A GORMAN-RUPP COMPANY

PATTERSON PUMP COMPANY

HVAC PUMPS & SYSTEMS

DESIGN MANUAL

A COMPLETE GUIDE TO HVAC PUMPS & SYSTEMS
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HVAC PUMPS AND SYSTEMS DESIGN MANUAL

PATTERSON PUMP COMPANY
A Gorman Rupp Company

James B. (Burt) Rishel, P.E.

INTRODUCTION

The development of digital electronics has brought many tools for the proper selection and application of centrifugal pumps, particularly in the HVAC world where there are so many different piping systems required for the distribution and reclamation of heating and cooling. Patterson Pumps, through its many years of applying large centrifugal pumps, has proved conclusively that the main causes of pump wear and failure are improper selection and application. This manual will provide guidelines for the selection and application of centrifugal pumps to the HVAC industry.

This manual is based upon forty years of applying pumps, variable speed drives, and controls to heating, cooling, and condenser water installations. Much of this information has been derived from factory testing of pumps themselves and field testing of complete water systems. Credit for many of the pump applications included herein must be given to forward-thinking engineers who have sought more efficient answers to age-old application problems in this field.

The fundamental purpose of this manual is to improve the **selection and application** of centrifugal pumps to hot, chilled, and condenser water systems. It has been developed for the HVAC engineer who has a general knowledge of pump application in this industry.

A very detailed evaluation will be made of the forces on centrifugal pumps which have a great effect upon the useful life of a pump. This will enable the HVAC system designer to select pumps so that long life with a minimum of maintenance will be achieved for them.

The procedures now available for the analysis of a prospective HVAC water system enable us to compute carefully the flow and pump head for that system. A number of hydraulic gradients will be provided to enable the designer to determine carefully the use of pump head in typical water systems. Old piping methods such as cross-over bridges and the use of energy-wasting devices such as multiple duty valves, and balance valves are eliminated through the hydraulic evaluation of these water systems.

Relatively new technology such as Variable Primary Pumping Systems on chilled and hot water will be reviewed carefully for designers of HVAC systems. The use of variable speed drives to minimize energy consumption and over-pressuring of water systems will be evaluated in detail. The use of kW input for programming multiple pump installations will be described with data that will indicate the points of addition and subtraction of individual pumps.

The development of the factory assembled pumping system has provided a new tool in the acceleration of production time and cost reduction of pumping systems in the HVAC industry. Complete central chilled water plants are now available. The advantages for these systems will be provided along with typical general arrangement drawings.

The development of variable speed pumping systems and pressure independent coil control valves has eliminated the need for manual balancing of most HVAC water systems. This has not reduced the need for the commissioning of these systems.

Proper commissioning techniques along with factory and field testing of HVAC pumping systems will provide the minimum of energy consumption along with extended useful life with little maintenance.

Particular emphasis will be made on pump selection. Improper selection of pumps dooms the HVAC water system to poor operating efficiency, increased noise, and unnecessary pump maintenance. Likewise, use of outdated mechanical devices such as cross-over bridges, multiple duty valves, and balance valves increase the energy consumption of the HVAC water system.

Reiterating, the purpose of this manual is to assist the HVAC designer in the selection and application of centrifugal pumps. It is hoped that the detailed evaluation of centrifugal pump performance and many examples of system evaluation will accomplish this purpose.

James B. (Burt) Rishel, PE

Note: Many of the drawings in this manual are from the author's books and magazine articles. These include the “HVAC Pump Manual”, McGraw-Hill, 1996, and “Water Pumps and Pumping Systems”, McGraw-Hill, 2002.

Another excellent book on detailed pump design is the “Pump Handbook” by Karassik, Krutzsch, Frazer, and Messina, Second edition, McGraw-Hill, 1986.

Chapter 1

PATTERSON PUMP COMPANY

A Gorman-Rupp Company

Patterson Pump Company, located in the beautiful, northeast Georgia, mountain city of Toccoa, is a worldwide leader in the design and manufacture of pumping equipment. Patterson has become the company of choice when it comes to reliable pumping needs around the world. Our pumping equipment is there to satisfy urban water and waste water demands as well as the protection of life and property through our fire pump lines.

Patterson is a wholly owned subsidiary of The Gorman-Rupp Company. Gorman-Rupp was founded in 1933 during the worst depression in U.S. history. Today, the company stands as a leader in its field and boasts a history of innovation, improvement and quality that continues to set the standards for the industry.

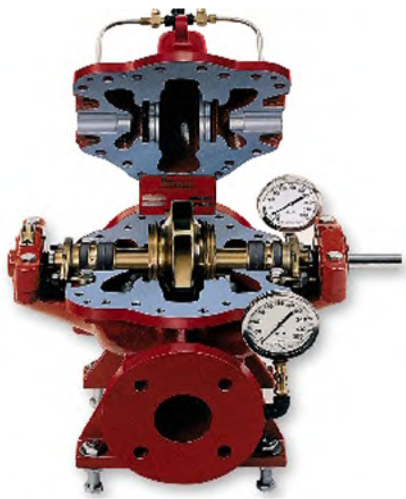
Patterson's main domestic facilities consist of a modern 25,000 square foot office complex as well as over 250,000 square feet of manufacturing facilities. In addition, we operate Flo-Pak, a business unit of Patterson Pump Company, located in Buford, Georgia. Flo-Pak manufactures packaged pumping stations for the Heating, Ventilation and Air Conditioning (HVAC), Plumbing, Municipal, Power, and Industrial markets.

Patterson reliability is enhanced by coordinated training in proper operation and maintenance of pumping products at its modern training facility in Toccoa. One of very few manufacturers to furnish such training, Patterson considers it essential to providing full service to its industrial, municipal and governmental agency customers.

On the international side, our facilities include: Patterson Pump Ireland, Limited, a subsidiary of Patterson Pump Company located in Mullingar, Ireland. We have regional sales offices in Athens, Greece; Nottingham, England; and Chiangmai, Thailand all with worldwide, factory trained, representation and service.

Our ISO 9001 certification attests to our world class quality and dependability. We are committed to maintaining our longstanding record of product reliability through expanded production fabrication facilities and the latest in high tech, computer aided fabrication tools and techniques.

Implementation of a Six Sigma program emphasizes our desire to continue to achieve world-class performance which results in delivering top quality products and services.



Patterson Pump Company continues to set the standard for the industry by offering a full line of modern, high-performance pumps in both domestic and international markets by offering:

- Industrial and commercial pumps;
- Horizontal and vertical centrifugal pumps;
- Fire pumps for a variety of safety applications;
- Non-clogging waste and sewage pumps;
- Axial and missed flow pumps for flood control and irrigation;
- Multi-purpose vertical turbine pumps;
- Patterson vertical turbine pumps;
- General service pumps; and prepackaged pump systems.

Quality workmanship, superior pump design knowledge, efficient production capability and careful attention to testing details ensure that every Patterson pump will perform its intended function efficiently, economically and durably. Whether the systems are electric, diesel engine, turbine or dual drive all provide guaranteed satisfaction of individual application time after time.

Since our first day in business, Patterson Pump Company has been committed to producing superior products that meet or exceed the individual needs of our customers. That commitment is even stronger today and is evidenced by the increasing growth and diversification of Patterson Pump Company.

The full range of Patterson Pump's leadership in design, engineering and fabrication is available now for application to your pumping needs. Discover for yourself why Patterson is the world leader in reliability.

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9201 Ayersville Road
Toccoa, Georgia 30577

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Chapter 2

PHYSICAL DATA FOR HVAC SYSTEMS

HVAC PUMPS AND SYSTEMS
DESIGN MANUAL

PATTERSON PUMP COMPANY

This chapter provides physical data for water and air that is used in the proper design of water systems and operation of these pumps. All of this information is important in this design. It is printed here, as it is not found in many text books.

INTRODUCTION:

It is important to understand the standards that exist for the design and operation of HVAC pumps. These standards have been prepared by technical societies, governmental agencies, trade associations, and as codes for cities, counties, states, and the federal government. The HVAC water system designer must understand the standards and codes that apply to each water system.

STANDARD OPERATING CONDITIONS:

All HVAC equipment has its design based upon some particular operating conditions such as maximum temperature or pressure; usually, these conditions are established by the manufacturer. The HVAC pump designer should understand the conditions that exist for each application. Variations in electrical service as well as maximum ambient air temperature should be confirmed for each installation.

STANDARD AIR CONDITIONS:

There are two standard air conditions that must be defined for each installation. These are: **ambient air** and **ventilation air**. Ambient air is the air in which all HVAC equipment must operate. Ambient air standard is usually 70° F while maximum ambient air temperature can be 40° C (104° F). For some boiler room work, the ambient air may be as high as 60° C (140° F). The designer must insure that the equipment is operable with such ambient air conditions.

The designer must be concerned also with the quality of **ventilation air.** This air cools the running equipment and provides ventilation for the building. The designer must check to insure that the equipment room is free from chemicals that might exist nearby. Likewise, dusty processes must be kept way from this equipment.

ASHRAE Standard 62 defines ventilation requirements for various buildings as well as for HVAC processes. This document, while controversial, should be available to all designers of HVAC water systems.

OPERATING PRESSURES:

Following is the basic equation for gauge, absolute, and atmospheric pressures.

$$\textbf{psia} = \textbf{psig} + p_e \qquad \textbf{(2.1)}$$

where: psia = Absolute pressure, pounds per square inch
psig = Gauge pressure, pounds per square inch
p_e = Atmospheric pressure, pounds per square inch

If a water system is operating at a pressure 90 psig at an elevation of 2,000 feet, the absolute pressure would be 90 plus 13.7 (from table 2-1) or 103.7 psia

TABLE 2-1: Variation of Atmospheric Pressure with Altitude

Altitude in Feet	Atmospheric Pressure, p_e	Atmospheric Pressure, ft. of Water, Up to 85°F
0	14.7	34.0
500	14.4	33.3
1,000	14.2	32.7
1,500	13.9	32.1
2,000	13.7	31.6
2,500	13.4	31.0
3,000	13.2	30.5
4,000	12.7	29.3
5,000	12.2	28.1
6,000	11.8	27.2
7,000	11.3	26.1
8,000	10.9	25.1
9,000	10.5	24.2
10,000	10.1	23.3

THERMAL EQUIVALENTS:

It is necessary to establish the relationship between values of heat and power. The values in this manual are based upon Keenan and Keyes' *Thermodynamic Properties of Steam* which defines1 BTU as 778.26 ft-lb.

One BTU (British Thermal Unit) = 778.26 ft-lb

One brake horsepower, BHP = 33,000 ft lb/min

One brake horsepower hour, bhphr = 2,544 BTU/hr

= 0.746 Kilowatt hour, KWH

One KWH = 1.341 BHP

= 3,413.0 BTU/hr

VAPOR PRESSURES AND SPECIFIC WEIGHTS OF WATER:

Vapor pressures and specific weights of water are provided for NPSHA calculations a well as for high temperature water installations. Vapor pressure is the absolute pressure at which water turns from a liquid to a vapor at a specific temperature.

Table 2-2: Up to 212°F for NPSHA Calculations:

Temperature °F	Absolute Pressure Feet of Water	Specific Weight, γ lb/ft³
32	0.20	62.42
40	0.28	62.42
45	0.34	62.42
50	0.41	62.38
55	0.49	62.38
60	0.59	62.34
65	0.71	62.34
70	0.84	62.31
75	0.99	62.27
80	1.17	62.19
85	1.38	62.15
90	1.62	62.11
95	1.89	62.03
100	2.20	62.00
105	2.57	61.92
110	2.97	61.84
115	3.43	61.77
120	3.95	61.70
130	5.20	61.53
140	6.78	61.39
150	8.75	61.20
160	11.19	61.01
170	14.19	60.79
180	17.85	60.57
190	22.28	60.35
200	27.60	60.13
210	33.97	59.86
212	35.39	59.84

Derived from Keenan and Keyes *Thermodynamic Properties of Steam,* John Wiley and Sons, New York, 1936

Table 2-3: 212 to 450° F for High Temperature Water Installations:

High temperature water systems are not common in HVAC installations, but they are still used for heating large facilities such as military bases. It is critical that the pumps be selected for these higher temperatures and that adequate expansion provisions are included in the piping.

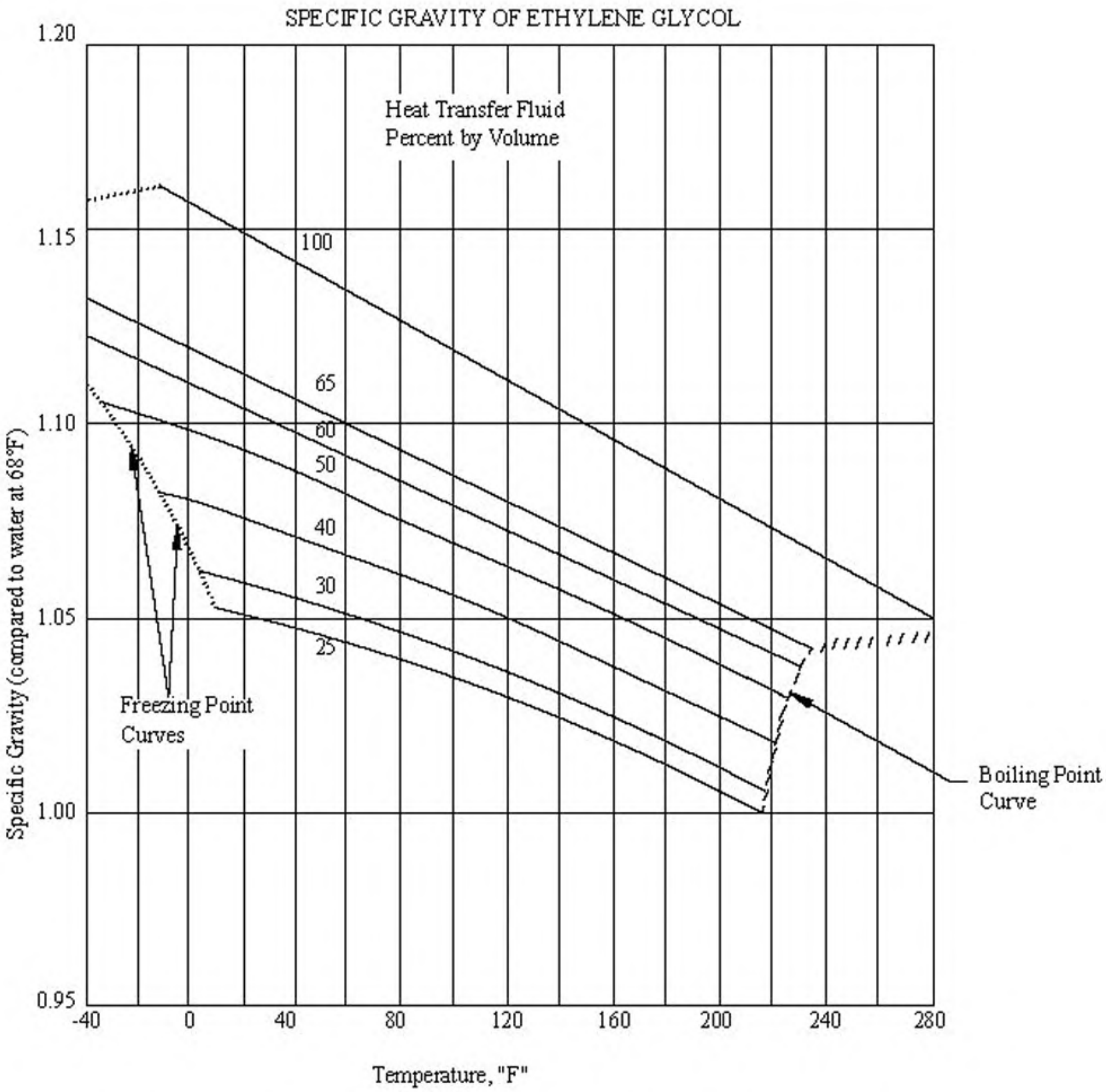
Temperature °F	Absolute Pressure psia	Specific Weight, γ lb/ft³
212	14.71	59.84
220	17.19	59.60
230	20.78	59.38
240	24.97	59.10
250	29.83	58.82
260	35.43	58.51
270	41.86	58.24
280	49.20	57.94
290	57.56	57.64
300	67.01	57.31
320	89.66	56.66
340	118.01	55.96
360	153.03	55.22
380	195.77	54.47
400	247.31	53.65
420	308.83	52.80
440	381.59	51.92
450	422.60	51.55

Source: Keenan and Keyes *Thermodynamic Properties of Steam,* John Wiley and Sons, New York, 1936

GLYCOL BASED HEAT-TRANSFER-FLUID (HTF) SOLUTIONS:

These solutions are used to reduce the freezing point of the liquid to avoid actual freezing in HVAC equipment. They are also used to transfer heat from energy storage tanks that use ice. There are several types of glycol such as propylene and ethylene. Figures 2-1 and 2-2 describe typical specific gravities and specific heats for ethylene solutions.

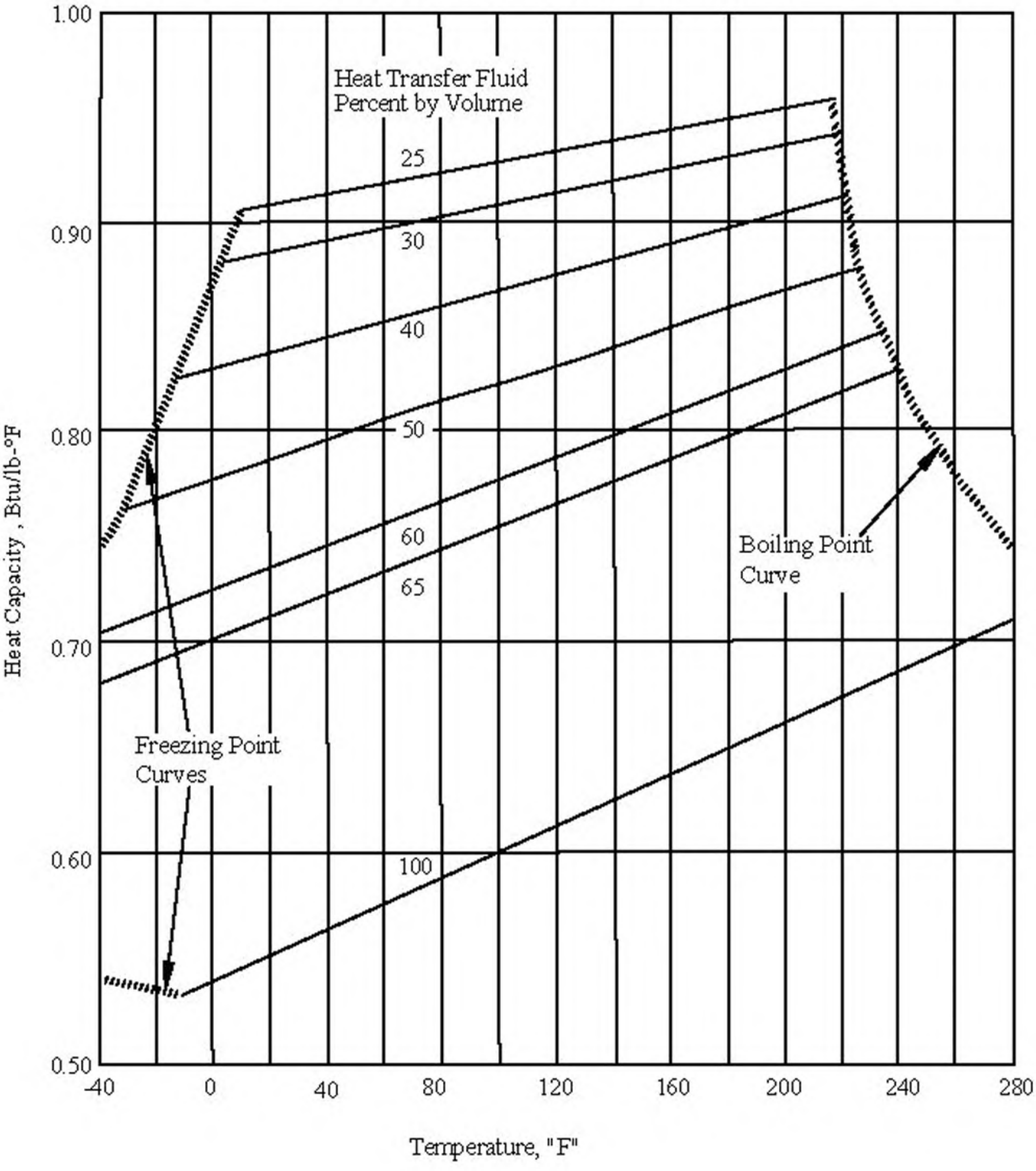
Figure 2-1



Specific gravity of heat transfer fluids.(From Engineering Data for Ethylene Glycol-Based Heat Transfer Fluids, Union Carbide Corporation, Danbury, Conn. 1993

