TECHNICAL MANUAL

TEH-375A

Pump & System Curve Data for Centrifugal Pump Selection and Application

Bell & Gossett

a xylem brand

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INTRODUCTION

This section of the B&G Engineering Design Manual covers centrifugal pump performance curve and system curve data. Other sections provide additional information on the selection and application of centrifugal pumps. It should be noted that example curves used herein are illustrative only and manufacturers' data should be consulted for specific information.

DEVELOPING THE PUMP CURVE

Pump performance is shown by means of a plotted curve which relates the flow (gallons per minute) to the total head produced (head in feet of fluid.) The pump curve is established by the manufacturer under carefully controlled test conditions. All B&G pump test curves are plotted from performance data obtained by using standard production pumps selected at random. A typical head capacity curve is illustrated in Figure 1.

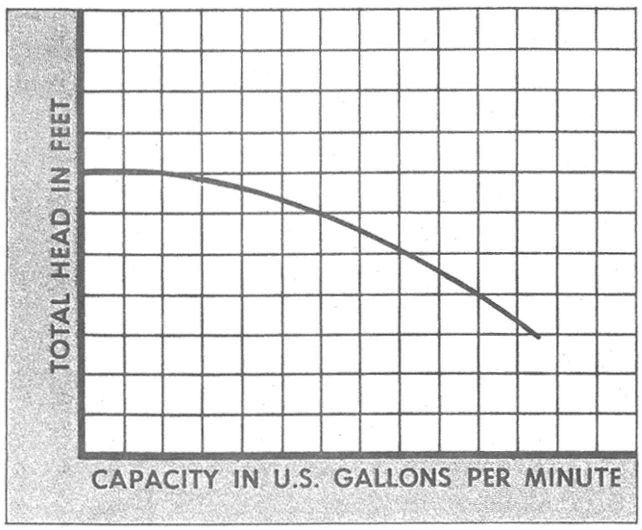


Figure 1 - Typical Head Capacity Curve

The developed pump curve is almost always illustrated as foot head vs GPM because this gives a general description of pump operation completely unaffected by water temperature or density. Some pump curves show PSI vs GPM or PSI vs lbs./hr. These curves are not general, but specific, and require definition in terms of water temperature and density. PSI curves are sometimes used for boiler feed pump description but should not be used in selecting pumps for Heating-Air Conditioning application. The GPM vs foot head capacity curve is general because of the physical characteristics of the centrifugal pump. The centrifugal pump produces energy in the form of foot pounds per pound of water pumped; and dependent on the volume flow rate passing through the impeller. In a manner of speaking, the pump raises each pound of water passing through it to an energy level at its discharge which is higher than that at the suction by the difference in foot head.

Energy as foot pound per pound is shortened to foot head by mathematical term cancellation. Since foot head is a simple energy statement, a pump curve defined by this term is not affected by water temperature change. This is because energy, as such, is not affected by temperature change. Likewise, water density has no effect on the pump curve – though density does affect pump power requirements.

A pump curve defined by a statement of foot head vs GPM is then a completely generalized curve – independent of liquid temperature and density. This pump curve, although established by 85°F. water tests, will apply to 40°F. water, 20°F. water or 40°F. water without change.

EFFECT OF VISCOSITY
ON THE PUMP CURVE

A change in liquid viscosity can affect the pump curve. However, the change must be greater than the change in water viscosity from 40°F. to 400°F. before the curve is affected. Thick, viscous liquids will markedly change the curve because of a great increase in the viscous shear force of the liquid being pumped. (see Figure 2) Information covering glycols and heat transfer oils is provided in another section of the Engineering Design Manual. Increased fluid viscosity will also affect pump power requirements because of the increased "drag" of the liquid within the pump.

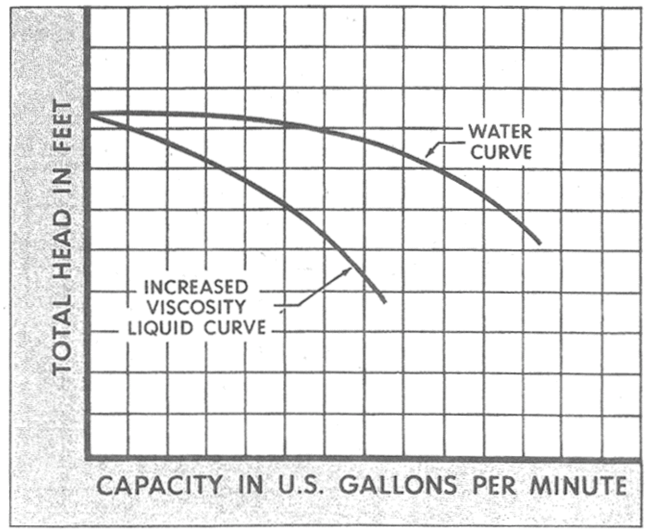


Figure 2 -Effect of Increased Viscosity on the Pump Curve

PUMP POWER REQUIREMENTS
A pump curve stated as GPM vs foot head is a power statement. At any point on the curve, the power put into the water (water horse power) is the water energy input rate. This is foot head (foot pound per pound) X GPM (converted to pounds per minute) andexpressed as water Horsepower (WHP).

$$\text{Water HP} = \frac{\text{GPM X Head X sp. gr.}}{3960}$$

A curve illustrating water horsepower input can be established for any specific pump as illustrated in Figure 3.

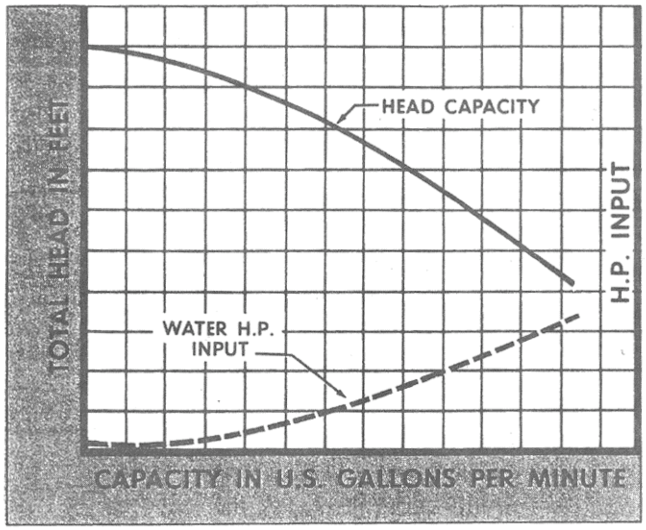


Figure 3 – Water Horsepower Input Curve

Water horsepower is zero at no delivery and increases with increasing flow; illustrating one of the characteristics of the centrifugal pump – power requirements generally increase with flow – even though head decreases. This is a most important point since an oversized pump (a unit operating at more than design flow rate) will draw more horsepower and may

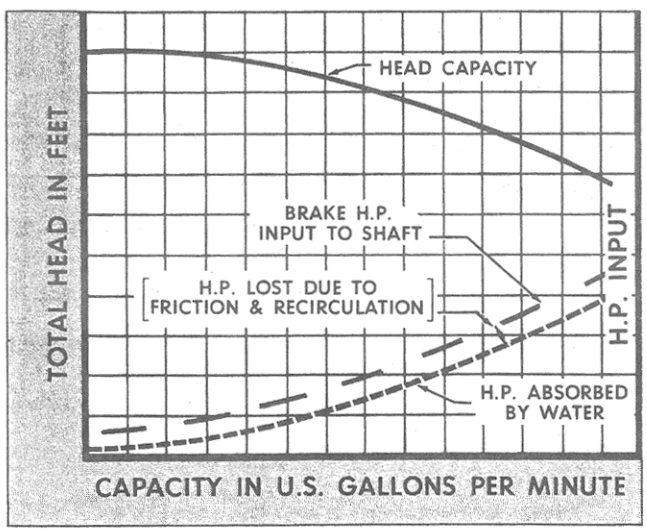


Figure 4 – Total BHP Requirement Curve

lead to a motor overload condition. Pump motor overload on the hydronic system must be avoided because water flow must be maintained in order to prevent freeze up, etc.

Actual power requirements at the pump shaft will be greater than the power absorbed by the pumped water. This is because of friction losses in the bearings, water friction itself and recirculation within the pump. These additional power losses define the total brake horsepower requirement (BHP) at the pump shaft as illustrated in Figure 4.

Water horsepower also increases with fluid density – even though the head capacity curve is not changed. This is because at any fixed GPM point more mass (more pounds per minute of fluid) is being pumped at the higher fluid density. If a fluid with twice the density of water were pumped, the required water horsepower would be doubled. (See Figure 5) The effect of fluid density must then be taken into account when evaluating horsepower requirements for fluids other than water.

Example

A fluid “A”, having a viscosity equal to water but with a specific gravity of 1.5 is to be pumped. A water base curve indicates a need for 3.2 BHP at the pump selection point. Problem: determine actual BHP and size of motor needed.

corrected BHP (fluid A) = BHP (water) X sp. gr. (fluid A) = 3.2 x 1.5 = 4.8 BHP

Solution: a 5 H.P. motor is selected.

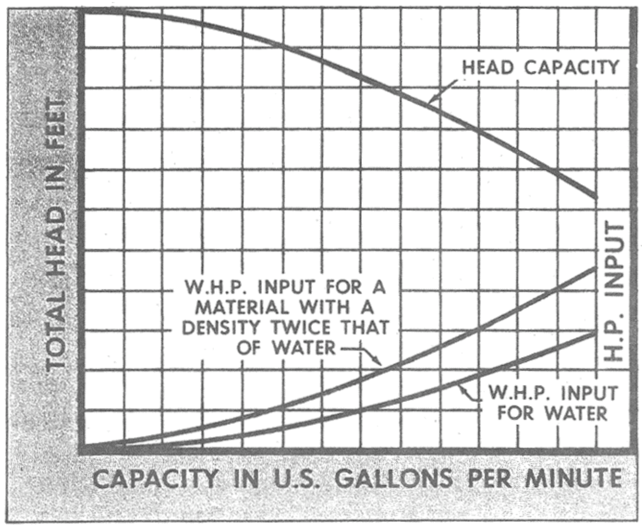


Figure 5 – Water H.P. & Fluid Density Curve

Pump test data is usually based on a water temperature of about 85° (specific gravity approximately 1). Pump power requirements apply to all water temperatures. Since the power requirements are based on near maximum water density the curve illustrates maximum pumping power. Pumping power will reduce slightly as water temperature is increased because of decreased water density; but this factor is seldom taken into account.

