. Brine - Specific Gravity = 1.2

$$\frac{1400 \times 60 \times 1.2}{.84 \times 3960} = 30.04 \text{ BHP}$$

In other words, the BHP for water should be multiplied by the specific gravity of the liquid being pumped to obtain the horsepower necessary for this liquid at the required rating.

NOTE: There will be no change needed either in the speed of the pump, or in the diameter of the impeller. The only change will be in the horsepower required. The pump should be selected directly from the "Water Characteristic Curves" with allowance made for the horsepower as indicated above.

B. Pressure - The pressure that a Centrifugal Pump develops is a pressure head (TDH) produced and the weight of the liquid being pumped. A given pump will develop, for example, 100 feet of head, regardless of the specific gravity of the liquid being pumped. The pressure however will change in direct proportion to the specific gravity of the liquid being pumped. For example, the 100 foot head pump will develop about 43 psi while pumping water. The same pump will develop 100 foot head pumping fuel oil (SG 0.7) also, but the pressure will be only about 30 psi (43 psi time 0.7). This point is very important when a pump is to be selected to develop a given pressure with a liquid having a specific gravity other than that of water.

IV. Viscosity Considerations in Pump Selection

Viscosity is the characteristic of a fluid that indicates its resistance to flow. Technically speaking, viscosity is the internal friction of a fluid which tends to oppose flow. As a measure of the coefficient of viscosity is in terms of CGS (centimeter gram second) units, the force of one dyno required to move one square centimeter of a fluid a distance of one centimeter. This unit is called a poise; a centipoise is 1/100th of a poise. Thus, absolute viscosities are given in centipoise units. In the United States the standard unit of viscosity is the SSU (Seconds Say bolt Universal) for medium viscosities such as motor oils, etc, and the SSF (Seconds Say bolt Furol) for high viscosities such as tar, molasses, etc. To convert SSU and SSF standards to Centistokes, refer to Table No. 24 Paragraph XVI.

Viscosities vary indirectly with temperatures, arid with some fluids, a moderate decrease in temperature shows a marked increase in viscosity. Therefore, it is essential that the viscosity be specified at the desired temperature. Viscosities of lubricants are usually measured at 100°F and 210°F while fuel oils are measured at 77°F and 122°F. Water has a viscosity of 31.5 SSU (1.1 centistokes) at 60°F while certain oils may have viscosities of 3000 SSU (650 cenitstokes) at the same temperature.

The performance of centrifugal pumps handling viscous fluids as related to water performances has been investigated by numerous authorities. However, due to the many variations in construction, design and rotative speeds, accurate correlation and prediction of results when pumping viscous fluids has not been obtained. The Hydraulic Institute has published in their standards a method of selecting suitable pumps for viscous fluids when the water performance is known.

A less accurate but shorter method is to be had by the use of the Correction Curves on page 3. After determining the discharge size of the pump and knowing the viscosity of the fluid, the correction factors for capacity, head, and efficiency can be determined and applied to the water performance curves of the pump.

For example: a pump is required to handle 225 GPM at 40 feet of liquid having a viscosity of 275 cenitpoise and a specific gravity of 1.25.

Change first from absolute units (cenitpoise) to kinematic viscosity units (centistokes).

$$\text{Since Centistokes} = \frac{\text{Centipoises}}{\text{Specific Gravity}} \quad \text{or} \quad \frac{275}{1.25}$$

= 220 Centistokes

From the conversion chart (Table No. 24 Paragraph XVI) this is very close to 1000 SSU. Referring to DEMING Curve No. PC 1415-A Page 4, it appears that a Figure 4011 Size 3S pump will be the tentative selection since the 225 GPM at 40 feet falls to the left of the best efficiency range. From the charts on Drawing No. 114-021 the correction factor for water capacity and head is .80 and for efficiency is .53. Thus the water performance should be 225 / .80 or 282 GPM at 40 / .80 or 50 feet.

Delivering 282 GPM at 50feet the pump shows an efficiency of 78%. By applying the correction factor of .53 the efficiency is reduced to 41.5%.

Thus the input power will be:

$$BHP = \frac{GPM \times TDH \times \text{Specific Gravity}}{\text{Efficiency} \times 3960}$$

$$BHP = \frac{225 \times 40 \times 1.25}{.415 \times 3960} = 6.85$$

requiring a 7½ HP - 1750 RPM motor.

The above method should be used only for tentative selection. Where there is a guaranteed performance it will be necessary to test the pump under actual conditions to determine its performance.

