BUILDING ENERGY MANAGEMENT AND REDESIGN RETROFIT (BEMARR)

PUMPS AND PUMPING SYSTEMS

CONTENTS PAGE

Figures

CONTENTS PAGE

1. GENERAL

1.01 This section provides information concerning pumps and pumping systems for building mechanical systems. The section covers pump efficiencies, zone pumping, replacement of 3-way valves with 2-way valves on constant speed pumping systems, and variable speed pumps. The material used in this section has been extracted from the *Building Energy Management and Redesign Retrofit (BEMARR) Manual* issued with GL-76-10077 (EL-4857), dated October 7, 1976.

1.02 Whenever this section is reissued, the reason(s) for reissue will be listed in this paragraph.

2. REDUCTION IN PUMPING ENERGY

2.01 Centrifugal pumps, the type of pump generally used for water flow in building mechanical systems, are major energy users. The power required for pumps is determined by the flow rate of the heating or cooling water and the head or pressure loss in the system.

2.02 The following energy conservation strategies can reduce energy consumption:

(a) Replacement of less efficient pumping equipment

(b) Proportional balance of water flow and impeller trim

(c) Zone pumping

(d) Two-way control valves

(e) Variable volume pumping.

3. PUMP EFFICIENCIES

3.01 Replacement or modernization of mechanical equipment can increase system efficiency and reduce energy consumption. Pumps are a good example. Pump efficiencies of 30 percent are not uncommon, while pump efficiencies of 80 percent are available. Selecting a pump with a low efficiency could increase energy consumption as much as 166 percent. Instead of immediately using the first pump that meets the design requirements, the efficiency of that pump should be reviewed. For example, for the same requirements [160 foot head and 60 gallons per minute (gpm)], a pump could be selected with impeller sizes 6, 6-1/2, and 7 inches having motors of 7-1/2, 10, and 15 horsepower (hp), respectively.

> ***Note:*** An increase in pump efficiency means that less horsepower or energy will be required to do a given pumping job.

SYSTEM CURVE

3.02 In the past, not much attention was paid to the system curve. The system curve illustrates the variation in total system head (system pressure loss) as the system flow rate varies (Fig. 1). It provides the designer with the information required to evaluate the benefits available from such energy conservation strategies as variable volume pumping, zone pumping, etc.

3.03 The system curve is constructed using three points that are calculated from the following relationship:

$$\frac{H_1}{H_{2,3}} = \left(\frac{Q_1}{Q_{2,3}}\right)^2$$

Where:

H_1 and Q_1 are design conditions
$H_{2,3}$ and $Q_{2,3}$ are intermediate (reduced load) conditions.

PUMP CURVES

3.04 Pump performance is shown by the pump-head capacity curve which relates the flow in gallons per minute to total head (feet) (Fig. 2). The pump curve is established under test conditions by the pump manufacturer and based on the type of pump, size of impeller, motor horsepower, and shaft revolutions per minute (rpm).

EVALUATING PUMP CURVES

3.05 Plotting or constructing the system and pump curves (for a pump with specific characteristics on the same graph) will yield the point of pump operation. This point is at the intersection of the system curve and the pump curve (Fig. 3).

PUMP EFFICIENCY CURVES

3.06 Pump efficiency curves are plotted with the basic pump curve. The maximum efficiency occurs at a particular point or within a range on the pump curve (Fig. 4). Pump efficiency curves are generally shown with brake horsepower (bhp) curves to enable selection of pump size and motor for the job conditions.

PUMP EFFICIENCY ANALYSIS

3.07 A 1 bhp, continuously operating pump with the efficiency of 100 percent will use:

$$1 \text{ bhp} \times 0.7457 \frac{\text{kW}}{\text{hp}} \times 8760 \frac{\text{hours}}{\text{year}}$$

$$= 6532 \text{ kWh/year}$$

3.08 With lower efficiency, the amount of energy required to deliver the same flow rate against the same head increases. For example at 80 percent efficiency, the power consumption increases to:

$$\frac{6532 \text{ kWh}}{80\%/100} \times 100 = 8165 \text{ kWh/year}$$

This is on an increase of 25 percent in power consumption over the ideal or 100 percent efficient pump.

3.09 However, 80 percent is about the best actual efficiency available from a pump. Therefore, the 80 percent efficient value should be used as the datum. At 30 percent efficiency, energy consumption increases to:

$$\frac{6532 \text{ kWh}}{30\%/100} = 21{,}773 \text{ kWh/year}$$

This represents an increase of 166 percent in energy consumption over the 80 percent efficient pump.

3.10 The true cost savings for a specific application can be determined by using life-cycle cost techniques.

4. PROPORTIONAL BALANCE OF WATER FLOW AND IMPELLER TRIM

4.01 It has been common design practice to add a 10 to 100 percent safety factor to the system head (pressure loss) to provide a measure of spare capacity for the future and to assure adequate terminal flow rates. This resulted in little gain in terms of increased terminal heat transfer for the large increase in pumping costs. It is charcteristic of centrifugal pumps for the power requirement to increase with flow even though the head decreases.

4.02 A proportional balance of water flow and impeller trim can reduce pumping energy on oversized pumps as previously described by matching the system water flow to the required heating or cooling load.

4.03 Proportional balance is accomplished by setting each terminal or coil to receive its percentage share of the total available water flow. Balancing valves are required at each terminal and pump. Adjusting the flow through these balancing valves throttles the system flow, thereby reducing the quantity of water pumped, and moves the operating point up along the pump curve into a more efficient operating range.

4.04 Consider, for example, a basic system in which the design conditions were 2000 gallons per minute at 60 feet head. The pump installed in this system was originally selected for 2000 gallons per minute at 100 feet head, is 70 percent efficient, and requires 100 brake horsepower. The system flow required for the load is 2000 gallons per minute.

(a) We begin our analysis with the pump-head capacity curve and a plot of the system curve at the original design conditions. The intersection of the design system curve with the pump curve (Fig. 5) is 2460 gallons per minute at 92 feet of head. This is shown at Point 2. For example, inspection of the pump pressure gauge reveals that system head loss is actually 85 feet. Remembering that the pump curve represents the conditions of flow at which the pump will operate, the pump is actually pumping 2900 gallons per minute at 85 feet head. This is Point 1 of the pump curve. This also means that the system curve, drawn from calculated design conditions, is in error. All operating conditions must fall on the system curve. Redraw the actual system curve based on field measured conditions as shown in Fig. 5. This curve shows that the design flow of 2000 gallons per minute can be attained with a head loss of 40 feet as opposed to the design value of 60 feet.

(b) The brake horsepower requirements can be calculated from the following:

$$\text{bhp} = \frac{\text{gpm} \times \text{Head (ft)} \times \text{spec gravity}}{3960 \times \text{Efficiency}}$$

Specific gravity of water = 1.0

For the pump operating at Point 1:

$$\text{bhp} = \frac{2900 \times 85 \text{ feet} \times 1.0}{3960 \times 70/100}$$

bhp = 89

For the pump operating at Point 2:

$$\text{bhp} = \frac{2460 \text{ gpm} \times 92 \text{ feet}}{3960 \times 70/100}$$

bhp = 82

Note: Analysis of the actual operating point (Point 1) for this pump indicates that the pump selected was not a good choice to begin with since it is operating near the end of the pump curve.

Pumps are generally selected for the operating point to fall within the midpoint ±1/4 as shown in Fig. 5. Pumps operating towards either end of the pump curve are inefficient as shown in Fig. 4.

(c) System balancing to reduce system flow can be accomplished with the terminal and pump balancing valves by increasing the head loss to cause the pump operating point to move left on the pump curve. For example, increasing the head loss to 92 feet (Point 2) will move the operating point of the pump to the point of intersection with the design system curve. A reduction of 7 brake horsepower (89 bhp - 82 bhp) will be accomplished; however, the flow rate is determined to be 2460 gallons per minute which is still in excess of the system flow rate requirements of 2000 gallons per minute.

(d) Further reduction in flow rate to that which is required may be possible by further increasing the system head with the balancing valves. If flow is further reduced to 2000 gallons per minute, the brake horsepower becomes 72.5 (Point 3); however, this will probably cause flow problems in the individual flow circuits due to the large amount of excess pump head being absorbed in the balance valves. This is referred to as sharply set balance valves.

(e) Inspection of the impeller curves for the particular pump installed indicates that the system requirements of 2000 gallons per minute at 40 feet head can be met with a smaller impeller.

The ideal impeller would be one that would intersect the system curve at 2000 gpm and 40 ft. (See Fig. 6, Point 5.) The power draw at this point would be 28 bhp. However, the best size (trimmed) impeller available for our pump will pump approximately 2350 gpm at 53 ft. (This is the point of intersection between the impeller curve and the system curve.) To match the flow rate to that required, a balance valve is used to increase the system head to 60 ft which produces a flow of 2000 gpm (Fig. 6, Point 4). Thus, the power reduction obtained is:

$$89 - 43 = 46 \text{ bhp}$$

4.05 For example, the savings in electrical energy and costs can be calculated using the following assumptions:

(a) \$0.05 kWh electrical cost.

(b) Pump runs continuously or 8760 hrs/year.

(c) Electric motor efficiency is 80 percent.

(d) Electric service is 3 phase, 230 volt.

(e) Power factor is 0.8.

