SECTION 9.21

WATER PRESSURE BOOSTER SYSTEMS

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TYPES OF WATER PRESSURE BOOSTER SYSTEMS

Plumbing fixtures and equipment connected to a water source will function properly only when a minimum water supply pressure is consistently available. Whenever this minimum pressure cannot be maintained by the supply source, a means of boosting the water pressure should be considered.

The commonly used methods for boosting pressure are

1. Elevated gravity (tank) systems
2. Hydropneumatic tank systems
3. Variable-speed-drive centrifugal pump booster systems
4. Tankless constant-speed multiple centrifugal pump systems
5. Limited-storage, constant-speed multiple centrifugal pump systems

Table 1 identifies the significant advantages and disadvantages of each system.

Although gravity and hydropneumatic tank systems have some distinct advantages, most installations favor constant-speed multiple-pump and variable-speed-drive systems. Because gravity and hydropneumatic tank systems are now seldom specified, they will not be discussed in detail here. Multiple-pump systems are well suited for high-rise buildings, apartment buildings, schools, and commercial installations. Variable-speed-drive systems are particularly suited for industrial applications, where control precision is required and maintenance capability for complex electronic apparatus is present.

It is important to remember that the primary function of a pressure booster pump for a building, whether variable- or constant-speed, is to maintain the desired system pressure over the entire design flow range. It is also important to recognize that the cold water service piping configuration is an open-loop system, not a closed-loop system as it is in

TABLE 1 Comparison of significant factors for pressure booster systems

System type	Advantages	Disadvantages
Elevated gravity tank	Simplicity, large storage capacity, low energy usage, reserve storage capacity for fire protection	Size and weight, water damage potential, freeze potential if roof-mounted, corrosion and contamination potential, limited pressure for floor immediately below tank, possible unsightly appearance, periodic cleaning and painting
Hydropneumatic tank	Location not critical, low energy usage, limited storage capacity, pump does not run when there is no demand, operation in optimum pump flow range	Requires compressed air source, corrosion and contamination potential, large pressure variation, relatively large, standby provision costly
Variable-speed drive: Fluid couplings	Simple controls, off-the-shelf motor, standby provision less costly than for above units	Slow response to sudden demand change, may require heat exchanger to cool drive, slip losses result in lower maximum speed and higher motor power, requires selective application, no water storage
Variable-speed drive: ac type	Low motor current inrush, precise pressure control, higher than 3600 rpm possible, few mechanical devices, large power capability at lower first cost	Complex electric circuitry, high initial cost for low-power units, may require special motors, motor low-speed limitation, rapidly changing technology, requires selective application
Tankless constant-speed multiple pump	Relatively low first cost, uses time-proven components, compact size, inherent partial standby capacity, extra standby capacity inexpensive, location not critical, good pressure regulation	Continuously running lead pump, difficult to accurately determine capacity split among pumps, no water storage, problems associated with low flow rates
Limited-storage constant-speed multiple pump	Shuts down during very low water demand, uses time-proven components, standby capacity inexpensive, location not critical, limited water storage, no air-to-water contamination with diaphragm tank	No significant disadvantages, tanks with high maximum working pressure may be difficult to obtain

heating and cooling systems. With an open-loop system, the static head above the pump is not canceled by the down leg of the piping, and therefore the pump must develop a head equal to or greater than the static head, even at zero flow (with suction pressure at design). Review Sections 8.1 and 8.2 for further discussions on closed-loop systems.

Variable-Speed-Drive Pressure Booster Systems Initial variable-speed drives were fluid couplings, magnetic couplings, and liquid rheostat (wound rotor motor) drives. The coupling-type drives were driven by a constant-speed motor, with the coupling output shaft varying in speed. With the advent of solid-state electronics technology and circuit miniaturization, most variable-speed drives currently used for pressure booster applications are of the solid-state, ac, adjustable-speed type. These drives are discussed in Subsection 6.2.2.

Variable-speed drives are usually specified for their low operating cost potential. To achieve the energy-savings goal, the system conditions should cause the drive speed to vary between 50 and 75% of full speed during most of the operating period.

The pump speed, head, and flow relationships are expressed by the affinity laws:

- Flow varies directly as speed:

$$\frac{Q_2}{Q_1} = \frac{N_2}{N_1} \tag{1}$$

- Head varies directly as the square of the speed:

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

where Q = flow rate, gpm (m^3/h)
N = pump speed, rpm
H = pump total head, ft (m)

The required pump total head at design conditions is

$$H = (H_d - H_s) + H_f \tag{3}$$

where H = required pump total head at design conditions, ft (m)
H_d = design system pressure at point of control, ft (m)
H_s = minimum design suction pressure, ft (m)
H_f = sum of all losses between H_d and H_s at design flow, ft (m)

At flow rates less than design conditions and suction pressures higher than the minimum design pressure, the pump head requirement is reduced by the change in pipe frictional losses H_f and the additional suction pressure H_s available. Also, because of the rising characteristic of the centrifugal pump performance curve as the flow decreases, an excess head is developed by the pump. All of these changes will alter the required pump total head and the pump speed from the design (maximum) values.