Figure 2-2

SPECIFIC HEAT OF ETHYLENE GLYCOL



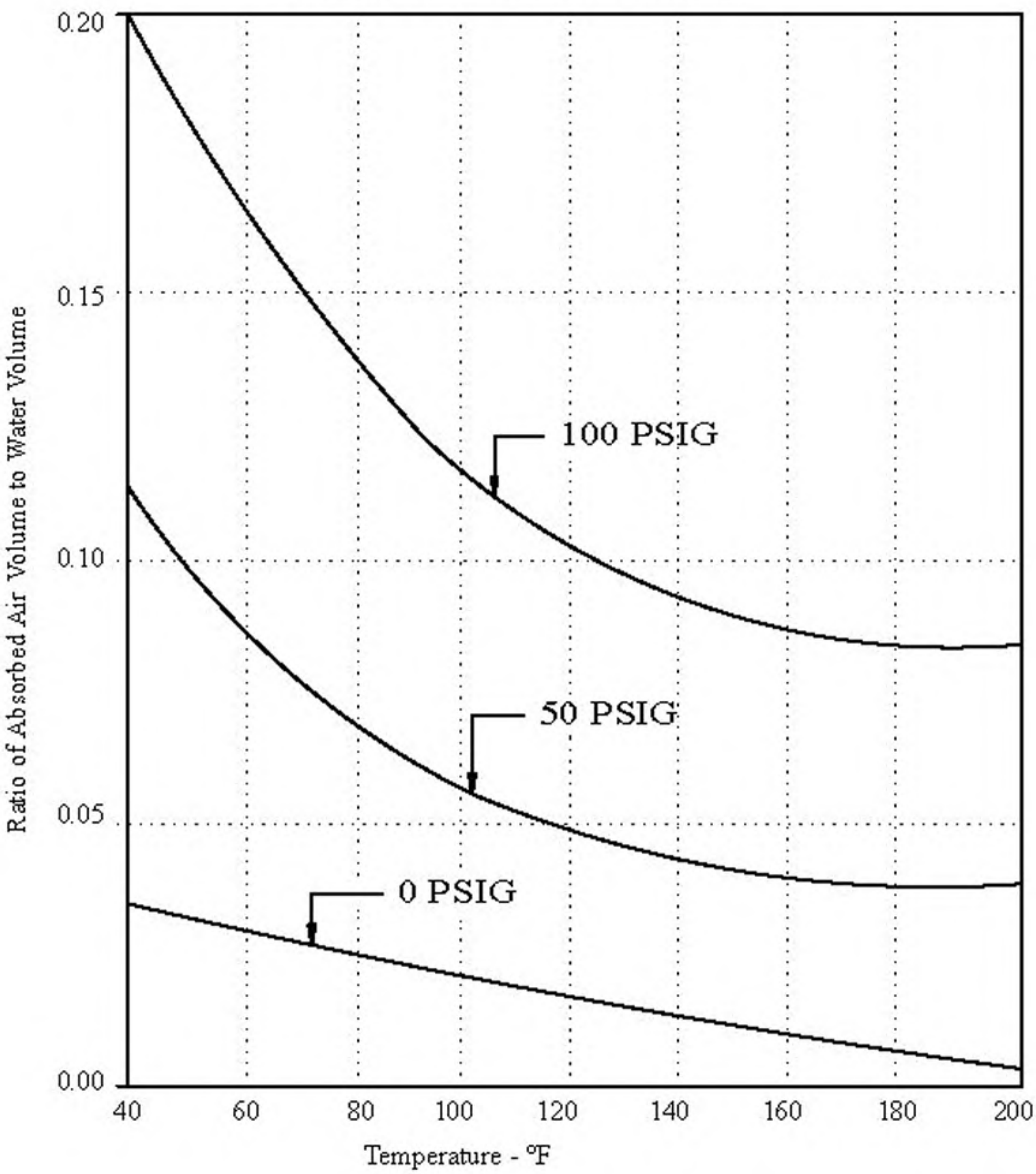
The designer of the HVAC system will select the type and percent solution required for a specific application. From such figures, knowing the particular specific weight and heat, the actual volume, in gpm, can be calculated along with the brake hp required for the actual specific gravity.

SOLUBILITY OF AIR IN WATER:

Air In water is an unsatisfactory condition that can lead to a number of troubles with pumps and corrosion in HVAC water systems. Figure 4-17 in Chapter 4 describes the loss of pump capacity/ when air is present. Obviously, the presence of air in steel piping systems contributes to the oxygen attack and the resultant rusting of the pipe.

Figure 2-3 describes the solubility of water that decreases with the water temperature and increases with the system pressure. This demonstrates that the lower the system pressure, the less ability that the water has to retain air and, therefore oxygen. Well designed HVAC water systems minimize overall water pressure and include air separation devices and air vents that help in the removal of oxygen from the system. Also, chemical feeders are now available to eliminate any residual oxygen.

FIGURE 2-3
SOLUBILITY OF AIR IN WATER



SOURCE: WATER PUMPS AND PUMPING SYSTEMS, McGRAW-HILL, NEW YORK, 2002.

STEAM DATA:

Steam data is needed for the selection of boiler feed pumps. This is not a broad field in the HVAC industry, but it can be troublesome if the actual suction and discharge conditions are not defined thoroughly for these pumps. .

There are two steam pressure ranges established by the ASME Boiler Codes: (1) up to 15 PSIG (250 degrees F.) and (2) above 15 PSIG steam pressure. These boiler codes are: (1) the Heating Boiler Code for pressures up to 15 PSIG steam pressure or hot water at or below 250°F and (2) the Power Boiler Code for pressures above 15 PSIG steam pressure as well as high temperature water boilers. See Chapter 8 for additional information on boilers.

TABLE 2-4: Basic Steam Data

ABSOLUTE PRESSURE lb/in²	STEAM TEMPERATURE °F	ENTHALPY, BTU/lb		
		SAT. LIQUID	EVAP.	SAT. VAPOR
14.7	212.00	180.07	970.3	1150.4
16	216.32	184.42	967.8	1152.0
18	222.41	190.56	963.6	1154.2
20	227.96	196.16	960.1	1156.3
22	233.07	201.33	956.8	1158.1
24	237.82	206.14	953.7	1159.8
26	242.25	210.62	950.7	1161.3
28	246.41	214.83	947.9	1162.7
30	250.33	218.82	945.3	1164.1
35	259.28	227.91	939.2	1167.1
40	267.25	236.03	933.7	1169.7
45	274.44	243.36	928.6	1172.0
50	281.01	250.09	924.0	1174.1
60	292.71	262.09	915.5	1177.6
70	202.92	272.61	907.9	1180.6
80	312.03	282.02	901.1	1183.1
90	320.27	290.56	894.7	1185.3
100	327.81	298.40	888.8	1187.2
125	344.33	315.68	875.4	1191.1
150	358.42	330.51	863.6	1194.1
174	370.29	343.10	853.3	1196.4
200	381.79	353.36	843.0	1198.4

NOTE: **Absolute pressure = Gauge + Atmospheric pressures.**

SOURCE: **Keenan & Keyes,** *"Thermodynamic Properties of Steam"*,
John Wiley & Sons, New York, NY

AREAS AND VOLUMES OF STEEL PIPE AND TANKS:

The following Table 2-5 provides the area, in square feet, and the volume, in gallons, of commercial steel pipe and circular tanks per linear foot of such pipe and tanks. This data is valuable for computing HVAC water system and tank volume.

TABLE 2-5: Areas and Volumes of Steel Pipe and Tanks

Inside Diameter	Area-ft^2	Volume Gal/Ft	Inside Diameter	Area-ft^2	Volume Gal/Ft
1¼"-1.380"	0.0104	0.078	66"	23.758	177.71
1½"-1.610"	0.0141	0.106	72	28.274	211.49
2" -2.067"	0.0233	0.174	84	38.485	287.86
2½"-2.469"	0.0334	0.250	90	44.179	330.46
3" -3.068"	0.0512	0.383	96	50.266	375.99
4" -4.026"	0.0884	0.662	102	56.745	424.45
5" -5.047"	0.1389	1.039	108	63.617	475.86
6" -6.065"	0.2006	1.501	114	70.882	530.20
8" -7.981"	0.3474	2.599	120	78.540	587.48
10"-10.02"	0.5476	4.096	12'	113.097	845.97
12"-11.938"	0.7773	5.814	14	153.938	1,151.46
14"-13.124"	0.9394	7.027	16	201.062	1,503.94
16"-15.000"	1.2272	9.180	18	254.469	1,903.43
18"-16.875"	1.5533	11.619	20	314.159	2,349.91
20"-18.812"	1.9302	14.438	24	452.389	3,383.87
24"-22.624"	2.7917	20.882	30	706.858	5,287.30
30"-29.00"*	4.5869	34.310	36	1,017.87	7,613.71
36"-36.0"**	7.0686	52.873	42	1,385.44	10,363.11
42"-42.0"**	9.6211	71.966	48	1,809.55	13,535.49
48"-48.0"**	12.5664	93.997	60	2,827.43	21,149.18

Notes: * All pipe sizes up to 24" are Schedule 40 while 30" is Schedule 20

** Pipe sizes 36, 42, and 48" are nominal inside diameters.

To convert the above volumes in gallons to pounds of water, multiply gallons by:

$$\text{Lbs of Water} = \frac{\gamma}{7.48}$$

where γ is the specific weight of the water at the operating temperature.

For example, water at 60°F has a specific weight of 62.34 lb/ft^3, so a 10", Schedule 40 steel pipe has 4.096 gal/ft $\cdot \frac{62.34}{7.48}$ or 8.33 lb/ft of length.

From: "HVAC Pump Handbook", James Rishel, PE, McGraw-Hill, New York, 1996.

ELECTRICAL DATA:

Following is information on electrical power supply in the HVAC industry. Electric power utilities are allowed a variation of ±5% from the distribution system voltages listed in these tables. The standard for most electrical distribution in the U. S. is 60 Hz although some remote areas may be supplied with 50 Hz power.

Table 2-6: Standard 60 HZ Voltages:

Nominal Distribution System Voltage	Motor Nameplate Voltage
Polyphase	
208	200
240	230
480	460
600	575
2,400	2,300
4,160	4,000
6,900	6,600
13,800	13,200
Single-phase	
120	115
208	200
240	230

Source: *AC Motor Selection and Application Guide*, Bulletin GET-6812B, General Electric Company, Ft. Wayne, IN, p. 2; used with permission.

TABLE 2-7 Standard 50 HZ Voltages

Distribution System Voltage	Motor Nameplate Voltage
Polyphase	
	200
(See Note)	380
	415
	440
	550
	3,000
Single-phase	
(See note)	110
	200
	220

Source: *AC Motor Selection and Application Guide*, Bulletin GET-6812B,General Electric Company, Ft. Wayne, IN, p.2; used with permission.

The voltage for most three-phase service is 480volts. Single-phase power is mostly 115 volts although 230 volt motors may be available up to 7½ HP. 208 volt service may be used in hospitals for three phase motors as high as 60HP. The higher voltages of 2,400 and 4,160 are economical on motors in sizes of 750HP and larger.

Electrical machinery, such as motors and variable speed drives, must have voltage tolerances greater than those of the electrical utility. The building power distribution system must have a voltage drop that does not exceed the voltage tolerances of the electrical equipment. The voltage tolerance of most electric motors is ±10%, while variable speed drives appear to have +10% and -5% tolerances.

Power Factor correction may be an additional charge above a certain size of motor. This charge may be applied by the public utility when the electrical distribution for an installation exceeds 500kVA.

Table 2-8: Unit Conversion Tables

UNIT x	FACTOR =	UNIT
Acceleration gravity	9.80665	meter/second^2
Acceleration gravity	32.2	feet/second^2
Acceleration gravity	9.80665	meter/second^2
Acceleration gravity	32.2	feet/second^2
acre	4,046.856	meter^2
acre	0.40469	hectare
acre	43,560.0	foot^2
acre	4,840.0	yard^2
acre	0.00156	mile^2 (statute)
acre	0.00404686	kilometer^2
acre	160	rods^2
acre feet	1,233.489	meter^2
acre feet	325,851.0	gallon (US)
acre feet	1,233.489	meter^3
acre feet	325,851.0	gallon
acre-feet	43560	feet^3
acre-feet	102.7901531	meter^3
acre-feet	134.44	yards^3
ampere	1	coulombs/second
ampere	0.0000103638	faradays/second
ampere	2997930000.0	statamperes
ampere	1000	milliamperes
ampere/meter	3600	coulombs
angstrom	0.0001	microns
angstrom	0.1	millimicrons
atmosphere	101.325	kilopascal
atmosphere	1.0332	kg/cm^2
atmosphere	0.10133	megapascal
atmosphere	14.7	pound force/inch^2
atmosphere	101325.0	newtons/meter^2
atmosphere	760	torrs
atmosphere	1.01325	bars
atmosphere	33.8995	feet of H_2O @ 40°F
atmosphere	1033.29	cm of H_2O @ 4°C
atmosphere	76	cm of Hg @ 0°C
atmosphere	29.530	inches of Hg @ 32°F
atmosphere	760	mm of Hg @ 0°C
bars	.98692	atmosphere
bars	.1	kilopascal
bars	14.50377	pound force/inch^2
bars	1019.72	grams force/cm^2
bars	75.0062	cm of Hg @ 0°C
bars	29.530	feet of H2O @ 40°F
bars	76	inches of Hg @ 0°C
bars	14.5038	psi
barrels of oil(US)	42.0	gallons (US)
barrels of oil(US)	5.61458	feet^3
barrels of oil(US)	163.6592	liters
board feet	144	inch^3
board feet	0.08333	foot^3
board feet	2359.74	cm^3
british thermal unit (BTU)	777.649	foot pound-force
british thermal unit (BTU)	1,055.056	joule
british thermal unit (BTU)	25020.1	foot poundals
british thermal unit (BTU)	251.996	calorie,g
british thermal unit (BTU)	0.2520	kg-calorie
british thermal unit (BTU)	0.000292875	kw-hours
british thermal unit (BTU)	0.00001	therms
british thermal unit (BTU)	0.000393	hp-hours
british thermal unit (BTU)	1054.35	watt-seconds
british thermal unit (BTU)	10.544 x 10^3	ergs
british thermal unit (BTU)	0.999331	BTU (IST)

UNIT x	FACTOR =	UNIT
BTU/min	0.01758	kilowatts
BTU/min	0.02358	horsepower
byte	8.000001	bits
calorie, g	0.00397	british thermal unit
calorie, g	0.00116	watt-hour
calorie, g	4184.00 x 10^3	ergs
calorie, g	3.08596	foot pound-force
calorie, g	4.184	joules
calorie, g	0.000001162	kilowatt-hour
calorie, g	42664.9	gram-force cm
calorie, g/hr	0.00397	btu/hr
calorie, g/hr	0.0697	watts
candle/cm^2	12.566	candle/inch^2
candle/cm^2	10000.0	candle/meter^2
candle/inch^2	144.0	candle/foot^2
candle power	12.566	lumens
carats	3.0865	grains
carats	200.0	milligrams
celsius	1.8 C°+ 32	fahrenheit
celsius	273.16 + C°	kelvin
centimeter	0.39370	inch
centimeter	0.03281	foot
centimeter	0.01	meter
centimeter	10	millimeter
cm grams -force	0.0000723	foot pound-force
cm of Hg	0.1934	pound/inch^2
cm/sec	0.0328	feet/sec
cm/sec	1.9685	feet/min
cm/sec	0.0006	km/min
cm/sec	0.0194	knots
cm/sec	0.000373	miles/hour
cm/sec/sec	0.0328	feet/sec/sec
cm/sec/sec	0.01	meters/sec/sec
chains	66.0	feet
chains	20.117	meter
circles	360	degrees
circles	400	grades
circles	6.2832	radians
circles	12.0	signs
circular inches	0.7854	inch^2
centimeter^2	0.15500	inch^2
centimeter^2	0.00108	foot^2
centimeter^2	127.324	circular mm
centimeter^2	100.0	mm^2
centimeter^2	0.0001	meter^2
centimeter^2	155000.0	mils^3
centimeter^3	0.06102	inch^3
centimeter^3	0.00042	board feet
centimeter^3	0.000035315	feet^3
centimeter^3	0.000001	meters^3
centimeter^3	0.27051	drams
centimeter^3	0.06102	gallons (US)
centimeter^3	0.001	liter
centimeter^3	0.03381	ounces
centimeter^3	0.00211	pints
centimeter^3	0.00106	quarts
centipose	0.001	pascal-second
centistokes	0.000001	meter^2/second
coulombs	1.0	amp-hours
coulombs	0.000010364	faradays
coulombs	2997900000	statcoulombs
days	24.0	hours

Table 2-8: Unit Conversion Tables			(continued)
UNIT x	FACTOR =	UNIT	
days	1440.	minutes	
days	0.00273	years	
days	86400	seconds	
decimeter	10.	centimeters	
decimeter	3.937	inch	
decimeter	0.32808	feet	
decimeter^3	61.02	inch^3	
degrees	60.0	minutes	
degrees	3600.0	seconds	
degrees	0.01111	quadrants	
degrees	0.01745	radians	
degrees	1.111	grades	
dynes	0.00001	newtons	
dynes/cm^2	0.000001	bars	
electron volts	1.6021 x 10^{-12}	ergs	
ergs	9.4845 x 10^{-11}	british thermal unit	
ergs	1.0 x 10^{-7}	joules	
ergs	7.376 x 10^{-8}	foot pound-force	
ergs	2.3885 x 10^{-8}	grams-calorie	
ergs	0.278 x 10^{-10}	watt-hours	
ergs	1.0	dynes-cm	
ergs/sec	1.341 x 10^{-10}	horsepower	
fahrenheit	(F°-32)/1.8	celsius	
fahrenheit	0.55556	celsius	
fahrenheit	459.72 + F°	rankin	
farads	100000	statamperes	
farads	1.00049	statfarads	
farads	100000	microfarads	
fathoms	6.0	feet	
fathoms	1.828	meters	
fathoms	2	yards	
feet of H_2O	2.98898	kilopascal	
feet of H_2O	0.4336	pound force/inch^2	
feet/second	0.508	cm/second	
feet/second	0.00508	meter/second	
feet^2/second	0.000001	meter^2/second	
foot	304.80	millimeters	
foot	30.480	centimeter	
foot	0.30480	meter	
foot	0.015151	chains	
foot	0.000189	miles	
foot	0..166667	fathoms	
foot - poundals	3.9968 x 10^{-5}	british thermal unit	
foot - poundals	0.010072	cal, gram	
foot - poundals	0.03108	foot pound-force	
foot - poundals	0.042133	joule	
foot - pound force	1.35582	joule	
foot - pound force	0.00128	british thermal unit	
foot/hour (linear)	0.508	Cm/minute	
foot/min	.00508	meter/sec	
foot/sec	.3048	meter/sec	
foot pound force	1.35582	newton meter	
foot^2	92,903.04	millimeter^2	
foot^2	929.0304	centimeter^2	
foot^2	0.09290	meter^2	
foot^2	0.11111	yard^2	
foot^2	0.00002	acre	
foot^2	3.5873 x 10^{-8}	mile^2	
foot^3	0.00781	cords of wood	
foot^3	12.0	board feet	
foot^3	1728.0	inches^3	

UNIT x	FACTOR =	UNIT
foot^3	28316.8	centimeter^3
foot^3	0.02832	meter^3
foot^3	28.32	liter (liq.)
foot^3	59.842	pint (liq.)
foot^3	29.922	quart (liq.)
foot^3	7.48052	gallon (liq.)
foot^3	0.03704	yard
foot^3/hour	.0283168	meter^3/hour
foot^3/hour	0.0167	feet^3/minute
foot^3/hour	7.4805	gallons/hour
foot^3/minute	0.283168	meter^3/minute
foot^3/minute	471.95	centimeter^3/second
foot^3/second	448.8306	gallon/minute
foot^3/second	0.02832	meter^3/second
foot^3/second	28.31658	liter/second
foot^3/second	120.0	foot^3/hour
foot^3/pound	120.0	centimeter^3/gram
foot^3 H_2O	28.31413	Kilogram
foot^3 H_2O	62.42197	pound
foot^3 H_2O	28.31413	Kilogram
foot^3 H_2O	62.42197	pound
furlongs	660.0	feet
furlongs	20116.8	centimeters
furlongs	201.17	meters
furlongs	7920	inches
furlongs	220.0	yards
gallon (US liq.)	8.0	pint
gallon (US liq.)	4.0	quart
gallon (US liq.)	3.0689 x 10^{-6}	acre feet
gallon (US liq.)	0.00379	meter^3
gallon (US liq.)	3.785	liter
gallon (US liq.)	0.13368	foot^3
gallon (US liq.)	8.33	pounds
gallon H_2O	3.78625	kilogram
gallon H_2O	3.78625	kilogram
gallon H_2O	8.34725	pound
gallon/minute	0.00006	meter^3/second
gallon/minute	0.06309	liter/second
gallon/minute	0.00144	million gallons/day
gallon/minute (gpm)	0.00223	foot^3/second (cfm)
gallons/inch/mile/day	0.03259	liter/mm/km/day
gallons/inch/mile/day	0.03259	liter/mm/km/day
gausses	10000.0	gamma
gausses	6.4516	lines/inch^2
gausses	6.452 x 10^{-8}	webers/inch^2
gram/centimeter^3	1,00.00	kilogram/meter^3
grades	0.0025	circles
grades	0.0025	circumfrencees
grades	0.9	degrees
grades	54	minutes
grades	0.0025	revolutions
grades	3240	seconds
grains	0.32399	carats
grains	0.01667	drams (troy)
grains	0.03657	drams (avdp)
grains	64.7989	milligrams
grains	0.00017	pounds (troy)
grains	0.00014	pounds (avdp)
grams	5.0	carats
grams	0.2572	drams (troy)
grams	0.5644	drams (avdp)
grams	15.432	grains