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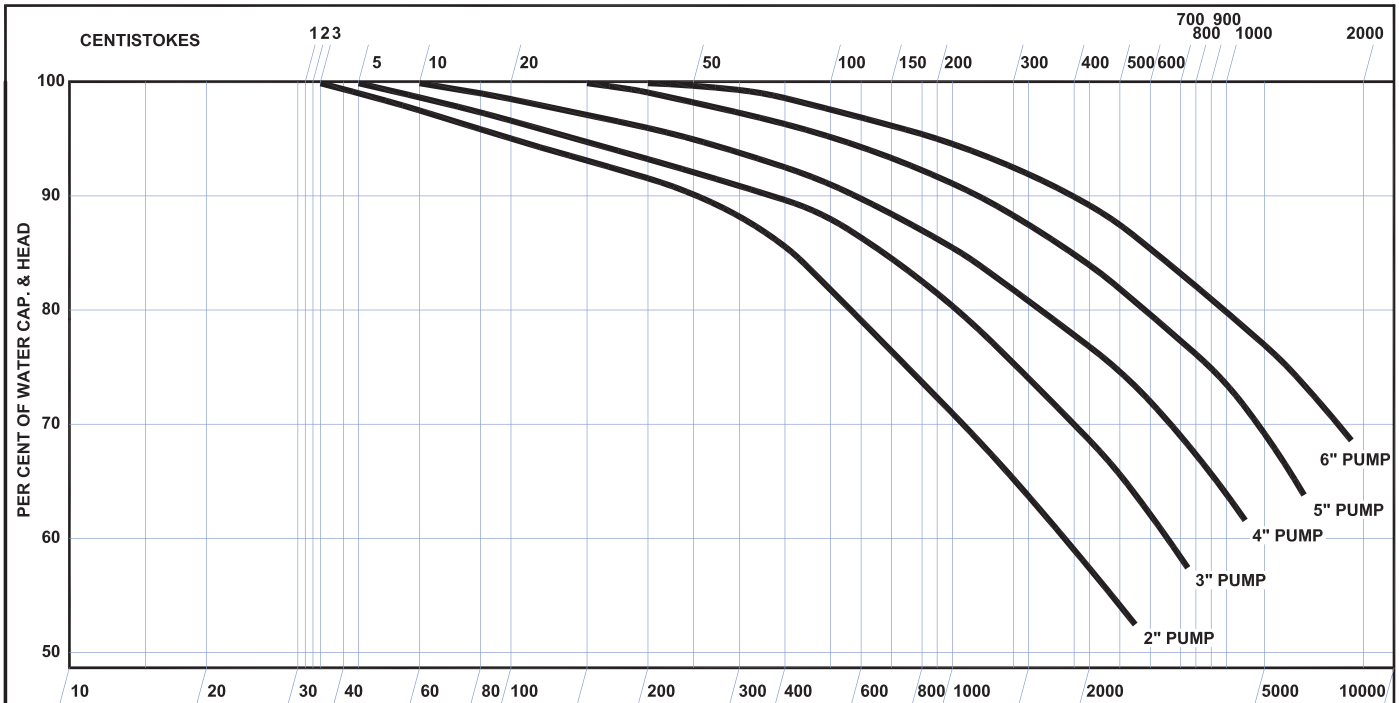
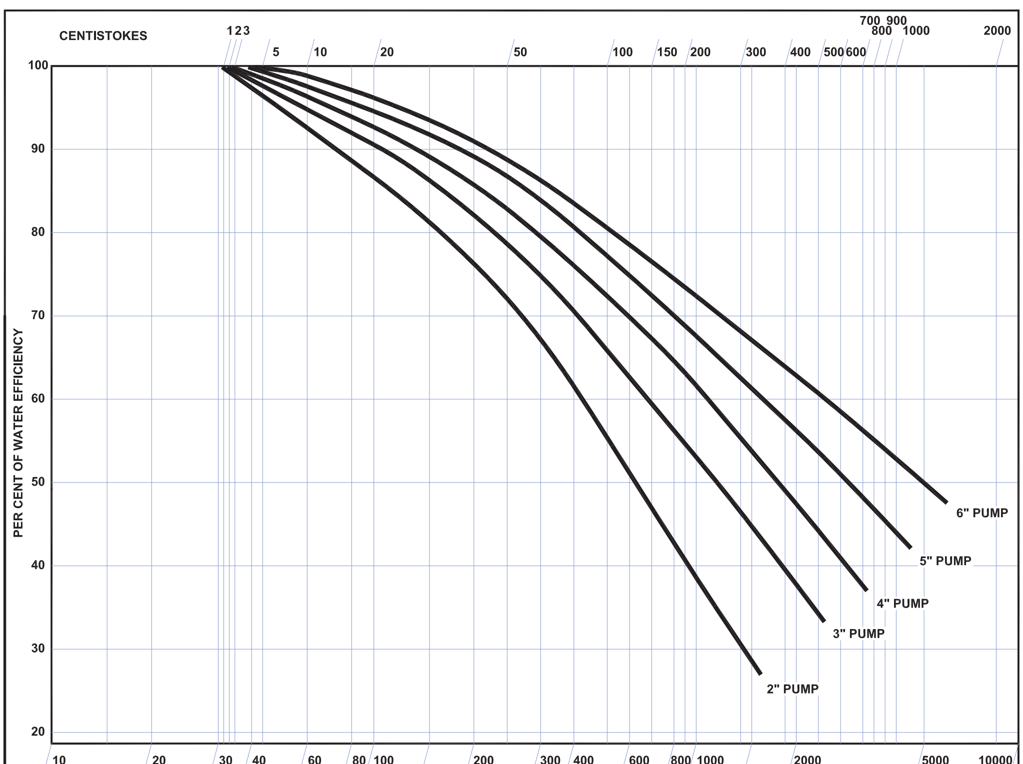
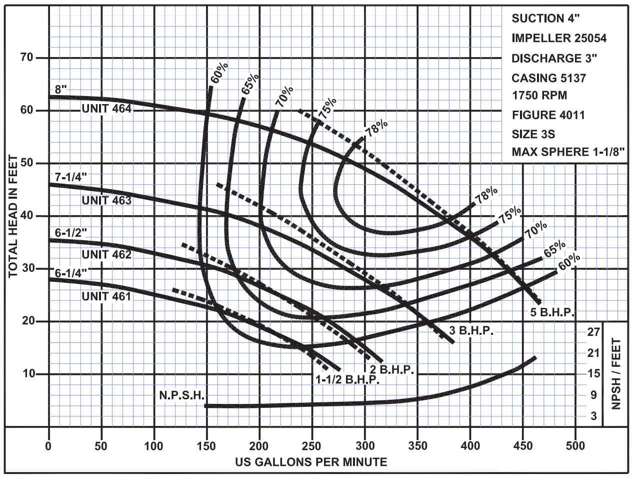