For a single operating pump unit, design maximum speed will be required at design minimum suction head and design maximum flow. Minimum speed will be required at maximum suction head and minimum flow. The procedure required to determine the speed range of the pump driver requires construction of the system-head curve, the pump head-capacity curve (see Section 8.1), and the affinity curve. Rearranging the affinity equations (Eqs. 1 and 2), the speed changes are calculated:

$$N_2 = N_1\left(\frac{Q_2}{Q_1}\right) \tag{4}$$

$$N_2 = N_1\left(\frac{H_2}{H_1}\right)^{1/2} \tag{5}$$

where the subscripts 1 and 2 represent the higher and lower speed values, respectively, for pump conditions at the same specific speed (see Subsection 2.3.1) or along the same affinity line.

The following examples illustrate how changes in flow rate, pipe frictional losses, and suction head affect pump speed.

EXAMPLE 1 With no change in suction pressure—first (design) conditions:

H_d = 193 ft (58.8 m)
H_s = 50 ft (15.2 m)
H_f = 7 ft (2.1 m)
Q = 190 gpm (43.1 m³/h)
N = 3500 rpm
$H = (193 - 50) + 7 = 150$ ft (45.7 m) (Eq. 3)

Second conditions:

H_d = 193 ft (58.8 m)
H_s = 50 ft (15.2 m)
H_f = 1.9 ft (0.58 m)
Q = 100 gpm (22.7 m³/h)
N = to be calculated
$H = (193 - 50) + 1.9 = 144.9$ ft (44.18 m) (Eq. 3)

The 3500-rpm pump head-capacity curve, the system-head curves, and the affinity (square) curves are shown in Figure 1. The operating points are lettered. Point *A* is for maximum flow at minimum suction head (design conditions). Point *B* is for lower flow at minimum suction head at a speed to be determined. The reduced piping losses between head points H_d and H_s are represented by the system-head curves.

To determine the approximate speed at point *B,* it is necessary to use trial and error because the flow and head ratios in Eqs. 4 and 5 are not known. The following procedure may be used to estimate the speed (Figure 1):

1. Draw an affinity (square) curve passing through the zero-flow/zero-head point and the lower operating point (point *B)* and intersecting at head-capacity curve of known speed (point *C).*

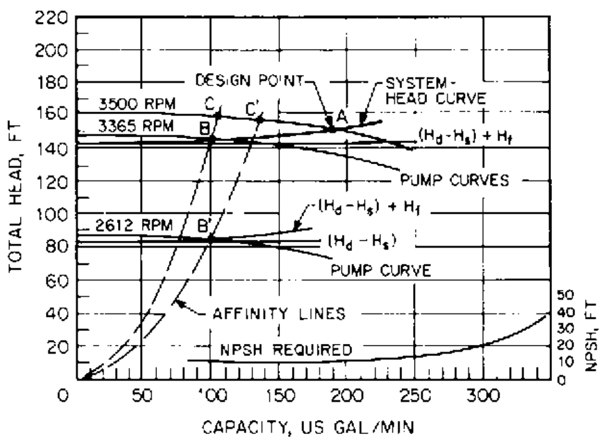


FIGURE 1 System and pump curves to determine speed reduction for the 2-in (51-mm) pump in Examples 1 and 2. (ft × 0.3048 m; gpm × 0.2271 = m³/h)

2. Read the probable flow rate at point C, the point of intersection between the affinity curve and the pump curve of known speed.
3. Calculate the head relative to the flow rate read at point C, based on the square curve relationship:

$$H_1 = \frac{H_2}{\left(\dfrac{N_2}{N_1}\right)^2} \tag{6}$$

where subscripts 1 and 2 represent the values at the 3500-rpm and lower speeds (points C & B), respectively.

4. Compare the head calculated in step 3 with the head of the pump curve (point C) of known speed at the probable flow rate of step 2. If the values differ significantly, try another flow rate and repeat steps 3 and 4.
5. Using the accepted head from step 4, calculate the speed at the operating point (point B) using Eq. 4 or 5. The equation not used may then be used for verification.

For this example, the estimated flow rate at point A (step 2) is 104 gpm (23.6 m³/h) and the calculated head at the same point (steps 3 and 4) is

$$H_1 = \frac{144.9}{\left(\dfrac{100}{104}\right)^2} = 156.7 \text{ ft} \quad \text{or} \quad \frac{44.18}{\left(\dfrac{22.7}{23.6}\right)^2} = 47.76 \text{ m}$$

The speed at the operating point (point B) is

$$N_2 = 3500\left(\frac{100}{104}\right) \quad \text{or} \quad 3500\left(\frac{22.7}{23.6}\right) = 3365 \text{ rpm} \qquad \text{(Eq. 4)}$$

Check:

$$N_2 = 3500\left(\frac{144.9}{156.7}\right)^{1/2} \quad \text{or} \quad 3500\left(\frac{44.17}{47.76}\right)^{1/2} = 3366 \text{ rpm} \qquad \text{(Eq. 5)}$$

The speed change is

$$\frac{3500 - 3365}{3500} \times 100 = 3.9\%$$

EXAMPLE 2 With change in suction pressure—first conditions: same as Example 1.