PUMP EFFICIENCY

Pump efficiency is defined as water horsepower output divided by pump shaft brake horsepower input.

Pump efficiency in percent is: $\frac{\text{WHP X 100}}{\text{BHP}}$

and is determined on the basis of manufacturers' tests using calibrated motors. As previously noted B&G tests are run on production pumps selected at random. The efficiencies illustrated could be increased by laboratory type modifications; smoothing of the impeller and internal water passageways. However, this modification would not be representative of production pumps.

An efficiency curve is shown plotted with the basic pump curve in Figure 6.

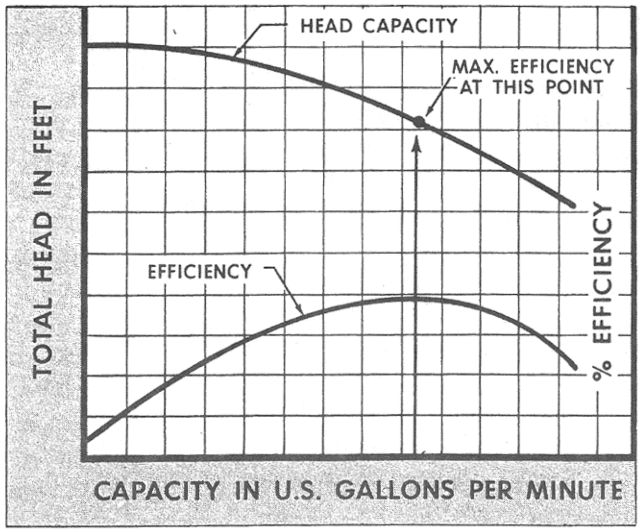


Figure 6 – Pump Efficiency Curve

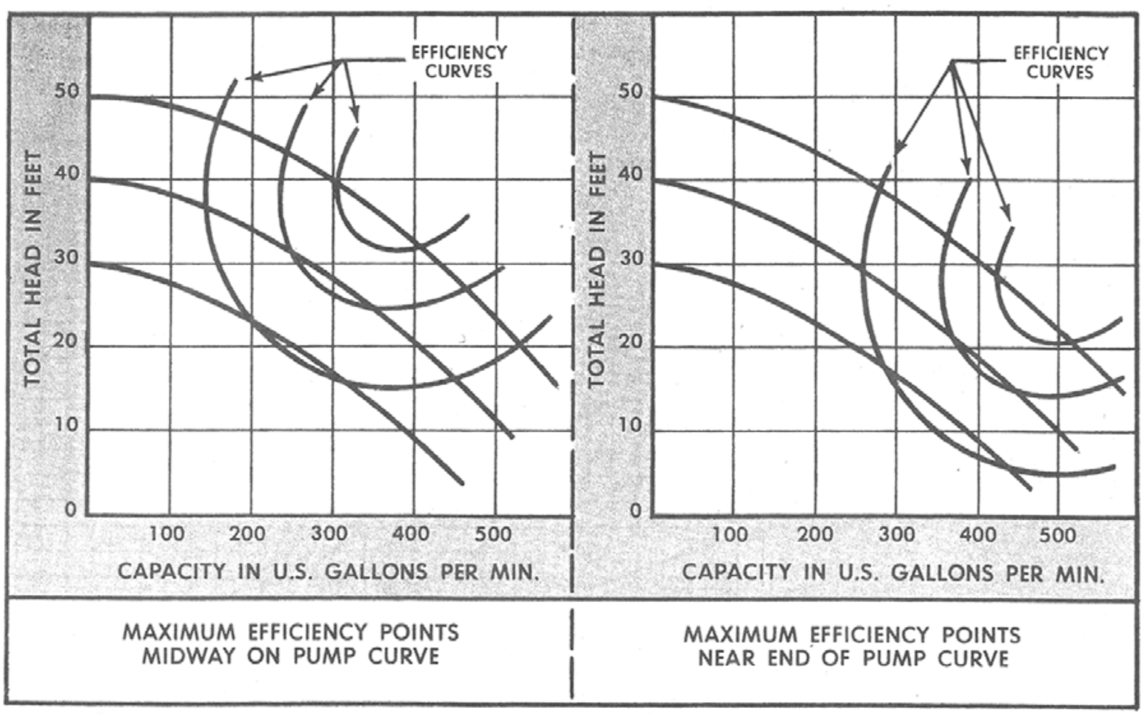


Figure 7 – Comparative Efficiency Range-Midway vs End of Pump Curve

Maximum efficiency occurs at a particular point or within a range on the pump curve. Efficiency decreases as flow either increases or decreases from its design point. The maximum efficiency point is dependent on impeller and volute design and indicates an area of flow where minimum impeller and volute turbulence is encountered. Pumps are often selected in the area of maximum efficiency. Selection of pumps to an efficiency only criteria may, however, be detrimental to both pump and system operation – especially when the maximum efficiency points are close to the end of the curve. Most hydronic system pumps have their maximum efficiency range about midway plus or minus ¼ of the published pump curve. Such pumps are most suitable to hydronic system application. (See Figure 7)

Pumps operating at 3450 RPM are not recommended for hydronic system application. This is because of pronounced shifting of the maximum efficiency pumping area to the right on the curve. Noise transmission difficulties may also be encountered.

Pump efficiency tends to increase with larger pump size. This is because bearing and other mechanical and internal hydraulic losses become a smaller proportion of required pump shaft BHP. When pump efficiencies are considered by the engineer as a basic criteria for pump selection, he must closely define the actual point of operation and predict with certainty the pump operational point. Oversized pumps will lead to greatly decreased system pumping efficiencies.

PUMP AFFINITY LAWS

The pump affinity laws provide methods for approximation of the effects of changed impeller diameters and rotational speeds on the pump curve.
These laws state:
1. Pump GPM capacity varies directly as the speed (RPM) or impeller diameter ratio change.
2. Pump head varies directly as the square of the speed (RPM) or impeller ratio change.
3. BHP varies directly as the cube of the speed (RPM) or impeller ratio change.

The relationships of the pump affinity laws can be formulated as follows:

	GPM Capacity	Ft. Head	BHP
Impeller Diameter Change	$Q_2 = \frac{D_2}{D_1} Q_1$	$H_2 = \left(\frac{D_2}{D_1}\right)^2 H_1$	$P_2 = \left(\frac{D_2}{D_1}\right)^3 P_1$
Speed	$Q_2 = \frac{R_2}{R_1} Q_1$	$H_2 = \left(\frac{R_2}{R_1}\right)^2 H_1$	$P_2 = \left(\frac{R_2}{R_1}\right)^3 P_1$

Where Q = GPM, H = head, P = BHP
D = impeller dia., R = RPM

Pump affinity laws can be used for resizing impellers on pumps known to be providing excessive system flows. However, system curve analysis methods provide more definite procedures.

Pump affinity laws can also be applied to pump selection for speeds different than the illustrated curve RPM base.

Example

A particular piping circuit may have a high flow-lowhead requirement and a suitable pump cannot be found at normal speeds (1750 RPM). While small pumps in parallel will often provide a solution, it is decided to apply speed reduction to 1150 RPM.

Design conditions are 250 GPM @ 20' head. Since most pump curves are stated to 1750 RPM, the

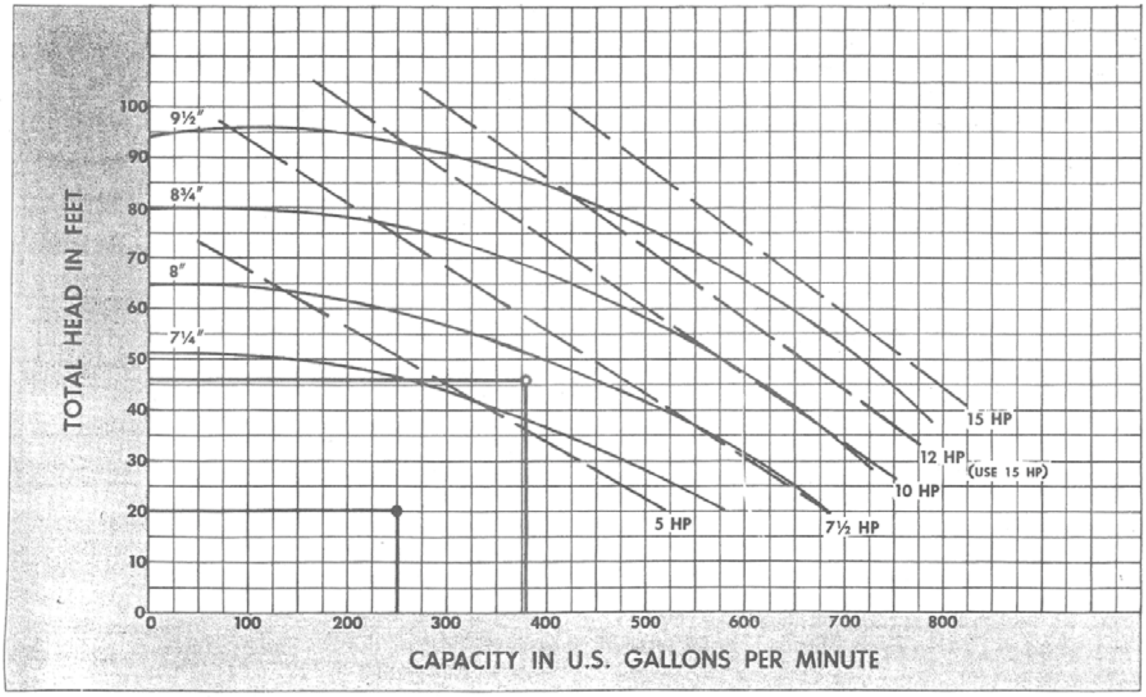


Figure 8 – Typical 1750 RPM Pump Curve

selection problem is to restate 250 GPM @ 20' head and at 1150 RPM to equivalent 1750 RPM conditions.

speed Change Capacity Formula: $\frac{R_2}{R_1} \times Q_1 = Q_2$

New Flow Rate: $\frac{1750}{1150} \times 250 = 380$ GPM

Therefore: 250 GPM @ 1150 RPM is equivalent to 380 GPM @ 1750 RPM.