Table 2-8: Unit Conversion Tables (continued)

UNIT x	FACTOR =	UNIT	UNIT x	FACTOR =	UNIT
			inch of Hg	0.03342	Atmosphere
grams	0.001	kilograms	inch of Hg	0.03386	bars
grams	1000.0	milligrams	inch of Hg	34.532	grams force/cm^2
grams	0.03215	ounce (troy)	inch pound force	0.11299	newton meter
grams	0.03527	ounce (avdp)	inch^2	645.10	millimeter^2
grams	0.00220	pound	inch^2	6.4516	centimeter^2
grams force/cm^2	98.0665	pascal	inch^3	16.387	millimeter^3
grams force/cm^2	0.00034	pound force/inch^2	inch^3	16.39	centimeter^3
hectare	10,000.00	meter^2	inch^3	0.01639	decimeter^3
hectare	2.47105	acre	joule	0.73756	foot* pound force
henries	1000.0	millihenries	joule	0.00095	british thermal unit
henries	1.113×10^{-12}	stathenhenries	kilogram	35.274	ounce
horsepower (mech)	2542.47	btu/hr	kilogram	2.20462	pound
horsepower (mech)	0.746	kilowatts	kilogram	0.001	metric ton (tonne)
horsepower (mech)	64160.0	calories, gram/hr	kilogram	1000.0	grams
horsepower (mech)	7.457×10^9	ergs/second	kilogram force	9.80681	newton
horsepower (mech)	1980000.0	foot pound-force/hour	kilogram force/cm^2	98.0665	kilopascal
horsepower (mech)	0.076	horsepower (boiler)	kilogram force/cm^2	14.22335	pound force/inch^2
horsepower (mech)	0.9996	horsepower (electric)	kilogram force/meter^2	9.80665	pascal
horsepower (mech)	1.0139	horsepower (metric)	kilogram/meter^3	0.06243	pound/foot^3
horsepower (mech)	745.7	joules/sec	kilogram/meter^3	1.68554	pound/yard^3
horsepower (mech)	0.212	tons of refrig.	kilogram/meter^3	0.00835	pound/gallon
horsepower (mech)	745.7	watts	kilogram/meter^3	0.00084	ton/yard^3
horsepower (boiler)	33445.7	btu/hr	kilogram/meter^3	0.001	metric ton/meter^3
horsepower (boiler)	140671.6	calories, gram/min	kilogram/metre	0.67197	pound/foot
horsepower (boiler)	9.8097×10^{10}	ergs/second	kilometer	0.62137	mile
horsepower (boiler)	13.155	horsepower (mech)	kilometer	0.00000000000010	light years
horsepower (boiler)	13.1497	horsepower (electric)	kilonewton	100000000.0	dynes
horsepower (boiler)	13.337	horsepower (metric)	kilopascal	1,000.	pascal
horsepower (boiler)	13.149	horsepower (metric)	kilopascal	0.01	bar
horsepower (boiler)	9809.5	joules/sec	kilopascal	0.14504	pound force/inch^2
horsepower (boiler)	9.8095	kilowatts	kilopascal	0.33456	feet of H_2O
horsepower (electric)	2547.16	btu/hr	kilopascal	0.29637	inches of Hg
horsepower (electric)	178.298	calories, gram/sec	kilopascal	0.001	megapascal
horsepower (electric)	7.46×10^9	ergs/second	kilopascal	0.00987	atmosphere
horsepower (electric)	1.0004	horsepower (mech)	kilowatts	3414.4	btu/hr
horsepower (electric)	0.0745	horsepower (boiler)	kilowatts	2655000	foot-pound force/hr
horsepower (electric)	1.01428	horsepower (metric)	kilowatts	1.34	horsepower (elec&mech)
horsepower (electric)	0.99994	horsepower (metric)	kilowatts	0.1019	horsepower (boiler)
horsepower (electric)	746	joules/sec	kilowatts	1.3596	horsepower (metric)
horsepower (electric)	0.746	kilowatts	knots	0.868976	kilometers/hour
horsepower (metric)	2511.3	btu/hr	knots	1.688	feet/second
horsepower (metric)	632800	calories, gram/hr	knots	1.1508	miles/hour
horsepower (metric)	7.355×10^9	ergs/second	leagues	18240.0	feet
horsepower (metric)	0.9863	horsepower (mech)	liter	0.03531	foot^3
horsepower (metric)	0.07498	horsepower (boiler)	liter	0.001	meter^3
horsepower (metric)	0.9859	horsepower (electric)	liter	1,000.	milliliter^3
horsepower (metric)	0.98587	horsepower (water)	liter	2.113	pint
horsepower (metric)	735.499	watts	liter	1.057	quart
horsepower (metric)	0.7355	kilowatts	liter	0.2642	gallon
horsepower (water)	0.076	horsepower (boiler)	liter	1.	decimeter^3
horsepower (water)	1.00006	horsepower (electric)	liter/minute	0.0353	foot^3/minute
horsepower (water)	1.00046	horsepower (mech)	liter/minute	.26417	gallon/minute
horsepower (water)	1.0143	horsepower (metric)	liter/second	0.035315	foot^3/second
horsepower (water)	0.746043	kilowatts	liter/second	15.851	gallon/minute
inch	25.4	millimeters	liter/mm/km/day	10.800	gallons/in/mile/day
inch	2.54	centimeter	liter/mm/km/day	10.800	gallons/in/mile/day
inch	0.08333	feet	liter/second	0.001	meter^3/second
inch	0.0278	yards	lumens	0.0015	watts
inch	1000	mils	lumens/foot^2	10.7639	lumens/meter^2
inch of Hg	3.37416	kilopascal	lux	0.0929	foot-candles
inch of Hg	0.49116	pound force/inch^2	megapascal	1,000.	kilopascal

Table 2-8: Unit Conversion Tables (continued)

UNIT x	FACTOR =	UNIT
megapascal	145.0377	pound force/inch^2
megapascal	9.86923	atmosphere
megapascal	10.	bar
meter	3.28084	foot
meter	1.09361	yard
meter	0.00062	mile
meter	0.1988	rods
meter^2	10.76391	foot^2
meter^2	1.19599	yard^2
meter^2	0.00025	acre
meter^2	0.0001	hectare
Meter^3	0.00081	acre feet
meter^3	35.315	foot^3
meter^3	264.17	gallon
meter^3	1.308	yard^3
meter^3	1,000.	liter
Meter^3	0.00081	acre feet
meter^3/second	35.315	foot^3/second
meter^3/second	15,850.3	gallon/minute
meter^3/second	1,00.	liter/second
meter^3/second	22.82447	million gallons/day
meters/second^2	3.280840	feet/second^2
metric ton (tonne)	2,204.6	pound
metric ton (tonne)	1.1023	ton (US)
metric ton (tonne)	1,000.	kilogram
metric ton/meter^3	0.84277	ton/yard^3
micrometers	10000.0	angstroms
mile (statute)	1,609.344	meter
mile (statute)	1.60934	kilometer
mile (statute)	5,280.	foot
mile (statute)	1,760.	Yard
mile^2	640.0	acre
miles/hour	.447	meter/sec
miles/hour	88.0	feet/minute
miles/hour	1.609344	meter/sec
miles/hour	1.6093	kilometers/hour
miles/hour	1.852	knots
miles/hour	1.6093	kilometers/hour
miles/hour	1.852	knots
millimeter^2	0.00155	inch^2
millimeter^2	0.00155	foot^2
millimeter^3	0.00006	inch^3
milliliters	1.00	cm^3
milliliters	0.06102	inch^3
milliliters	0.001	liters
milliliters	0.0338	ounces (fld)
milliliters	0.00211	pints (fld)
millimeters	0.03937	inches
millimeters	0.00328	foot
millimeters	0.01	centimeters
millimeters	0.001	meters
millimeters	39.37	mils
millimeters	1000.0	microns
millimeters	1000.0	micrometers
million gallons/day	694.44	gallon/minute
million gallons/day	0.04381	meter^3/second
newton	0.22481	pound force
newton	0.10197	kilogram force
newton meter	0.73756	foot pound force
newton meter	8.85073	inch pound force
newton/meter^2	0.00015	pound force/inch^2
newton/meter^2	1.0	pascal

UNIT x	FACTOR =	UNIT
ohms	100000.0	micro ohms
ounce	28.3495	gram
ounce	437.5	grain
ounce	0.02835	pound
ounce	0.2835	kilogram
ounce-force/inch^2	4.3942	gram-force/cm^2
ounce-force/inch^2	0.0625	pound force/inch^2
parts/million	0.05842	grains/gallon (US)
parts/million	1.0	grams/ton (metric)
parts/million	0.0001	percent
pascal	1.	newton/meter^2
pascal	0.00750062	torr
pint	0.4732	liter
pint	0.01671	feet^3
pint	28.875	inch^3
poise	0.100	pascal-second
pound	7000	grains
pound	453.5924	gram
pound	0.45359	kilogram
pound	0.00045	metric ton (tonne)
pound	0.0005	ton
pound	16.	ounce
pound	0.0005	ton
pound (apoth or troy)	0.82286	pound (avdp)
pound force	4.44822	newton
pound force/inch^2	6,894.757	pascal
pound force/inch^2	6.89476	kilopascal
pound force/inch^2	0.00689	megapascal
pound force/inch^2	0.07031	kilogram force/cm^2
pound force/inch^2	6,894.757	newton/meter^2
pound force/inch^2	0.06895	bar
pound force/inch^2	0.06805	atmosphere
pound force/inch^2	2.307	feet of H_2O
pound force/inch^2	2.036	inch of Hg
pound of H_2O	0.01602	feet^3
pound/foot	1.48816	kilogram/metre
pound/foot^3	16.01846	kilogram/meter^3
pound/foot^3	0.0135	ton/yard^3
pound/gallon	119.82640	kilogram/meter^3
pound/yard^3	0.59328	kilogram/meter^3
quart	0.9463	liter
quart	2.0	pint
radians	57.2957	degrees
rods	502.92	centimeter
tablespoon	180	drops of liquid
teaspoon	60	drops of liquid
ton	0.90719	metric ton (tonne)
ton	907.18	kilogram
ton/yard^3	1,186.553	kilogram/meter^3
ton/yard^3	1.18655	metric ton/meter^3
ton/yard^3	74.07407	pound/foot^3
torr (Torricellis)	1.0	mm of Hg
watts	0.000948	btu/sec
watts	680	lumens
watts	0.00134	horsepower
yard	0.91440	meter
yard	91.44	centimeter
yard	0.0005682	miles
yard^2	0.83613	meter^2
yard^2	9.0	foot^2
yard^2	0.00021	acre
yard^3	0.7646	meter
yard^3	27.0	foot^3

SUMMARY:

This chapter has brought together data that is not included generally in reference books. There are many reference books extant in the HVAC field. One that must be referenced is the Hydraulic Institute *Engineering Data Book.* This book has detailed information on pipe and fitting dimensions, friction losses in steel pipe and pipe fittings as well as the viscosity of liquids. It should be in the library of any designer of HVAC water systems.

Chapter 3

HVAC PIPING

HVAC PUMPS AND SYSTEMS
DESIGN MANUAL

PATTERSON PUMP COMPANY

INTRODUCTION:

A basic understanding of pipe friction and design is important for the applier of centrifugal pumps in the HVAC industry. A poor computation of system friction will have a lasting and negative effect on pump selection and operation. An important subject facing water system designers is to have at hand procedures and sources for determining pipe friction head for these systems.

By far, the most important document for steel pipe design is the Engineering Data Book of the Hydraulic Institute. This book should be owned by every serious pipe designer and applier of HVAC pumps. Table IIIB-4 of that book provides a comprehensive source for pipe sizes as well as velocities and frictbn losses for a broad range of flows.

Maximum Velocities of Actual Piping:

In the HVAC field, there are “rules of thumb” as to the allowable velocity of water in pipe. There are limits on velocity such as 4 fps for small pipe and 10 fps for large steel pipe while the plastic pipe industry has a limitation of 5 fps. Noise, erosion, and hydraulic shock are reasons often given for these limits on velocity of water in pipe.

The selection of pipe size is a complicated process that must take into consideration first cost and operating cost. Seldom is pipe velocity a reason for selecting a certain size of pipe. Friction loss is the major factor, as it affects the pump and motor size for a particular installation. The other factor is the size of the water system. Small compact systems do not have the same pipe sizing procedure as the large systems with thousands of feet of pipe. Large piping can have average velocities as high as 15 fps.

Steel Pipe Friction Formulas:

The calculation of pipe friction is normally accomplished by two different formulas. These are the Darcy Weisbach and the Hazen-Williams formulas:

<u>Darcy Weisbach</u>:

$$h_f = f \cdot \frac{L}{d} \cdot \frac{v^2}{2g} \qquad (3.1)$$

where:

h_f = Friction Loss - Ft. of Liquid
L = Pipe Length – Feet
d = Average Inside dia. - Ft.
f = Friction Factor

The friction factor, f, is usually derived from the Colebrook Equation:

$$\frac{1}{f^{0.5}} = -2 \log_{10} \left(\frac{\varepsilon}{3.7D} + \frac{2.51}{R_E \cdot f^{0.5}} \right) \qquad (3.2)$$

(Note: Parentheses are around all of second and third terms)

where:
R_E = Reynolds Number
ε = Absolute Roughness Parameter
(typically 0.00015 for steel pipe)

The piping tables in the Engineering Data Book of the Hydraulic Institute are based upon these formulas.

Another formula that can be used is the Hazen Williams. It is the basis of much design in the Civil Engineering industry. It will produce about the same results as the Williams Hazen formula for cold water.

<u>Hazen & Williams</u> :

$$h_f = 0.002083 \cdot L \cdot \left(\frac{100}{C}\right)^{1.85} \cdot \frac{GPM^{1.85}}{D^{4.8655}} \qquad (3.5)$$

*Parentheses across complete term

Where:

h_f is the friction loss in ft.

L is the length of pipe in ft.

C is a design factor determined for various types of pipe.

D = inside diameter of pipe, inches

BEFORE USING ANY FORMULA OR DATA ON PIPE FRICTION, BE SURE THAT THE PIPE SIZE UNDER CONSIDERATION HAS THE SAME INSIDE DIAMETER AS THE PIPE IN COMPUTER SOFWARE OR IN THE TABLES!

Reynolds Number:

Reynolds Number is used by pipe designers where the calculation of pipe friction must be made under varying velocities and viscosities. It is a dimensionless number that is not used extensively in the HVAC industry. It should be used with high temperature water with its changes in specific gravity and viscosity.

$$\textbf{Reynolds Number, } R = \frac{v \cdot d}{\nu} \qquad \textbf{(3.3)}$$

where:

v = Velocity, ft per sec
d = Pipe diameter in ft
ν = Kinematic viscosity in ft^2/sec.

Plastic Pipe:

Plastic pipe is used sparingly in the HVAC field. It is found around cooling towers where the water may be laden with oxygen, and steel pipe would be susceptible to rusting. The types of plastic pipe used are PVC (polyvinyl chloride), Schedules 40 and 80, CPVC (Chlorinated polyvinyl chloride), Schedules 40 and 80, and High Density Polyethylene (HDPE).

Plastic pipe itself has lower friction than steel pipe, but the plastic fittings generally have a higher friction loss that those for steel pipe. The plastic pipe industry uses the Hazen-Williams formula for computing pipe friction with a "C" factor of 150.

PIPE FITTING LOSSES:

Pipe fitting losses comprise a sizeable percentage of the total friction loss of an HVAC water system. There are so many more fittings than in domestic water system of the same capacity. Hydraulic Institute's Engineering Data Book provides the best source for friction losses for steel and cast iron fittings. Also, this book demonstrates the variation that can occur in these fittings due to actual manufacturing tolerances.

Basically, fitting losses vary with the velocity head of the water flowing in the pipe, $v^2/2g$. Hydraulic Institute has produced a "k" factor for many of the popular pipe fittings that results in a fitting loss, h_f that is computed by the following equation:

$$h_f = k \cdot \frac{v^2}{2g} \qquad (3.4)$$

There are charts on “k” factors for many popular pipe fittings that are included in the Hydraulic Institute’s Engineering Data Book.

ASHRAE Laboratory Testing of Fittings:

The American Society of Heating, Refrigeration, Air-Conditioning Engineers (ASHRAE) has completed a series of tests on pipe fittings to update the data on friction losses. There were five different test projects.

Project No.	Fitting Size	Material
RP-968	2 and 4”	Iron and Steel
1034-RP	12, 16, 20, & 24”	Steel
1035-RP	4”	Steel*
1116-RP	6, 8, and 10”	Steel
1193-RP	6, 8, and 10”	PVC, Sch. 80

* The purpose of this test was to determine the effect of closely connected fittings. The tests consisted of connecting two welded steel elbows in four different configurations which were 1) in plane, “Z” connection, 2) in plane, “U” connection, 3) out of plane, torsional connection, and 4) out of plane, swing connection. (See Chapter 9 for the effect of packaged systems on pipe fitting losses.)

Complete reports of these tests can be secured from ASHRAE headquarters in Atlanta.

RP-968: The actual k-factors when compared with published data are shown in Table 3-1. New data for reducing elbows is shown in both the reducing and expansion modes.

1034-RP: Table 3-1 describes variations between the test data and past values for these fittings. New data is provided for reducing fittings in both the reducing and expansion modes.

1035-RP: These tests proved false the widespread belief that there is an increase in the friction loss of fittings when they are closely connected. The development of factory-assembled systems was resisted on the theory that closely assembling pipe fittings increases the friction loss of the pumping system. These tests proved this theory wrong and that there is an energy savings for factory assembled pumping systems.

1193RP: These tests proved that the losses through most plastic pipe fittings are much greater than comparable steel fittings. This is due to the sharper radii and shoulders found in some plastic fittings. The data in these tests should be reviewed by designers of plastic piping for HVAC water systems.

Table 3-1 provides a comparison in “k” values for popular steel pipe fittings. This was compiled for ASHRAE by Utah State University; they conducted several of the research projects listed above. A significant fact uncovered by Utah Sate University was the variation in the “k” factor with velocity of the water in the pipe fitting. This demonstrates that practical experience is an appreciable factor in the design of HVAC piping.

This is a synopsis of a very detailed subject. The HVAC water system designers must continuously improve their techniques and sources of information to insure that the pump head for those systems is close to that actually required to achieve design flow. HVAC piping has often been the recipient of questionable design and excessive pump head.

TABLE 3-1: DATA SUMMARY OF TEST DATA FOR ELLS, REDUCERS AND EXPANSIONS

	Past	USU 4fps	USU 8 fps	USU 12 fps
2” S.R. Ell (R/D=1) Thread	0.60 to 1.0 (1.0)*	0.60	0.68	0.736
4” S.R. Ell (R/D=1) Weld	0.30 to 0.34	0.37	0.34	0.33
1” L.R. Ell (R/D=1.5) Weld	0.39 to 1.0	-----	-----	-----
2” L.R. Ell (R/D=1.5) Weld	0.50 to 0.7	-----	-----	-----
4” L.R. Ell (R/D=1.5) Weld	0.22 to 0.33 (.22)*	0.26	0.24	0.23
6” L.R. Ell (R/D=1.5) Weld	0.25	-----	-----	-----
8” L.R. Ell (R/D=1.5) Weld	0.20 to 0.26	-----	-----	-----
10” L.R. Ell (R/D=1.5) Weld	0.17	-----	-----	-----
12” L.R. Ell (R/D=1.5) Weld	.16	0.17	0.17	0.17
16” L.R. Ell (R/D=1.5) Weld	.12	0.12	0.2	0.11
20” L.R. Ell (R/D=1.5) Weld	.09	0.12	0.10	0.10
24” L.R. Ell (R/D=1.5) Weld	.07	0.098	0.098	0.098
Reducer (2”x1.5”) Thread	-----	0.53	0.28	0.20
Reducer (4”x3”) Weld	0.22	0.23	0.14	0.10
Reducer (12”x10”) Weld	-----	0.14	0.14	0.14
Reducer (16”x12”) Weld	-----	0.17	0.16	0.17
Reducer (20”x16”) Weld	-----	0.16	0.13	0.13
Reducer (24”x20”) Weld	-----	0.053	0.053	0.055
Expansion (1.5”x2”) Thread	-----	0.16	0.13	0.02
Expansion (3”x4”) Weld	-----	0.11	0.11	0.11
Expansion (10”x12”) Weld	-----	0.11	0.11	0.11
Expansion (12”x16”) Weld	-----	0.073	0.076	0.073
Expansion (16”x20”) Weld	-----	0.024	0.021	0.022
Expansion (20”x24”) Weld	-----	0.020	0.023	0.020

Past (published data by Freeman, Crane, Hydraulics Institute)
USU (Utah State University Data)
( )* Data published in ASHRAE Fundamentals
S.R. – short radius or regular Ell
L.R. – long radius Ell

Chapter 4

PHYSICAL DESIGN OF HVAC PUMPS, MOTORS, AND DRIVES

HVAC PUMPS AND SYTEMS DESIGN MANUAL

PATTERSON PUMP COMPANY

INTRODUCTION:

The use of water pumps in the HVAC industry consists of pumping chilled, hot, and cooling tower water. Cooling tower pumps are also known as “condenser” pumps. When temperatures are encountered below 32°F, various percentages of glycol are used to lower the freezing temperature of the solution.

