Peerless Pump Company

System Analysis for Pumping Equipment Selection

PEERLESS PUMP® LaBour Taber

Brochure B-4003

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1

Introduction

"Turn us on and we'll come running" is a phrase used by an investor-owned water utility in advertising their service. The faucet which we open to water the lawn or wash our hands is one end of a very large and complex piping system; at the other end is a pump. Many people are aware of a pump only when it does not function; others must be concerned about it from the time the piping system is conceived, through its design, construction, and operational life. By considering the pump early in the system design and applying energy evaluation procedures, long-term operating cost benefits can result.

This book has been prepared to guide readers at all levels of experience through a basic understanding of centrifugal pumps, pump characteristics, and system operation and control. These topics form the foundation for Total System Evaluation®.

Energy evaluation procedures are applicable to all systems, existing or proposed, large or small. In many systems the procedures will show that a reduction in operating power requirements can be achieved. Whether the reduction is large or small, you must determine its true significance.

We have all been challenged to use our available energy more efficiently. This book will serve as a working tool to assist you in meeting that challenge.

2

Centrifugal Pumps

Introduction

The selection of a centrifugal pump for an energy-efficient pumping system requires an understanding of the principles of mechanics and physics which can affect the pumping system and the pumped liquid. The efficiency of a centrifugal pump is also dependent upon the behavior of the liquids being pumped. The principles of centrifugal pump operation which govern head and flow must be clearly understood before pump performance can be accurately evaluated.

Fluids

The behavior of a fluid depends upon its state, liquid or gas. A liquid is a fluid with a free surface; its volume does not change significantly with a large pressure change. A gas is a fluid without a free surface; thus, a gas will completely fill a containment vessel of any size and shape.

Liquids and gases offer little resistance to changes in form. Typically, fluids such as water and air have no permanent shape and readily flow to take the shape of the containing enclosure when even a slight shear loading is imposed. Factors affecting behavior of fluids include:

- Viscosity
- Specific gravity
- Vapor pressure.

Viscosity is the resistance of a fluid to shear motion—its internal friction. The molecules of a liquid have an attraction for each other. They resist movement and repositioning relative to each other. This resistance to flow is expressed as the viscosity of the liquid. Dynamic viscosity can also be defined as the ratio of shearing stress to the rate of deformation. The viscosity of a liquid varies directly with temperature; therefore, viscosity is always stated at a specific temperature.

Liquid viscosity is very important in analyzing the movement of liquids through pumps, piping, and valves. A change in viscosity alters liquid handling characteristics in a system; more or less energy may then be required to perform the same amount of work. In a centrifugal pump an increase in viscosity reduces the pressure energy (head) produced while increasing the rate of energy input. In piping systems a liquid with a high viscosity has a high energy gradient against which a pump must work and more power is required than for pumping low-viscosity liquids.

Specific gravity is the ratio of the density of one substance to that of a reference substance at a specified temperature. Water at 4° C is used as the reference for solids and liquids. Air is generally used as the reference for gases. The specific gravity of a liquid affects the input energy requirements, or brake horsepower, of centrifugal pumps. Brake horsepower varies directly with the specific gravity of the liquid pumped. For example, water at 4° C has a specific gravity of 1.0. Table 1 includes some specific gravity values for water at selected temperatures.

Table 1

Water Temp.		Water Specific Gravity	Effect of Specific Gravity on Energy Input for Constant Flow	
°C	°F		KW	BHP
4	39.2	1.0	74.6	100
60	140	0.983	73.3	98.3
100	212	0.958	71.5	95.8
125	257	0.939	70.0	93.9
150	302	0.917	68.4	91.7

Specific gravity affects the liquid mass but not the head (usually expressed as feet of liquid) developed by a centrifugal pump as shown in Table 2.

Table 2

Specific Gravity	Effect of Specific Gravity on . . . Linear Head Feet	Linear Head Meters	Gauge Pressure PSI	Gauge Pressure Bar
0.75	246	75	79.9	5.51
1.00	246	75	106.5	7.34
1.20	246	75	127.8	8.81

The specific gravity also affects the energy required to move the liquid and, therefore, must be used in determining the pump's horsepower requirement.

Vapor pressure is the pressure at which a pure liquid can exist in equilibrium with its vapor at a specified temperature. Fluids at temperatures greater than their specified (critical) temperatures will exist as single-phase liquids (vapors) with no distinction between gas and liquid phases. At less than the critical temperature, two fluid phases can coexist; the denser fluid phase exists as a liquid and the less dense phase as a vapor. At a specific temperature the liquid phase is stable at pressures exceeding the vapor pressure and the gas phase is stable at pressures less than the vapor pressure.

For a fluid to exist in a liquid state its surface pressure must be equal to or greater than the vapor pressure at the prevailing temperature. For example, water has a vapor pressure of 0.1781 psia at 10°C and 14.69 psia at 100°. The vapor pressure of a volatile liquid (such as ether, alcohol, or propane) is considerably higher than that of water at the same temperature; consequently, much higher pressures must be applied to maintain volatile materials in their liquid states. The surface pressure of a liquid must be greater than its vapor pressure for satisfactory operation of a centrifugal pump.

Centrifugal pump performance

Centrifugal pump characteristics remain constant unless an outside influence causes a change in operating conditions. Three conditions can alter pump performance:

- Changes in impeller or casing geometry.
- Increased internal pumping losses caused by wear
- Variation of liquid properties

For example, if the impeller passages become impacted with debris, the head-flow relationship will be reduced. Similarly, performance will decline if mechanical wear increases the clearance between the rotating and stationary parts of the pump.

Except for specially designed pumps, most centrifugal pumps can handle liquids containing approximately 3 to 4 percent of gas (by volume) without an adverse effect on performance. An excess of gas will reduce the flow of liquid through the pump and, under certain conditions, flow will cease, setting up a condition which may damage the pump.

The function of a pump is to move liquids by imparting pressure energy (head) to the liquid. The ability of the pump to perform its function is based on the law which states that energy cannot be created or destroyed, but can only be converted in form. A pump converts mechanical energy into pressure energy. Part of the converted energy is required to overcome inertia and move the liquid; most of the remaining energy is stored in the liquid as elevated pressure which can be used to perform useful work outside the pump. A centrifugal pump is basically a velocity machine designed around its impeller. The interaction between the impeller and its casing produces the characteristics of head or pressure energy.

The developed head is a function of the difference in velocity between the impeller vane diameter at entrance and the impeller vane diameter at exit. The expression of theoretical head can be related to the law of a falling body:

$$H = \frac{V^2}{2g}$$

where: H = height or head, ft
V = velocity of moving body, fps
g = acceleration of gravity (32.2 ft/sec²)

When the height of fall is known (for example, H = 100 ft) the terminal velocity can be determined (in this case V = 80.3 fps). Conversely, if the direction of motion is reversed, a liquid exiting through an impeller vane tip at a velocity of 80.3 fps reaches a velocity of zero fps at 100 ft height above the impeller tip. Fig. 1.

When the developed head required is known, the theoretical impeller diameter for any pump at any rotational speed can be determined from the equation for peripheral velocity of a round rotating body:

$$D = \frac{(229.2)V}{N}$$

where: D = unknown impeller diameter, in.
V = velocity (derived from $V = \sqrt{2gH}$), fps
N = rotational speed of the pump, rpm

If pump head is 100 ft and rotational speed is 1750 rpm, the formula indicates a 10.52 in. theoretical diameter impeller is required.

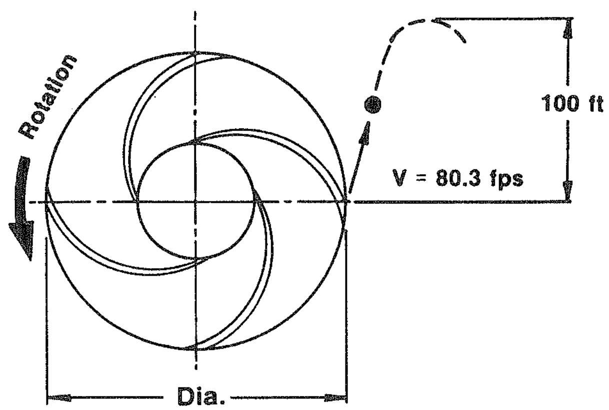


Figure 1

The function of a centrifugal pump is to transport liquids by transferring and converting mechanical energy (foot-pounds of torque) from a rotating impeller into pressure energy (head). By the transfer of this energy to a liquid, the liquid can perform work, move through pipes and fittings, rise to a higher elevation or increase the pressure level. Because a centrifugal pump is a velocity machine, the amount of mechanical energy per unit of fluid weight transferred to the liquid depends upon the peripheral velocity of the impeller, regardless of fluid density. This energy per unit of weight is defined as pump head and is expressed in feet of liquid. If the effect of liquid viscosity is ignored, the head developed by a given pump impeller at a given speed and flow rate remains constant for all liquids.

Head is the vertical height of a column of liquid. The pressure this liquid column exerts on the base surface depends upon the specific gravity of the liquid. A 10 ft column of liquid with a specific gravity of 0.5 exerts only 2.16 psi; a 10 ft column of liquid with a specific gravity of 1.0 exerts 4.33 psi.

The formula for converting feet of head to pressure is:

$$H = \frac{P \times 2.31}{SG}$$

where: P = pressure, psi
SG = specific gravity

The proper selection of a centrifugal pump requires that pressure be converted to feet of head. If not, a pump that is incapable of imparting the required energy to a liquid may be installed before the inadequacy is discovered. The heads required to produce a pressure of 100 psi for three liquids of different specific gravities are given in Table 3.

Table 3

Head equivalent of 100 psi for liquids of different specific gravity	
Specific Gravity	**Head, ft of liquid**
0.75	308
1.0	231
1.2	193

From this analysis, a single-stage pump selected for pumping a liquid with a specific gravity of 1.2 will have the lowest peripheral velocity at the impeller tip, while the pump for a liquid with a 0.75 specific gravity will have the highest peripheral velocity.

Flow (the volume of liquid a centrifugal pump can move per unit of time) is a function of the peripheral velocity of the impeller and the cross-sectional areas in the impeller and its casing; the larger the passage area, the greater the flow rate.

Consequently, the physical size of the pump increases with higher flow requirements for a given operating speed. The liquid flow rate is directly proportional to the area of the pump passages and can be expressed as:

$$Q = \frac{V \times A}{0.321}$$

where: Q = flow, gpm
V = velocity, fps
A = area, sq. in.

For example, a centrifugal pump having a 15 fps liquid velocity at the discharge flange can pump approximately 330 gpm through a 3 in. diameter opening. If the flow requirements are increased to approximately 1300 gpm and the velocity remains at 15 fps, a 6 in. diameter opening will be required. Liquid flow is also directly proportional to the rotational speed which produces the velocity. The four preceding equations establish two relationships:

- Head is directly proportional to the square of the liquid velocity
- Flow is directly proportional to the peripheral velocity of the impeller.

Cavitation in a centrifugal pump can be a serious problem. Liquid pressure is reduced as the liquid flows from the inlet of the pump to the entrance to the impeller vanes. If this pressure drop reduces the absolute pressure on the liquid to a value equal to or less than its vapor pressure, the liquid will change to a gas and form vapor bubbles. The vapor bubbles will collapse when the fluid enters the high-pressure zones of the impeller passages.

This collapse is called cavitation and results in a concentrated transfer of energy which creates local forces. These high-energy forces can destroy metal surfaces; very brittle materials are subject to the greatest damage. In addition to causing severe mechanical damage, cavitation also causes a loss of head and reduces pump efficiency. Cavitation will also produce noise.

If cavitation is to be prevented, a centrifugal pump must be provided with liquid under an absolute pressure which exceeds the combined vapor pressure and friction loss of the liquid between the inlet of the pump and the entrance to the impeller vanes.

NPSH

NPSHA (net positive suction head available) is the absolute pressure of the liquid at the inlet of the pump. NPSHA, a function of the elevation, the temperature, and the pressure of the liquid, is expressed in units of absolute pressure (psia). Any variation of these three liquid characteristics will change the NPSHA. An accurate determination of NPSHA is critical for any centrifugal pump application.

The net positive suction head required (NPSHR) by a specific centrifugal pump remains unchanged for a given head, flow, rotational speed, and impeller diameter, but changes with wear and liquids.

Specific speed is a correlation of pump flow, head, and speed at optimum efficiency. It classifies pump impellers with respect to their geometric similarity. Specific speed is usually expressed as:

$$N_s = \frac{N\sqrt{Q}}{H^{3/4}}$$

where: N_s = pump specific speed
Q = flow at optimum efficiency, gpm

The specific speed of an impeller is defined as the revolutions per minute at which a geometrically similar impeller would run if it were of a size that would discharge 1 gpm against a head of 1 ft. Specific speed is indicative of the impeller's shape and characteristics.

Centrifugal pumps are divided into three classes:

- Radial flow
- Mixed flow
- Axial flow.

There is a continuous change from the radial flow impeller (which develops head principally by the action of centrifugal force) to the axial flow impeller (which develops most of its head by the propelling or lifting action of the vanes on the liquid). Typically, centrifugal pumps can also be categorized by physical characteristics relating to the specific speed range of the design, Fig. 2 through 6. Once the values for head and capacity become established for a given application, the pump's specific speed range can be determined and specified to assure the selection of a pump with optimal operating efficiency.

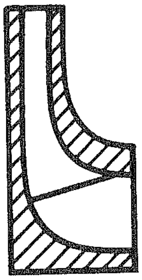


Fig. 2. Single suction, horizontal or vertical centrifugal pumps with narrow port impellers have low capacities and deliver high heads. Specific speed range is 500 to 1000.

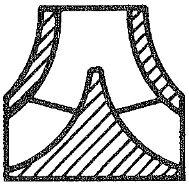


Fig. 3. Single or multistage, double (illustrated) and single suction, volute and diffuser design centrifugal pumps deliver medium capacity and medium heads. Specific speed range is 1000 to 2000.