Speed Change Foot Head Formula: $\frac{R_2}{R_1} \times H1 = H2$

New Foot Head: $\frac{1750^2}{1150} \times 20' = 46'$

Therefore: 20' head @ 1150 RPM is equivalent to 46' @ 1750 RPM.

Reference is now made to standard 1750 RPM curves or the condition 380 GPM & 46'. The selection point appears on pump curve as illustrated in Figure 8.

A 7 ¾" impeller size is indicated and 6 ½ BHP (approximately) is required for non-overload operation at 1750 RPM. BHP requirements at 1150 RPM will be:

Speed Change Power Formula: $\frac{R_2{}^3}{R_1} \times P_1 = P_2$

New Power: $\frac{1150^3}{1650} \times 6.5 =$ I .9 BHP @ 1150 RPM

A selection for 250 GPM at 20' head is met by using a 7 ¾" impeller and 1150 RPM 2 HP motor.

The procedure outlined above can also be followed for pump selection when using 50 cycle motors. The 50 cycle motor operates at 5/6 of 60 cycle speed. The comparative 1750 RPM is then 1750 X 5/6 or 1450 RPM. Selection procedure is the same as the preceding example; the only difference being substitution of 1450 RPM for the 1150 RPM illustrated.

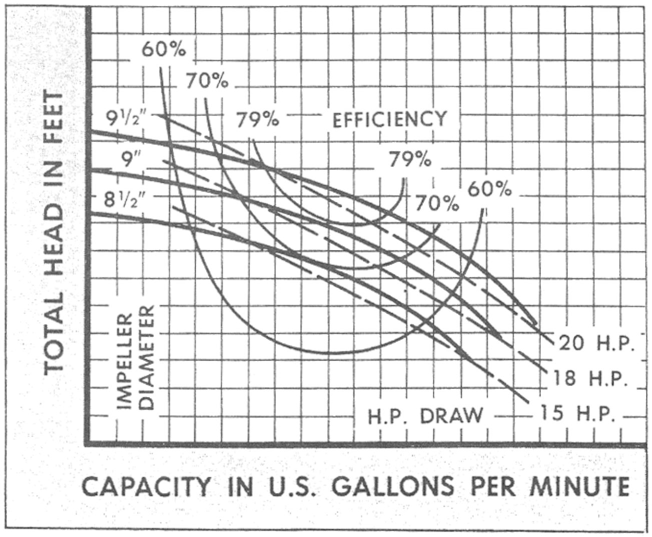


Figure 9 – Impeller size Capacity Curve

MANUFACTURERS PUMP SELECTION DATA

The capacity curve for a particular pump volute will vary as a function of impeller diameter. This permits the manufacturer to provide a series of pumps, using a single volute design by changing impeller diameters. Figure 9 illustrates a series of such curves, which include pump efficiencies and required motor H.P. draw for the various impeller sizes shown.

Quiet operating pump selection curves do not include efficiency information-despite relatively high pumping efficiencies. This is because of conviction that the selection basis for these specially designed pumps should be clearly and unmistakably defined as quiet operation. The best selection range for these pumps is indicated by a clear or lightly shaded area. See Figure 10.

The best selection range is about the midpoint of the capacity curve plus or minus ¼ (see Figure 10). This is the normal intended area of pump usage and provides for good pump operation. Pumps are often selected, or operate, to the left of this area; in the low flow, high head range. This is often proposed because of the “drift” of the actual operating point to the right of the specified point and due to highly complex variables including piping circuitry system pressure drop and control valve operation. Operation to the left of the midway operating range does not introduce operating problems-except for heat dissipation and bearing thrust loads when large pumps are used at or near to dead shut-off. It should be noted that two way open-close or modulating type control valves will often cause a shift in the actual pump operating point from a specified “midway” point to almost shut-off. Selection of a pump toward the left of the midpoint of the curve does not introduce the possible hazards of a selection made to the right of the midpoint of the curve.

Selection to the right of the curve midpoint will not provide a cushion against possible operation beyond the end of the published curve. Operation in this area causes noisy cavitational operation, possible pump damage and reduced flow to the system “end circuits”.

A shift in the actual pump operating point to the right of the specified point will occur if circuit flows are left unbalanced, if three way control valves are applied or if improper pressure drop charts are used. A pump selected midway or toward the left of the curve will permit considerable shift to the right before trouble is encountered. The pump selected close to the right end of the curve has no “shift allowance” and operating trouble can occur.

The importance of providing shift allowance can only be illustrated after system curve analysis methods have been explained. Detailed pump selection procedures are covered in another section of the Engineering Design Manual.

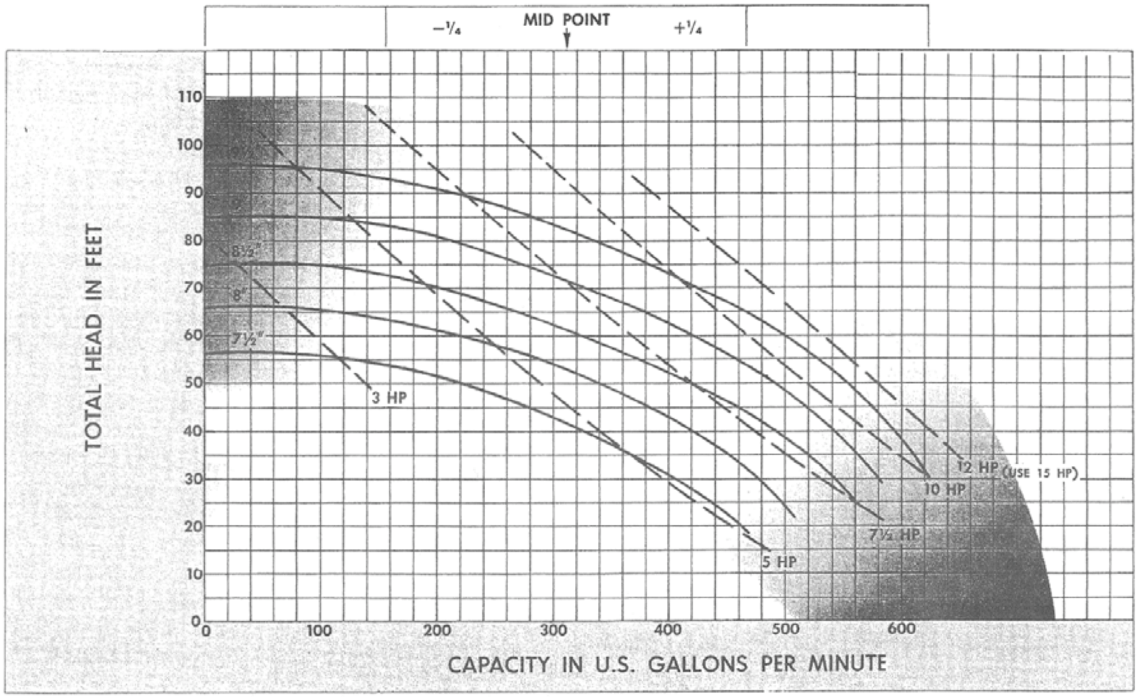


Figure l0 – Best Selection Range

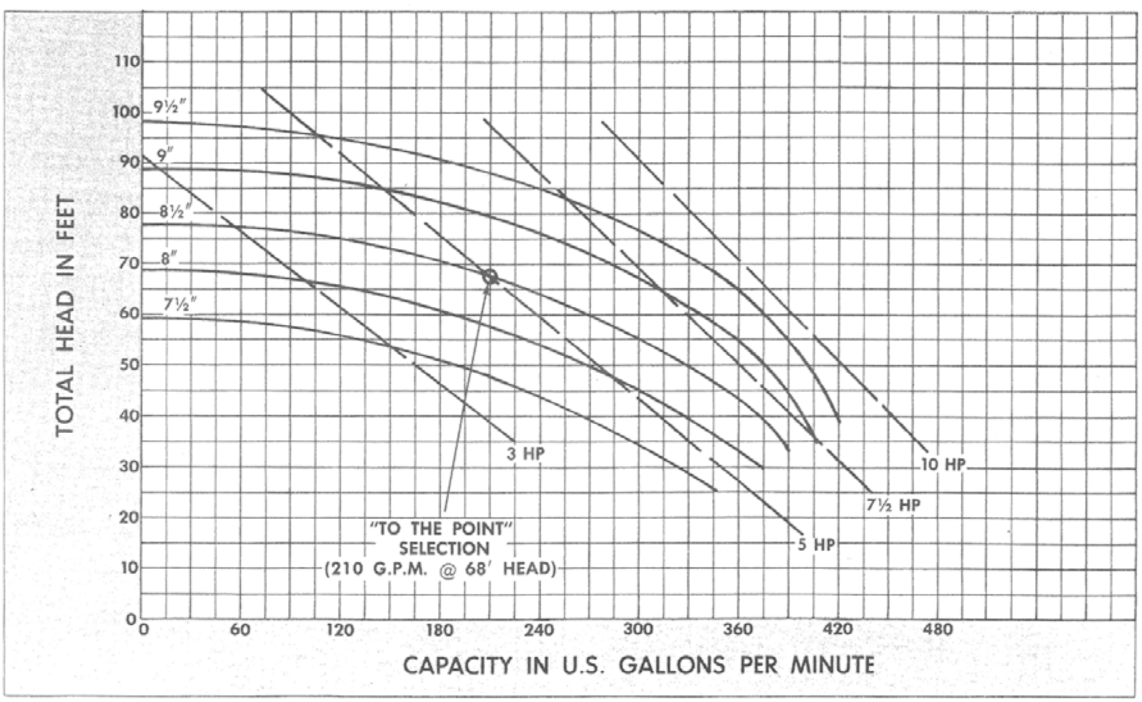


Figure 11 – Motor Selection on Pump Curve

The manufacturers composite pump curve illustrates power requirement (pump shaft BHP) in standard motor H.P. increments. The BHP for a specific operating point can be interpolated between the motor ranges shown.