VISCOSITY - SECONDS SAYBOLT UNIVERSAL
PUMP CAPACITY & HEAD - VISCOUS FLUIDS




VISCOSITY - SECONDS SAYBOLT UNIVERSAL
PUMP EFFICIENCY - VISCOUS FLUIDS

## Engineering Data




Also, use these correction curves only for centrifugal pumps of conventional design in the normal operating range, with open or closed impellers. Do not use for mixed flow or axial flow pumps. To avoid the effect of cavitation make certain there is ample NPSH.
Use only on uniform liquids. Gels, slurries, paper stock and non-uniform liquids may produce widely varying results.
For most centrifugal pump applications the viscosity of the liquid should not exceed 3000 SSU unless experience has indicated a higher figure is practical.

V. Liquid Temperature
Temperature of the liquid should be taken into consideration with regard to:

1. Viscosity - (Generally not important factor with water at normal temperature) See Par IV.
2. Vapor Pressure - Very important for hot water and vaporous liquids (See NPSH Par XII).

VI. Solids in Suspension
Solids in suspension may restrict the use of a pump. Refer to DEMING characteristic curves for maximum allowable solids. Should there be any doubt regarding the use of a pump for liquids containing solids in suspension, please refer to the factory. Sizes of solids should always be indicated when referring to a sump or sewage pump.

VII. Well Information
At Sea Level a shallow Well Water Pump may generally be used with a suction lift of up to about 25 feet maximum. A deep well pump is necessary for greater suction lifts at sea level. At higher altitudes, however, the maximum suction lift becomes less and less. (See Table No. 9 Par XVI Column 5).
For Deep Well Installations it is important that the pumping level be ascertained. This is the level of the water while the pump is operating at its rating. It is this figure upon which the deep well pump "lift" is based, rather than the static level.
The well diameter is important for all Jet, deep well plunger, and especially Vertical Turbine and Submersible installations.
The well capacity should also be determined before the pump selection is made, least a pump of too great a capacity be installed, causing the well to be pumped dry. A pump which will over-pump the well can result in considerable damage to the well, the pump, and the motor or engine. Since wells will often produce varying capacities at different depths, it is often recommended that a well be tested to make certain that the required capacity can be obtained and at what depth or "pumping level".

VIII. Total Dynamic Head
Total Dynamic Head - (TDH) is the sum, in feet, of:

1. Suction Lift from lowest water level to pump. (If positive suction head - subtract this figure from the sum of No.'s 2 thru 5).
2. Friction in suction line (See Tables 1, 3, & 5 Par. XVI)
3. Discharge Head or Vertical Distance from Pump to point of discharge.
4. Friction in discharge line (See Tables No. 1, 3, & 5 - Par XVI)
5. Discharge Pressure required (if any) at point of discharge.

Since Centrifugal Pumps develop a "velocity head" measured in feet, it is important that "pressure head" requirements in psi be changed to "feet" before determining the total dynamic head. Thus under Item 5, above, if a discharge pressure of 20 psi were required, this should be changed to 46.2 feet of water since each psi at sea level is equal to 2.31 feet of water. For liquids having a specific gravity other than that of water, this figure must be corrected by dividing the number of "feet of water" by the specific gravity of the liquid being pumped. Using the above example 20 psi discharge pressure of fuel oil having 0.7 SG would be:

| | | |
|---|---|---|
| PSI x 2.31 / SG | = | Feet Head |
| 20 x 2.31 / .7 | = | 66' |

Reciprocating pumps, on the other hand, are generally selected on the basis of pressure in pounds per square inch. Therefore, "head in feet" will generally have to be converted into psi.

$$PSI = \frac{Feet\ Head \times SG}{2.31}$$

NOTE: In any case, if a liquid is other than water, special attention should be given to specific gravity and to viscosity. Information on selecting pumps for these liquids is covered in Par. III and IV.