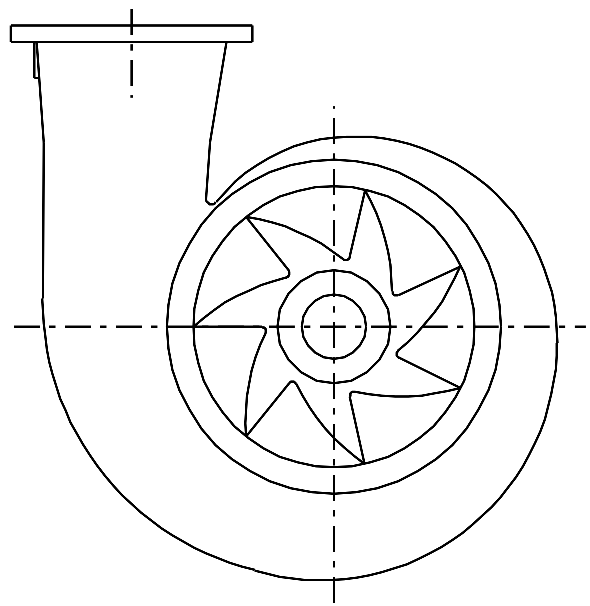
Chilled water pumps usually operate at temperatures of from 26 to 60°F while hot water pumps operate from 80 to 250°F. At water temperatures above this, they are known as high temperature hot water pumps that may operate as high as 450°F. Condenser pumps operate at 45 to 95°F.

HVAC pumps are all centrifugal type of either volute or diffuser design, Figure 4-1. The flow through the volute pump is perpendicular to the pump shaft, and it is parallel to the pump shaft in diffuser pumps. Diffuser pumps are also called “axial flow” pumps. Self priming pumps are seldom used in this field; they will be found when lifting chilled water that is stored underground. Positive displacement pumps are only used for pumping small quantities of chemicals.

TYPES OF PUMPS:

Most of the pumps used in this industry are **volute** type, and they are available in 1) **single suction** and 2) **double suction** designs, Figure 4-2. Single suction pumps are also called “**end suction**” while double suction pumps are better known as “**split case**” pumps. Single suction pumps have several basic configurations. They can be **close coupled,** Figure 4-3, and they can be **frame mounted**, Figure 4-4, or **in-line**, Figure 4-5. High pressure boiler feed pumps may require multi-stage, single suction pumps, Figure 4-6. Unfortunately, they are often called multi-stage, split case pumps without mentioning that they are actually single suction.

Split case (double suction) pumps are generally of one configuration, Figure 4-7; they can be mounted horizontally or vertically. These split case pumps are axially split along the pump shaft to accommodate inspection and cleaning without disturbing the rotating assembly. There are vertically split pumps in the HVAC



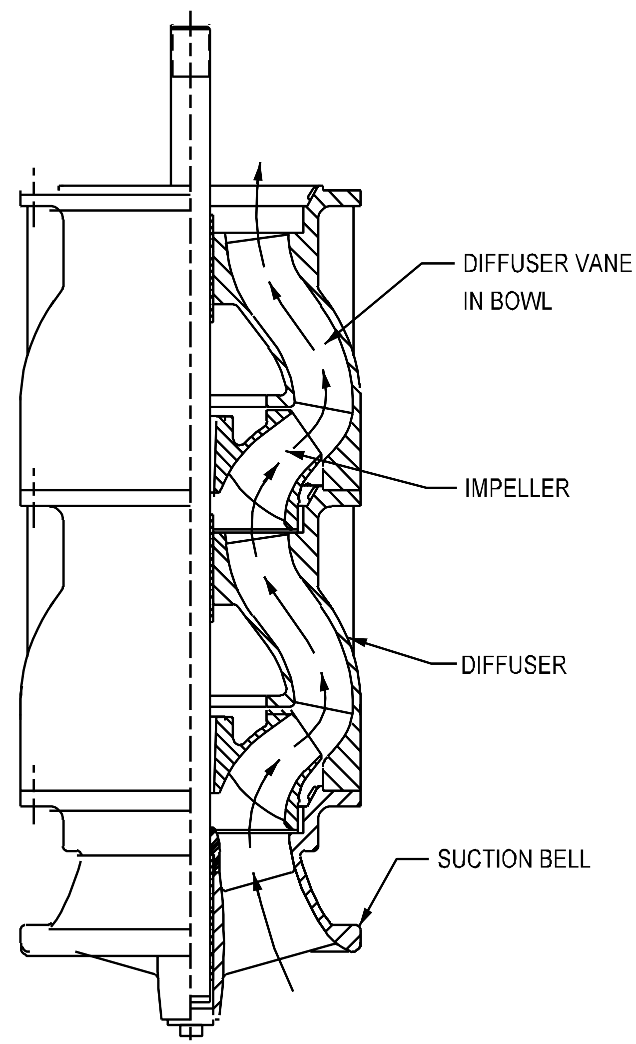
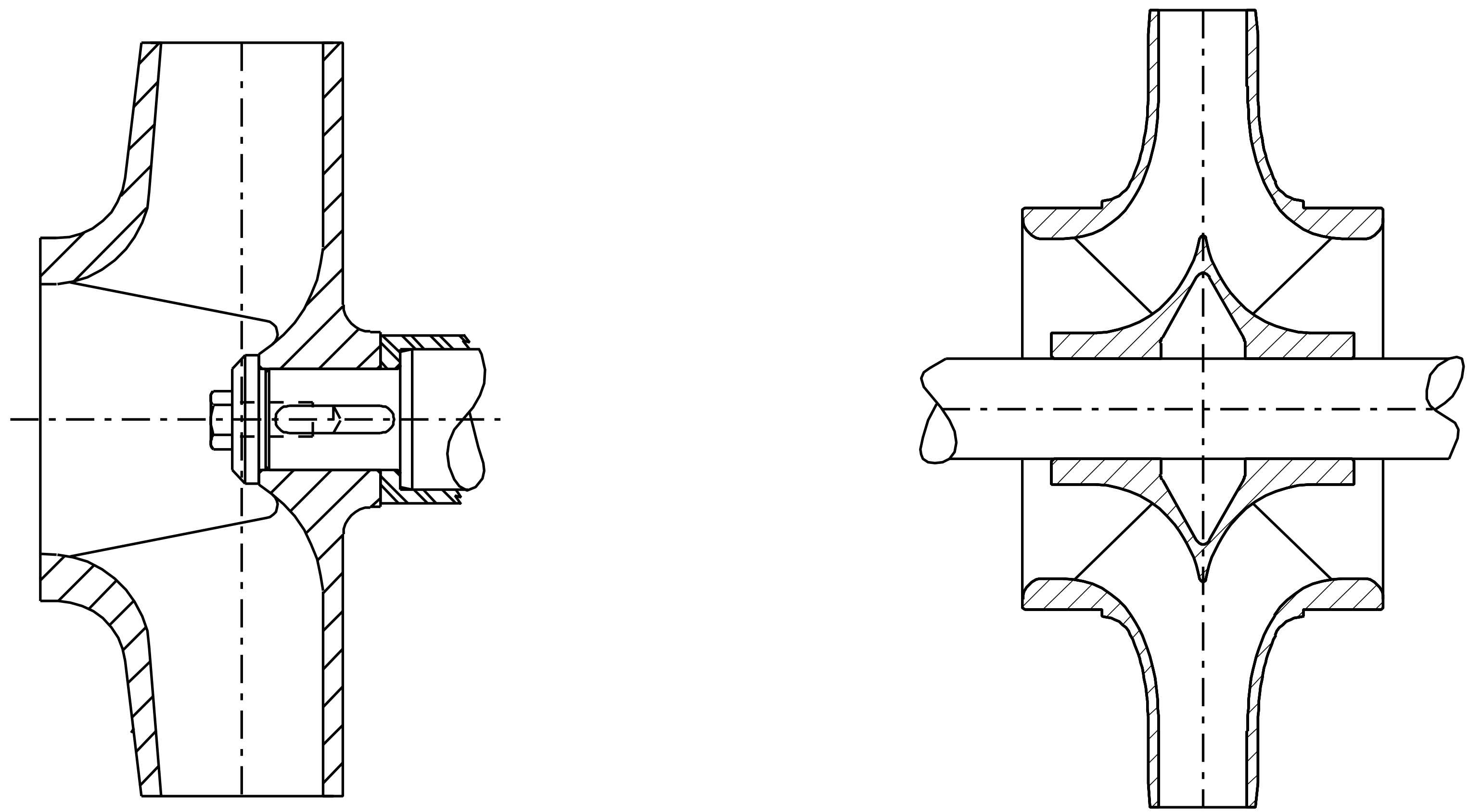


FIGURE 4-1 Basic centrifugal pump configurations



(a) SINGLE SUCTION (b) DOUBLE SUCTION

FIGURE 4-2 Basic configurations of volute impellers

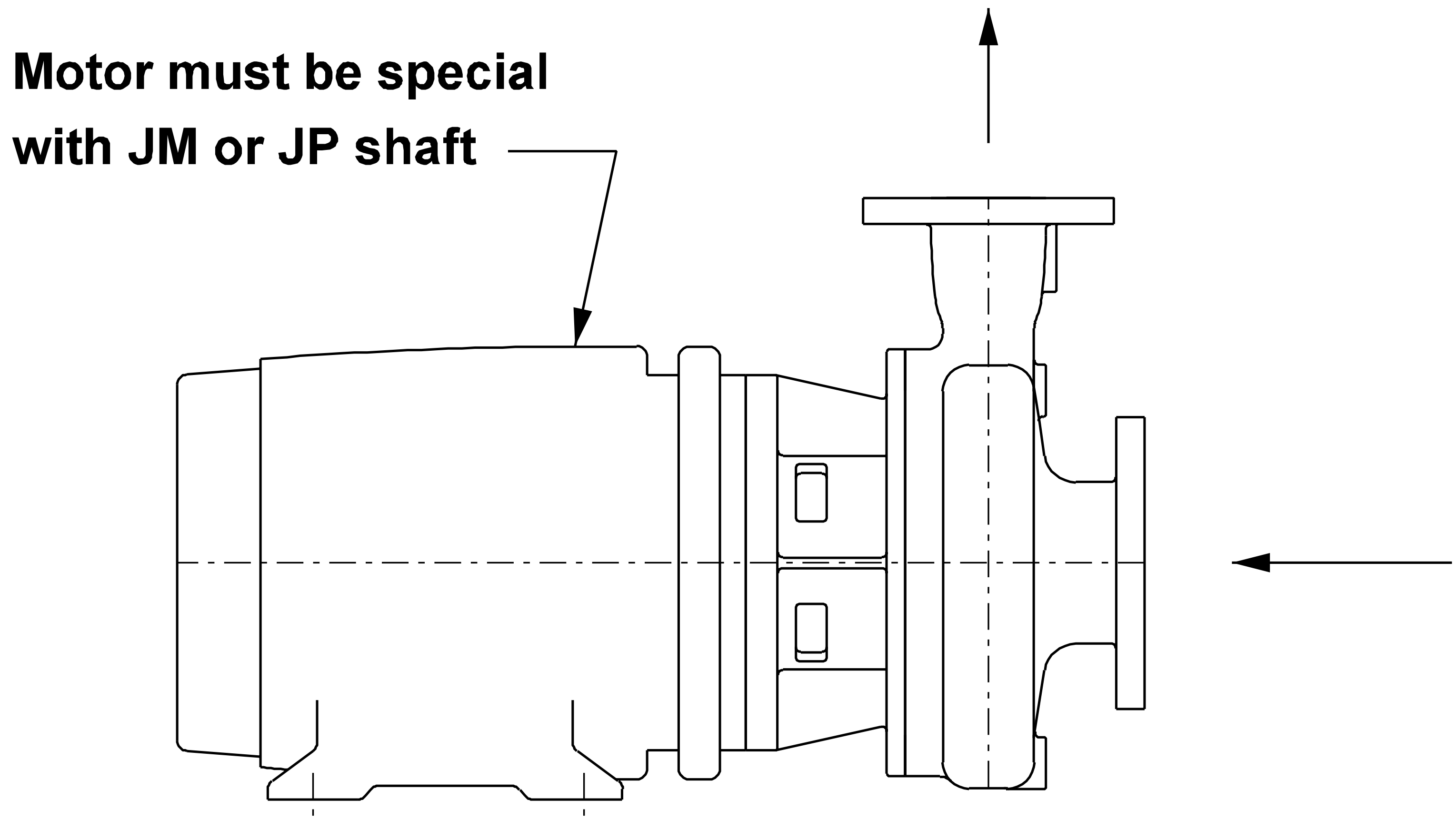


FIGURE 4-3 Close-coupled single suction pump.

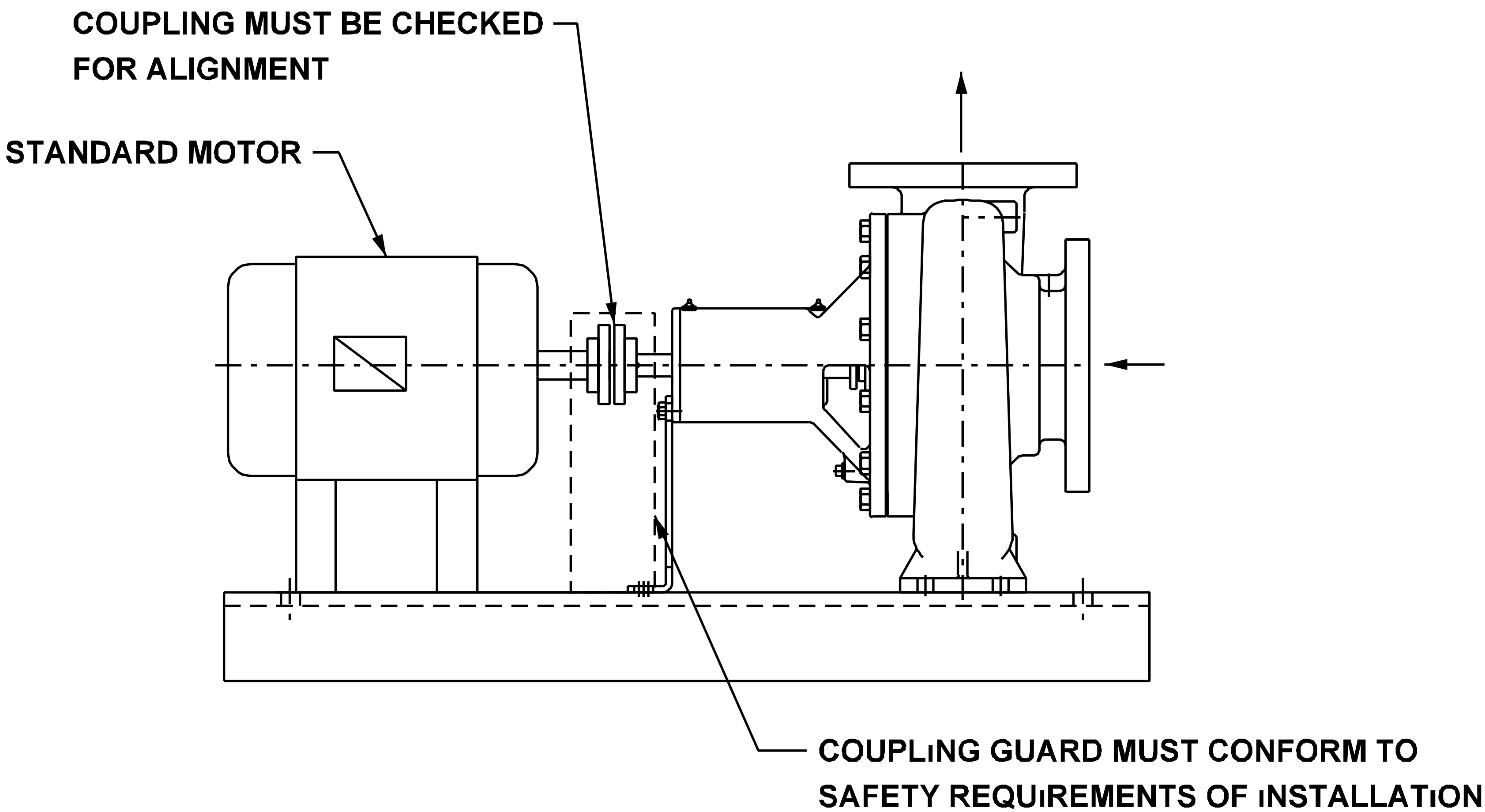


FIGURE 4-4 Flexible coupled single suction pump horizonally mounted.

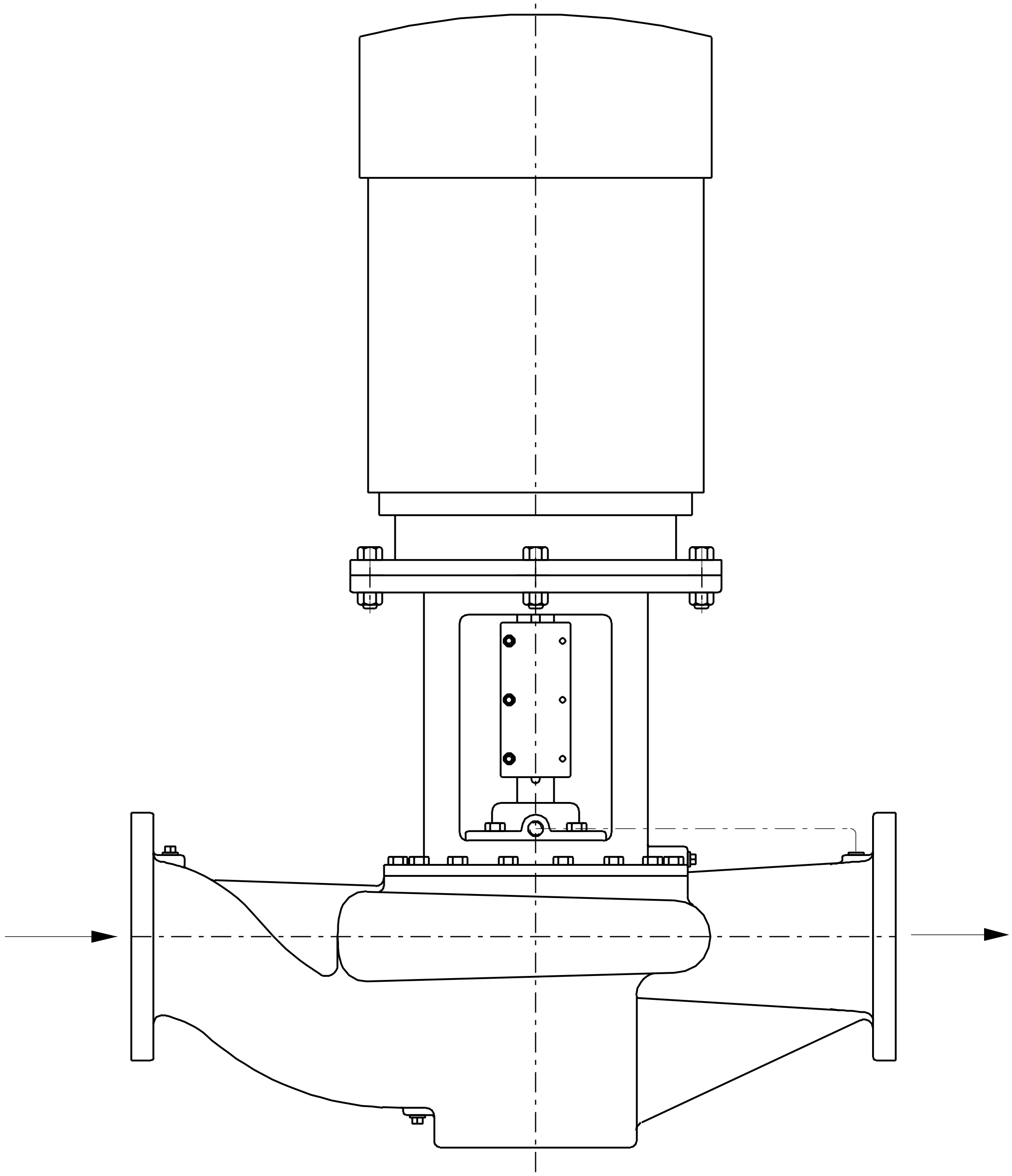
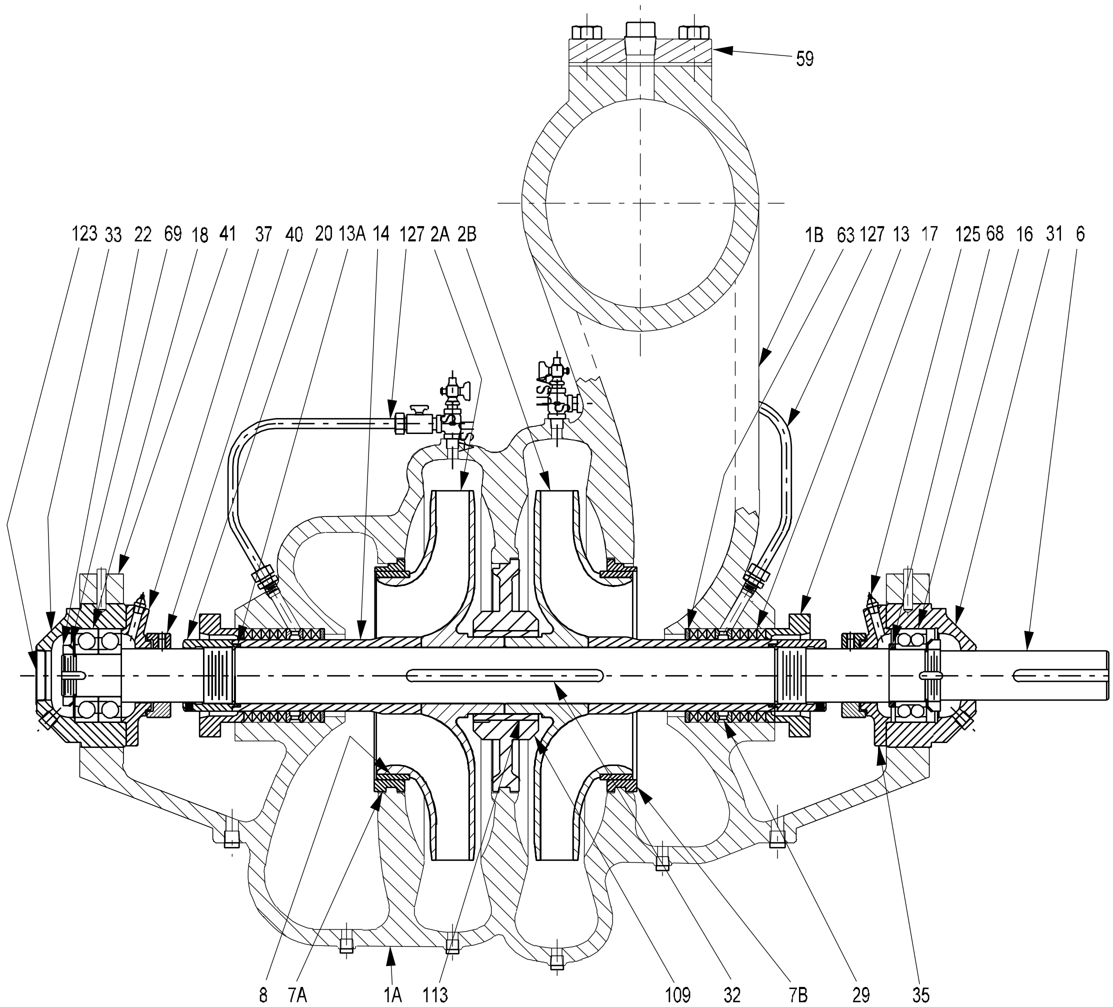