Motors can be selected for either non-overload characteristics or “to the point.” On the pump curve illustrated in figure 11; a 5 H.P. motor selection would be “to the point” for the 8 ½” impeller curve at 210 GPM and 68 ft. head.

The “to the point” motor selection does not provide for pump operating point shift to the right and may lead to motor overload with consequent system flow stoppage and freezeup possibilities.

For the example illustrated a 7 ½ H.P. motor selection would be non-overloading over the entire 8 ½” impeller curve range.

Use of a 7 ½ H.P. motor for an approximate 5 H.P. draw does not appreciably increase total motor power requirements. A comparison is illustrated in figure 12 for motor power draw between a 5 H.P. motor at full load and for a 7 ½ H.P. motor at a 5 H.P. load.

The “to the point” selection for the pump using the 5 H.P. motor (210 GPM @ 68’ Head) indicates exactly 5 H.P. draw at 1750 RPM operation. When using the 7 ½ H.P. motor the power draw will increase only slightly beyond 5 H.P. As illustrated in figure 12 the pump curve will shift up slightly because of the RPM increase of the larger motor operating at partial load. The new curve can be plotted in accordance with the pump affinity laws. However, the actual power draw change will be even less than shown because of the higher efficiency of larger motors. The new point of operation for the 7 ½ H.P. motor and pump combination can be determined by application of the system curve to the new pump curve. Because of the negligible change in power draw between a “to the point” motor selection and a non-overloading motor selection it is not necessary to follow the above procedure in actual practice. Provision for non-overload motor operation will not significantly increase actual power draw.

Pumps should be selected as non-overloading over the entire pump curve. This characteristic is most important for hydronic system operation, especially when pumps are to be used in parallel or when the pump operating point is relatively indeterminate and subject to shift; as in the normal heating-air conditioning application.

Motor selection to the operating point frequently results in a smaller motor than the non-overloading

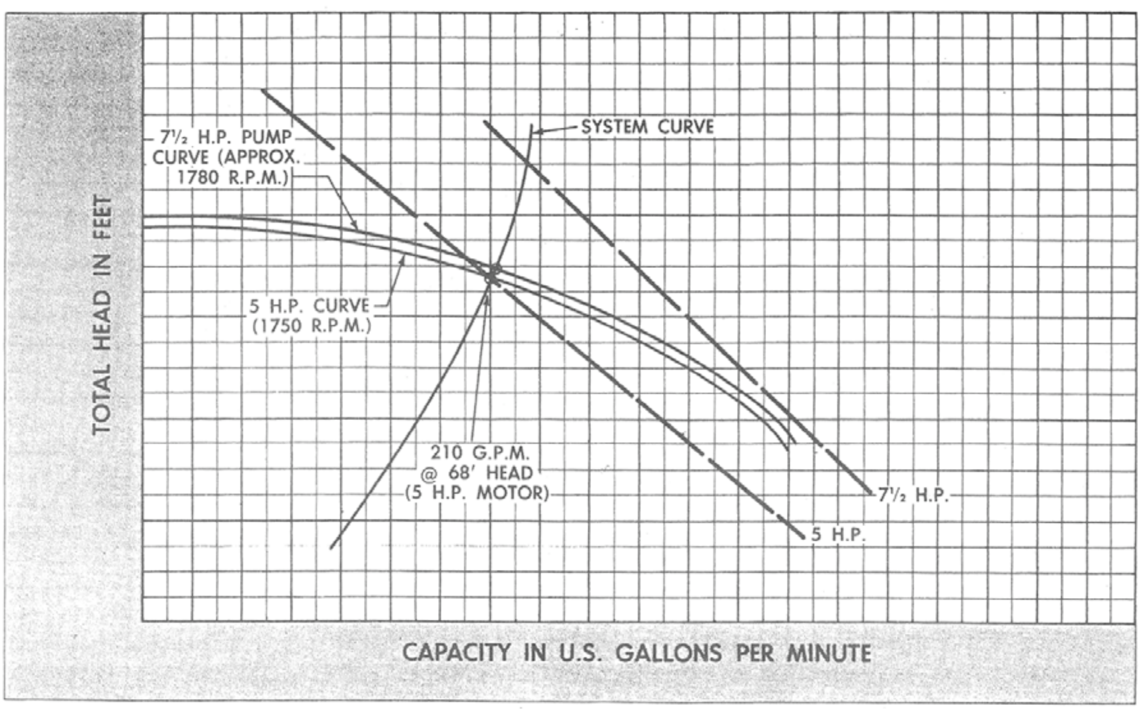


Figure 12 – Comparison of 5 H.P. vs 7 ½ Motor Power Draw

motor selection. Thus, the initial cost of the pump-motor combination is less. However, selection "to the point" must be made with careful analysis by system curve methods and all variables must be properly evaluated.

The BHP information included on the composite pump curve permits an approximation of pump efficiency.

Example

@ 210 GPM, 68 foot head and 5 BHP – the approximate efficiency will be:

$$WHP = \frac{GPM \text{ X } FT\ HD}{3960} = \frac{210 \times 68}{3960} = 3.6$$

3961

BHP =5

$$EFF = \frac{WHP}{BHP} = \frac{3.6}{5} = 72\%$$

The fact that a large motor may be applied for non-overload operation does not affect pump efficiency since pump shaft BHP usage will remain the same.

Care should be taken when applying pump curve H.P. for an efficiency approximation that the illustrated H.P. does not include motor service factor.

Motor service factor is the motor manufacturers safety factor and "down rates" the actual motor H.P. ability to provide against low voltage operation etc.

On standard open type A.C. motors of 3 H.P. and over a service factor of 1.15 is applied. The service factors increase with decreased H.P. as illustrated in table 1.

HORSEPOWER	SERVICE FACTOR
1/20	1.4
1/12	1.4
1/8	1.4
1/6	1.35
1/4	1.35
1/3	1.35
1/2	1.25
1	1.25
1-1/2	1.20
2	1.20
3 & up	1.15
Standard Open A.C. Motors Only (40°C. Rise)	

TABLE 1 – ELECTRIC MOTOR SERVICE FACTOR

B&G stock pumps have motors which provide for non-overload operation for their complete curve range.

Pump Curve for 3450 RPM Operation.

Pump curves are provided for several motor speeds. While 1750 RPM operation is recommended for the hydronic system, 3450 RPM operation is often used for industrial and some cooling tower applications.

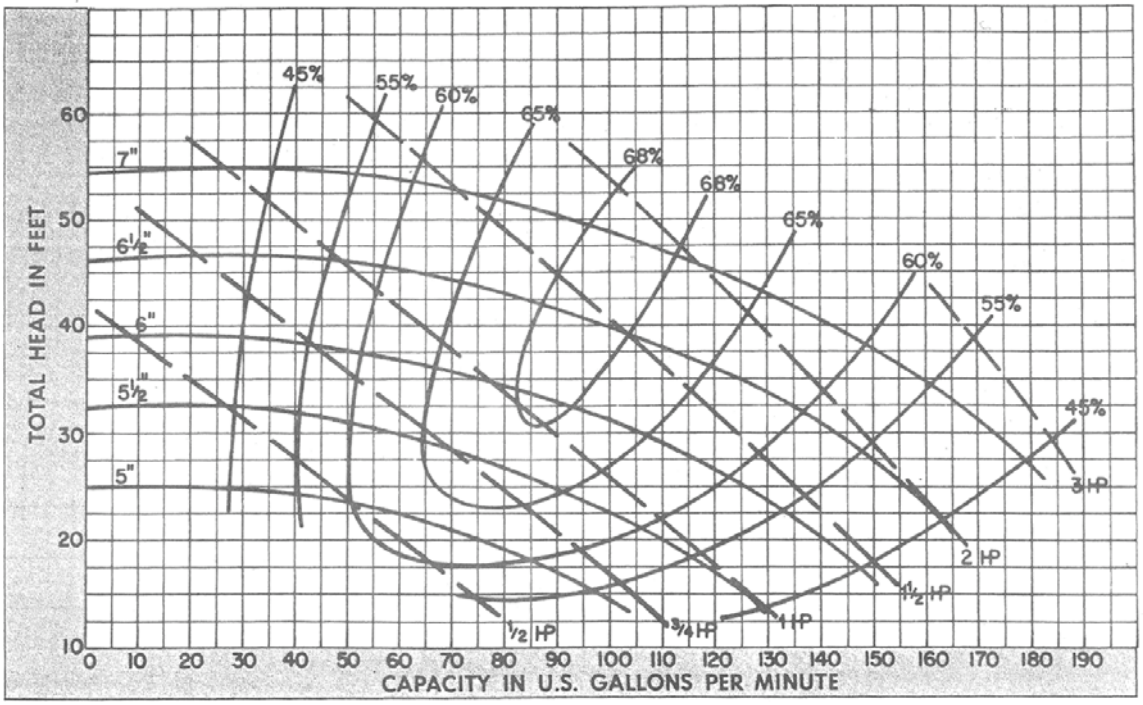


Figure 13 – Pump Curve for 1750 RPM Operation

The 3450 RPM pumps are not recommended for comfort heating air-conditioning application, because of the possibility of noise transmission into the system.