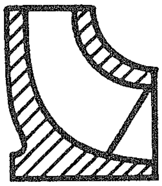


Fig. 4 Single or multistage, and single (illustrated) and double suction, Francis-type impellers, operating in volute or diffuser type casings produce medium to high capacity at medium to low speeds. Specific speed range is 2000 to 4000.

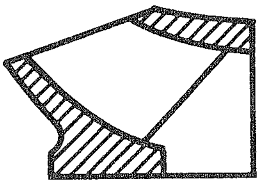


Fig. 5 Single or multistage single suction centrifugal pumps, in volute or diffuser-type casings with mixed flow impellers, deliver high capacity at low head. Specific speed range is 4000 to 6000.

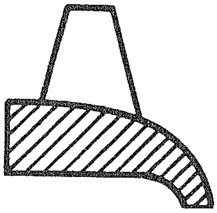


Fig. 6 Single or multistage pumps with mixed flow and propeller-type impellers have very high capacities and deliver very low heads. Specific speed range is 6000 to 10,000.

3

Pump Characteristics

Differing pump hydraulic characteristics will result in one pump being better suited for a given application than another. This section examines head-flow curves, pump performance curves, variations in curve shape as a function of specific speed and pump affinity laws.

Head-flow curve

The head-flow curve of an ideal pump with ideal (frictionless) fluid is a straight line whose slope from zero flow to maximum flow varies with the impeller exit vane angle. For example, a given impeller and casing combination with an impeller vane exit angle of 25° will have a greater maximum flow than a similar impeller with a 15° vane exit angle (Figure 7). However, the actual head-flow characteristic of a centrifugal pump is not an ideal straight line. Its shape is altered by friction, leakage, and shock losses which occur in impeller and casing passages.

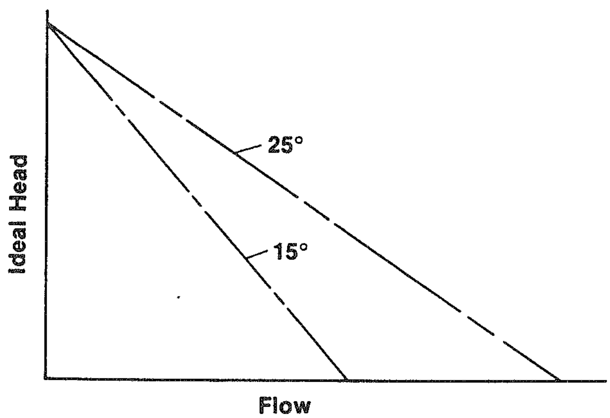


Figure 7

Friction losses in a centrifugal pump are proportional to the surface roughness and the wetted areas of the impeller and casing. Leakage losses result from the flow of liquid between the clearance of rotating and stationary parts, such as impeller to case wear ring clearances. Shock losses occur as the liquid enters the impeller entrance vanes and as the liquid flows from the impeller into the casing. These internal losses characteristically reduce pump performance from the ideal to the actual total head-flow curve shown in Figure 8. The flow at which the sum of all these losses is the least determines the point of maximum efficiency.

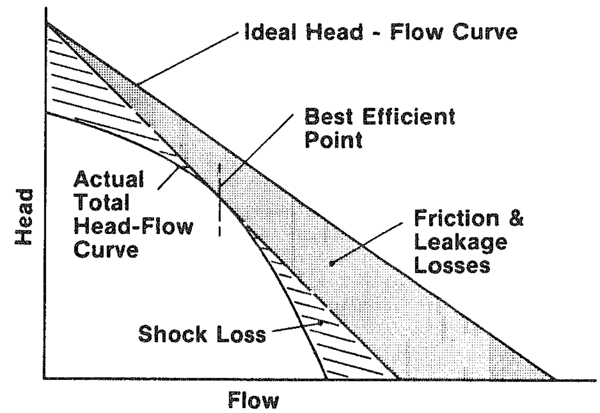


Figure 8

Mechanical losses in bearings, packings and mechanical seals further reduce pump efficiency. Although mechanical losses may be calculated, the results are generally not accurate; actual pump performance can only be determined by testing.

A centrifugal pump is designed around its impeller. The function of the casing is to collect the liquid leaving the impeller and convert its kinetic energy into pressure energy. A case is designed in conjunction with an impeller to achieve the most efficient match for conversion of liquid velocity to pressure energy. The interaction between the impeller and its casing for a given pump determines the pump's unique performance characteristics.

Performance curves

Pump performance curves depict the total head developed by the pump, the brake horsepower required to drive it, the derived efficiency, and the net positive suction head required over a range of flows at a constant speed. Pump performance can be shown as a single line curve depicting one impeller diameter (Figure 9) or as multiple curves for the performance of several impeller diameters in one casing (Figure 10).

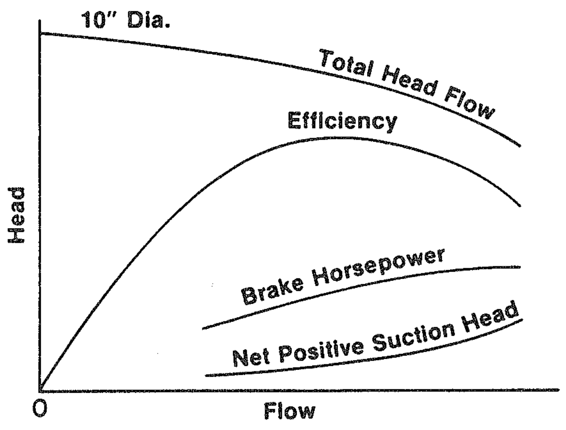


Figure 9

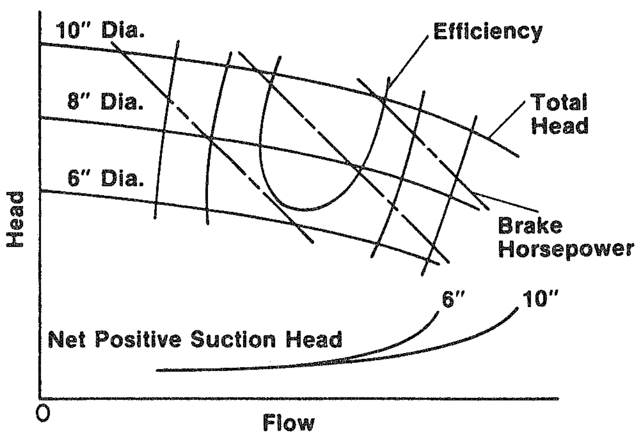


Figure 10

Performance related to specific speed

The performance characteristics of pumps are classified by discharge specific speed and have the approximate curve shapes shown in Figure 11 on page 9.

Curve shape definition

According to shape, centrifugal pump head-capacity and horsepower curves are classified as follows:

Drooping head characteristics are those in which the head at zero flow is less than the head developed at some greater flow (Figure 12).

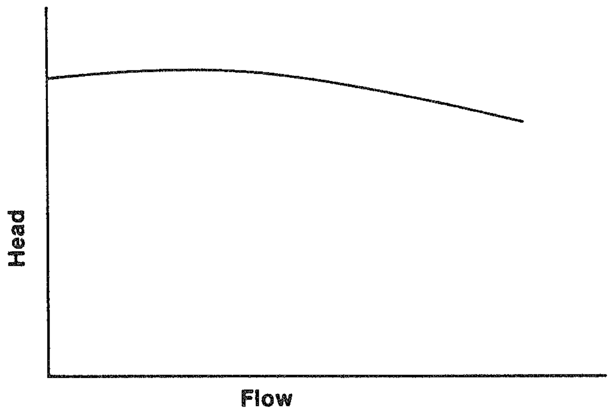


Figure 12

Continually rising head characteristics are those in which the head rises continuously as the flow is reduced to zero (Figure 13).

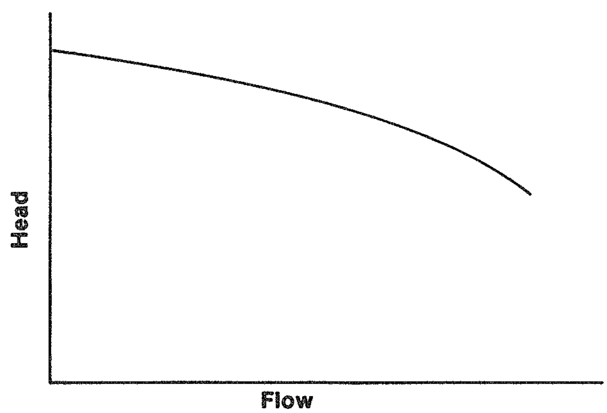
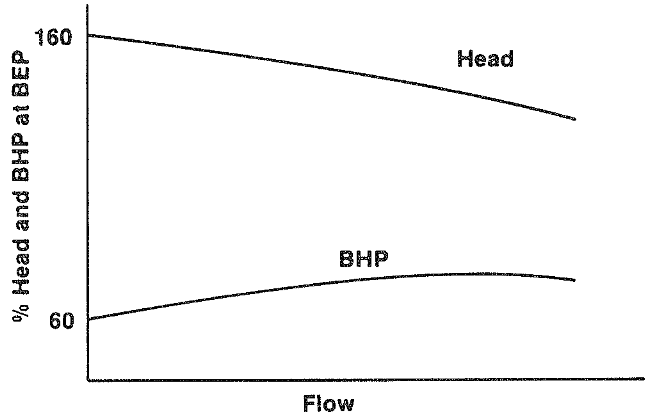


Figure 13

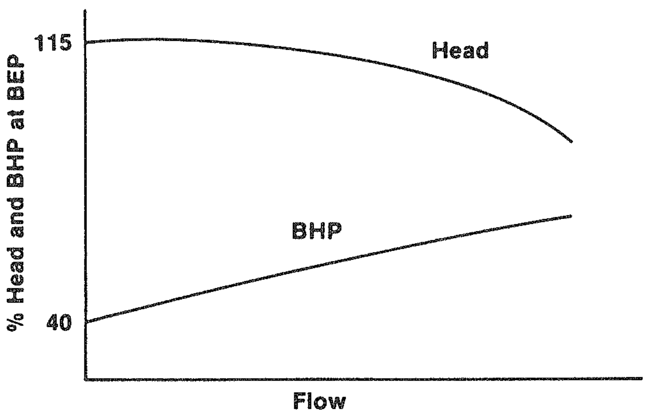
BEP is the reference to the flow point at which the best pump efficiency occurs.

All specific speed values are in English units.

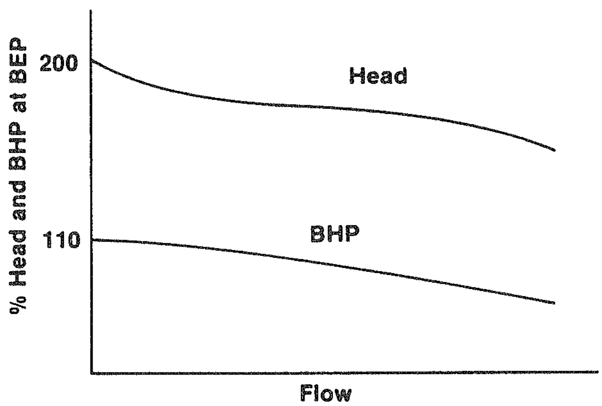
Refer to Peerless Pump Technical Information Bulletin No. 14 for specific speed calculation details.



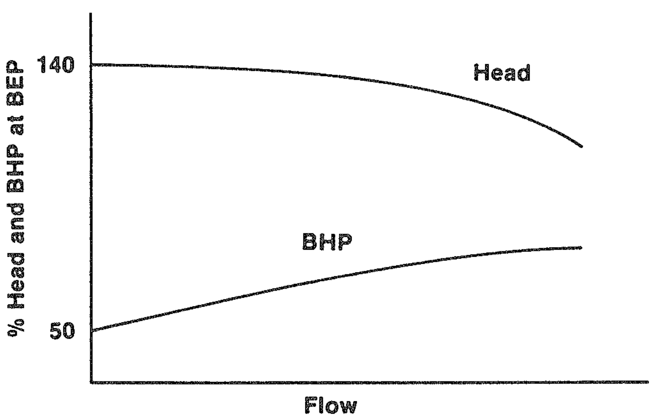
N_s = 2000 - 4000



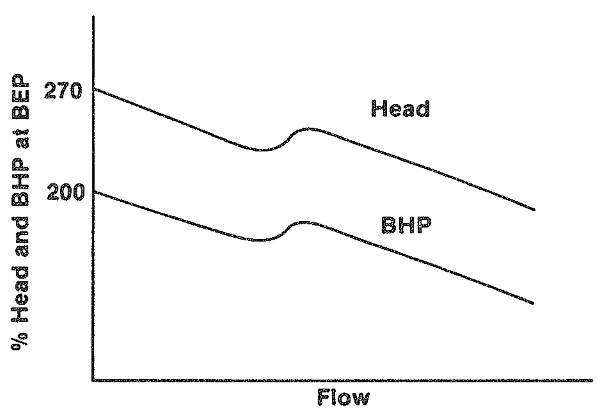
N_s = 500 - 1000



N_s = 4000 - 6000



N_s = 1000 - 2000



N_s = 6000 - 10000

Figure 11

Steep head characteristics are those in which the head rises steeply and continuously as the flow is reduced. Curve steepness is only a relative term because there is no defined value of curve slope for comparison (Figure 14).

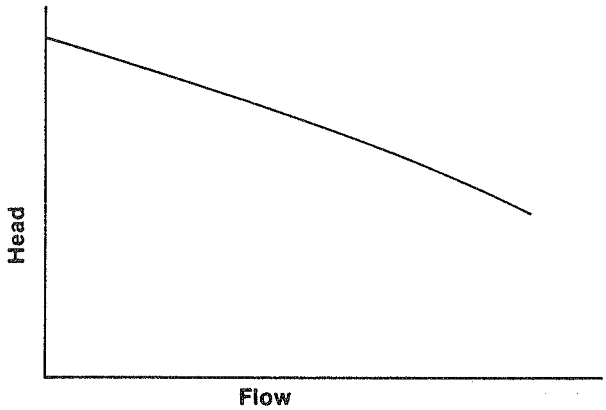


Figure 14

Flat head characteristics are those in which the head rises only slightly as the flow is reduced. As with steepness, the magnitude of flatness is a relative term (Figure 15).