Second conditions: same as Example 1, except

$$H_s = 110 \text{ ft (33.5 m)}$$
$$H = (193 - 110) + 1.9 = 84.9 \text{ ft (25.88 m) (Eq. 3)}$$

Operating point B' and other curves related to this example are shown in Figure 1. Again by trial and error, the flow rate and pump head at the intersection of the affinity curve and the 3500-rpm pump curve (point A') are 134 gpm (30.4 m³/h) and 152.4 ft (46.45 m). The speed at the operating point is

$$N_2 = 3500\left(\frac{100}{134}\right) \quad \text{or} \quad 3500\left(\frac{22.7}{30.4}\right) = 2612 \text{ rpm} \qquad \text{(Eq. 4)}$$

Check:

$$N_2 = 3500\left(\frac{84.9}{152.4}\right)^{1/2} \quad \text{or} \quad 3500\left(\frac{25.88}{46.45}\right)^{1/2} = 2612 \text{ rpm} \qquad \text{(Eq. 5)}$$

The speed change is

$$\frac{3500 - 2612}{3500} \times 100 = 25.4\%$$

These examples indicate that the speed of the pump is not significantly affected unless there is an appreciable change in suction pressure.

For optimum performance, it should be confirmed, after the speed has been determined, that the most often occurring flow demand range is ideally near the pump's maximum efficiency.

Generally, the design point at maximum speed should be selected to the right of the pump's best efficiency point. The operating flow range at different speeds should be within the hydraulically and mechanically stable ranges of the pump.

The pump shaft power at any specified speed is

in USCS units
$$\text{hp} = \frac{QH(\text{sp. gr.})}{3960E}$$

in SI units
$$\text{kW} = \frac{QH(\text{sp. gr.})}{367E} \qquad (7)$$

where H = pump total head, ft (m)
Q = flow rate, gpm (m^3/h)
sp. gr. = specific gravity of fluid
E = pump efficiency, % (expressed in decimal equivalent)

H, Q, and *E* are values from the pump performance curve for the specified speed and impeller size.

The motor power is the same as the shaft power for pumps driven directly by the motor. For pumps driven through intermediate variable-speed couplings, the motor power must also include the drive slip and fixed loses.

When the required design flow exceeds the capacity of a single pump, several pumps, including possibly one constant-speed unit, can be arranged to operate in parallel. Two methods of staging the pumps are usually used:

1. The first pump is operated in the variable-speed mode until its maximum speed is reached; the second pump, also a variable-speed unit, is energized. Now both pumps are operating in parallel at the same reduced speed. This sequence is repeated for the other pumps.
2. The first pump is operated in the variable-speed mode until its maximum speed is reached and is then locked in at this speed to operate as a constant-speed pump. The second pump is energized and operates in the variable-speed mode. The sequence is repeated for the other pumps. The speed variation of the second pump is relatively small because it must develop the same head as the first pump operating at maximum speed. Because the flow rate through the second pump is less, its speed is affected by the increase in total head available from the rise in the pump performance curve.

Which method of sequencing is selected depends on economics, equipment redundancy, and other considerations. For example, with the first method, both pumps must be furnished with variable-speed drive units. With the second method, only one variable-speed drive unit is required if it is an electronic type because the first pump is locked in by separate electric means.

The sole function of the pumps is to maintain constant pressure at the control point; therefore, the controller selected must be pressure-actuated. The type (follower signal) of controller chosen depends on the type of speed change signal acceptable by the variable-speed control unit. For electronic motor speed controls, such as variable-voltage and variable-frequency units, typical follower signals are low-voltage dc, milliamp dc, 135-ohm potentiometer, and pneumatic.

Tankless Constant-Speed Multiple-Pump System The major components of this system (Figure 2) are