IX. Pump Speed
The Speed of a centrifugal pump is important in that any variation in speed will create a change in the capacity, head, and horsepower. Changing the speed on a centrifugal pump will affect the performance as follows:

Capacity - Changes in direct proportion to the change in speed.
Head - Changes as the square of the change in speed.
Horsepower - Changes as the cube of the change in speed.
Efficiency - Remains approximately the same.

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Therefore, a pump delivering 100 GPM at 50 feet TDH requiring 1.7 BHP at 1800 RPM would deliver 122 GPM at 75 feet TDH requiring 3.1 BHP at 2200 RPM. It is easy to see that a slight change in speed will cause a slight change in capacity; a moderate change in head; and a comparatively large change in horsepower.

This speed factor is very useful in variable speed applications (Gasoline or Diesel Engine Drives, Belt Drives, etc.) since often a pump, which on the standard curves may appear somewhat too small, can fit the application very well by a slight increase in speed.

CAUTION: Since the horsepower changes are so proportionately greater, this will often limit the allowable increase in speed. Increases in speed of more than approximately 20% above standard speeds (1450 RPM, 1750 RPM, 2900 RPM, 3500 RPM) should have prior factory approval.

As a practical centrifugal pump illustration, using the slide rule, let us assume that a pump is required to deliver approximately 240 GPM at 120 feet TDH. The pump will be driven by a gasoline engine at 2200 RPM. Since curves at 2200 RPM are not readily available we will refer to the 1750 RPM curve, making the following calculations with the slide rule set as in the illustration below.

at the indicated ratings would remain about 66%. To figure the horsepower required at 2200 RPM, we would use the formula:

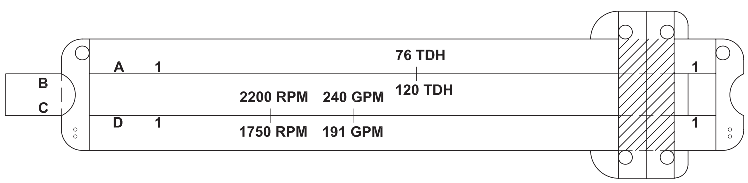
$$\frac{\text{GPM x TDH x SG}}{\text{Efficiency x 3960}} = \text{BHP}$$

$$\frac{240 \times 120 \times 1}{.66 \times 3960} = 11 \text{ BHP}$$

A little practice with this scaling of a centrifugal pump curve will enable one to select a suitable unit for many installations where selections based only on the standard printed curves would indicate little possibility of an available unit.

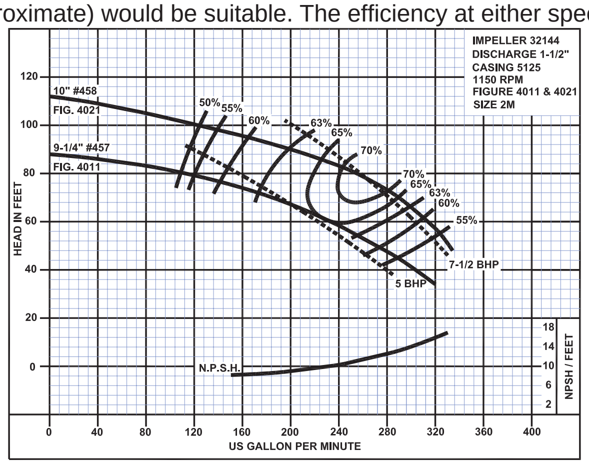
50 Cycle - Note particularly that this scaling procedure is invaluable for calculating performance characteristics for pumps driven at 1450 RPM and 2900 RPM by 50 cycle motors. In each case, work from the 1750 RPM curves for 1450 RPM performance, and work from 3500 RPM curves for 2900 RPM performance.

In reciprocating pumps an increase in speed (strokes per minute) will cause a proportionate increase in capacity. A loss in pressure will result from this change unless the horsepower is increased in proportion to the increase in speed. The horsepower formula given above would also apply.




Set curve speed on "D" scale under scaled speed on "C" scale: Read scaled capacity on "C" scale over curve capacity on "D" scale. Read scaled head in feet on "B" scale under curve head in feet on "A" scale.