FIGURE 4-5 Vertically mounted, separately coupled pump.



1A	LOWER HALF CASING	18	OUTBOARD BEARING	46	COUPLING KEY
1B	UPPER HALF CASING	20	SHAFT SLEEVE NUT	59	CROSSOVER CAP
2	IMPELLERS	22	LOCK NUT	63	STUFFING BOX BUSHING
6	SHAFT	29	LANTERN RING	68	SHAFT COLLAR
7	CASING RING	31	INBOARD BRG HOUSING	69	BEARING LOCK WASHER
8	IMPELLER RING	32	IMPELLER KEY	109	CENTER PIECE
13	PACKING	33	OUTBOARD BRG HSG	113	CHANNEL PIECE BUSHING
13A	SHAFT SLEEVE O-RING	35	INBOARD BRG COVER	123	BEARING END COVER
14	SHAFT SLEEVE	37	OUTBOARD BRG COVER	125	GREASE FITTING
16	INBOARD BEARING	40	DEFLECTOR	127	SEAL PIPING
17	GLAND	41	BEARING CAP		

FIGURE 4-6 Horizontal, multi-stage, volute pump, flexible coupled for clear service.

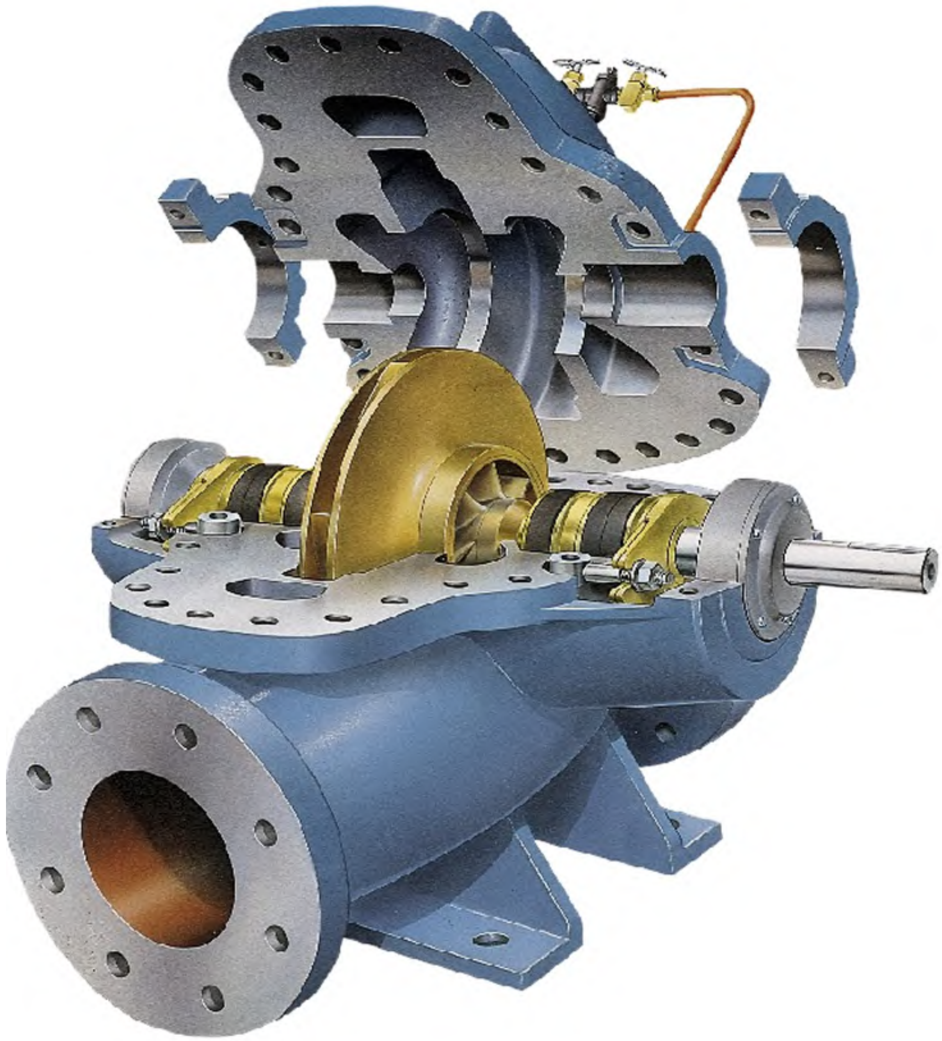


FIGURE 4-7 Double suction axially split, volute pump.
(Courtesy Patterson Pump Company, A subsidiary of The Gorman-Rupp Company)

industry, but they must be disassembled to inspect the rotating assembly and require appreciable removing space at the end of the pump.

The word, **diffuser**, describes how the water flows through the impeller and bowl assembly of this pump. **Diffuser** pumps can be high head, low specific speed, Figure 4-8, and are called **vertical turbine.** With medium head and specific speed, they are called **mixed flow** while low head, high specific speed diffuser pumps are called **axial flow or propeller pumps.** Mixed flow and axial flow pumps are found on large cooling tower installations where condenser water is pumped from a basin under the cooling towers. For above ground installations, vertical turbine pumps are enclosed in a steel assembly and are called "**can**" pumps, Figure 4-9. These pumps are also known as **vertical turbine, close coupled** pumps. Another small diffuser pump is used for boiler feed to small, high pressure steam boilers, Figure 4-10.

SELECTION OF TYPE OF PUMP:

There is much customer preference in the selection of the pump type in the HVAC field. Generally, single suction pumps are used for smaller flows while split case pumps are used on larger installations. The determining factors for pump selection are usually 1) pump efficiency, 2) first cost, 3) ease of maintenance, 4) floor space in the equipment rooms and 5) environment. Some engineers prefer frame type, end suction pumps while others like the close coupled type since there is a register fit and no possibility of misalignment. The close coupled pumps are lower in first cost in the smaller sizes but become higher priced in the larger motor sizes. No longer is a there a preference based upon certain pump flows as there was in the past. (See Chapter 8 for a detailed discussion of pump selection.)

Most HVAC pumps are located in equipment rooms where the ambient air temperature and cleanliness are controlled. The exception is condenser or cooling tower pumps that are often located outdoors near large cooling towers. Under very unusual conditions, chilled water pumps may be located outdoors. Care should be taken by installing pumps indoors where possible.

PUMP MATERIALS AND ACCESSORIES:

Pump materials and accessories for HVAC pumps are quite simple, as almost all of the pumps are "clear service" meaning that they do not pump solids or stringy material. Following are some elementary construction specifications.

1. **Pump materials** are normally cast iron volutes and bowls with bronze sleeves, impellers, and case rings. Volutes and bowls can have 125 psig construction for 175 psig service or 250 psig for 400 psig service up to 150°F. For higher temperatures, these ratings are reduced in accordance with ANSI/ASME B16.4.

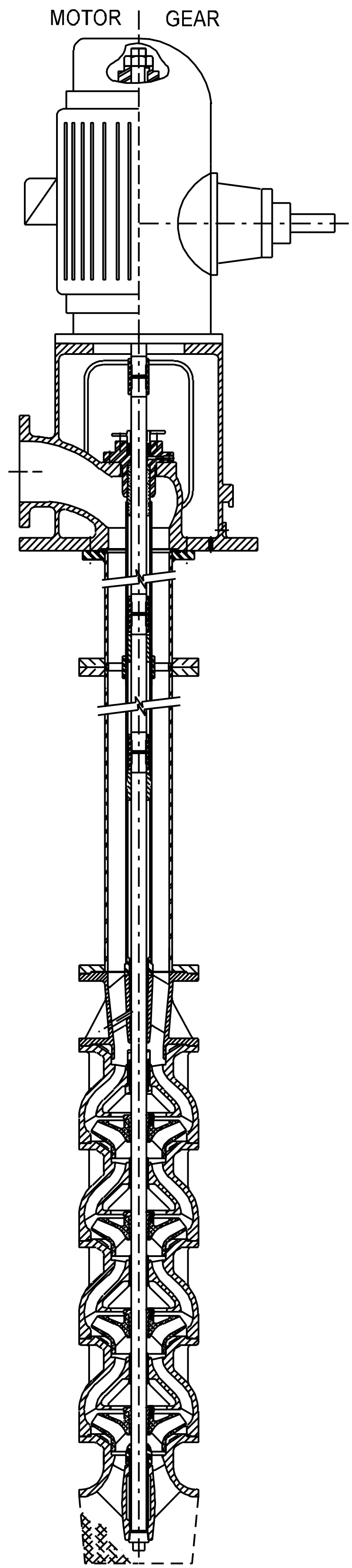


FIGURE 4-8 Diffuser pump, enclosed impeller, enclosed lineshaft.

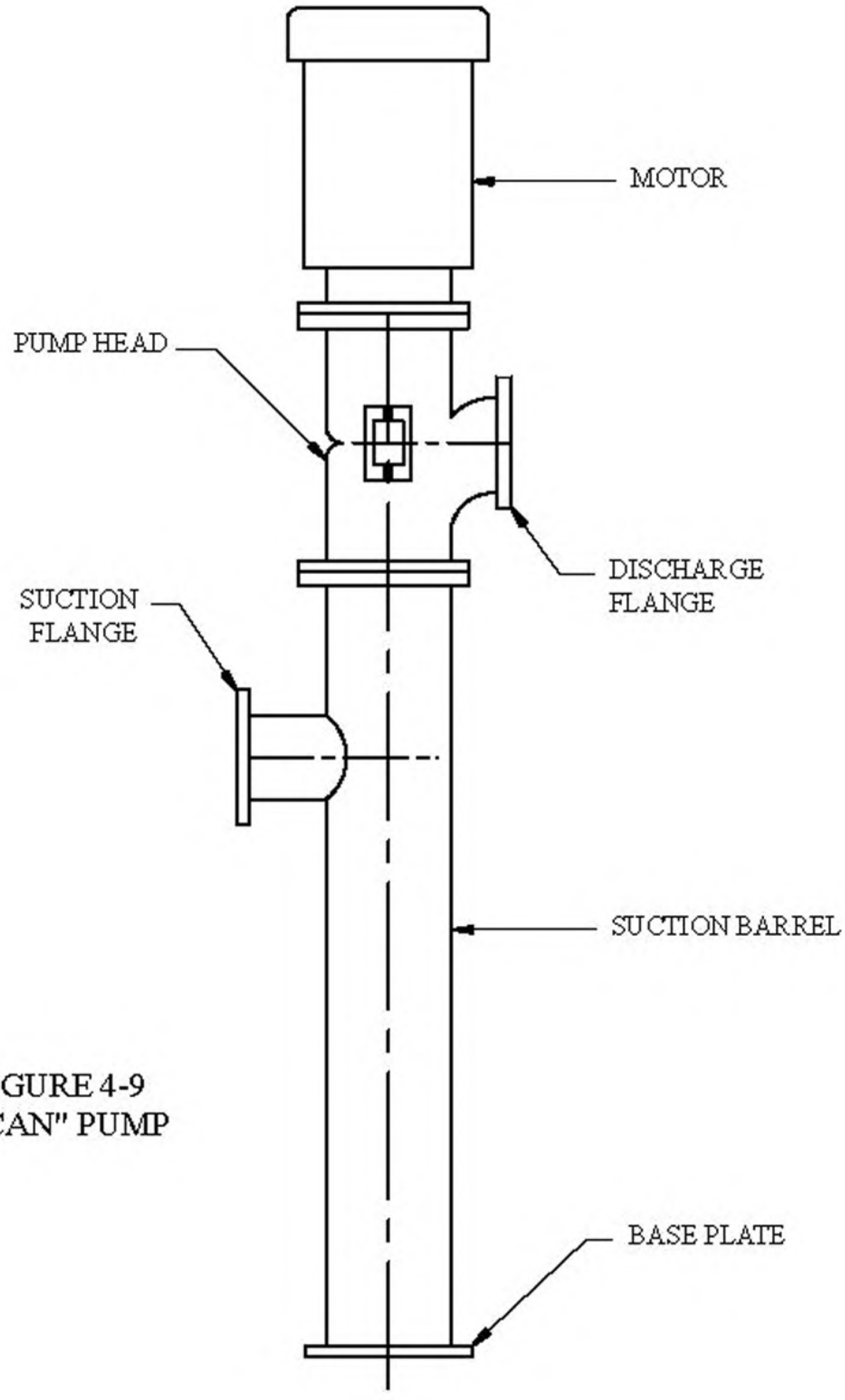


FIGURE 4-9
"CAN" PUMP

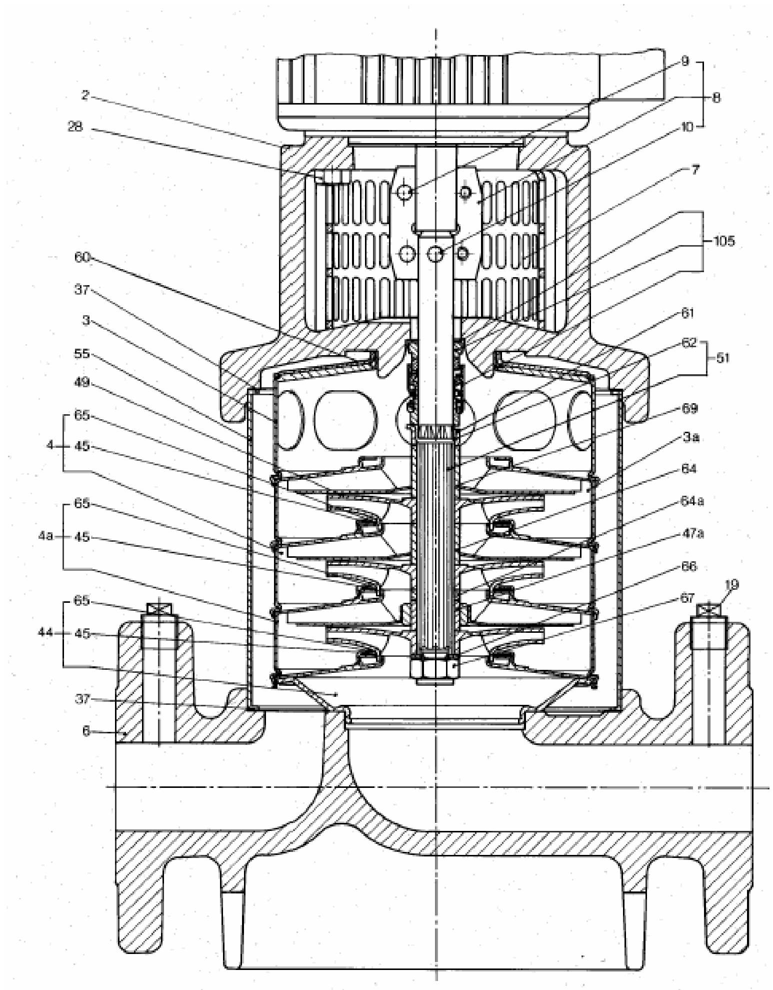


FIGURE 4-10. Vertical, multi-stage pump, flexible coupled.

Ductile iron in Classes 55, 56, and 57 can be provided for high pressure service in lieu of cast iron. Pump shafts are carbon steel or stainless steel.

2. **Pump Impellers** are always enclosed type, never open or semi-open. The only exception might be very large diffuser pumps for cooling tower service.

3. Most HVAC pumps are equipped with **mechanical seals**, not packing. There may still be some customer preference for packing on condenser pumps due to the material that gets into the cooling tower water.

4. All pumps flexibly coupled to their motors or other drivers must be equipped with **OSHA Approved** coupling guards. This now includes the couplings on diffuser pumps.

5. Most HVAC pumps are equipped with a suction strainer of the in-line or suction diffuser type. It is generally recommended that the strainer for condenser pumps be installed on the discharge rather than the suction of the pump. The strainer is there to protect the tubes of the chiller condenser. A strainer is normally provided on the cooling tower that is adequate to protect the pumps.

6. A manual shut-off valve should be installed on the suction and discharge of most HVAC pumps. A non-slam type check valve should be installed on the pump discharge.

7. Pump bases for flexibly coupled end suction and split case pumps should be of structural steel that insures stability and maintains the alignment of the pumps and motors. Drip-rim bases are discouraged in this industry due to the possibility of Legionnaires disease. Instead, the pumps should have taps near the mechanical seals to remove drainage through piping.

CENTRIFUGAL PUMP DESIGN CONSIDERATIONS:

The pump designer must recognize the effects that the water flow has on the centrifugal pump. The pump must be designed to accommodate the forces and leakages that occur in the pump as well as the suction and discharge pressures that exist on a pump. These are design concerns and will be further discussed in Chapter 5 on Pump Performance.