(1) Calculate the 3-Phase Amps =

$$\frac{746\text{W/hp} \times \text{hp savings}}{\sqrt{3} \times \text{Volts} \times \text{Efficiency} \times \text{Power Factor}}$$

$$= \frac{746 \times (89\text{-}43)}{1.732 \times 230 \times 80/100 \times 0.80}$$

$$= 134.6 \text{ Amperes}$$

(2) Calculate the 3-Phased kW =

$$\frac{\text{Volts} \times \text{Amperes} \times \text{Power Factor} \times 1.732}{1000}$$

$$= \frac{230 \times 134.6 \times 0.80 \times 1.732}{1000}$$

$$= 42.9 \text{ kW}$$

(3) Cost savings =

$$42.9 \text{ kW} \times 8760 \text{ h} \times \frac{\$0.05}{\text{kWh}}$$

$$= \$18{,}790 \text{ per year}$$

5. ZONE PUMPING

5.01 When terminal balance valves are sharply set [paragraph 4.04(d)] to throttle the flow of water, the valve will absorb the excess head available at the pump with a resultant waste of pump power. For example, examine the single pump, 2-zone system in Fig. 7. At 70 percent efficiency:

$$\text{bhp} = \frac{2000 \text{ gpm} \times 60 \text{ feet}}{3960 \times 70\%/100} = 4.32$$

5.02 The head loss and capacity of each zone must equal the available head and capacity of the pump. Therefore, a balance valve is required in Zone 2 to limit the flow through this zone to 150 gallons per minute and increase the head 30 feet to balance the system. The power loss through a balance or throttling valve is determined from the same relationships as power input by a pump:

$$\text{bhp (loss through valve)} = \frac{\text{gpm} \times \text{Head (feet)}}{3960 \times \text{Efficiency}}$$

for this example:

$$\text{bhp loss} = \frac{150 \times 30}{3960 \times 70\%/100}$$

$$= 1.62$$

This power loss can be avoided by zone pumping with individual zone pumps. This will simplify the circuit balancing. The head loss common to all zone pumps must be considered because as it varies, so does the flow in each loop. For example, the common head loss to each zone in Fig. 8 is the loss in line A-B or 10 feet (Fig. 8).

5.03 The brake horsepower requirements for this system are:

$$\textit{Zone 1:}\quad \text{bhp} = \frac{50 \times 60}{3960 \times 70/100} = 1.08$$

$$\textit{Zone 2:}\quad \text{bhp} = \frac{150 \times 30}{3960 \times 70/100} = 1.62$$

The total bhp = 2.70 or a savings of (4.32 − 2.70) = 1.62 bhp

6. REPLACEMENT OF 3-WAY VALVES WITH 2-WAY VALVES IN A CONSTANT PUMPING SYSTEM

6.01 Replacing the 3-way diverting valve with a 2-way valve in a pumping system will reduce actual pumping energy. A 2-way valve system results in variable flow.

6.02 In a building system where the loading in cooling or heating water varies throughout its season so that the diversity between the mean operating point and the maximum design point is significant, the 2-way valve system will save energy. Therefore, a typical 2-way valve system with constant speed pump reduces power over a 3-way valve system by 24 to 612 percent.

6.03 In building systems with low load diversity, for example, high core loads such as computer centers and where heating and cooling loads are relatively constant, it will probably not be cost effective to convert to 2-way valve system.

6.04 Figure 9 represents a typical hydronic system using 3-way diverting valves and a constant speed pump. As the terminal unit is satisfied, the 3-way valve modulates to allow the supply water to divert around the unit. This diverted water consitutes wasted pumped energy.

6.05 Figure 10 refers to a typical 2-way valve system with a constant-speed pump system. When the terminal unit is satisfied, the 2-way valve closes. As the total flow of the system is decreased, the actual head pressure imposed on the pump is increased, thereby decreasing the horsepower.

6.06 In Fig. 11, assume that the pump has a 1-1/2 horsepower motor and 55 gallons per minute at 37 foot head with a 6-1/2 inch impeller. When the 2-way valve closes to restrict the flow, the pump responds by backing up on its curve to convert its flow energy to potential energy (head pressure). Point 1 is the system with flow 100 percent through all 2-way valves. At this point, the brake horsepower requirement is 1.3. When the flow is decreased to Point 2, the brake horsepower is 0.8. At Point 3, it is 0.65.

6.07 As the brake horsepower requirement is reduced, the power consumption of the pump motor is also reduced.

6.08 There are two reasons why the flow should not go to zero. First, the pump casing and impeller will not withstand the pressures involved; second, the heat exchanger device (such as a chiller) may require a minimum flow to maintain proper operation.

6.09 Two methods can be used to eliminate these problems. As shown in Fig. 10, a bypass valve with a flow meter measuring flow in the return pipe is used to maintain minimum flow through the heat exchange device. If a minimum flow is of no concern, then a bypass around the pump itself is required. The minimum flow rate through the bypass must be established by the pump manufacturer. Either one of these methods will ensure pump longevity and safety.

> ***WARNING: In a blocked or valved-off pump discharge, the motor horsepower is transferred into the water causing steam, and the steam pressure that is generated can cause the pump casing to explode!***

6.10 As the flow and friction loss decrease, the head developed by the pump increases. This results in increased pressure drops across the control valves, making control more dificult, and sometimes resulting in wire cutting of the valve seat.

6.11 To ensure that this does not happen, a differential pressure control with a pump discharge valve must be used (Fig. 10). This will ensure a constant pressure drop in the system and allow normal control action of the 2-way valves.

6.12 The graph in Fig. 12 illustrates the difference between 2- and 3-way valves. The important point to remember on this graph is the ***average*** load. This is the point where the system will normally operate. The connected load is the maximum design of the system with all systems working 100 percent. Normally, a system does not operate at 100 percent. For this example, the average load is two-thirds of the maximum connected load, and the minimum load is 25 percent of the design load. The ability of the system controls and pumping equipment to match the required water flow with actual flow will determine energy efficiency.

6.13 Referring to Fig. 12, the power consumption for the 3-way valve and constant volume system is described as the horizontal line at the top of the curve. The water flow is constant regardless of the building load, and there is no way to limit the pump horsepower.

6.14 Replacing the 3-way valves with 2-way valves will result in horsepower savings as shown in Fig. 12.

7. VARIABLE VOLUME PUMPING

7.01 A constant pumping, 2-way valve system can be improved upon to obtain additional energy savings through variable volume pumping with variable speed pumps.

7.02 A pump's energy usage is in direct ratio with the flow requirements. Therefore, a variable flow (variable speed) pump will save energy over a constant flow pump (Fig. 12).

7.03 In a variable speed system, a differential pressure control is used to maintain a constant pressure drop (head loss) across the terminal control valves. As this head loss increases (closing of valve), the pump speed and water flow will decrease resulting in reduced power requirements.

7.04 Consider the system depicted by the system and pump curves in Fig. 13. The flow requirements for this 2-way control valve, constant speed pump system vary from 1100 to 550 gallons per minute. As this flow rate is varied by the control valve, the pump horsepower requirements will vary from 40 brake horsepower at 1100 gallons per minute to 30 brake horsepower at 550 gallons per minutes.

7.05 There is a reduction in pump horsepower requirements of 10 brake horsepower for a constant speed, 2-way valve system when compared to a 3-way valve system. However, it is noted that 17 brake horsepower is absorbed by the control valve which represents wasted power. This wasted power can be eliminated by using a variable speed pump so that the pump curve nearly intersects the system curve at the minimum flow rate. Assuming that the pump will be controlled with a differential pressure controller to maintain a head loss of 20 feet across the control valve, the system curve is plotted on a family of pump speed curves. From Fig. 14, it is noted that the minimum flow rate for the systems can be met with the pump operating at 62 percent of full load speed. The pump power requirement is now 8 rather than 30 brake horsepower as previously seen.

7.06 The pump motor drive efficiency must be considered in determining energy reductions with variable speed pumps. The motor drives exhibit a decrease in drive efficiency at lower speeds (Fig. 15).

7.07 At 62 percent speed, the drive efficiency of the pump is only 40 percent; therefore, the motor must draw $\frac{8 \text{ bhp}}{40/100} \doteq 20$ horsepower to provide the power required at the pump.

$$20 \text{ hp} \times 0.746 \frac{\text{kW}}{\text{hp}} = 14.9 \text{ kW}$$

By comparison at the same flow rate of 550 gallons per minute, the constant speed pump requires 30 brake horsepower and at 70 percent efficiency:

$$\frac{30 \text{ bhp}}{70/100} = 42 \text{ hp}$$

$$42 \text{ hp} \times 0.746 \frac{\text{kW}}{\text{hp}} = 31.3 \text{ kW}$$

8. ANALYSIS OF A PUMPING SYSTEM

8.01 To arrive at the optimum pumping system(s) for a building, study the water needs of the system to ensure that controls and pumping equipment are the most efficient without undue equipment cost. We should:

(a) Develop the following loads.

(1) Maximum design (connected) load

(2) Minimum design load

(3) Annual average or mean operating load.

(b) Select several pump and piping sizes and control schemes for the system.

(c) Calculate a system head curve for each proposed system to obtain a profile of head and flow requirements.

(d) Plot pump horsepower curves for each of the schemes.

(e) Determine first costs for the various schemes.

(f) Evaluate the energy costs for each of the schemes.

(g) Select the most suitable pump scheme that meets the requirements.

9. BIBLIOGRAPHY

9.01 Information in this section is based on the following reference sources:

1. *ASHRAE Handbook & Product Directory,* 1973 Systems; Published by American Society of Heating, Refrigerating, and Air Conditioning Engineers, 1791 Tullie Circle, N.E., Atlanta, Georgia 30329.