Therefore to obtain a centrifugal pump performance of 240 GPM at 120 feet TDH, at 2200 RPM, a pump capable of delivering 191 GPM at 76 TDH at 1750 RPM should be selected from the 1750 RPM curves. From Curve 1270 on this page it appears that FIg. 4021, Size 2M with a $9\frac{5}{8}$" diameter impeller (approximate) would be suitable. The efficiency at either speed




X. Suction Piping

Suction piping and suction problems probably deserve more consideration than any other factor in pump installation. It would be well to refer to Table No. 9 Paragraph XVI for maximum practical suction lift on pumps at various altitudes. Other considerations may be outlined as follows:

1. Suction piping should be free from air leaks, sharp curves, and loose connections to eliminate the formation of air pockets.
2. A foot valve should be considered where possible to eliminate the need for priming the pump after each shut-down.
3. A strainer (often combined with foot valve) is necessary in preventing foreign matter from entering the suction line and clogging the pump.
4. Suction entrance should always be located well below the minimum water level. A pump that "breaks suction" is liable to be severely damaged and damage to the motor or engine may also result.
5. Suction lines should never be smaller than the suction inlet of the pump.

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Refer also to Paragraph XII for additional information on suction factors.

XI. Mechanical Seals and Stuffing Boxes
Mechanical Seals and Stuffing Boxes are used to seal off the liquid end of a pump from the power end.

The stuffing box consists of a material having a low coefficient of friction and good sealing qualities, packed around the pump shaft. This packing can be tightened or loosened by a stuffing box gland. Under normal operating conditions, however, the stuffing box should permit a slight flow of liquid to escape. This slight leakage keeps the packing in good condition and is necessary for satisfactory stuffing box operation.

Mechanical seals can be furnished when a pump is handling a hazardous or expensive liquid or liquids where the necessary stuffing box leakage is objectionable.

The mechanical seal consists of a rotating element and a stationary element. The rotating element is fastened to and rotates with the shaft. The stationary element is mounted in, or on the casing seat. The sealing faces are highly lapped surfaces of materials selected for their low coefficient of friction and their resistance to corrosion by the liquid being pumped. The faces have a minute running clearance and normally run on a dry thin film of liquid. A spring (or springs) or a flexible member of suitable material provides a means of positioning the seal and providing flexibility.

Mechanical seals are made in many designs and each installation must be carefully reviewed and analyzed before making a selection.

When the pump is equipped with a mechanical seal, no attention or adjustment to the seal is normally required. Except for possible slight initial leakage, the seal should operate with negligible loss.

A single mechanical seal is used for handling clear liquids being pumped recirculated through the seal chamber to serve both as a lubricant and as a coolant.

The double mechanical seal is used when the liquids are abrasive, non-lubricating, or have excessive temperature. The seal chamber here is filled with a sealing fluid from an outside source.

For both stuffing box applications and for mechanical seals, special materials are available to suit nearly every type of liquid being pumped. For any special application, or for any assistance in selecting material for a stuffing box packing or for a mechanical seal, please refer to the factory.

XII. Net Positive Suction Head - NPSH
For some reason NPSH presents a difficult problem to many pump professionals. The following explanation will be of assistance. We are covering two methods of calculation, the first of which is suitable for normal cold water installations, whereas the second is suitable for all installations regardless of the liquid being handled. Definitions:

1. Available Net Positive Suction Head (NPSH) is the DIFFERENCE between Barometric Pressure (See Table No. 9, Par XVI, Col. 4) and Dynamic Suction Lift (Sum of Vertical Suction Lift and all friction losses in Suction Line).

2. The required Net Positive Suction Head at the suction inlet of the pump is shown at the bottom of each performance chart and is in the form of a curve, and has been determined from actual tests.
3. The available NPSH must be at least equal to or exceed the required NPSH.

EXAMPLE NO.1: For installations at or near sea level and involving water at normal temperature a 3 foot safety factor is deducted from the Barometric reading of 34 feet. This safety factor is recommended for any elevation.

Selection A:
Refer to Curve Chart of Fig. 4011, Size 2S 1750 RPM below.

Pump to deliver:
150 GPM against 54 feet total head of which 14 feet is suction lift, including all friction losses, etc.

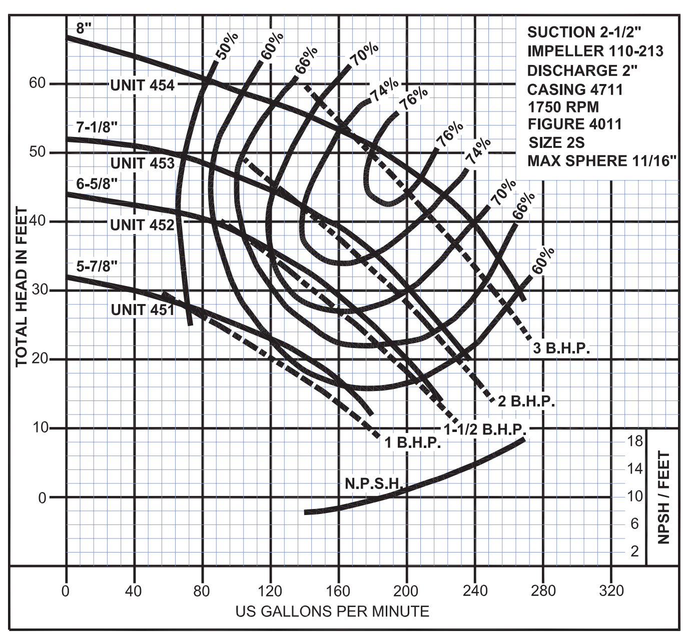
| | | |
|---|---|---|
| | 34' - 3' = 31' | approx. local Barometer |
| | -14' | Dynamic Suction Lift |
| | 17' | NPSH available |
| as per curve | - 8' | NPSH required |
| leaves | 9' | NPSH available |

in excess of the required, and pump is therefore suitable for the specified service.