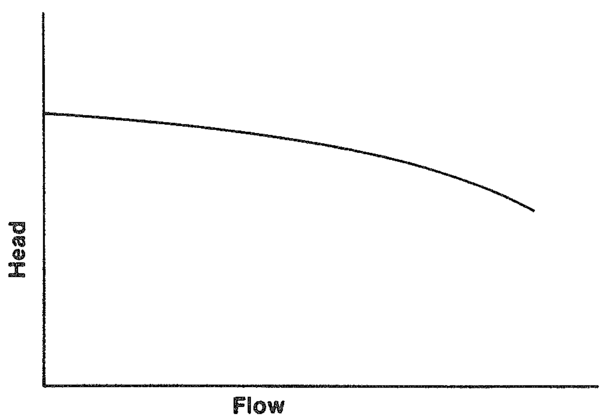


Figure 15

Discontinuous head characteristics are those in which a given head is developed by the pump at more than one flow rate. Many pumps in the high specific speed ranges have this characteristic (Figure 16)

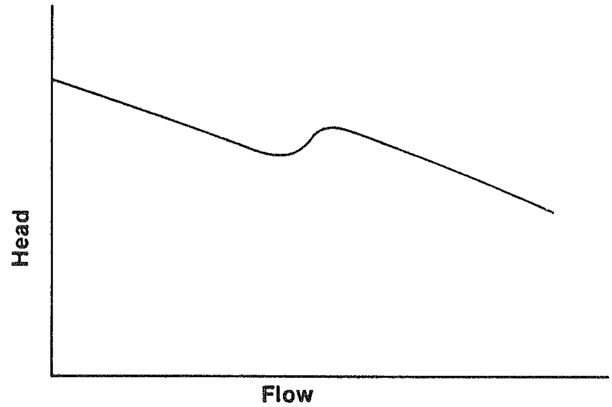


Figure 16

Continually increasing horsepower characteristics are those of low specific speed pumps where the horsepower increases at flows greater than the best efficiency point (BEP) and decreases at flows to the left of BEP, (Figure 17).

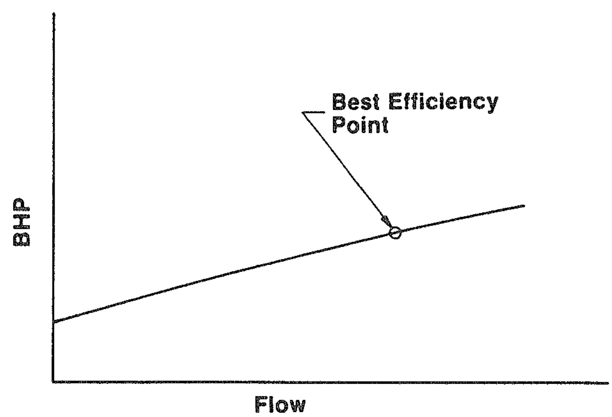


Figure 17

Peaking horsepower characteristics are those of medium specific speed pumps where maximum horsepower occurs in the BEP range and decreases at all other values of flow (Figure 18).

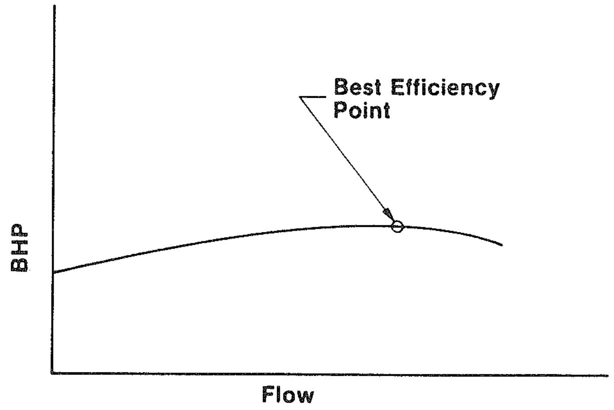


Figure 18

Discontinuous horsepower characteristics are those of high specific speed pumps where horsepower increases at flows less than BEP and decreases with flow to the right of BEP. The discontinuity usually occurs at flows less than BEP (Figure 19).

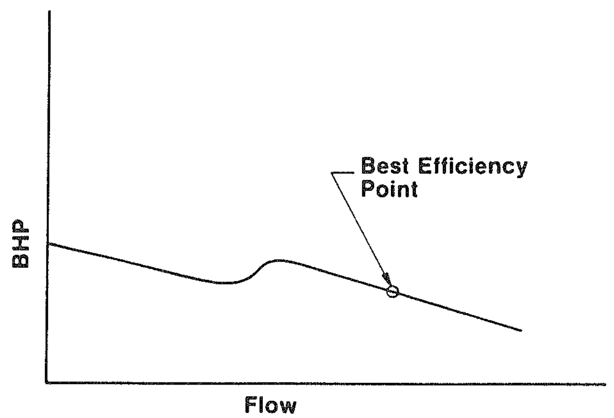


Figure 19

Pump affinity laws

The pump affinity laws state that flow is proportional to impeller peripheral velocity and head is proportional to the square of the peripheral velocity. Specifically:

Flow (Q) will vary directly with the ratio of change of speed (N), or impeller diameter (D).

$$\frac{Q_2}{Q_1} = \frac{N_2}{N_1} \text{ or } \frac{Q_2}{Q_1} = \frac{D_2}{D_1}$$

Head (H) will vary as the square of the ratio of speeds or impeller diameters.

$$\frac{H_2}{H_1} = \left(\frac{N_2}{N_1}\right)^2 \text{ or } \frac{H_2}{H_1} = \left(\frac{D_2}{D_1}\right)^2$$

The pump required horsepower (BHP) will vary as the cube of the ratio of speeds or impeller diameters.

$$\frac{BHP_2}{BHP_1} = \left(\frac{N_2}{N_1}\right)^3 \text{ or } \frac{BHP_2}{BHP_1} = \left(\frac{D_2}{D_1}\right)^3$$

A basic premise throughout these analyses is that pump efficiency remains constant for speed or impeller diameter changes. For example, the data in Table 4 show a projection of pump performance from 1760 RPM to 1450 RPM. The subscript 2 is used for the calculated performance and subscript 1 for the values selected from the 1760 RPM curve.

$$Q_2 = \left(\frac{N_2}{N_1}\right) Q_1$$

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

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

Table 4

Tabulated Performance Data from 1760 RPM Curve				Calculated Performance Data for 1450 RPM Operation			
Q_1 gpm	H_1 feet	Eff_1	BHP_1	Q_2 gpm	H_2 feet	Eff_2	BHP_2
1000	184	.61	76.2	824	125	.61	42.6
1500	175	.76	87.2	1236	119	.76	48.9
2000	166	.84	99.8	1648	113	.84	56.0
2500	151	.86	110.8	2060	103	.86	62.3
3000	128	.82	118.3	2478	87	.82	66.4
3250	110	.73	123.7	2678	75	.73	69.5

A similar table can be developed for changing impeller diameter. In projecting pump performance, care should be taken to project from the speed or diameter which is closest to the calculated value. Changes in both speed and diameter may alter the efficiencies taken from the originating pump performance curve.

Pump performance curves are developed as a means of conveniently viewing the complete characteristics of a given pump and selecting a pump which will successfully meet system requirements. The pump is a source of pressure energy and, within its limits of design and operating speed, has the capability of providing the liquid in a system with the potential to do work.

Pressure energy capability is dependent upon the velocity the impeller imparts to a liquid. For a given pump, this velocity can be changed by varying the impeller diameter (Figure 20) or by changing the operating speed. Therefore, by controlling impeller angular velocity, a centrifugal pump can be used as a variable energy source to satisfy pumping systems having variable pressure energy requirements.

When system requirements dictate that the centrifugal pump *must* have particular curve shape characteristics, the approximate discharge specific speed index range may be used to define the general requirement for the specified pump.

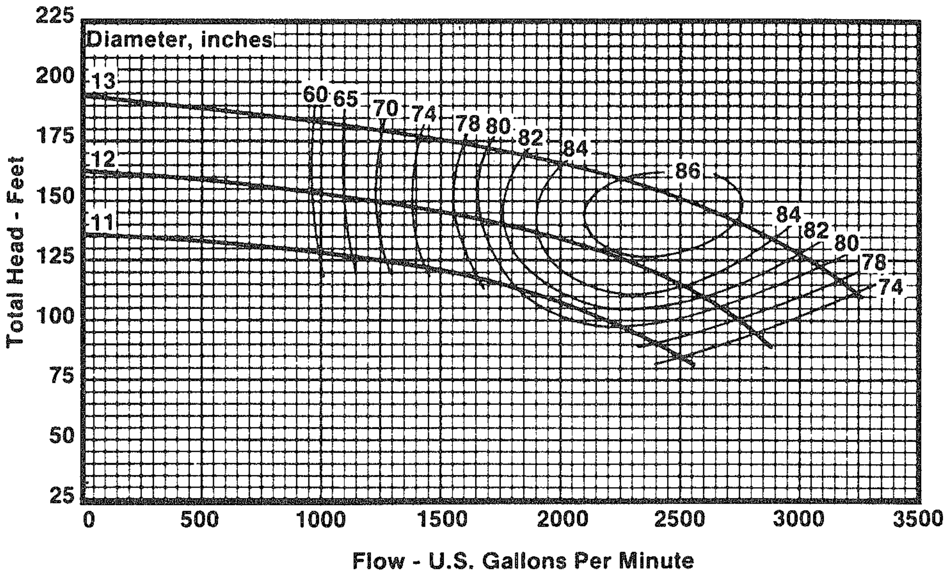


Figure 20

4

System Characteristics

Introduction

A pumping system is the arrangement of pipe, fittings and equipment through which liquids flow. Only a few pump applications are such that a pumping unit can operate alone, one example being a pump lifting water from a well into an irrigation ditch. The ditch acts as a conduit distributing the water by gravity flow. Pumps are required when liquids need to be transported from lower to higher elevations, moved over long distances, or distributed in grids, circulated in loop or pressurized systems to perform work.

Classification of systems

Mechanically, systems can be classified by path of liquid flow as follows:

Nonreturn systems where all the liquid is discharged from the system.

Return systems
a) Where none of the liquid is discharged from the system.
b) Where some of the liquid is discharged from the system.

Functions of systems can be classified by the characteristics of work performed as follows:

Thermal exchange where some form of thermal exchange is conducted for the purpose of satisfying a design condition.

Removal/delivery where the system is designed to remove the liquid from or deliver it to some point to satisfy a specific design service.

The following lists illustrate typical pump application names classified by system function:

Thermal exchange applications
Chilled Water
Cold Well
Condenser Water
Cooling Tower
Heat Recovery
Hot Well
Mill Roll Cooling
Plant Circulating Water
Plant Cooling Water
Spray Pond
Strip Mill Quench

Removal/delivery applications
Ash Sluice
Boiler Feed
Condensate
Domestic Water
Effluent
Filter Backwash
Flood Irrigation
High Service
Low Service
Municipal Booster
Raw Water
River Intake
Sewage Ejector
Sprinkler Irrigation
Storm Water

Piping system design

Some of the items which effect the cost and operation of piping systems include noise, water hammer, pipe aging and friction loss.

Flow noise

Flow noise or "hiss" contains all sound frequencies at an evenly distributed sound level, "white noise". This noise is generated by turbulent flow through system pipes and it increases with velocity; or it is generated by cavitation in the system. Flow noise occurs in pumps, valves, elbows, tees or wherever flow changes direction or velocity.

Noise conduction

Conduction of noise proceeds through the liquid as well as through the pipe material and is usually controlled by flexible fittings, sound insulating pipe supports and pipe insulation.

Pipe resonance

Pipe resonance can result in objectional sound in a piping system. Resonance is a single tone sound magnified and emitted by a length of piping. It occurs when the natural frequency of a length of piping is matched by the frequency of some regular energy source such as the vanes in an impeller passing the volute tongue (cutwater) as shown in Figure 21. An illustration of this type of resonance is shown in Figure 22. Other regular energy sources such as the rotating elements in a bearing or the power supply frequency can also cause pipe resonance. Pipe resonance can be corrected by increasing or decreasing the resonating length or weight of pipe under the direction of an acoustical consultant.

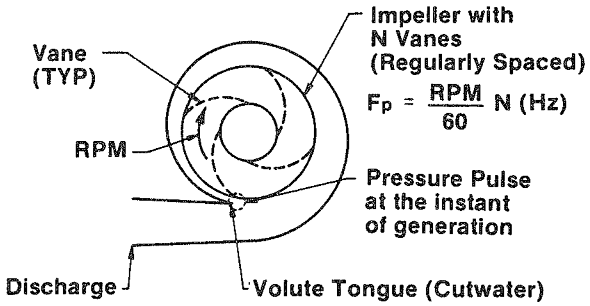


Figure 21

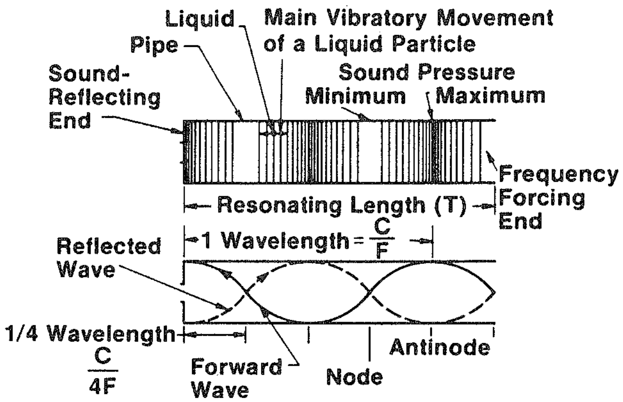


Figure 22

Water hammer

Water hammer is the result of a strong pressure wave in a liquid caused by an abrupt change in flow rate. As an illustration, for the maximum possible instantaneous head increase above the normal head due to a water hammer pressure wave, the following expression is applicable.

