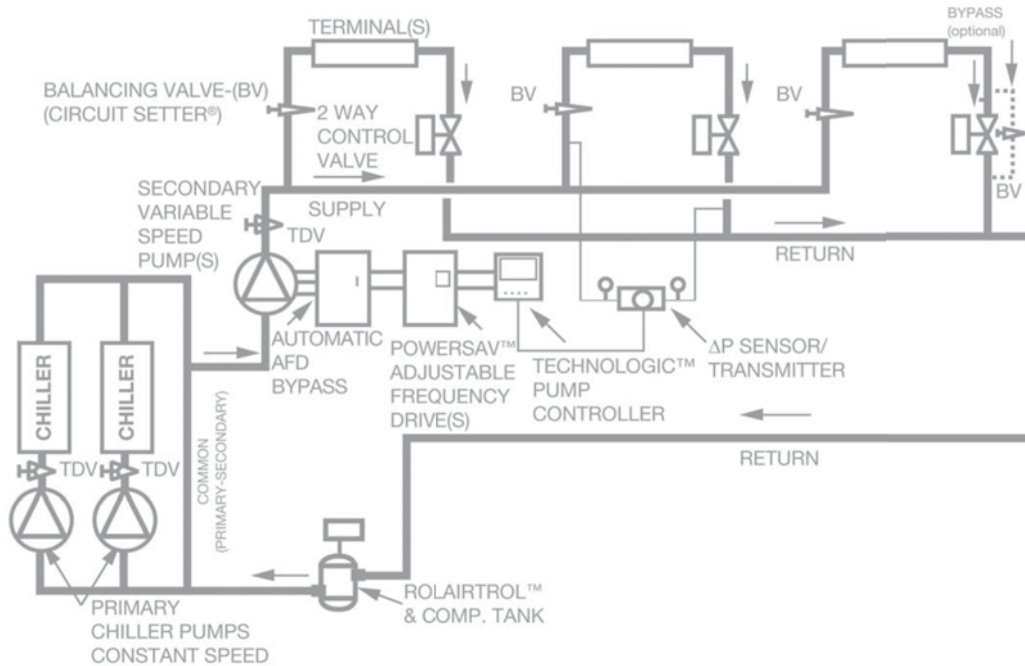


VARIABLE SPEED CHILLED WATER PUMPING CONTROL  
TWO-WAY VALVE REVERSE RETURN PIPING SYSTEM



# Variable Speed/Variable Volume

## PUMPING FUNDAMENTALS

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## INTRODUCTION

The present trend towards using variable speed (V/S) pumping systems is based on proven engineering economics since operating costs for constant speed (C/S) pumps have risen dramatically. Table I shows the cost of C/S operation per HP for various time periods, using an 85% efficient motor at specific \$/KWH utility rates.

TABLE I  
PUMPING COST PER HORSEPOWER WITH 85%  
MOTOR EFFICIENCY

	\$/KWH	.01	.02	.04	.06	.08	.10	.12	.14	.16	.18	.20
OPERATING TIME	1 Hr.	\$ .0088	.0176	.0352	.053	.07	.088	.105	.123	.141	.158	.176
	24 Hr.	\$ .21	.42	.84	1.26	1.68	2.10	2.52	2.94	3.36	3.78	4.20
	30 Day (One Month)	\$6.30	12.60	25.20	37.80	50.40	63	75.60	88.20	100.80	113.40	126
	1 Yr.	\$ 77	154	308	464	613	770	920	1078	1235	1385	1540

TABLE I:            C/S Pumping Cost/HP Based on 85% Motor Efficiency

A similar table was published in 1971, when it was included in an article prepared at Bell & Gossett. The article was entitled "Pump Energy Saving Basics", and is available from Bell & Gossett Representatives. The pump energy saving ideas expressed in this article are still valid, and can provide valuable additional information to the design engineer interested in pump energy saving. At that time (1971) utility rates averaged about \$.015/KWH, and the Table extended only to \$.04/KWH. It was considered inconceivable that rates could increase beyond \$.04/KWH at that time.

At the present time (1985) national commercial utility rates average about \$.065/KWH (\$500/Yr./HP), and reach \$.14/KWH in some areas (about \$1,000/Yr./HP). Further, utility rates are expected to double by 1992.

The relentless rise in electric utility rates, combined with reduced costs for variable frequency (variable speed) drive have resulted in attractive payback periods for both new and retrofit V/S installations. It should be noted, however, that the variable speed pump is not a cost saving panacea.

A meaningful energy savings payback with Variable Speed Systems is only possible if the designer has followed time tested design fundamentals in the application of a V/S pumping system. Therefore, it's the purpose of this discussion to re-emphasize those fundamentals, outline logical application areas and to provide specific design and control fundamentals for maximizing energy conservation.

1. HISTORIC CONSTANT SPEED (C/S) PUMPING PROBLEMS AS RELATED TO VARIABLE SPEED (V/S) PUMPING

Prior to examining variable speed (V/S) application fundamentals, it will be of value to review past (C/S) pumping problems and their solutions in terms of impact on variable speed pumping applications.

For example, two way control valves were applied to large heating systems many years ago. The application lead to trouble because of control problems. This control problem is illustrated in Figure 1.

In the example, two way control valves were selected for a 20 ft. head loss at design flow. Aside from the control valves, head loss for the remainder of the system (equipment room, piping, terminals, fittings, etc.) is established at 80 ft. head loss for design flow. As the two way valves modulate and reduce terminal flow in accordance with load demand, head loss in the piping system is reduced. The head loss reduction is a squared function of flow change and is illustrated by the flow head loss curve in Figure 1.

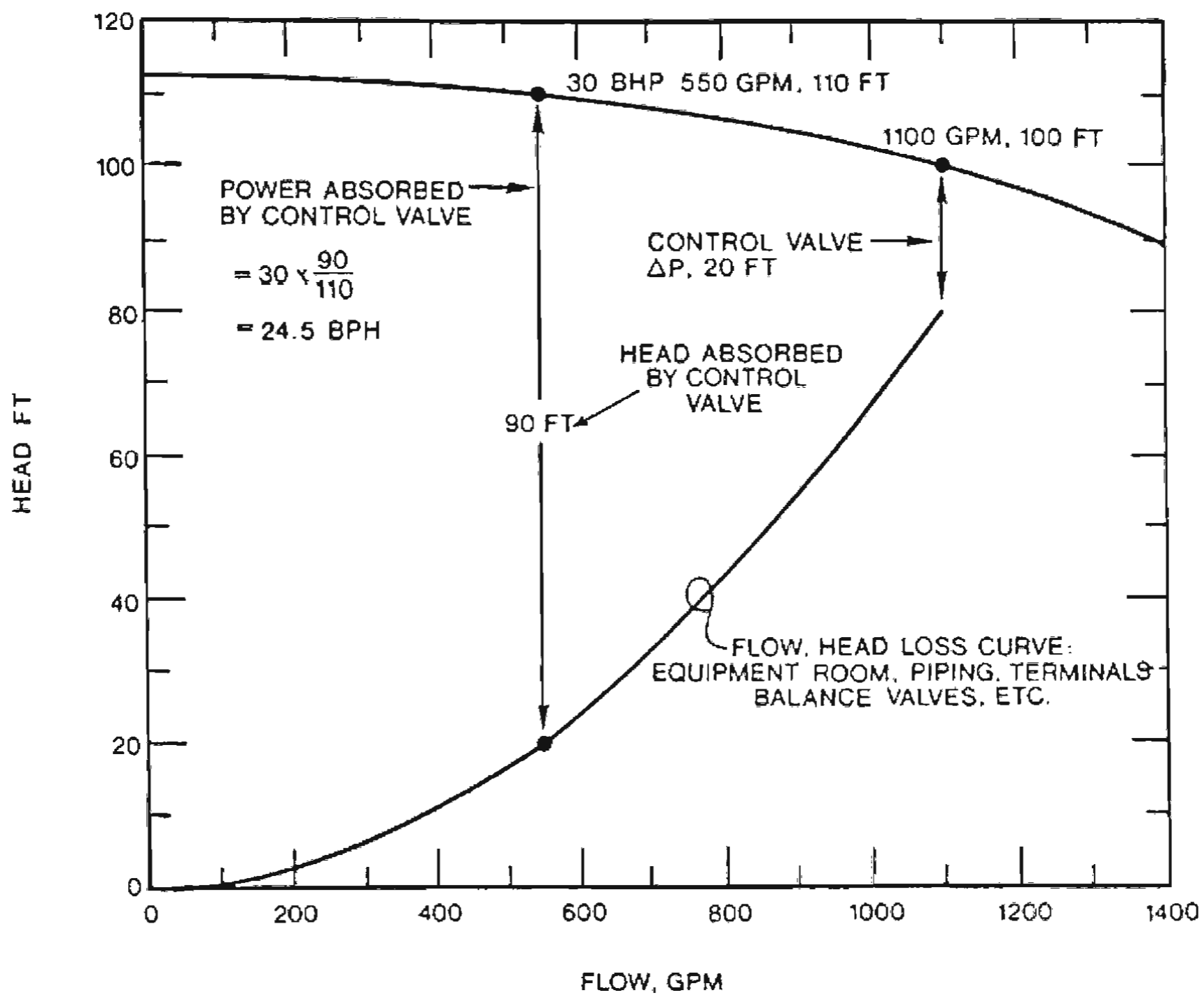


FIGURE 1. Control Valve Differential Head Increases with Flow Decreases When C/S Pump is Used

The principle problem is that constant speed pump head increases with flow reduction at the same time that piping head loss decreases. The control valve is now the victim of an increased differential head caused by increased pump head and decreased piping head loss. This is illustrated in Figure 1., where control valve differential head has increased from 20 ft. (full system design flow) to 90 ft. at half flow.

The increase in differential head can modify control valve characteristics and may cause loss of all control because the valve may be forced open during light load operation. This problem is further aggravated by overheaded, safety factored pump head selection, and by very low control valve head loss specification.

The solution used many years ago to solve this problem was "reset". When reset is applied to a hot water system, its water temperature is changed (reset) in accordance with changes in outside air temperature so that as outdoor temperature increases, hot water temperature decreases. When properly applied, reset establishes a constant system flow. The control valve then simply performs a "topping off" control function. This solves the problem. Conventional "Constant Flow" reset is shown in Figure 2.

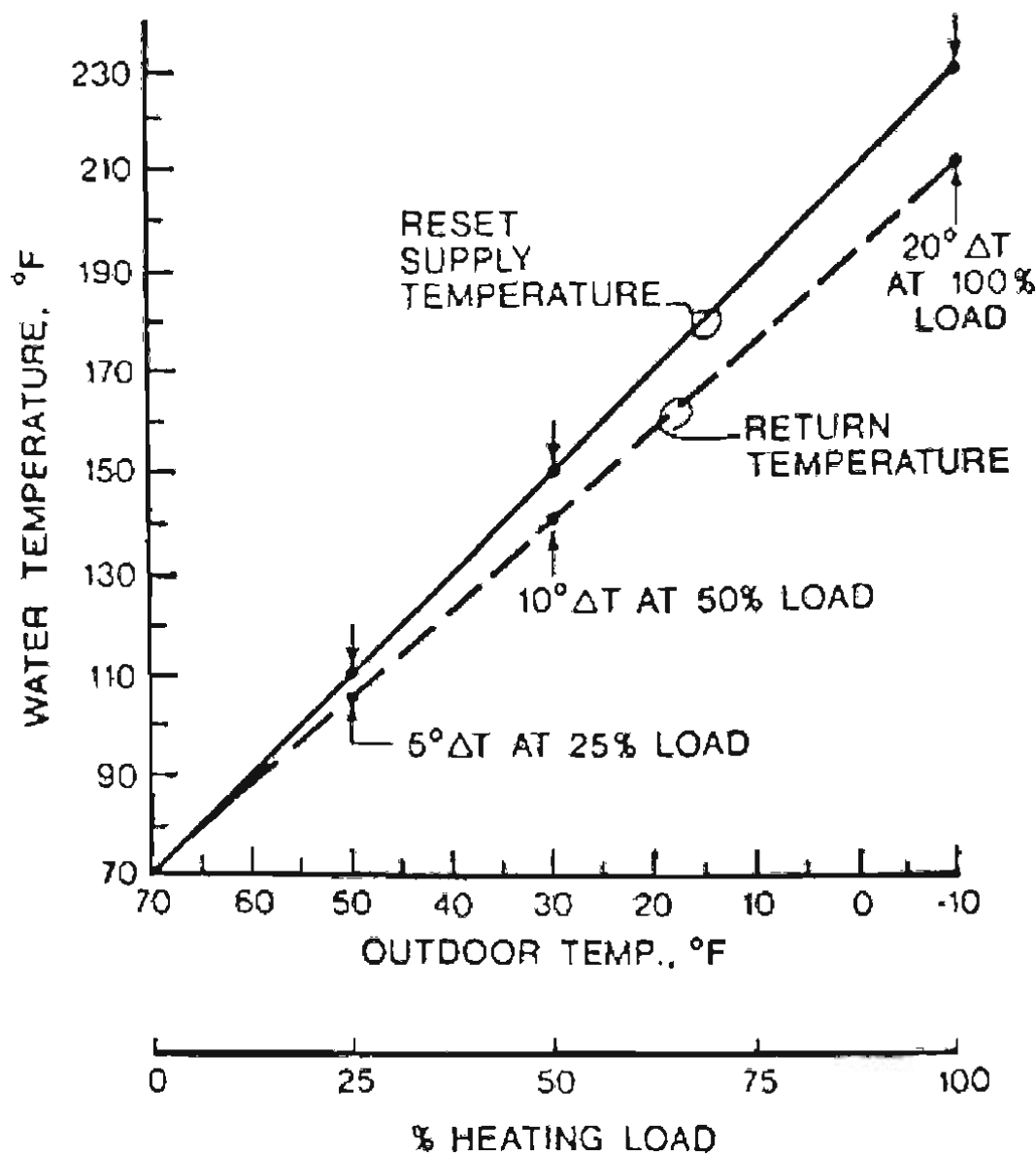


FIGURE 2. Conventional Reset Schedule Provides Constant Flow Because Temperature Drop ( $\Delta T$ ) Varies Directly with Load

The "reset" reference to variable speed pumping should be clear. Conventional reset should not be applied to variable volume (variable speed) systems because it leads to constant flow; the anti-thesis of pump energy conservation.

While conventional reset should not be used, a "modified" reset directed to variable volume flow should be applied to hot water systems. This is because reset by boiler temperature change (not conventional 3 way mix with constant boiler temperature) can be used to improve seasonal boiler operating efficiency. This subject is discussed under heading 14: V/S Application to Hot Water Heating Systems.

The basic two way control valve pressure differential increase problem occurred again when large chilled water systems were introduced. The solution to this problem was different, because chilled water systems cannot be reset sufficiently to overcome the problem. One solution for chilled water was to employ three way bypass control valves at the terminal units. This eliminates the control valve pressure rise problem, but again leads to constant flow.

A chilled water system using three way valves may carry a "double whammy" in terms of pump power wastage. This is because many three way control valves are not balanced at the bypass. The resulting major flow increases (pump power draw increase) during light load operation can be substantial!. The typical three way controlled chilled water system is shown in Figure 3.

Another solution, widely used for three pipe systems, was the "choke" valve applied to the C/S pump discharge. Use of the choke valve is described in the article: "Variable Volume Pumping Considerations", prepared by Bell & Gossett, and available from Bell & Gossett Representatives.

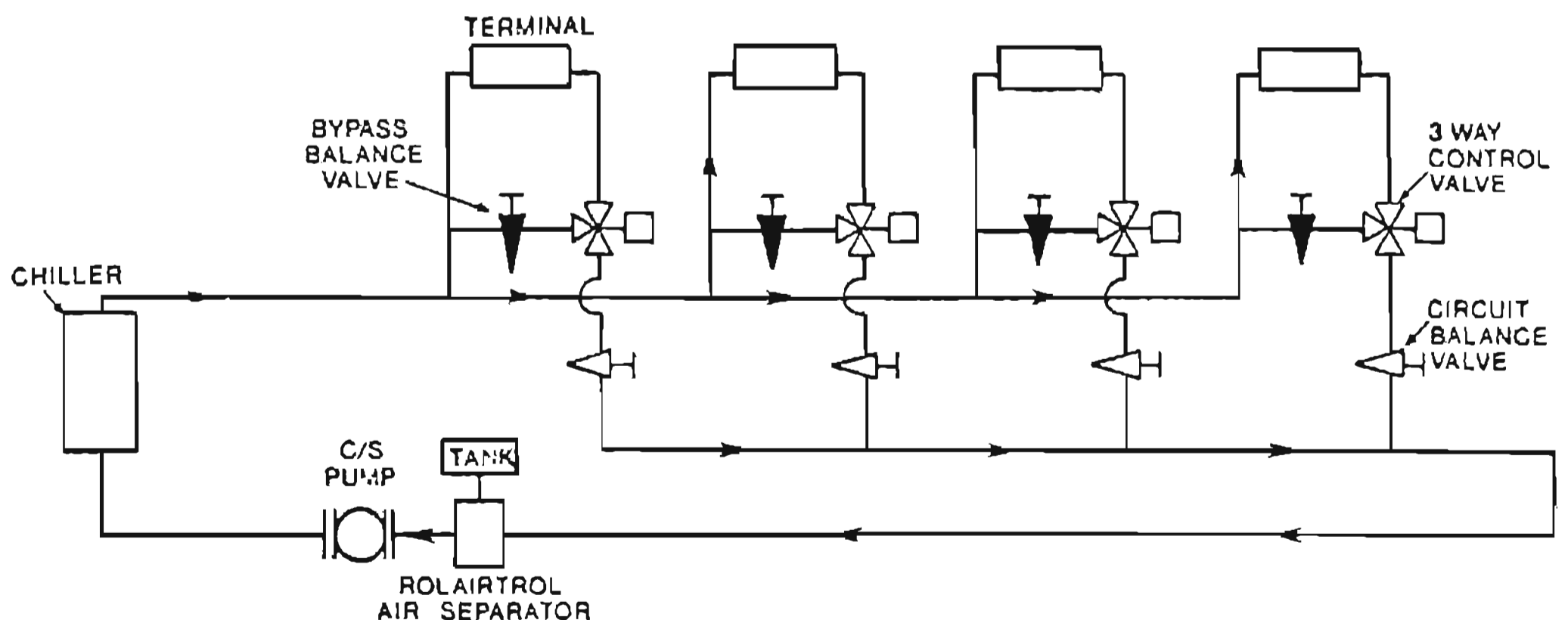


FIGURE 3. Existing Chilled Water System Using Three Way Control Valves; Results in Pump Power Wastage



It should be emphasized that with the application of variable speed pumping systems, the fundamental reason for three way valve control is eliminated. This is because the variable speed pump can control the differential pressure increase across two way control valves, while primary-secondary pumped chillers overcomes the often undesirable effects of variable flow through the chillers.

Existing pump energy wasting three way valve controlled systems can be changed to modified two way valve control with V/S pumps. Pump energy savings are substantial. Necessary modifications are described under heading 15; Conversion of Existing C/S Three Way Valve Systems to V/S.

A cautionary note about V/S pumping should be kept in mind regarding upward reset of the chiller leaving water temperature during light load. It is well known that chilled water reset reduces chiller KW/Ton draw. But, reset also increases terminal unit flow need and works against the system flow reduction needed for maximized pump energy conservation. The chiller power reduction from chiller reset should be weighed against the pump power increase caused by the resultant "reset" flow increase on V/S systems.

## 2. GENERALIZED POTENTIAL FOR PUMP POWER REDUCTION WITH VARIABLE SPEED (V/S) PUMPING

The basic pump shaft HP draw formula is shown below for water.

$$\text{Pump Shaft HP Draw} = \frac{\text{Pump Flow (GPM)} \times \text{Pump Head (Ft.)}}{3960 \times \text{Pump Efficiency}}$$

Pump flow and head must exactly match piping system flow and head.

For any piping system, head reduces as a squared function of flow.

$$\text{Thus: Ft.Head (New)} = \text{Ft.Head (Design)} \times \left[ \frac{\text{GPM Flow (New)}}{\text{GPM Flow (Design)}} \right]^2$$

Application of this formula illustrates the change in system head loss with flow change and is commonly called the "System Curve". The change in head with flow change is shown in Figure 4. as the system curve. V/S pump efficiency is assumed constant along the specific system curve shown in Figure 4.

Figure 4. also illustrates pump curve change with pump speed. A method for plotting the V/S curve changes with speed change is described under heading 7: V/S Operating Principles and Operational Cost Estimation.

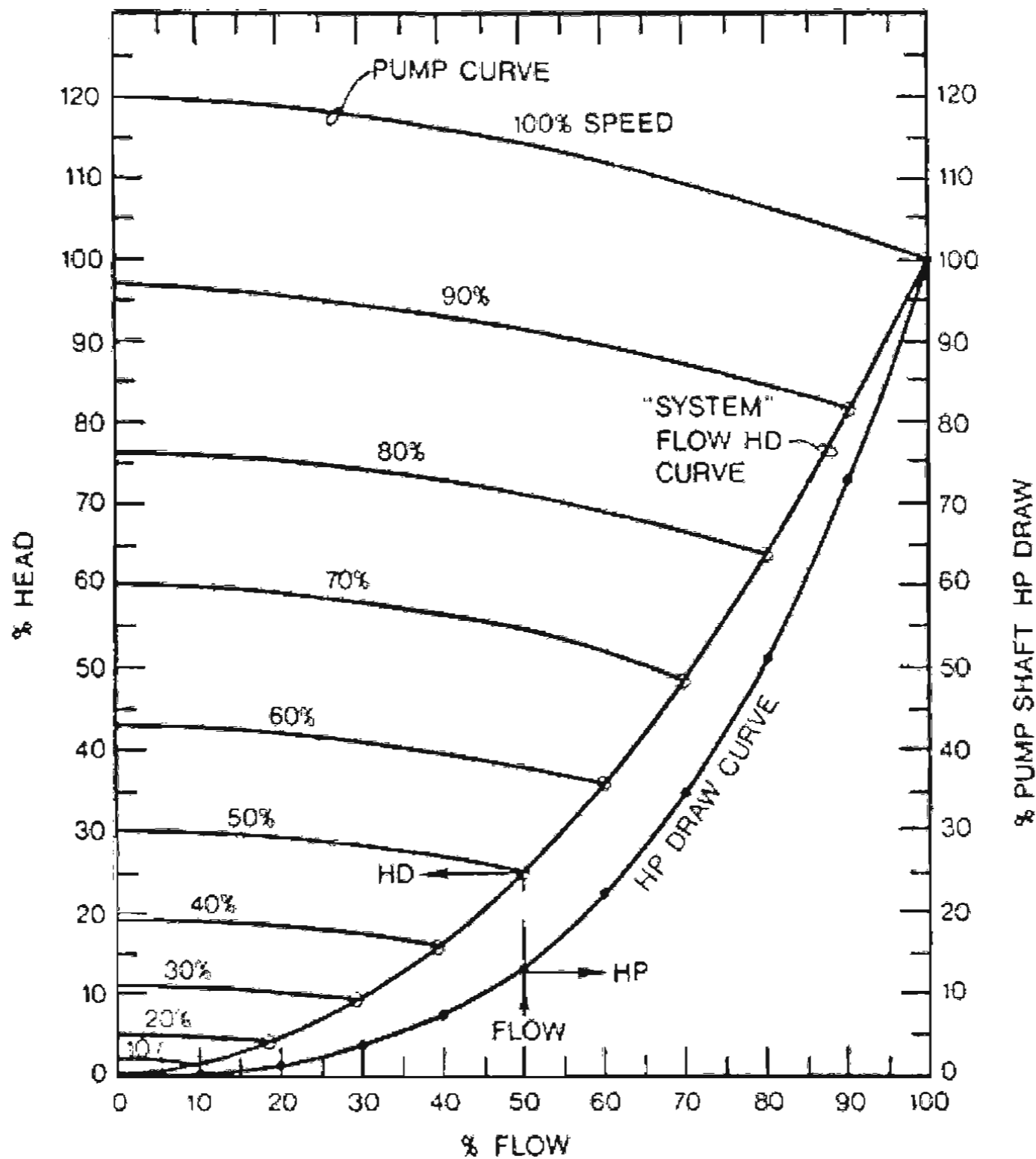


FIGURE 4. Theoretical Pump HP Reduction with Flow Reduction for V/S Pump

The V/S Pump must operate at the intersection of its "speed" curves with the system curve. Thus, at 50% speed, the V/S pump will provide 1/2 (50%) design flow and 1/4 (25%) design head.

The potential HP reduction for V/S pumping is also shown on Figure 4. The basic pump power draw formula states that if flow is reduced by 1/2, and head by 1/4, the HP reduction will be  $1/2 \times 1/4$  or 1/8 that required by design flow.

The HP reduction curve shown in Figure 2 has been used for many years as the basis for V/S pump applications. It implies, however, that V/S pump applications must result in the HP savings illustrated simply because a V/S pump has been used. Unfortunately, it's not that simple. The key to V/S pump savings is in the proper application of a V/S pump and its controls in the system.

One of the factors that modifies theoretical V/S pump power saving, as shown in Figure 4., is the actual proportion of system head loss subject to change with flow reduction in comparison with that proportion maintained as constant. These proportions will be affected by system types.



3.

GENERALIZED V/S PUMP HP SAVING AS AFFECTED BY SYSTEM TYPES

The theoretical V/S pump HP reduction with flow reduction shown in Figure 4. applies only in specific circumstances. One such circumstance relates to a closed loop pumping system, in which the V/S pump is used as a temperature control instrument as illustrated in Figure 5.

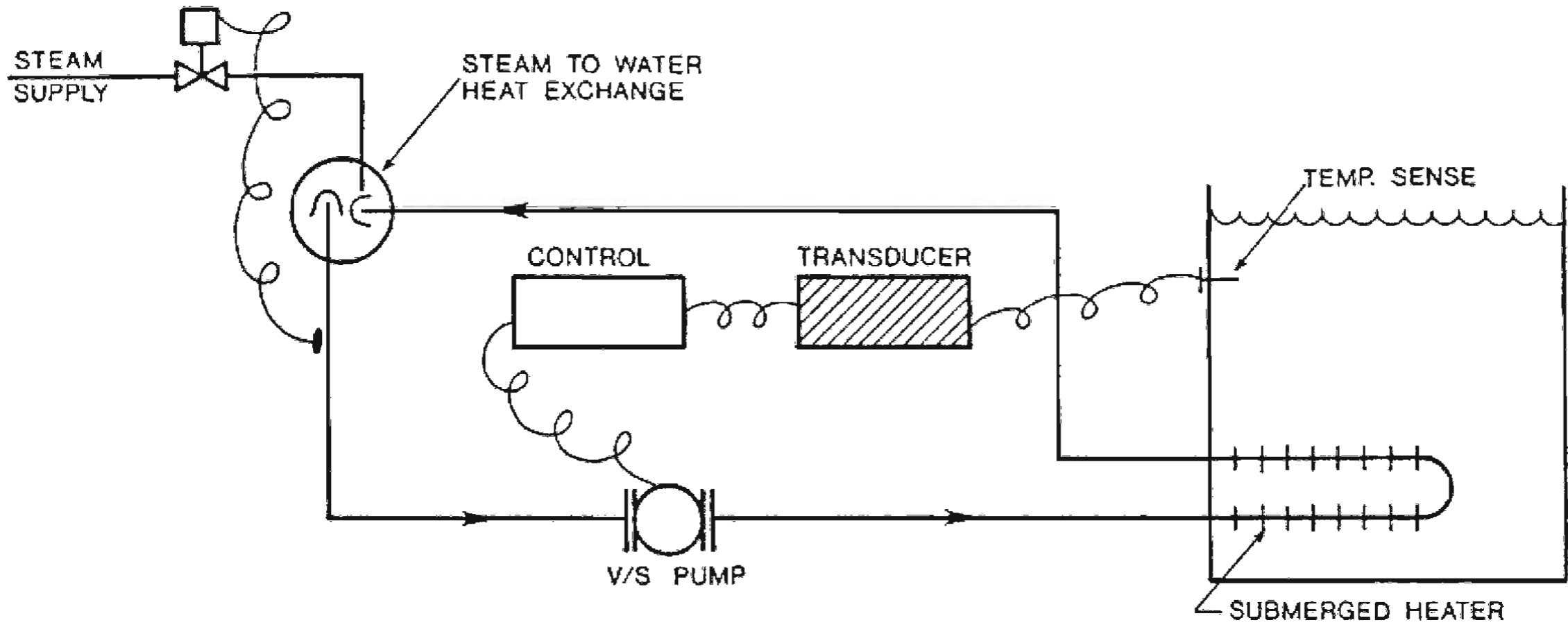


FIGURE 5. V/S Pump Used for Control of Constant Temperature Batch Process in Closed Loop Piping System

The V/S pump in Figure 5. will control heat transfer (batch temperature) by flow variation. As flow varies, head loss in the piping system and heat exchangers will vary as the square of the flow change. The V/S HP change shown in Figure 4. will apply because there is not a maintained head difference - the entire system head loss decreases as flow reduces.

Most V/S pumps are applied, however, to systems which have a requirement for a maintained or constant difference as a percentage of total system head.

The great majority of "open systems" require that water be raised from a low level to a higher level. The difference in water level is a constant head difference that must be maintained at all times by the V/S pump despite flow change. A cooling tower, using a V/S pump, is a good illustration as shown in Figure 6.

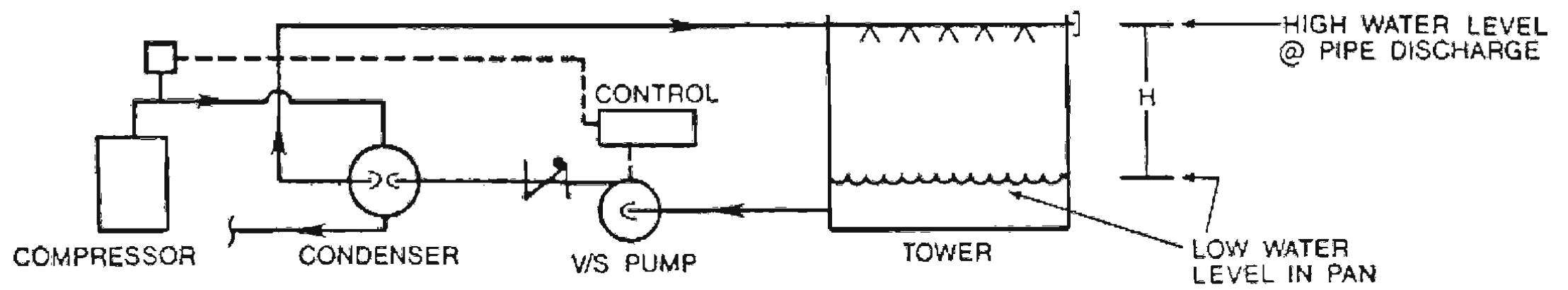


FIGURE 6. Open System; Constant Head (H) Must Be Provided By Pump at All Times

The closed loop HVAC system will also require a constant head differential at all times. This is for control reasons and to insure the necessary differential head across any controlled terminal sub-circuit is maintained for full load operation. The typical HVAC application is shown in Figure 7.

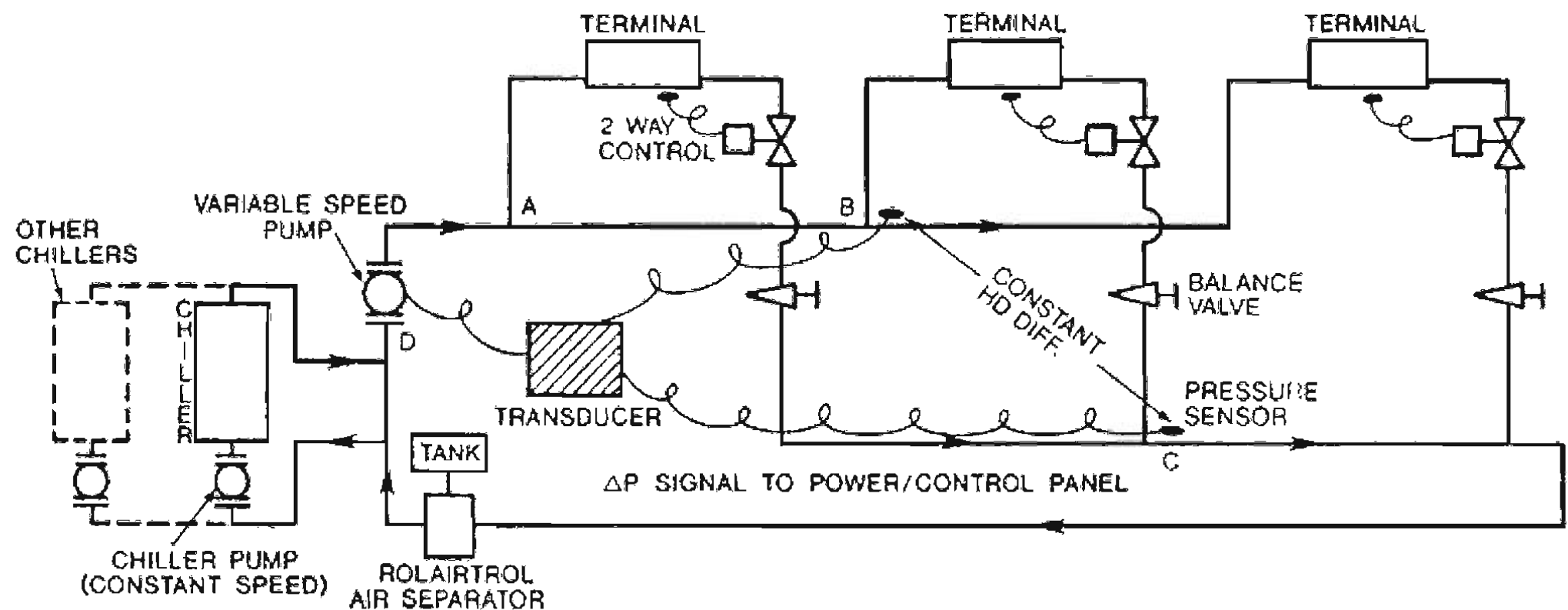


FIGURE 7. Closed HVAC System; Constant Head Difference Must Be Provided by Pump at All Times

Potential HP savings for both the open and closed loop systems can be significantly affected by maintaining the required constant head difference. V/S pump power savings with flow change will be the greatest when the constant head difference is only a small proportion of total pump head so the variable head loss dominates.