Selection B:
Same pump as above but to deliver:
240 GPM against 40 feet total head of which 20 feet is suction lift.

| | | |
|---|---|---|
| | 34' - 3' = 31' | approx. local Barometer |
| | -20' | Dynamic Suction Lift |
| | 11' | NPSH available |
| as per curve | -15' | NPSH required |
| which is | 4' | more than available, therefore |

the Suction Lift must be reduced by at least 4 feet or a different pump selected as shown in Selection "C".



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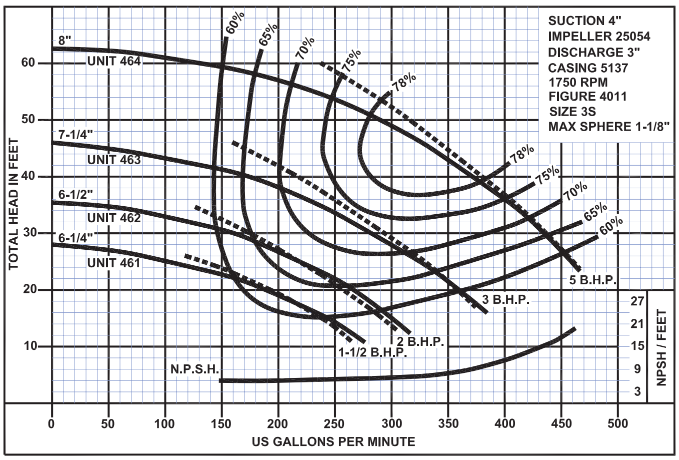
Selection C:
Pump requirements are the same as in Example 2, but using a larger pump. Refer to Curve Chart below of Fig. 4011, Size 3S, 1750 RPM.

| | | |
|---|---|---|
| | 34' - 3' = 31' | approx. local Barometer |
| | -20' | Dynamic Suction Lift |
| | 11' | NPSH available |
| as per curve | - 6' | NPSH required |
| leaves | 5' | NPSH available |

in excess of the required, and therefore suitable. In this instance a 7½" cut diameter impeller would be used with performance shown along heavy dash curve.




EXAMPLE NO. 2: NPSH calculations involving hot water or liquids with different vapor pressures would bring an additional factor into consideration - Vapor Pressure. The following example will give the required calculations:

Step No. 1. Figure the equivalent of the Barometric pressure in feet of the liquid being pumped at the elevation of the intended installation. This will be the maximum theoretical suction lift. (Note carefully that specific gravity will have to be taken into consideration).

Using Table No. 9 Par. XVI, the figures in Column 5 will be divided by the specific gravity of the liquid be in pumped. Call this result "A".

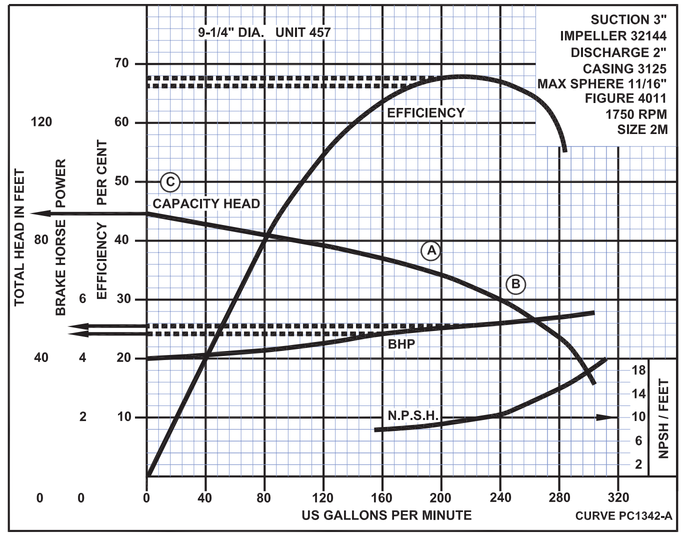
Step No. 2. Take the required NPSH of the pump and add it to the suction friction losses in feet, and the vapor pressure converted into feet of water. Call this result "B".