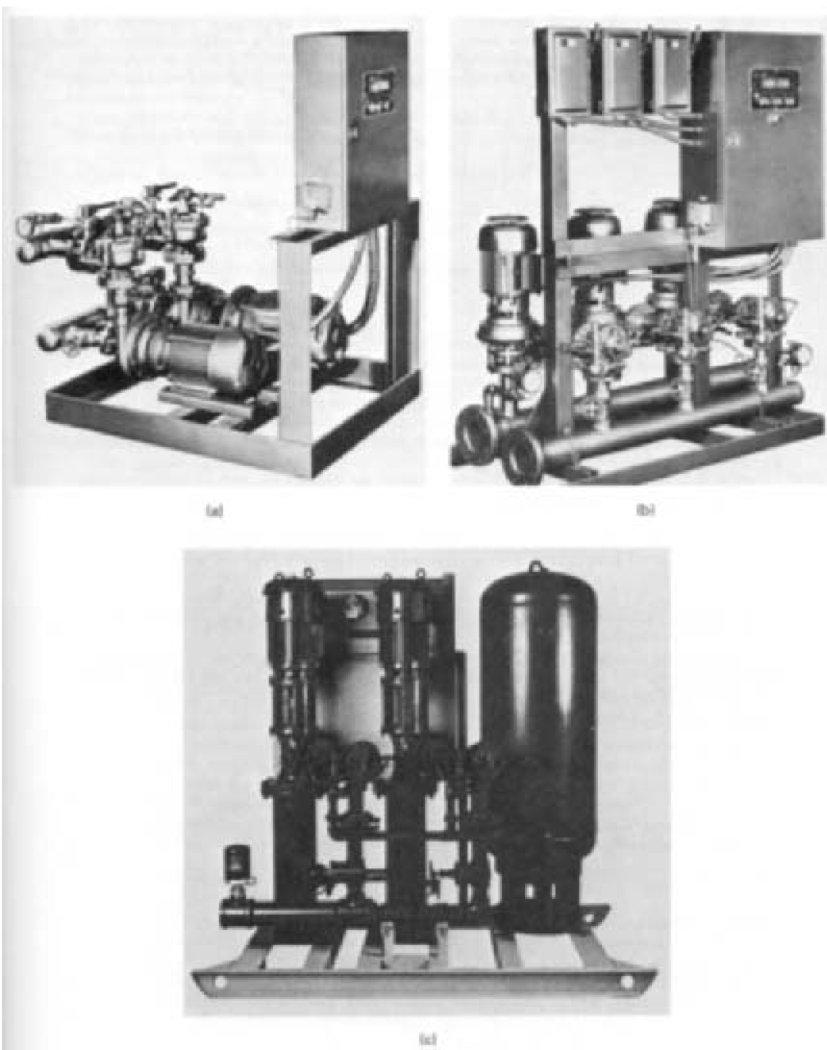


FIGURE 2 Constant-speed multiple-pump pressure booster systems: (a) horizontal end-suction volute pumps (ITT Bell and Gossett), (b) vertical end-suction volute pumps (ITT Bell and Gossett), (c) vertical turbine pumps with limited storage (SynchroFlo)

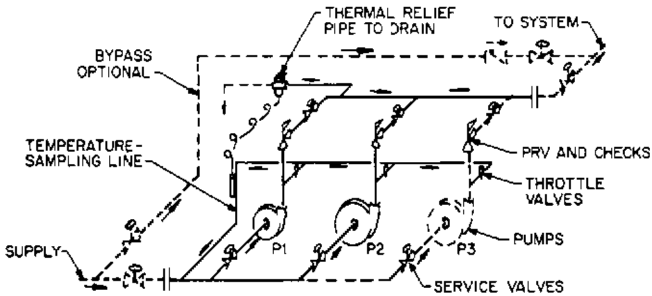


FIGURE 3 Constant-speed multiple-pump pressure booster schematic (ITT Bell and Gossett)

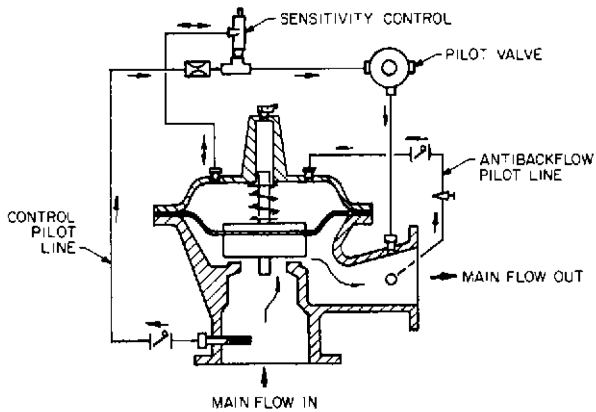


FIGURE 4 Typical pressure-reducing and check valve to maintain constant system pressure and prevent backflow (Cla-Val)

1. One or more pumps, with two or three most common
2. A combination pressure-reducing and check valve (PRCV) for each pump (parallel-piped PRCVs are used with larger sizes; separate pressure-reducing valves and check valves may be used)
3. An automatic sequencing control panel
4. When factory assembled, a steel frame for the entire unit

PIPING ARRANGEMENT AND FLOW PATH A schematic of the piping arrangement and flow path in a constant-speed multiple-pump system is shown in Figure 3. Supply water under fluctuating pressure enters the suction header and flows into the pump, where it is boosted to a higher pressure. This varying high-pressure water enters the PRCV (Figure 4), and the pressure is reduced to the constant pressure desired over the design flow range. Flow reversals through the idle and parallel pump circuits are prevented by the checking feature of the PRCV, which also dampens the pressure fluctuations caused by sudden flow changes.

When only minor changes in supply water pressure are anticipated, such as from a nonpressurized tank, and the pump head-capacity curve is relatively flat, silent check valves without pressure-reducing feature may be used. This will result in a slight increase in discharge pressure above the desired design constant pressure. When pumps of different sizes are used, care should be exercised to avoid having the higher-head pump force the lower-head pump to shut off or to operate at less than minimum design pump flow. A decrease in flow from a centrifugal pump must be accompanied by an increase in pump total head as required by the pump head-capacity curve. If check valves only are used, it is preferable to have all pumps and valves identical to avoid unbalanced flows.