In Figure 7., the constant head difference is established by the pressure differential sense points across the terminal circuit ( B to C ). The variable head loss is stated by the piping head loss; in this case pipe length A-B plus C-D, including valves and fittings.

The effect of changing ratios of constant versus variable head loss in potential V/S power saving is shown in Figure 8.

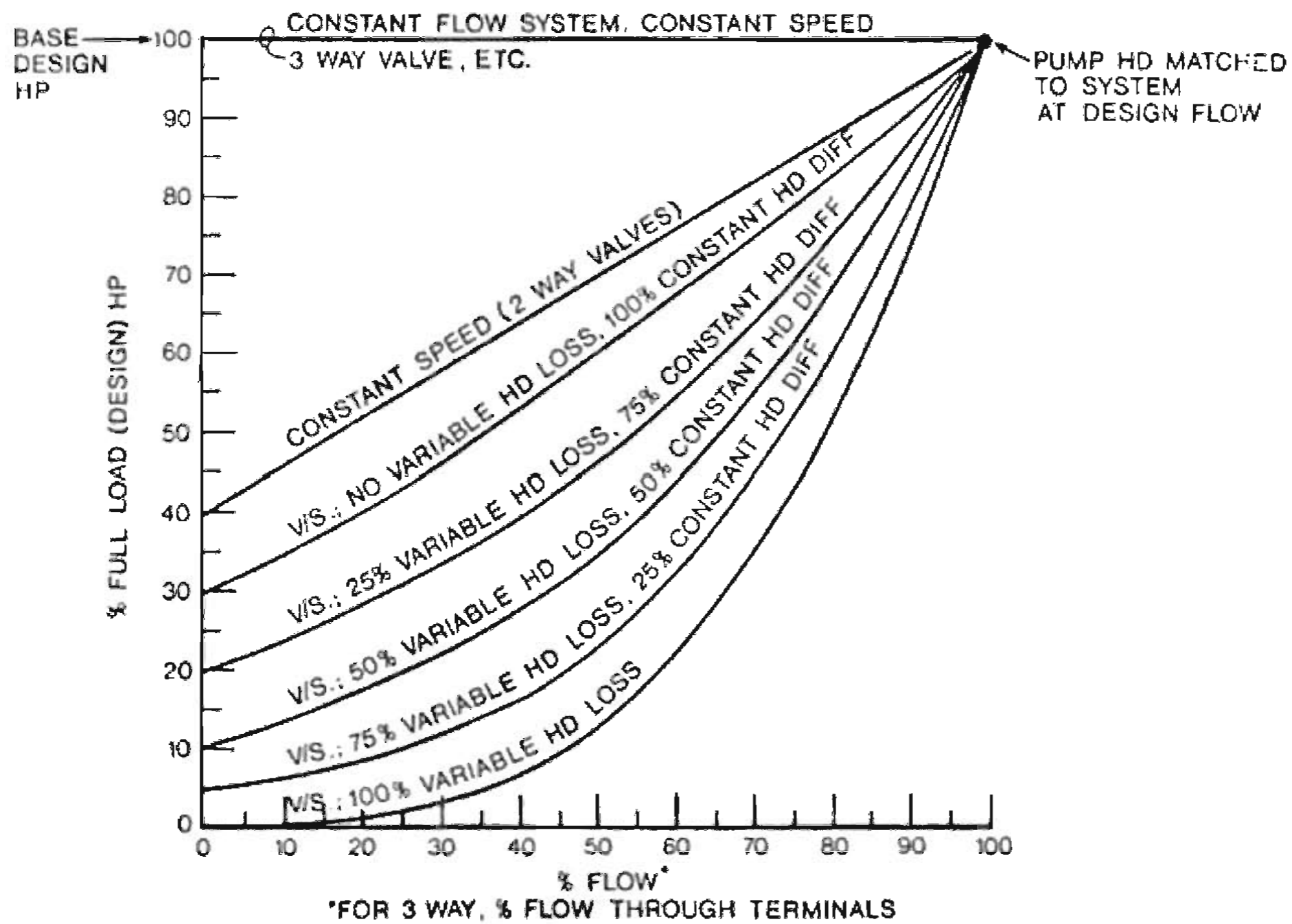


FIGURE 8. Potential HP Saving as Affected by Maintaining a Constant Head

It should be emphasized that the V/S pump must work against a high variable head loss if significant speed reduction (power saving) is to occur. Without a variable head loss, the V/S pump acts similar to C/S; the only difference being that the V/S pump eliminates the "rising" pump curve characteristic of the C/S pump with decreased flow.

As an example, in an equipment room serving several large air handlers located close to the chillers etc., minimal V/S power savings will occur because there is limited pipe length (variable head loss). The "best" V/S application would have long pipe runs and minimal terminal unit and control valve head losses.

The same considerations apply to open pumping systems. The "best" application would have high variable head losses and minimal constant (elevation) heads.

It is evident that in V/S system design, variable head loss should be maximized and constant head difference minimized. For HVAC application, this means that terminal unit head loss should be as low as possible.

Since all terminal circuits must be balanced, the highest head loss terminal circuit will establish the required constant differential head. Very high head loss terminal circuits can be primary-secondary pumped in order to reduce the constant head difference need and increase the percentage of variable head loss. This will maximize V/S speed reduction and consequent power saving. The modification is shown in Figure 8A.

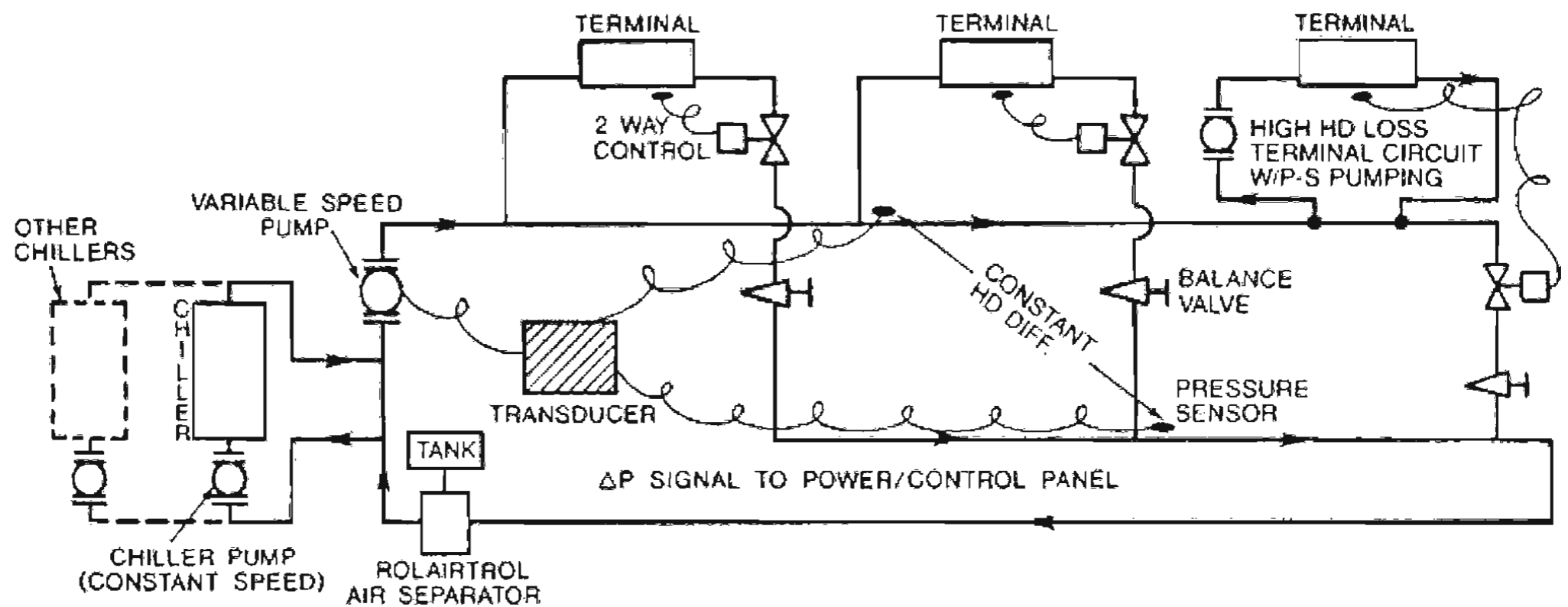


FIGURE 8A. Constant Head Differential Reduced by Primary-Secondary Pumping of High Head Loss Circuit

Figure 8 also shows the potential power savings by modifying existing three way valve control systems. Potential power savings are high when such systems are properly modified; especially when C/S pumps are safety factor overheaded and the system is not balanced.

Pump overheating has always been a major factor relative to excessive pump power draw. Figure 9 illustrates the bad effect of pump overheating on pump power consumption.

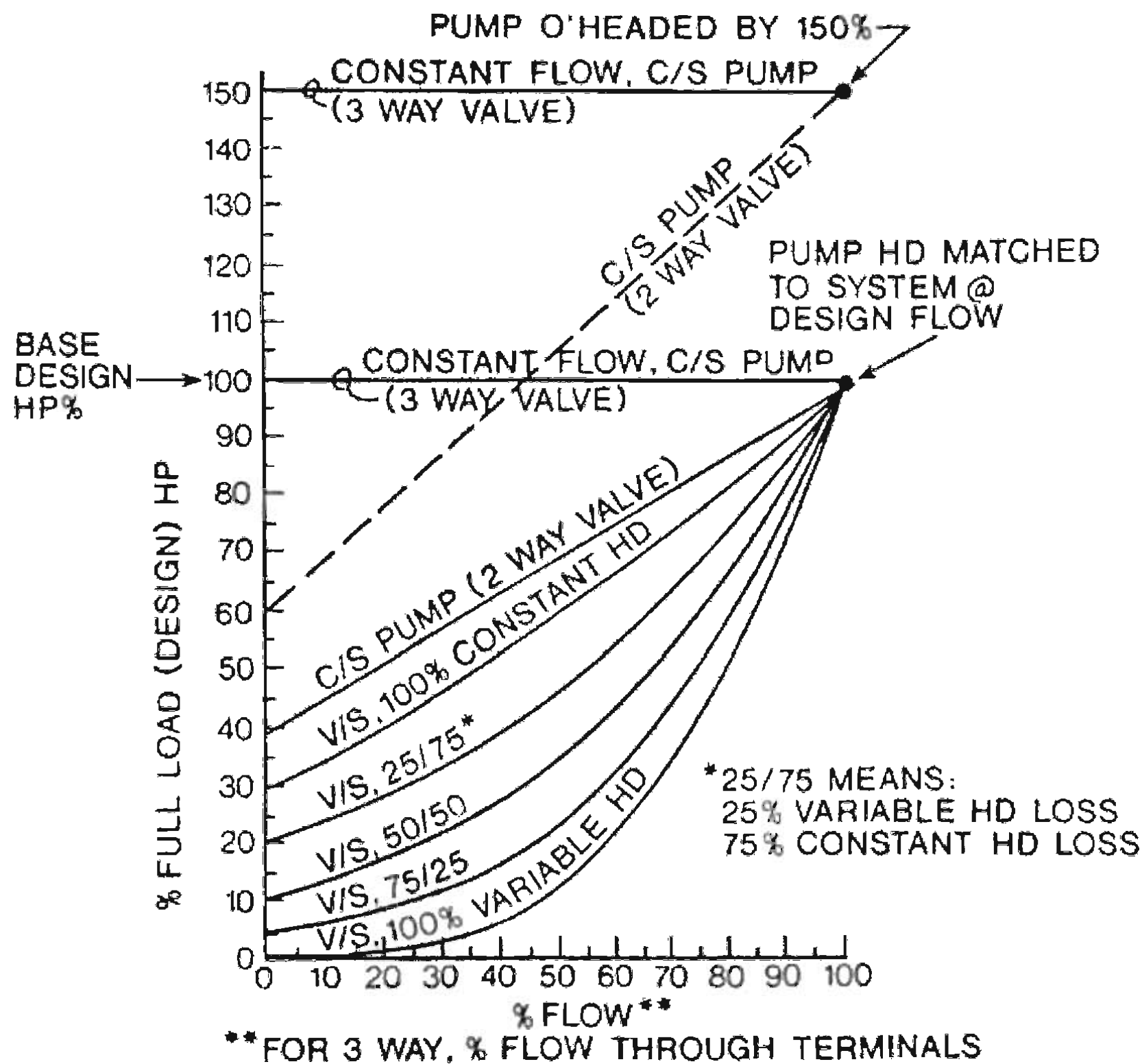


FIGURE 9. Potential Savings for V/S Application to Existing Systems with Overheaded Constant Speed Pumps

Potential V/S power saving for existing three way valve controlled systems using overheaded constant speed pumps are dramatic when these systems are properly modified to V/S.

The same consideration applies to overheaded pumps now used in existing two way valve controlled systems. Potential savings, available by simply "dialing down" pump speed through use of a V/S controller, are high.

#### 4. GENERALIZED PUMP POWER SAVING POTENTIAL FOR PARALLEL CONSTANT SPEED PUMPS

Parallel constant speed (C/S) pumps have been used for many years in a simple method for achieving pump power reduction for variable volume (two way valve controlled) HVAC systems. Paralleled C/S pumps compete favorably with V/S pumps, particularly in smaller systems where variable system piping head loss is a small proportion of total system head needs.

The power saving potential for parallel C/S pumps is derived from the basic pump HP draw formula:

$$HP = \frac{\text{Flow (GPM)} \times \text{Head (Ft.)}}{3960 \times \text{Pump Efficiency}}$$

Parallel C/S pumps are usually selected so that each pump provides 50% design flow. At 50% flow, the basic pump HP draw formula then states that each pump provides 50% of design HP needs.

The parallel C/S pump power draw correlation with a single C/S pump, assuming similar pump efficiencies, is shown in Figure 10.

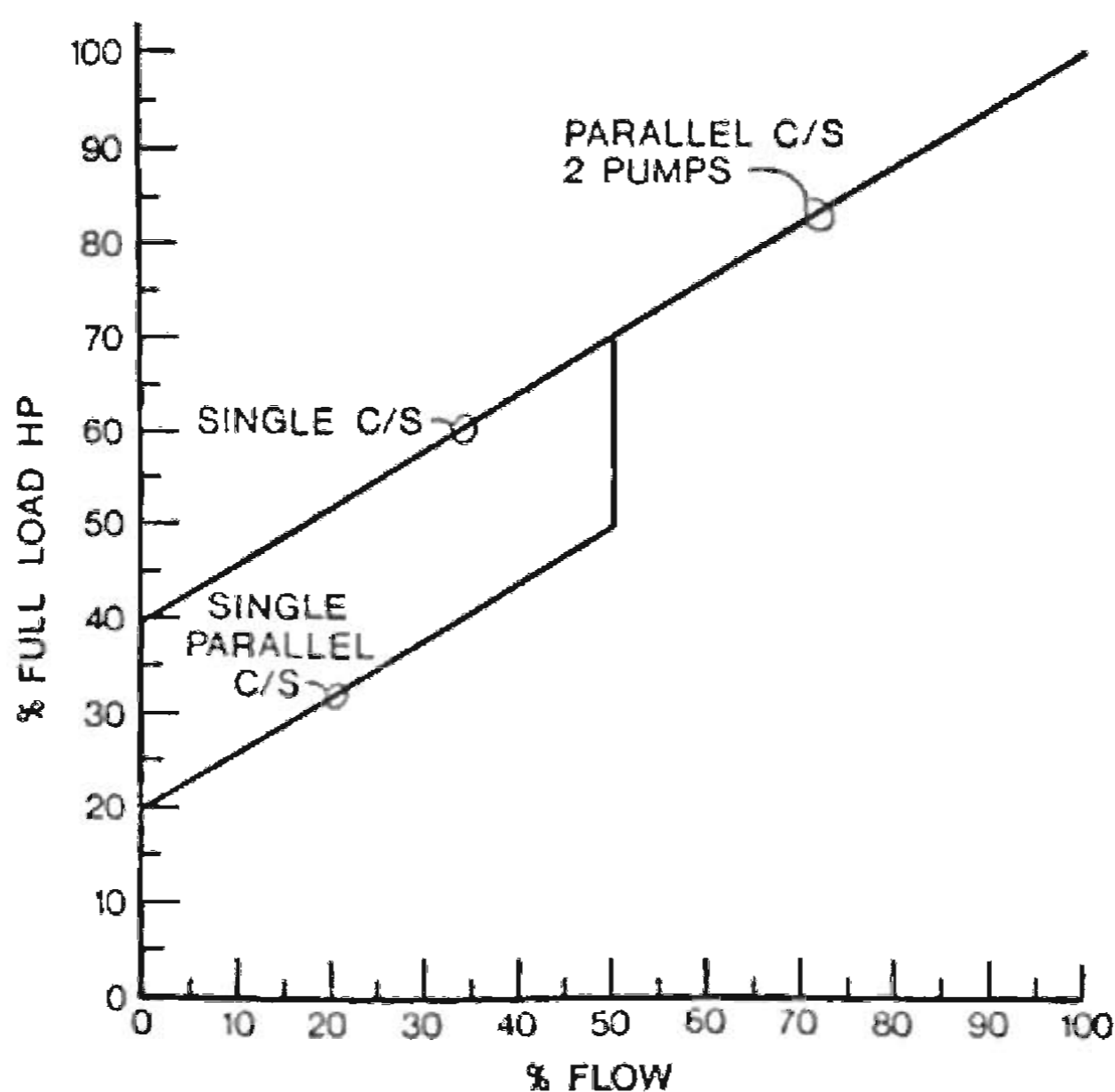


FIGURE 10. Parallel C/S Pump Power Saving Advantage Over Single C/S at Low Flow

A similar advantage for parallel C/S pumps is comparison with V/S pumps occurs when the variable head loss in a V/S pumped system is a low percentage of total pump head. This is shown in Figure 11.



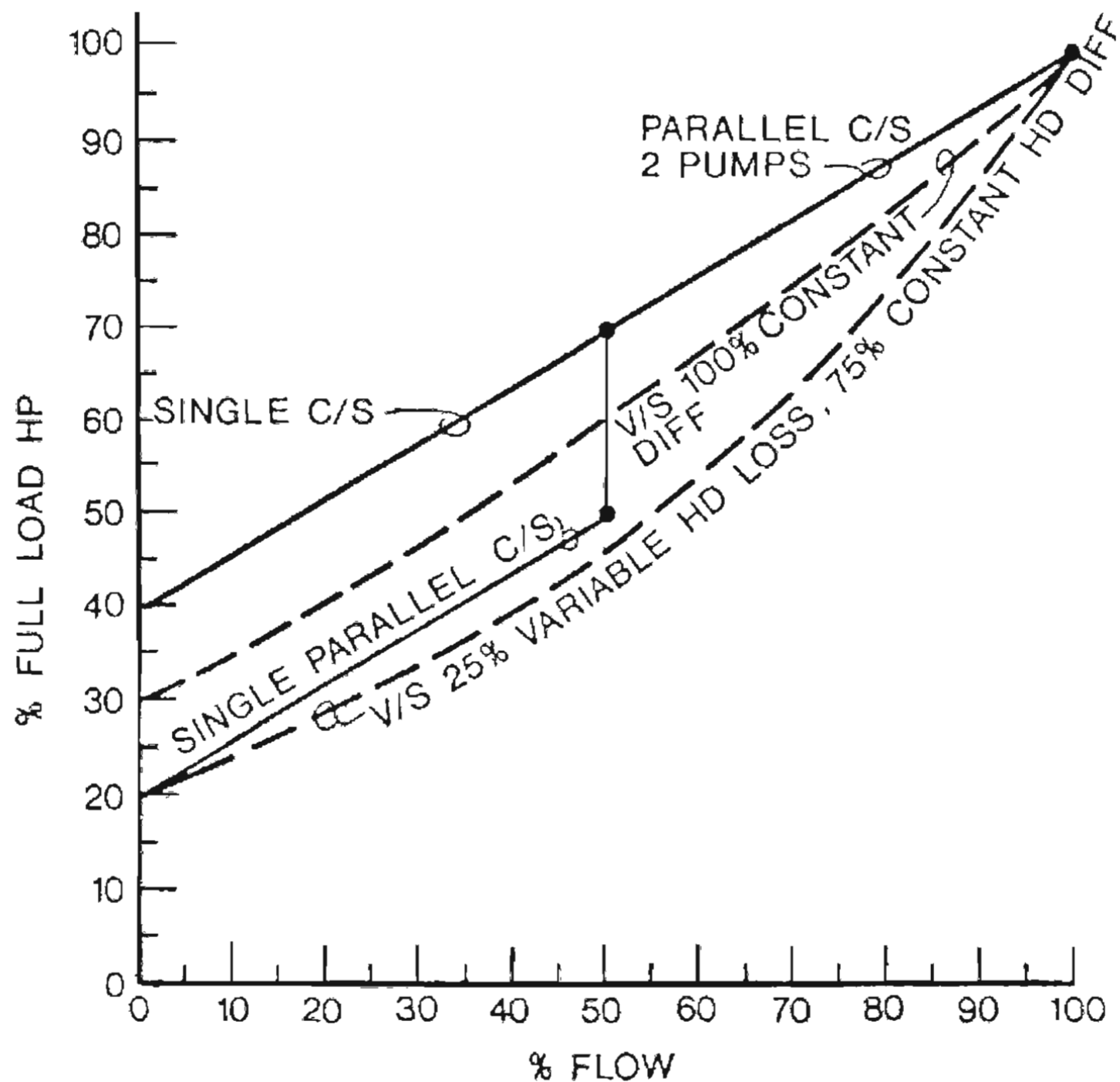


FIGURE 11. Best Application Area for Parallel Equally Sized C/S Pumps in Comparison with V/S Occurs When System Variable Head Loss is Low

It should be noted that the HP draw comparison in Figure 11 is theoretical, and will be modified by actual pump efficiency changes. In general, low flow pumps are less efficient than high flow pumps. Pump HP draw for parallel C/S pumps will also be increased by head loss through check valves and/or pressure reducing valves placed at each pump discharge. While conventional check valves applied to parallel C/S pumps have a generally low head loss, the combination check and pressure reducing valve used on parallel C/S Pressure Booster Pumps is relatively high; in the order of 15 ft. at design flow. The effect of this additional head loss on power consumption is shown in Figure 55.

5. GENERALIZED PUMP POWER SAVING POTENTIAL FOR PARALLEL V/S WITH C/S PUMPS

The principle advantage for using one V/S pump in parallel with a similarly sized C/S pump is that V/S controller size (cost) can be reduced. When properly selected and controlled, HP savings will correspond with the savings potential for a single V/S pump and control (see Figure 8.).

There are several variations to the idea of using parallel C/S and V/S pumps.



One variation is to use two equally sized pumps and motors with a single V/S controller. The single V/S controller can control either pump. At start-up, the V/S controller works with pump #1 and increases pump speed in accordance with system need until full speed is reached. At this time, Pump #1 is "locked" into full speed, and V/S control is changed to pump #2, which now changes pump #2 speed in accordance with system needs. A similar, but reversed cycle occurs as system load decreases. The control variation as described leads to the potential for "lead-lag" pump operation.

A similar, but less costly control establishes one pump as C/S only, the second as V/S only. In this case, at start-up, the V/S increases pump speed in accordance with system needs until a set control point is reached, at which time the C/S pump is started. The V/S continues in operation, meeting increasing system flow needs as required. Again, a similar but reversed cycle occurs as load decreases.

It would not seem that the extra cost associated with lead-lag pumping is justified. B&G centrifugal pumps have long lives and are easy to service. The principle service problem with any V/S system is associated with the V/S controller which, in this case, cannot be lead-lag controlled.

Parallel V/S and C/S pump selection and operational characteristics must be carefully considered if the combination V/S and C/S application is to operate successfully. Basic concerns are discussed in detail under heading 9.

The preceding discussion has provided a general background for the application of V/S pumps, but has not detailed V/S curve construction methods or operational cost approaches.

The following material will outline methods for determination of comparative operational cost for C/S, V/S and paralleled V/S and C/S pumps. The comparative power consumptions, as finally developed, can be used for initial evaluations as to which pumping system would be most advantageous to use for generalized circumstances. In addition, the need for operational cost estimates requires a detailed examination of V/S pump curve applications, and should lead to a better understanding of how V/S pumps actually work.

## 6. PUMP POWER DRAW FUNDAMENTALS, AND DEVELOPMENT OF SYSTEM PUMPING COSTS FOR SINGLE C/S PUMP

Any discussion concerning pump energy conservation must start with pump power draw relationships. This relationship is shown below and applies to any pump; constant or variable speed.

$$\text{BHP}_p = \text{Ft. Hd.} \times \text{GPM}/3960 \times E_p$$

A constant speed pump selection stated for 1200 GPM at 70 ft. is shown in Figure 12 (9 1/4" impeller). At the selection point, pump efficiency (Ep) is 83%. For this condition (1 in Figure 12) pump power draw will be:

$$BHP_p = 70 \times 1200/3960 \times .83 = 25.6 \text{ HP}$$

Given motor efficiency of 89%, the motor KW draw at this 1200 GPM flow point will be 25.6 HP x .746 KW/HP x 1/.89 = 21.45 KW.

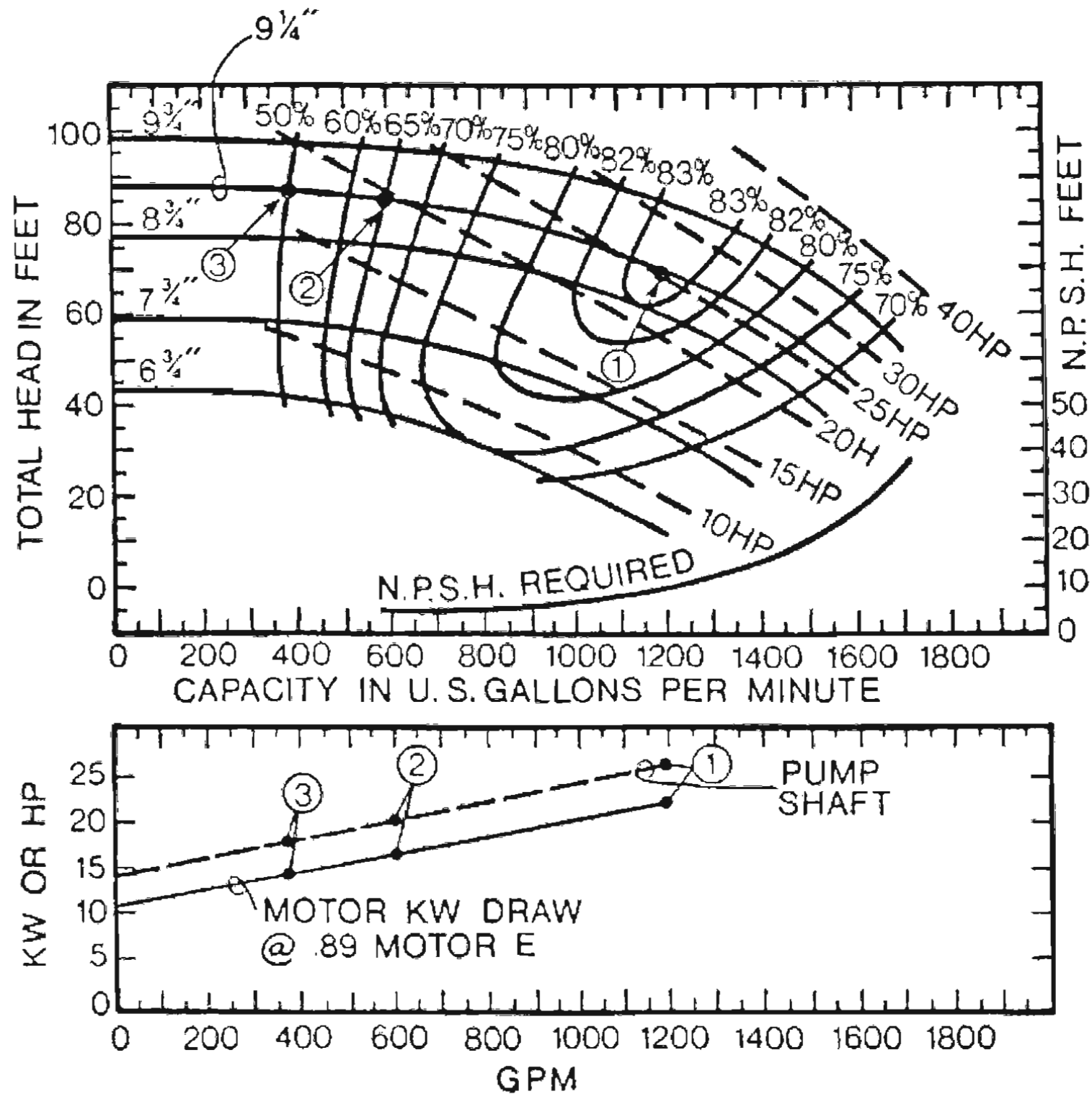


FIGURE 12. Pump Curve and Resultant Power Draw Curve Over Flow Range for 9 1/4" Impeller

Since this pump will be applied to a two way valve controlled variable volume pumping system, (see Figure 13), we are interested in pump power consumption over the complete pump flow range. Keep in mind, as two way valves modulate, flow will reduce.

Power consumption can be determined over the entire flow range by selecting pump head and efficiency at various flow points on the selected pump curve. For example, at point 2, head is 85 ft., flow is 600 GPM and efficiency is 65%. Application of these points to the basic HP draw formula states 19.8 pump shaft brake HP draw and 16.6 KW motor draw with the motor at .89E.

Similar calculations for other flow points are used to establish the power vs. flow curve shown in Figure 12.

Note that the example power draw curve is based on usage of a single full load motor efficiency over the entire flow range. This is despite the fact that pump motor HP draw has reduced from about 25 HP at 1200 GPM to about 14 HP at no flow, and is contrary to the generally accepted belief that motor efficiency varies greatly with motor loading. There are two reasons for using a constant motor efficiency and avoiding the time consuming complications inherent to the use of specific motor rating curves.

First, reference to the pump power draw formula illustrates that a 1% increase in pump head, over that needed by the system, is equivalent to about a 1% decrease in either pump or motor efficiency. A pump head determination made from piping plans, is not too accurate and is clearly the most important variable relevant to establishing the pump power draw correlation.

The second reason is motor efficiency, for motors over 10 HP, change very little in operating efficiency down to about 1/3 load. The change in efficiency is not "enough of a difference to make a difference"; most especially when the over-riding importance of the pump head term is considered relative to power draw.

The pump shown in Figure 12 will be applied to the system shown in Figure 13. Terminal unit plus control valve pressure drop is 30 ft., and piping head loss at full design flow is 40 ft. (about 1600' total equivalent pipe length (TEL)). The chillers are primary-secondary pumped because system flow is highly variable. The system has been properly balanced so that short circuiting during high load draw will not occur. The pump selection is exactly matched to system need.

2 WAY VALVE CONTROLLED, REVERSE RETURN PIPING

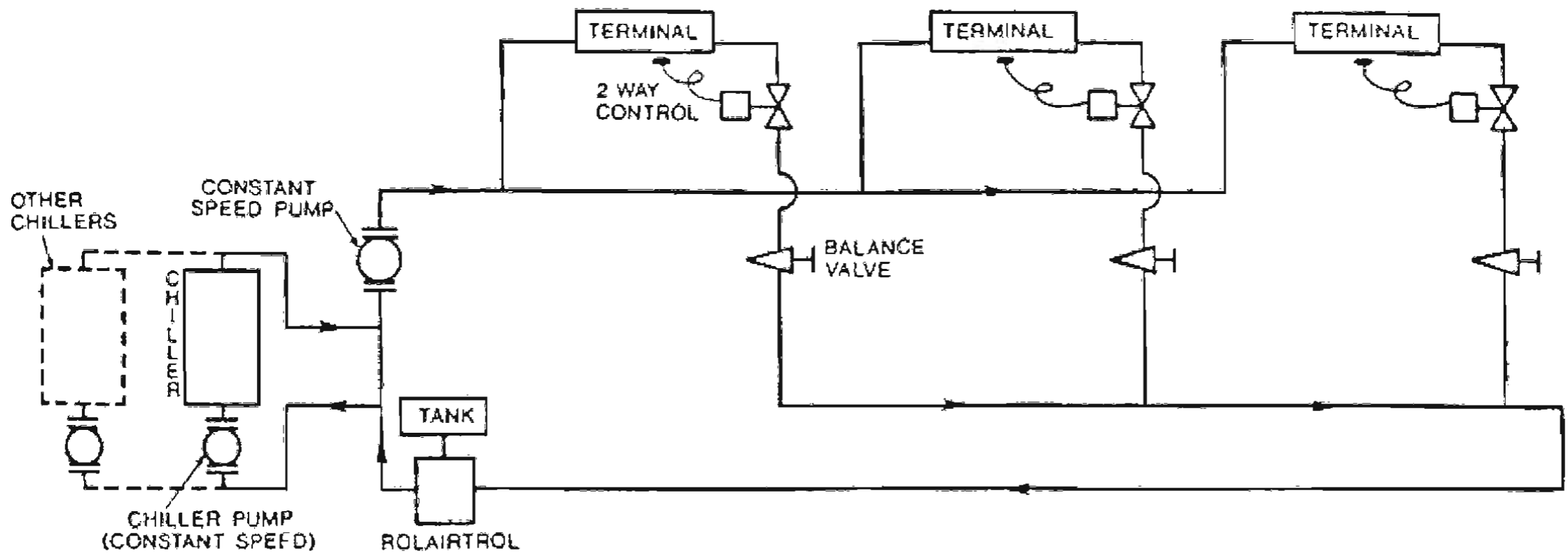


FIGURE 13. Example of C/S Pump Exactly Matched to System Needs

A load profile, in combination with the specific flow vs. power consumption curve in Figure 12 will establish power consumption and operational costs for the system. The power consumption for the example is shown in Table 2.