Inches of Mercury x .491 = PSIA
PSIA x 2.31 = Feet in Water

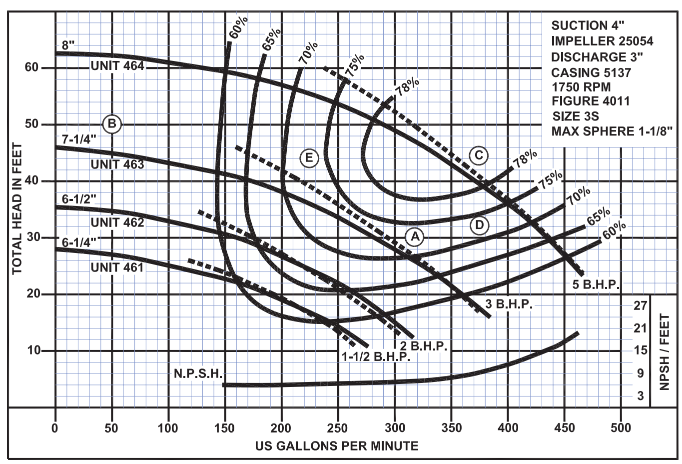
Step No. 3. Comparing the results of the first two steps:
If "A" is greater than "B" this difference will be the maximum allowable suction lift.

If "A" is less than "B" this difference will be the minimum required suction head.

XIII. Reading Understanding and Evaluating the Centrifugal Pump Performance Curve
The most convenient way to illustrate the operating characteristics of a centrifugal pump is through use of the characteristic curve. Three common types of curves, as used by Crane Pumps and Deming brand pumps, are illustrated in this section and are accompanied by explanatory material.




Curve No. 1 shows the performance characteristics of a Fig. 4011 No. 2M Pump at one speed (1750 RPM) with one impeller diameter (9¼").




Curve No. 2 shows the performance characteristics of a Fig. 4011 No. 3S Pump at one speed (1750 RPM) with four different impeller diameters.

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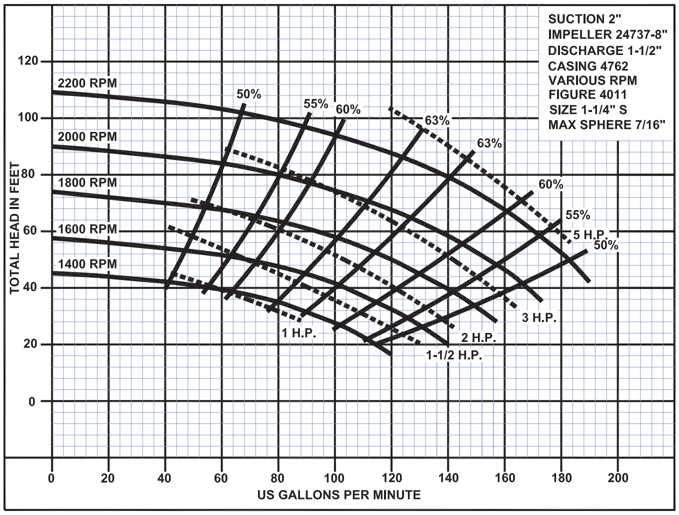


Curve No. 3 shows the performance characteristics of a Fig. 4011 No. 1½S Pump at five different speeds (1400 RPM to 2200 RPM) with one impeller diameter (8").

A. To Select the Capacity and Head:

Curve No. 1 - Read from Curve marked "capacity-head". Read GPM from bottom scale and head in feet from extreme left vertical scale. Thus, at point "A", the pump will deliver 180 GPM at a total head of 70 feet. At point "B" the pump would deliver 240 GPM at 59 TDH.

Curve No. 2 - As with Curve No. 1, read GPM from horizontal scale and read head in feet from vertical scale at left. At Point "A", therefore, this pump would deliver 280 GPM at 30 feet TDH with a 7¼" diameter impeller. (Read impeller diameter from Figure "B"). This pump would be, then, a Deming Fig. 4011 Size 3S, Unit No. 463. Should a pump delivering 360 GPM at 42 feet TDH be required, select at Point "C". This unit would have an 8" diameter impeller and would be ordered as a Deming Fig. 4011 size 3sm Unit No. 464. Should a pump be required to deliver, for example, 360 GPM at 30 feet TDH, as shown at Point "D", it is possible to operate this pump with a curve characteristic as indicated by the dotted Curve "E" by "cutting" the impeller to a diameter of approximately 7⅝". It is often very helpful to cut the impeller to keep the pump from overloading a motor, when a capacity and head developed by a standard diameter impeller is somewhat more than is required. Curves showing characteristics of standard diameters are shown on these curves, however, diameters falling between these sizes are available.

Curve No. 3 - Selection is made from this type of curve in a similar manner, except that these curves show characteristics of a given impeller diameter at various speeds. For speeds falling between those shown by the curves, the head and capacity may be found by interpolating between curves. For example, at 1750 RPM the pump shown in Curve No. 3 would operate along the dotted characteristics curve.

B. To find horsepower required:

Curve No. 1 - Read required horsepower from middle scale at left and from BHP Curve. Thus approximately 5 BHP. Operating at Point "B" the pump would require approximately 5.3 BHP.

Curve No. 2 and No. 3- The horsepower is read directly from the curve. Thus at Point "A" this pump would require just less than 3 HP at Point "C" just less than 5 HP, and at Point "D", approximately 4 HP.