Forces in a Centrifugal Pump:

Mechanically, the HVAC volute type pump must be designed to withstand both axial and radial thrusts as shown in Figure 4-11 for an end-suction pump. Axial thrusts for both end-suction and split-case pumps are shown in Figure 4-12. Radial thrusts for these volute type pumps vary with the flow through the pump

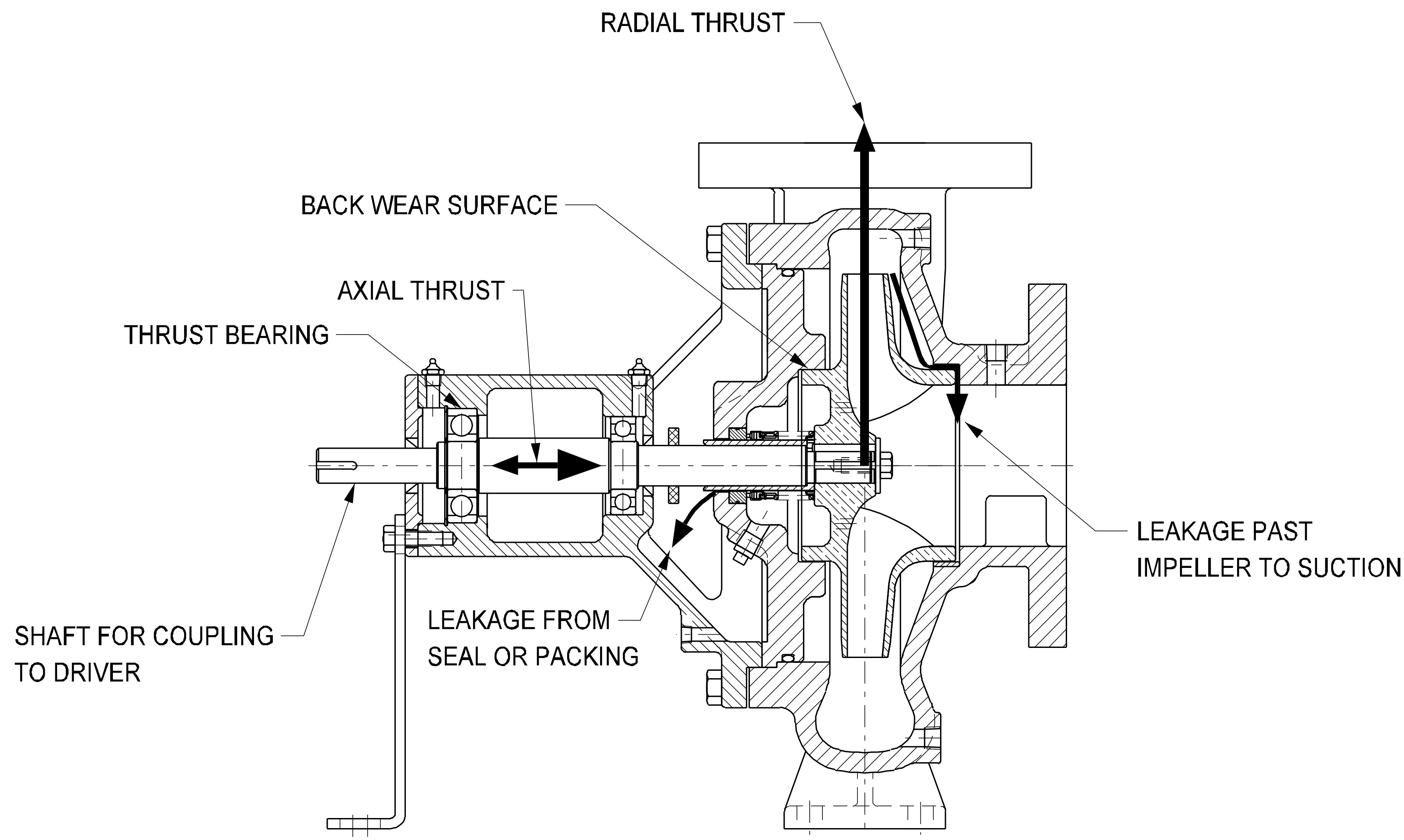
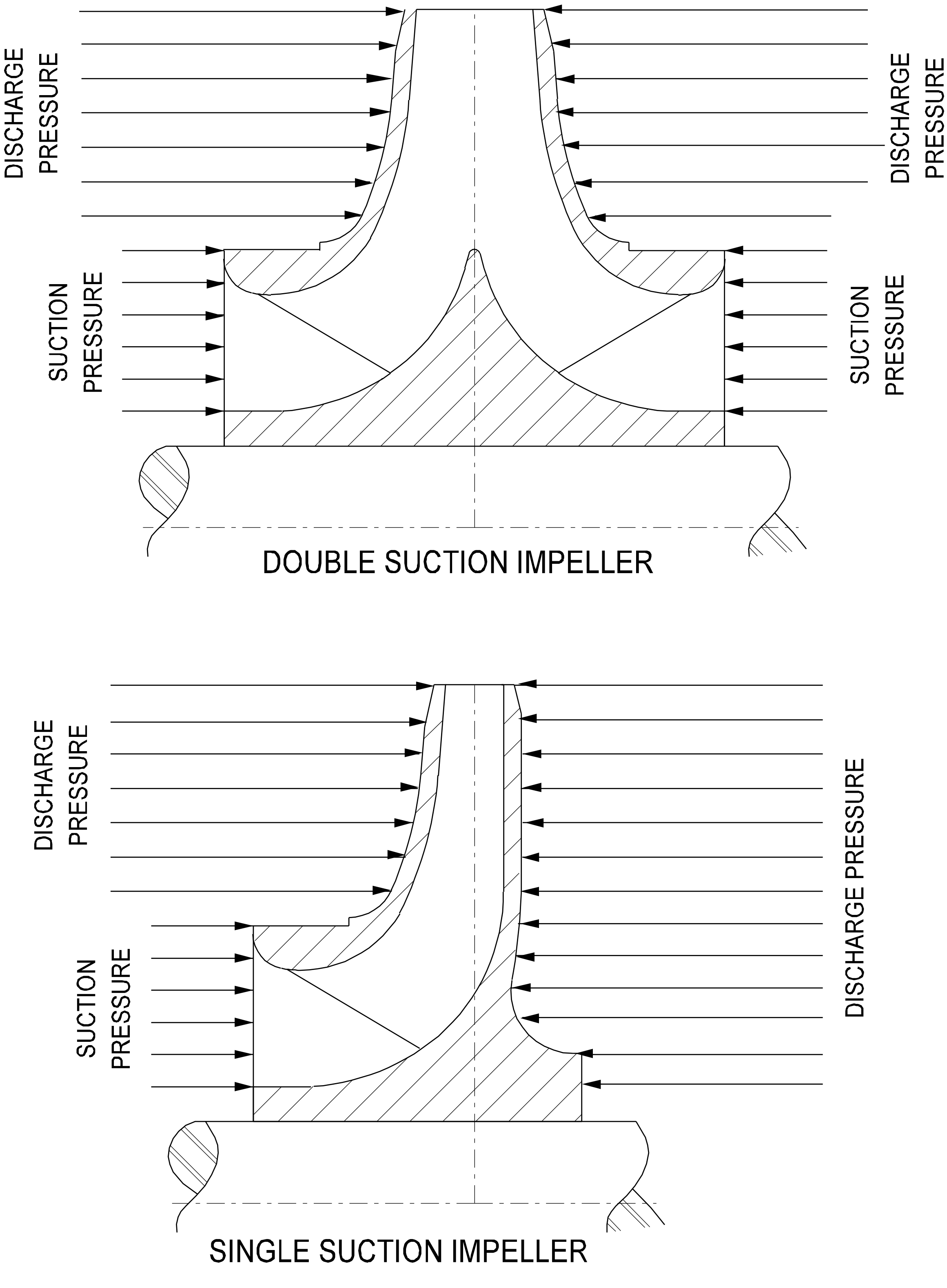


Figure 4-11. Forces and leakages in a volute-type pump.



From: "Pump Handbook", Karassik et al, 3rd edition, McGraw-Hill, New York, 2001.

Figure 4-12. Axial thrust in volute-type pumps.

and the pump speed, Figure 4-13. The minimum radial thrust for volute pumps usually occurs near the point of best efficiency (BEP).The designer must recognize these forces and provide a pump structure that accommodates them. The HVAC system engineer who selects pumps must understand these forces when selecting pumps for a particular application. It is obvious that volute type pumps should be selected and operated as closely as possible to the BEP to avoid undue wear in the pump. One of the great values of variable speed operation is the reduction in radial thrust with lower operating speeds.

Diffuser pumps have very little radial thrust compared to volute pumps. It is easily designed for by the pump manufacturer. Axial thrust must be recognized, as it varies with the type of impeller, Figure 4-14. This is not a concern in most HVAC applications since the system heads are seldom above 100 ft.

Leakages in Centrifugal Pumps:

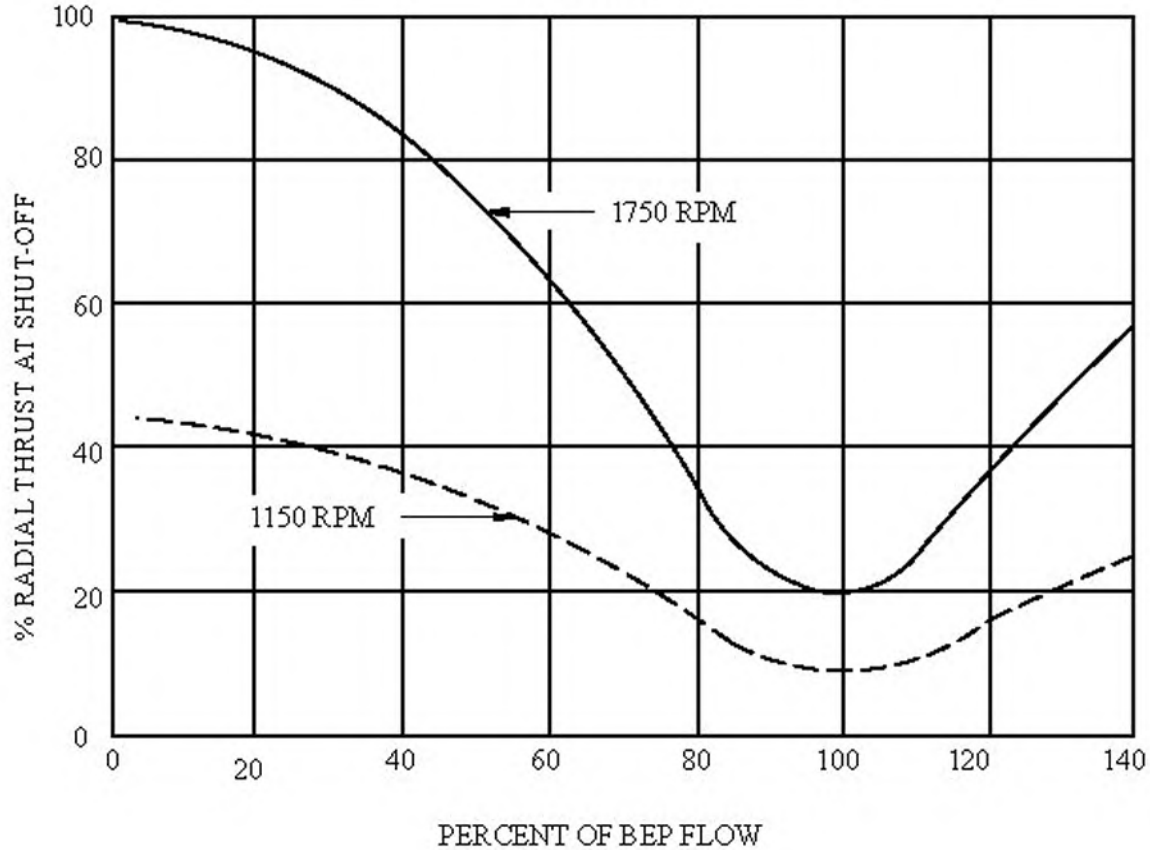
Figure 4-11 describes the two leakages that occur in all types of centrifugal pumps, namely, leakage from mechanical seals or packing and leakage from the discharge to the suction of the impeller. This latter is also called "bypassing". Leakage from the seal or packing should be drained from the pump by a pipe connecting it to the nearest floor drain. Leakage or bypassing should be held to a minimum by the clearance between the impeller and case ring. This is one of the most important design considerations for a centrifugal pump. The pump designer must determine the minimum clearance possible for the flexibility of the pump shaft. **Obviously, shorter and larger diameter pump shafts will have less flexibility at the casing ring and, therefore, can have a smaller clearance. This results in higher pump efficiency since the bypassing isreduced.**

Suction Conditions:

The entrance of water into a centrifugal pump is important, as it determines the performance and useful life of the pump. All centrifugal pumps have a net positive suction head curve showing the required suction pressure at various flows through the pump. This is called the NPSHR curve and is shown on the pump curves in Chapter 5. NPSHR is not a concern on most HVAC pump applications, only when taking water from a storage tank. The net positive suction head available, NPSHA, must be calculated for such installations, Figure 4-15. The pressures that are exerted on a pump suction include the atmospheric pressure at a specific altitude and the static pressure in the tank above the pump suction. If the water is lifted from the tank, the lift becomes a negative pressure. The vapor pressure of the water and the friction of the suction piping must deducted to determine the net positive suction head available for the actual installation. In Figure 4-15, if the altitude is 1,000 ft, from table 2-1, the atmospheric pressure is 32.7 ft. of water. From Table 2-2, the vapor pressure for 85°F water is 1.4 ft. With a static head of 8 feet on the pump suction and 6 ft. of friction loss in the suction piping, the total NPSHA is 33.3 ft. Any pump taking

FIGURE 4-13

RADIAL THRUST CURVES
FOR
VOLUTE PUMPS



THE PHYSICAL DESIGN OF CENTRIFUGAL PUMPS FOR WATER

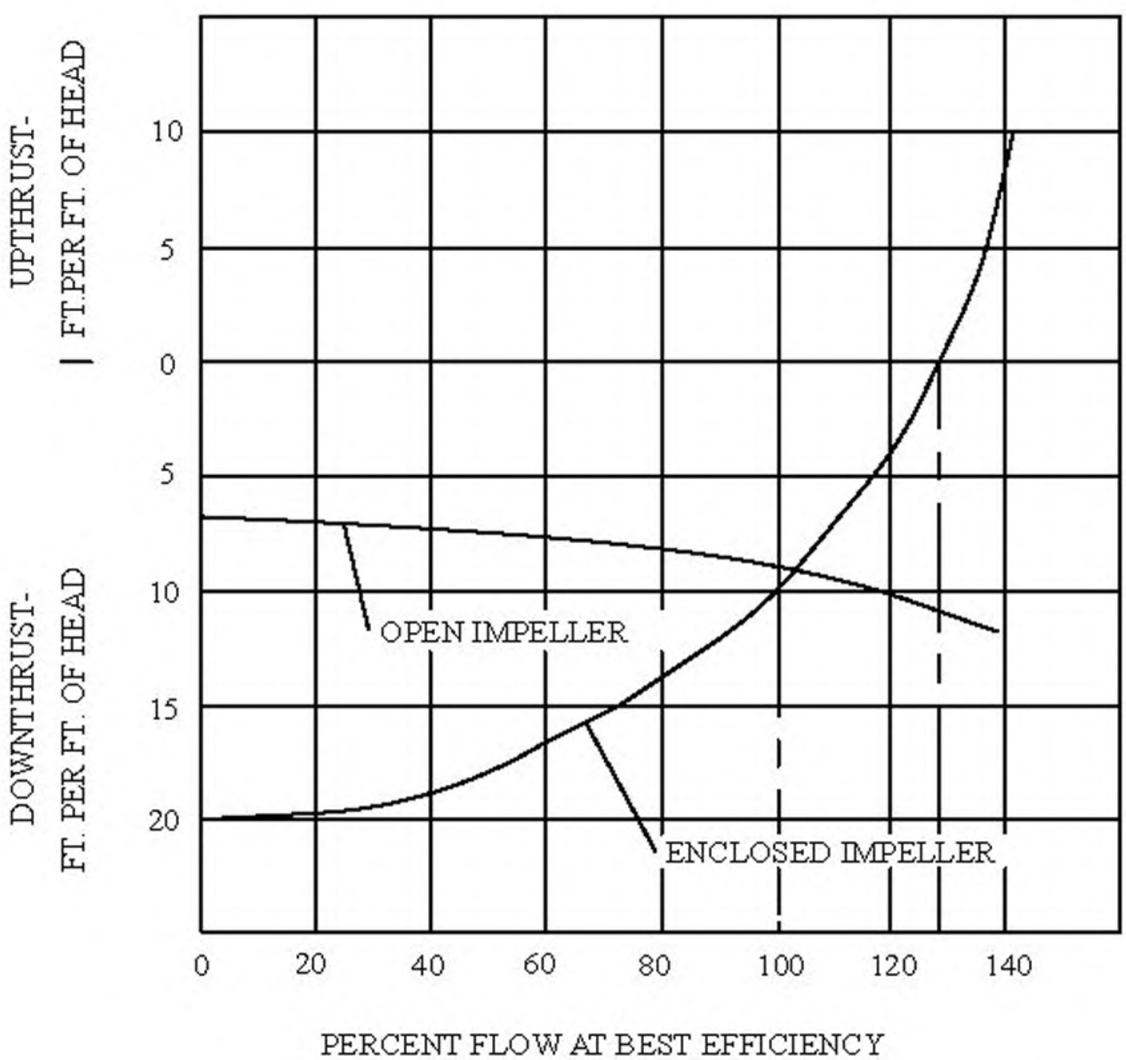
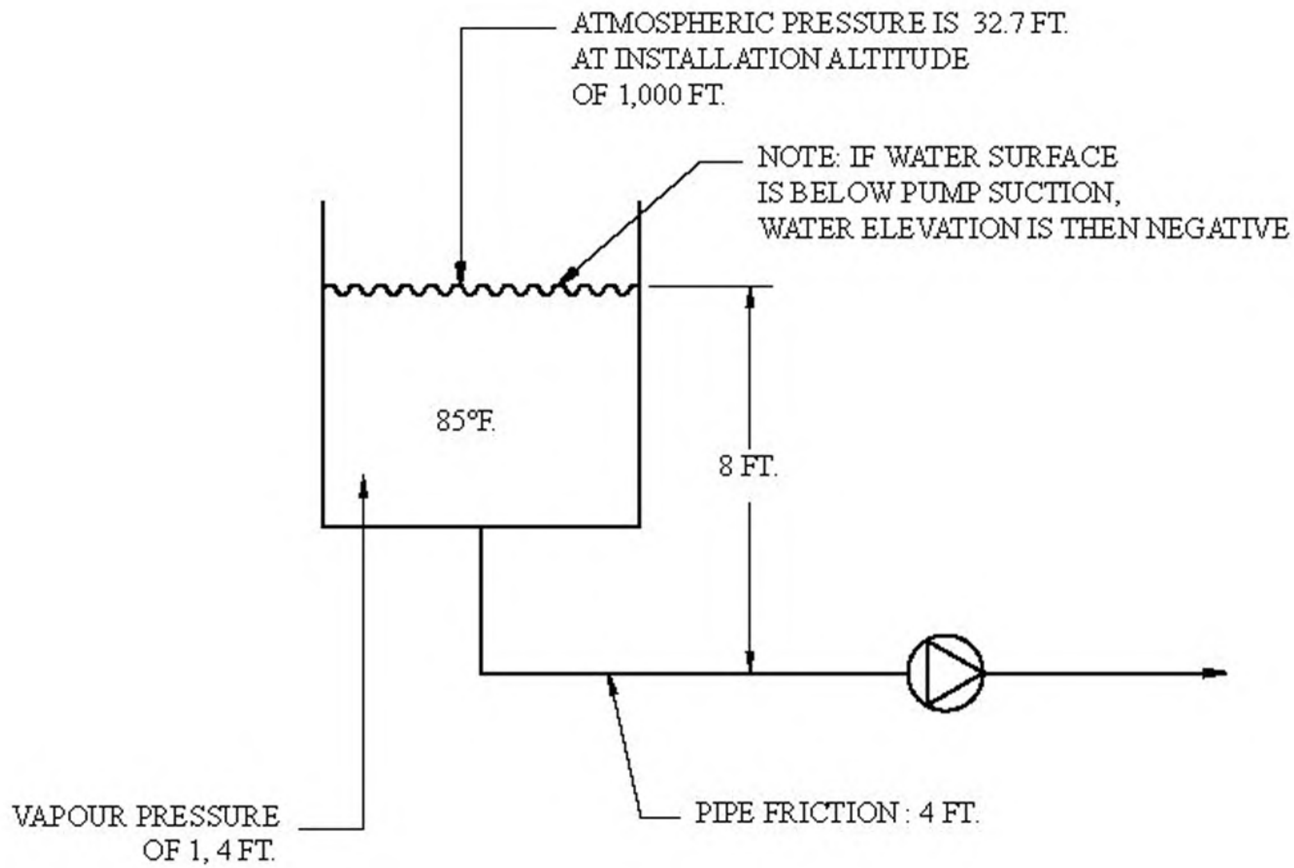


FIGURE 4-14

Axial thrust vs. rate of flow curves for axial flow pumps.

From: "Water Pumps and Pumping Systems", James Rishel, PE,, McGraw-Hill, New York, 2002.

FIGURE 4-15



ATMOSPHERIC PRESSURE IS 32.7 FT.
AT INSTALLATION ALTITUDE
OF 1,000 FT.
NOTE: IF WATER SURFACE
IS BELOW PUMP SUCTION,
WATER ELEVATION IS THEN NEGATIVE
85°F.
8 FT.
VAPOUR PRESSURE
OF 1, 4 FT.
PIPE FRICTION : 4 FT.

water from such a tank must have an NPSHR of less than 33 ft. at any possible flow rate.

Vortexing in open tanks is a serious problem that must be addressed when applying pumps that take suction from them. On small pumps, a vortex plate should be installed above the side entrance from a tank, Figure 4-17b. On large cooling tower pumps, the system designer should study carefully ANSI/HI 9.8-1998, “Pump Intake Design”, to insure that the installation will not have problems with flow into the pumps.