TABLE 2

Load Profile & KWH Usage Calculation for C/S Example

% LOAD	% TIME	HRS	GPM	KW	KWH
0-10	2	175	120	12.5	2188
10-20	3	262	240	13.3	3485
20-30	5	437	360	14.5	6337
30-40	15	1311	480	15.5	20321
40-50	20	1748	600	16.5	28842
50-60	30	2625	720	17.8	46725
60-70	15	1311	840	18.7	25516
70-80	5	437	960	19.8	8653
80-90	3	262	1080	21	5502
90-100	2	175	1200	22	3850

8760 HRS/YR TOTAL 151,419  
151,419 x \$.10/KW = \$15,140

At \$.10/KWH, the yearly operating cost of this example is \$15,149/Yr., based on a yearly draw of 151,140 KWH. The example is based on 1200 GPM: at 2400 GPM, power draw will be approximately doubled, so that at \$.10/KWH, operating cost would be \$30,280/Year. If utility costs were \$.05/KWH, operating costs stated above would be halved.

Operational costs for similar systems, can be identified. These costs will vary as a function of pipe length, while other base variables are retained; 1200 GPM, 30 ft. combined head loss for terminal and control valve, \$.10/KWH utility cost, etc. A plot of operating cost for the two way valve controlled variable system with constant speed pumps is shown in Figure 14.

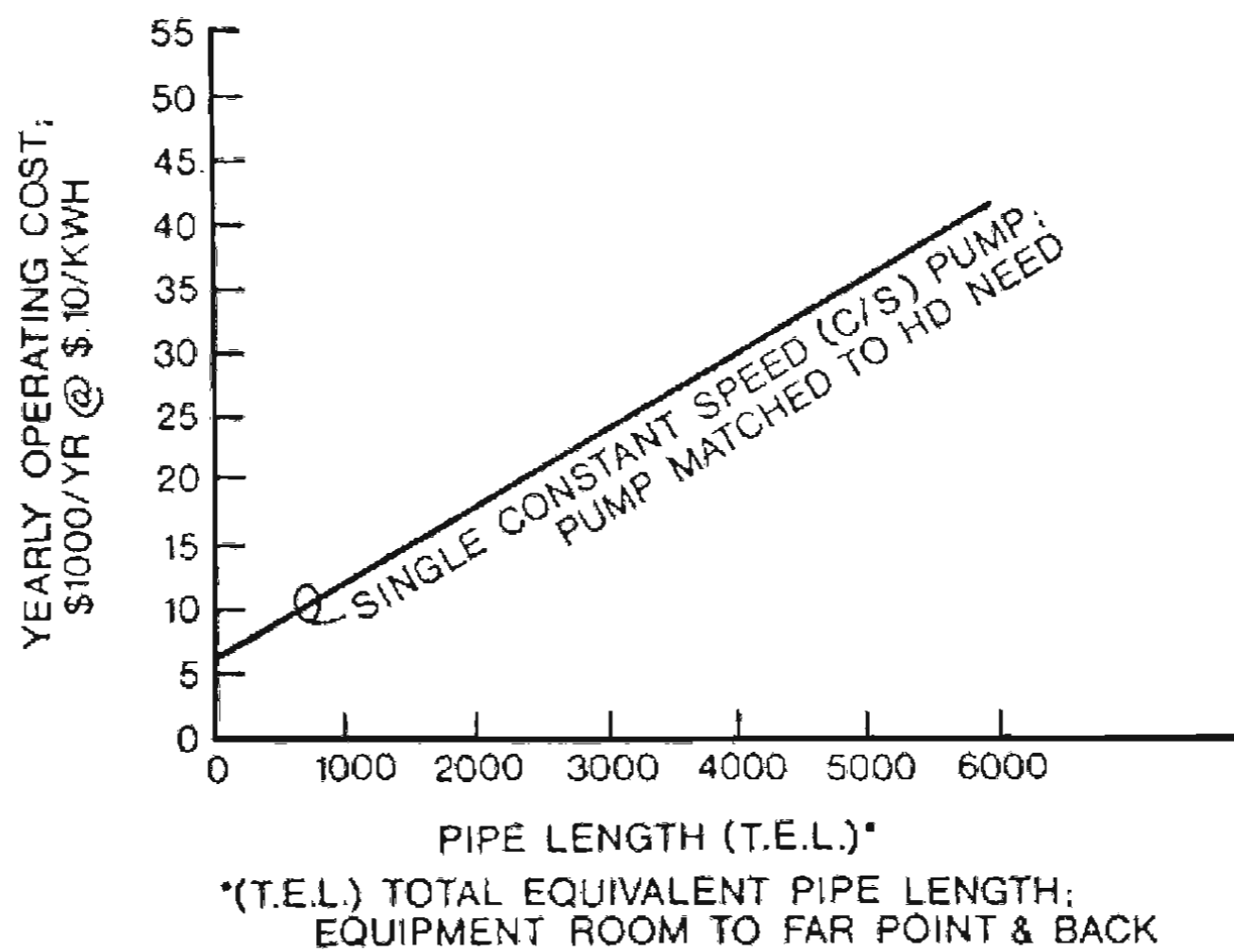


FIGURE 14. Generalized Example: Yearly Operating Cost for 1200 GPM at \$.10/KWH

The operating costs shown in Figure 14 are based on a pump selection exactly matched to system need. If the pump head is twice that actually needed, operating cost will be twice that shown in Figure 14. The increase in operating cost for overheaded pumps is shown in Figure 15.

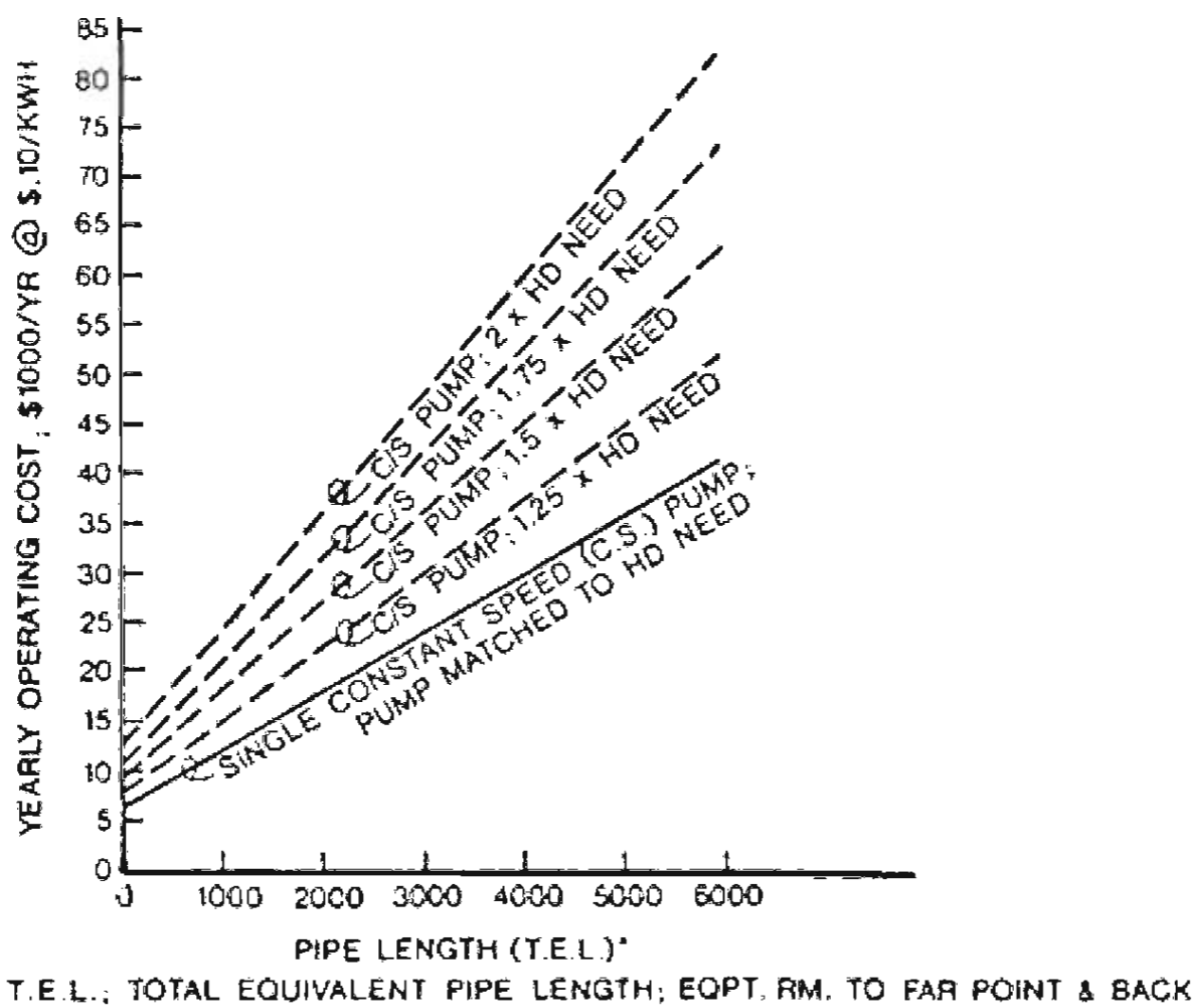


FIGURE 15. Increased Operating Cost for Two Way Valve Controlled System with Overheaded Pumps

It's obvious, operational costs for constant speed pumps increase greatly when pumps are overheaded in terms of actual need. Unfortunately, most pumps are selected for a pump head greater than that needed for the system. This is because of safety factor; the design engineer wants to make sure that the pump is adequate. Head safety factor is also used because of unknowns; the engineer is not sure of terminal unit, control valve and chiller head losses because these may vary due to the bid selection process. In addition, concern about system flow balance (short circuiting) may influence the selection.

There are two answers to the very high operating costs developed by constant speed pump overheading:

- (1) Preset flow balance followed by impeller trim while retaining constant speed pump operation.
- (2) Use of variable speed pumps.

Operational cost saving opportunities for preset flow balance and impeller trim can be very high, but is not a part of a discussion of variable speed pumping. Preset will be covered in a separate discussion - and would drop operating cost to the level described in Figure 4; pump matched to system need.



The variable speed pump, properly applied, will automatically reduce the overhauled pump to minimum power input. Additional pump power saving over and above that forecast by an exact pump to system match, will result because the variable speed pump further reduces power need due to the reduction of piping head loss as flow reduces.

The comparative operating cost reduction for variable speed pumps can be illustrated in a discussion of variable speed operational principles.

7. PUMP POWER DRAW FUNDAMENTALS AND DEVELOPMENT OF SYSTEM OPERATING COST FOR VARIABLE SPEED PUMPING

The example two way valve controlled system, operated with a constant speed pump as illustrated in Figure 13, is changed to variable speed shown in Figure 16.

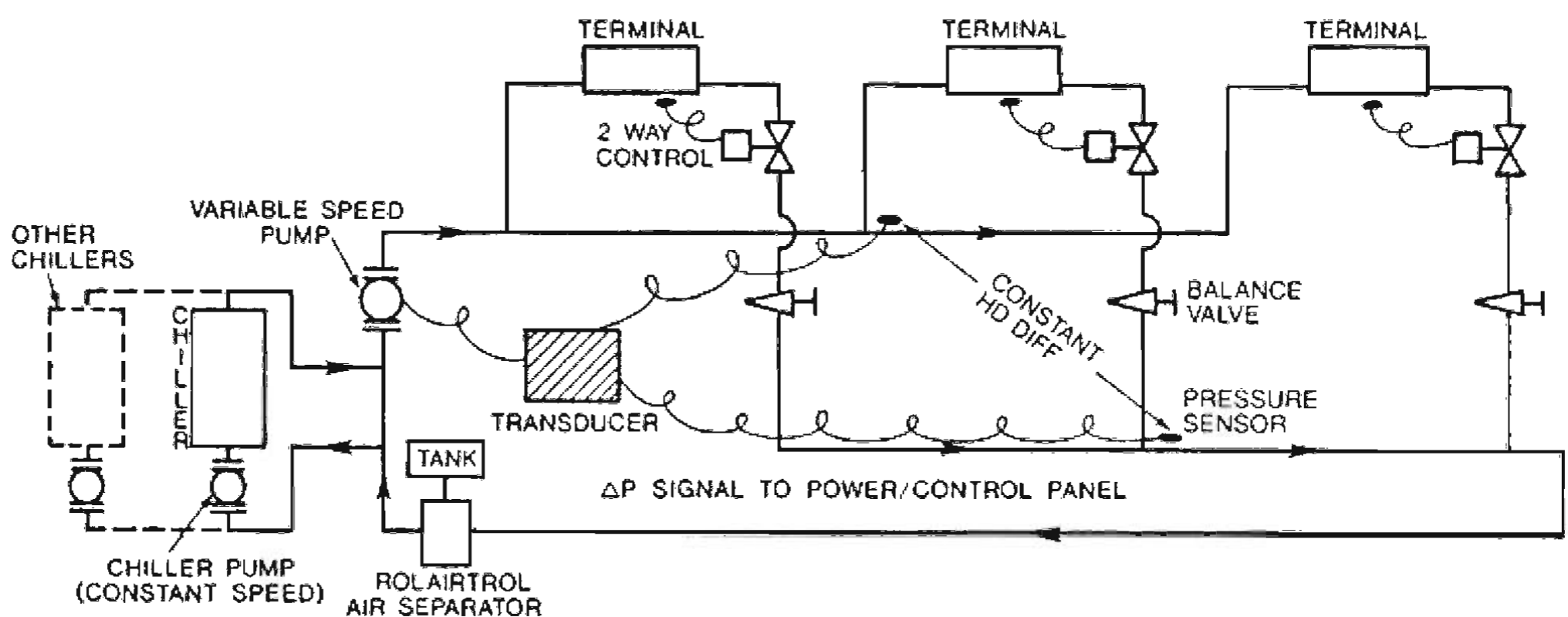


FIGURE 16. Typical Variable Speed Pump System Using Two Way Valve Control and Reversed Return Piping

Note that reversed return piping is shown for the variable speed system illustrated in Figure 16. This is because reversed return piping simplifies pressure differential sensor location and because it maximizes pump energy savings. A discussion of the difference between reversed and direct return piping systems in reference to variable speed pumping systems will be found under heading 16: "V/S Operating Savings as Affected by Application Variables."



Calculations concerning the example variable speed system are based on the same considerations used for the constant speed evaluation previously described. These conditions, at full design load are: 1200 GPM, 40 ft., piping loss for the 1600 ft. (TEL) length and 30 ft. differential head loss across the terminal units and their control valves.

In order to maximize pump power saving, the sub-circuits containing the terminal unit and their control valves have all been flow balanced so that design flow occurs at 30 ft. differential head. This will eliminate short circuiting at terminal loads greater than design when the two way valves are wide open.

For the conditions described, the V/S pump will be selected for 70 ft. head and 1200 GPM. The pump selection is shown in Figure 17, and is identical to that illustrated for the previous constant speed example (6 x 8 x 9-3/4 - H) with a 9 1/4" impeller.

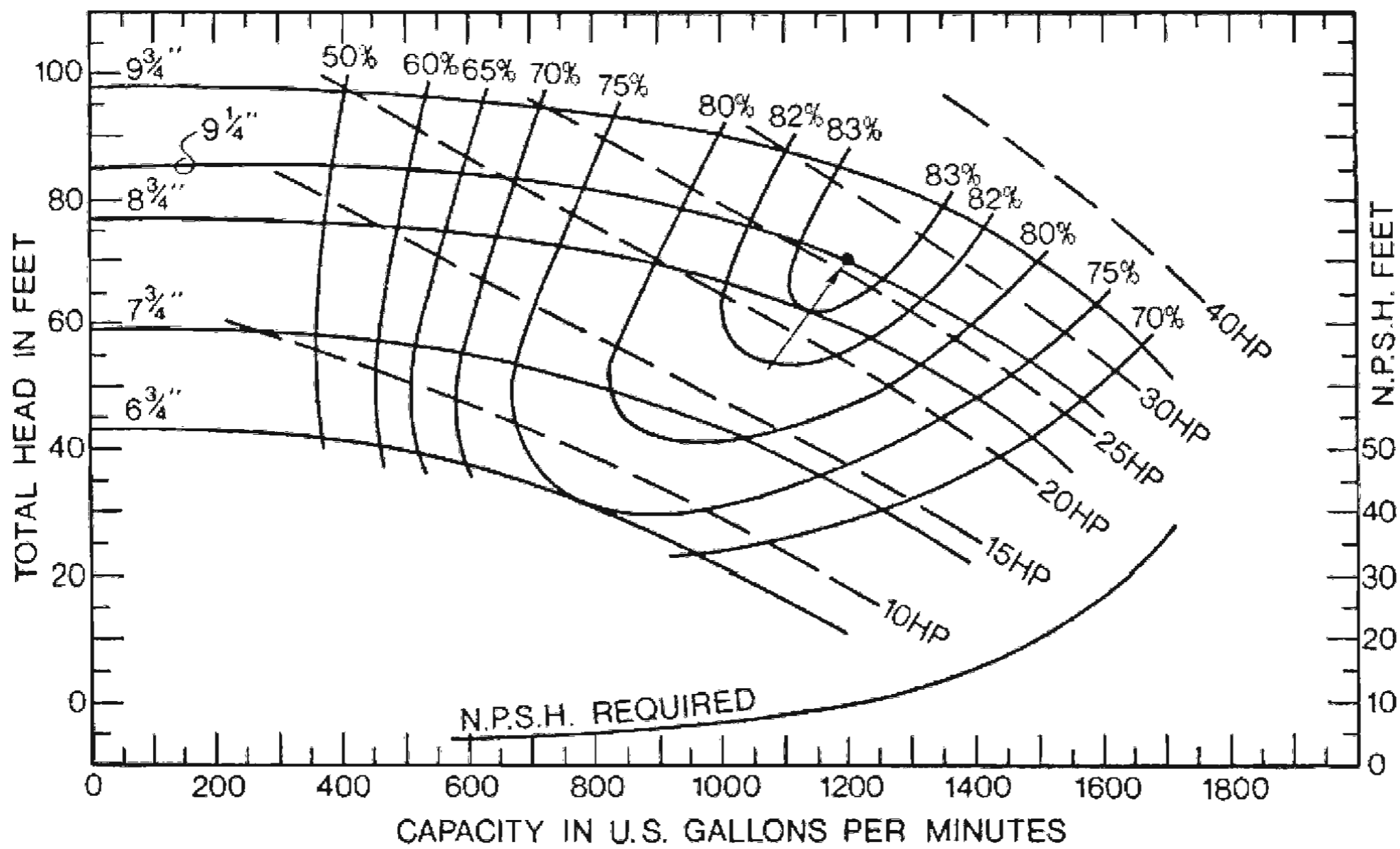


FIGURE 17. Variable Speed Pump Selection with 9 1/4" Impeller Producing 1200 GPM at 70 Ft.

An important question concerning V/S pump application centers on variation of head and flow as pump speed changes. The variable speed pump curve can be constructed from a conventional constant speed curve by observing that for any specific flow and head point on a pump curve:

- (1) Pump flow changes directly as a function of pump speed.
- (2) Pump head changes as a squared function of the pump speed change.

Unlike the relatively inefficient "high slip" V/S drives (variable voltage, fluid drive, etc.), the variable frequency drive does not exhibit slip. For any specific flow head point on the pump curve, and with the pump curve selected for 100% speed using a variable frequency drive, the following tabulation (Table 3) can be developed.

TABLE 3

Flow-Head Variation with Speed

PUMP SPEED %	100	90	80	70	60	50	40	30	20	10	0
PUMP FLOW %	100	90	80	70	60	50	40	30	20	10	0
PUMP HEAD %	100	81	64	49	36	25	16	9	4	1	0

Figure 18 shows the specific pump curve example, selected for 100% speed. The curve illustrates flow-head points for each of several efficiency points on the pump curve. The illustrated flow-head points are used to calculate curve variation with speed change.

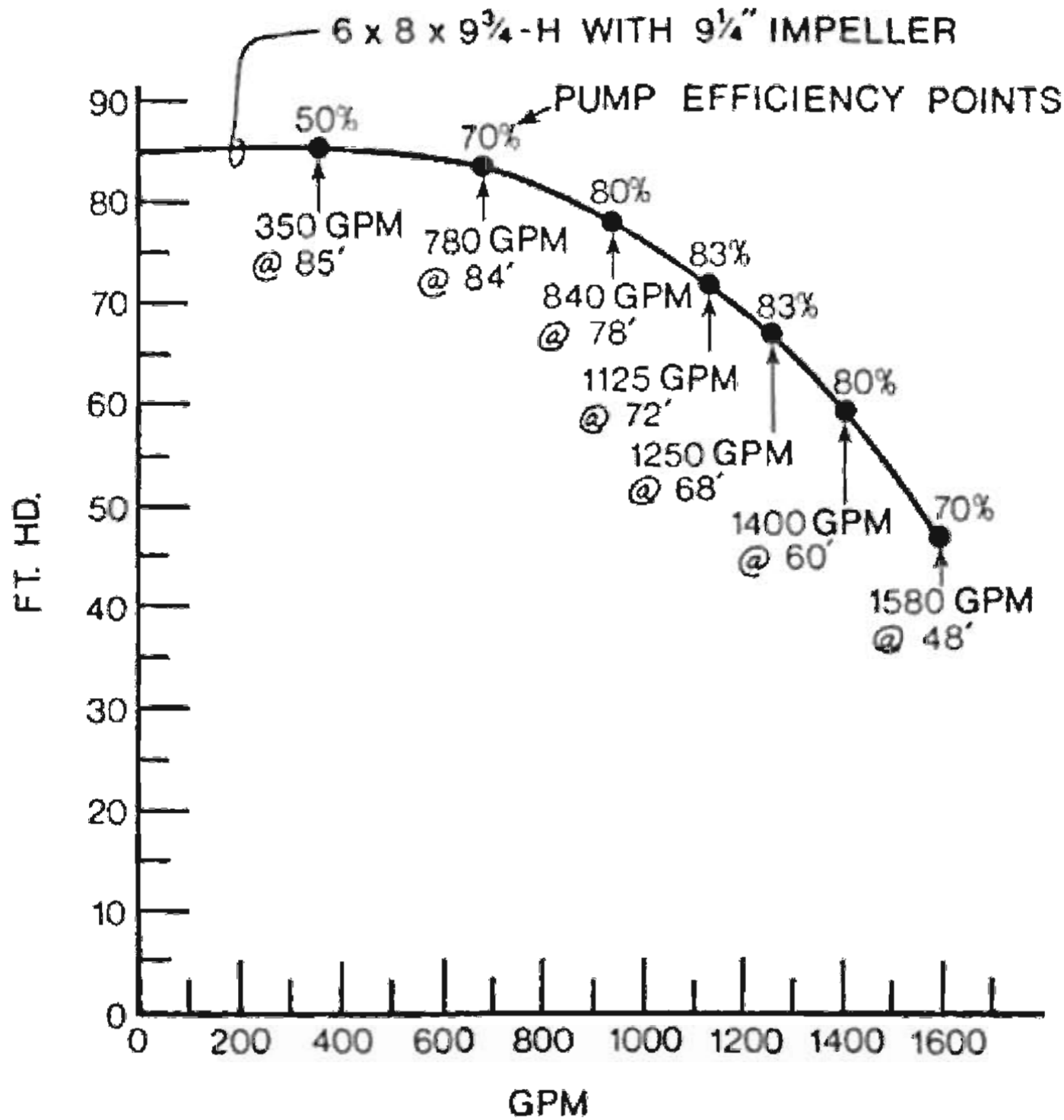


FIGURE 18. Selected Pump Curve Shows Flow-Head at Various Efficiency Points at 100% Speed

Using Table 3 and the specific pump curve, the flow head points can now be used in tabulated form to establish flow head variation with speed change. The calculation for 50% E at 350 GPM and 85 ft. head is shown in Table 4. The resultant curve is called a constant efficiency curve since pump efficiency remains constant along this curve regardless of speed change.

TABLE 4

Flow Head Variation with Speed at Selected Efficiency Point

PUMP EFFICIENCY = 50%, 350 GPM @ 85'

PUMP SPEED %	100	90	80	70	60	50	40	30	20	10	0	
PUMP FLOW	%	100	90	80	70	60	50	40	30	20	10	0
	ACTUAL	350	315	280	245	210	175	140	105	70	35	0
PUMP HEAD	%	100	81	64	49	36	25	16	9	4	1	0
	ACTUAL	85	69	52	42	31	21	14	8	3.5	1	0

Similar calculations will identify flow head variation with speed change for other efficiency points. These points can be plotted and interconnected on a base pump curve, as shown in Figure 19, to provide a complete picture of V/S pump curve characteristics with speed change.

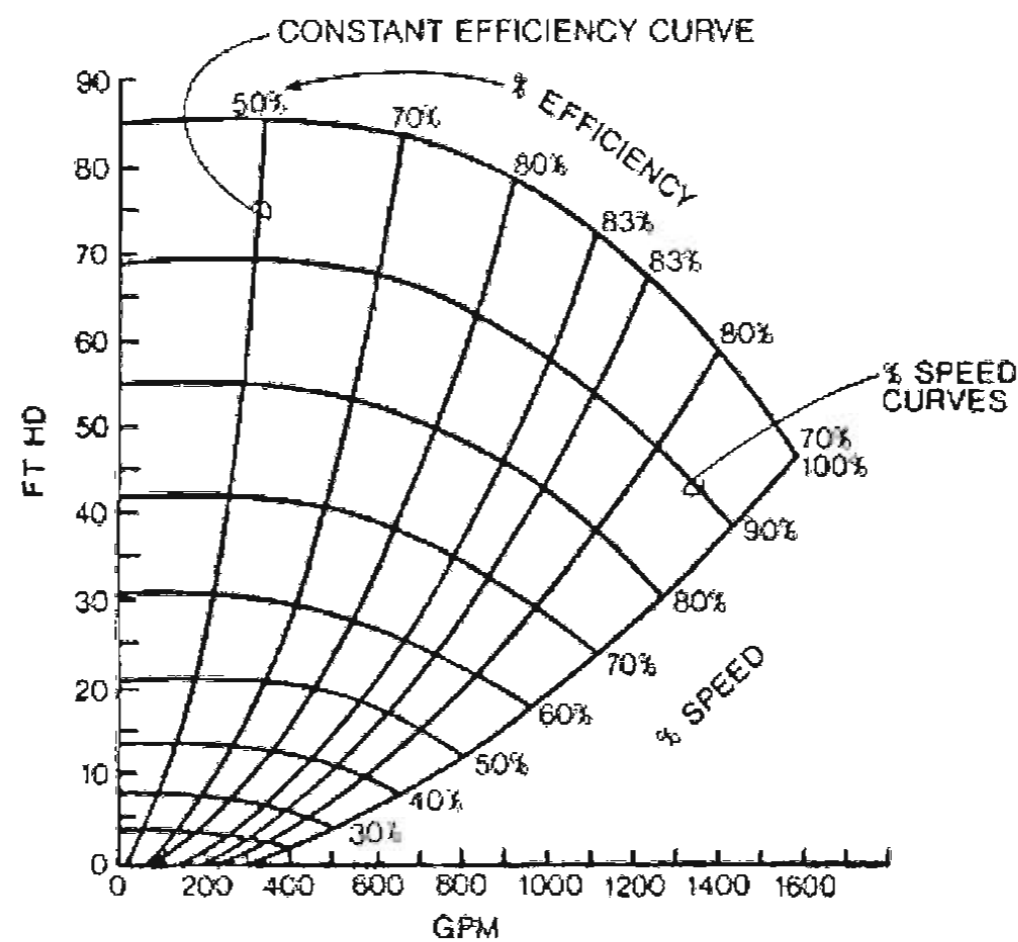


FIGURE 19. Pump Curves and Pump Efficiency Plotted as Function of Pump Speed

Operational points for the V/S pump as applied to the specific piping system must now be determined. This will require a piping system evaluation.

The piping pressure drop for this example is stated as 40 ft. at the full 1200 GPM design flow. The system, designed for V/S operation, uses two way control valves. As load decreases, the two way valves will modulate to closure, reducing flow in the piping system. Piping system flow reduction will cause a head loss reduction. The head loss reduction will be a squared function of flow change. The change in piping head loss with flow change for this example is plotted in Figure 20.

The piping diagram (Figure 16) also illustrates pressure differential pressure control sensors located across one of the piping sub-circuits containing terminal units and its control valves. Proper location of this sensor insures that 30 ft. differential head will be maintained across all sub-circuits at all times. This constant pressure differential must be maintained by the pump at all times so if any valve opens to full load flow, adequate differential pressure is available to provide full load draw.

\*It should be noted that the change in piping head loss with flow change is an averaged curve based on the assumption that all terminal units are subject to an equal % flow change with building load change. While conditions may vary because of different terminal load requirements (exposure, occupancy, etc.), the assumption is useful for basic analysis.

When the constant pressure differential (30 ft.) is added to the piping head loss curve as in Figure 20, a summation head curve is obtained. This curve is called the "control valve", and illustrates the actual pump head that must be maintained by the pump as system flow changes.

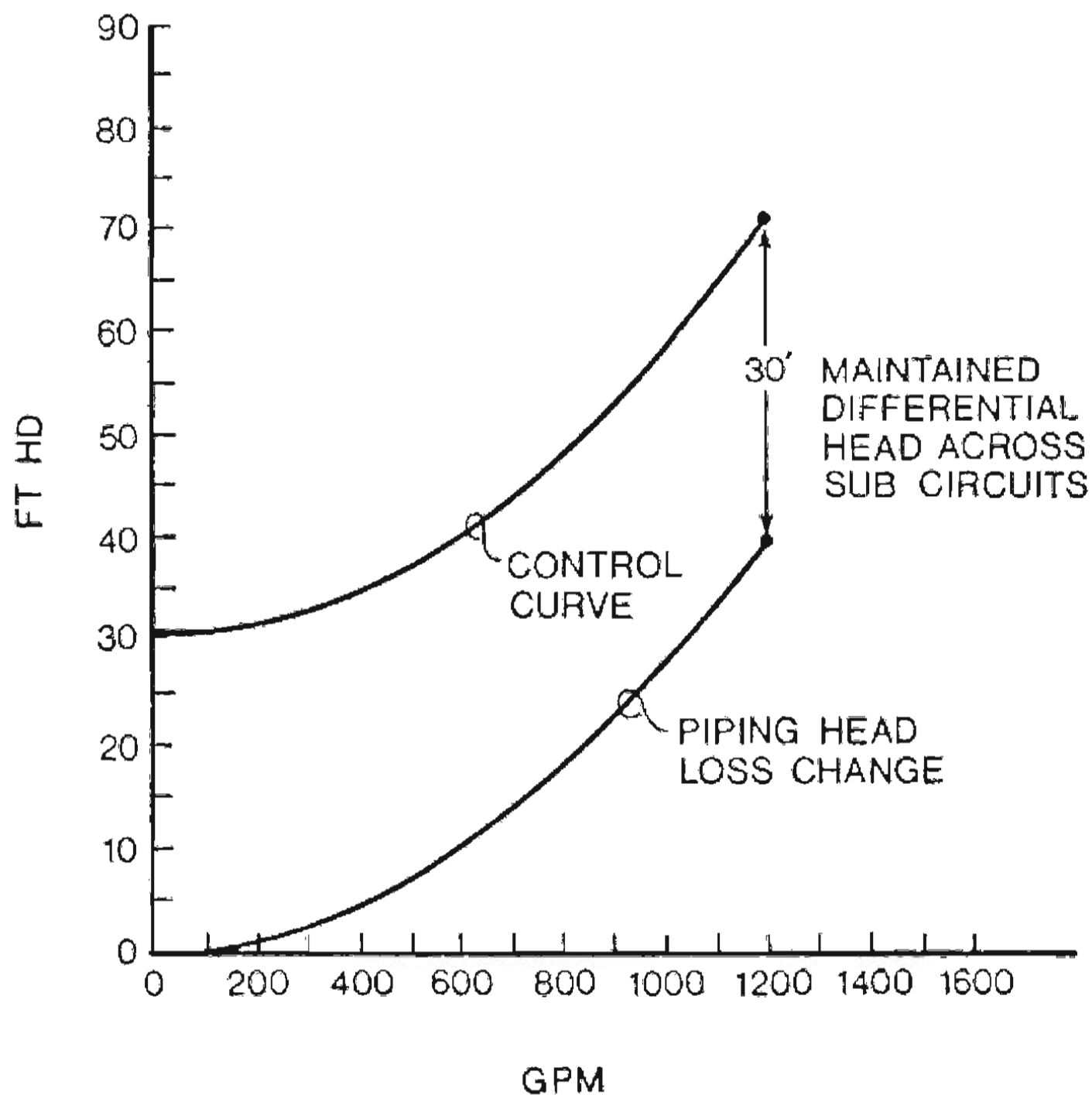


FIGURE 20. Control Curve Development; Constant Differential Added to Piping Head Loss Change Caused by Flow Change

The operating points for the variable speed pump will be determined by its percentage speed curve intersections with the control curve. In order to determine operational points, the variable speed pump curves can be plotted on the control curve or vice-versa. In this case, and for simple illustration, the V/S pump curves will be plotted across the "control" curve as in Figure 21.