Operation at a higher speed raises the pump capacity curve. The pump curve efficiencies and brake horsepower draws are established by test and illustrated in the same manner as for 1750 RPM operation. Figures 13 and 14 show the pump capacity curve comparison for 1750 RPM and 3450 RPM operation using the same volute and impeller diameters.

Variable Speed Pumps

Variable speed pumps are used in many pumping applications. Variable speed pumping is covered in detail in another section of the Engineering Design Manual. A variable speed pump curve is illustrated in Figure 15.

"Available from Stock" – Pump Curves

Line mounted pumps are usually illustrated as a family of pump curves. These pumps are generally, but not always, powered for non-overloading characteristics throughout their entire pumping range. The pumps are mass produced and immediately available from stock. Their illustration as a family of curves allows simple use of system curve analysis for pump selection. A family of pump curves for line mounted pumps is illustrated in Figure 16.

Families of pump curves are also illustrated for quiet operating base mounted integral horsepower pumps designed for hydronic systems. (see Figure 17) These pumps are available for quick delivery. Like the line mounted pumps, they can be powered to the end of their published curve and can be used without fear of motor overload. Again, this method of illustration permits simple application of system curve analysis – a basic tool used by the engineer in his pump selection.

Net Positive Suction Head (NPSH) Curve

Net Positive Suction Head (NPSH) is a pump term that has extremely limited usage for closed loop piping circuit application and properly applied cooling tower pumps. It is of great importance, however, in open circuit industrial application where low

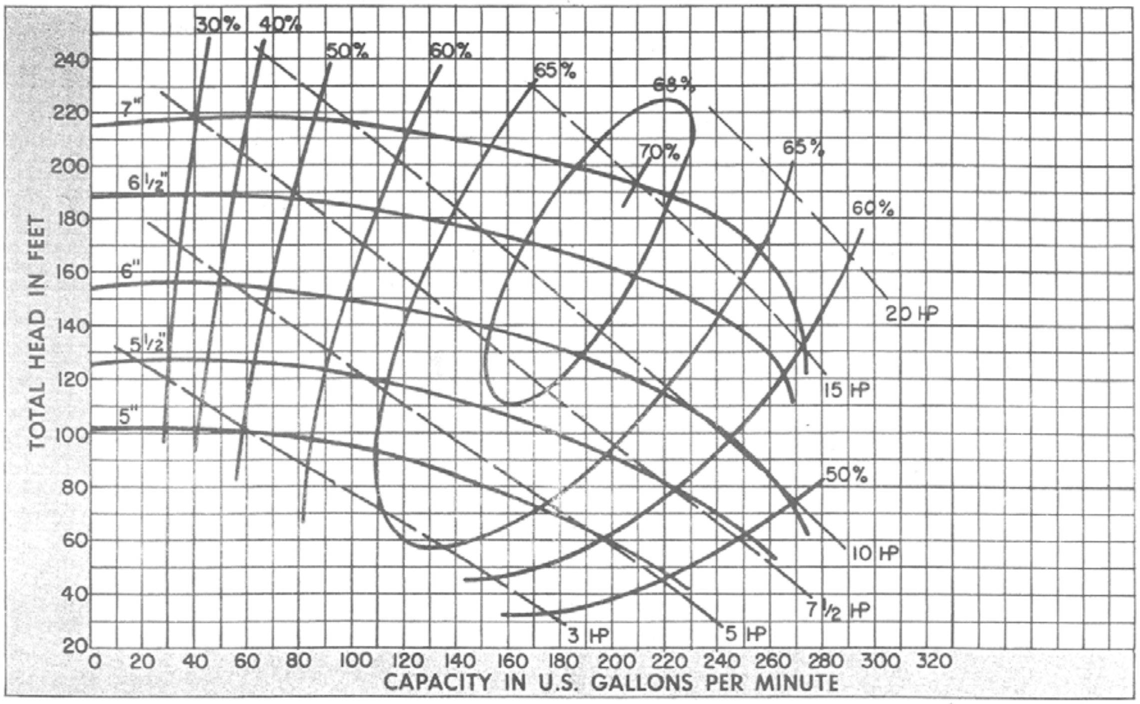


Figure 14 – Pump Curve for 3450 RPM Operation

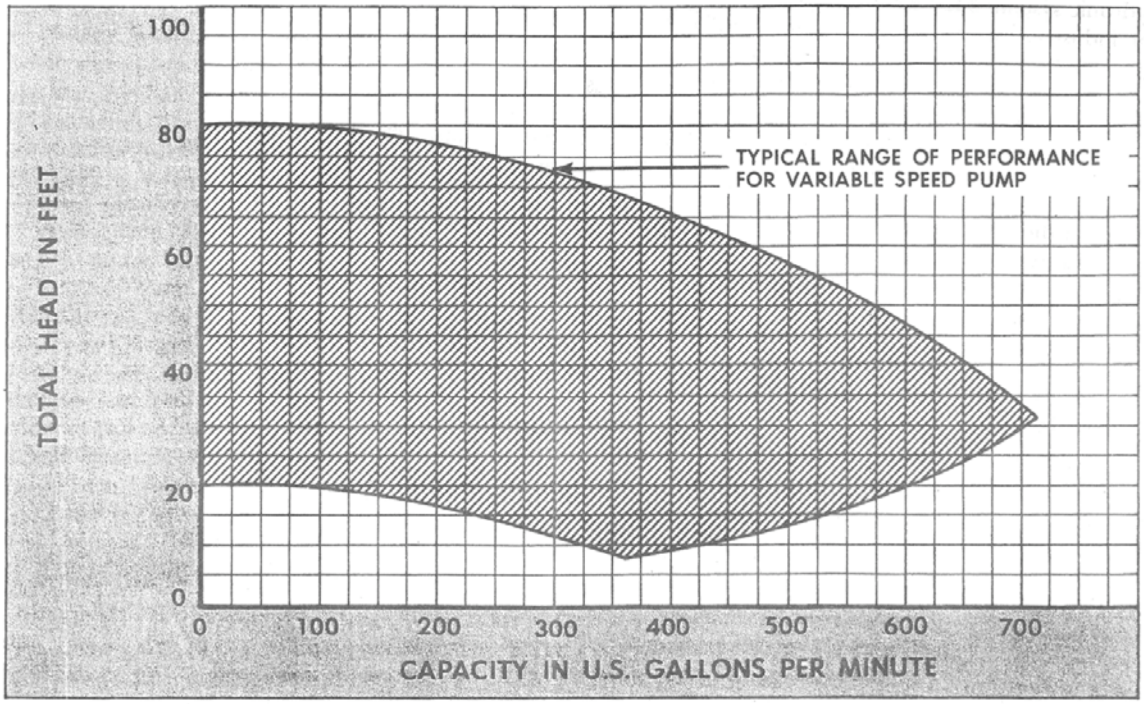


Figure 15 – Variable Speed Pump Curve

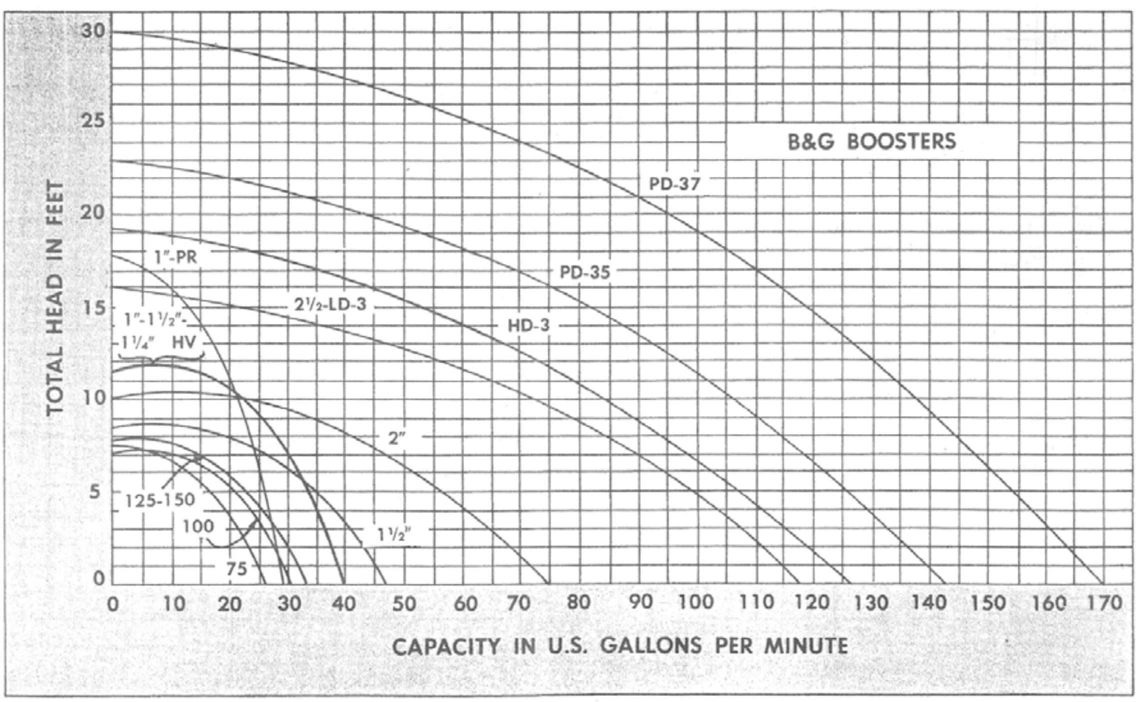
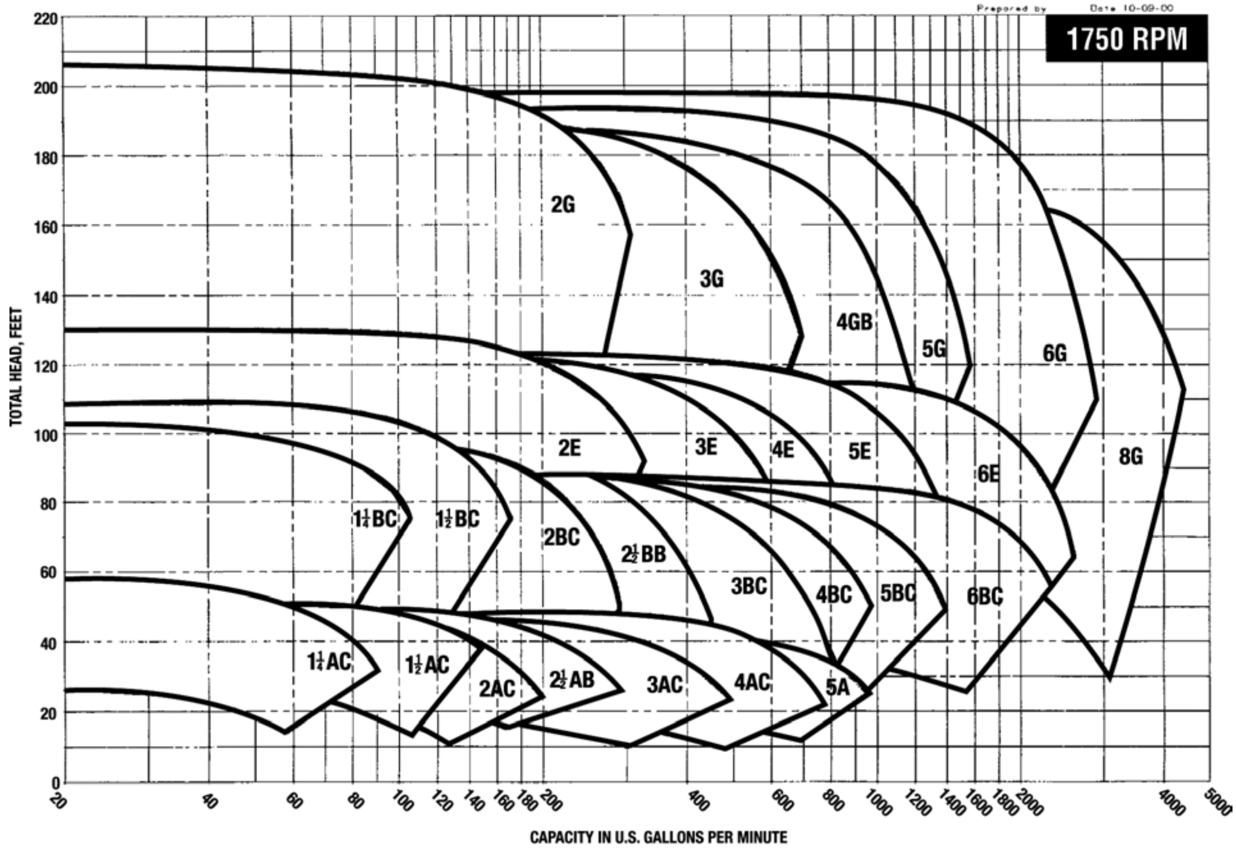