$H_{wh} = \frac{CV}{g}$ (Joukowsky's law) where:

C = velocity of sound in the liquid (ft/sec) e.g., C for 15.6° C water is 4820 ft/sec.

V = normal velocity in the conduit before closing valve (ft/sec).

g = acceleration due to gravity (32.2 ft/sec^2).

Assuming V = 15 ft/sec, H_{wh} for 15.6° water is 2245 ft. or a pressure surge of 972 psi. This is the maximum possible pressure rise by instantaneous closing of the valve and may be more than the system can withstand.

Pipe deterioration

Some deterioration of the inside surfaces of a piping system normally occurs with age. This deterioration may result from corrosion, deposits, or a combination of both which impede flow by increasing the relative roughness ratio e/D

where: e = height of protrusion on a side in feet or inches

D = inside diameter of pipe in feet or inches

Table 5 contains some representative values for the height of the internal surface protrusions:

Table 5

Material (New)	**e (feet)**	**e (inches)**
Copper or brass tubing	5×10^{-6}	4.2×10^{-7}
Steel pipe	1.5×10^{-4}	1.3×10^{-5}
Galvanized steel	5×10^{-4}	4.2×10^{-5}
Concrete	4×10^{-3}	3.3×10^{-4}

For old material, the above values may be increased by 100 or more times due to corrosion or deposits. Past experience is generally the most accurate guide to determine the extent of pipe deterioration caused by aging.

Pipe friction

Pipe friction is resistance to flow and results in a loss of head which is expressed as friction head (H_f) in feet of liquid. One method of calculating friction head is the Darcy-Weisbach formula:

$H_f = \dfrac{f\ L\ H_v}{D}$ where:

L = length of pipe in feet

D = inside diameter of the pipe in feet

$H_v = \dfrac{V^2}{2g}$ velocity head in feet where:

V = flow velocity (ft/sec)

g = acceleration of gravity (32.2 ft/sec²)

f = friction factor

A relationship exists between Reynolds number, friction factor and relative roughness as shown in Figure 23. Reynolds number is based on pipe diameter and the fluid properties of viscosity, density, and velocity. For Reynolds numbers smaller than 2000, the pipe roughness factor e/D does not apply and the friction factor can be calculated, $f = \dfrac{64}{N_r}$.

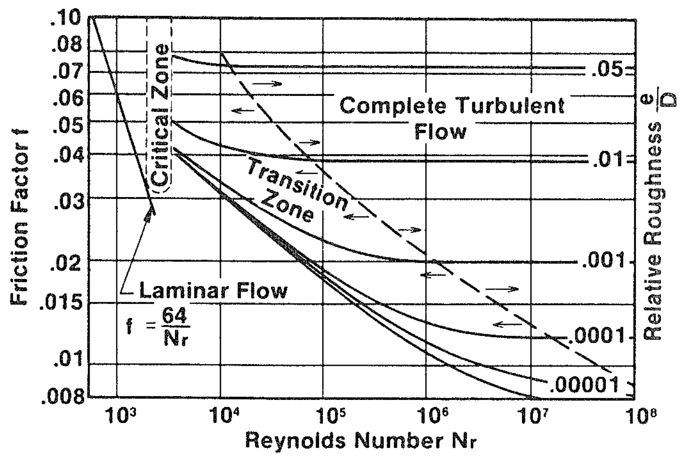


Figure 23

Reynolds numbers in the range of 2000-4000 are in a critical zone and the values of "f" are indeterminate. For Reynolds numbers above 4000, in both the transition and complete turbulence zones, Reynolds numbers and relative roughness must be used to determine the friction factor.

Reynolds Number $N_r = \dfrac{VD}{k}$ where:

V = velocity ft/sec

D = pipe inside diameter in feet

k = Kinematic viscosity in ft²/sec where:

$k = \dfrac{a}{w/g}$

a = absolute viscosity lb-sec/ft²

w = specific weight, lbs/ft³

g = 32.2 ft/sec²

Viscosity conversion methods. If absolute viscosity is expressed in centipoise (cp), use the following conversion:

$2.089 \times 10^{-5} \times cp$ = lb-sec/ft² which in turn can be converted into kinematic viscosity (ft²/sec) by dividing by the mass density (lb-sec²/ft⁴).

If kinematic viscosity is expressed in centistokes (cs), use the following conversion:

$1.076 \times 10^{-5} \times cs = ft^2/sec.$

If viscosity is expressed in Seconds Saybolt Universal (SSU), there is no distinction between the absolute and the kinematic designations. Convert SSU to kinematic viscosity as follows:

For SSU larger than 100:

$$1.076 \times 10^{-3} \times (0.0022\ SSU - \frac{1.35}{SSU}) = ft^2/sec.$$

For SSU equal to 100 or less:

$$1.076 \times 10^{-3} \times (0.00226\ SSU - \frac{1.95}{SSU}) = ft^2/sec.$$

Example 1:
Given: 40° C water with kinematic viscosity of 1.58 centistokes. Convert viscosity into ft^2/sec units.
Solution:
$1.076 \times 10^{-5} \times 1.58\ cs = 1.70 \times 10^{-5}\ ft^2/sec.$

Example 2:
Given: Ethylene glycol at 21.1° C with a viscosity of 88.4 SSU. Convert viscosity into ft^2/sec units.
Solution:

$$1.076 \times 10^{-3} \times (0.00226 \times 88.4 - \frac{1.95}{88.4}) =$$

$1.912 \times 10^{-4}\ ft^2/sec.$

Example 3:
Given: Ethyl alcohol at 20° C with specific weight of 49.4 lb/ft^3 and absolute viscosity of 1.2 centipoises (cp). Convert viscosity into ft^2/sec units.
Solution:
$2.089 \times 10^{-5} \times 1.2 = 2.507 \times 10^{-5}\ lb\text{-}sec/ft^2$

$$\text{Mass Density} = \frac{49.4\ lb/ft^3}{32.2\ ft/sec^2} = 1.5342\ \frac{lb\text{-}sec^2}{ft^4}$$

$$\frac{2.507 \times 10\ lb\text{-}sec/ft^2}{1.5342\ lb\text{-}sec^2/ft^4} = 1.63 \times 10^{-5}\ ft^2/sec$$

Friction formula variations. Through many years, simplified variations of the Darcy-Weisbach formula have been devised for limited areas of application. The accuracy of such variations is supported by successful use and allows the user to bypass the diagram interpretation and calculation necessary with the basic formula. The use of specific formula variations outside their known areas of successful application may lead to inaccurate projections of friction head.

This simplified derivation of the Williams-Hazen formula can be used to calculate friction head for liquids having a kinematic viscosity of 1.1 centistokes. (Water at 60° F has a viscosity of 1.13 centistokes.)

$$\text{Friction Head, } H_f = 10.45 \left(\frac{GPM}{C}\right)^{1.85} \left(\frac{L}{D^{4.87}}\right)$$

where: GPM = Gallons per minute
C = Pipe roughness factor ranging from 60 to 160.
D = inside diameter of pipe in inches
L = length of pipe in feet

The Williams-Hazen formula is generally used for cast iron pipes of 3 inch and larger diameter. As pipes deteriorate, the roughness factor "C" decreases. This decrease in C depends on the pipe material, pipe linings, pipe age, and the characteristics of the liquid.

For liquids with viscosities other than 1.1 centistokes, methods more accurate than the Williams-Hazen formula should be used in determining friction head. For example, friction head computed with the Williams-Hazen formula will increase by 20% at a viscosity of 1.8 centistokes and decrease by 20% at a viscosity of .29 centistokes. In addition, friction loss will occur in fittings such as elbows, tees, and valves. Many handbooks and manuals provide means for determining these losses. Manufacturers' data must be used to determine friction loss for mechanical equipment installed in the system.

System head curves

A system head curve is the graphical representation of the head required at all flows to satisfy the system function. Regardless of the mechanical configuration, function, or means of control, the system head curve or system head band is used to define total head versus flow for any piping system. System head curve development is a combination of calculations and the designer's "best feel" for the variable conditions.

System head components

The three components which make up total system head are static head, design working head, and friction head. These components are defined as follows:

Static head is the vertical difference in height between the point of entry to the system and the highest point of discharge.

Design Working Head is that head which must be available in the system at a specified location to satisfy design requirements.

Friction head is the head required by the system to overcome the resistance to flow in pipes, valves, fittings and mechanical equipment.

Total system head at a specified flow rate is the sum of static head, design working head, and friction head. The values of static head and design working head may be zero. Figure 24 illustrates a total system head curve made up of the friction head shown in Figure 25 and the design working head and static head shown in Figure 26.

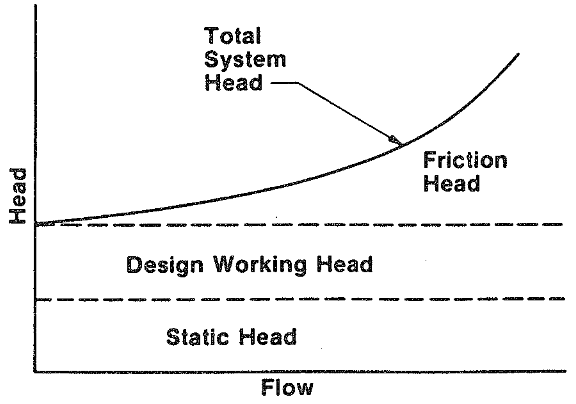


Figure 24

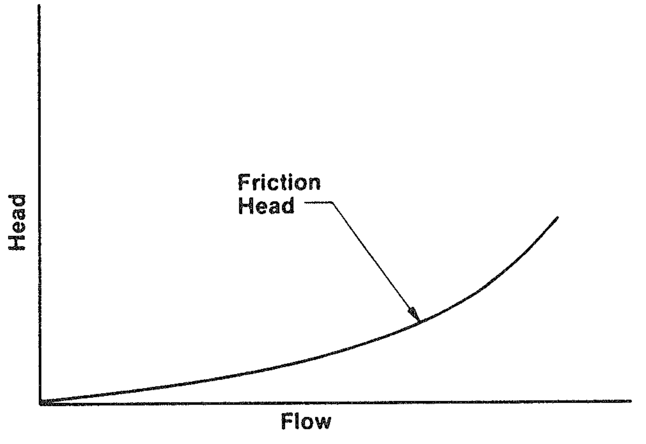


Figure 25

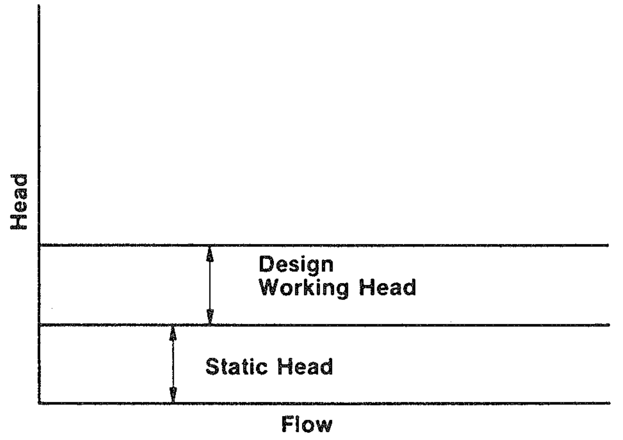


Figure 26

Total system head variables

For most systems, the total system head requirements are best illustrated by a band formed by two total system head curves. This band of system requirements is a result of variable factors which affect total system head calculations. The following list of variables should be considered in projecting system requirements.

Static head will vary as a result of change in elevation of highest point of discharge of the system.

Design working head is usually treated as a constant component of total system head.

Friction head at any specified flow will vary as a result of:

a) Method of calculation or source of tabulated data.

b) Change in viscosity resulting from a change in liquid temperature.
c) Deterioration of the piping system.
d) Load distribution.
e) Systems differences between design and "as-built".
f) Manufacturing tolerances of mechanical equipment.
g) Accumulation of solids in the system.

Figure 27 illustrates a band of system head requirements based on variations in friction head only; Figure 28 illustrates variation in friction head and static head.

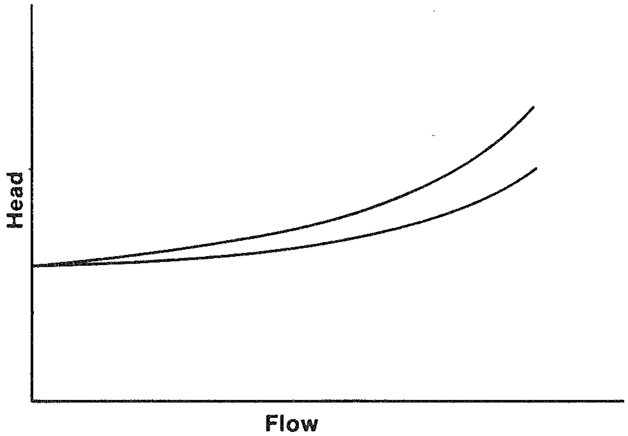


Figure 27

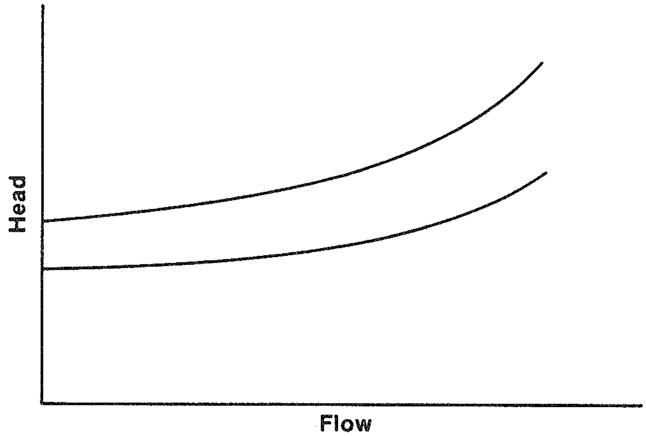


Figure 28

Conclusion

Most systems are not completely defined by a single line system head curve. Those variables which are applicable determine the band of maximum and minimum system head requirements. The band between maximum and minimum head requirements at any given flow will be significant in some systems and not in others. For all systems, interaction of total head available and total head required must be evaluated to achieve satisfactory system operation.