Water flow into pump suctions should be smooth without any twisting velocities. Manufacturers of complete pumping systems take this into account in their pump suction piping. Field suction piping should be in accordance with the recommendations of the pump manufacturer.

Air in Pumps:

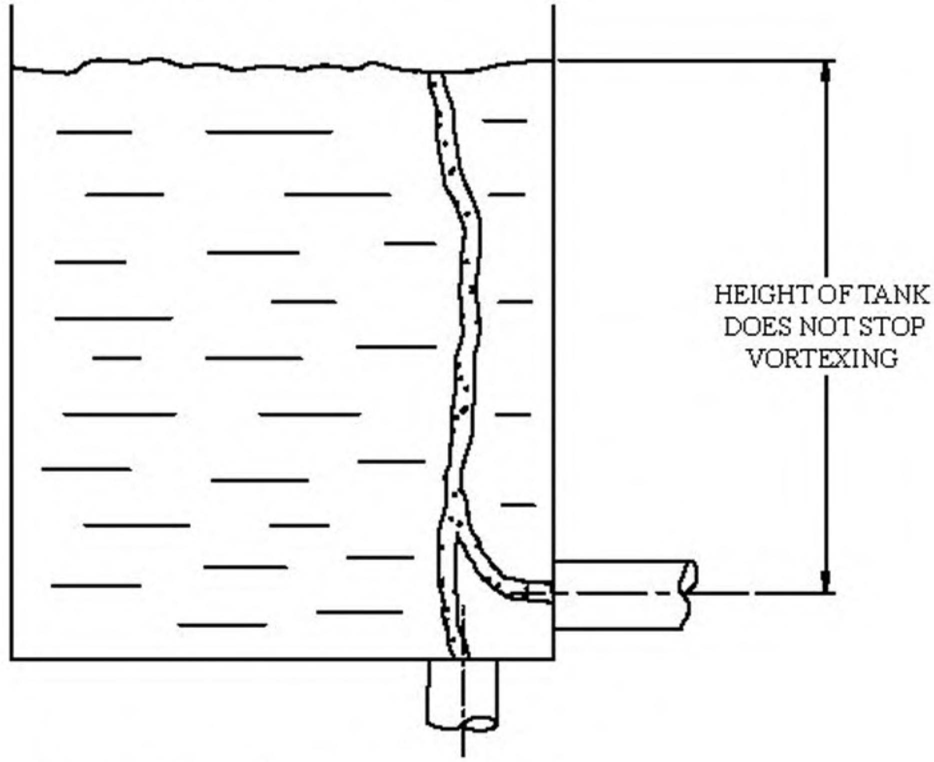
Entrainment of air in pumps has an appreciable effect upon their performance. Fortunately, HVAC chilled and hot water systems usually have air separators and air vents that remove the air from them. Such is not the case with condenser or cooling tower pumps where the water is laden with air. Figure 4-17a describes the effect of air entrainment upon the performance of a centrifugal pump while sources for air entrainment are shown in Figure 4-17b.

Air separators are normally installed on the suction line to pumps for chilled and hot water systems. Cooling tower and condenser pumps present different problems. The suction piping for these pumps must be designed for lower velocities and great care should be taken in calculating the NPSHA and the NPSHR. Generally, strainers should not be installed on these pump suctions to avoid fouling and reduction of the NPSHA. If strainers are required to protect the tubes of the condensers, they should be installed on the pump discharges.

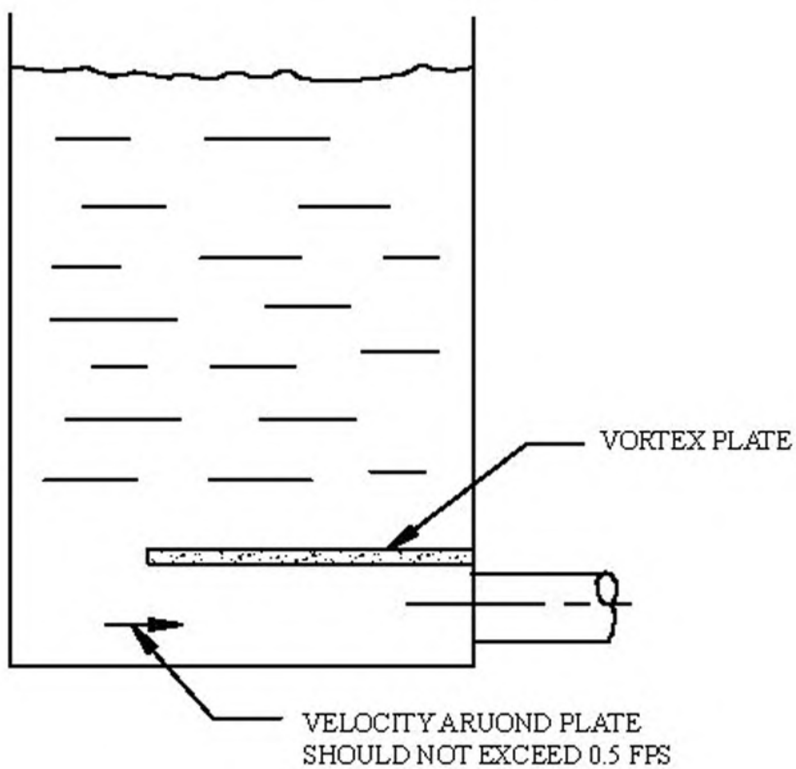
QUALITY FEATURES OF PATTERSON PUMPS:

1. All Patterson HVAC pumps contain a **case wear ring**, Figure 4-18a. To not install such a ring results in the impeller running against a cast iron surface, Figure 4-19b. Fine particulate can be found in HVAC pumps. As this particulate passes through the opening between the impeller and casing, the surfaces of the impeller and casing become scored. Also, any internal rusting of the casing or bowl increases the clearance. The result is loss of pump efficiency and ultimate replacement of the pump casing. This is an expensive procedure since the entire pump must be removed, requiring disconnection from the piping and the motor. The casing ring does not rust, and maintains the desired clearance. If it does become damaged, it can be replaced without disconnecting the casing from the piping or motor.

FIGURE 4-16

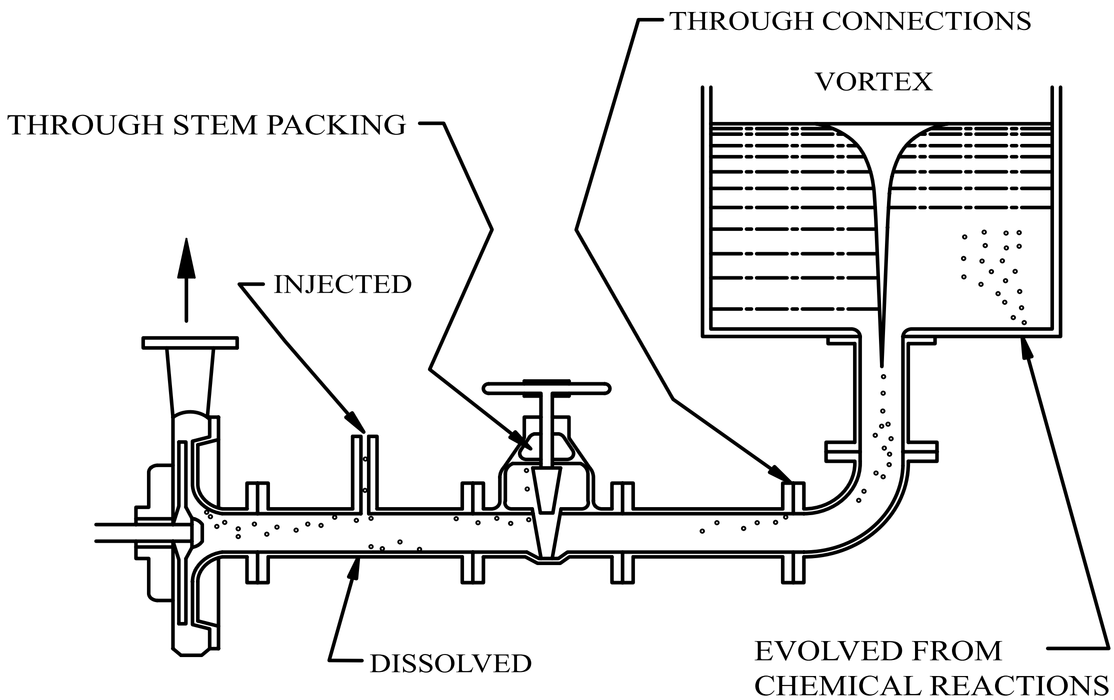


a. Vortexing in an open tank.

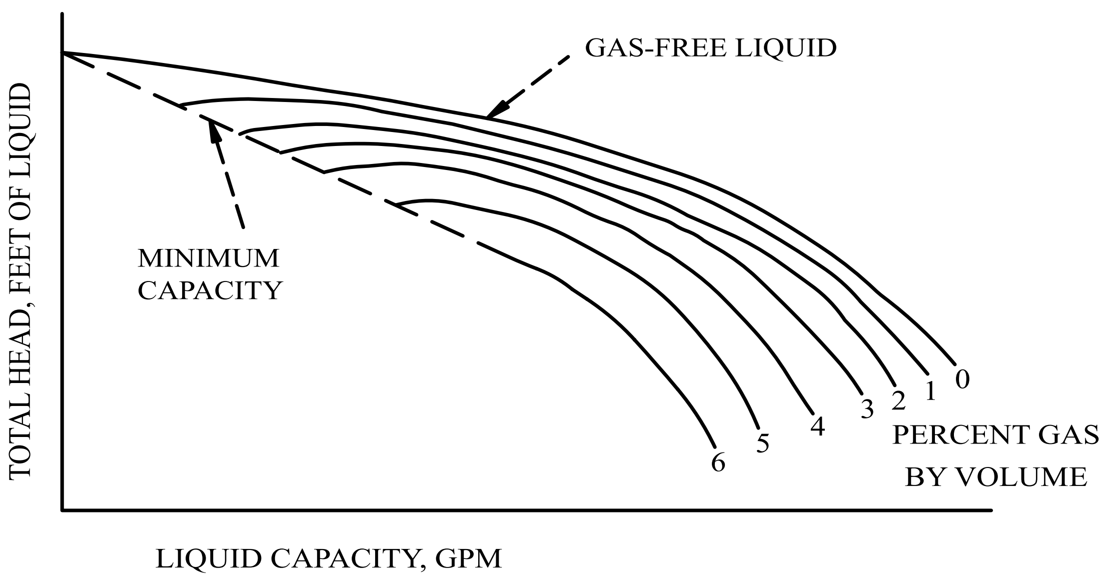


a. Vortex plate for side exit from tank

From: "Suction Side Problems, Gas Entrainment", John H. Ingram, Pumps and Systems Magazine, September 1994, p. 34.



b. Centrifugal pumps and entrained-air problems.



a. Effects of various amounts of entrained gas on pump characteristics.

FIGURE 4-17 Air in pump suctions.

(From: John H. Doolin, "Centrifugal Pumps and Air Entrainment Problems". Chemical Engineering Magazine January 7, 1963, p. 103.)

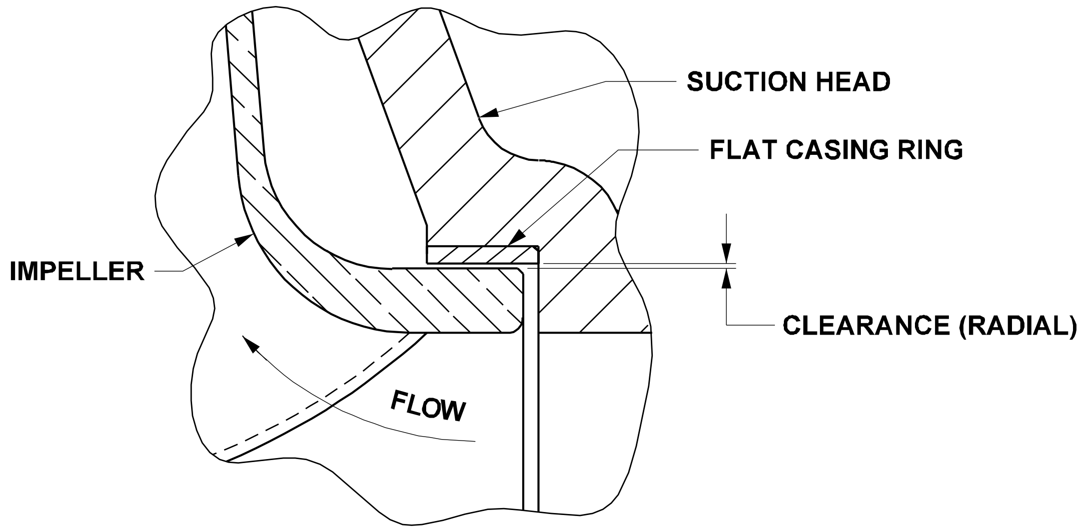


FIGURE 4-18a Single flat casing ring construction.

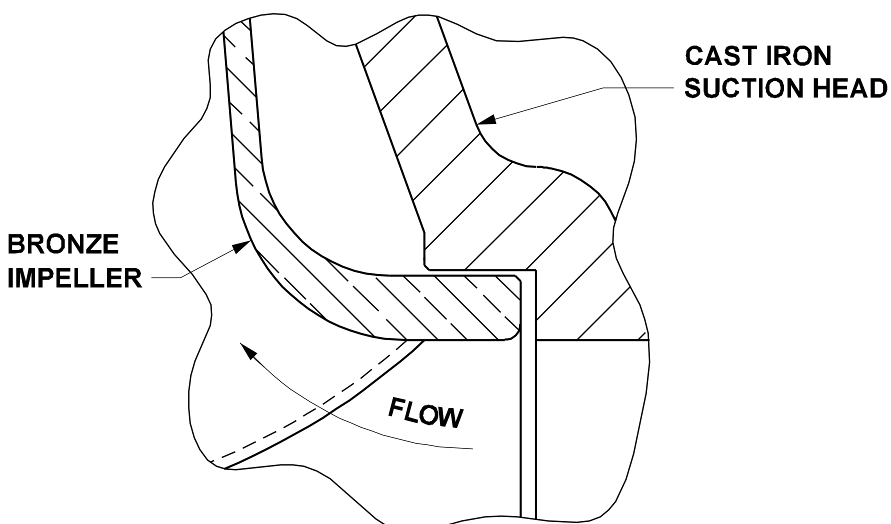


FIGURE: 4-18b

No casing ring, bronze against cast iron.

2. The distance between the impeller and case ring is called the **"radial clearance"**, Figure 4-18a. As indicated above, the overall efficiency of a pump is determined by this clearance. The allowable clearance to prevent wear is determined by the stiffness (diameter) of the pump shaft. Patterson pumps are designed with shafts that limit deflection in the allowable operating range (AOL) to 0.002". The result is minimal wear at the impeller and case ring.

3. The smoothness of the entry of water into a pump determines the quality and efficiency of that pump. Figure 4-19 describes various suction configurations. Figure 4-19a describes the ideal suction design for pumps taking lift from a tank or lake. There is no possibility for air to collect in the suction. Figure 4-19b describes quality construction for pumps taking water under pressure. Figure 4-19c shows abrupt suction design that creates turbulence and reduces efficiency.

4. Rough or uncontrolled surfaces of the casing or bowl for a centrifugal pump can reduce its efficiency. Patterson's specification for interior surface finish in hydraulic passages is 200 RMS.

5. All Patterson HVAC pumps include sleeves over the shaft. This reduces the maintenance required and insures that the impeller is located at the position on the shaft that insures maximum efficiency for the pump.

6. All Patterson HVAC pumps can be repaired without disturbing the piping or motor, excepting the close-coupled, end suction pumps. All frame mounted, end suction pumps are of the **"back pull-out"** design, so they can be serviced without disturbing the motor when they are specified with a spacer type coupling

The above statements demonstrate the need for the design engineer or owner of pumps to carefully inspect a pump cross–section drawing before specification or purchase. On large projects including many pumps, an actual pump should be inspected internally for compliance to the specifications.

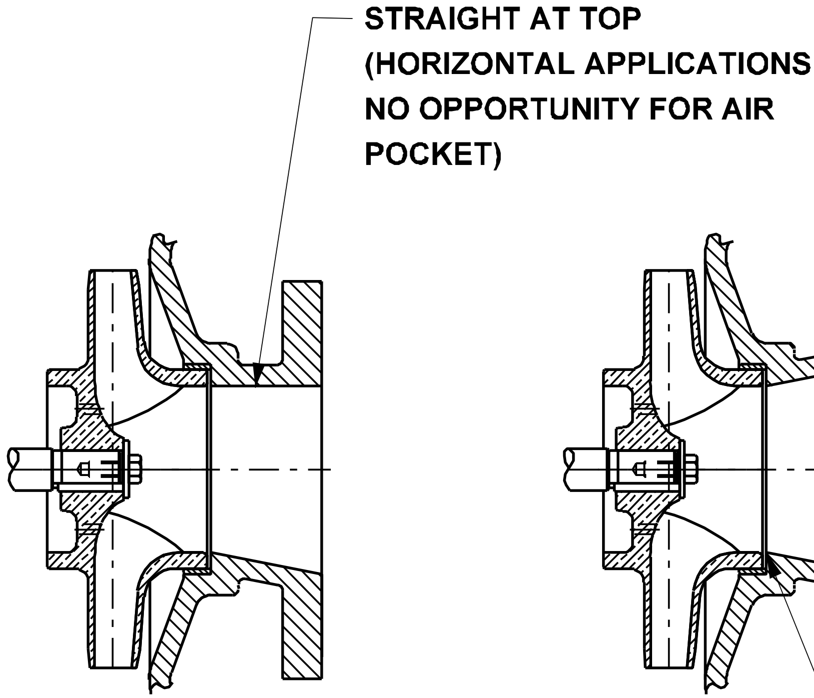
Centrifugal Pump Nomenclature:

It is well to know the principal parts of centrifugal pumps. Figure 4-20 describes these parts for an end-suction, frame mounted pump.

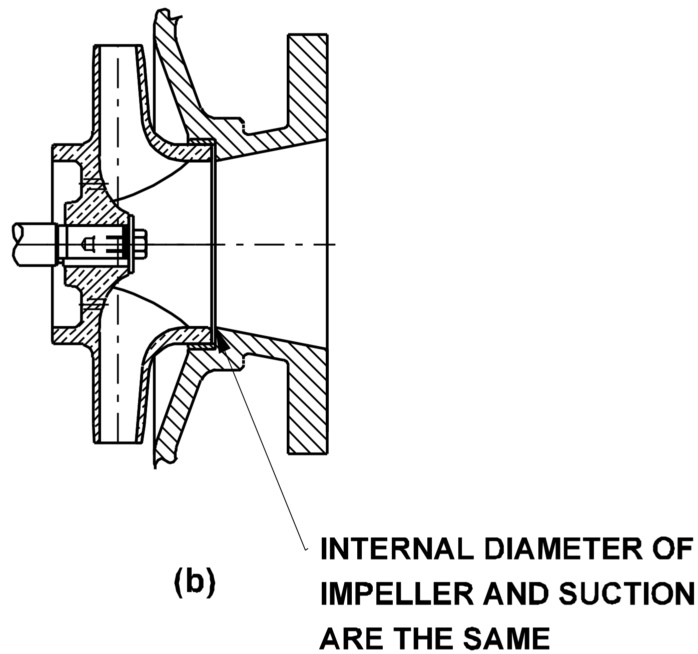
ELECTRIC MOTORS:

Most HVAC pumps are driven by electric motors. In some stand-by situations, these pumps may be driven by diesel engines. In large boiler plants, steam turbines may be used to drive HVAC pumps.

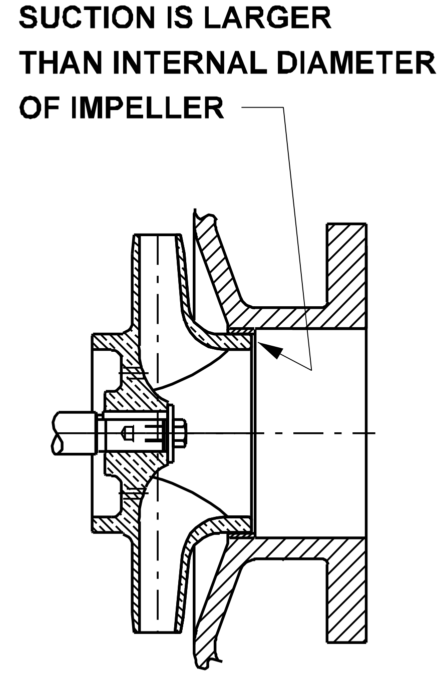
The power supply for most of these pumps is 480 volt, 3 phase, 60 Hz. Smaller circulating pumps can be operating on 120 or 230 volt, single phase power. Some buildings such as hospitals may have 208 volt, 3 phase, 60 Hz power.