Figure 16 – Family of Pump Curves For Line Mounted Pumps



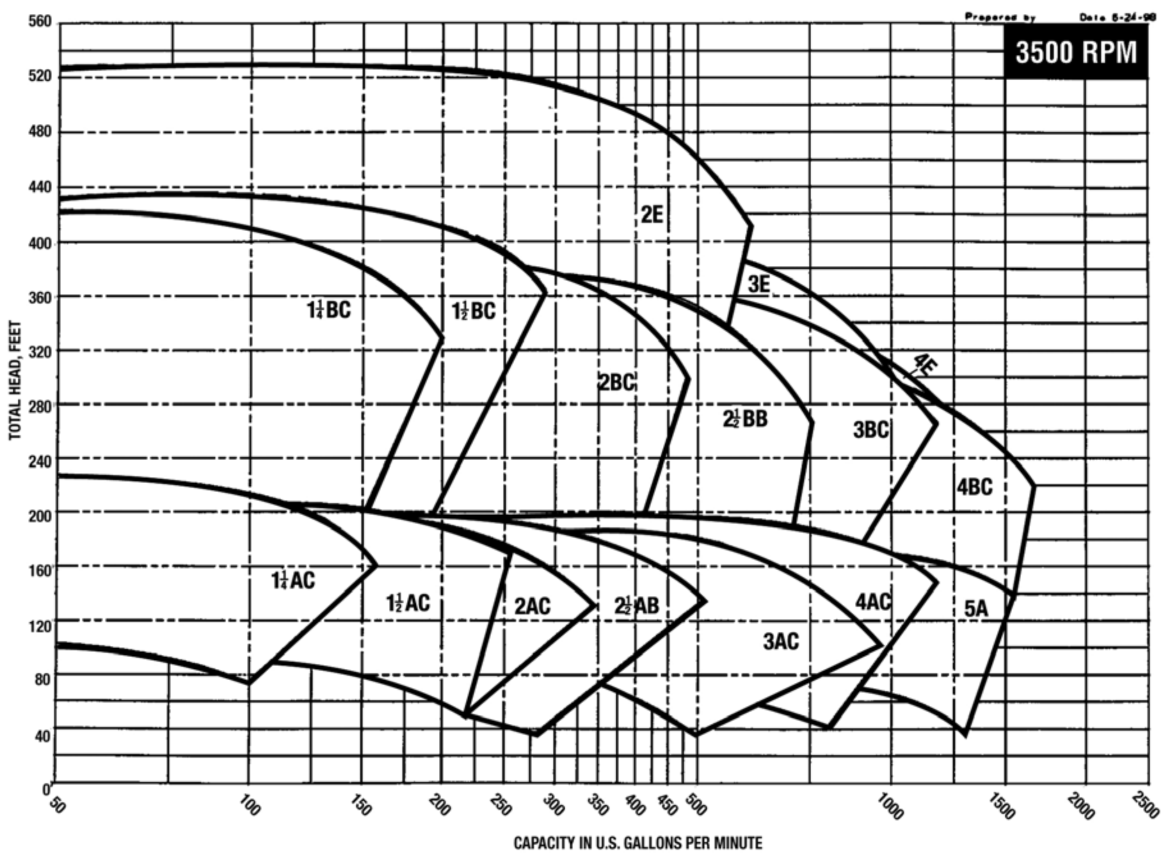


Figure 17 – Family of Pump Curves For Base Mounted Pumps

Suction Pressures are encountered. Pump manufacturers provide NPSH curves for specific pumps. A typical NPSH curve is shown in Figure 18.

NPSH curves are needed because all centrifugal pumps operate at a lower pressure in the impeller eye than the pressure existing at the pump suction flange. The decrease in pressure within the impeller eye is caused by greatly increased water velocity as water enters the working pump parts. The NPSH curve defines the pressure over and above fluid flash point or vaporization pressure which is needed at the pump impeller eye and takes into account decreased pressures within the pump. It should be noted that NPSH shown on the pump curve indicates required NPSH and increases with increased flow (increased water velocity).

NPSH evaluation is often needed for industrial open system pump application, especially when using volatile liquids. This is because interior pump pressure reduction can cause cavitation and pump damage. Cooling tower pump selection seldom needs NPSH evaluation provided proper system design and installation procedures are followed.

Most closed loop heating and air conditioning system pump application do not require NPSH evaluation because it is possible to obtain adequate pump suction pressurization.

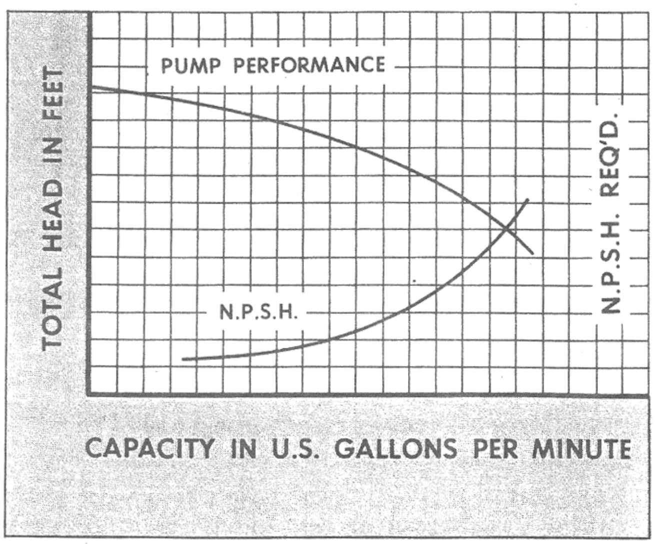


Figure 18 – Typical NPSH Required Curve

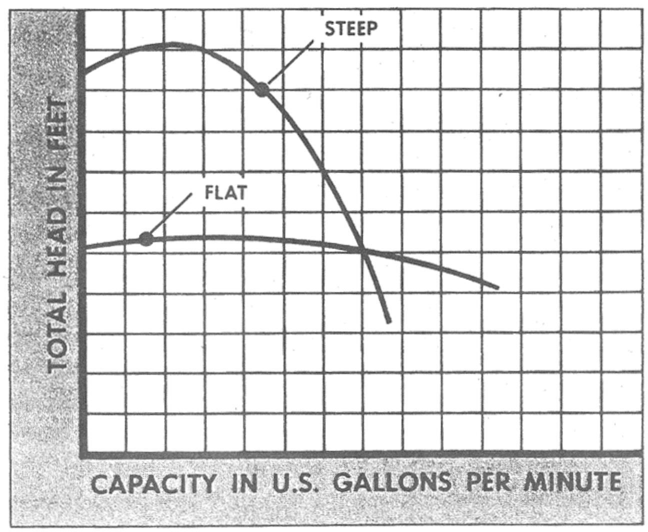


Figure 19 – Flat vs Steep Pump Curve

GENERAL PUMP CURVE CHARACTERISTICS

Pump curves are arbitrarily classified as “flat” or “steep.” This refers to the general shape of the head capacity curve as illustrated in Figure 19.