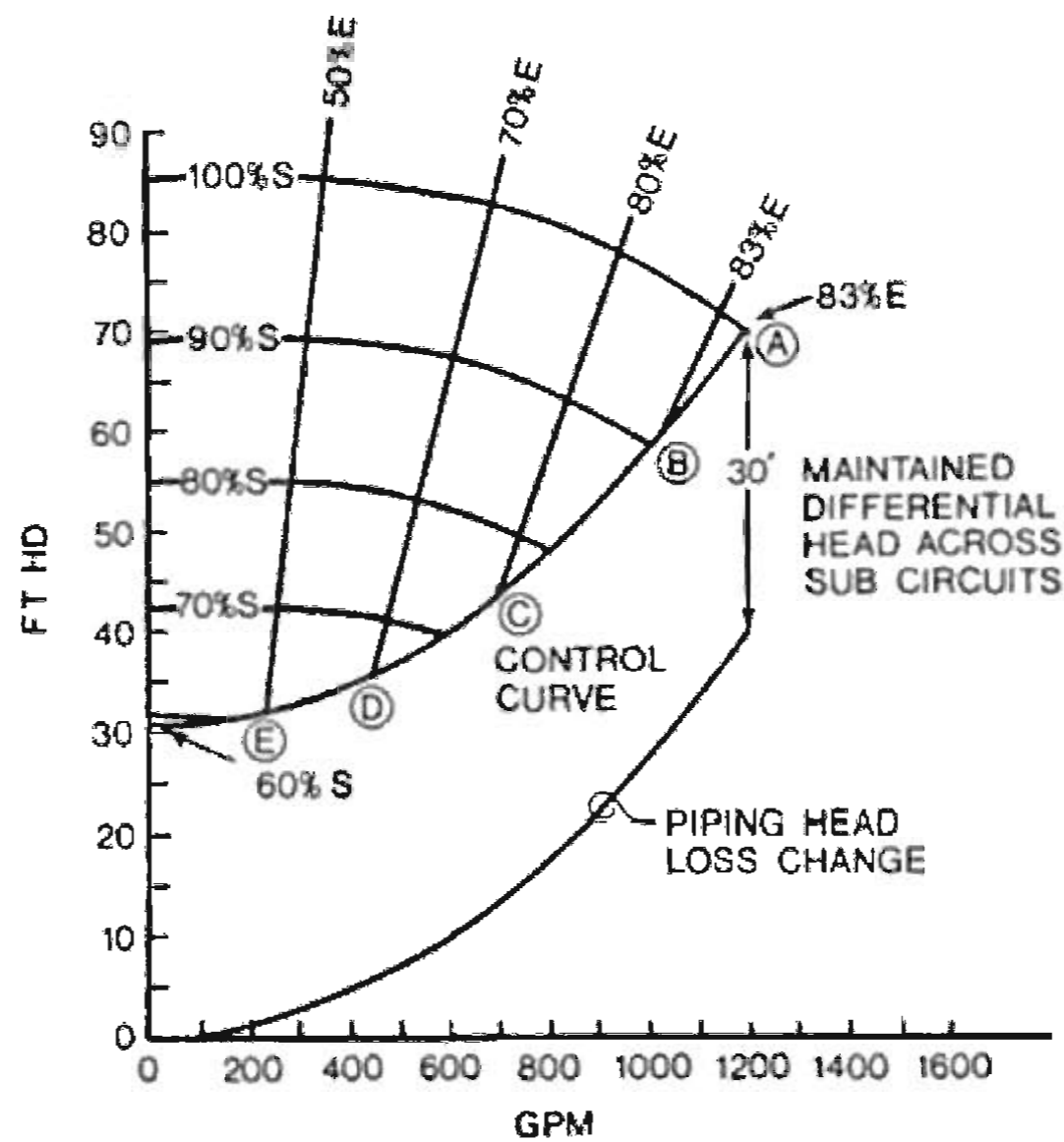


FIGURE 21. Variable Speed Operational Points: A, B, C, Etc.

Variable speed pump operational points are defined at the intersection of the V/S pump curves with the control curve. It will be noted that the minimum pump speed (0 flow) will be at 60% of maximum; 1750 RPM  $\times$  .6 = 1050 RPM. At 800 GPM; system flow, pump speed will be 80% or 1400 RPM and at full load flows (1200 GPM) pump speed will be 1750 RPM.

The power draw curve for the variable speed pump application is established by power draw determined at the V/S, efficiency curves intersection with the control curve. Pump shaft HP draw is assessed as outlined below by use of the basic pump shaft HP draw formula:

$$HP = GPM \times Ft. Hd. / 3960 \times Ep$$

Table 5 shows development of pump shaft HP draw for points A, B, C, D, & E as derived from Figure 21.

TABLE 5

Development of V/S Pump HP Draw with Flow Change Example

POINT	APPROX. %S	EFF	FLOW	HD	PUMP HP	% LOAD (HP)
A	100	83	1200	70'	25.6	100
B	90	83	1000	50'	17.9	70
C	75	80	700	44'	9.72	38
D	67	70	440	36'	5.7	22
E	63	50	230	32'	3.7	14

In order to develop KW draw variation with flow, efficiency must be established for both the V/S drive controller, and for the controlled motor. While difficult to establish, V/S drive controller efficiency seems to follow the relationship shown in Figure 22 since drive efficiency is related to % speed.

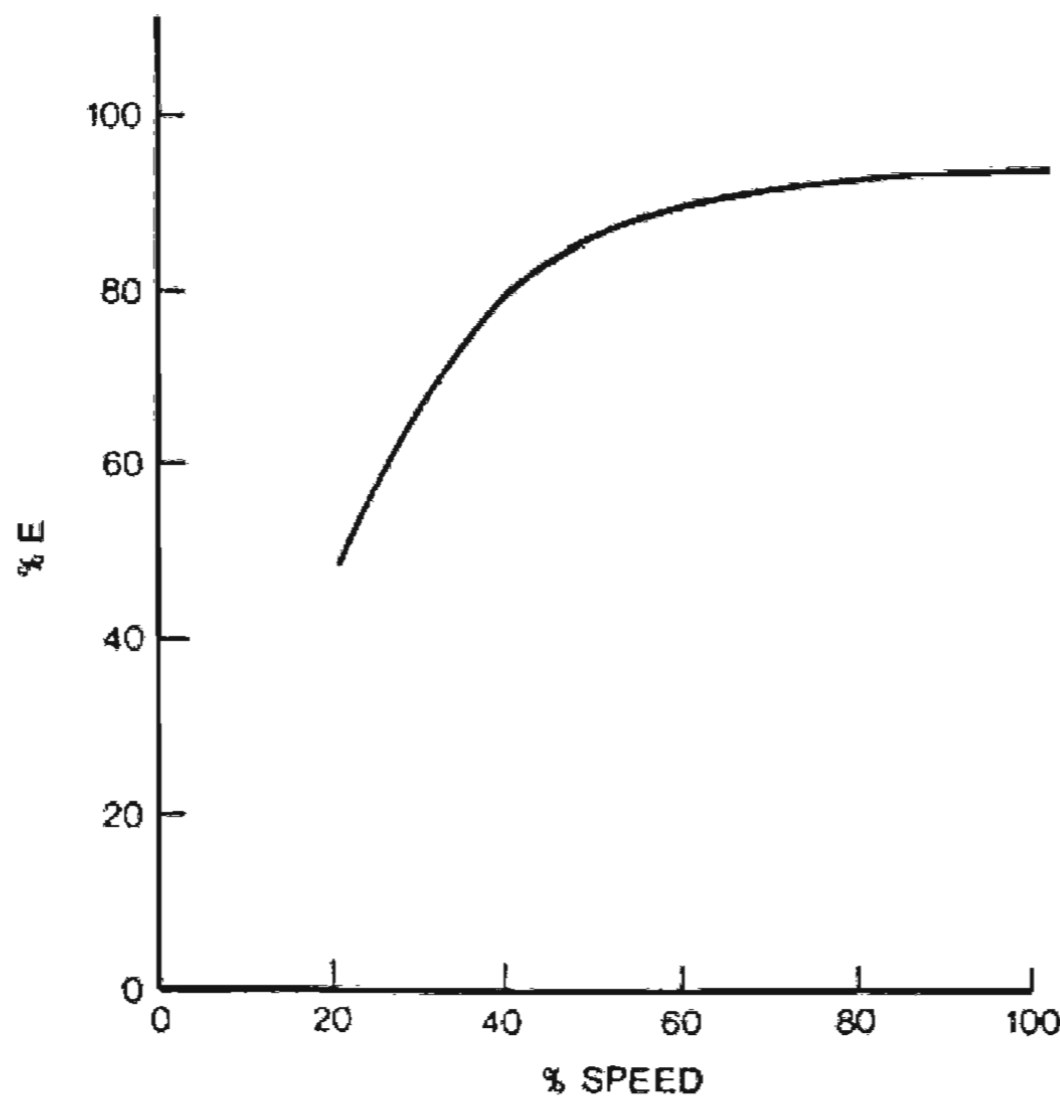


FIGURE 22. Variable Frequency Drive Control Efficiency



Observe that V/S drive efficiency is maintained at high values above 50% speed. Since few V/S drives, as applied to HVAC systems, will operate at speeds less than about 50% full speed, an average drive efficiency of about 92% should be used.

HP loading for the conventional constant speed pump system will seldom drop below 50% of design. Within this range, motor efficiency is maintained close to full load efficiency. For this reason, constant speed motor efficiency change is not taken into account. A similar rationale applies to V/S power draw determination since the power draw at low flow is so low, and operating time in a low flow mode so short that changes in motor efficiency do not make any appreciable difference in the final results.

It should be remembered, however, that for other V/S systems operating for sustained periods at low flow and low motor speed, changes in both motor efficiency and drive control efficiency should be taken into account.

The final calculation, establishing KW draw as a function of flow change is shown below for this example.

TABLE 6

Development of V/S KW Draw with Flow Change

POINT	% S	PUMP E	PUMP FLOW	PUMP HEAD	PUMP HP	MOTOR E	CONTROL E	KW
A	100	83	1200	70	25.6	89	92	23.3
B	90	83	1000	59	17.9	89	92	16.3
C	75	80	700	44	9.7	89	92	8.8
D	67	70	440	36	5.7	89	92	5.2
E	63	50	230	32	3.7	89	92	3.4

The load vs. KW draw is plotted in Figure 23 for the example. The difference in power consumption between the C/S and V/S pump applications is clearly shown.

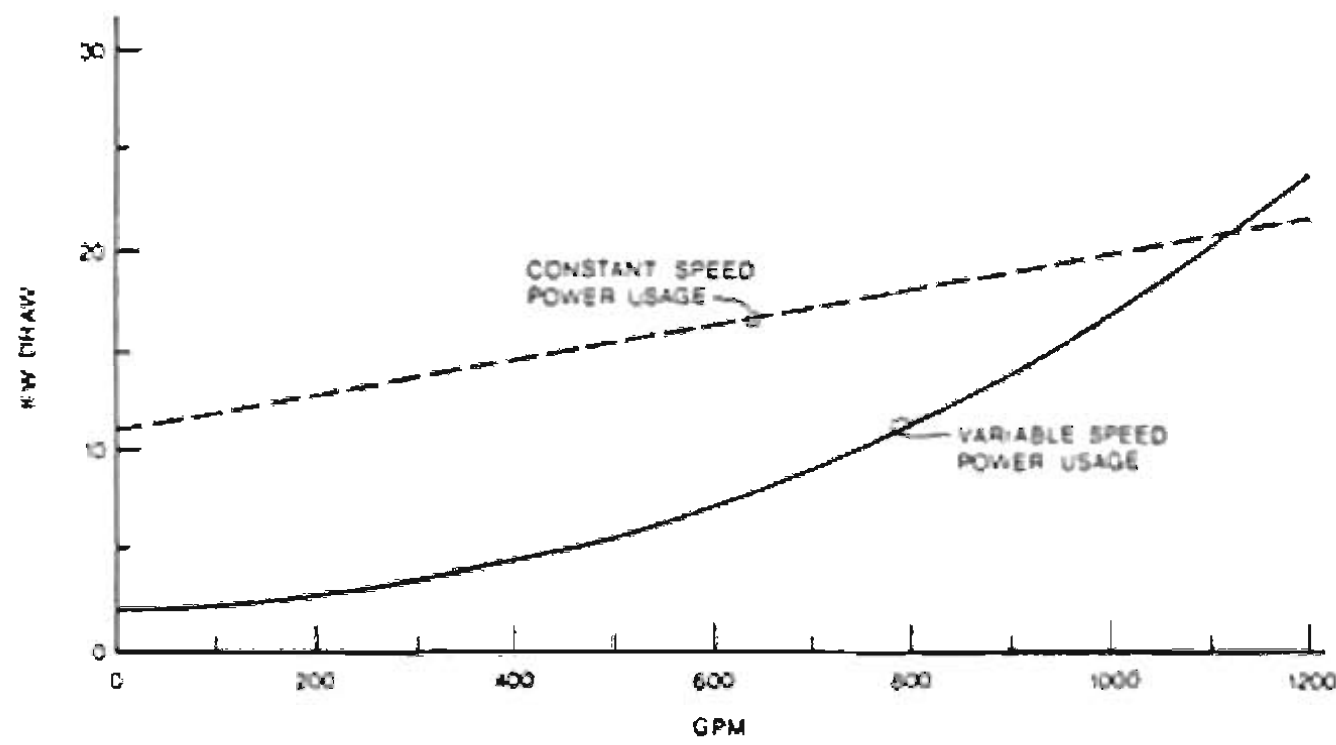


FIGURE 23. Variable Speed Power Usage as Contrasted with Constant Speed

The load profile in the example can now be applied to the flow-power correlation. Final calculation of total KW in the example is shown in Table 7.

TABLE 7

KWH Draw/Year for V/S Example

LOAD PROFILE

% LOAD	% TIME	HRS	GPM	KW	KWH
0-10	2	175	120	2.5	438
10-20	3	262	240	3	786
20-30	5	437	360	4	1748
30-40	15	1311	480	5.5	7210
40-50	20	1748	600	7	12236
50-60	30	2625	720	10	26250
60-70	15	1311	840	13	17043
70-80	5	437	960	16	6992
80-90	3	262	1080	19	4978
90-100	2	175	1200	23.1	4165

TOTAL = 81846 KWH/YR.

8760 HRS/YR. TOTAL

= \$8,184/YR. @ \$.10/KWH

Similar calculations for other piping systems with different pipe lengths (TEL), but otherwise maintaining base conditions\*. Yearly operating cost estimates for the V/S application are shown in Figure 24.

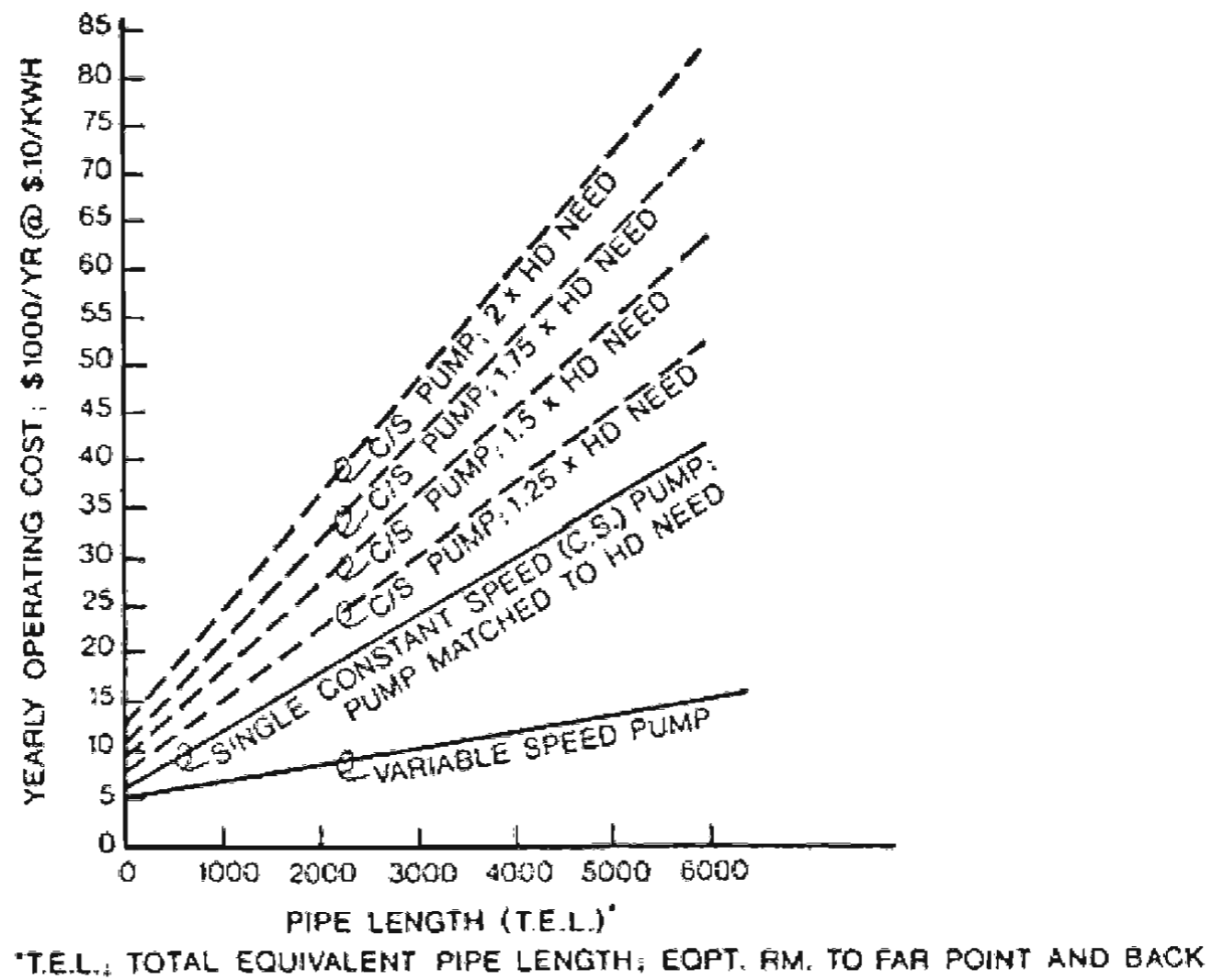


FIGURE 24. Operating Cost Comparison Variable Speed Vs. Constant Speed with Two Way Control Valves

The curves shown in Figure 24 illustrate the very large potential operating cost saving for V/S pump application - especially when piping lengths are long and/or when substituted for overheaded constant speed pumps

\*Base Conditions are:

- (1) 1200 GPM
- (2) Maintenance of 30 ft. differential head across sub-circuits.
- (3) Pipe head loss at 2.5 ft./100 ft. for TEL shown.
- (4) Load profile as previous identified (Table 7).

NOTE: An increase in flow rate to 6000 GPM would increase operating cost by 6000/1200 or 5 times.

An actual KWH cost of \$.05 would decrease cost by \$.05/\$.10 or one-half.

PUMP POWER DRAW FUNDAMENTALS AND DEVELOPMENT OF SYSTEM OPERATING COST FOR PARALLEL CONSTANT SPEED PUMPS

As previously noted, parallel C/S pumps can give substantial savings over single C/S pumps for variable volume systems using two way control valves. Parallel C/S pumps also compare favorably with V/S pumps when system variable head loss (pipe head loss) is a relatively small proportion of total required pump head.

The power draw evaluation for parallel C/S pumps is based on the same conditions as in the example previously used for C/S and V/S, i.e., 70 ft. head at 1200 GPM. The 70 ft. head need is made up of 30 ft. variable pipe friction head loss.

Parallel C/S pumps are generally selected so that each pump provides 1/2 design flow at design head need. In this case, each pump would be selected for 600 GPM at 70 ft.; a 4 BC pump with 9-1/8" impeller diameter operating at 1750 RPM. A 15 HP motor is used.

The parallel application is shown in Figure 25.

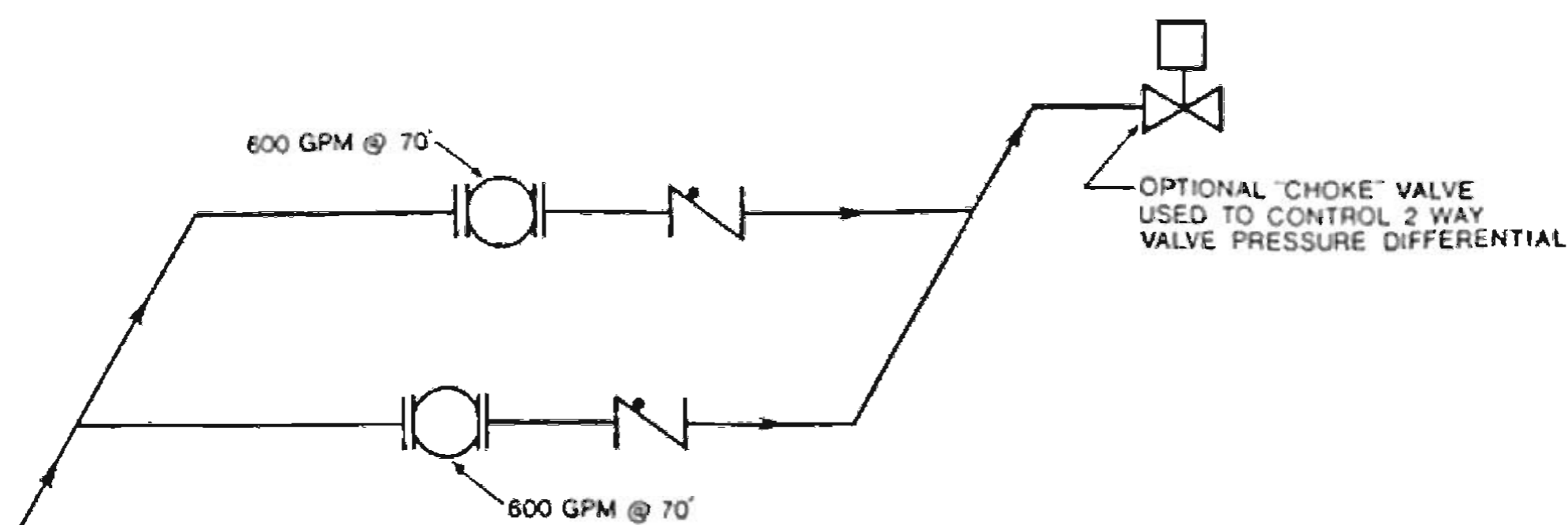


FIGURE 25. C/S Parallel Application Example

When parallel pumps are equal in size, when operating, each pump will provide 1/2 system flow at identical heads. This is shown in Figure 26. Point "A", illustrates that the single pump will provide 500 GPM at 75 ft. head; when both pumps operate, the combination will provide 1000 GPM at 75 ft. head; "A"<sub>2</sub> in Figure 26.

Figure 26 also illustrates piping head loss change with flow. Flow will change because of the operation of the two way valves. Piping head loss in this example is 40 ft. at 1200 GPM. Since head loss varies as the square of flow change, at 1/2 flow (600 GPM) piping head loss will become 10 ft.;  $(1/2)^2 \times 40 \text{ ft.} = 10 \text{ ft.}$

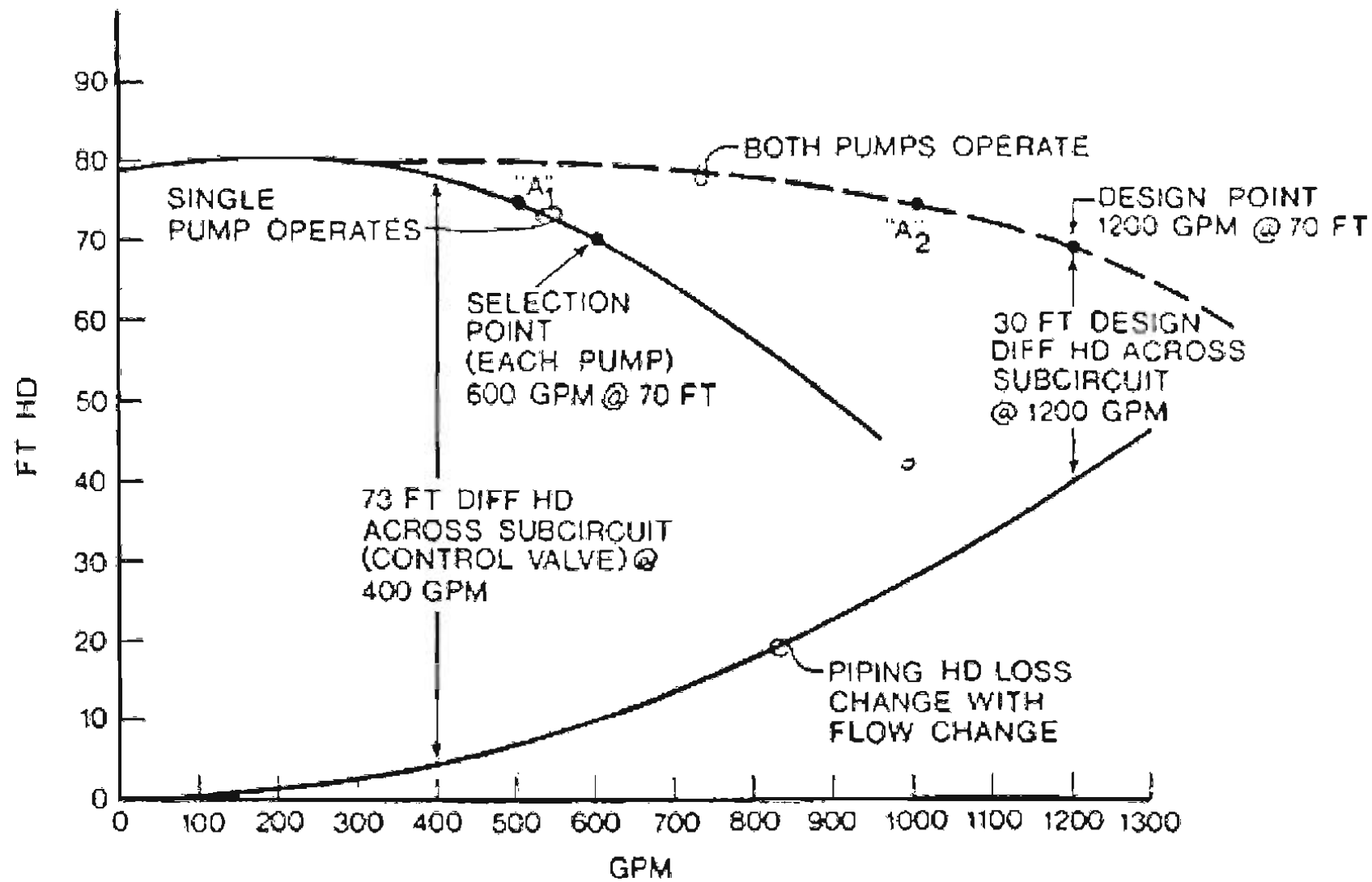


FIGURE 26. Parallel Pump Application of Variable Volume Example

The problem illustrated in Figure 16 is that the two way control valve is subject to increasing differential head as the valves reduce system flow. This can cause operational difficulties.

A commonly used solution for this problem is to apply a "choke" valve at the paralleled pump discharge as shown in Figure 25. The choke valve is controlled by a differential pressure sensor taken across a system sub-circuit in the same manner that differential pressure control taps are taken across sub-circuits for V/S application. As flow decreases, the differential pressure sensors will detect the differential pressure rise, and the choke valve will move to close in order to maintain set control pressure. The choke valve thus substitutes its pressure drop for the pressure drop change occurring in the piping distribution system as flow changes - and maintains a set pressure drop across its controlled sub-circuits. This eliminates the two way control valve pressure differential rise problem.

The use of the choke valve illustrates the "control curve" as applied to parallel pumping. Control curve construction is exactly the same as for V/S application. In this case, 30 ft. is the maintained head difference, and this constant head difference is added to the piping head loss curve to establish the control curve as shown in Figure 27.

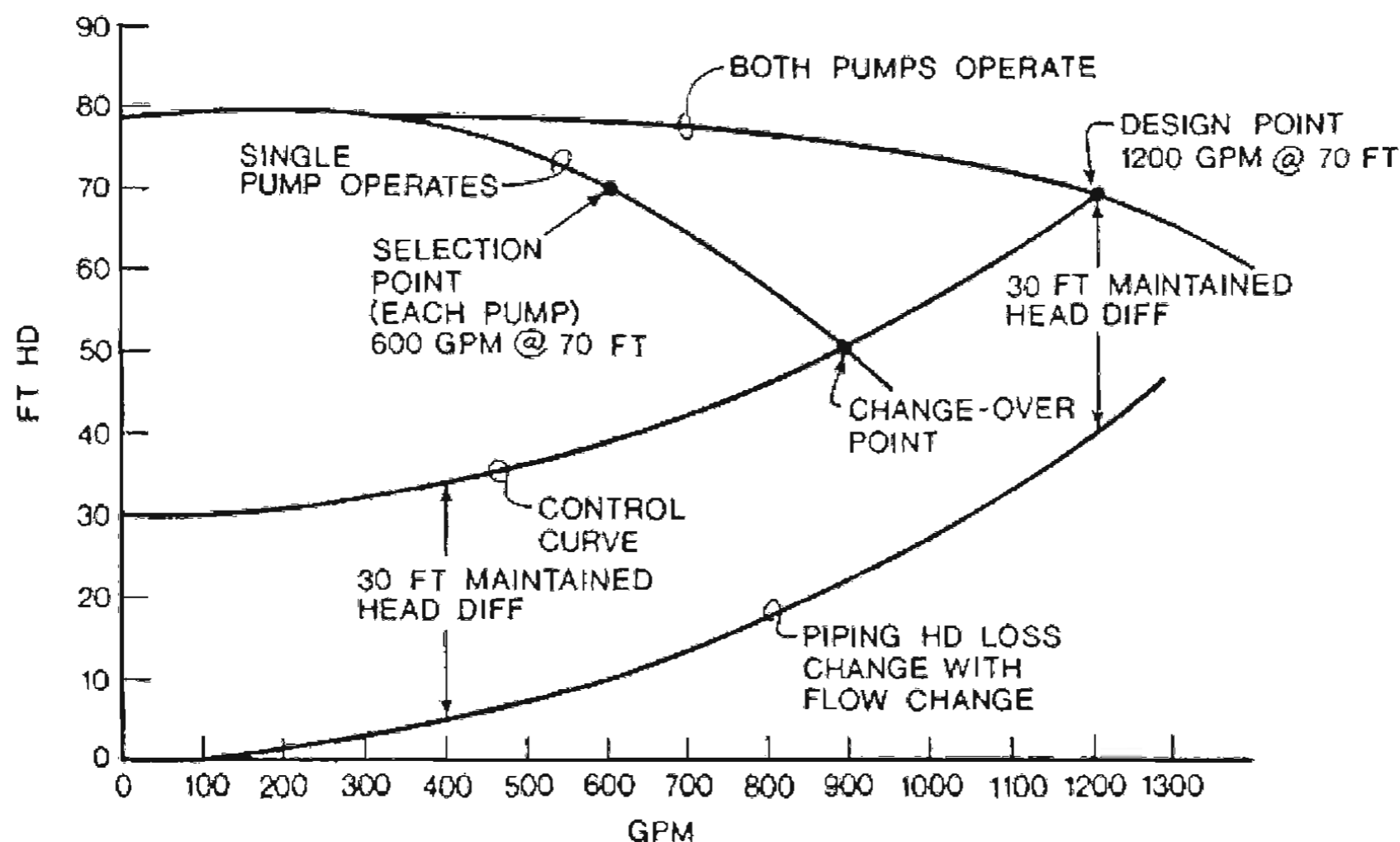


FIGURE 27. Illustration of Control Curve and Change-over Point for Parallel C/S Pump

The change-over point for operation with one pump to operate both pumps is at the intersection of the single pump curve with the control curve. It's important that a "Control Curve" be constructed for parallel pump application to determine the change-over period even when choke valves are not used.

The change-over point should be carefully evaluated in terms of pump selection and motor size in order to insure that the single pump is capable of driving out to this flow point without difficulty. In this example, the change-over point at 875 GPM is considerably in excess of the original "selection intended" point of 600 GPM. This will tend to reduce operating cost since the farther out single pump operation extends in terms of flow, the greater the operational saving.

Having developed the change-over point, the selected pump must be evaluated in terms of applicability. Figure 28 shows the selected single pump curve, and its efficiency. From this curve, we can develop pump shaft HP draw. At 875 GPM (projected change-over point from one pump to two pumps), pump HP draw is less than the proposed 15 HP pump motor selection. Change-over flow rate at 875 GPM is also within operational confines of the pump curve so that the paralleled pumps can be used as described.

Should the above analysis indicate pump motor overload, motor size will have to be increased and/or the change-over point shifted to meet pump requirements. The change-over point may also have to be modified if the proposed change-over point exceeds pump curve capability.