Operating the pumps at shutoff (no flow) will cause the water temperature in the pump castings to rise (Subsection 2.3.1). To keep the temperature in the pumps within a safe limit, the heated water is relieved through the thermal relief valve. This valve may be either a self-actuated (thermostatic) type or a solenoid valve actuated by a temperature controller.

PUMP CONTROL PANEL One of the advantages of a factory-assembled pressure booster package is the prewired, pretested, pretubed, mounted control panel that requires a minimum of field connections. In addition to providing for the proper sequencing of the pumps, the control panel should contain electrical interlocks for the operating and safety controls and circuit connections for remote control units.

Standard items and optional equipment vary considerably from one manufacturer to another. The components usually included with a standard panel are listed:

Steel enclosure
Control transformer
Control circuit protector
Selector switches
Low-suction pressure control
Starters
Sequencing controllers
Pump failure interlocks
Minimum-run timers
Time delays
Pilot lights

Optional features that may be available:

Power supply fused disconnects or circuit breakers
Enclosure door interlock
High water temperature control
High system pressure control
Pump alternation
Program time switch
Elapsed time meters
Low system pressure control
Low water level control
Emergency power switchover
Unit failure alarm
Low-flow shutdown
Miscellaneous enclosure types
Power economizer circuit
Additional pilot lights

Most factory-wired panels conform to one or more of the consumer safety agencies, such as Underwriters' Laboratories (UL), National Electrical Code (NFPA/NEC), and Canadian Standards Association (CSA) and are furnished with a label so indicating.

PUMP CONTROL SEQUENCE A typical elementary wiring diagram provides the following sequences of events. With both pump selector switches in the auto mode and all safety and operating controls in run status, the lead pump starter is energized, starting pump 1. As the system water demand increases, a staging control switch, which senses motor current, flow, or system pressure, starts pump 2. Pump 2 continues to operate until the decreasing water demand causes the staging control switch to open, stopping pump 2. If the circuit is provided with a minimum-run timer, pump 2 will continue to run for the set time period regardless of the staging switch status. This timer prevents pump 2 from short-cycling during rapidly fluctuating demand periods.

For test purposes and emergency operation, both pumps may be operated by placing the selector switches in the *hand* mode. In this position, most of the safety and operating controls are bypassed.

Should pump starter 1 fail to operate because of an overload heater relay trip or starter malfunction, a failure interlock switch automatically starts pump 2.

Pump 1 will run continuously unless the circuit is provided with shutdown features, such as high-suction pressure control and/or low-flow shutdown control.

Many units are specified for manual or automatic alternation of the equally sized pumps. This feature is intended to equalize the wear among the pumps and associated components. Alternation of any pump designated for *standby duty* (emergency use) may not be prudent. The standby pump should be preserved until needed, similar to an emergency power generator.

Limited-Storage Constant-Speed Multiple-Pump System The most notable shortcoming of the tankless multiple-pump system is the need for a continuously running pump, even at zero water usage. This drawback is remedied by the addition of a pressurized storage tank connected to the high-pressure side of the piping system with low-flow shutdown controls to stop the pumps.

The limited-storage system is not a scaled-down version of the hydropneumatic system. It differs in three important aspects:

1. The primary function of the tank is water storage.
2. Tank pressure is developed by the booster pump.
3. Air and water in the tank are separated by a flexible barrier (diaphragm, bladder) to eliminate direct interaction between the gas and water in the tank. The barrier also prevents the gas from escaping when the tank is emptied of water.

SEQUENCE OF OPERATION CONTROL CIRCUITRY During periods of normal water usage, the operating sequence is that of the tankless unit. As the water flow approaches zero, a low-flow sensing device stops the pumps. The tank provides the water needs during the shutdown period until the water pressure diminishes to the minimum allowable value. At this time, a pressure switch starts the lead pump to restore the pressure and recharge the tank with water.

The low-flow device may be a flow switch, a pressure switch, or a temperature-actuated switch and must be capable of switching at flows below 5 gpm (1.14 m^3/h).

The tank should be sized for sufficient drawdown volume (water available from tank during shutdown) to avoid excessive pump cycling.

RELATIONSHIPS BETWEEN TANK SIZE, DRAWDOWN VOLUME, AND PRESSURE The approximate formula for sizing prepressurized diaphragm tanks at constant temperature is

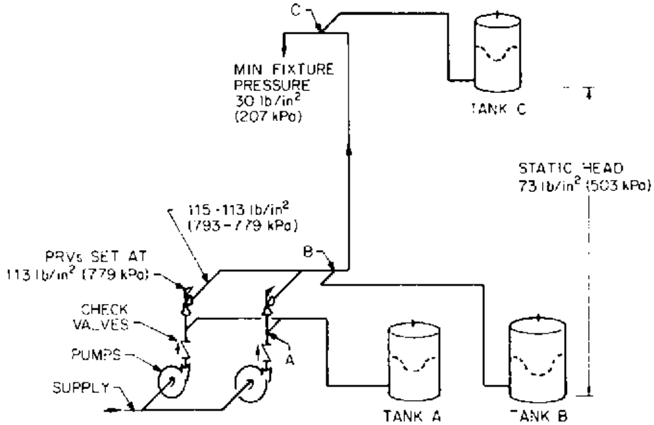
$$V_t = \frac{V_e}{1 - \dfrac{P_f}{P_0}} \tag{8}$$

where V_t = tank volume, ft^3 (m^3)
V_e = change of gas volume in tank, ft^3 (m^3)
P_f = final gas pressure lb/in^2 (kPa) abs
P_0 = initial gas pressure, lb/in^2 (kPa) abs