Flat-curve pumps are generally preferred for closed circuit systems because of the influence of the pump curve on the system operating components. Large changes in capacity can be achieved with a small change in head. This is advantageous in balancing multi-circuit systems. Flat curve pumps should also be selected for systems using valve control. The flat curve pump offers a more nearly stable pressure drop ratio across the valves as they go to the closed position and decreases control valve “force open” possibilities.

The pump curve permits use of system curves as a method of pump selection analysis. The system curve can be used as a tool to provide better understanding of pump selection and operation as it affects overall system operation.

GENERAL DESCRIPTION OF SYSTEM CURVE

System curve analysis methods provide means for determination of good pump application practice. Parallel and series pump installations require system curve analysis if operating problems are to be avoided. System curve analysis will help define the operating relationships between the pump, control valves, piping circuitry and heat transfer elements for heating-air conditioning systems. The analysis can be applied to permit engineered integration of operating variables. The system curve is simply a plot of the change in energy head resulting from a flow change in a fixed piping circuit. System curve construction methods differ between open and closed piping circuits.

SYSTEM CURVE FOR CLOSED SYSTEMS

A typical closed loop piping circuit is shown in Figure 20.

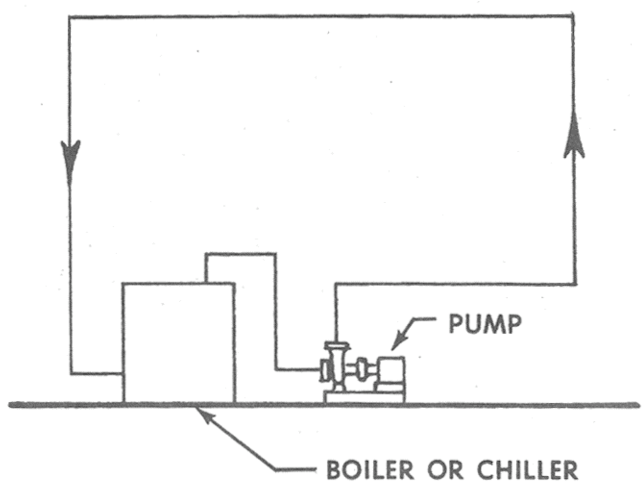


Figure 20 – Typical Closed Loop Piping System

From the pipe size and design flow rate, a calculated energy head pressure drop is determined. It should be particularly noted that system static height is of no importance in determining energy head pressure drop. This is because the static heights of the supply and return legs are in balance; the energy head required to raise water to the top of the supply riser is balanced by the energy head regain as water flows down the return riser.

Example

A design flow rate of 200 GPM establishes 30 foot pressure drop in a typical system. This particular point can be plotted on a foot head versus GPM chart as shown in Figure 21. What pressure drop would occur were the flow changed to 125 GPM through the piping circuit? Another calculation would indicate that 11.8 ft. of head is needed. The same procedure carried out for 75 GPM flow rate would result in a 4.2 foot pressure drop. These points can also be plotted on the foot head versus GPM chart as shown in Figure 21. Connection of these three points describes a “system curve.” The system curve is a statement of the change in pipe friction drop with water flow change for a fixed piping circuit. This is a most important working tool for pump application.

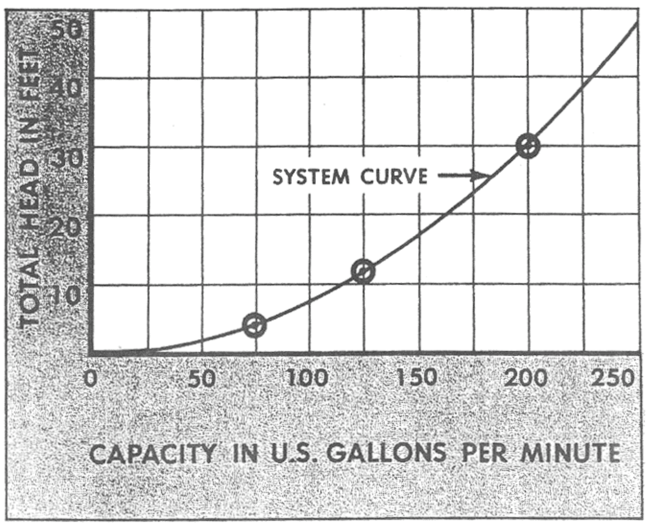


Figure 21 – System Curve Construction

The calculations described above are not needed to establish a system curve. This is because pipe friction drop varies in a mathematical ratio with the change in water flow rates. Head will change as the square of the change in water flow. This relationship is described below:

$$\frac{Q_2^2}{Q_1} = \frac{h_2}{h_1}$$

Q1 = known (design) flow
Q2 = final flow
H1 = known (design head
H2 = final head

Application of the above relationship is incorporated into scales of the System Syzer and provides a method by which the system curve can be quickly established. The preliminary calculations provide the known head and GPM values.

The “known” head is lined up to "known" GPM. For the example stated above; 30 ft. is lined up with 200 GPM. This is illustrated in Figure 22.

Figure 22 – Scale 5 of the B&G System Syzer

Head and GPM relationships are made immediately available. In this case 30 ft. head at 200 GPM are the known values. A tabulation for foot head requirement at other flows is illustrated below:

G.P.M.	FT. HD.
115	10
165	20
185	25
200	30
215	35
230	40

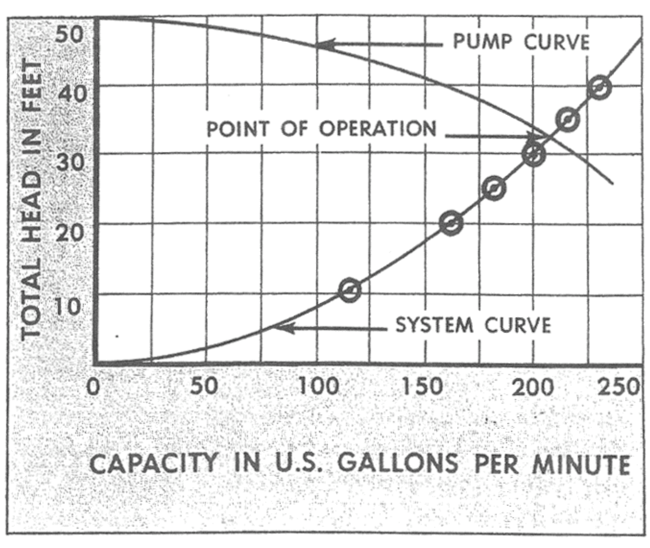


Figure 23 – System Curve Plotted on Pump Curve

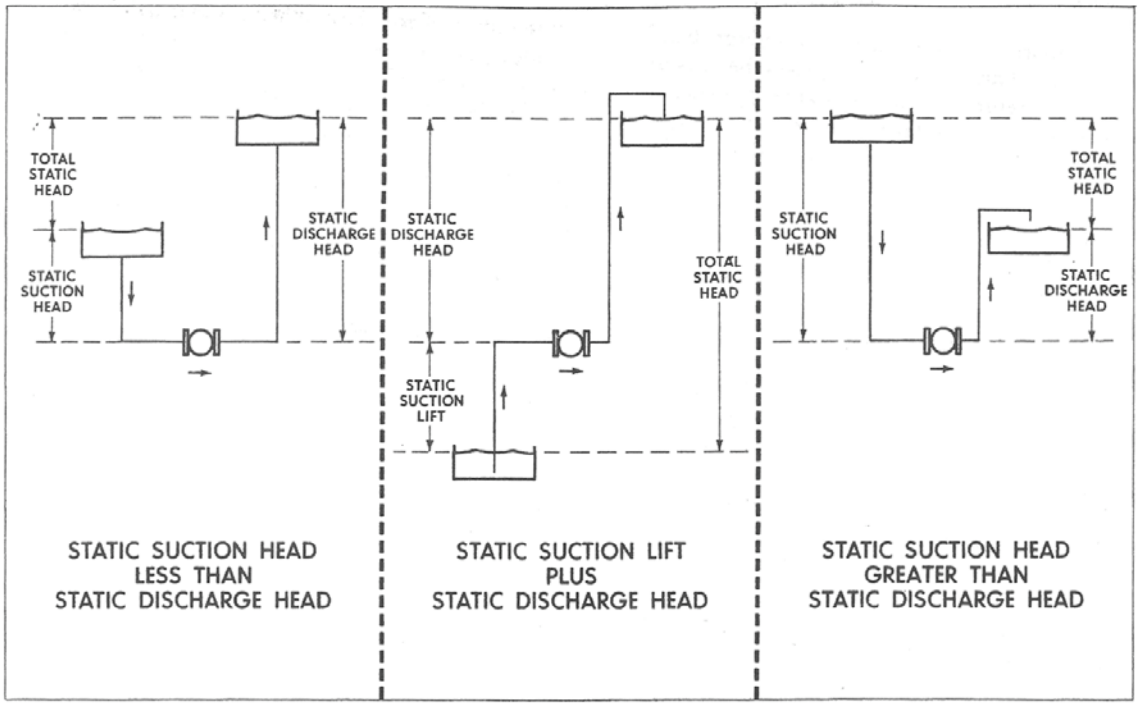


Figure 24 – Typical Open Systems

The complete system curve is illustrated in Figure 23 as plotted on a pump curve.

The operation of this pump on the piping circuit described by this system curve must be at the intersection of the pump curve with the system curve. This is because of the First Law of Thermodynamics – ENERGY IN MUST EQUAL ENERGY OUT. Energy put into the water by the pump must exactly match the energy lost by water as it flows through the piping circuit. Water flow through the piping circuit must match pump flow. The point of intersection is the only point that can meet this basic engineering law. For the example, the pump will operate at about 210 GPM and 32 ft. head- providing the specified selection points (200 GPM, 30 ft. head) are true statements of the system flow and pressure drop relationship.

The system curve, as constructed above, is continually used in system design and pump applications.