2. Becker, H.P., "Energy Conservation Analysis of Pumping Systems," *ASHRAE Journal,* April, 1975.

3. Becker, H.P., "Energy Conservation Characteristics of Various Pumping Systems Compared With Variable-Speed Systems," *ASHRAE Journal,* June, 1974.

4. Bell & Gosset, *Pumping Design Manuals,* Training and Education Dept, Fluids Handling Division, 8200 N. Austin Avenue, Morton Grove, Illinois 60053.

5. Endejaun, R.H., "Pumping System Design," *ASHRAE Journal,* June, 1974.

6. Richel, J.B., "Energy Conservation in Heating and Air Conditioning Pumping Systems," *ASHRAE Journal,* June, 1974.

7. *Handbook of Energy Conservation for Mechanical Systems in Buildings,* Van Nostrand Reinhold Company, New York, 1978.

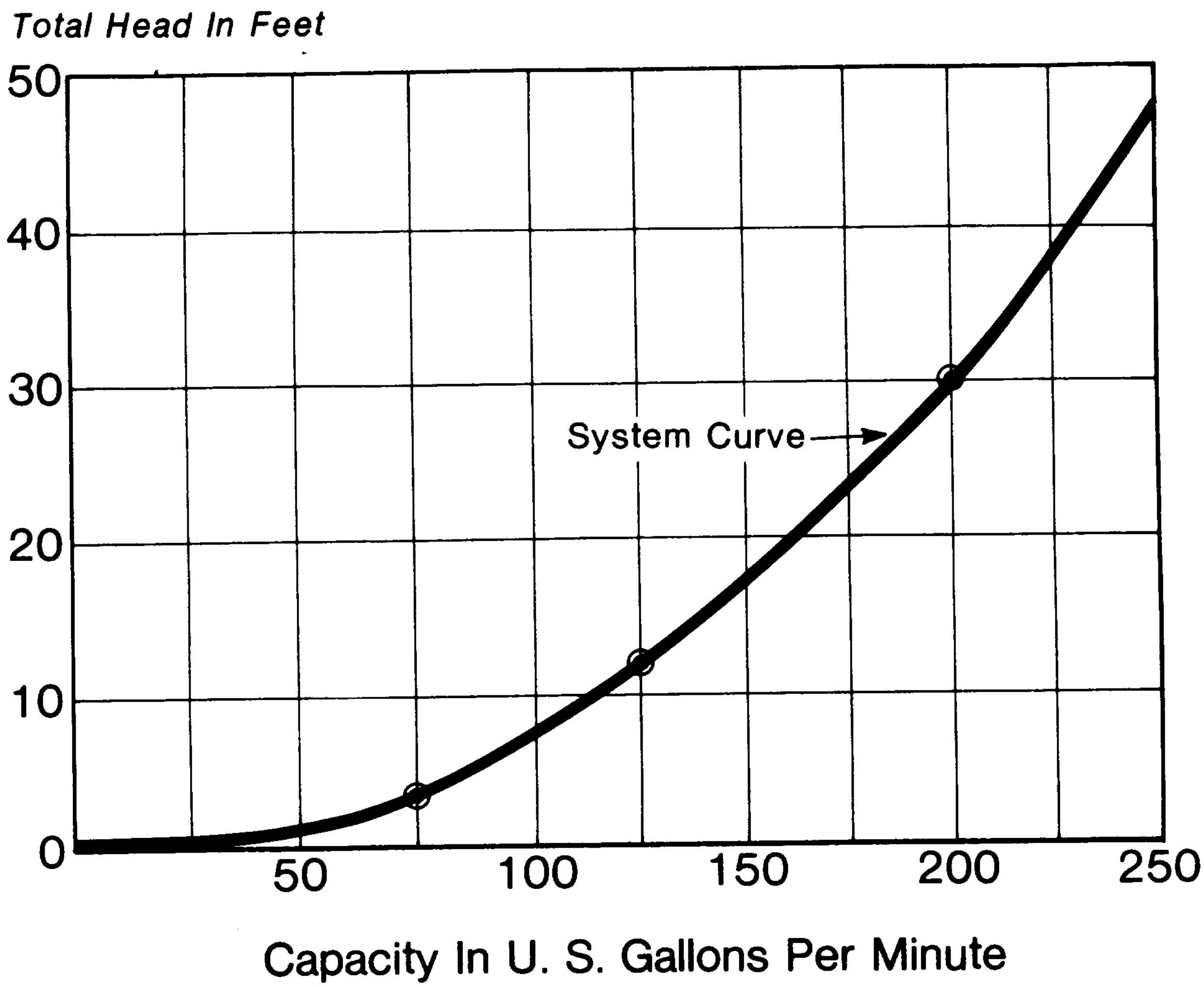

Fig. 1—Typical System Characteristic Curve

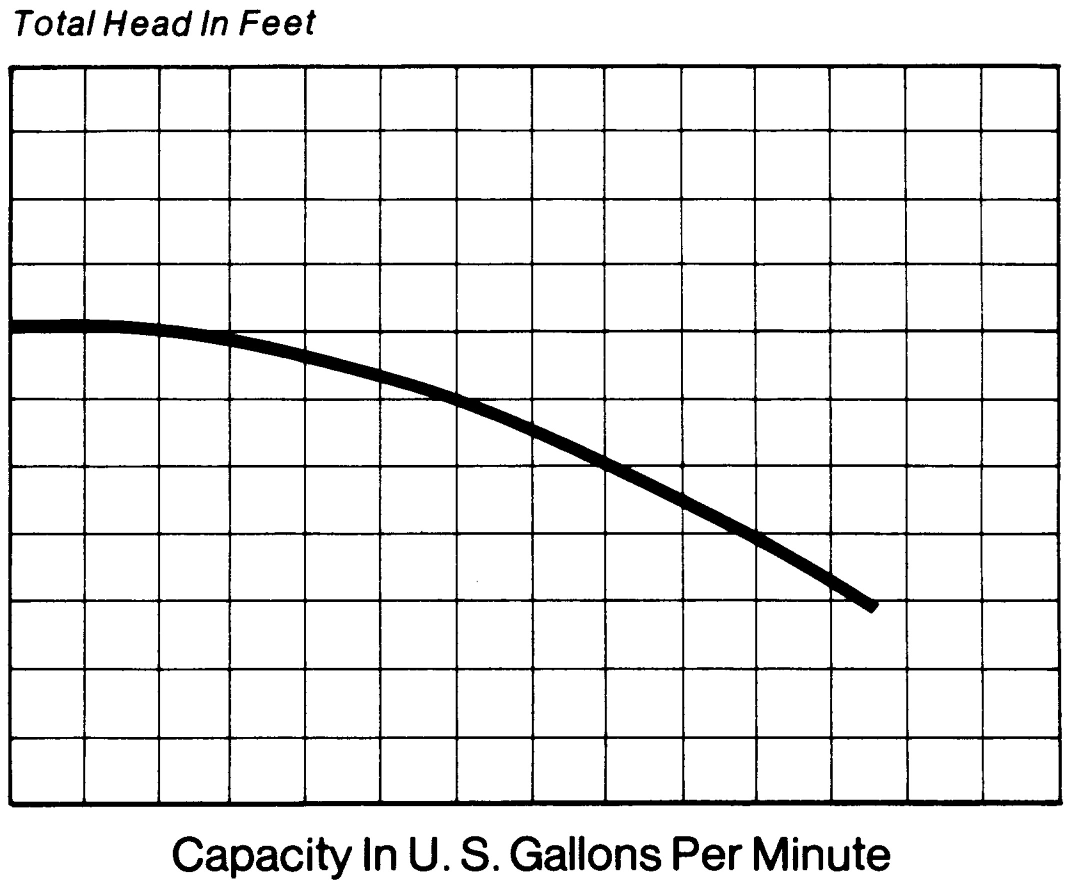

Fig. 2—Typical Pump-Head Capacity Curve

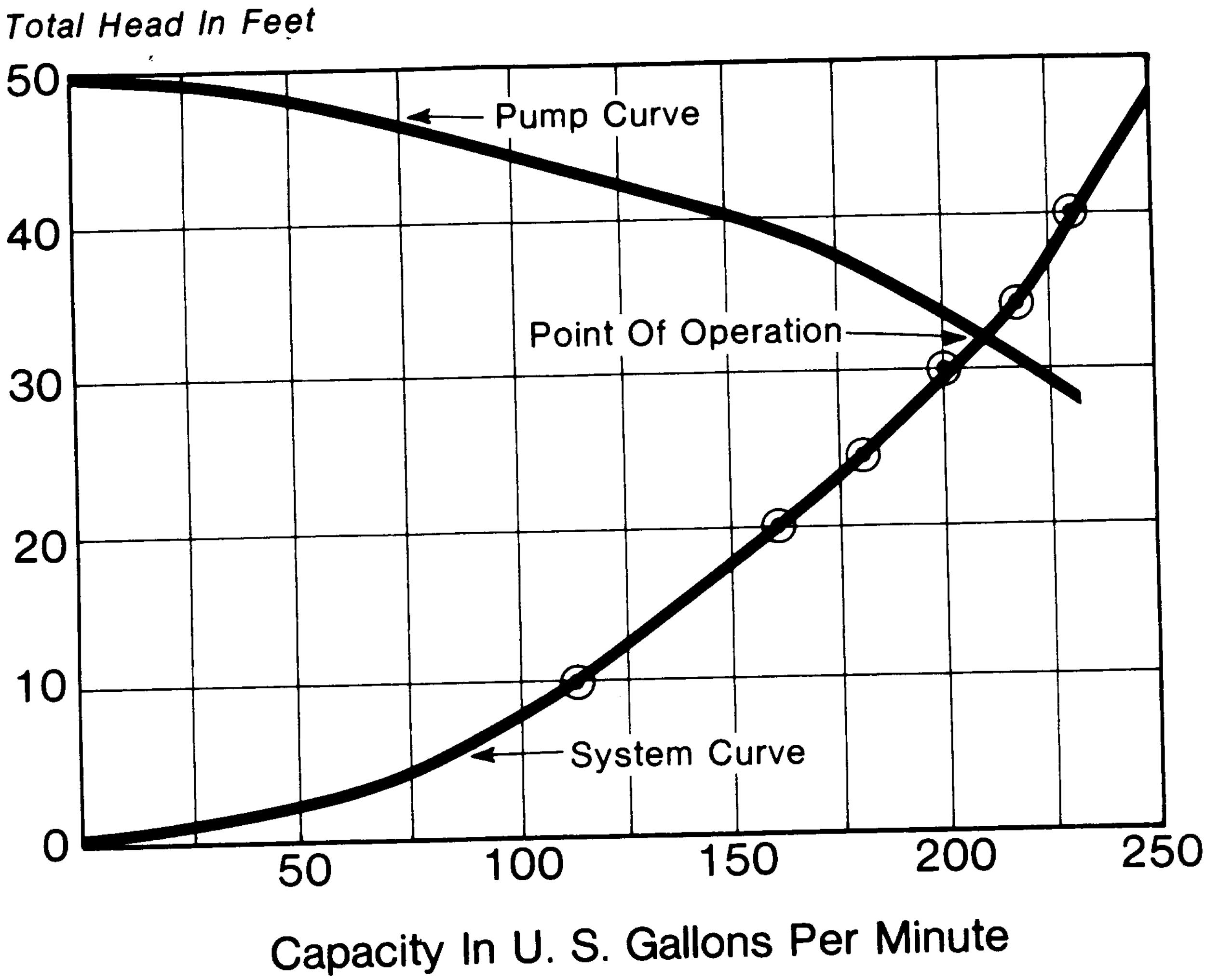

Fig. 3—Sytem Curve Plotted on Pump Curve

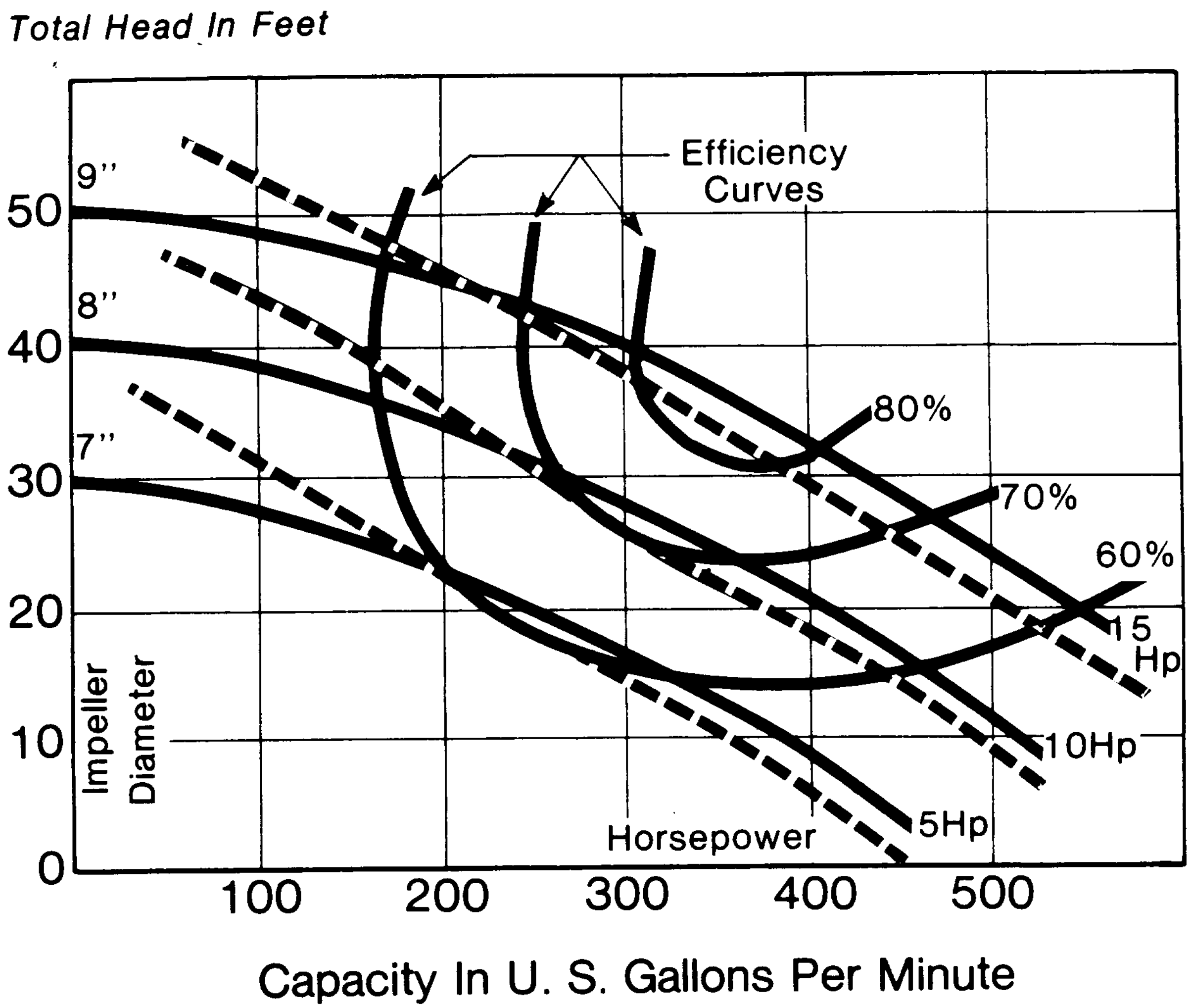

Fig. 4—Pump Efficiency Curves

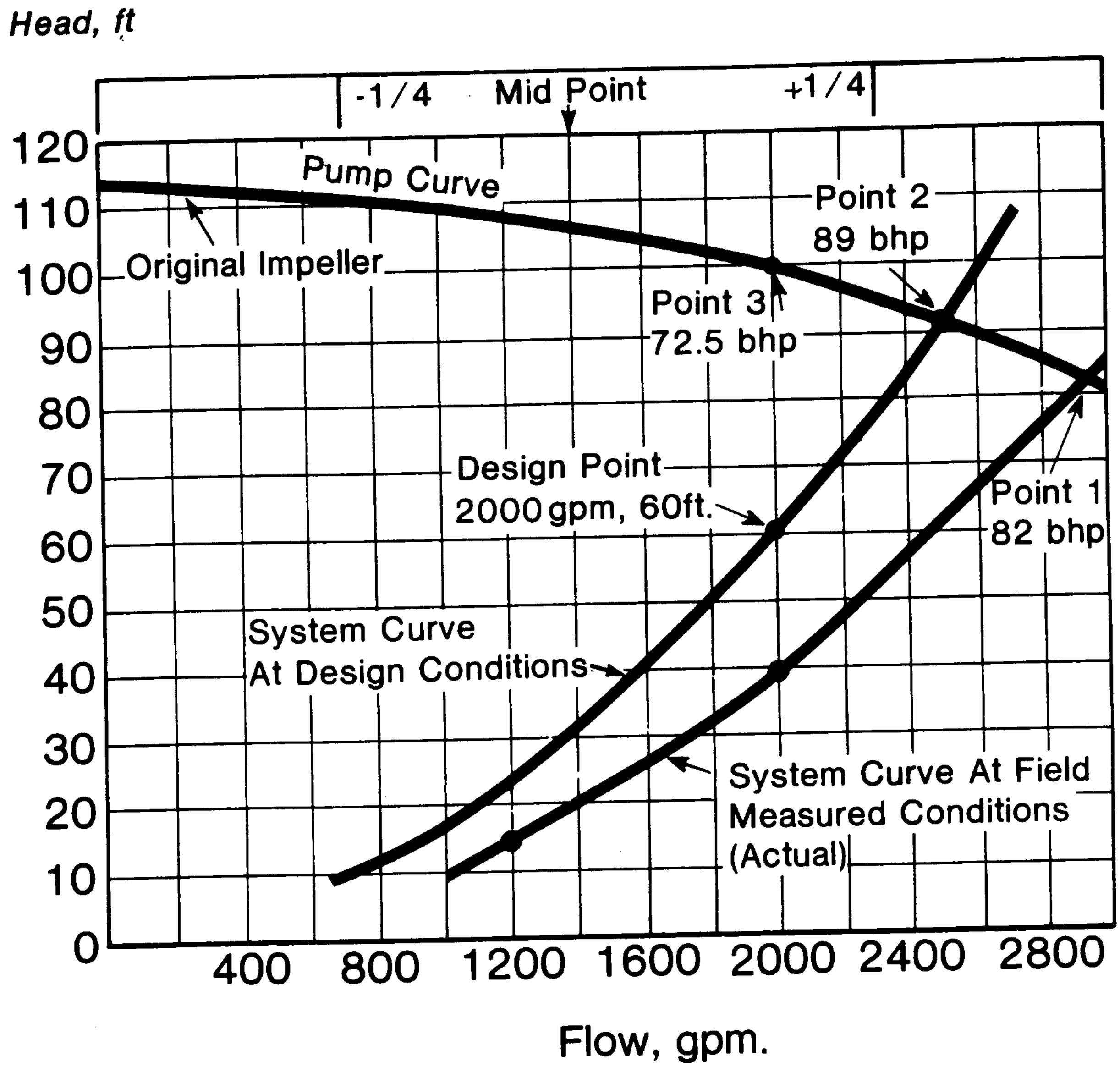

Fig. 5—Design System Curve, Actual System Curve, and Pump Curve of Installed Pump