C. To find pump efficiency:

Curve No. 1 - Pump efficiency is shown in a similar manner as is horsepower required. Simply use the curve marked "efficiency" and read from the right scale at the left of the graph. Thus, operating at Point "A", the pump efficiency would be about 65%; at Point "B", approximately 69%.

Curve No. 2 and No. 3 - Pump efficiency is shown directly on the curves in percent. At Point "A" for example, the efficiency is about mid-way between 70% curve and the 75% curve, or approximately 72.5%. 78% is peak efficiency for this particular pump and therefore, Point "C" is in the 78%
efficiency range.

D. To find the required Net Positive Suction Head:
(For example See Par. XII)

Curves No. 1 and No. 2 - Use same principle as finding efficiency on Curve marked NPSH in the lower portion of the graph. Thus, at Point "B" on Curve No. 1, the required NPSH would be 10 feet.

Curve No. 3 - NPSH information does not appear on variable speed curves, since each given speed would require a separate curve. These have been omitted therefore for the sake of clarity.

E. Shut-off:

The Shut-off Point is that height beyond which the pump will deliver no water. This height is shown on the vertical, left hand scale (ordinate). The point is different for each impeller diameter and for each speed. In Curve No. 1, for example, the Shutoff point would be at "C" or approximately at 89 feet. In other words unless the speed were increased, or the impeller diameter increased, this pump would not deliver any water above a total dynamic head of 89 feet. The importance of this Shut-off point can be shown from Curve No. 1. For example, if the pump were installed to deliver 180 GPM against the head of approximately 70 feet the pump would be operating at point "A", which is a point of rather high efficiency and the pump is well suited to this particular installation. However, if it were possible that the total head of this system would increase to a point beyond approximately 89 feet, this pump would not be suitable, since 89 feet is the shut-off point beyond which no water will be delivered.

F. Other information contained on performance curves:
Aside from capacity, head, horsepower, efficiency and required net positive suction head - additional information is also shown in many cases, on the performance curves. All of this information is important in planning a correct installation. Occasionally some of this information will be omitted, however, in general you may look for the following data:

Figure Number of Pump
Size of Pump
Unit Number
Sizes of Suction and Discharge Openings
Impeller Number and Diameter
Casing Number
Operating Speed
Maximum Size of Spheres which will pass through the pump

XIV. Electric Starting Equipment:
Since by far the majority of pumps are driven by electric motors, an outline of the various types of available starting equipment would be appropriate. This listing is by no means comprehensive, but should serve, rather, to suggest various possibilities to those who will be installing or purchasing pumping equipment. In selecting starting equipment, especially for larger motors, always refer to the rules and regulations set up by the local power company. Since these regulations will differ greatly from place to place, no generalization can be made here.

The most common methods of starting polyphase, squirrel cage motors are:

1. Full Voltage or Across the Line Starting - which makes use of a manual or automatic starting switch to throw the motor directly across the line. Full voltage starting is most common and can be used when ever the load can stand the starting shock and where no objectionable line disturbances are created.

2. Primary Resistance Starting - which makes use of a resistance unit in series with the stator of the motor, thereby reducing the starting current. This type has an advantage in that the motor is finally connected to full voltage without having its connection to the line interrupted at anytime.

3. Auto Transformer or "Compensator" Starting - which makes use of manual or automatic switching between taps of the auto-transformer to give reduced voltage starting. An automatic timer will then remove the auto transformer from the circuit and connect the motor across the line. The principle disadvantage of this type of starter is the open circuit transition.

4. Impedance Starting - which makes use of reactors in series with the motor. Not generally advantageous except in the case of high voltage and/or physical of resistors is a problem.

5. Star Delta Starting - in which the starter of the motor is star connected for starting and Delta connected for running. The motor must be specially wound with six leads brought out. The advantage of this type of starting lies with the lack of accessory voltage reduction equipment. While it does have the disadvantage of open circuit transition, a Star Delta starter does give a higher starting torque per line ampere than does a part winding starter.

6. Part Winding Starting - in which the motor stater windings are made up of two or more circuits, with each individual circuit connected to the line successively in starting, and in parallel for running. This type is somewhat less expensive than other types of reduced voltage starting and, as a further advantage the transition is inherently closed circuit. The disadvantage is that not all motors should be part winding started. The starting torque is rather poor, and the stater is almost always an increment start device. However, for many centrifugal pump applications the advantage of this type of starting equipment may well lead to its use. Always refer to the factory or to the motor manufacturer before installing a part winding starter.

In short, there are two fundamental methods of starting squirrel cage motors -

Full Voltage Starting (No. 1 above)
Reduced Voltage Starting (No. 2 thru 6 above)

Starting equipment is available in many NEMA enclosures for all types of installations. Start stop push buttons, hand off automatic selector switches and contacts for pilot devices are also available to be incorporated with any auxiliary, high or low, water cut off, etc.

For complex electrical controls, full details of what is required should be forwarded to the factory with your inquiry for pumping equipment.