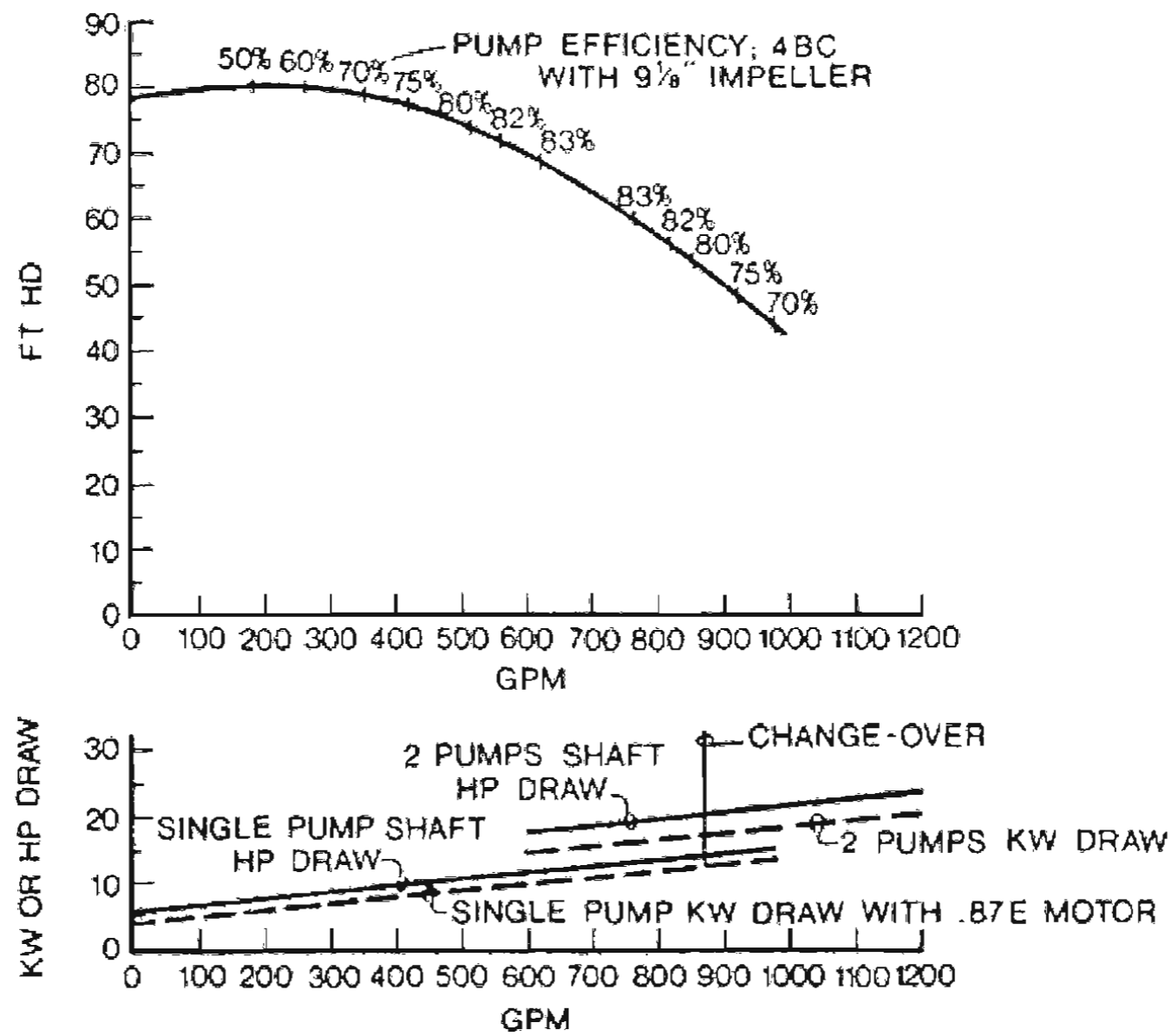


FIGURE 28. Pump Power Draw Analysis from Pump Curve for Single and Parallel Pump Operation

Figure 28 also illustrates total HP and KW draw when both pumps are in operation. When both pumps operate to provide full system design flow (1200 GPM), each pump provides 600 GPM. At 600 GPM, each pump draws 12 HP so power draw at 1200 GPM for two pumps is 24 HP. A similar assessment is made for other flow conditions. KW draw is assessed from pump shaft HP by assuming the 15 HP motor operates at 87% efficiency and that one HP equals .746 KW.

The KW draw for the paralleled pumps with variable volume flow can now be determined by using the single KW draw curve from 0 to 875 GPM to the change-over to two pumps, and then using the two pump KW curve from 875 GPM to 1200 GPM.

The KW draw curves can now be applied to the load profile to determine yearly operating costs as shown in Table 9.



TABLE 9

Load Profile and Power Usage for Parallel Pumps Example

LOAD PROFILE

% LOAD	%TIME	HRS	GPM	KW	KWH	
0-10	2	175	120	5	875	One Pump
10-20	3	262	240	6.5	1703	
20-30	5	437	360	7.5	3278	
30-40	15	1311	480	9	11,799	
40-50	20	1748	600	10.5	18,354	
50-60	30	2625	720	11.5	30,188	
60-70	15	1311	840	12.5	16,388	Two Pumps
70-80	5	437	960	18.5	8085	
80-90	3	262	1080	20	5240	
90-100	2	175	1200	21	3675	

TOTAL = 99,585 KWH

8760 HRS/YR TOTAL

@ \$.10/KWH, YEARLY COST = \$9,958

For the same conditions, yearly operational cost at about \$10,000/Yr. for the parallel pumps compares with operating costs of about \$15,000/Yr. for the single constant speed pump and about \$8,000/Yr. for the V/S pump.

It's a truism that, when properly applied, parallel pumps come close to matching the energy savings provided by the V/S pump. Safety factored overheading of C/S pumps will, however, increase operational cost comparison.

The single V/S pump maintains its advantage, however, since it can adjust its speed to meet actual system requirements.

PUMP POWER DRAW FUNDAMENTALS AND DEVELOPMENT OF SYSTEM OPERATING COST FOR PARALLEL VARIABLE AND CONSTANT SPEED PUMPS.

The economic reason for the use of paralleled V/S and C/S pumps is that the cost of the V/S drive can be reduced. This is evident from the base example thus far used; 1200 GPM at 70 ft. head with 30 ft. as maintained head and 40 ft. as variable pipe loss.

When single V/S pump is applied, a 6 x 8 x 9-34/ H with a 9 $\frac{1}{4}$ " impeller was used for the selection point of 1200 GPM at 70 ft. This pump requires 25.6 HP at design conditions, so a 30 HP motor and V/S drive is needed.

The paralleled C/S example (see page 31 ) illustrates, however, that each pump, selected for 600 GPM at 70 ft., requires only a 15 HP motor. This means that if V/S is applied in parallel, as with a C/S pump, only a 15 HP and V/S drive is needed. The difference in drive cost can be substantial.

Parallel V/S application, in any of its several variations acts the same as a V/S pump operating in parallel with a C/S pump. This is illustrated in Figure 29. The base example 1200 GPM at 70 ft. requires that each pump be selected for 600 GPM at 70 ft.

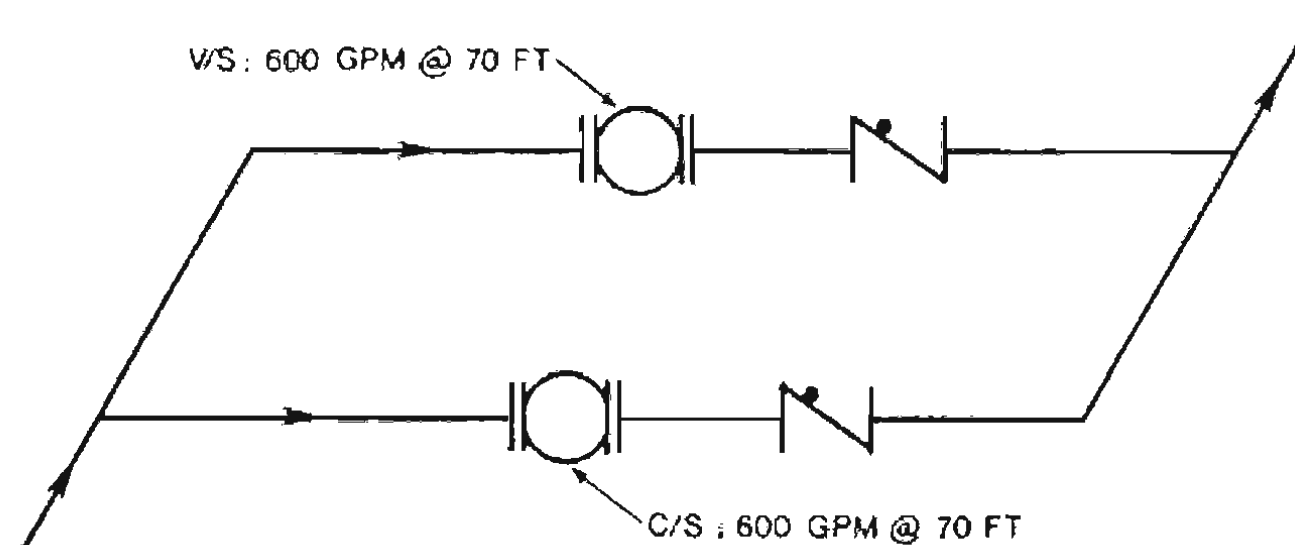


FIGURE 29. V/S Pump in Parallel with C/S

Operation of the parallel V/S pumping system is best illustrated by considering that the V/S pump will operate by itself from no flow to a change-over flow point as determined by the same procedure outlined for parallel C/S pumps.

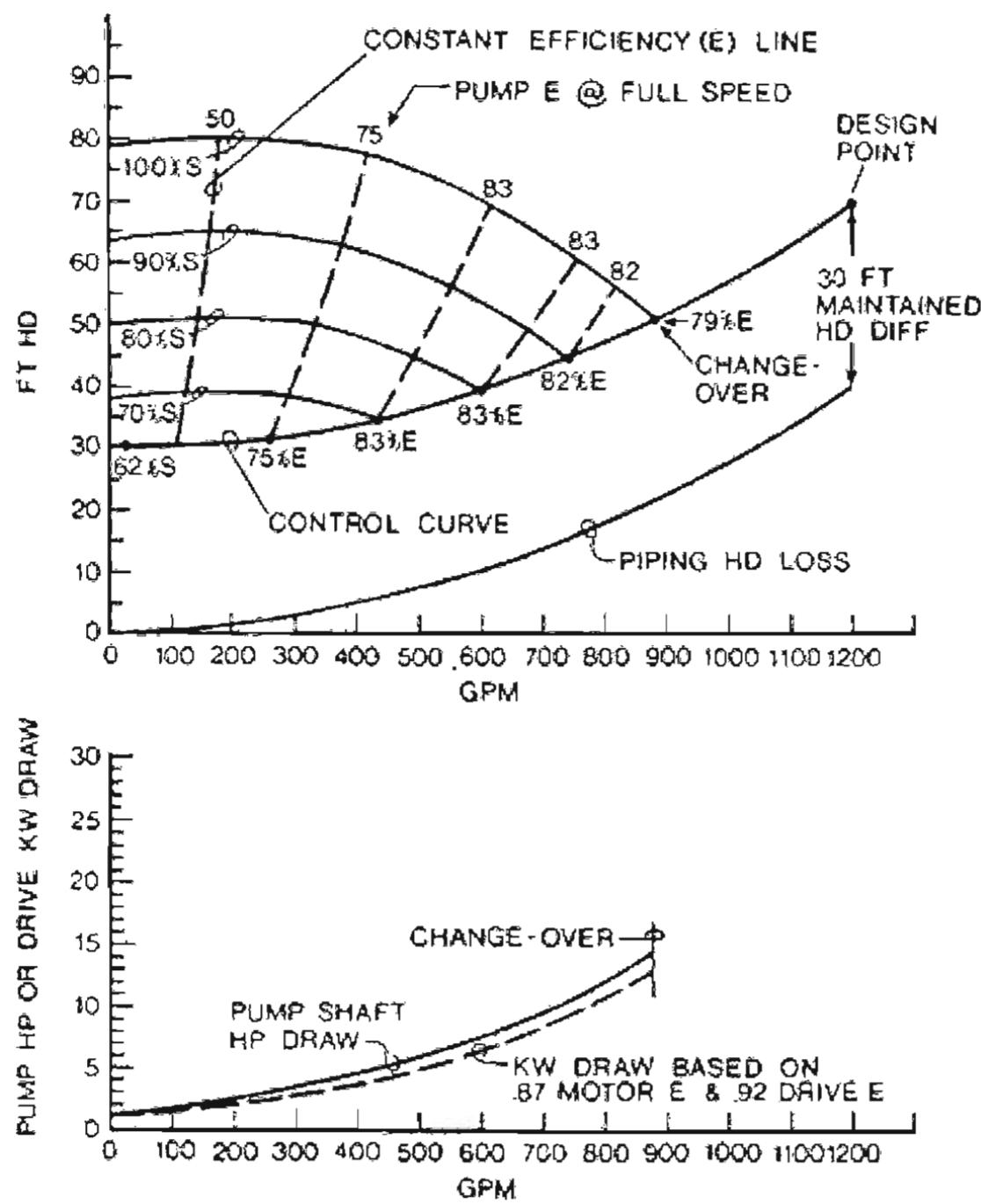


FIGURE 30. Operation of Paralleled V/S Pump at Low Flow, Up to Change-over Point

Power draw for the V/S pump, operating from no flow to the change-over point as shown in Figure 18, is developed in the same manner as previously shown for the single V/S pump.

When the V/S pump reaches 100% speed it can be locked into this speed and operate as a constant speed pump. V/S drive, in this option, is then switched over to the second pump.

In another approach, V/S drive is operated as the base pump at all times. When the V/S pump reaches 100% speed, or a pre-determined change-over point, the second parallel C/S pump, is started.

In either case, once the change-over point is reached, one of the pumps acts as C/S and the other operates as V/S. The V/S pump curve "humps" onto the C/S pump curve and until design flow is reached performs in accordance with basic unequal parallel pump application fundamentals.

To understand how the parallel V/S and C/S pumps operate, unequal parallel pump fundamentals must be understood. The basic idea is shown in Figure 31.

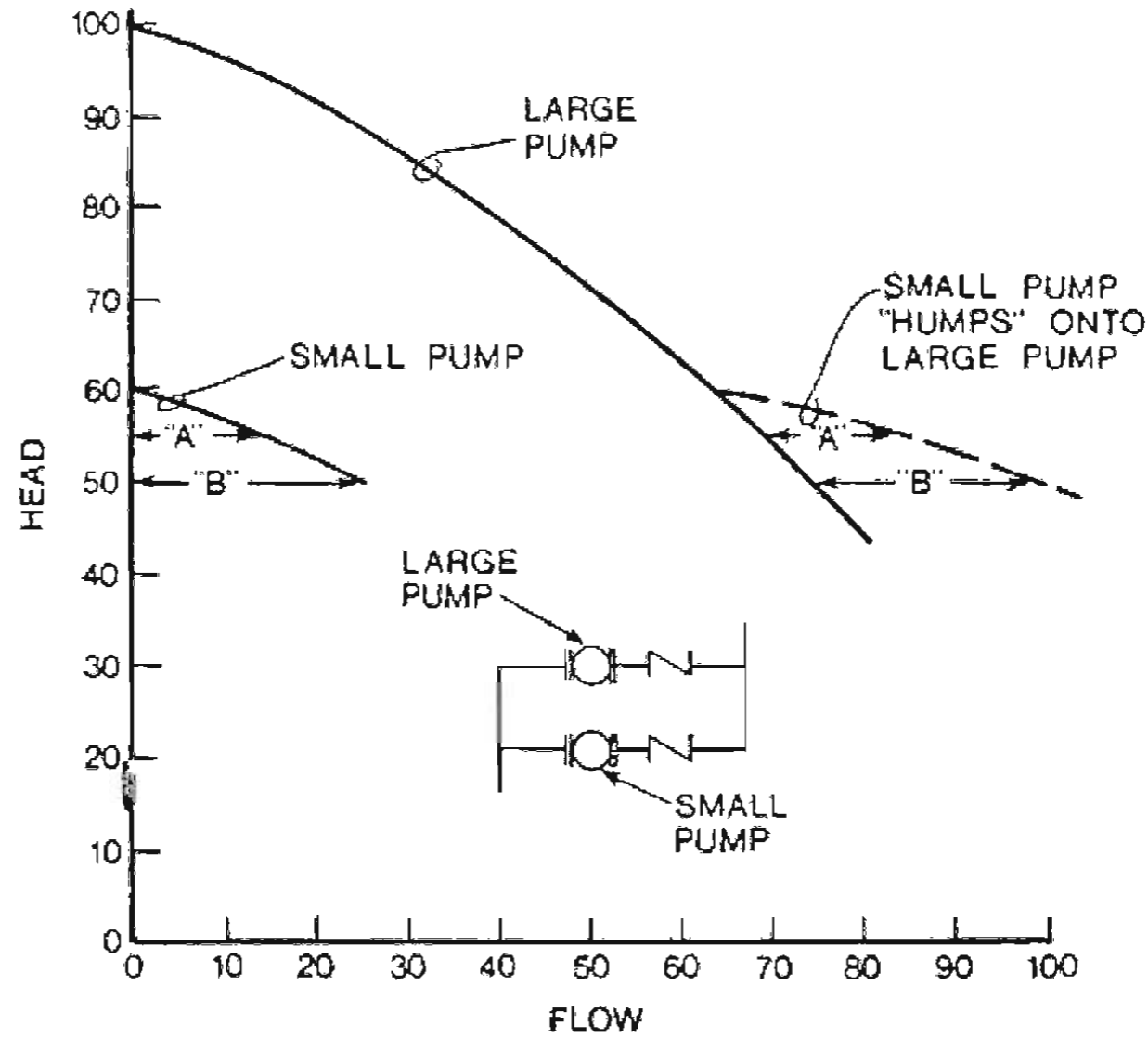


FIGURE 31. Small Lower HD Pump; "Humps" its Flow-Head Curve onto Larger Pump

When parallel, but unequal pumps are used, total system flow is established by adding flow capability for each pump at identical heads. This is shown in Figure 31, where at 55 ft. head, the large pump provides 70 GPM. At this head, the small pump provides 15 GPM as shown by "A". Total flow of 85 GPM is established by adding 15 GPM (small pump) to 70 GPM (large pump).

85 GPM will be delivered to the system. However, if system flow decreases below 65 GPM, the small pump cannot operate because its check valve will close. This is because at system flow rates less than 65 GPM, small pump head (at shut-off) is less than the head developed by the large pump.

The parallel V/S, C/S pump combination works in the same manner. In this case, the V/S pump curves "hump" onto the C/S curve as shown in Figure 32, during operation after the change-over point.

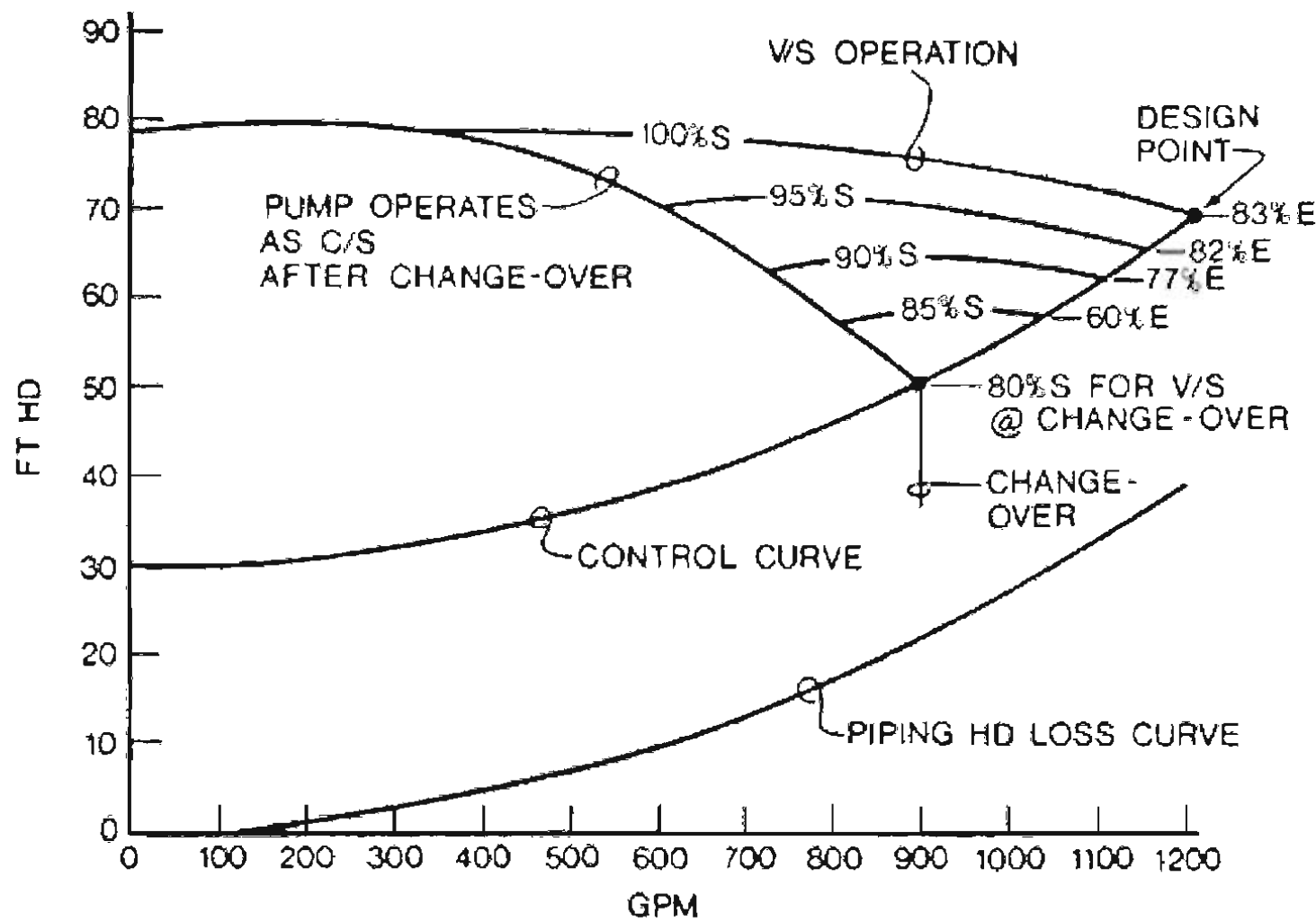


FIGURE 32. At High Flow Need; V/S Pump "Humps" onto C/S Pump Curve

It is of interest to note, that after change-over is reached and with increasing system flow, V/S pump flow increases while C/S pump flow decreases.

Power draw for the V/S parallel pump combination is similar to power draw established for the single V/S pump. While theoretical savings are equal to the single V/S pump, there are several important concerns regarding parallel V/S pump application.

The parallel V/S application works only because one of the pumps is treated as C/S. Each pump is equal in size. If a safety factor head is applied to the selection of these pumps, the change-over flow point may not be predictable under this condition. Unless the pump impellers are trimmed to match the pumps to the system, it may be necessary to increase the maintained head differential in order to establish workable change-over control points. This will waste pump energy.

Paralleled V/S application is not self correcting as is the single V/S application because one of the paralleled pumps must finally be treated as C/S.

Any paralleled V/S application requires careful attention be paid to switch-over flow points, both with increasing and decreasing flow modes. Unless care is taken, the V/S pump check may be closed by a higher head C/S pump. The V/S pump will then spin against a closed discharge.

All in all, the V/S parallel application requires much more attention to design detail than the more self compensating single V/S pump application.

The problems inherent to overheaded C/S, V/S parallel pump combination are best illustrated by an example. In this example, the V/S, C/S pumps are each selected for 600 GPM (1250 GPM) at 137 ft., even though head requirements are only 70 ft. The selected pump is 4CB, 12½" impeller @ 1750 RPM as shown in Figure 33.

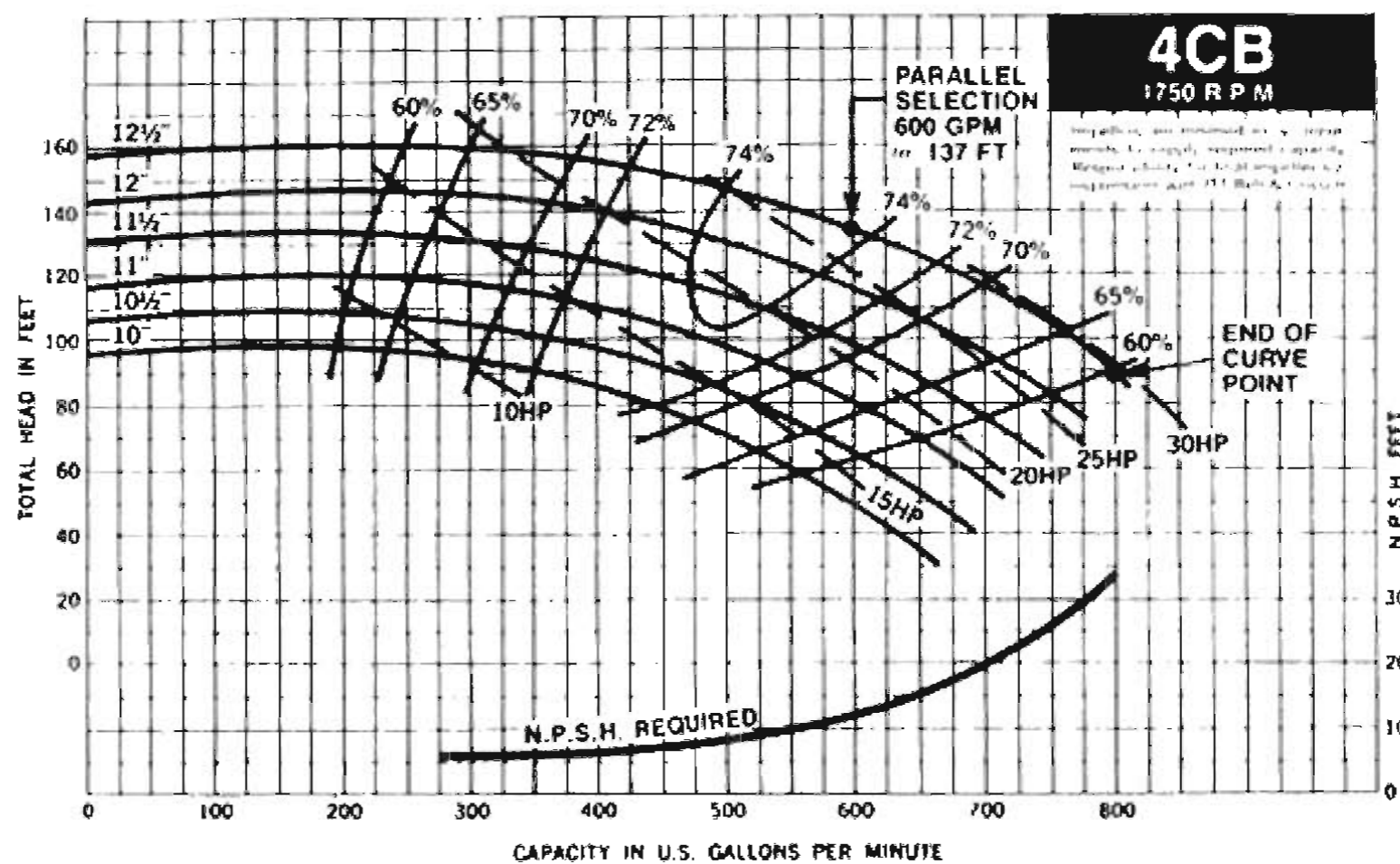


FIGURE 33. Overheaded Pump Selection for Parallel C/S, V/S Operation 4CB W/12½" Impeller

Parallel operation is intended as conventional. The V/S pump operates to 100% s, at this point the C/S pump is started and the V/S pump then operates as previously described and intended.

The problem is that the V/S pump cannot operate beyond 60% speed before it runs out of flow capacity. This is shown in Figure 34.

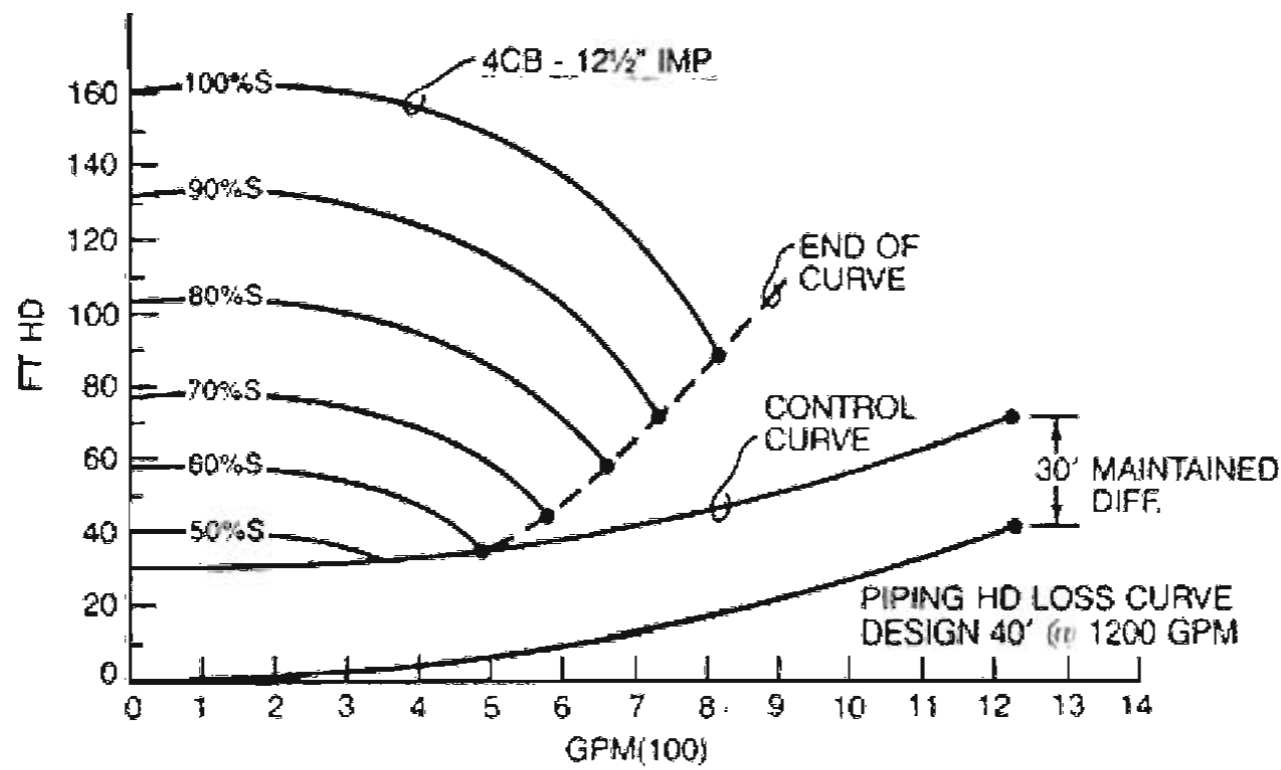


FIGURE 34. Overheaded Pump Runs Out of Flow Capacity;  
Exceeds Curve End at Over 60% Speed

In Figure 34, the maximum intended flow for this pump is shown by the "end of curve" plot. This curve plot is derived from the end of the end of the curve point taken from 100% speed and for the selected pump (see Figure 33). It is true that the pump can operate beyond the "end of the curve" given sufficient pump suction pressurization. The pump is then operating in an unknown operational area. Operation beyond the end of the curve is not recommended because of potential operational problems.

In the situation just described, the parallel V/S, C/S pump combination cannot operate as intended because of the pump system mis-match. There are two solutions to the problem the first of which wastes energy:

- (1) Increase the maintained head difference. In this case, the maintained head difference would be increased from 30 ft. to about 70 ft. This may require re-balancing, but will allow the pumping system to operate as intended. This approach will cause a significant reduction of pump energy saving because maintained head difference has increased. This unsatisfactory solution is shown in Figure 35.



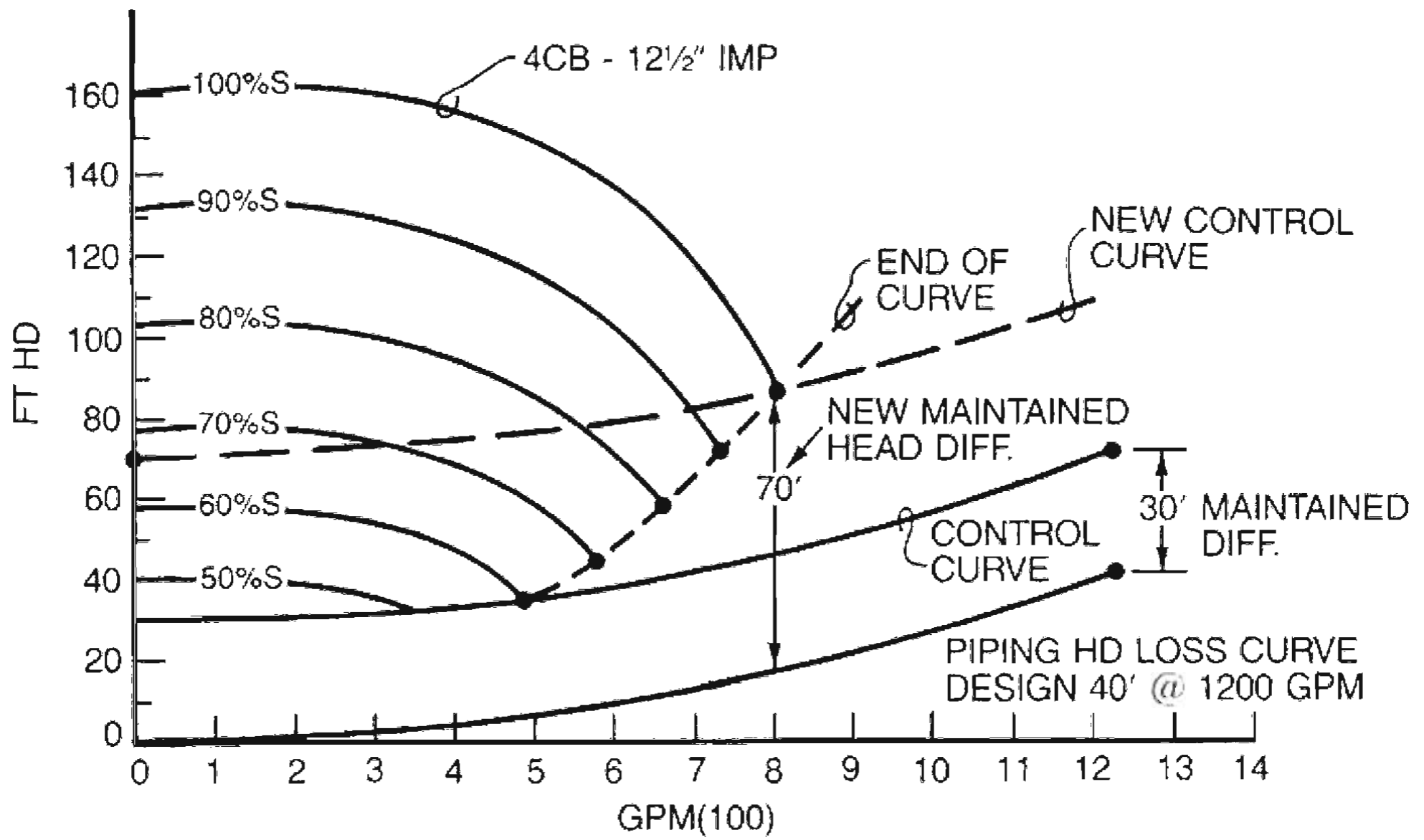


FIGURE 35. Energy Wasting Solution to Parallel Pump "End of Curve" Problem; Increase Maintained Constant Head Difference

(2) The best solution is to match the pump to the system. In this case, this can be accomplished by selecting the pumps to satisfy actual parallel pump need of 600 GPM (1200 GPM total) at 70 ft.