5

System Operation and Control

Introduction

For a system to operate at design flow, total head available to the system must be equal to the total system head required. For proper evaluation of system operation at flow rates other than design, a comparison of total available head and system required head must be made based on the means selected to control system flow. The reference point used to determine total available head must be the same as that used to determine total system head required. The point of entry to the system, which is also the pump discharge connection, is used in this text as the reference point.

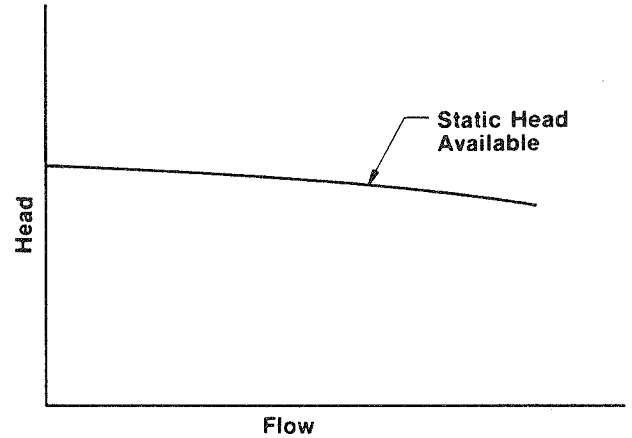


Figure 29

Total available head

Total available head at the point of entry to the system is the sum of static head and pump head.

Static head available is the vertical difference in height between the point of entry to the system and the liquid supply, minus any friction head in the supply pipe and fittings. Static head may be relatively constant as illustrated in Figure 29. The decrease in static head with increasing flow is the result of friction head in the supply pipe. In some applications static head will vary, and is normally illustrated by a band of maximum and minimum values as shown in Figure 30.

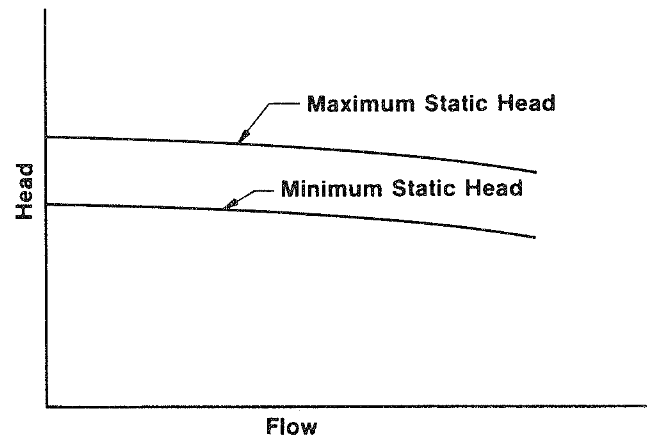


Figure 30

When the liquid supply is below the point of entry to the system as shown in Figure 31, static head available is a negative value.

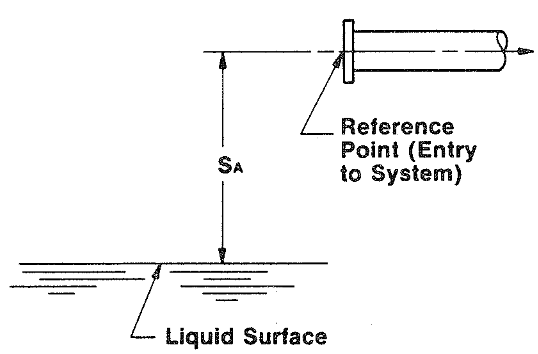


Figure 31

The difference in elevation between the fluid supply and the point of entry to the system must be developed by the pump. Therefore, pump head must include static head and friction head in the supply pipe and fittings.

In systems where the liquid supply is from a pressurized main, the pressure must be expressed in feet of liquid. The equivalent height of the liquid above the system entry point is static head available. In Figure 32, with a 100 psig pressure at point A and a liquid with a specific gravity of 1.0, the static head available will be 231 feet minus elevation height X. In return piping systems where none of the working fluid is discharged from the system, static head available is zero.

Pump head is the total head developed by a pump. Its value at any given flow rate must be obtained from the pump manufacturer's performance curve. Figure 33 illustrates a typical centrifugal pump head-flow curve.

Total available head curve

A curve of total available head versus flow is developed by adding static head and pump head at several values of flow. An example is shown in Figure 34 based on static head from Figure 29 and pump head from Figure 33. The total available head curve is slightly steeper than the pump head curve due to the slope of the static head curve.

To evaluate the operation of a system, the total system head curve or curves and the total available head curve are plotted on a common graph. The two curves will intersect at maximum flow for that system. As shown in Figure 35, system flow greater than 100% cannot occur since total system head required exceeds the total available head.

Consider a system where the total available head must be represented by two curves forming a band as shown in Figure 36. This is a common situation which is a result of maximum and minimum values of static head available. It is now possible for system flow to exceed design flow of 100% since the pump selection was based on the minimum value of static head available at 100% flow.

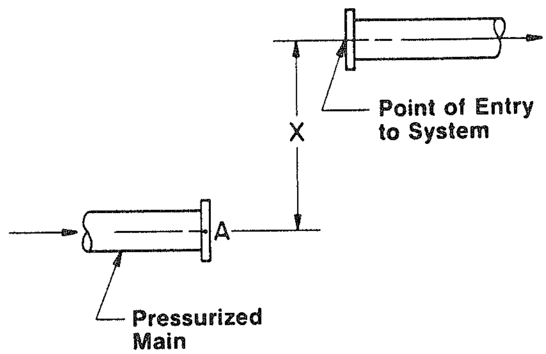


Figure 32

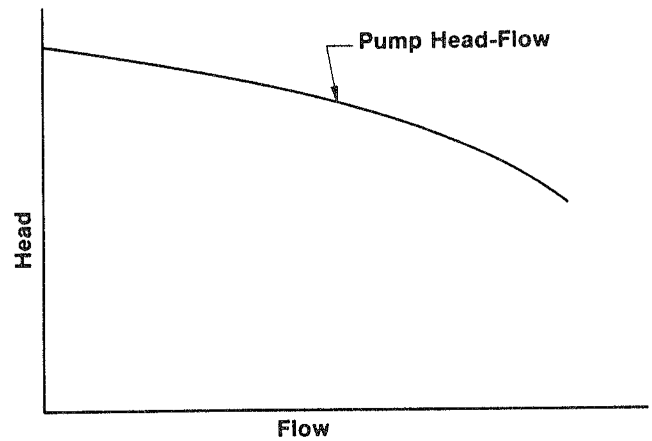


Figure 33

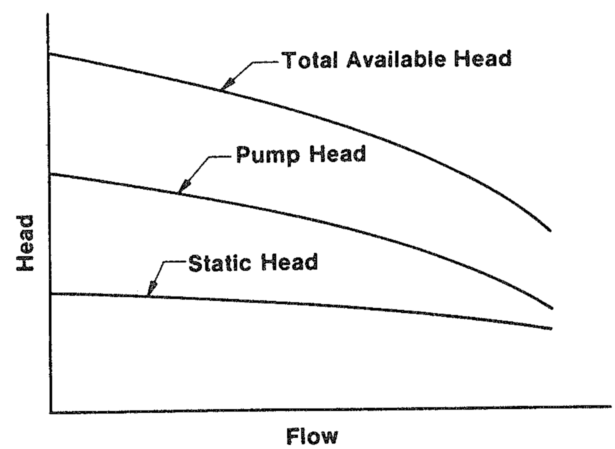


Figure 34

Figure 37 illustrates a system with a band of total system head and a band of total available head. At any given time, system operation can occur anywhere within the area A-B-D-C-A. While the pump is selected for proper operation at design flow, operation at flows other than design may result in problems such as noise, cavitation, low pump efficiency, pump or system damage, or poor system function. To avoid these problems, should they occur, some means of system control must be employed.

Control of system operation

Control of system operation is achieved by altering total system head, total available head or both. Each method has advantages and disadvantages depending on the system, its design function, and the characteristics of total available head. Methods of control should be selected which best satisfy the operation and cost objectives of the system.

Total system head alteration

Control of system operation by altering total system head is accomplished with valves which change the friction head component of total system head. Two methods of valve control are bypassing and throttling and each has a different effect on system head.

Valve bypassing is usually accomplished with valves which have three ports: one for entrance of liquid and two ports which determine the liquid flow path as shown in Figure 38. Movement of the valve mechanism reduces the cross sectional area of one port and simultaneously increases the cross sectional area of the other port. This action causes increased head through one flow path and reduced head through the other; however, the head required at the entrance to the valve remains relatively constant. Figure 39 illustrates how flow remains constant at the entrance of the bypass valve as flows change in the functional portion of the system. The bypass valves may be automatic or manual and can be located anywhere in the system. Regardless of type of operation or location, the end result of valve bypass control is

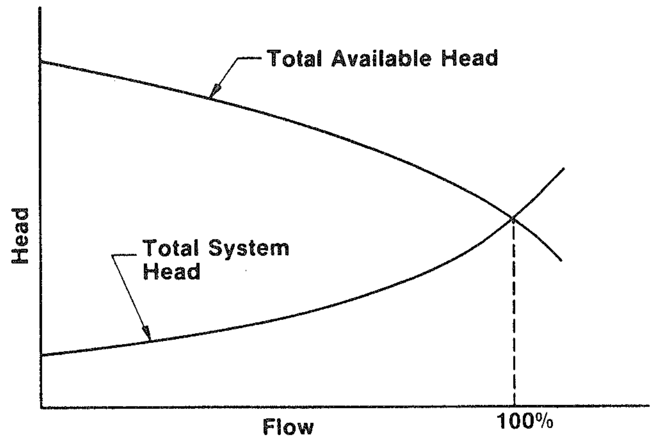


Figure 35

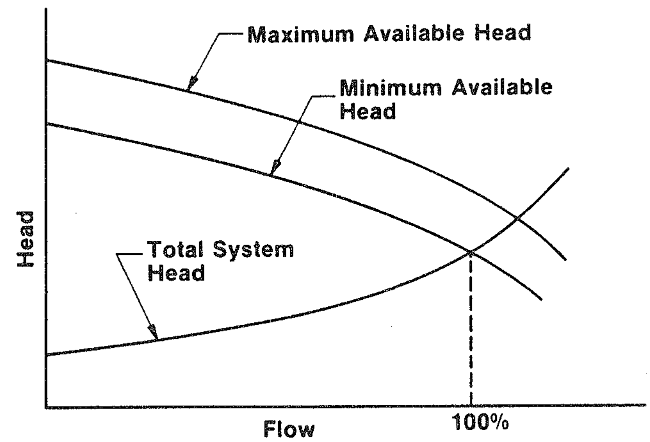


Figure 36

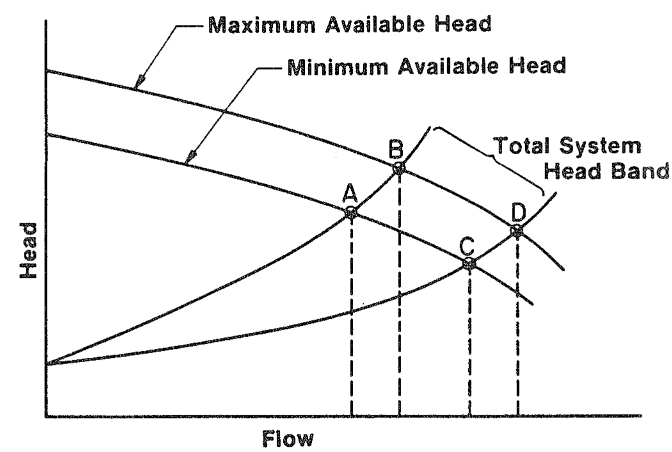


Figure 37

constant total flow from the pump for any given value of total available head.

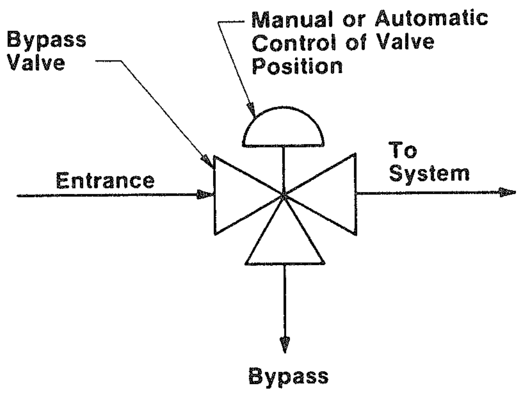


Figure 38

Valve throttling control of system operation is usually accomplished with valves which have two ports, one for the entrance and one for the exit of liquid. Movement of the valve mechanism changes the cross sectional area of the valve port causing friction head to increase to a greater value than the full open valve friction head. From fully open to fully closed position, valve throttling produces a series of system head curves as shown in Figure 40. System friction head will vary depending on relative valve opening and resulting system flow will occur at the point of intersection between the system head curve and the total available head curve.

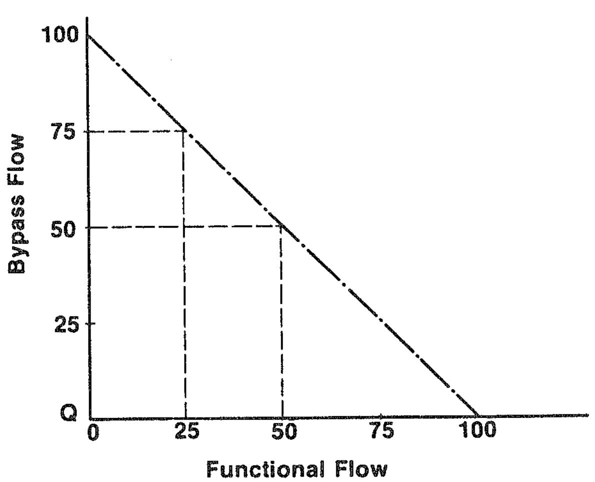


Figure 39

Total available head alteration

Control of system operation by altering the pump head portion of total available head can be accomplished by changing pump impeller diameter, selecting pumps for series operation or parallel operation, or by change of pump operating speed. Each method of control will have a different effect on system operation.