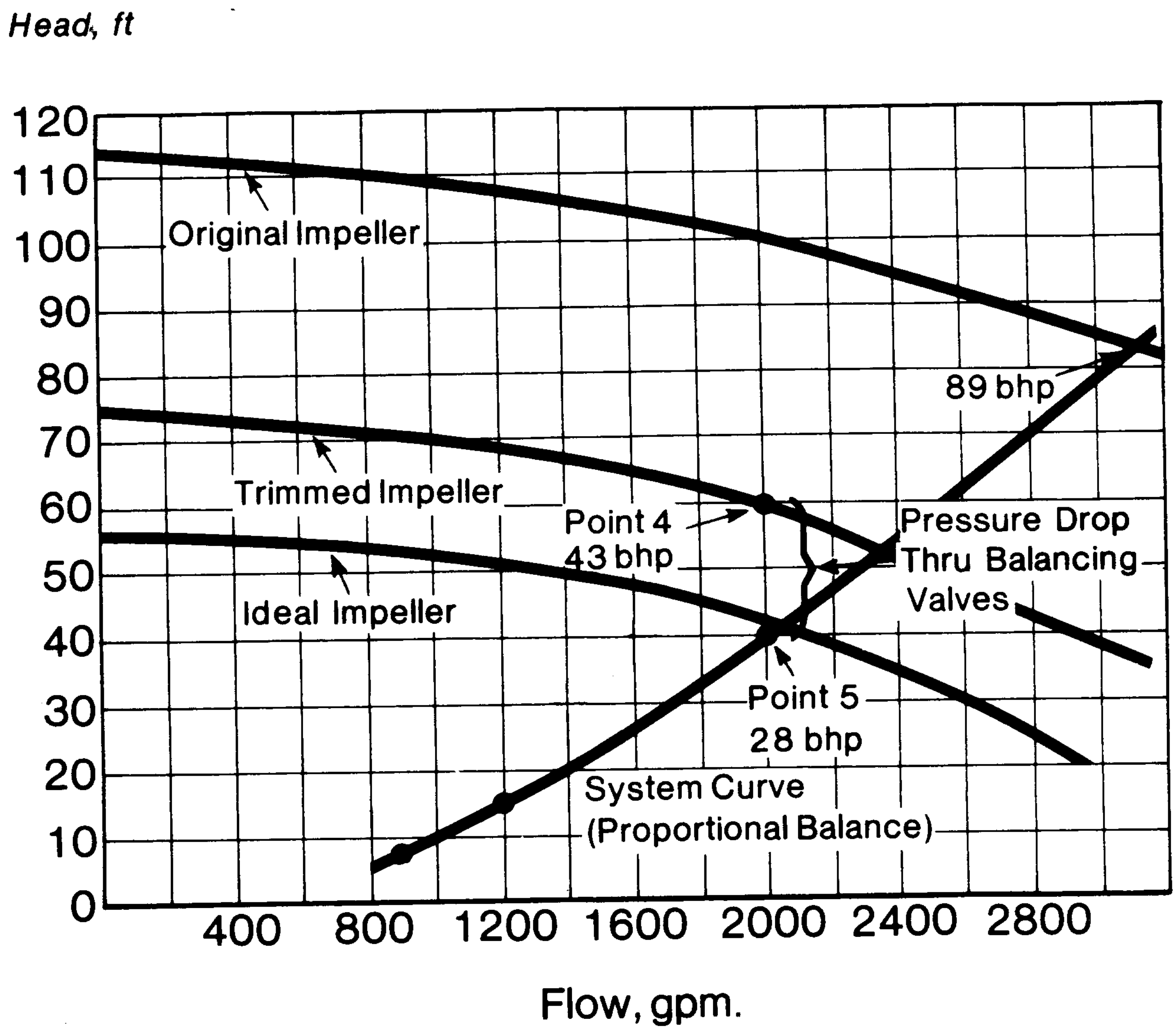

Fig. 6—Pump Curve for Proportional Balanced System and Trimmed Pump Impeller

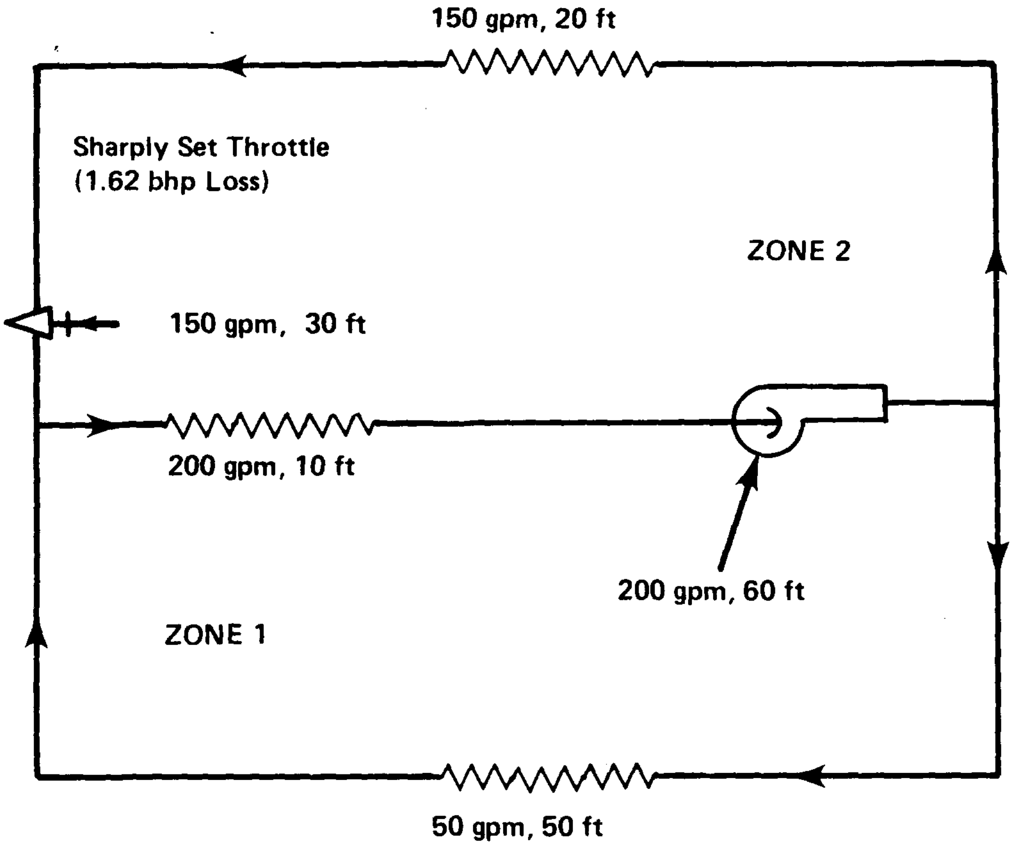

Fig. 7—Single Pump 2-Zone System

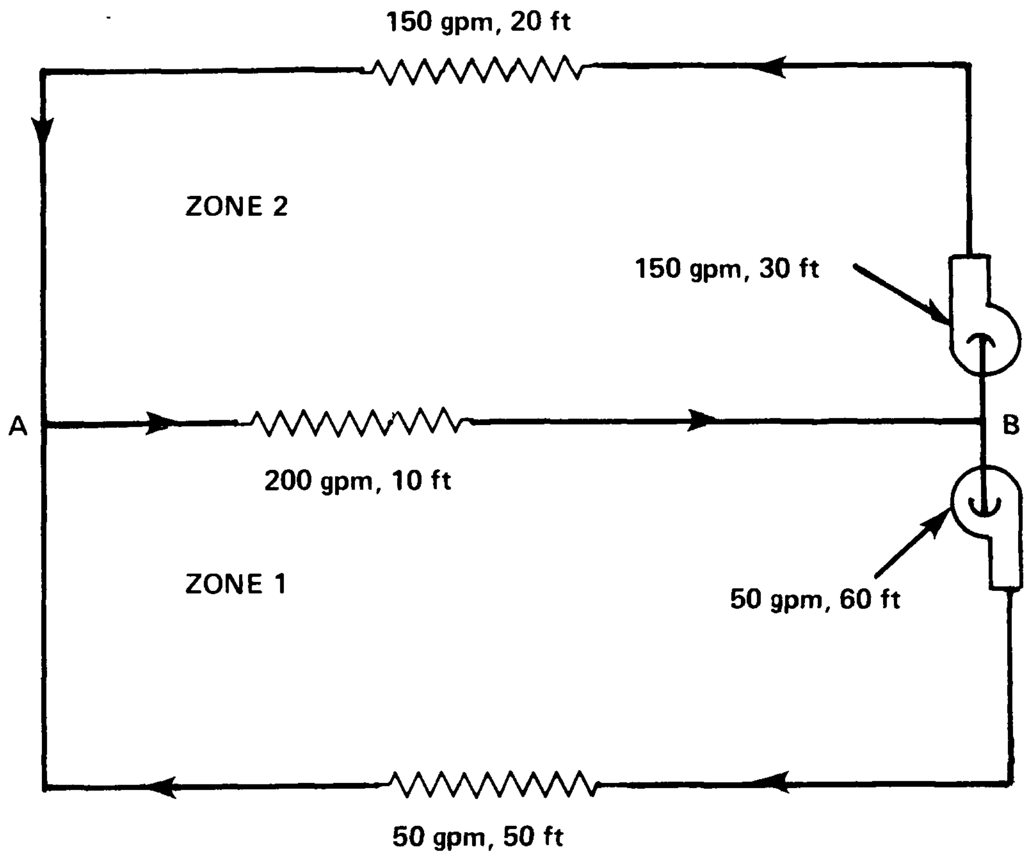