The corresponding relationships of water volume and pressures in the tank are

- V_e is equal to the drawdown volume.
- P_f is the water pressure at the end of the drawdown cycle, the minimum allowable system pressure.
- P_0 is the water pressure at the beginning of the drawdown cycle or at the termination of the charging cycle.

TANK LOCATION Because both P_f and P_0 are influenced by the static head above the tank and the available charging pressure, the location of the tank and point of connection to the system should be carefully selected. Connecting the tank at *A* in Figure 5 provides the highest available tank charging pressure because this pressure is not affected by the reduction through the PRVs. However, there are several disadvantages to this point of connection:



Tank Location Comparisons

Tank location	Drawdown volume gal	(liters)	Tank size gal	(liters)	Min. operating pressure lb/in²	(kPa)	Max operating pressure lb/in²	(kPa)
A	20	(76)	128	(484)	104	(717)	126[a]	(869)[a]
B	20	(76)	216	(818)	103	(710)	115	(793)
C	20	(76)	95	(360)	30	(207)	42	(290)

[a]Higher when suction pressure rises above design pressure

FIGURE 5 Relative effect of tank location on tank size and pressure for equal draw-down volumes (ITT Bell and Gossett)

1. The highest static pressure is applied to the tank because the pumps are usually located on the lowest floor of the building.
2. The tank may be subjected to high working pressure because any increase in suction pressure above the minimum design pressure is additive to the pump head.
3. The tank and all alternated lead pumps must be interconnected.
4. A silent check valve must be placed between the tank connection point and the pump discharge nozzle.

Connecting the tank in *B* in Figure 5, downstream of the PRVs, will eliminate disadvantages 2, 3, and 4. The available charging pressure is less, and therefore a larger tank is required. However, tank location is not critical.

Locating the tank on the upper elevation of the building, such as at *C,* reduces the static head pressure on it; consequently, a smaller tank will suffice. The maximum design working pressure is also lower, and therefore it may be possible to use a tank with a lower pressure rating at less cost.

PRESSURE BOOSTER SYSTEM SIZING

Design Data The proper sizing of a pressure booster system is subject to the accuracy of the design data available. Experience has indicated that the greatest single cause for unsatisfactory pressure booster performance is oversizing. Submitted data may be rough estimates or historical data not relevant to the location; therefore, their source and accuracy should be scrutinized. Pertinent factors are the flow demand rate and hourly profile, pressure conditions, and types of building occupancy.

FLOW DEMAND RATE AND HOURLY PROFILE Estimating water demand rate is one of the most difficult and controversial subjects. To determine the water demand, many designers refer to the classical Hunter curve (National Bureau of Standards, Report BMS79, by R. B. Hunter) or a modified version of it for lack of a more accurate method. Numerous studies have indicated that the Hunter method often results in very appreciable demand rate overstatement. It should be noted that the Hunter curve was intended for determining pipe sizes, not water demand.

The average hourly demand profile is helpful in proportioning the system demand rate among the pumps in the system. Optimum selection results in the lowest total energy consumption by the pumps while maintaining the demand requirements.

PRESSURE CONDITIONS Four accurate pressure values are required: design system pressure, minimum suction pressure, maximum suction pressure, and minimum allowable system pressure. The difference between design system pressure and minimum suction pressure is used to determine the developed head of the pump at design flow. Inaccurate values will result in incorrect impeller sizing and incorrect motor power selection.

The design system pressure can be calculated with reasonable accuracy. However, when the water supply is from the municipal water main, obtaining accurate values of suction pressures for the proposed site of installation is very difficult because of the absence of recent pressure data. Often the minimum and maximum pressures are haphazardly estimated from average citywide values. A minimum suction pressure error as small as 5 lb/in^2 (35 kPa) could result in selection of the next larger motor.

The maximum suction pressure will determine the working pressures required of the pumps and piping and whether there is sufficient pressure to justify a bypass connection (Figure 3) paralleling the pumps.

Knowing the minimum allowable system pressure is helpful in setting controls associated with this pressure.

TYPE OF OCCUPANCY If the hourly demand profile is not available, the type of building occupancy could be used to determine the standby capacity requirement, whether low-flow shutdown is applicable, and the number of pumps for the system. Most pressure booster manufacturers provide data for determining demand, capacity splits, and pump sizing.