Change of impeller diameter will alter the pump head component of total available head. In systems where constant speed pump impeller diameters have been selected for the *calculated* system design head, and the *actual* system head is less than that calculated, system operating efficiency can usually be improved by machining the impeller diameter to develop the actual operating head of the system. All aspects of pump and system characteristics explained in the preceding sections should be reviewed before considering changing impeller diameter to assure satisfactory pump and system operation.

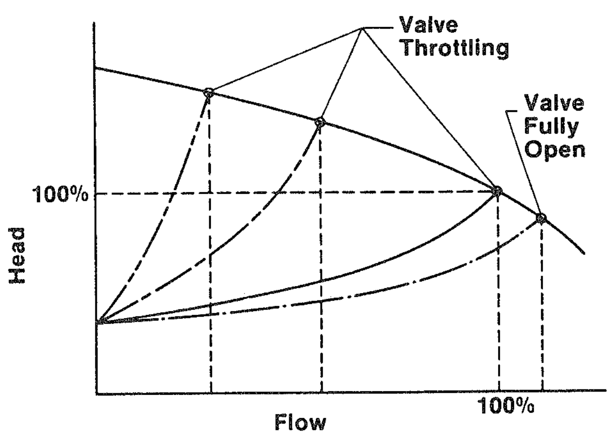


Figure 40

In series pumping the pump head component of total available head is the sum of the heads developed by each pump at any given flow. Each pump must be selected to operate satisfactorily at system design flow. Pumps operated in series can be referred to as "pressure additive," illustrated in

Figure 41. With one pump operating, system flow will occur at point A. With both pumps operating, system flow will occur at point B which is system design flow. In this mode of operation, both pumps are developing equal head at full system flow.

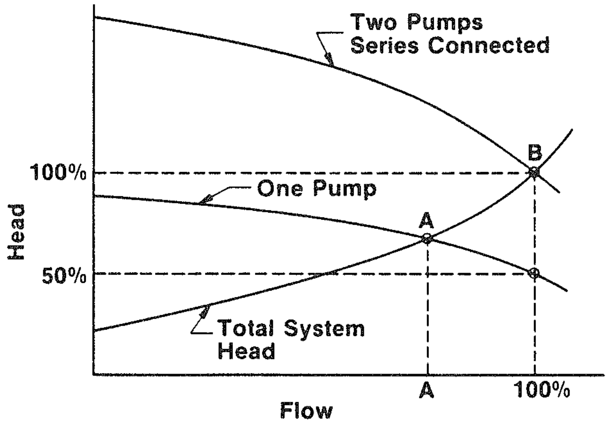


Figure 41

In parallel pumping the pump component of total available head is identical for each pump and the system flow is divided among the number of pumps operating in the system. The flows produced by individual pumps can represent any percentage of total system flow. Where series pumping is described as "pressure additive," parallel pumping is described as "flow additive". When operating in parallel, pumps will always develop an identical head value at whatever their equivalent flow rate is for that developed head and the sum of their capacities will equal system flow. Figure 42 illustrates parallel pumping in which each pump develops 50% of total flow at 100% head. With one pump operating, system flow will occur at point A; with both pumps in operation, flow will occur at point B.

Changing pump speed will alter the pump head component of the total available head. The performance characteristics of a pump at various speeds can be calculated by use of the affinity laws and will result in a pump head-flow curve for each operating speed as shown in Figure 43. The application of variable speed pumps in systems can result in system operation at the intersecting points shown in Figure 44.The number of intersection points will be dependent upon the number of pump speed changes.

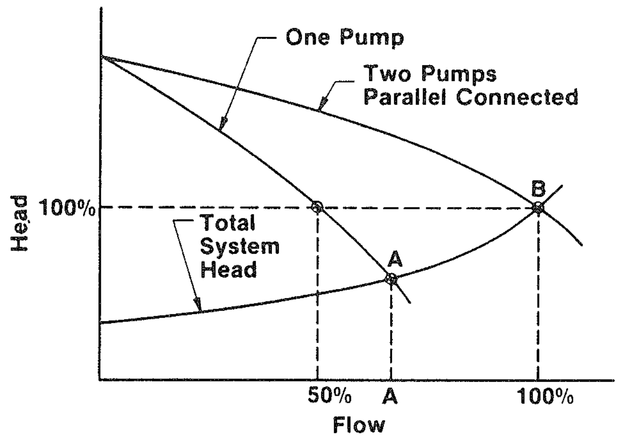


Figure 42

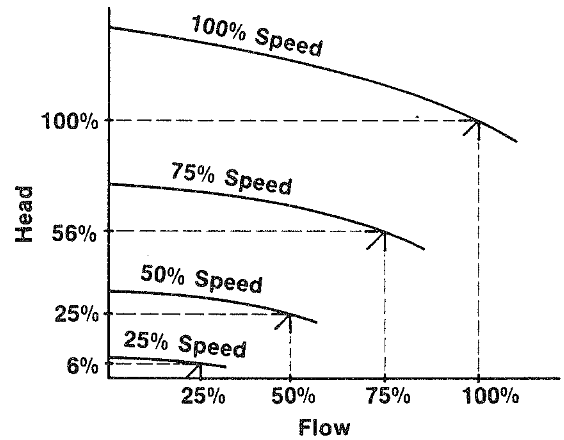


Figure 43

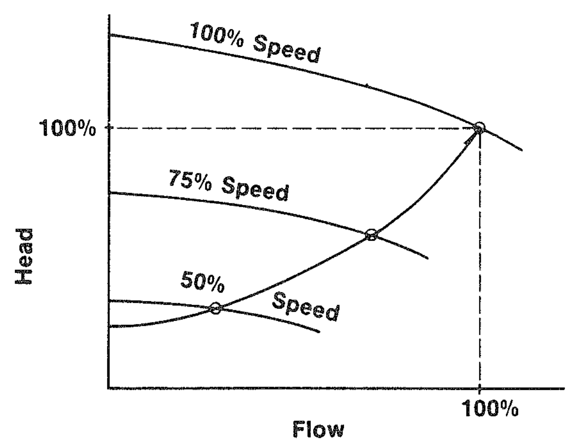


Figure 44

There are several devices available to vary pump operating speeds. Some cause fixed increments of speed change, such as multiple speed motors, while others operate at an infinite number of speeds within the design limits of the device. With drives which have fixed increments of speed change, a fixed number of intersection points occur between the pump head curves and the system head curve as shown in Figure 44. Pump drives which have an infinite number of operating speeds can produce an infinite number of intersecing points on the system head curve.

System sensing — variable speed pumps

When variable speed pumps are employed as a means of system control, the system sensing method which will control the pump speed must be determined. This is necessary because the total available head at the point of entry to the system is a function of the type and location of the monitor.

Functions — The sensing equipment must perform three functions: Monitor the system variable, compare the value of the system variable to the required value (set point), and cause the variable speed pump to restore the variable to the required value. System values typically monitored include level, flow, pressure, differential pressure, temperature, and differential temperature.

Types — Sensing equipment may be as simple as a piece of tubing connecting the piping system to the variable speed drive or as complex as electronic telemetry equipment which monitors variables many miles from the pump.

Location — The total operation of the system must be evaluated at the location where the system variable is being monitored.

Effect of system sensing — Figure 45 illustrates a pumping system where the design function is the removal/delivery of the variable inflow from point A to delivery point B. When the pump is equipped with a variable speed drive, the design function

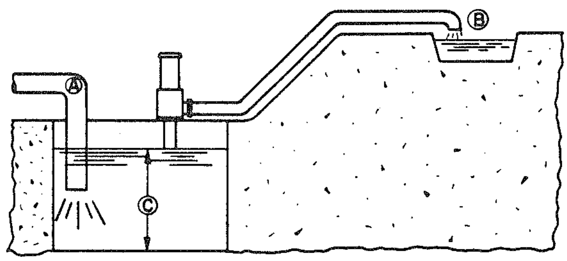


Figure 45

can be achieved by monitoring liquid level in the sump, flow in pipe A, or pressure in the sump. In each instance, an increase or decrease in the value of the monitored variable will cause an increase or decrease in pump speed. The pump will operate at a speed which will allow its total head curve to intersect the total system head curve at the value of actual flow at point A. All pump operating points will occur on the unaltered total system head curve regardless of which variable is monitored, or the physical location of the monitor. This type of operation will occur in systems where flow change results *only* from a change in available head.

When equipment in the system causes a change in flow, the path of pump operating points will not follow the unaltered system head curve. An infinite number of new system head curves will occur when system components cause flow change. For example, in a system which requires a fixed design working head, the path of pump and system operating points can be predicted once the location of the system monitoring device is established. This path of operation will be controlled by the location of the monitor and will affect the operating speeds and power requirements of the pump.

Figure 46 illustrates a simple system where flow change is caused by a valve near the end of the system. The system head curve for this example is shown in Figure 47, and includes static head (S_1) and friction head (H_f) between points A and C with the valve in a fixed position.

A variable speed pump with a pressure monitor at the point of entry to the system, A, will produce a constant total available head at A for any system flow. The setting for the pressure monitor is determined from the following formula:

$P_1 = P_2 + H_f + S$ where:

P_1 = Total available head in feet to be maintained at the pressure monitor.

P_2 = design working head in feet.

H_f = full flow friction head in feet between the monitor and the location in the system where P_2 is to be maintained, or from the monitor to the end of the system (A to C) when P_2 is zero.

S = Static head in feet between the monitor and the location in the system where P_2 is to be maintained, or from the monitor to the highest point of discharge. Static head, S, is applicable only in systems employing pressure monitors.

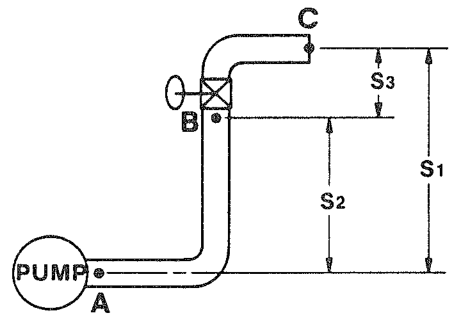


Figure 46

In Figure 46, P_2 is zero and $P_{1A} = H_f + S_1$.

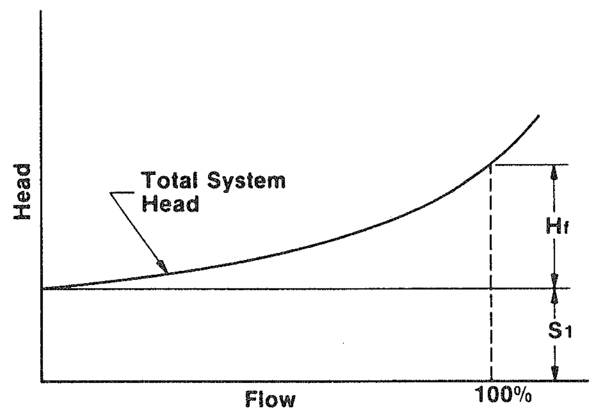


Figure 47

As system flow is throttled, a series of new system head curves will be produced as shown in Figure 48. System flow will occur at those points where the system head curves intersect the available head curve. At any flow less than 100% there is more head available than required and the excess is consumed as friction head across the throttling valve. Now consider the same system with the pressure monitor located in Figure 46 at point B. The system has not changed, and the total system head required is still as shown in Figure 47. The pressure to be maintained at the monitor is determined from the same formula:

$P_1 = P_2 + H_f + S$

Design working head, P_2, is zero, and static head is equal to S_3. H_f is the full flow friction from B to C. The pressure setting $P_{1B} = H_f + S_3$. The total head which must be available at point A in order to satisfy the monitor at point B will be equal to the pressure monitor setting, P_{1B}, plus the static head, S_2, plus the friction head from A to B. This is illustrated in Figure 49.

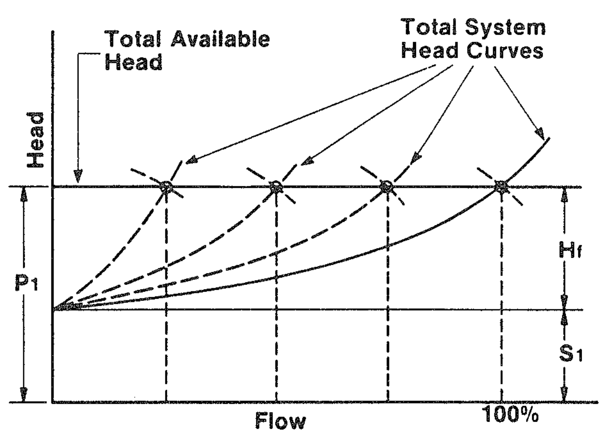


Figure 48

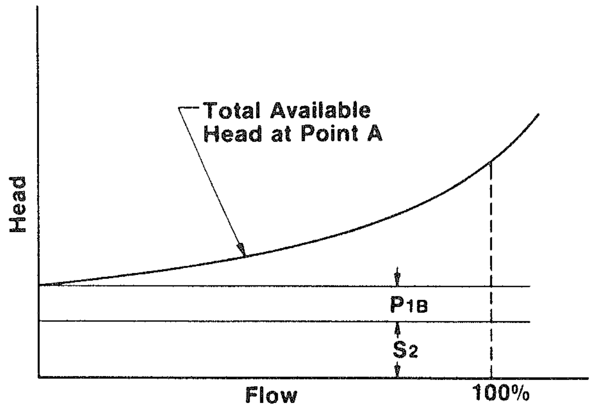


Figure 49

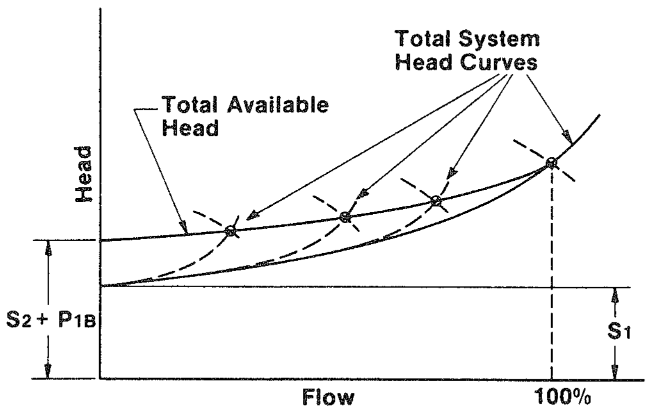


Figure 50

As system flow is throttled, a series of system head curves will be produced as shown in Figure 50. System flow will occur at those points where the system head curves intersect the total available head curve. By comparing Figures 48 and 50, the throttling losses are less with the pressure monitor at B than they are with the pressure monitor at A for specific values of flow. This can result in lower pump operating speed, lower horsepower required, and lower operating cost.