Fig. 8—Two-Zone System With Zone Pumps

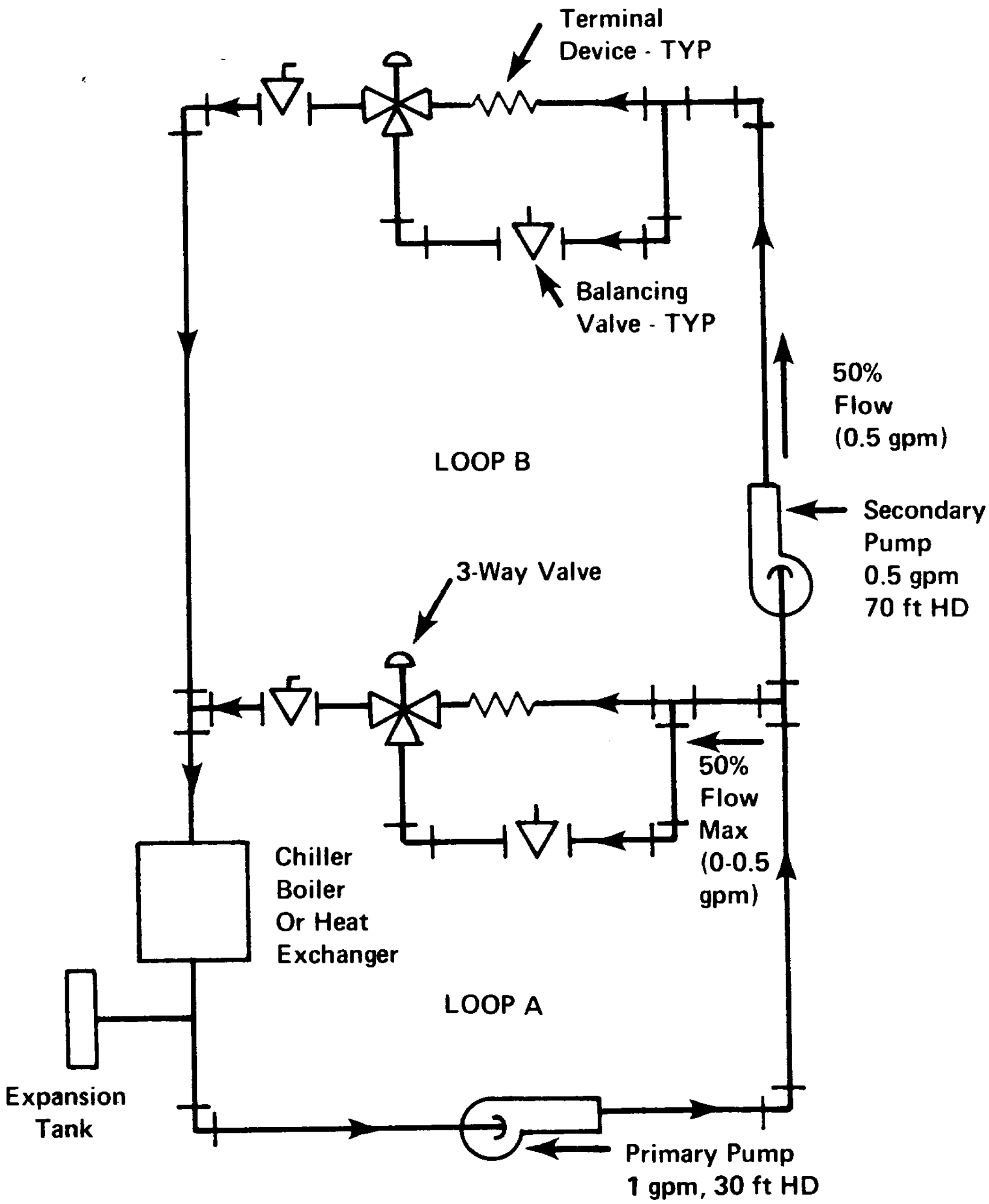

Fig. 9—Constant Speed Pumps With 3-Way Diverting Valves

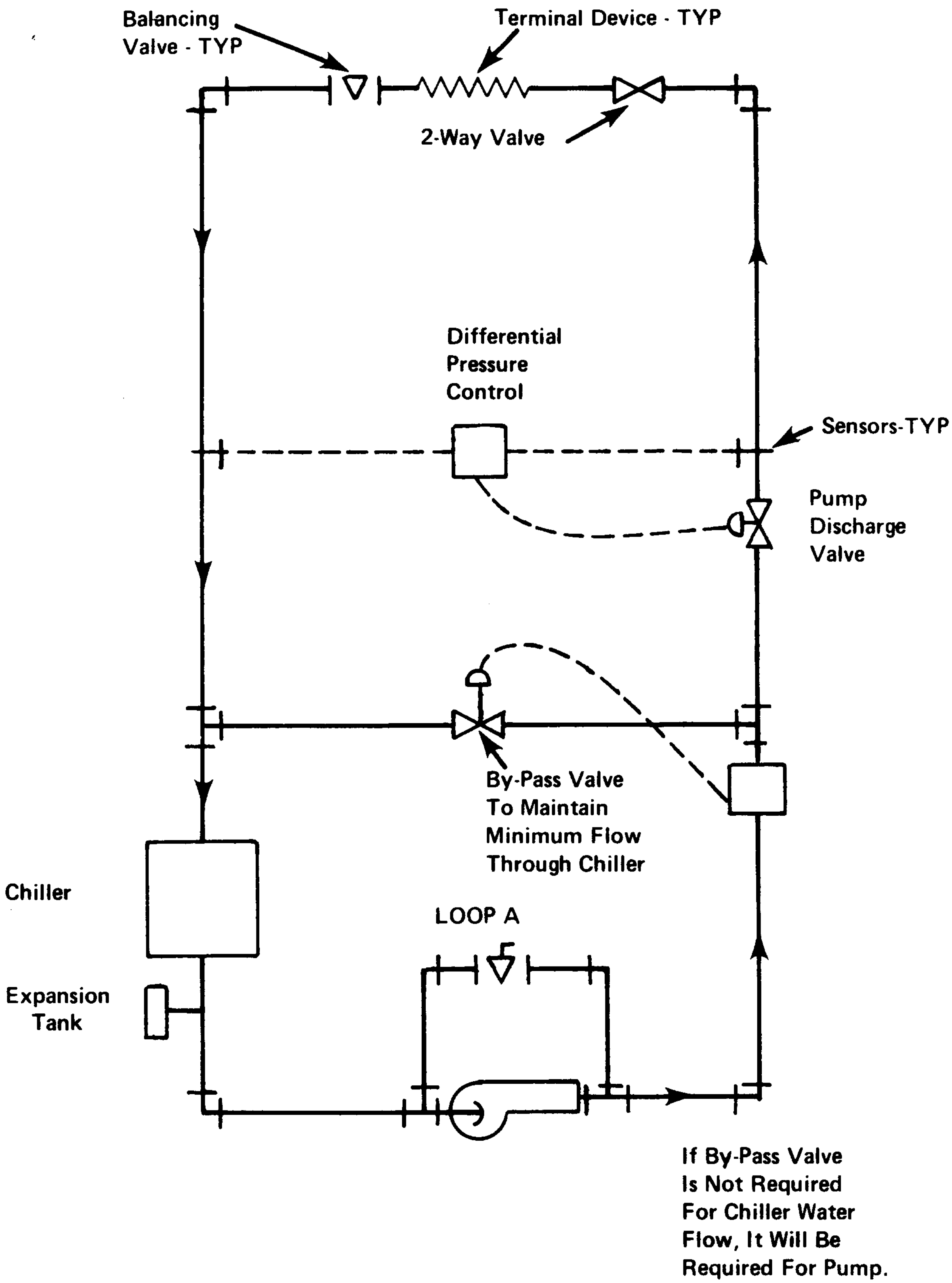

Fig. 10—Constant Speed Pump With 2-Way Valves