Pump Sizing Procedures Information should be collected on system information and component performance. The system information required is

Q_d = total design capacity, gpm (m^3/h)
P_d = design pressure at system header outlet, lb/in^2 (kPa)
P_s = minimum available pressure at suction header inlet, lb/in^2 (kPa)
Q_1, Q_2, Q_3 = design capacity of individual pumps, gpm (m^3/h)

Components performance data required are

- Pump performance characteristic curve
- Pressure-reducing valve (and check valve) flow chart* (Figure 6)

Next, the required pump total head at the maximum flow design condition should be calculated:

$$H = (H_d - H_s) + H_v + H_u \tag{9}$$

where H = required total head, ft (m)
H_d = design system pressure at PRCV outlet, ft (m)

*Add loss through separate check valve if not integral with PRV.

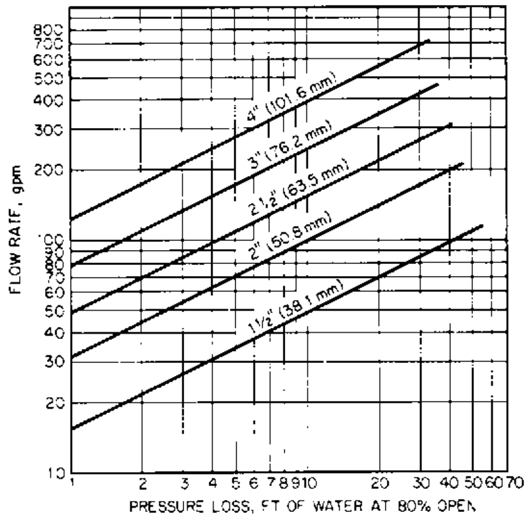


FIGURE 6 Typical PRV or PRCV flow chart (ft × 0.3048 = m; gpm × 0.2271 = m³/h; in × 25.4 = mm) (Cla-Val)

H_s = minimum design suction pressure, ft (m)

H_v = PRCV pressure loss, ft (m)*

H_u = sum of all unaccounted losses in the pressure booster package, ft (m)

For pressure booster applications, the PRCV is usually sized for a nominal pipe velocity between 8 and 18 ft/s (2.4 and 5.5 m/s). The pressure loss H_v is based on the valve being 80% open (Figure 6). Operating the valve at less than full-open position is desirable for good pressure regulation. The term H_u represents the allowance for piping frictional losses between the suction header inlet and the system header outlet at design capacity, exclusive of the PRCV loss. Manufacturers of booster packages differ in determining this value. Some ignore it completely to compensate for usual oversizing as the result of inaccuracies in the design capacity and minimum suction pressure statements. Others assign a fixed value, about 5 ft (1.5 m), or a percentage, about 3%, of the difference of system and suction pressures ($H_d - H_s$). It is recommended that some value be used.

Then determine the best combination of pump and PRCV sizes. The selection will depend on whether the design criterion is least capital cost or lowest operating cost. Usually the smallest pump and PRCV combination will result in lowest first cost and the pump equipped with the smallest motor will yield the lowest operating cost. Various pump, PRCV, and motor combinations may be examined by plotting system curves (Figure 7) for two or three PRCV sizes on one or more pump performance curves.

The pumps tentatively selected should exhibit the highest efficiency at the most frequent operating flow point on their head-capacity curves. When pumps are operated at suction pressures higher than design minimum, the PRCVs will throttle the excess suction head and maintain the same pump total head and constant design discharge pressure. When the flow demand is less than the design maximum, characteristically, the head on the centrifugal pumps will increase to some higher value on the pump head-capacity curve. The PRCVs will reduce this excess head to maintain the constant design discharge pressure. Therefore, each pump operates along its own head-capacity curve, which is also the adjusted system-head curve.

Add loss through separate check valve if not integral with PRV.

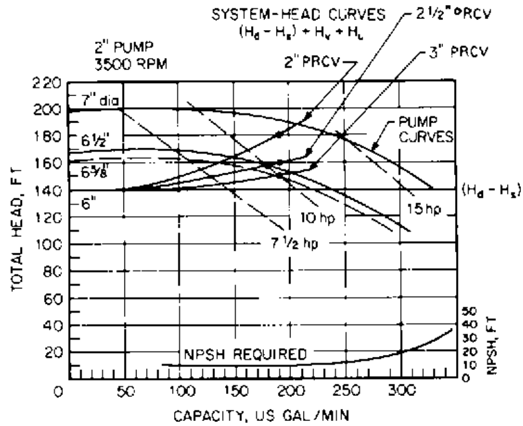


FIGURE 7 System curves for PRCV sizing (ft × 0.3048 = m; gpm × 0.2271 = m³/h; hp × 0.7457 = kW; in × 25.4 = mm) (ITT Bell and Gossett)

When several pumps are to be used, especially pumps of different sizes, their head-capacity characteristics should be compatible. A pump having a shutoff total head less than the system total head at any operating point (with or without other pumps operating) will not pump. Also, pumps should be so selected and controls should be such that no one pump will operate at too low a flow; that is, at a flow less than the minimum recommended for mechanical and hydraulic stability. If the minimum suction pressure is very low, the net positive suction head required by the pumps throughout their operating flow range must be checked (Subsection 2.3.1).