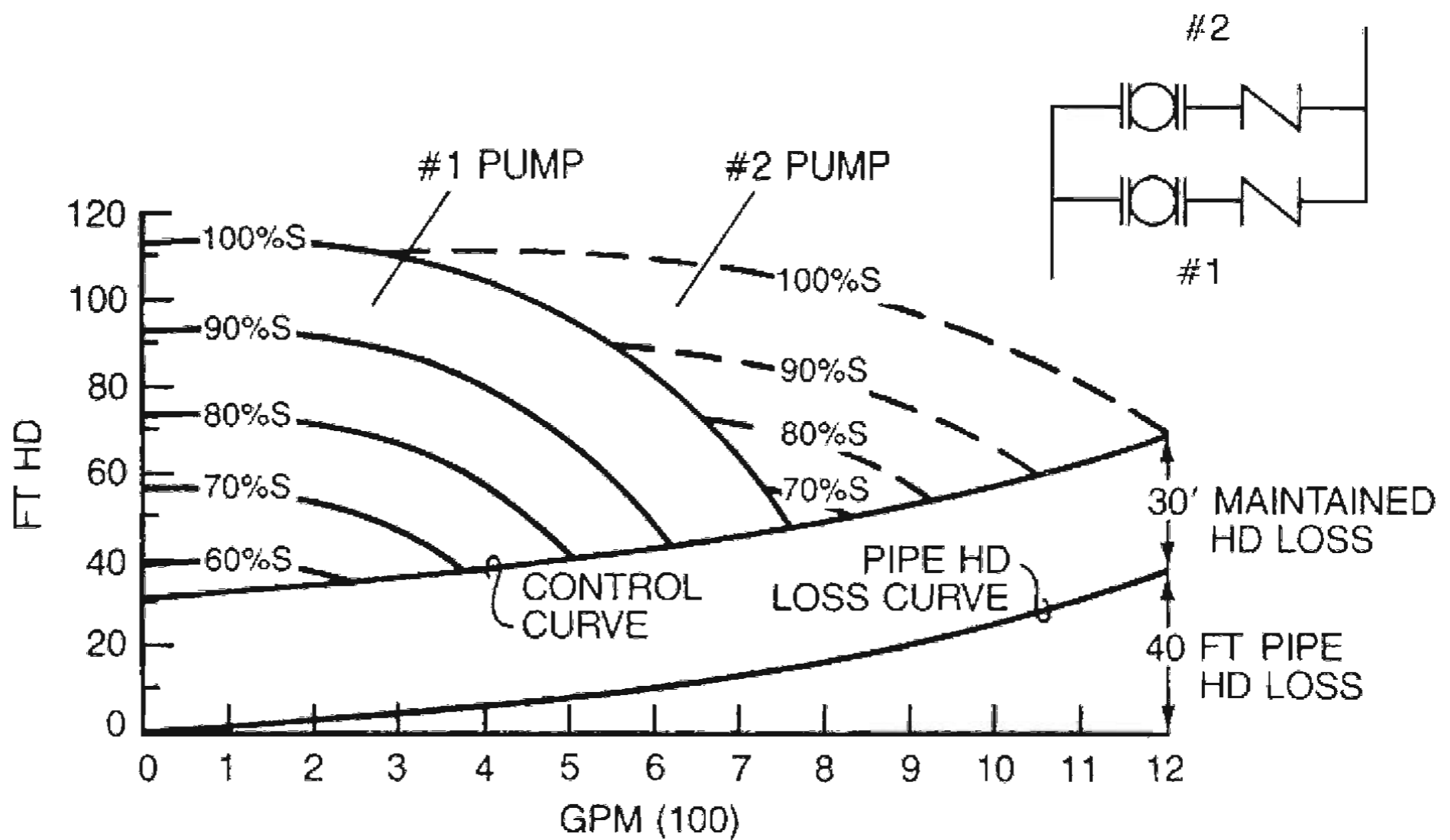


FIGURE 36. Best Solution to Parallel Pump "End of Curve" Problem; Match Pump to System and Avoid "End of Curve" Problem

It's important to remember the "end of curve" pumping problem can occur when V/S drives are used in any parallel combination; whether all V/S - or V/S in combination with C/S as described.

In some cases, particularly when multiple parallel V/S pumps are used, each with their own V/S drives, it may be necessary to establish a second pump start by a "curve end" operational signal from an operating pump. The curve end signal can be provided by percentage of speed integrated with pump differential head or flow.

## 10. V/S OPERATING SAVINGS AS AFFECTED BY APPLICATION VARIABLES

Actual V/S operating HP savings will be affected by a number of additional important variables which are:

- ... Location of pressure differential taps for controlling pressure sensors and transducer as affected by piping design.
- ... High load flow balance.
- ... Correlation of flow with load, especially at low system load conditions.
- ... System pick-up after shut-down as affected by piping system design.

The effects of these variables on V/S operating costs will be discussed next with reference to system diagrams. The variables are inter-related and their effects vary according to the V/S application.

### 10A. MAJOR DIFFERENCE BETWEEN REVERSED RETURN AND DIRECT RETURN PIPING SYSTEMS IN TERMS OF V/S PUMP APPLICATION

The major difference between reversed and direct return piping application to V/S operation has to do with pick-up after shut-down.

The properly balanced reversed return piping system lends itself to even pick-up after weekend shut-down. This is because at full load pick-up conditions, each terminal unit will receive its proportionate share of available flow.

The direct return piping system, on the other hand, may not provide even pick-up after shut-down for a weekend. This is because of a terminal sub-circuit balance problem, described in detail in the discussion concerning pressure differential sensor locations in direct return piping.

Uneven pick-up for direct return piping systems means that adequate flow will not appear at the "far" circuits until after a time lag and only when the near circuits are satisfied and their control valves start to move towards closure. The pick-up problems do not occur in direct return piping systems, where there is no continuous operation with no shut-down as in hospitals.

10B. LOCATION OF PRESSURE DIFFERENTIAL CONTROL TAPS AND HIGH LOAD FLOW BALANCE IN A REVERSED RETURN PIPING SYSTEM

The piping diagram shown in Figure 37 illustrates the reversed return V/S pumping system. Each sub-circuit, containing a terminal unit and its two way control valve (A-D, C-D and C-F), is balanced against its matching sub-circuits. As an example; if sub-circuits C-D has 30 ft. head loss at design flow while sub-circuits A-D and C-F have 15 ft. head loss at their design flow rate, the lower head loss sub-circuit will require an additional 15 ft. head loss to obtain balance. Such balance allows for even pick-up after week-end shut-down and helps prevent terminal unit sub-circuits from "stealing" flow under heavy load draw conditions.

Since flow in the system will be highly variable, the chillers are separately pumped, using primary-secondary techniques.

One of the requirements for V/S application in most HVAC systems is that each terminal unit have full load capability at all times. In order to drive full flow through any terminal the necessary differential head must be maintained as a relative constant by use of the differential pressure control sensors shown in Figure 37.

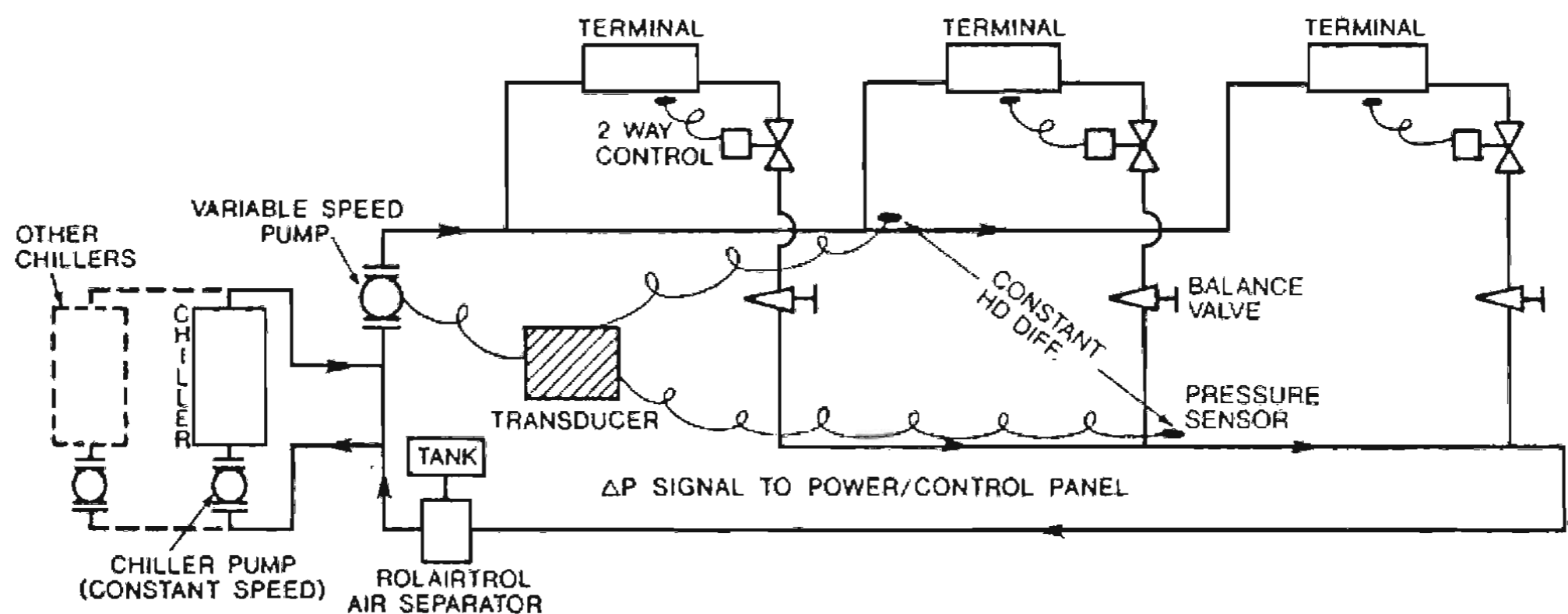


FIGURE 37. V/S Pumping System Pressure Differential Control Tap Location For Reversed Return Piping; Note -  $\Delta P$  Sensor Tap Location in Center of Circuit

Past experience with C/S HVAC piping system design has been to favor reversed return piping over direct return. This is because, if all terminal sub-circuits have the same head loss, the piping circuit will be balanced. When terminal sub-circuit head losses are different, simple adjustment of the balance valves will provide for required full load flow balance. The "common sense" rule of thumb to design the piping system as reversed return\*, would seem to indicate that it does not make any difference for V/S pumping as to which sub-circuits the differential pressure sensors are located across. In following this rule, the logical location would be across the closest terminal sub-circuit, A-B in Figure 37, since this would be the easiest and least costly to install.

Unfortunately, reversed return piping, when applied to V/S application does not necessarily follow the "common sense" rule outlined above. Some problems may occur in a reversed return piping circuit when sub-circuit loading varies greatly during a specific time period. As an example, when V/S differential control is applied across the first sub-circuit, and that circuit load drastically decreases relevant to other sub-circuits operating at high load, available differential head across the last sub-circuit will decrease. This is because supply line head loss will become greater than return line head loss. Depending on load requirements, the decrease may be enough to affect full load operation of that last sub-circuit. Placement of the tap points across the far circuit will not help because the basic problem is simply "mirror imaged".

Relocation of the controlling pressure differential taps to the center sub-circuit (C-D in Figure 37) provides a solution because the "center" tap location tends to equalize terminal circuit differential head despite unequal supply and return line head loss changes.

It should be emphasized that the potential problem is probability factor oriented.

The problem will be minimized when:

- (1) All terminal unit loads are expected to change by about the same ratio with respect to time.
- (2) The total head loss for the supply line plus return line is equal to or less than the terminal sub-circuit head loss used to set differential head control.

\*While this basic design rule is widely accepted, it is true that a direct return piping system, designed with knowledge, can be superior to a reversed return piping design used only because of the widely believed "rule" that reversed return must provide flow balance. A complete discussion of direct return vs. reversed return is in the B&G reprint entitled: "Hydronic Systems, Analysis and Evaluation".

The described problem will be maximized when:

- (1) Terminal unit loads (flow need) changes drastically within a given time as with two way ON-OFF control for terminal sub-circuits as an example.
- (2) The total head loss for the supply line plus the return line greatly exceeds the terminal sub-circuit head loss used to set differential head control. Should this ratio exceed the order of 2.5 to 1, it may be necessary to use two sets of differential pressure control tap points; one across the first sub-circuit, the second across the last sub-circuit. The differential controls in this case are over-riding. Should either control detect a less than required differential head, it will over-ride the other control and re-establish required differential by increasing pump speed.

10C. LOCATION OF PRESSURE DIFFERENTIAL CONTROL TAPS AND HIGH LOAD FLOW BALANCE IN A DIRECT RETURN PIPING SYSTEM

A direct return piping system is shown in Figure 38.

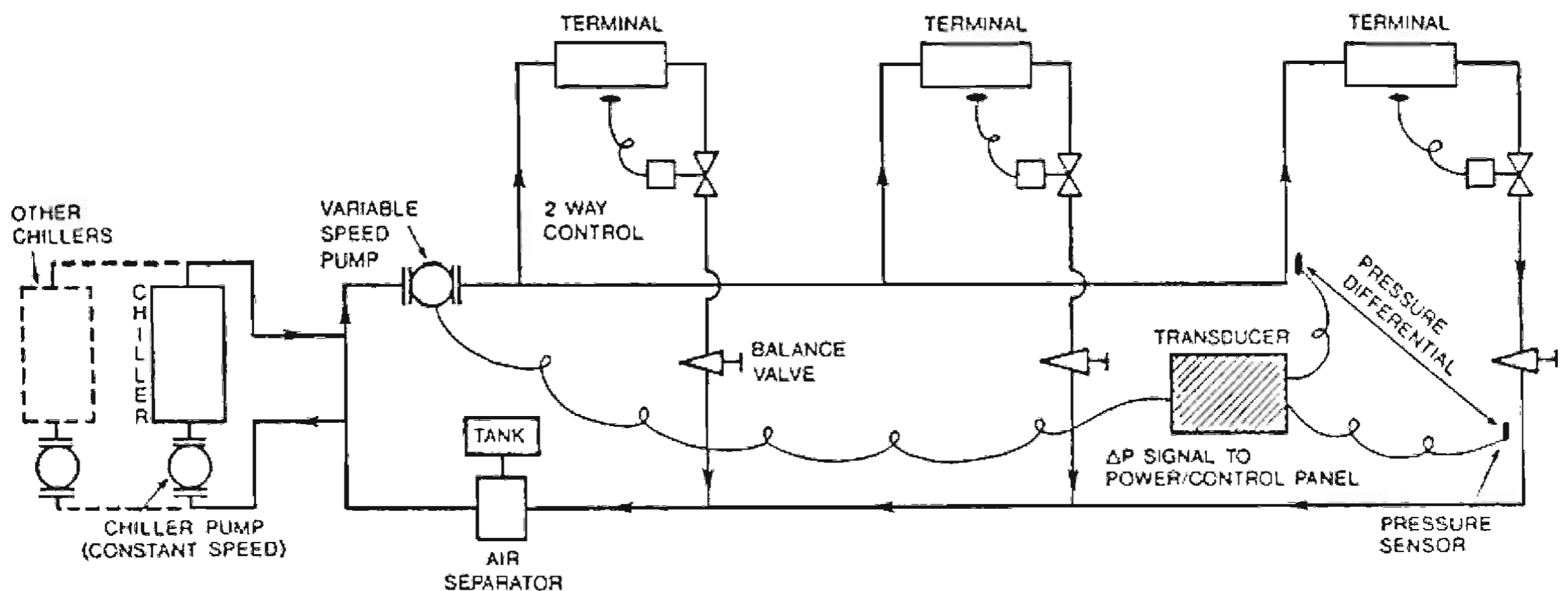


FIGURE 38. V/S Pressure Tap Location in a Direct Return Piping System

The direct return piping system generally requires that pressure differential sensors be placed across the farthest or last terminal. The terminal sub-circuits are balanced against each other, so the same head loss at design flow is maintained for all sub-circuits. In this type system, for V/S pumping, piping head loss is not taken into account in balance. The reason for this follows.

Figure 39 illustrates a direct return system conventionally balanced when C/S pumps are used. Each sub-circuit has 20 ft. head loss at design flow, and each supply and return pipe section has 4 ft. head loss at design flow. The circled numbers indicate the required energy head at junction points at full flow. The number adjacent to the balance valves illustrates their head loss setting for conventional C/S balance at design flow.

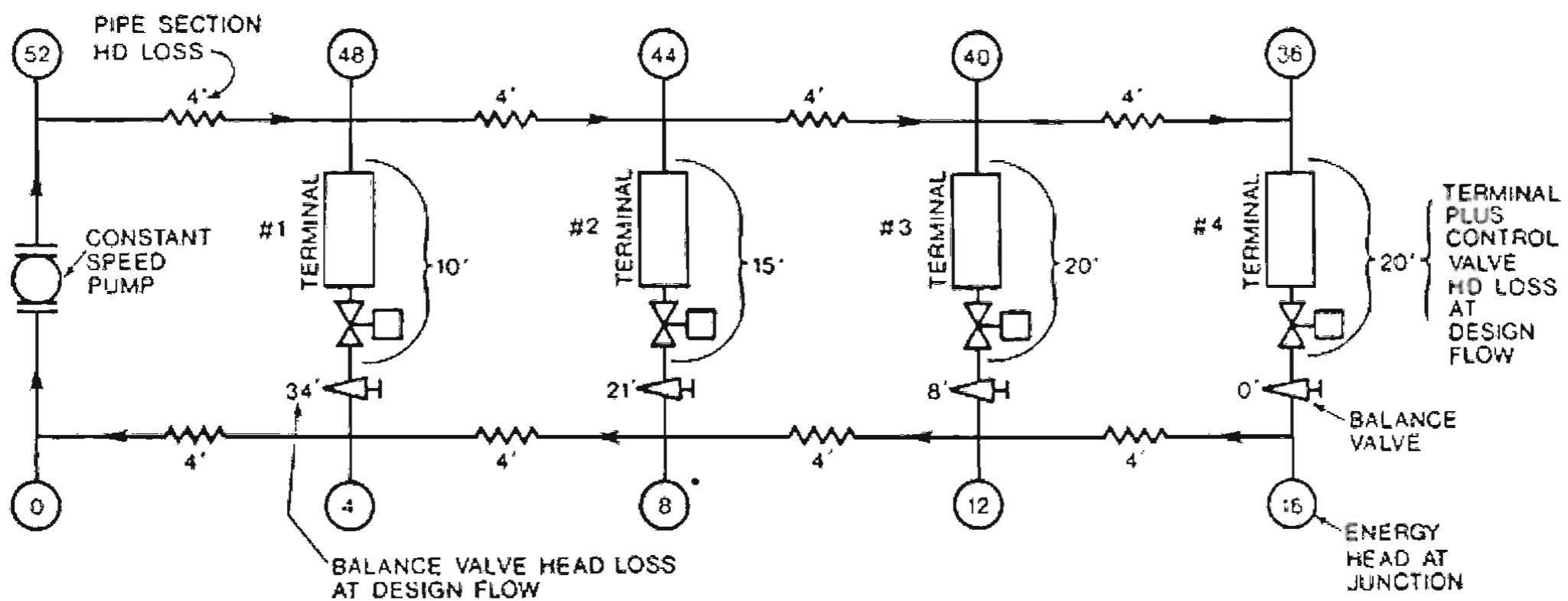


FIGURE 39. Balance Valve Setting for Constant Speed Pump and for Direct Return Systems

Now assume that a V/S pump is to be applied to the "balanced" circuit shown in Figure 11. The differential pressure sensor points are taken across the "far" circuit, and control is set for a maintained 20 ft. differential at this point. The system will operate satisfactorily at full flow, but operational difficulties may occur at low loads, as shown in Figure 40 at 50% flow.

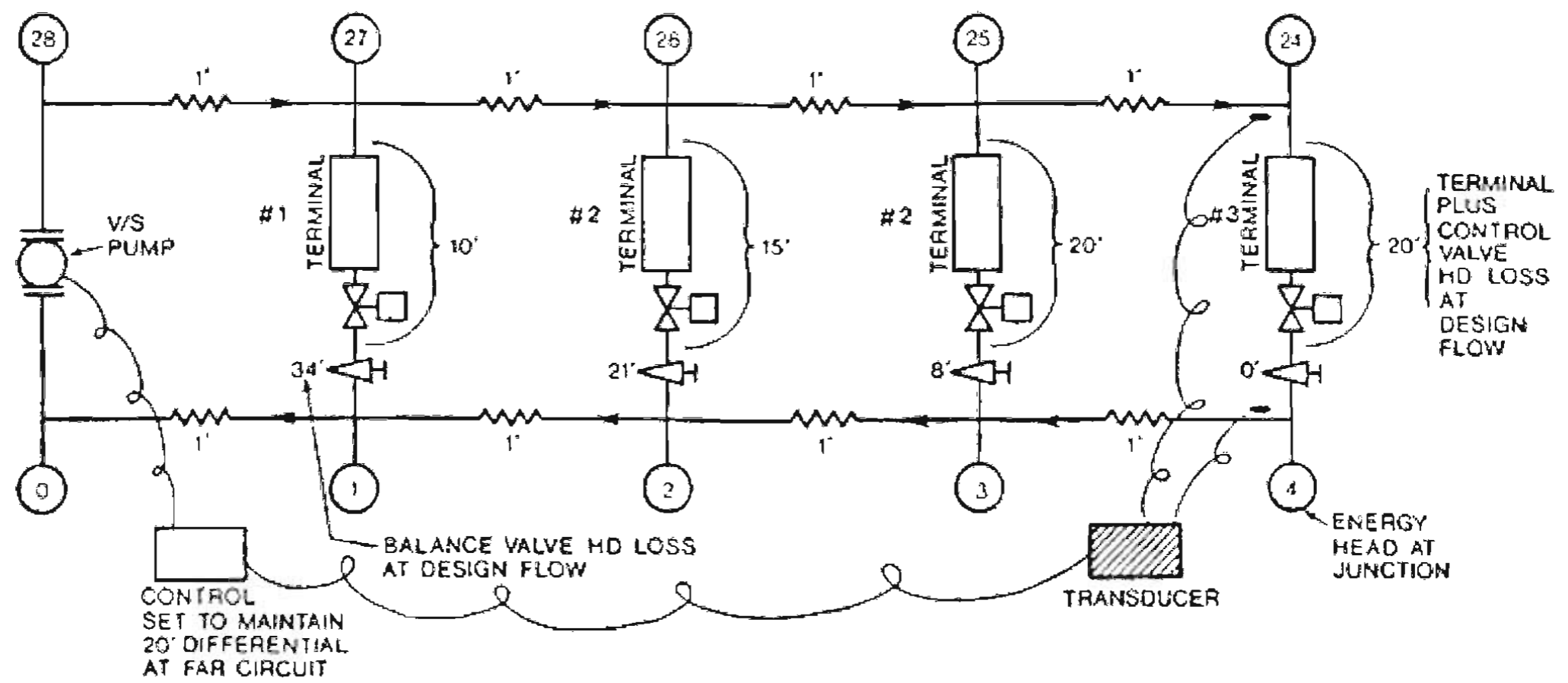


FIGURE 40. "Balanced" System Shown Previously, Converted to V/S, and Operating at 50% Flow

In Figure 40, circuit #1 cannot get its full design flow for the 50% flow condition illustrated because it has a total design flow resistance of  $(34' + 10')$  or  $44'$ , but only has an available head of  $(27' - 1')$  or  $26'$ .

Figure 41 illustrates the "best" balance valve settings for the direct return system shown in Figure 39 when using V/S pumping. The system is again shown at 50% flow. Now full flow capacity is available at all sub-circuits and they are balanced against each other so that each has an equal head loss; in this case 20 ft. at design flow.

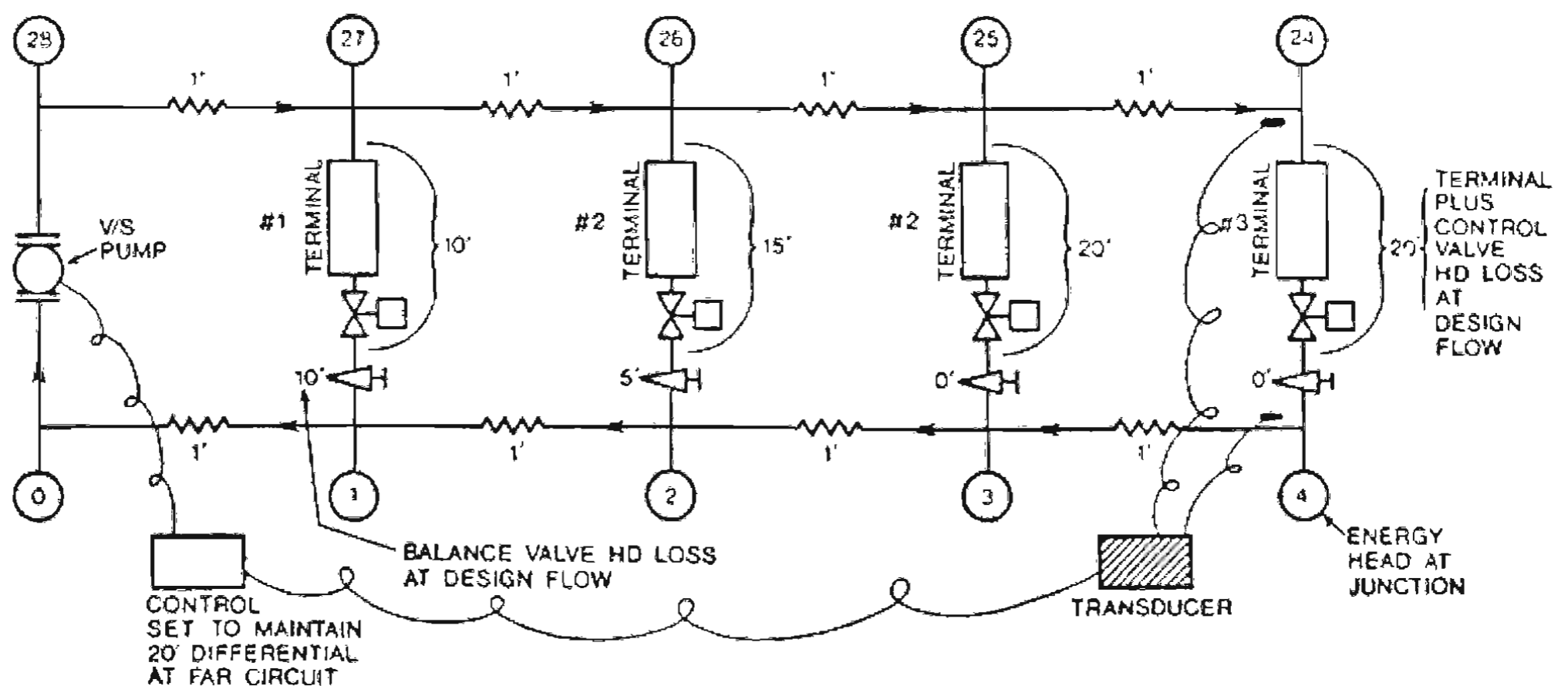


FIGURE 41. "Best" Balance Valve Head Loss Settings; Using V/S Pumping with Piping System at 50% Flow



Since piping head loss is not taken into account, the direct return V/S system may be subject to uneven pick-up after weekend shut-down. High load operating costs may also be greater for the direct return system, as compared with reversed return, because a wide-open control valve close to the pump may "steal" more than its required flow share. For example, at full system design flow, circuit #1 in Figure 41 will have 44 ft. differential head across its circuit, but has only 20 ft. head loss at design flow. With a wide-open control valve, flow would increase by a factor of  $(44/20)^{1/2}$  or about 50% over design.

10D. IMPROPER USAGE OF DIFFERENTIAL PRESSURE CONTROL SENSORS

Many V/S pumping systems do not provide their full saving potential because of improper placement of the pressure sensors. This is shown in Figure 42.

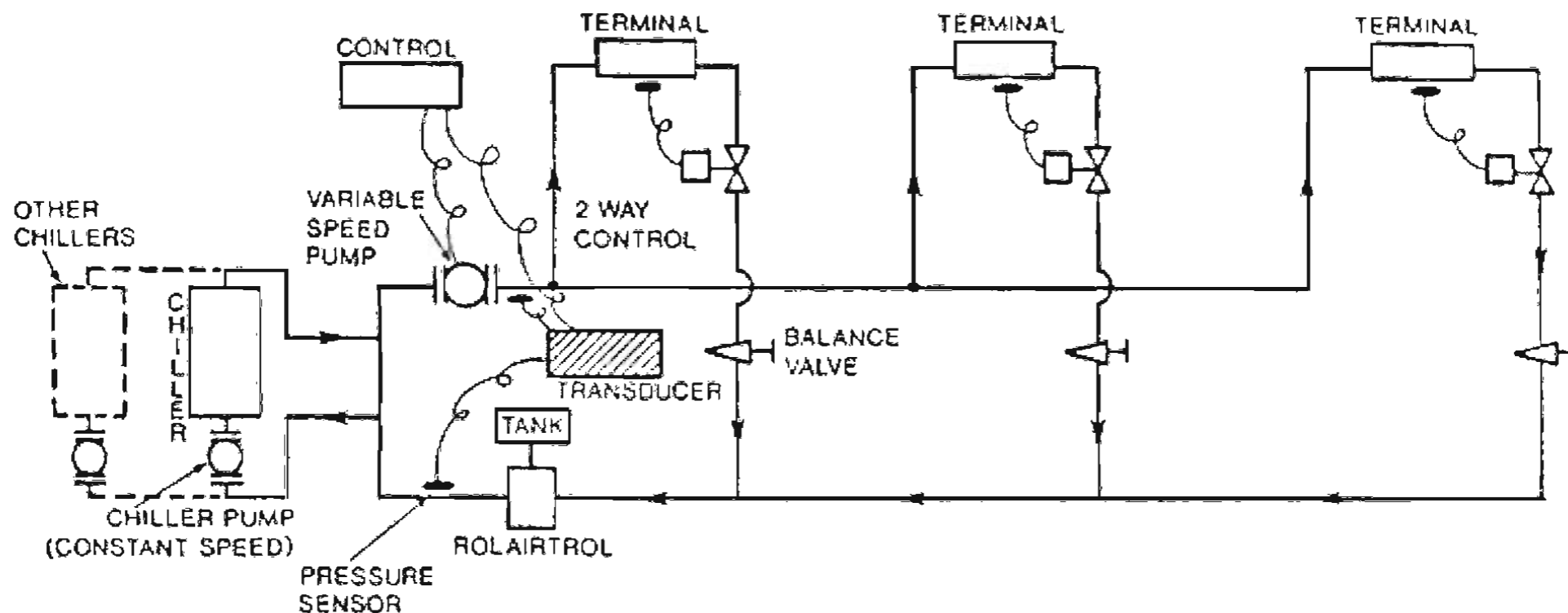


FIGURE 42. Pressure Differential Improperly Taken Across Supply and Return Mains in Equipment Room - A Common Error

The pressure differential signal is often improperly taken across the supply and return mains in the equipment room. This is usually done in the mistaken belief that -

- (1) The cost of running a long control line from a transducer at the end of the circuit back to the pump controller may be considered too high.
- (2) A naive belief that this is the best location.

This is probably the worst location since the result of taking pressure differential sensors across the pump is to essentially convert the V/S pump to a constant speed pump which has a "flat" pump head characteristic. Some limited HP savings will occur but only because most centrifugal pumps do have a rising head characteristic as flow is reduced.

HP saving results are minimal, however, when compared to locating the pressure differential taps across the sub-circuits as previously described. This is because properly located differential pressure taps react to reduced piping head loss at low flow and reduce the pump speed accordingly thus maximizing pump HP savings.

Variable speed pumps are at times controlled by a single pressure tap installed close to the pump discharge. While this may make some sense for a V/S Pressure Booster application, it is a poor application when used in a closed loop HVAC system, as shown in Figure 43.