SYSTEM CURVE FOR OPEN SYSTEMS

In plotting the system curve for an open system the statics of the system must be analyzed in addition to the friction loss. The different static conditions are illustrated in Figure 24.

A typical cooling tower application is illustrated in Figure 25. In this system, the pump is drawing water from the tower sump and discharging it through the condenser to the tower nozzles, at a 10 foot higher elevation than the sump level.

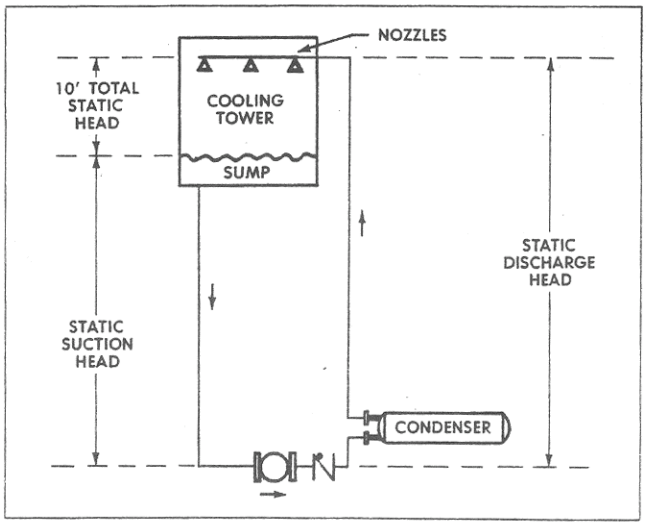


Figure 25 – Typical Cooling Tower Application

Total friction loss (suction & discharge piping, condenser, nozzles, etc.) is 30 foot at a design flow rate of 200 GPM. Using the design point of 30 foot at 200 GPM, the "slide rule procedure" previously described can be used to determine the change in piping pressure drop for a change in water flow rates.

This change is shown below:

G.P.M.	Foot Head
115	10
165	20
185	25
200	30
215	35
230	40

The points shown can be used to develop a system curve. This system curve cannot be applied directly to the pump curve and the intersection taken as the accurate pumping point for the open system. A false evaluation using this criteria, but without evaluating the static height of the tower is shown in Figure 26.

Figure 26 – System Curve for Open Circuit – False Operating Point

The illustration is false because the pump must also provide the necessary energy to raise water from the tower sump to the spray nozzles. In this case, the pump must raise each pound of water 10 ft. in height, or stated in another way, it must provide 10 ft. of energy head due to the static difference in height between the water levels.

The static difference of 10 ft. must be added to the piping pressure drop to provide total required head for each of the GPM points previously noted. The total required head is shown in Table 3.

GPM	0	115	165	185	Design 200	215	230
Pipe & Valve Pressure Drop	0	10	20	25	30	35	40
+Static Energy Head	10	10	10	10	10	10	10
Total Req'd Head for Flow	10	20	30	35	40	45	50

TABLE 3 – GPM VS TOTAL REQUIRED HEAD
(COOLING TOWER APPLICATION)

The correct procedure for plotting a system curve for the circuit shown in figure 25 is illustrated in Figure 27.

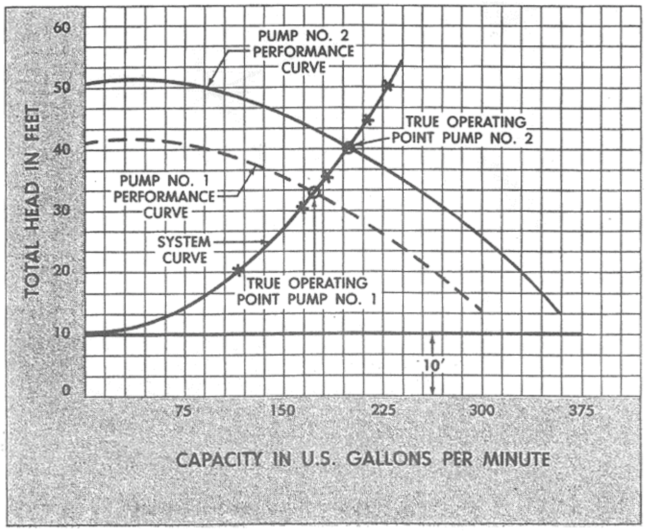


Figure 27 – System Curve for Open Circuit – True Operating Point

The same system curve construction method applies to a pump operating under a suction lift – and discharging to a tank at a higher elevation. (See Figure 28).

Figure 28 – Pump Operating Under Static Suction Lift

When the piping pressure drop curve is adjusted for the 10 ft. static head differential between the tanks; the point of intersection between the system curve and the pump curve will be the point of operation. The static suction lift of 5 ft. has no bearing on this analysis; it is only of importance to anti-cavitation operation (NPSH evaluation). The system curve for this circuit operating at 200 GPM and 30' Head is shown in Figure 29.

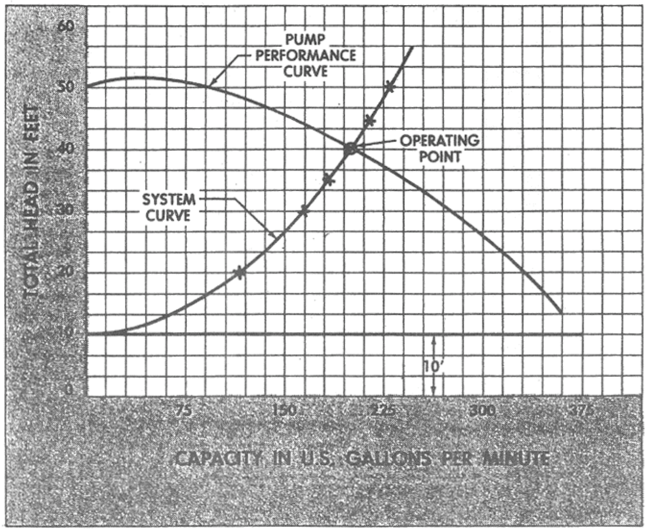


Figure 29 – System Curve for Static Suction Lift

When a pump is used as a booster, to increase water flow rates from an elevated tank at its suction to a tank at a lower elevation, a similar analysis will determine the actual pumping point. (See Figure 30).

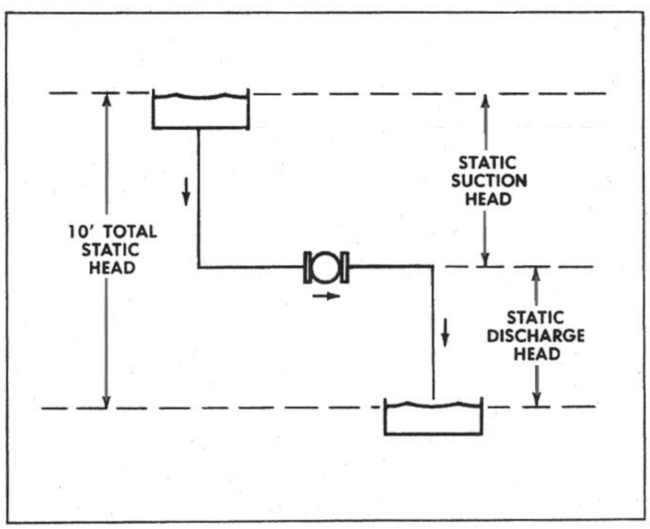


Figure 30 – Pump Used as Booster from an Elevated Tank

For the system shown in Figure 30 using the design points of 200 GPM at 30 foot piping and valve pressure drop the GPM versus total required head for the flow table would appear as shown in table 4.

GPM	0	115	165	185	Design 200	215	230
Pipe + Valve Pressure Drop	0	10	20	25	30	35	40
Minus 10′ Static Energy Head	−10	−10	−10	−10	−10	−10	−10
Total Req'd Head for Flow	−10	0	+10	+15	+20	+25	+30

TABLE 4 – GPM VS TOTAL REQUIRED HEAD
(USING PUMP AS BOOSTER)

In this case, an elevated static head at the pump suction provides energy head for water flow and the pump simply increases or "boosts" this head. Since the pump suction tank is at a 10 ft. higher elevation than the discharge tank; the pipe friction curve is dropped to a point registered at a minus 10 ft. on the pump curve. The composite curve then appears as shown in Figure 31.

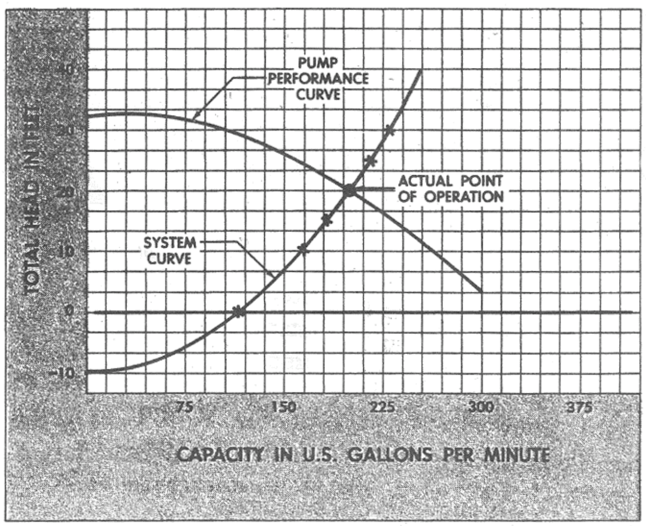


Figure 31 – Composite Curve for Elevated Tank Pump Booster Installation

It will be generally noted that in most cooling tower applications total pump head is usually composed of a high friction loss (piping, valves, condenser, etc.) plus a relatively low static discharge head. Many other open systems have high static head pumping requirements as compared with low piping friction loss. Open system pump selection by system curve methods becomes increasingly important as "open" static head energy requirements become a greater portion of the total pump head.

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Xylem Inc.
8200 N. Austin Avenue
Morton Grove, Illinois 60053
Phone: (847) 966-3700
Fax: (847) 965-8379
www.bellgossett.com