Sensing multiple branch systems — In parallel piping systems with two or more branches, a single pressure monitor must be set to satisfy the branch with the maximum calculated friction head. In some instances, load variation in the system can cause the pressure in non-monitored branches to drop below acceptable limits, even though the monitored branch is satisfied. This type of problem can be resolved by analyzing the various branches based on the location and setting of a single pressure monitor. Additional pressure monitors can be added to those branches which experience the low pressure condition to restore system operation to acceptable levels.

Conclusion

Evaluation of the total available head band and the total system head band over the expected range of system flow is necessary to make decisions which will affect the cost and operation of a pumping system. The decisions are based on problem statements and evaluation of the possible solutions with respect to the cost and functional objectives of the system design. The following questions outline some of the considerations that should be reviewed to assure satisfactory operation of a pumping system:

1. If the system is throttled for reduced system flow, does the maximum total available head exceed the system working pressure?

   If so, the alternatives include:
   a) Increase system working pressure
   b) Select pump with a "flatter" head capacity characteristic
   c) Consider variable speed pump drive
   d) Add bypass valve or pressure reducing valve to system

2. Does pump horsepower increase with decreasing flow?

   If it does, the alternatives include:
   a) Add bypass valve to system
   b) Consider variable speed pump drive
   c) Select several lower specific speed pumps to operate in parallel.

3. Will operation at reduced flows occur for significant periods of time?

   If so, pump efficiency will be reduced and the service life of pump bearings, packing, mechanical seals and close clearance wearing rings will usually be shortened and the alternatives include:

a) Select smaller pump for the reduced flow operation
b) Add bypass valve to system
c) Consider variable speed pump drive

4. Will actual system head at design flow be considerably less than the calculated value?

   If so, the system flow will exceed design flow and the alternatives include:

   a) Reduce the pump impeller diameter
   b) Install valve to throttle system flow
   c) Consider variable speed pump drive

5. In systems planned for two or more pumps operating in parallel, have all the operating possibilities been considered?

   For example, one pump alone can operate at higher flow than its rating as selected for parallel operation. For the higher flow operation the pump will require more net positive suction head (NPSH), and may be subject to undesirable hydraulic loads, motor overload, increased noise level and shortened life.

These are only a few of the many problems which can be encountered in the course of designing a pumping system. In many instances, several solutions can provide equally satisfactory results if operating power requirements are not considered. *If design objectives include minimum power consumption, an energy evaluation of the pumping system must be made.*

6

Total System Evaluation®

Introduction

Total System Evaluation® is the process of developing that information about the system and its pumping equipment which is necessary to satisfy a system's design requirements, concluding with a projection of total power requirements. To perform the evaluation, system head and total available head must be established. These two components are combined with pump power requirements and system load profile to project total system operating costs.

Load profile

Load profile expresses the measure of work executed in a system compared to a unit of time. Work performed has a direct relationship to flow; therefore load expressed as flow provides a common base for any system energy requirement projection. System load profile can be illustrated with a curve as shown in Figure 51; however, it is generally easier to use the flow/time relationship in tabular form as illustrated in Table 6. The tabular format organizes total operational time at specific flow values and simplifies identifying high flow/time concentration areas which should be the focal point for preliminary pumping equipment selection.

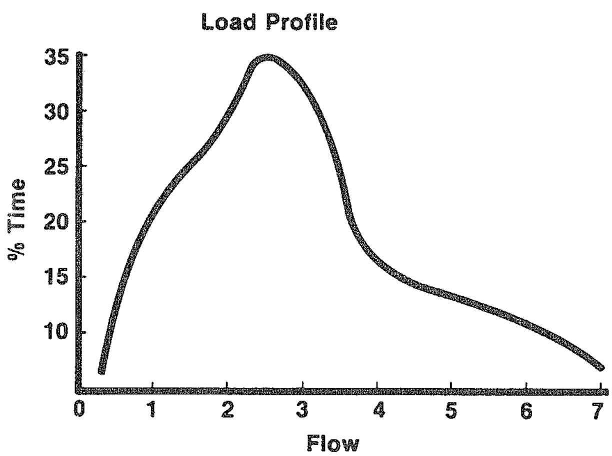


Figure 51

Table 6

FLOW GPM	HOURS
280	87
840	1,138
1,680	1,841
2,520	2,629
3,640	1,227
4,480	788
5,600	613
6,440	350
7,000	87
—	8,760

Figure 52 illustrates how the flow/time relationship can be combined with a system head curve to aid in pump selection procedures. In this example, a significant amount of system operational time will occur in the flow ranges of 20% to 30% and 60% to 75% of the total design flow. Pumping equipment should be selected to pro-

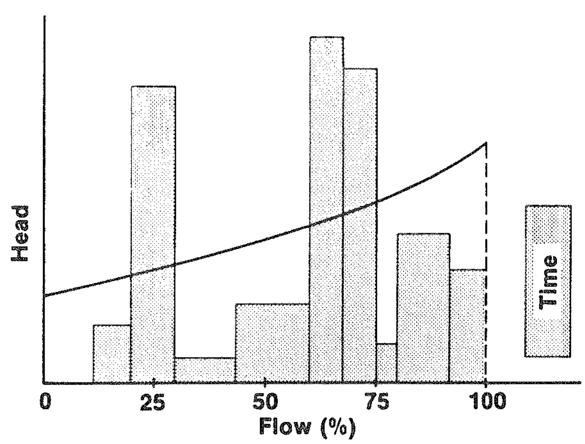


Figure 52

vide maximum operating efficiencies in these flow ranges. One possible pump equipment selection for this system would include a single pump selected for the system head and flow requirements in the 20% to 30% range and two identically sized pumps to share the total system capacity equally at full system flow, but selected so that each will be in its best efficiency range when operated independently in the 60% to 75% system flow range (Figure 53).

The time base for computing load requirements should be of sufficient length to incorporate all variables affecting system operation. The minimum practical load profile time length is one complete cycle, from minimum to maximum flow requirements. A twelve month time base for total system load evaluation generally encompasses all production, load, and climatic variables for a system and can be used to project system operating requirements and costs. System load profile is the central component of any energy evaluation process.

Variable available head — In projecting total system energy requirements, total available head must be reviewed. If static head available is variable with respect to system flow, time, or both, a profile of these relationships should be made. This projection should represent the magnitude of change compared to the length of time the change is expected to occur, thus developing a correlation with system load profile. If the accuracy of these data is questionable, a band of minimum to maximum values should be established from which a total system power requirement band can be developed.

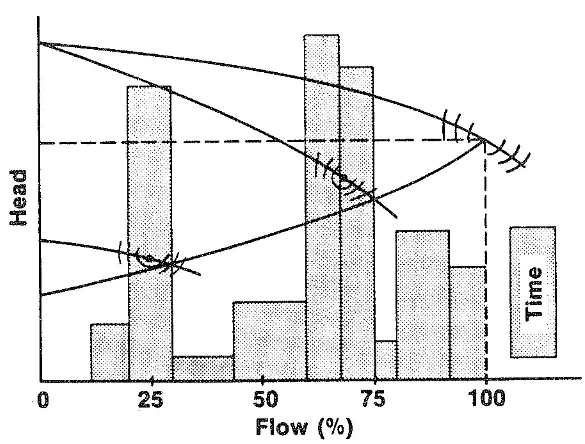


Figure 53

Determining power requirements

In projecting total power requirements for any liquid handling system, pump horsepowers at all significant flow rates between minimum and maximum conditions must be developed. Significant flow rates can be determined by reviewing the system load profile. To determine power inputs at various flow rates, it is necessary to know if the pumping equipment shares the total system flow or head equally or unequally, and whether the pumping equipment operates at fixed speed, infinitely variable speed, or a combination of both. The system monitoring means must also be known when infinitely variable speed pumps are employed.

Equal size — When pumps are combined to share total system flow or head equally, the system power requirement at any flow rate can be computed based on individual pump horsepower. This definition is compatible with any range or quantity of pumps employed to satisfy all system flow requirements. The energy input will be governed by the number of pumps necessary to satisfy those requirements. In this case, the flows or heads of individual pumps combine equally to satisfy total system requirements and the total power required at any flow rate is the sum of the power required by the individual pumps.

Unequal size — When unequal size pumps are combined to satisfy system flow or head requirements (i.e., unequal percentage of total required) special consideration must be given to the pump operating sequence to ensure that maximum system efficiency is achieved. A Pump Sequence Energy form for analyzing minimum power requirements for systems incorporating unequal size pumps is available from Peerless Pump for this purpose.

Initial pump selections should be compared to the system head band and system load profile to establish a basic operating sequence for significant system flow values. Horsepower computations for the individual pumps are then made and recorded on the data form. To ensure that the selected order of pump operation results in the minimum power input necessary to satisfy the system requirements, rearrange the pump operating sequence at individual system flow rates, compute new pump horsepower requirements, and enter this information on the data form.

This procedure will illustrate that for any system flow rate there is a single combination of pumps which requires minimum power input to satisfy the system requirements.

For systems where the pump contribution to the total available head is variable, or where the system required head is variable, these forms may need to be completed for all major operating conditions to determine if the pump operating sequence needs to be altered or modified during these variable conditions.

Projecting pump horsepower

Constant speed — To compute constant speed pump horsepower for parallel, series, equal, or unequal combinations, all pump characteristics necessary are available on the manufacturers' performance curves. For pump operation with fixed speed change increments, the characteristics of flow, head, and brake horsepower can be calculated from the manufacturers' curves using the affinity laws. Therefore, power requirements for all constant speed pumps,and pumps of fixed speed increments, can be projected from available data.

Variable speed — To compute variable speed pump horsepower it is first necessary to determine system monitor type and location. Type and location will determine head available, and head available will determine power required. Unfortunately there is no simple mathematical relationship which can be developed for computing the interacting characterteristics of systems and variable speed pumps. The derivation of the affinty laws,

$$Q_2 = Q_1 \sqrt{\frac{H_2}{H_1}},$$

is helpful in computing performance for individual variable speed pumps. In this formula, Q_1-H_1 represents any single value of flow and head taken from the total system head curve. Q_2-H_2 represents a point on a pump total head curve of known operating speed which will satisfy the formula. The relationship of these values is shown in Figure 54. When the formula has been satisfied, the pump operating speed necessary to produce system conditions Q_1-H_1 can be calculated.

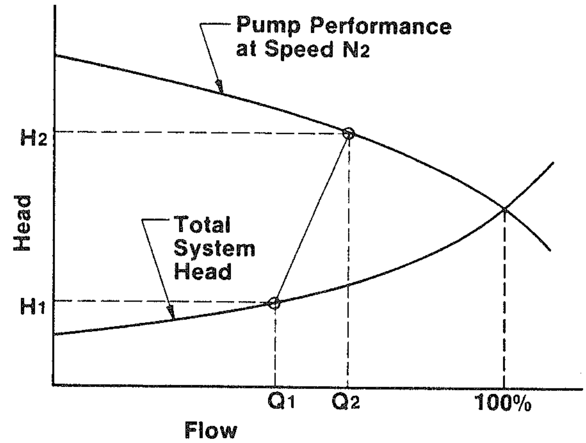


Figure 54

In using the proposed formula, values known include system flow (Q_1), system head (H_1), and full pump operating speed (N_2). Values unknown are the equivalent pump flow (Q_2), equivalent pump head (H_2), and the unknown speed (N_1) at which the pump must operate to produce the required system flow and head. When making variable speed pump brake horsepower calculations, it is necessary to know the equivalent point on the full speed characteristic curve which equates to the required system flow and head, since the efficiency at the equivalent point will reoccur at the reduced flow and head condition. Variable speed pump brake horsepower is computed from the formula:

$$BHP_1 = \frac{Q_1 \times H_1 \times \text{Specific Gravity}}{3960 \times \text{Eff.}_2}$$

where Eff._2 is the efficiency from the full speed pump curve at Q_2, H_2 expressed as a decimal value, Q_1 is USGPM and H_1 is feet of head. The reduced pump operating speed necessary to satisfy the system flow and head can be projected from the affinity law derivation:

$$N_1 = \left(\frac{Q_1}{Q_2}\right) N_2$$

This value is used when computing brake horsepower requirements for variable speed equipment where drive efficiency is a function of output to input speed ratio. The expected range of pump operating speed should be determined and compared by the pump manufacturer to the critical speed of the pump. If operating speed occurs at or near pump critical speed it will be necessary to modify the pump or to select a different pump for the application.

Determining the precise Q_2-H_2 point on the pump curve will usually require reiterative calcuation. The triangulation procedure illustrated in Figures 55 through 60 locates an intercept point on the pump curve which closely approximates the actual flow and head solution, Q_2-H_2. The accuracy of the approximated point is affected by the relative magnitude of H_1, H_2 and the slopes of the pump and system head curves. To satisfy the formula, and thus determine the point on the pump curve of the known speed which will produce the required system values Q_1-H_1 at reduced speed, it is first necessary to calculate the initial Q_2-H_2 condition as a reference point. This is done by entering known system conditions Q_1-H_1, and an initial value of H_2 in the formula. The initial value of H_2 is taken from the known pump performance curve at system flow Q_1. By solving the formula for the initial value of Q_2, a reference point for initial Q_2-H_2 is established. By connecting the initial (reference) values of Q_2-H_2 with required system condition Q_1-H_1, an intersection point with the known pump curve is established. This intersection point becomes the approximate Q_2-H_2 value which will, at reduced operating speed, produce system conditions Q_1-H_1.