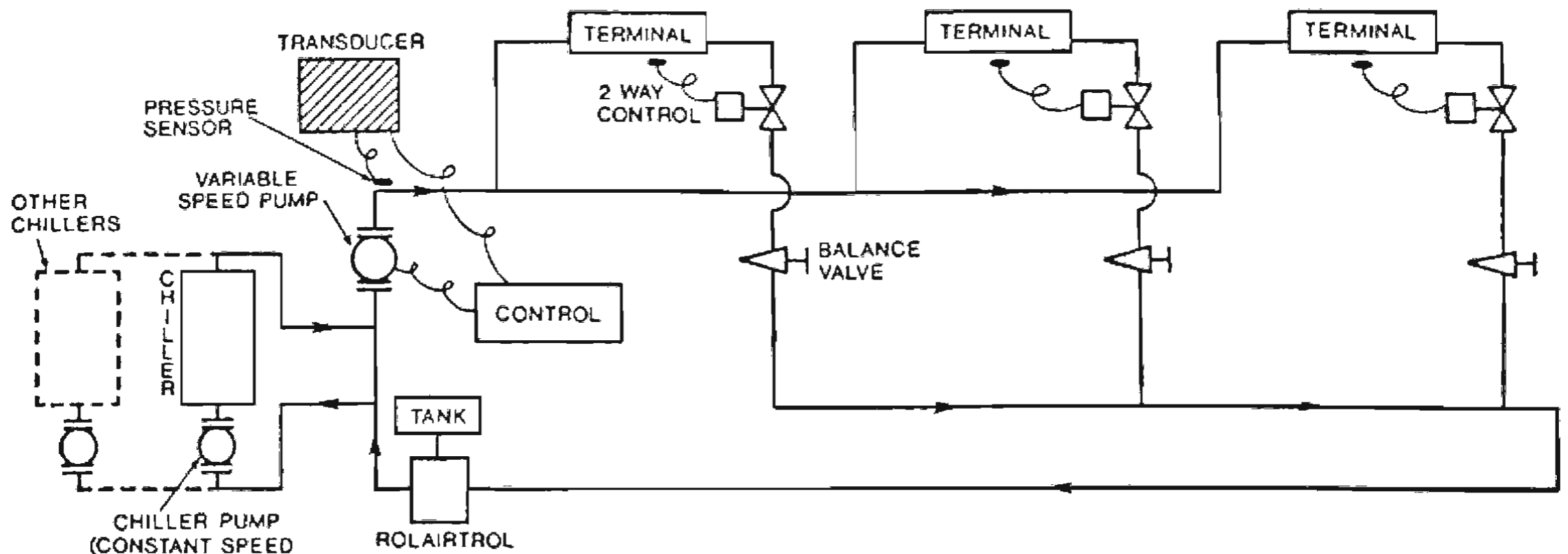


FIGURE 43. Illustration of Single Pressure Sensor Improperly Used for Control of V/S Pump on HVAC System

The single pressure sensor installed at the pump discharge is only influenced by pressure. When the system static pressure increases for other reasons such as an increase in compression tank pressure the pump will slow down. When static pressure decreases, pump speed will increase. This is self-defeating since now pump speed is not truly influenced by system load change, but rather by system water expansion and by operation of the system pressure reducing valve.

Pump speed may also be influenced slightly by the rising pump curve characteristic as flow reduces, but this will be a minor factor compared with static pressure changes.

Control of pump speed by a single pressure sensor at the pump discharge is not recommended in closed loop applications because it is not related to system flow and head loss change. Savings would be minimal.

11. MULTIPLE ZONE V/S PUMPING SYSTEMS

When multiple circuits are required, as in Figure 44, each circuit or zone may be subject to different time loading conditions. In this case separate transducers and their differential pressure sensors are used for each circuit.

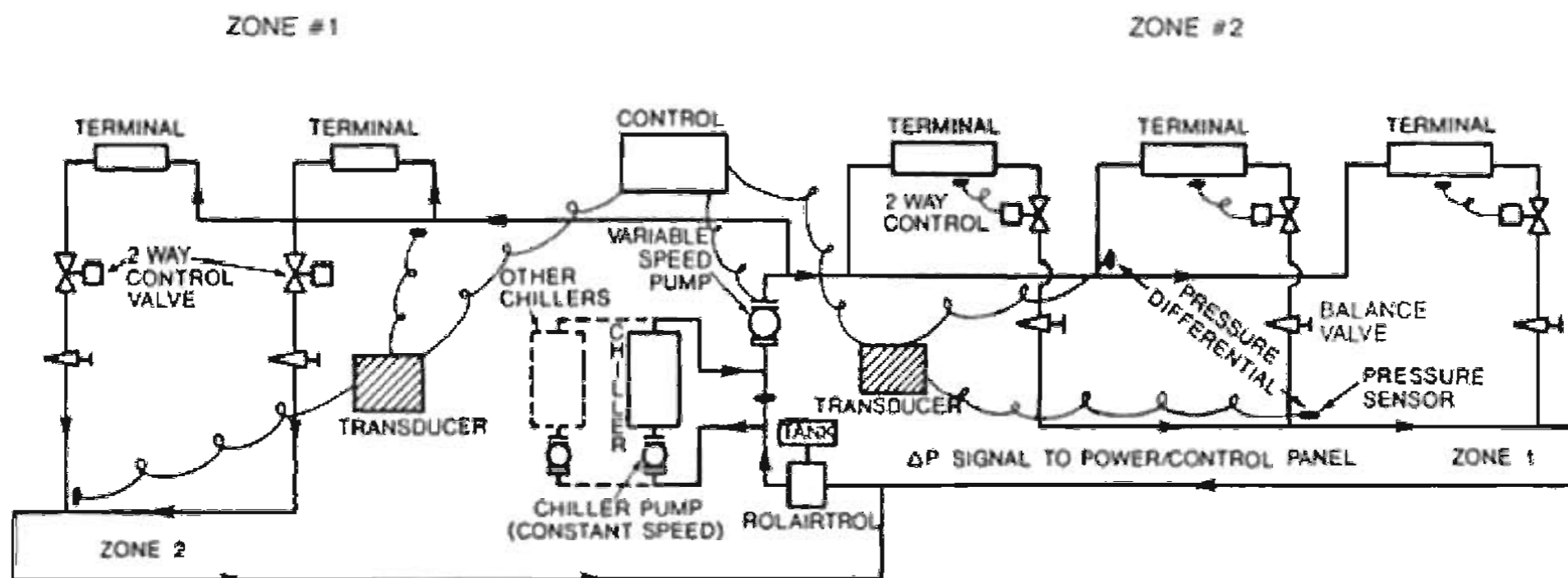


FIGURE 44. Separate Transducers Applied to Separate Zones for V/S Pumping

In Figure 44, the controlling transducers are selected by a special selector control. If, for example, the transducer in Zone 2 finds that its differential pressure is less than required, it will over-ride Zone 1 transducer and increase pump speed until satisfied. The same over-ride function applies to the transducer in Zone 1.

Maximum power savings will result when each zone is provided with its own pump and control.

A zoned V/S pumping system is shown in Figure 45. Zoning eliminates troublesome time-load differences as they occur in several zones. When the V/S pumps are located in a single equipment room, care must be taken that the piping head loss common to all pumps (A-B-C-D-E-F) in Figure 45 is low enough to prevent intra pump operating interference. Separately pumped chillers are needed.

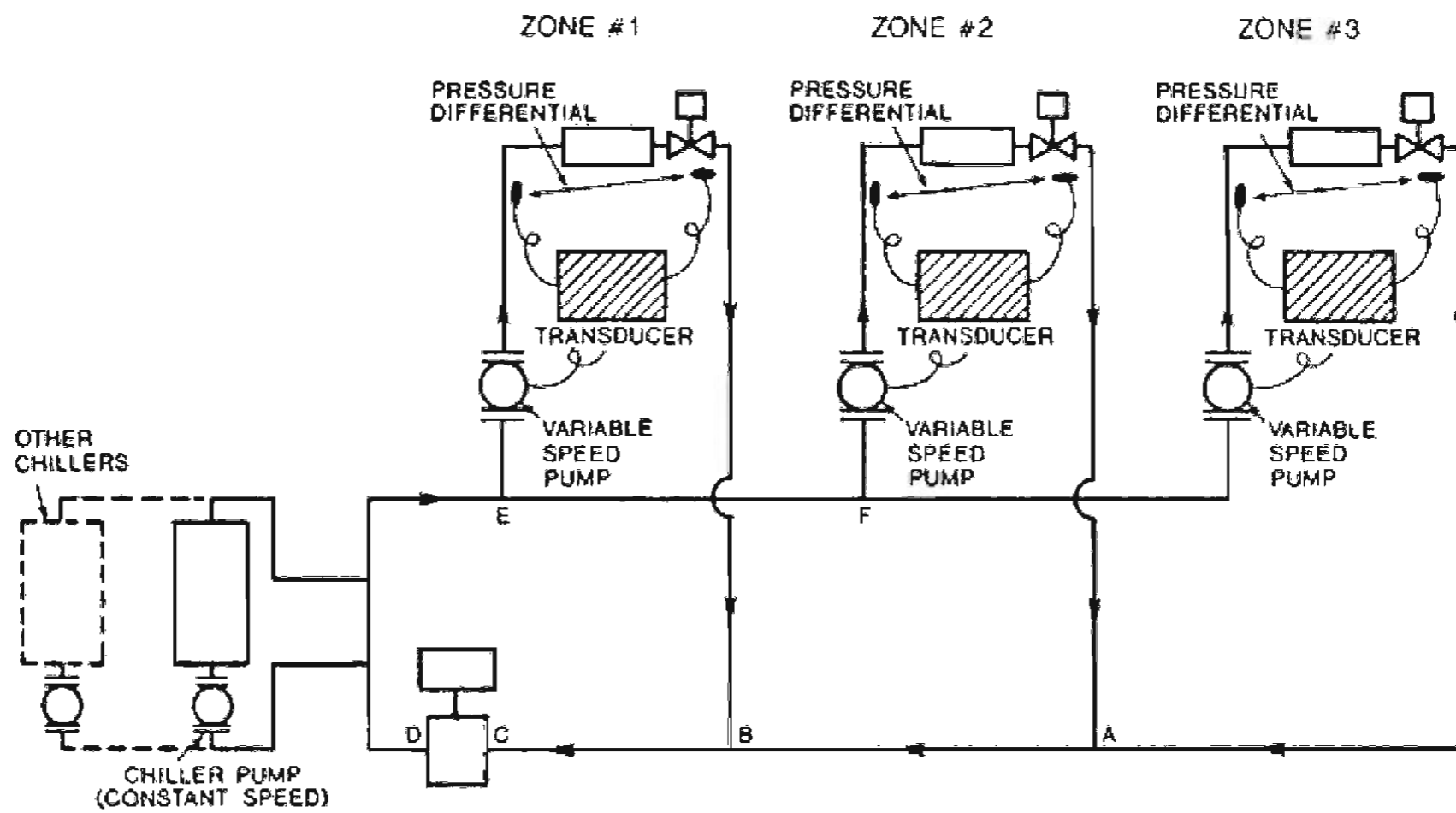


FIGURE 45. Zone Variable Speed Pumping; Each Zone May Represent a Large Multi Terminal Circuit (See Previous Discussion for Pressure Differential Tap Location)

Zone V/S pumps are also used on large campus type distribution systems, where distribution piping loss (A-B-C-D-E-F) is significant. Figure 46 illustrates such a system, where head loss in each pipe section; A-B, B-C, C-D, etc. is 10 ft. at full load flow. The circled numbers on Figure 46 shows the full flow energy head requirements at each junction point: A, B, C, etc.

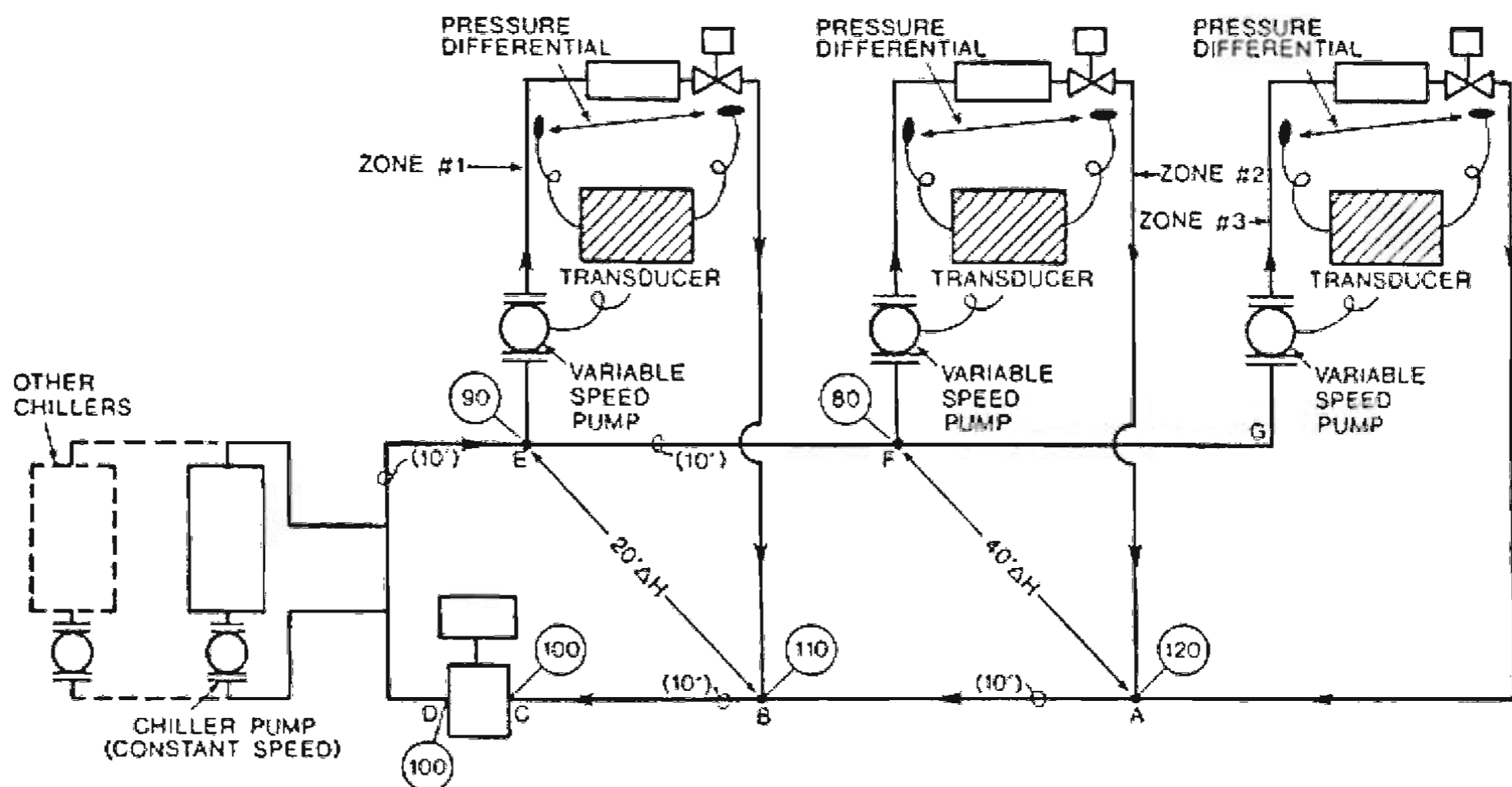


FIGURE 46. Zone Variable Speed Pumping with High Distribution Piping Head Loss; Each Zone Shown May Represent Multiple Circuits (See Previous Discussion for Pressure Differential Tap Control)



In Figure 46, each zone V/S pump must be able to assist in circulating water in the distribution main plus providing circulation for its own zone. As an example, the V/S pump for Zone 2 must be selected for its own zone needs plus an additional 40 ft. head necessary to overcome distribution piping head loss across A-F. During operation, each zone pump will be markedly effected by other zone pumps, because of their effect on distribution line head loss.

Keep in mind the system illustrated in Figure 46 may be subject to expansion problems. As an example, if a new zone were to be added as an extension from G-A, distribution piping energy head needs at each junction point could change significantly. This would require re-assessment of V/S pumping head needs for each of the already installed V/S zone pumps.

## 12. CORRELATION OF LOAD WITH FLOW; ESPECIALLY AT LOW FLOW CONDITIONS

V/S pump application for HVAC requires that flow track load. Therefore, when load is reduced by 1/2, flow should be approximately reduced by the same amount. A two way valve will provide this function - though some problems may be introduced at very low, and very high (pick-up) load conditions.

A two way control valve, selected for a design temperature rise of 10°F. through a coil at design flow and load will maintain (or increase) temperature rise as it throttles. This means that flow will track load through the majority of load conditions. However, there are some exceptions to this.

When separately controlled sub-circuits (coil and valve) are not balanced against each other, and a heavy pick-up load occurs, the valve will move into its wide open position, and much more flow than needed will pass through the coil and its valve. This excessive flow will not significantly increase coil capacity, but will cause a reduction in chilled water temperature rise. This will upset the desired flow-load correlation. The V/S system must be balanced if maximum savings are to be obtained.

Consideration should also be given to a low load condition. All air handling coils are selected for full load conditions with turbulent water flow through the coils. As a two way valve throttles flow, at some point (approximately 25% design flow) the coil flow will change from turbulent to laminar flow. This change will reduce overall heat transfer in such a way as to greatly increase flow relative to load.

The principle tools used to provide a constant flow-load track on large V/S pumping systems are:

- ... Large air handling coils can use sequencing control valves - one per bank. As load decreases, the coil banks are sequentially shut down, but those coil banks in operation remain in turbulent flow. This maintains a high water side  $\Delta T$  and a predictable flow to air handler load correlation.
- ... Return water temperature control has also been used because with a fixed supply water temperature, a fixed return water temperature must establish a fixed flow to load correlation at any load condition.

### 13. CHILLED WATER V/S PRIMARY-SECONDARY PUMPING SYSTEM

Large central station chilled water systems utilizing primary-secondary pumping are particularly well suited for V/S pumping, because they have long distribution lines and a minimal maintained constant head difference. A variable speed pumped system is shown in Figure 47.

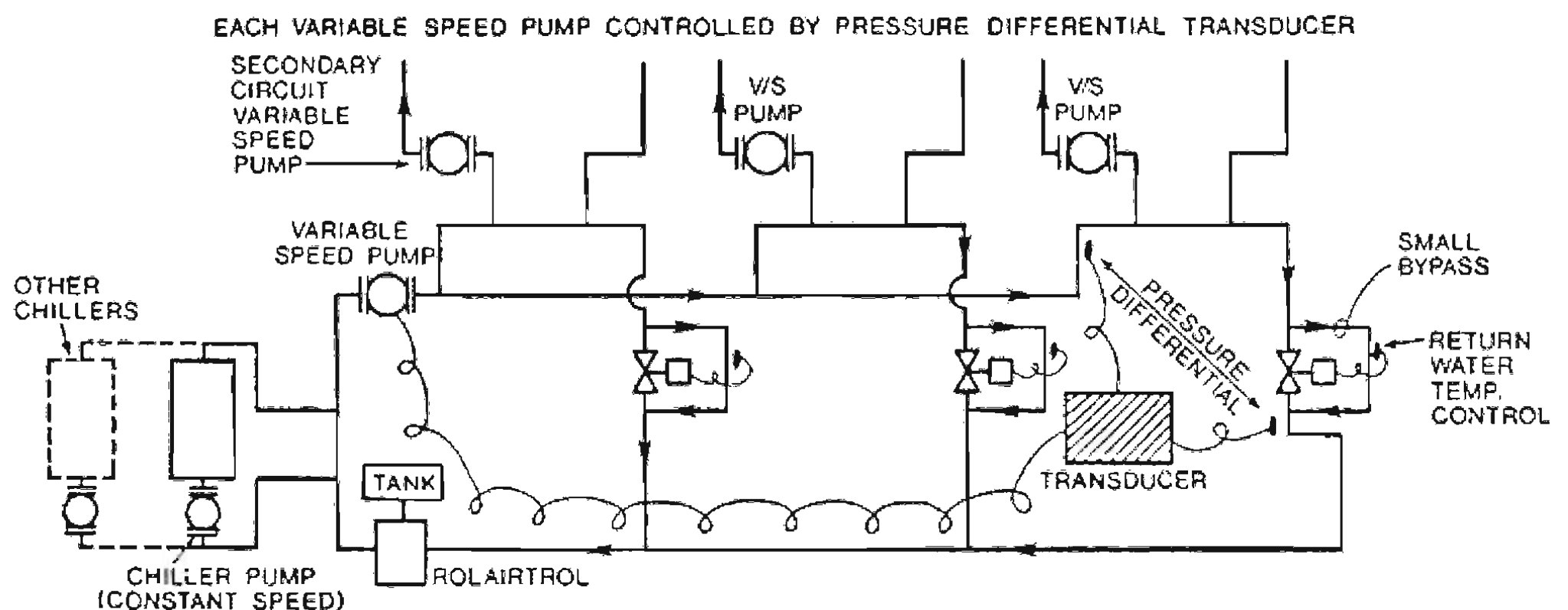


FIGURE 47. Primary-Secondary System Using V/S Pumping and Return Water Temperature Control

One of several versions of return water temperature control is shown in Figure 47. In this case, a small amount of return water is in a continuous bypass around the return water temperature control valve. This protects against unstable control that could otherwise occur if the control valve were to close with the control bulb inserted in the main return line. At no flow, water in this line would be stagnant and could not be used as the control signal.

Secondary circuit pumps can be either V/S or constant speed depending on circuit size, HP, return on investment, etc.

For historic reasons (see pages 2 - 4), most large hot water systems use "reset". Reset changes hot water supply temperature to terminal units in accordance with changes in outdoor temperature. Figure 48 shows a typical "conventional" reset schedule for a HW system designed for  $20^{\circ}$   $\Delta T$  at a design supply water temperature.

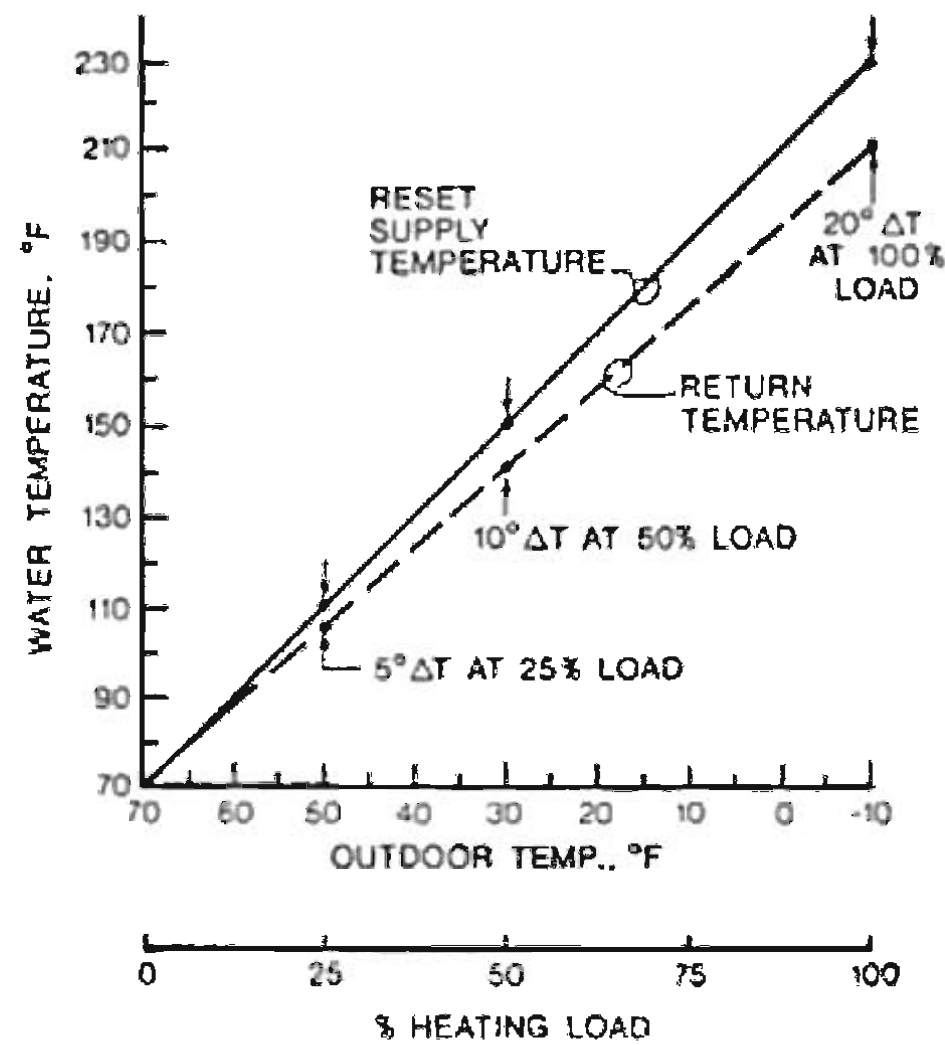


FIGURE 48. Conventional "Theoretical" Reset Schedule Provides Constant Flow

As previously noted, one of the reasons for reset was to provide a basically constant flow in order to overcome two way valve differential pressure control problems. Constant flow is provided by conventional reset because of the heat balance between reset radiation output and water flow rate (head conveyance). It should be remembered that heat output from HW radiation is set by the temperature difference between the mean water in the radiation and the air being heated, while flow rate is set by radiation load and the water temperature difference across the radiation.

Using Figure 49 as a reference for supply water temperature with load change, and assuming a 5,000,000 B/Hr. design load, the 100% load heat balance is shown below when outdoor temperature is  $-10^{\circ}$  F.



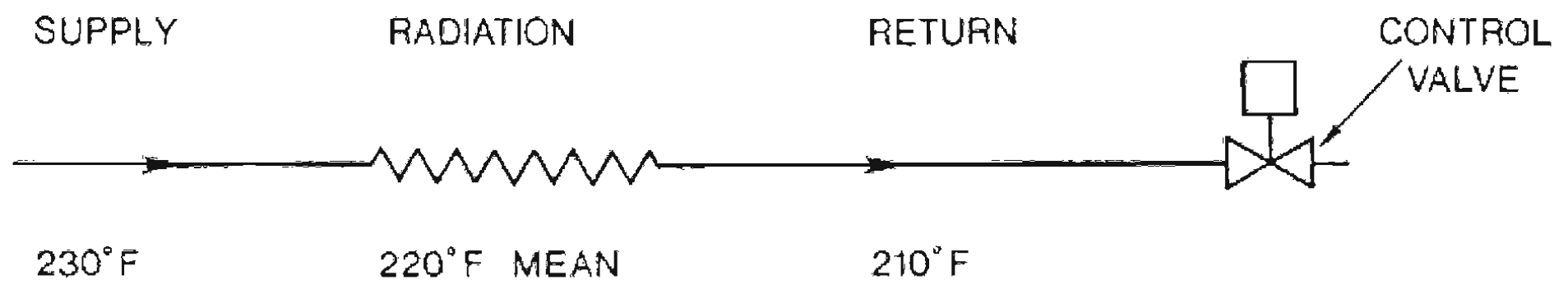


FIGURE 49. Radiation Condition at 100% Load

AT 100% LOAD:

- MEAN AIR TO WATER  $\Delta T = 220 - 70 = 150^\circ F \Delta T$
- WATER TEMP DIFF =  $230 - 210 = 20^\circ F$
- WATER FLOW =  $5,000,000 / 500 \times 20 = 500 \text{ GPM}$

Note that, with reset, supply water temperature will decrease to  $150^\circ F$ . at 50% load ( $30^\circ$  outdoor temperature).

The heat balance will be established by the temperature conditions shown. At 50% load ( $30^\circ$  outdoor temperature), the following conditions will be present:

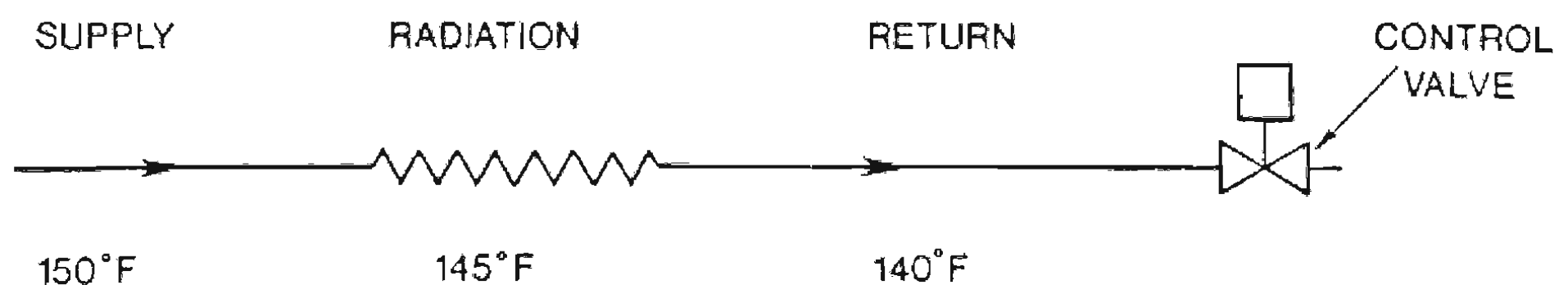


FIGURE 50. Radiation Temperature Conditions at 50% Load with Conventional Reset

AT 50% LOAD:

- MEAN AIR TO WATER  $\Delta T = 145 - 70 = 75^\circ F \Delta T$
- WATER TEMP DIFF =  $150^\circ F - 140^\circ F = 10^\circ F$
- WATER FLOW =  $2,500,000 / 500 \times 10 = 500 \text{ GPM}$

In This example the flow rate has remained constant and the initial historic reset objective has been attained. Constant flow is, however, directly opposite to the variable flow requirements for V/S needs.

Differential pressure controls as applied in V/S pump applications eliminate the historic reason for reset. Reset should still be employed, however, for HW systems because of high potential savings due to increased seasonal boiler efficiency - providing the boiler temperature itself is reset. Reset should not be accomplished by conventional means; that is by maintaining boiler temperature in combination with a three way mix valve, except when required for domestic water heating, etc.

If reset is provided by resetting boiler temperature, the lowest boiler temperature that should be used is the condensing flue gas temperature inside the boiler (about 130°). Use of about 130° as the lowest reset boiler temperature provides a new reset schedule, shown on Figure 51, as reset schedule "A".

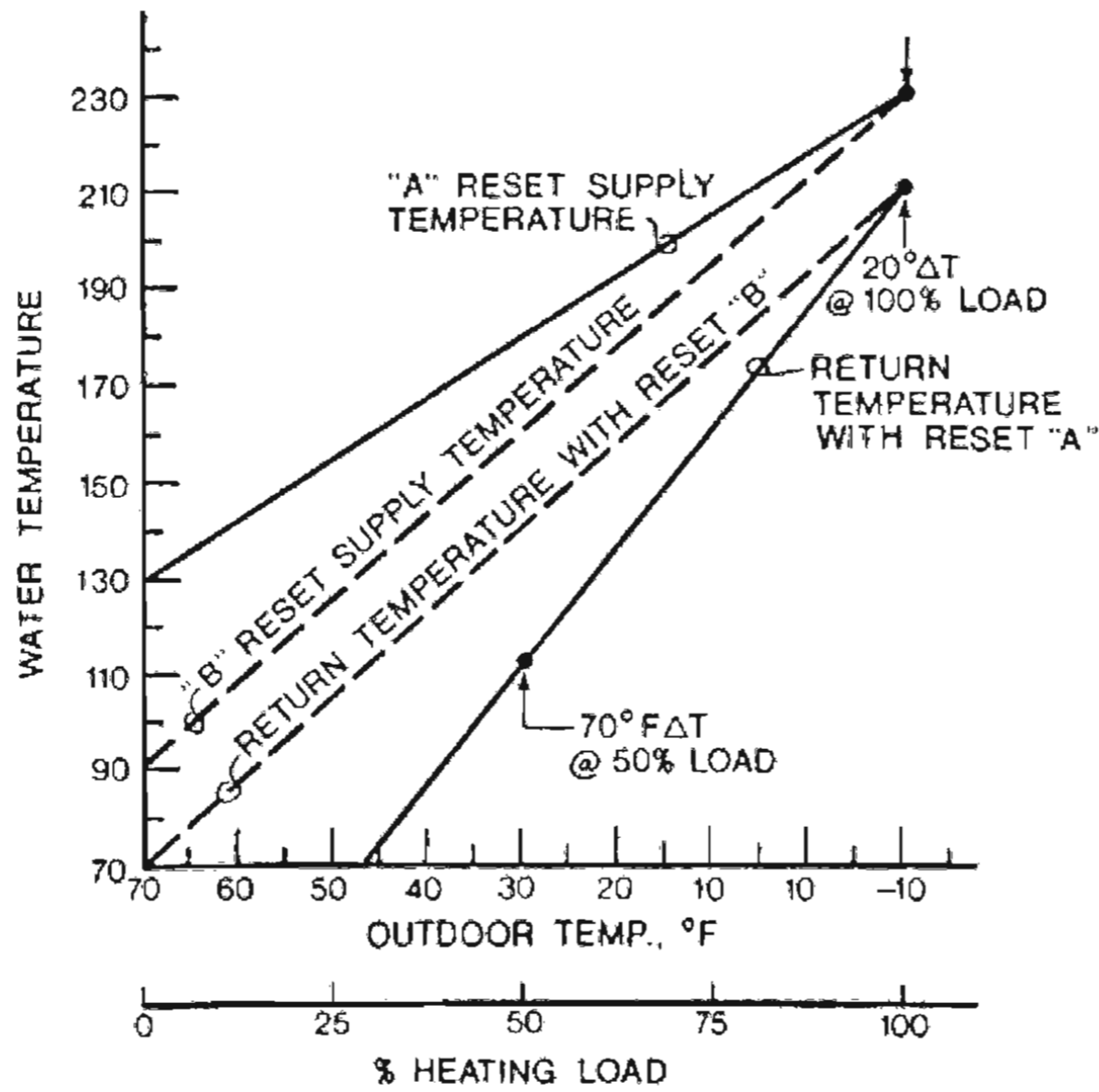


FIGURE 51. Reset Schedules for HW Systems and for V/S Application

Figure 51 also illustrates reset schedule "B" which is intended to provide a constant 20°F. supply to return water temperature difference - and thus a direct flow to load match.