PUMP AND PRCV SELECTION EXAMPLE

Total system design capacity	Q_d = 380 gpm (86.3 m³/h)
Design capacity	Q_1, Q_2 = 190 gpm (43.1 m³/h)
Design pressure at system header outlet	H_d = 210 ft (64.0 m)
Minimum pressure at suction header	H_s = 70 ft at 100 ft *NPSH* inlet (21.3 m at 30.5 m *NPSH*)
PRCV loss, Figure 6	H_v = 36 ft (11 m) for trial 2-in (51-mm) valve
Allowance for internal frictional losses	H_u = 5 ft (1.5m)

Calculate H:

In USCS units $H = (210 - 70) + 36 + 5 = 181$ ft* (Eq. 9)

In SI units $H = (64.0 - 21.3) + 11 + 1.5 = 55.2$ m

Table 2, calculate and tabulate H for all applicable PRCV sizes up to the design flow rate of 190 gpm (43.1 m³/h). H_v and H_u terms vary as the square of the flow rate.

Now locate H at 190 gpm (43.1 m³/h) on the applicable pump curve. Select the pump and PRCV combination that best satisfies the design criterion of least first cost or lowest energy input.

*Due to the many approximations, decimal amounts are not used in the final results.

TABLE 2 Required head for pressure reducing valves

	Flow, gpm (m^3/h)				
PRV size, in (mm)	0 (0)	50 (11)	100 (23)	150 (34)	190 (43)
2 (51)	140 (42.7)	143 (43.6)	151 (46.0)	166 (50.6)	181 (55.2)
$2\frac{1}{2}$ (64)	140 (42.7)	141 (43.0)	146 (44.5)	153 (46.6)	160 (48.8)
3 (76)	140 (42.7)	141 (43.0)	143 (43.6)	147 (44.8)	151 (46.0)

Note: All values are head in ft (m).

Plot H values on the selected pump curve to obtain the system curves for the three PRCV sizes (Figure 7). The plots reveal that the 3-in (76-mm) valve, a 10-hp (7.5-kW) motor, and $6\frac{3}{8}$ -in diameter (162-mm) impeller are the proper selections.

Note that if the design capacity of the pump could be reduced to about 178 gpm (40.4 m^3/h, a $2\frac{1}{2}$ -in (64-mm) PRCV with the impeller sized for $6\frac{1}{2}$ -in (165-mm) diameter may be used without additional power input. This choice reduces the cost and improves the PRCV performance at low flow rates.

Using a 2-in (51-mm) PRCV would require a 15-hp (11-kW) motor operating at about 12.5 hp (9.3 kW) load. This choice is not energy effective.

The power required to drive the pump may be approximated by using the pump curve and Eq. 7. The available *NPSH* is in excess of the required pump *NPSH* (Figure 7).

PUMP TYPES AND MATERIALS

Single-stage volute centrifugal pumps, followed by multistage vertical turbine diffuser pumps in a suction tank, are most commonly used for domestic water pressure booster systems. The volute pumps can be end-suction with a vertically split case, double-suction with an axially split case, or in-line. For high-pressure service, two-stage axially split case pumps or multistage vertical turbine pumps may be selected.

In selecting the type of pump best suited for the application, such inherent characteristics as low-flow recirculation, low-flow cavitation, high-flow cavitation, steepness of the pump curve, noise, and operating efficiencies should be evaluated. These factors are discussed in Subsection 2.3.1.

The most often discussed factor in booster application is low-flow recirculation. Low pump flow conditions do occur; they cannot be completely designed out of the system. Most well-designed small pumps with positive (above atmospheric) suction pressure can operate safely in the low-flow region, with the only point of concern being water temperature rise at or near shutoff flow. Should radial thrust be a major consideration, double-volute pumps or diffuser (vertical turbine) pumps offer advantages.

Vertical pumps generally require less space. Pumps may be close-coupled to their drivers, or, if not, a coupling requiring careful alignment is required. Close-coupled pumps may require less maintenance, as there are no pump bearings. Shaft sealing may be either packing or mechanical seal.

All pumps and drivers in packaged units are mounted on a common steel frame, but individual bases for each pump and driver may be necessary for larger units.

The materials of construction for pumps, valves, and piping should be suitable for the water quality and conditions furnished to the system. Because water impurities vary from location to location, careful analysis is recommended. Federal and local agencies, such as the Food and Drug Administration, may also prescribe allowable materials.

FURTHER READING

American Society of Heating, Refrigerating, and Air-Conditioning Engineers. *Equipment Handbook,* ASHRAE, Atlanta, GA, 1983.

ITT Bell and Gossett. "Domestic Water Service." Bulletin TEH-1175, Morton Grove, IL, 1970.

ITT Fluid Handling Division. "Pressurized Expansion Tank Sizing/Installation." Bulletin TEH981, Morton Grove, IL, 1981.

Potter, P. J. *Steam Power Plants,* Ronald Press, New York, 1949.

Steel, A. *High Rise Plumbing Design,* Miramar, Los Angeles, 1975.