In a parallel pumping system where system flow is shared equally by all pumps, the procedure is identical to that for a single pump system, except that system flow (Q_1) must be divided by the number of pumps (P_n) operating. Figure 56 illustrates the triangulation process for a system incorporating two equally sized pumps operating in parallel.

When pumps selected to share unequal portions of total system flow are operating, the individual

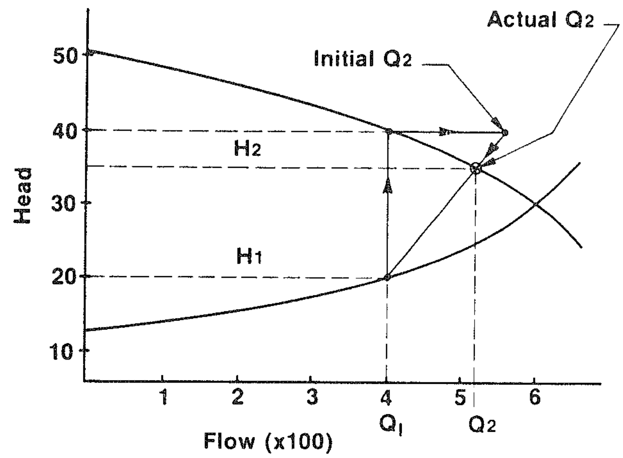


Figure 55

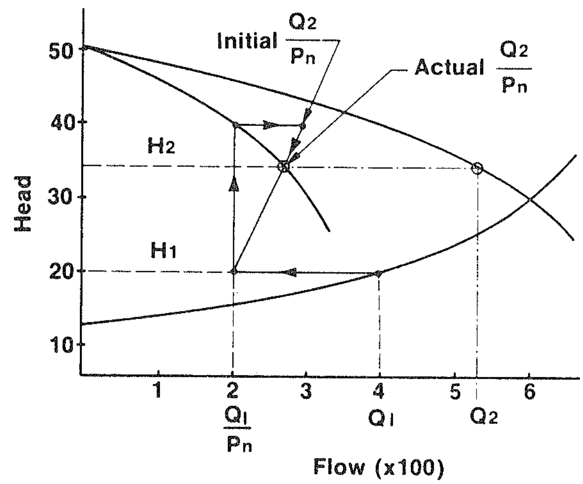


Figure 56

pump solution formula must be modified to $Q_1 = Q_2\sqrt{H_1/H_2}$. This is necessary since the flow produced by each pump for a selected system flow (Q_1) is unknown. To determine the flow and head developed by unequal size variable speed pumps when they are both contributing to total system flow, it is first necessary to solve the basic triangulation process for system flow Q_1 on the combined pump characteristic curve (Figure 57). The intersection point which occurs on the combined pump curve establishes the H_2 value for the individual pumps. Use of the formula, $Q_1 = Q_2\sqrt{H_1/H_2}$, determines individual pump flow $Q_1(P_1)$ and $Q_2(P_2)$ for the system operation at Q_1. System power requirements at flow Q_1 are determined by adding the horsepower requirements of each pump.

When a variable speed pump is combined with a constant speed pump, the constant speed pump will always produce flow at the point where the developed head matches required system head. When system flow requirements are such that the constant speed and the variable speed pumps must be operated in parallel, flow produced by the variable speed pump will be the difference between required system flow Q_1 and constant speed pump flow at system head H_1.

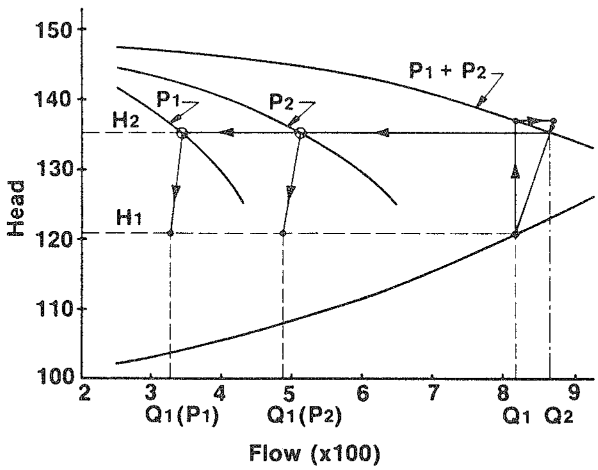


Figure 57

Figure 58 illustrates a two pump system where both pumps are of equal size, one constant speed and one variable speed. With a required system flow of Q_1 and head of H_1, the two pumps will develop unequal percentages of the total required flow. As system flow changes, the percentage of flow contribution by each pump will change.

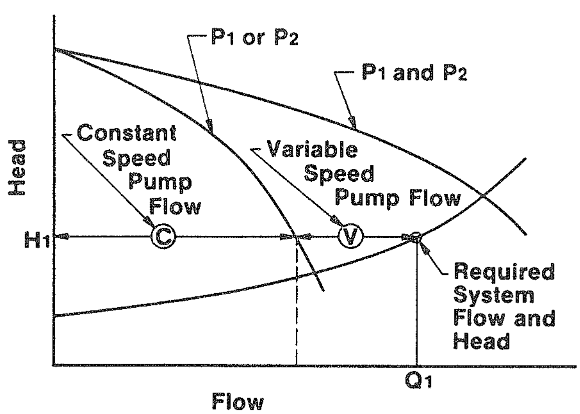


Figure 58

To determine total system power requirements under these circumstances, it is necessary to exercise the basic triangulation procedure for the variable speed pump based on the known capacity it will develop at the required head of the system. The horsepower of the variable speed pump is calculated and added to the horsepower required by the constant speed pump at its developed head and flow.

Projecting variable speed pump performance under conditions of variable static available head requires the formula solution to be equated to the actual developed head of the pump. Use of the normal projection formula is sufficient, and the equivalent point on the pump head curve will result from projecting the actual Q_2, H_2 solution and subtracting the static head. This will accurately determine the equivalent point on the pump head curve which equates to the system required head and flow. Figure 59 illustrates the effect of variable available static head.

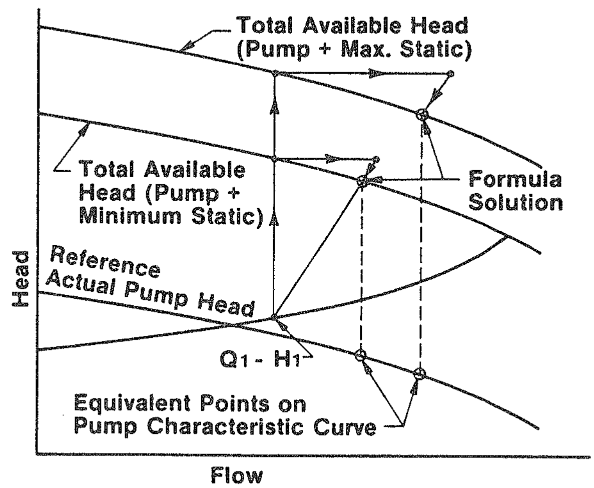


Figure 59

When friction head in the supply pipe and fittings significantly reduces pump head, it is necessary to project the triangulation solution to the pump head curve only. This is illustrated in Figure 60. The projection values can be computed from either the total available head curve or the pump curve but the intersection solution must always be related to the equivalent point on the individual pump head curve.

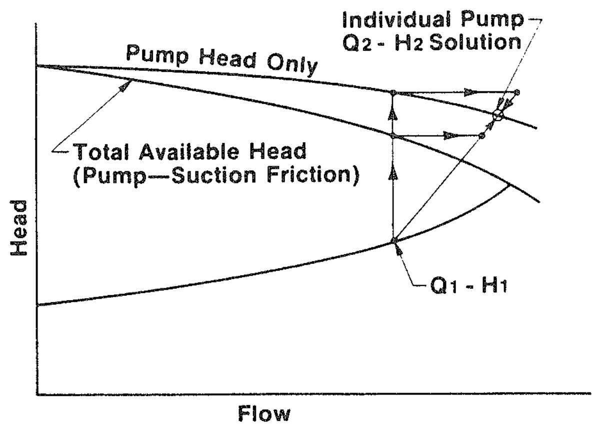


Figure 60

The accuracy of the triangulation procedure will depend to a great extent on the accuracy of pump and system curve data used.

Special considerations — In the employment of variable speed drives whose efficiency is a function of output to input speed, system operating efficiency can be improved when major system operating conditions require pump speed changes below the nominal speeds available from standard multiple speed AC induction motors. Careful comparison of the operating speed range to the system load profile should be made. If the time/load relationship is of sufficient magnitude, motor speed can be reduced to improve the variable speed drive efficiency by altering output to input speed ratios.

Numerous contingencies must be considered in the analysis of this type of application to ensure that all mechanical and hydraulic considerations are taken into account.

Equipment power comparisons

Accurate comparisons of the various kinds of pumping equipment under consideration for a given application make it necessary to establish a common point of reference. That reference point is the power which the prime mover must supply at its output shaft.

Constant speed — To project the total system power required for a constant speed pump, divide the pump horsepower by the prime mover efficiency.

Variable speed — The horsepower at the output shaft of the prime mover for a pump driven by a variable speed device is derived by dividing the pump brake horsepower by both the prime mover efficiency and the variable speed device efficiency. This applies whether the variable speed device is between the pump and the prime mover or ahead of the prime mover as shown in Figure 61.

Do not be misled by the lower net efficiency of the equipment which includes the variable speed device. Via pump operation which is closely aligned to system needs, significantly valuable operating cost savings can be obtained, particularly in systems which have variable loads. The greater initial cost of the variable speed equipment can often be amortized in a relatively short time after which a net reduction in overall system operating costs can be obtained.

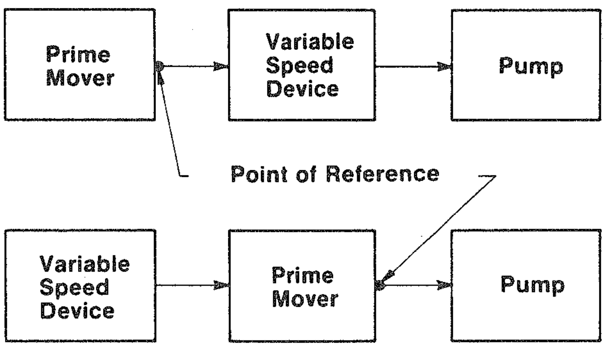


Figure 61

Total energy projection

To project the total energy requirements for any system with any combination of pumping equipment and to assure that all answers are projected from a common base, a structured procedure for the analysis should be followed; such a procedure is outlined in Table 7. Each item in the procedure will have a significant impact on the total energy requirement of a specified pump-equipment combination and system operating procedure.

To make comparisons of dissimilar equipment or to evaluate the impact of system operation change, it is necessary to follow the evaluation procedure for each specified condition. The individual steps of the procedure are defined as follows:

Step 1 — At the point of entry to the evaluation procedure, it is necessary to have established and assembled all information relative to the system design requirements. An accurate projection of system flow and head requirements, as illustrated by a system head curve or band, is necessary at this stage of evaluation.

Step 2 — The load profile of the system should be reviewed to make preliminary pump equipment selections and to establish the evaluation base for the system under consideration. If future system conditions such as expansion, etc. will significantly alter the load profile these considerations should be confirmed at this time.

Step 3 — Pump selection evaluation. Beyond the normal mechanical and hydraulic criteria associated with pump selection for a specific application, the selection procedure should encompass the primary operating efficiency at the point of selection and the amount of efficiency deviation system flow variation will cause. These values should then be compared to the operating time at specified flow rates to assure that the selection offers the maximum potential operating efficiency through the range of maximum flow/time concentration.

Step 4 — Preliminary decisions must be made relative to the control means which will be employed to satisfy system flow and head requirements. The decision to control system required head, total available head, or both, will be a significant factor in the total energy requirements projected for the system being evaluated.

Step 5 — This stage of the evaluation requires a projection of the pumping equipment power requirements over the entire system operating flow range. The pump required power can be organized in tabular or graphical form as best suits the user. In tabular form, the power requirements of the pumping system, compared to required system flow, should be of sufficient quantity to allow the user an analysis of all

Table 7

1	**Enter with new or revised system requirement data**
2	**Review preliminary load profile to establish system operating parameters**
3	**Select pumps based on system requirements and load profile**
4	**Determine system control**
5	**Determine power requirements**
6	**Compute total energy requirements in energy/time units**
7	**Summarize and review data**

significant operating areas as dictated by the flow/time profile.

Step 6 — The load profile is used to compute the energy requirements of the selected pumping equipment and system operating procedures. When compared to the power projection in Step 5, the load profile will determine the *total* energy requirements of the system, thus concluding the evaluation in energy time units (brake horsepower hours, kilowatt hours, etc.).

Step 7 — Determination of the total energy required to satisfy all projected system operating conditions is the basis from which all other considerations and comparisons can be made. This is the reviewing and decision making phase of the total system evaluation procedure. If the energy summary is adequate, the evaluation procedures are concluded; if not, the evaluation can be repeated with new or modified data to determine what effect the change will have.

Conclusion

When the total system energy answer has been determined, normal decision making processes to select that combination of pumping equipment and system operating procedures which most effectively satisfy the design goals of the system can begin. Total system evaluation ensures that all projections, comparisons, and decisions are based on as much factual information as possible.

When comparisons of various combinations of pump equipment and system operating procedures are desired, total system evaluation allows the comparisons to be made on an equal basis — the requirements of the system. Total system evaluation is a structured, fundamental process for determining the pump and system combination that best suits the cost-benefit-functionality requirements of the user.

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