Comparative system flow rates for each of the reset schedules are shown in Figure 52.

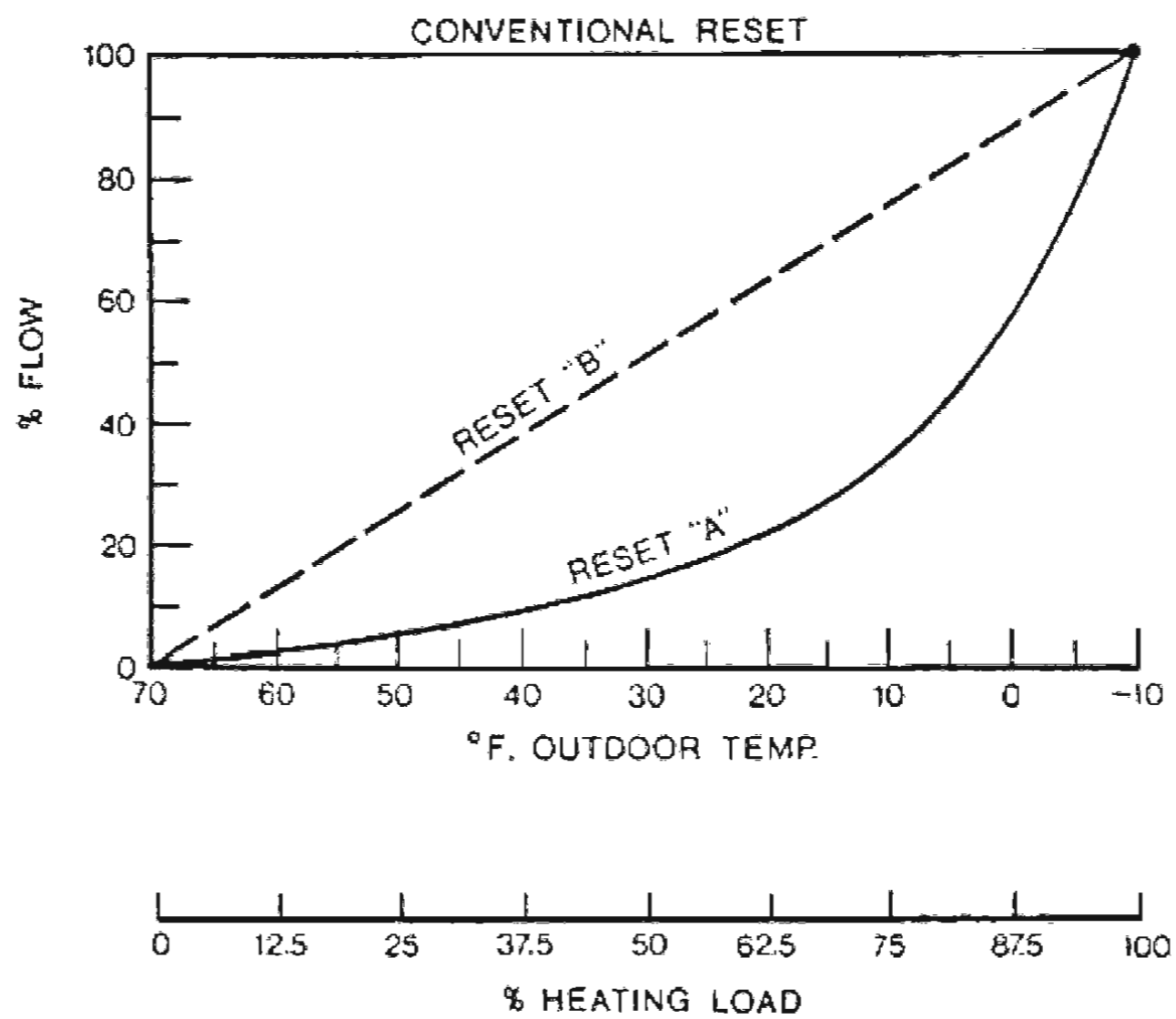


FIGURE 52. Comparative System Flow Rates Established by Reset Schedules

The reset evaluation illustrates that flow rate and temperature drop for the HW system are highly depending on the line "slope" used for reset control.

Reset "A", for example, shows the greatest flow reduction, and therefore, the highest potential power saving for V/S application. Reset "A", however, also provides very high system temperature drops, so care should be taken to avoid boiler temperature "shock". In order to avoid temperature "shock", separate boiler recirculation will be needed as shown in Figure 53.

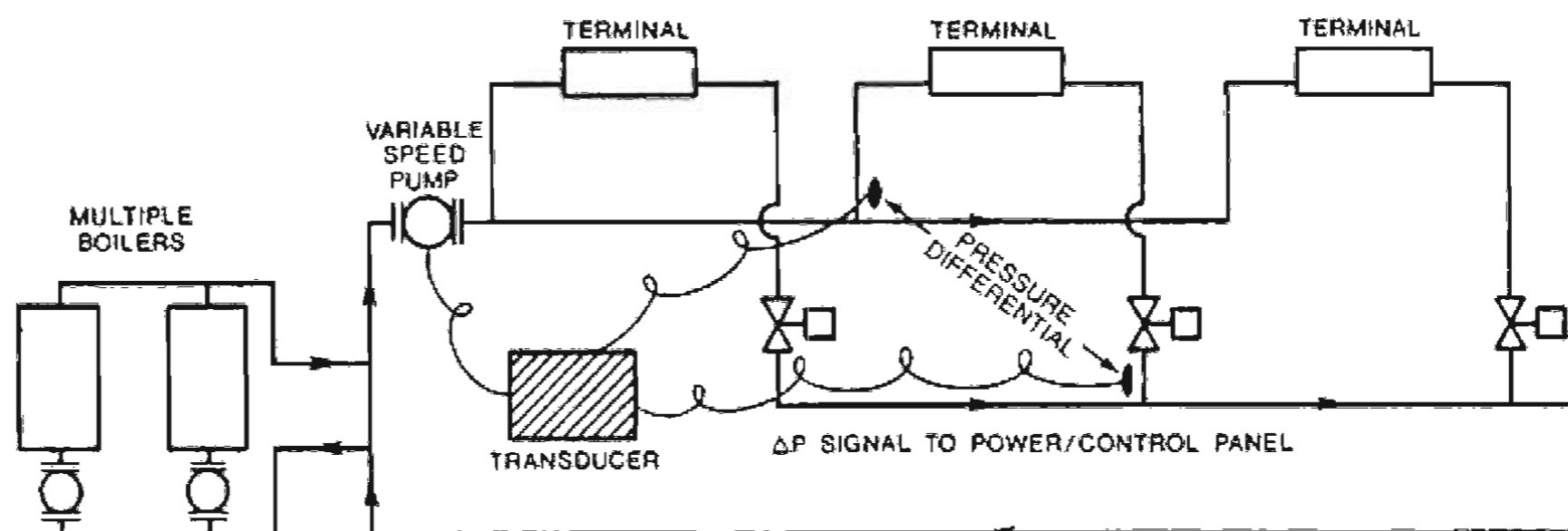


FIGURE 53. HW Systems Using V/S Pumps with Separate Boiler Recirculation Pumps

A complete discussion concerning boiler temperature "shock" and its avoidance is contained in B&G Bulletin TEH-475 and is available from Bell & Gossett Representatives.

In Conclusion, the combination savings inherent in properly applied V/S pumping and increasing seasonal boiler efficiency by use of modified "reset" is high and should be considered for HW systems.

15. CONVERSION OF EXISTING C/S THREE WAY VALVE SYSTEM TO V/S

As previously noted, pump operating power saving potential is very high for existing three way valve controlled systems. Cost saving potential is shown in Figure 54.

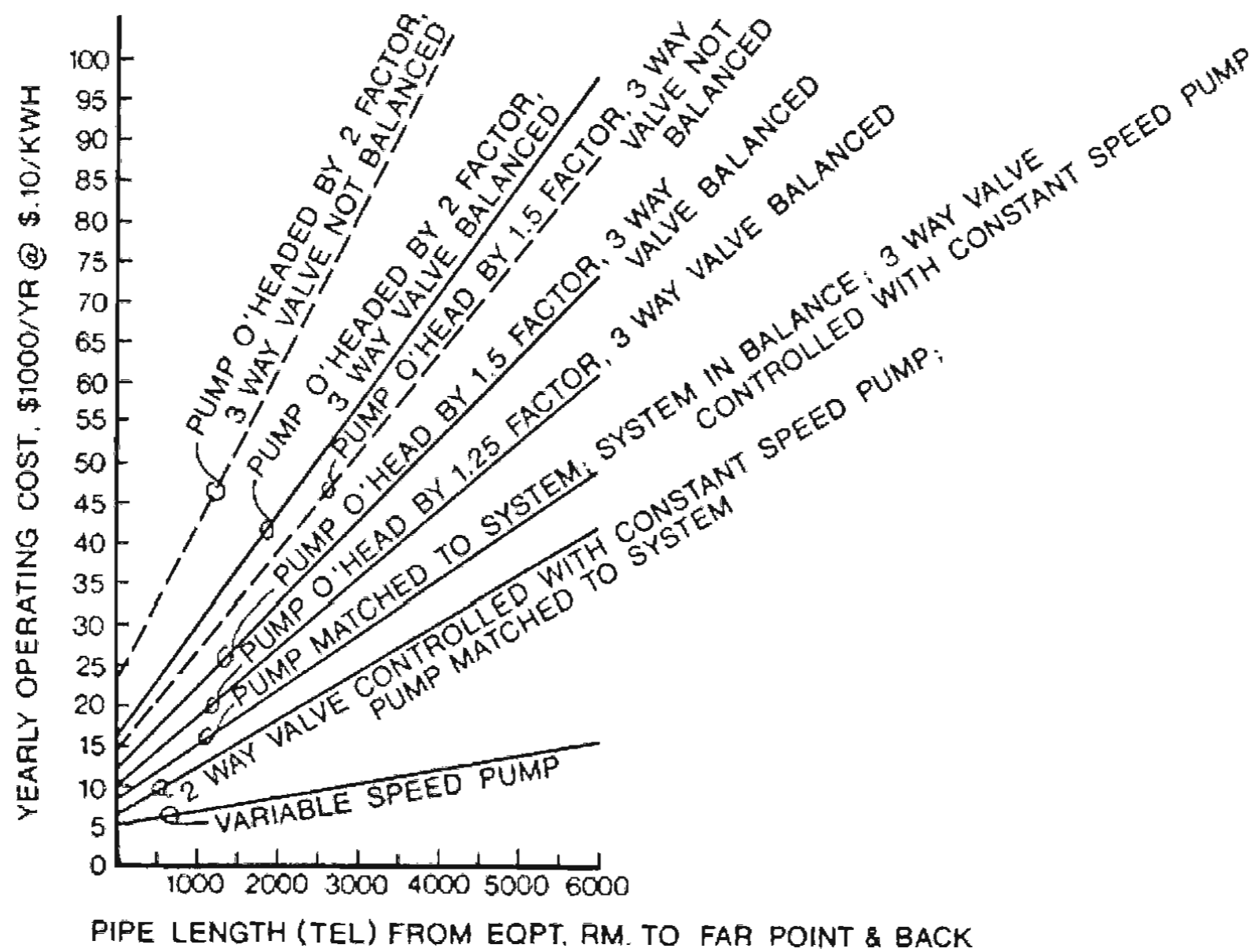


FIGURE 54. Three Way Valve System Operating Cost Comparison with Variable Speed

High operating costs result because of the constant flow established by this system type. Operating costs will also increase because of general lack of system balance. The typical three way valve controlled system as used for chilled water is shown in Figure 55.

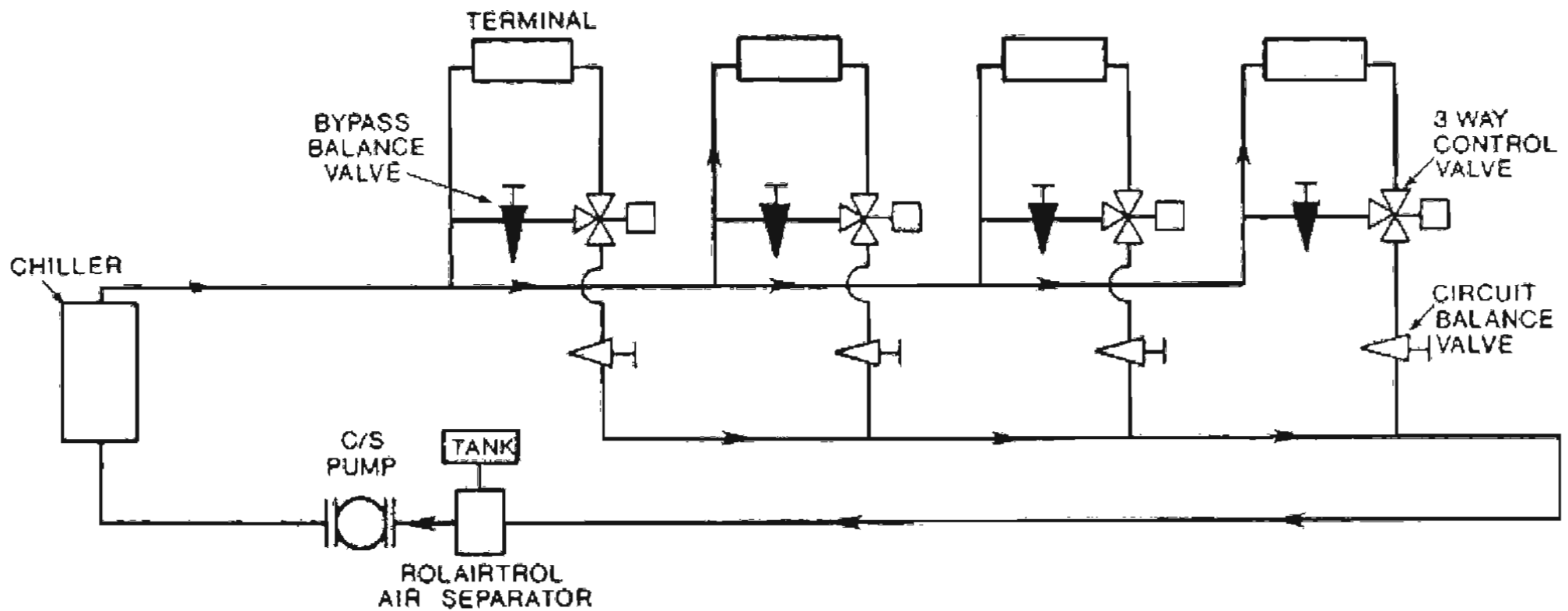


FIGURE 55. Existing Chilled Water System Using Three Way Valves, Pump Power Wastage

The basic approach to conversion of the existing three way valve system is simple; convert the system to two way control. This introduces variable flow volume, and is accomplished by simply closing the bypass balance valve. Closure of the bypass balance valve changes the three way valve to a modified two way control.

System changes introduced by closure (or elimination) of the bypass requires attention.

The basic change from constant flow to variable system flow means that the chillers must be primary-secondary pumped in order to maintain constant chiller flow when system flow is reduced. This is shown in Figure 56.

Figure 56 also illustrates application of a V/S pump. One of the main functions of the V/S pump, in addition to operational cost saving, is to control the otherwise difficult problem of pressure differential increase across the three way valve as modified to two way when system flow decreases. (See two way valve pressure differential increase problem, page 2).

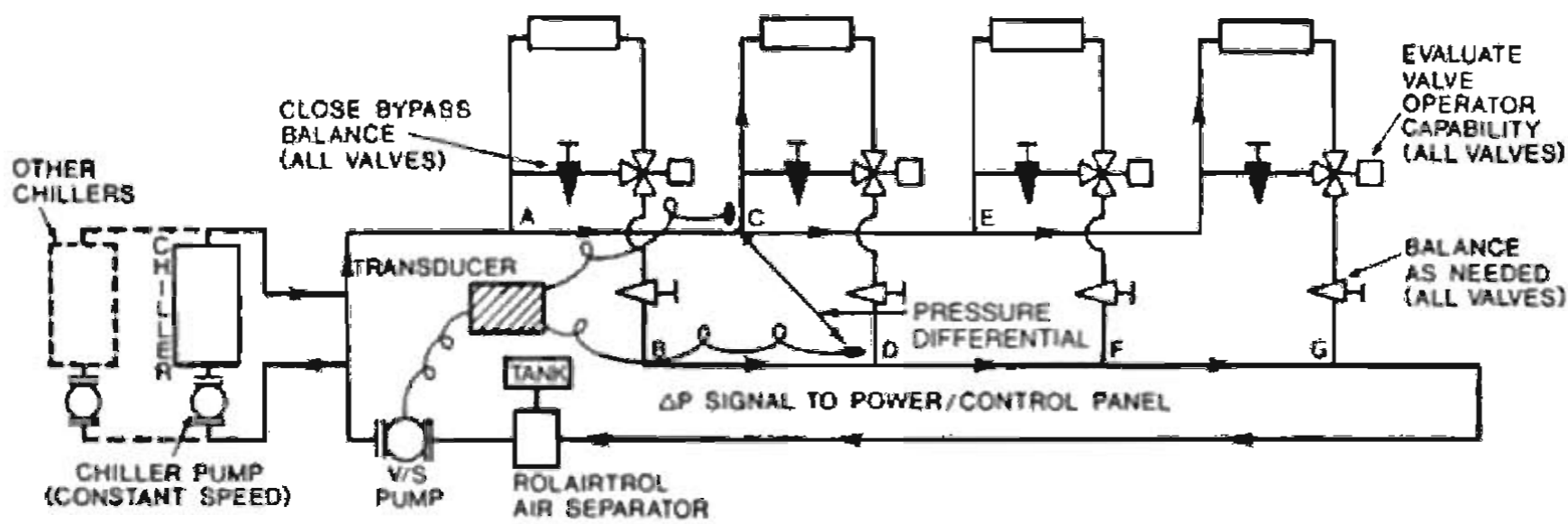


FIGURE 56. Three Way Valve System Modified to Two Way V/S Pump Systems

When the three way valve system is changed to modified two way with V/S, the capability of the valve operator to close against the V/S controlled pressure differential (A-B in Figure 56) must be evaluated. As a generality, most three way valve operators are selected to a low operator power need. This is because conventional three way bypass application does not require closure against a substantial pressure difference.

The constant pressure difference to be maintained by the V/S pump as across A-B in Figure 56, is determined by an assessment of sub-circuit head losses as across A-B, C-D, E-F, etc. This assessment will include piping, terminal coil and control valve head loss. The controlling differential would be the highest head loss sub-circuit, and all other circuits would be balanced against that head loss by use of the Circuit Setter Balance Valve.

It should be noted that particularly high terminal head loss sub-circuits can be primary-secondary pumped if necessary to reduce the pressure difference to be maintained by the V/S pump.

Should the preceding evaluation uncover any valve operator with insufficient power to close against the maintained pressure difference, it will be necessary to change the operator (increase operator power).

The relatively high flow rates required by the condenser on a chilled water system appears to establish a sound base for V/S pump application. A projected application is shown in Figure 57.

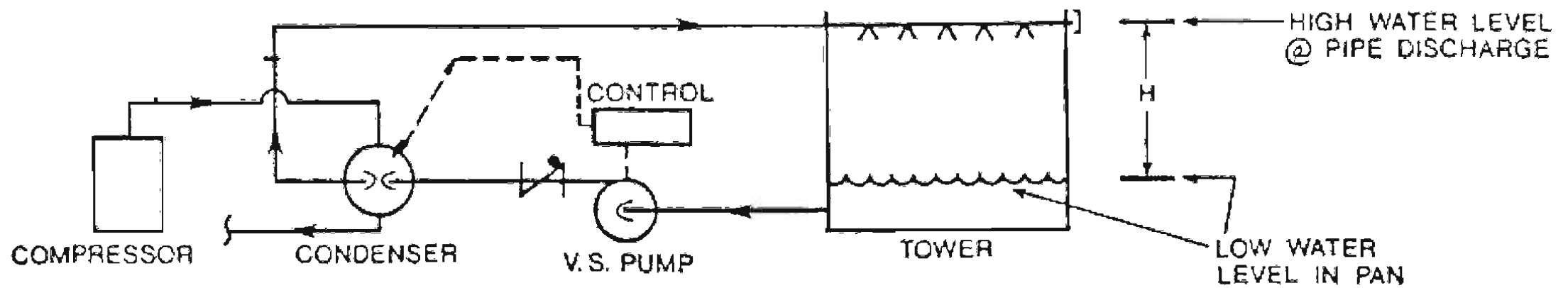


FIGURE 57. Cooling Tower Pumping with V/S

The V/S cooling tower-condenser pump is generally controlled from condenser pressures shown in Figure 57. While this is satisfactory for a single condenser, difficult control problems can result when multiple condensers (chillers) are used, unless separate V/S pumps are used for each condenser.

In terms of the cooling tower, maximum V/S pump power saving will result when high pressure drop tower spray nozzles are used. This is because variable system head loss is increased as a proportion of total pump head. Care should be taken, however, when flow is reduced through tower nozzles because of the potential for "misting" in the tower.

The assessment concerning V/S pump adaptability to the cooling tower circuit should include the following considerations:

- ... Chillers operate at reduced KW/Ton draw rates when leaving tower water temperature is reduced. This power saving should be compared with V/S pump power saving.
- ... Minimum condenser flow rates should also be considered, since too low a flow rate may lead to laminar heat transfer and a potential for tube fouling.
- ... When V/S pumps are applied to the cooling tower circuit, both the chiller and tower manufacturer should be consulted because of potential inter-locking problems.

V/S pumps have not been widely applied for condenser-tower pumping, apparently because of the concerns expressed above.



Power saving in the tower-condenser circuit is presently established by tower fan cycling; and by the use of separate C/S condenser pumps applied to each condenser. In both cases, power consumption is reduced when fan and pump motors are shut-down.

Very large power saving will be established when "free cooling" is provided by a plate exchanger. The plate type exchanger introduces chilled water to the chilled water systems without chiller operation, when tower water is sufficiently cold. It should be noted that the plate exchanger maintains operation of the chilled water system as closed and sealed against the introduction of tower water contaminants (oxygen, etc.). The plate exchanger can also be used to prevent fouling of the condenser - a very large energy waster for the chilled water system.

17. PRESSURE BOOSTER PUMPING

Pressure Booster Pumps are used to maintain pressure at the top of the hi-rise buildings when city main pressure is too low. The V/S application is shown in Figure 58.

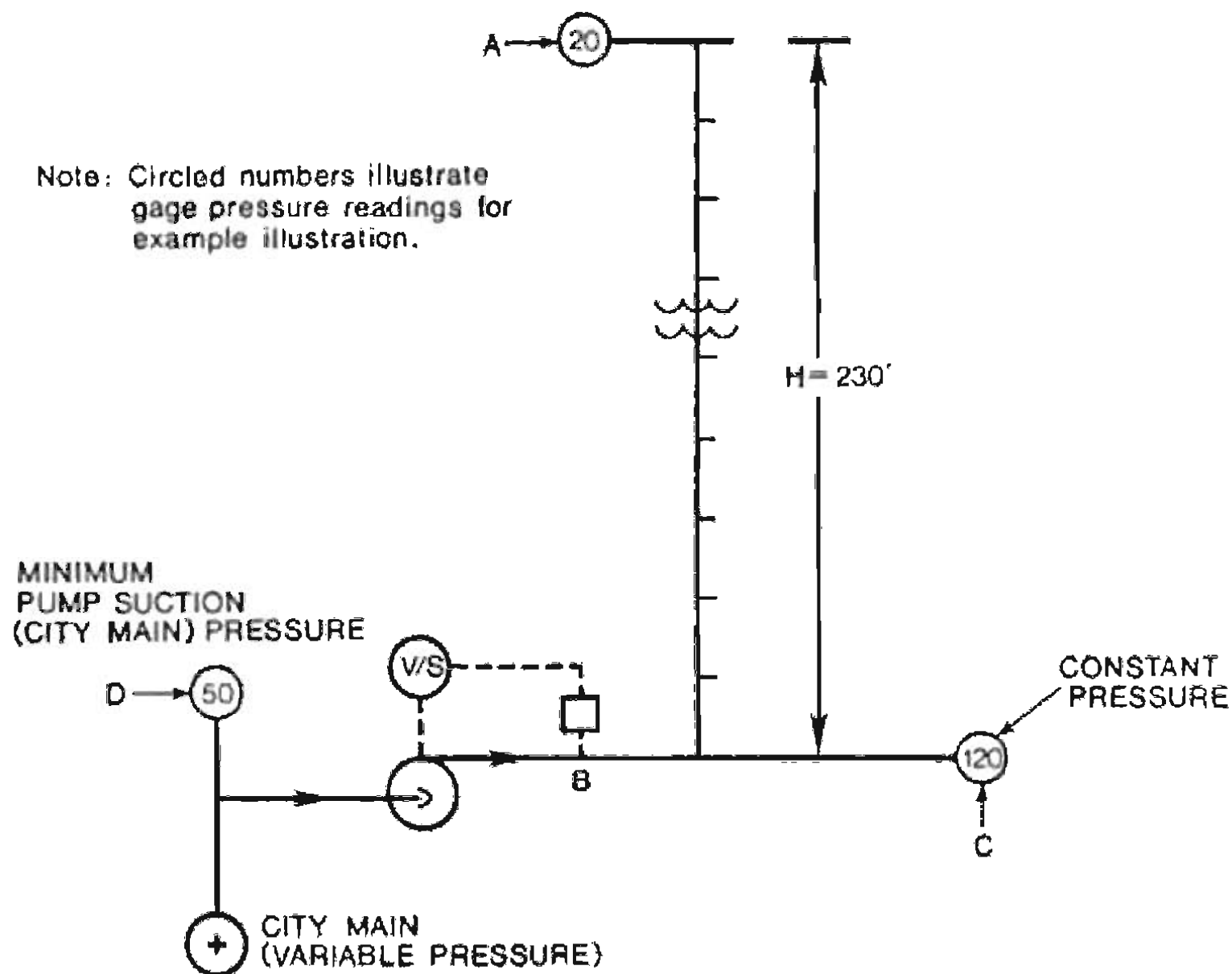


FIGURE 58. V/S Pressure Booster Pump Provides Constant Building When Street Pressure Varies

V/S Pressure Booster Pumps are generally used when street pressures are highly variable. When street pressures remain fairly constant, paralleled constant speed pumping, in combination with pressure reducing valves, are most often used.

In Figure 58, the V/S pump provides a constant 120 PSI pressure at "C" in order to establish a relatively constant 20 PSI pressure at the top of the building (at "A"). Given the conditions shown, the pump would be selected for 120 PSI - 50 PSI or 70 PSI (160 ft.) at design flow.

If the city main pressure did not change, the pump would be working with a constant head differential. Potential power saving for the V/S, as contrasted with paralleled constant speed would be minimal, as shown on Figure 59.

In Figure 59, the approximate comparative power draw for paralleled pumps vs. the V/S pump (Curve "A" in Figure 59) operating with a constant differential pressure is shown. It will be noted that the paralleled constant speed pumps are penalized at high flow because of power loss established by the pressure reducing valve, but that at low flows (one pump operating) the constant speed parallel pumps show a comparative power reduction.

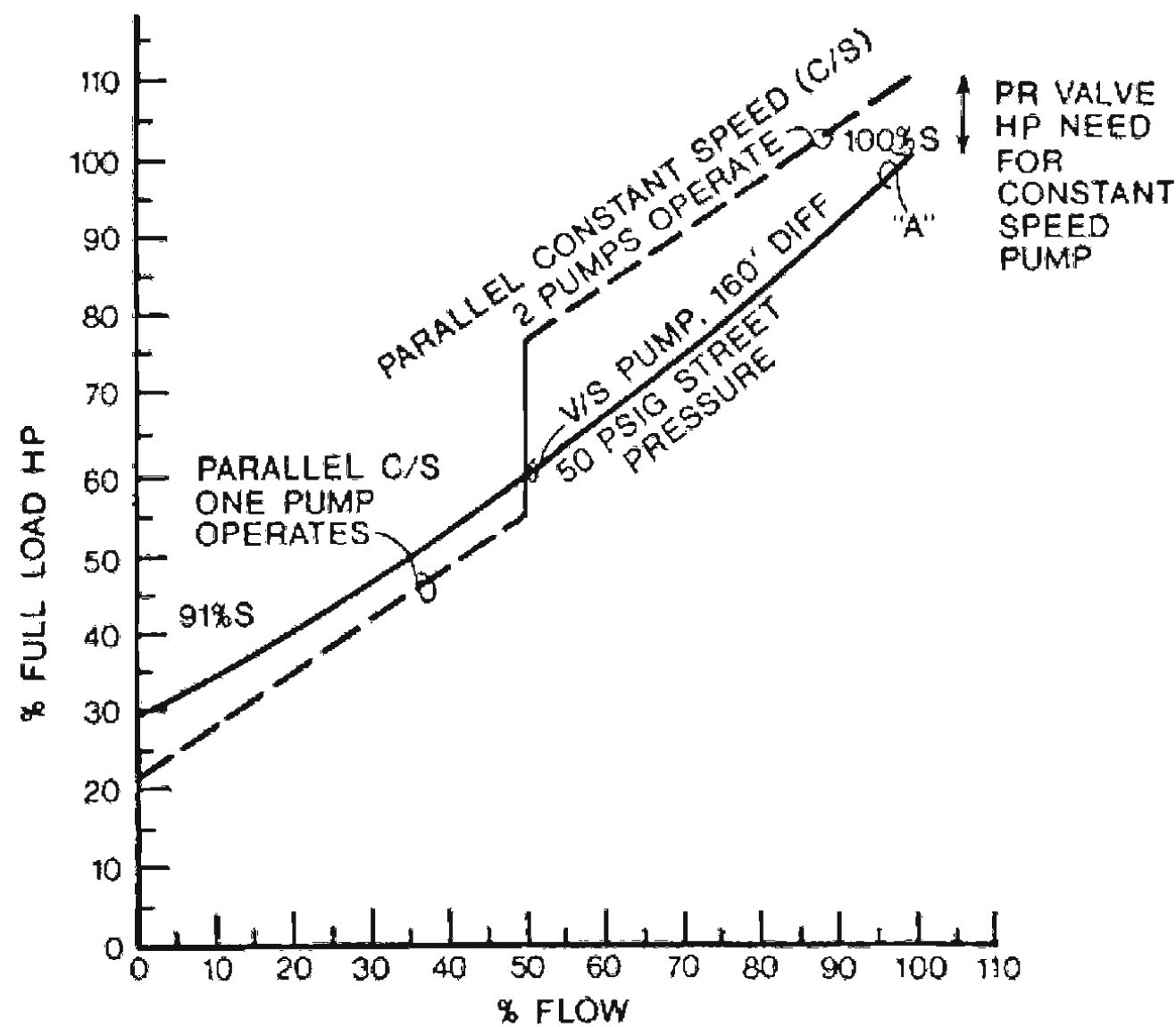


FIGURE 59. Paralleled Constant Speed Versus V/S Pump Power Draw Change with Flow; Pumps Operate with Constant Head Difference

Let us assume now that for a time period, the example street pressure will rise; as from 59 PSIG to 60 PSIG. The differential pressure across the pump is now 120 - 60 or 60 PSI (115 ft.), not the original 120 - 50 or 70 PSI (160 ft.). Similar consideration will be given to a rise in street pressure to 70 PSIG.

The reduction in pump power draw with increases in street pressure is shown in Figure 60.

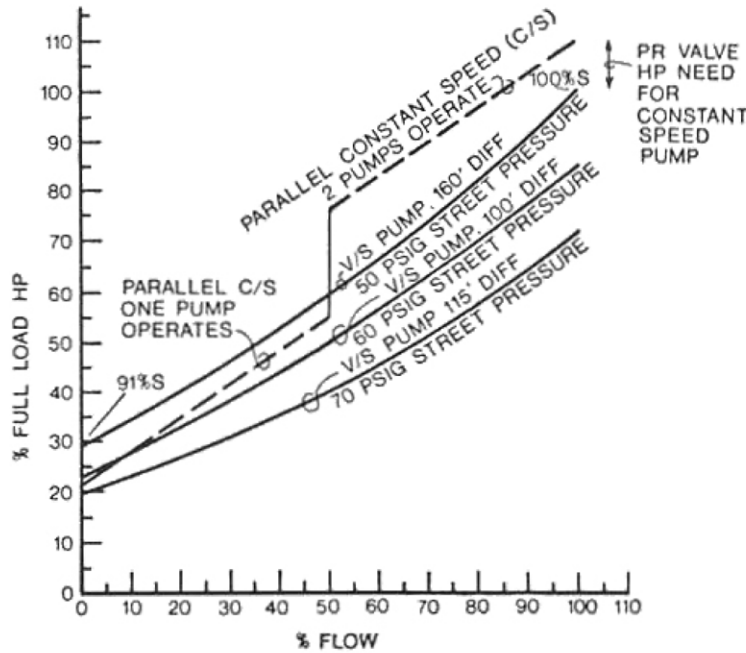


FIGURE 60. Paralleled Constant Speed Versus V/S Pump Power Draw Change with Flow; Pumps Operate with Variable Street Pressure

It will be noted that the V/S Booster System does provide substantial savings when street pressure varies. Remember, however, that the increase in street pressure usually occurs during periods of very low draws, and at a time period when the parallel constant speed pumps can be shut down using no-flow shut-down techniques. It will be agreed that there is no pump that will draw less power than one that is shut down.

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