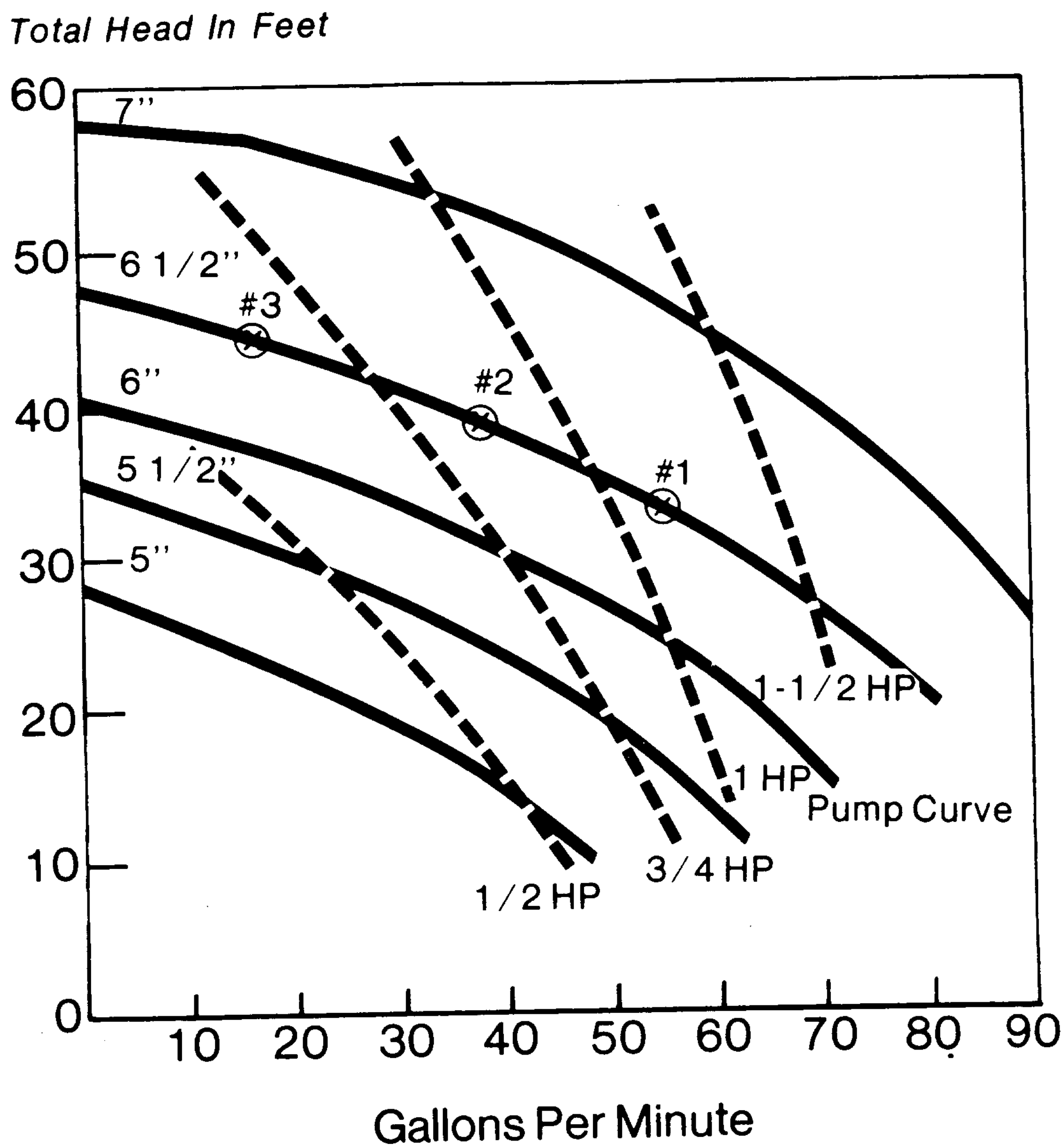

Fig. 11—Typical Pump Performance Curves

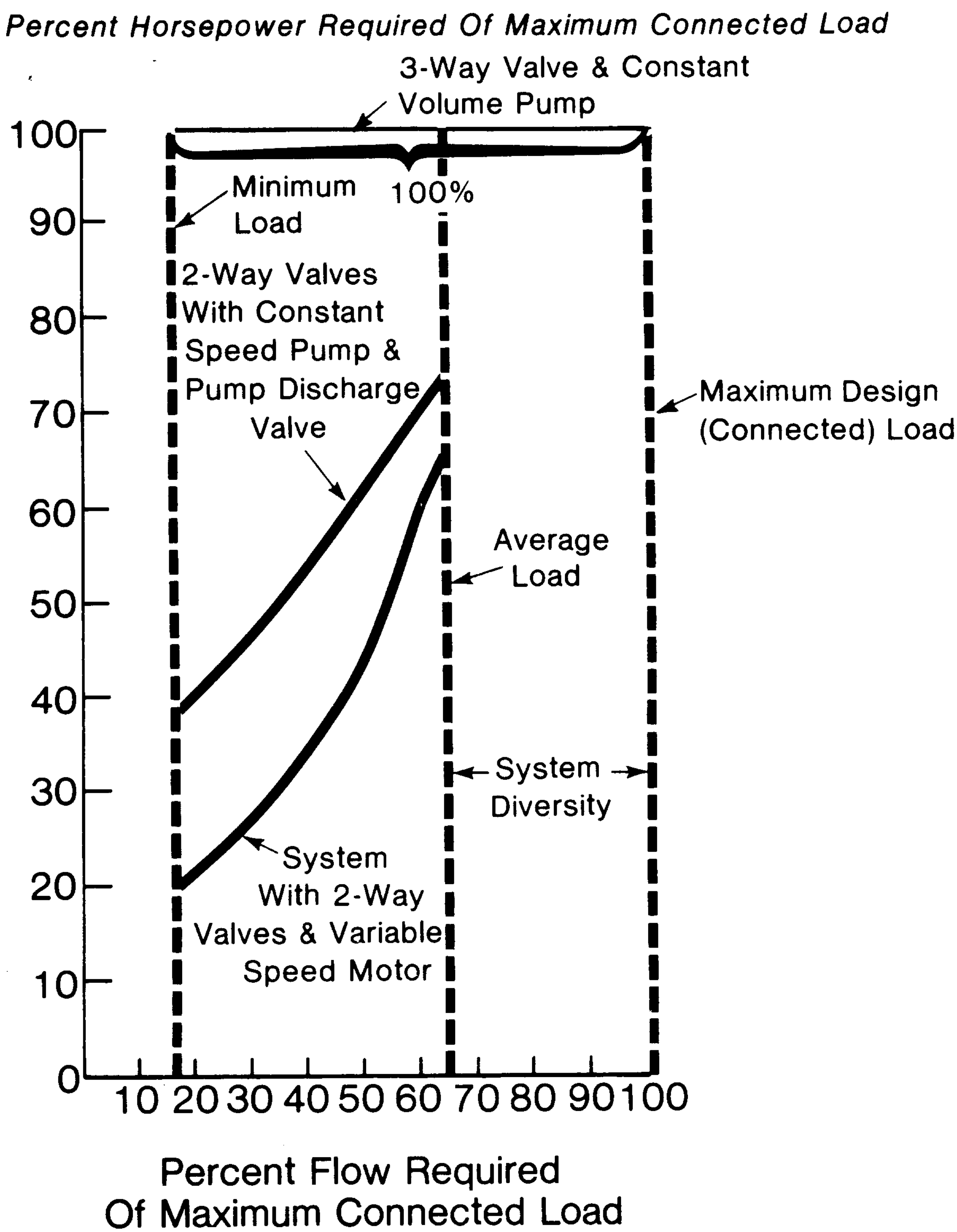

Fig. 12—Power Savings of a 2-Way Valve System

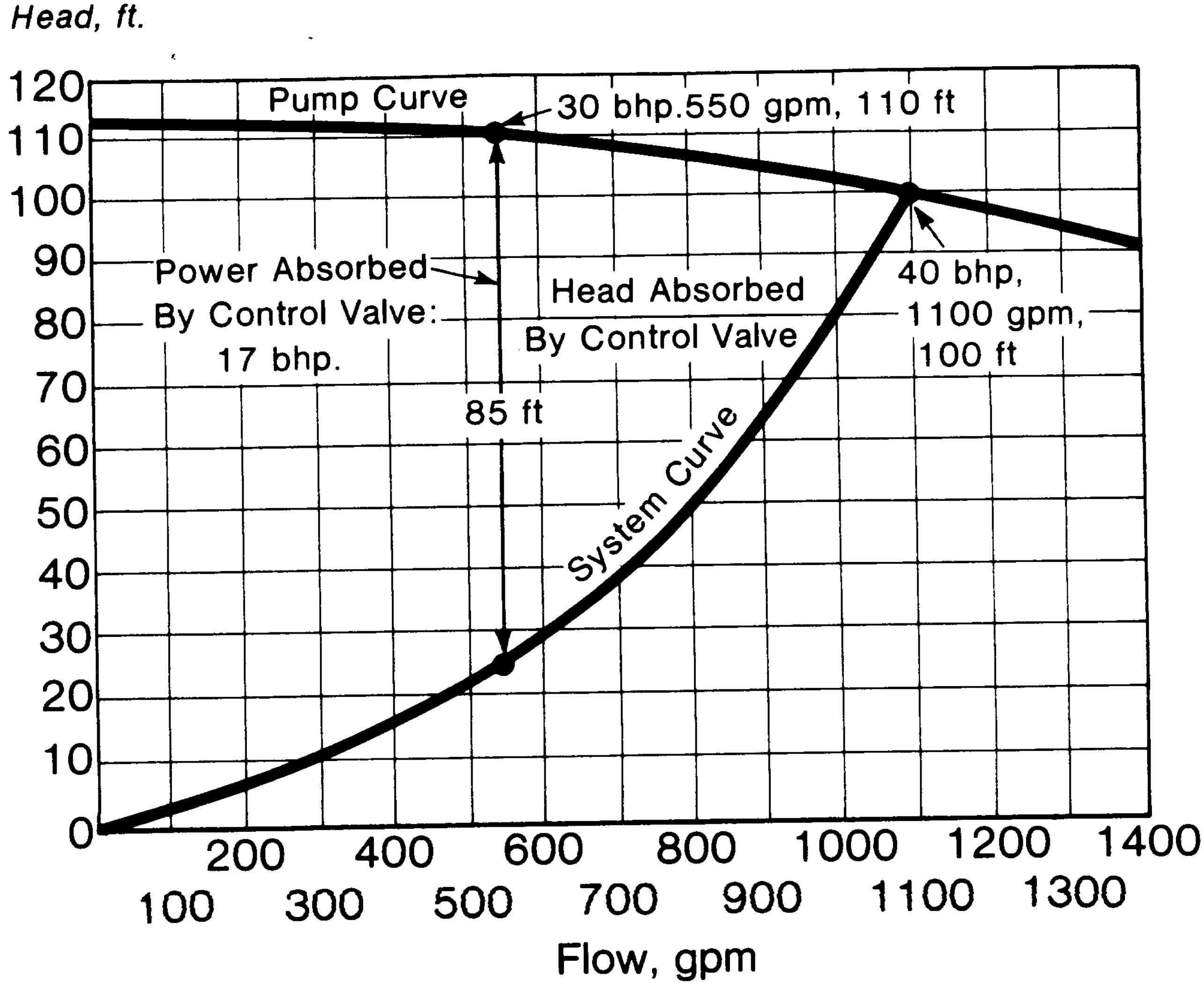

Fig. 13—Pump Power Requirements With Varying Flow

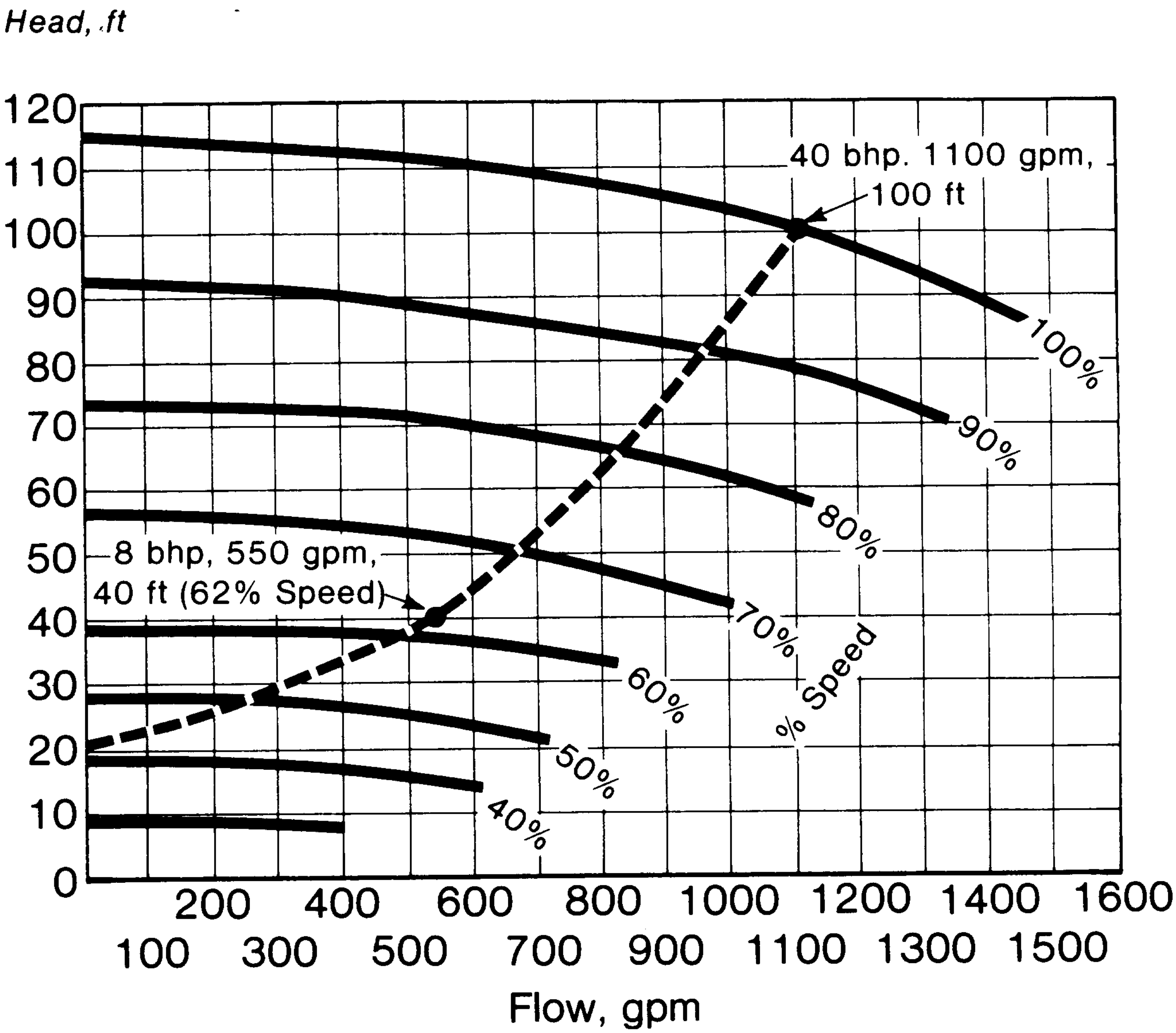

Fig. 14—Pump Curves For Various Speeds

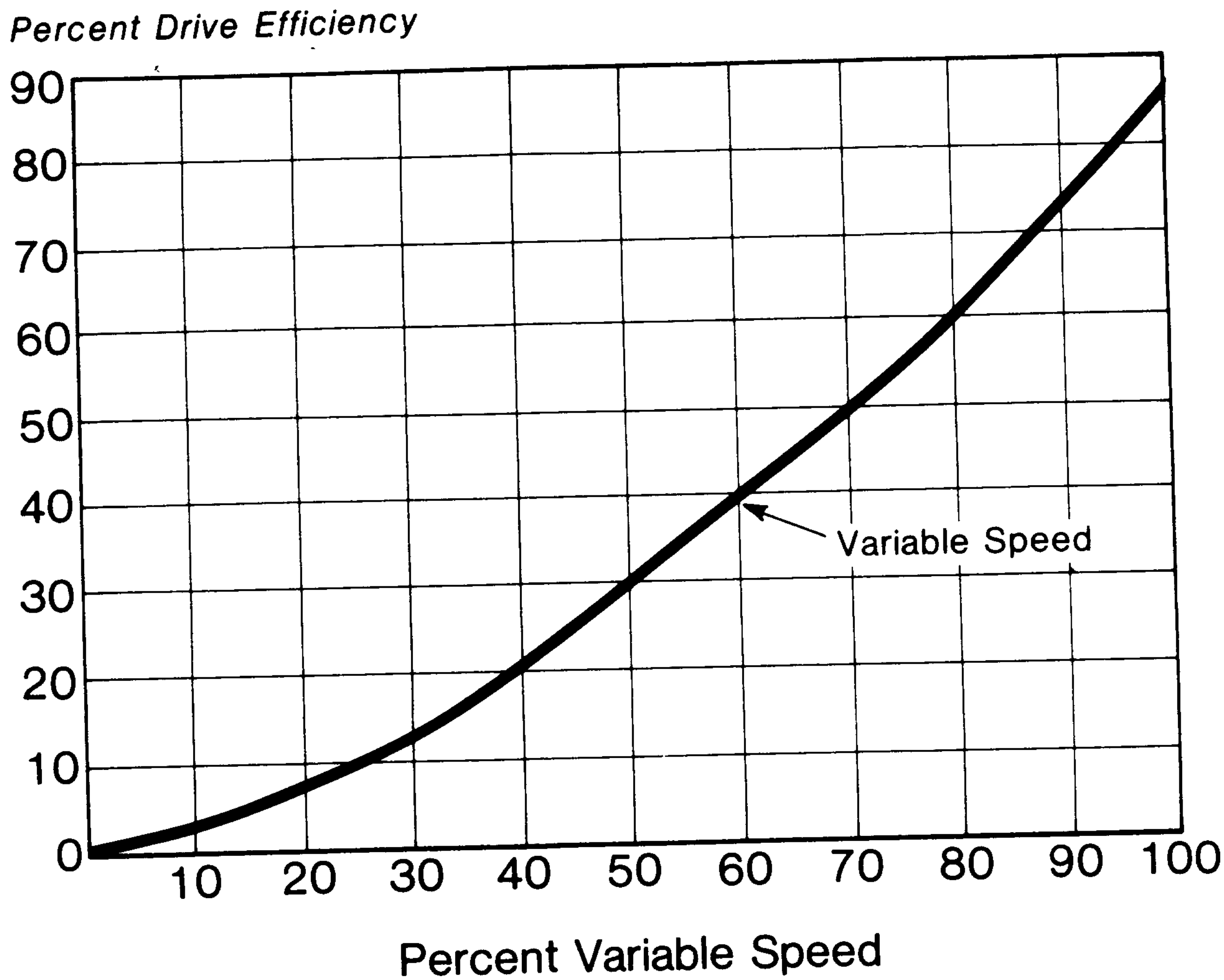

Fig. 15—Variable Speed Drive Efficiencies