(a)



(b)

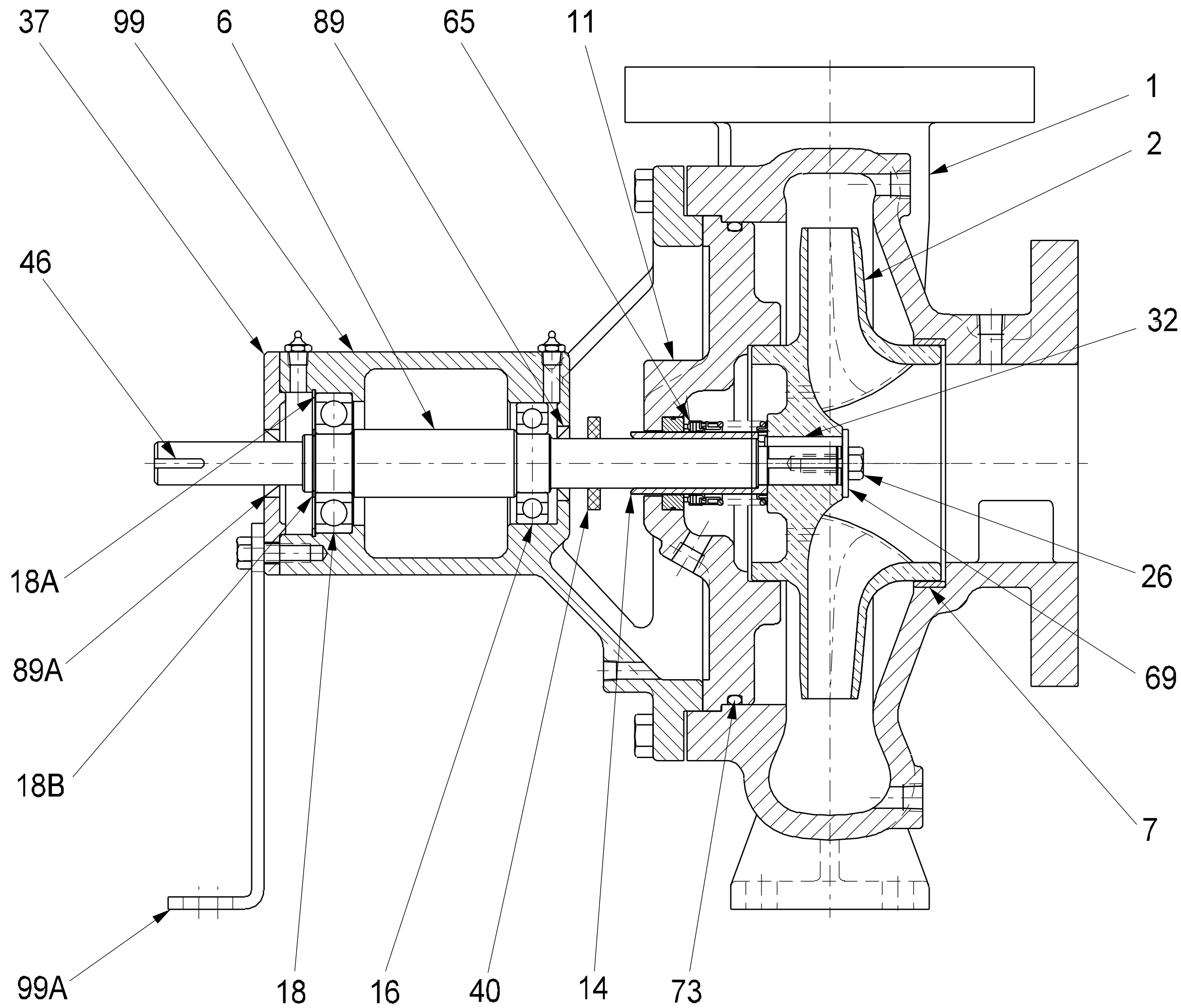


(c)

FIGURE 4-19a
Quality pump suction design

FIGURE 4-19b
Quality pump suction design

FIGURE 4-19c
Average pump suction design



Overhung impeller, separately coupled, single stage, frame mounted

1	CASING	18A	OUTER RETAINING RING	65	MECHANICAL SEAL
2	IMPELLER	18B	INNER RETAINING RING	69	IMPELLER WASHER
6	SHAFT	26	IMPELLER SCREW	73	O-RING
7	CASING RING	32	IMPELLER KEY	89	INBOARD SEAL
11	VOLUTE COVER	37	BEARING COVER	89A	OUTBOARD SEAL
14	SHAFT SLEEVE	40	DEFLECTOR	99	BEARING HOUSING
16	INBOARD BEARING	46	COUPLING KEY	99A	BRG. HOUSING SUPPORT
18	OUTBOARD BEARING				

FIGURE 4-20 Cross-sectional of an End-Suction, volute type pump.

Some remote areas of the United States still have 50 Hz power. In these cases, a careful evaluation should be made of the available service before pumps or motors are selected. Similarly, foreign countries operating on 50 Hz power require a careful evaluation of the actual service before pump or motors are selected.

The motor enclosure for most HVAC motors can be open frame, drip-proof. There is no need for TEFC enclosures in most HVAC equipment rooms. Outdoor installations require evaluation of the ambient conditions to determine the proper enclosure. Vertical diffuser pumps are usually equipped with WP-1 weather proof enclosures for their motors.

Motor selection has changed now that variable speed drives are equipped with current limiting features that prevent overloading when the pump runs out its curve. Motors and drives can now be selected at the design condition of flow and head (See Figure 8-2).

VARIABLE SPEED DRIVES:

Almost without exception, variable speed drives for pumps and fans in the HVAC world are of the variable frequency type. Since their first introduction into this field in the 1970's, they have improved greatly, both in efficiency and reliability. Usually, there are no problems any longer with magnetic and radio interferences. Also, their size and cost have been reduced considerably. For the larger sized pump motors, above 50 hp, they are probably close to the cost of reduced voltage starters and special motors.

Most of these drives now have efficiencies in the range of 96 to 98%. Their power factor is above 90% for the drive and motor. Performance of these drives will be reviewed in Chapter 5 on pump performance.

Most variable frequency drives are pulse width modulated (PWM) at 480 v. up to around 750 hp. At 750 hp and above, medium voltage drives for 2,400, 4,160 or 6,900 volts should be evaluated. There are a number of different designs for medium voltage drives such as load commutated, neutral-point-clamped inverters, and multi-level series-cell inverters. This is an active market with new designs being offered continuously. Various designs should be studied for the motor hp and voltage of each installation. So far, ABB and Robicon are the principal manufacturers of these medium voltage drives.

There are some mechanical design features of all variable speed drives that should be included.

1. Approval: All HVAC drives should be approved by UL - Underwriters Laboratories and ETL – Electric Testing Laboratories. In Canada, the drives should be approved by CSA – Canadian Standards Association. Care should be taken to insure that all accessories furnished with these drives bear these labels also.

2. Bypass Starters: In the past, the reliability of these drives was questioned, so they were equipped with these starters that would run a pump at full speed in event of failure of its drive. This created havoc on some systems that were not equipped with the necessary design features to accommodate continuous operation of a pump at full speed. The overall cost of utilizing these bypass starters is so great that it is better to install another standby pump and drive for emergency conditions.

3. Drive Enclosures: Most HVAC drives require only Nema 1 enclosures. Seldom is it necessary to go to Nema 3, non-ventilated drives that require air conditioners to maintain acceptable, internal temperatures.

4. Instrumentation: Minimum instrumentation on any of these drives is as follows:

a. Ammeter
b. Percent Speed meter
c. Hand-off automatic switch
d. Manual speed control
e. Common fault alarm

Application of Variable Speed Drives:

VFD's will provide years of uninterruptible service if installed and operated properly. The manufacturer's instructions should be followed carefully. Following are some installation conditions that should be addressed.

1. Ventilation: On variable torque applications such as HVAC pumps and fans, the heat expended by a variable speed drive is:

$$\textbf{BTU/hr} = \textbf{Max kW of drive} \times 3412 \times (1 - \eta_d) \qquad \textbf{(4-1)}$$

Where: η_d is the efficiency of the drive as a decimal.
3412 is the thermal equivalent of one kW.

2. Cleanliness: VFD's operate very well on HVAC pumps and fans if they are kept clean. Applications that are dusty can result in dust on the electronic components of the drives and reduced life of the drive. Such applications may require Nema 3R enclosures with air conditioners.

3. Chemical Attack: This is not a problem on most HVAC installations, but it must be emphasized that these drives must not be exposed to air borne chemicals such as hydrogen sulfide.

4. Maximum Temperature: Like most electrical equipment, VFD's are usually designed for 40°C (104°F) and should be equipped with internal air conditioners if the ambient temperature is such that it will cause higher internal temperatures in the drives.

5. Location: VFD's should be located in dry areas where they cannot be wetted by overhead pipes or surface water. If it is necessary to locate the drives under piping, the enclosures should be equipped with drip shields. Do not locate drives outdoors where they are exposed to direct sunlight. If it is necessary to locate the drives outdoors, protect them with a sun shield and insure that the acceptable internal temperature is maintained with internal air conditioners.

6. Number of Drives: There is no reason in HVAC applications to connect more than one motor to a VFD. Each pump should have its own drive.

Variable speed drives have proved to be very reliable for HVAC pump and fan installations. Substantial energy has been saved by installing variable speed pumps.

SUMMARY:

This is a brief discussion of pump, motor, and drive design. It is intended to provide a basic knowledge of this equipment. The various engineering and sales personnel at Patterson Pumps can provide more detailed responses to any aspect of this design.

Chapter 5

HVAC PUMP PERFORMANCE

HVAC PUMPS AND SYSTEMS
DESIGN MANUAL

PATTERSON PUMP COMPANY

INTRODUCTION:

An evaluation of HVAC pump performance must recognize that these pumps vary from small circulators with flows of less than 10 gpm up to large condenser pumps with thousands of gpm. Since all of these pumps are for clear service, concerns for viscosity and specific gravity arise only when pumping solutions of glycol or high temperature water. All of the equations listed in this chapter are for clear water with a specific gravity of one.

Before reading this chapter, it is urged that the pump suction conditions of Chapter 4 be reviewed that pertain to NPSH and air in pumps.

PUMP HEAD - FLOW CURVES:

The fundamental fact of centrifugal pump performance is that the pump head (pressure) developed by these pumps is dependent upon the flow through them. The basic curve showing this is in Figure 5-1 for various pump speeds. Most of the volute type pump curves have a continuously rising characteristic from maximum flow to the shut-off or no-flow condition. In the HVAC industry, these curves are normally published for pumps driven by four and six pole electrical motors. These are 1800 and 1200 rpm, induction motors; very large pumps with 750 hp and larger motors may be driven by synchronous motors. Originally, the curves were always published at 1750 and 1150 rpm. Now that the higher efficiency motors are available, these curves may be available at the actual full load speed of the motors, such as 1780 and 1185 rpm.

The basic configuration of head-flow curves has the flow, in gpm, plotted horizontally on the “x” axis while the head, in feet, is plotted vertically on the “y” axis. Pump head is always shown in feet of liquid rather than pounds of pressure. With the pressure expressed in feet of head, the pump curve is applicable to any liquid of any specific gravity.

Pump efficiency curves are also included on pump head-flow curves as shown on Figures 5-1 and 5-2. Do not even consider a pump that does not have them shown on the flow-head curve. Unfortunately, some small HVAC pumps provide only the motor size.

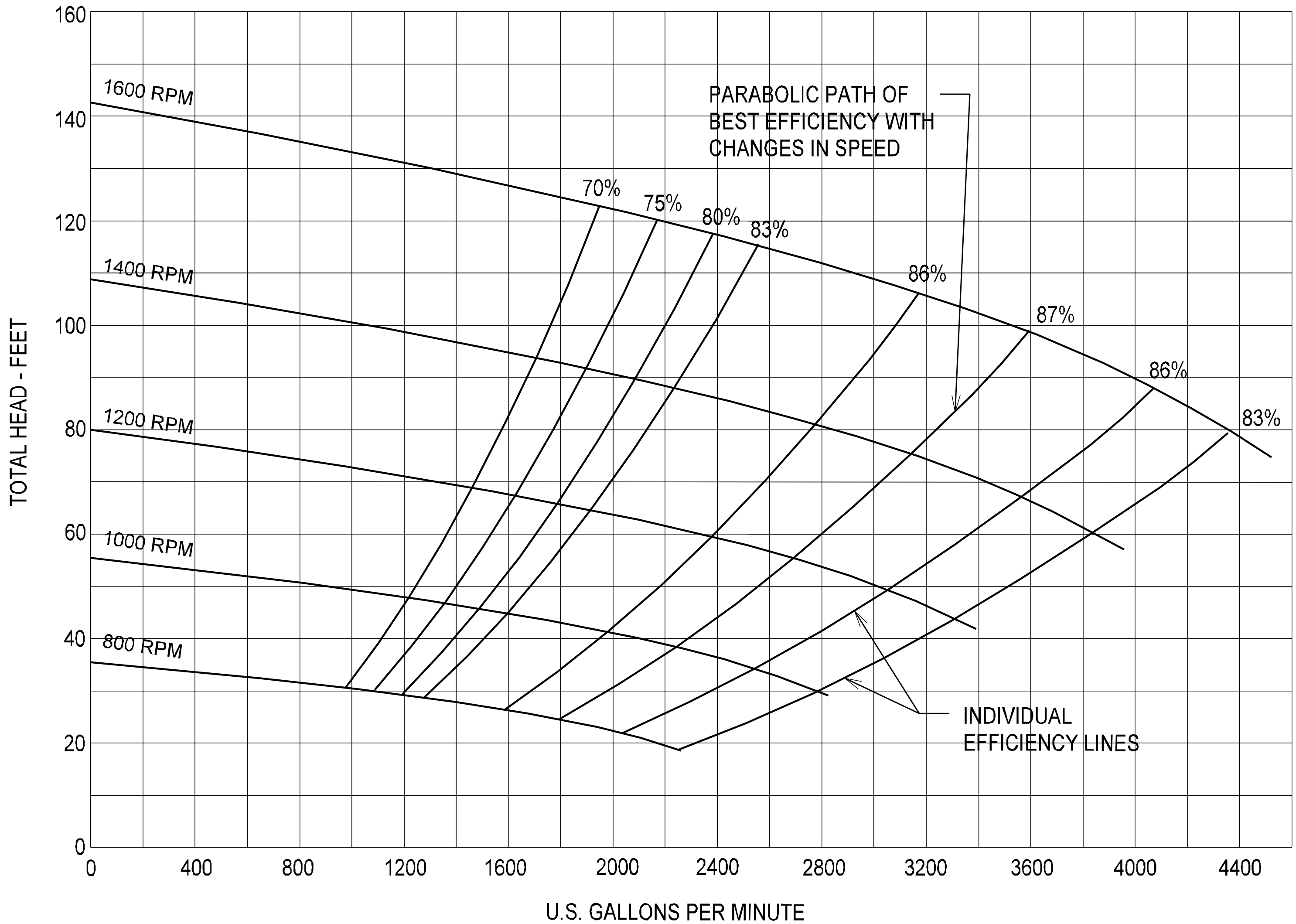
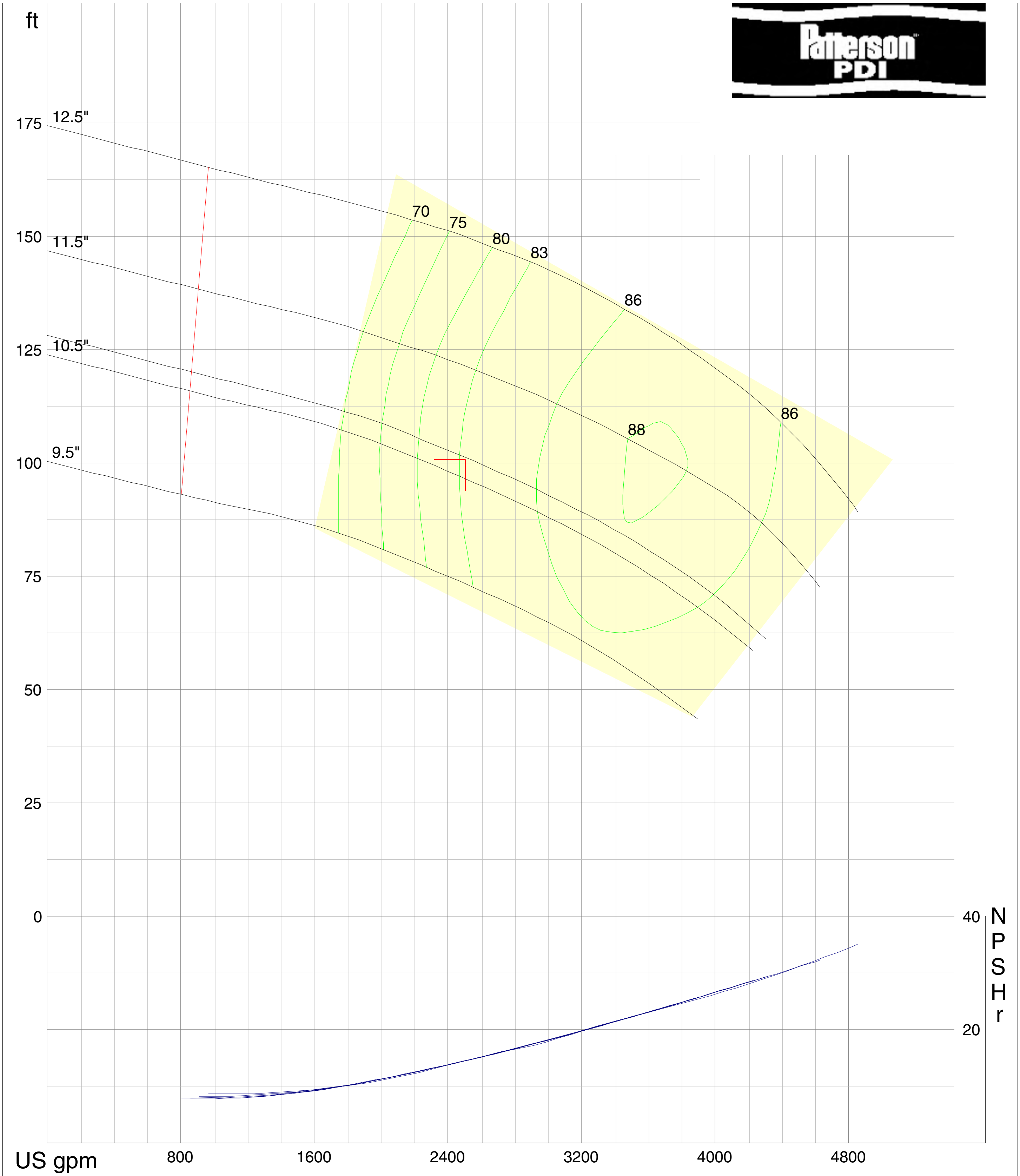


FIGURE 5-1. Head - Capacity curves for a variable speed pump.



Curve No.: 10MAD-A

Patterson Pump Company

07/19/04

.ufs

FIGURE 5-2

Patterson 60 Hz Pumps

Catalog: Patterson Pumps 60HZ, vers 1

HSC - 1800

Design Point: 2500 US gpm, 100 ft

Size: 12x10 MAD-A

Speed: 1770 rpm

Dia: 10.6875 in

Motor horsepower curves for HVAC pumps are normally included on the flow-head curves as shown in Figure 5-1. The motor horsepower curve can be shown below the flow-head curve, Figure 5-2, but it is not as accurate for various impeller diameters as are the horsepower curves included on the flow-head curves of Figure 5-1.

Before continuing this basic description of centrifugal pump performance, a clarification of the relationship of pump head in feet to gauge pressure should be explained. The feet of head per pound of pressure for any liquid are:

$$\textbf{Feet of head, } h = \frac{144}{\gamma} \tag{5.1}$$

Where γ = the specific weight of the liquid in lb/ft^3 at the actual water temperature.

For example, from Table 2.2, water at 60°F has a specific weight of 62.34 lb/ft^3. Therefore,

$$h \text{ for } 60°\text{F water, is } \frac{144}{62.34} \text{ or } 2.31 \text{ feet per pound of pressure}$$

With the exception of glycol solutions and high temperature water, the above figure of 2.31 is used for most HVAC applications.

Figure 5-2 describes a centrifugal pump's performance in the form that is usually provided by the pump manufacturer. The allowable impeller diameter range is shown with the efficiency and brake horsepower curves imposed upon them. The efficiency curves indicate that the efficiency reduces as the flow subsides. The Best Efficiency Point (BEP) is the peak efficiency for a pump, and selection should always be made as close to the BEP as possible. The pump manufacturer develops a number of pumps of different impeller diameters with various suction and discharge connections of a certain type such as end suction pumps. The result is a family of curves as shown in Figure 5-3.

It is important to know what causes the loss of efficiency in a centrifugal pump as the flow varies through it. Figure 5-4 describes the losses in these pumps. Recirculation is the major loss at lower flows while hydraulic losses are the greatest at flows beyond the flow at the BEP.

Diffuser pumps, particularly axial flow pumps, can have much higher specific speeds than volute pumps; there may be a dip in their head-flow curves as shown in figure 5-5a. Usually, these pumps are used for very high flows at relatively low heads for installations such as large condenser pumps. These pumps have another characteristic that is different than volute pumps, and that is their horsepower curve which may be near its maximum at the no-flow condition.

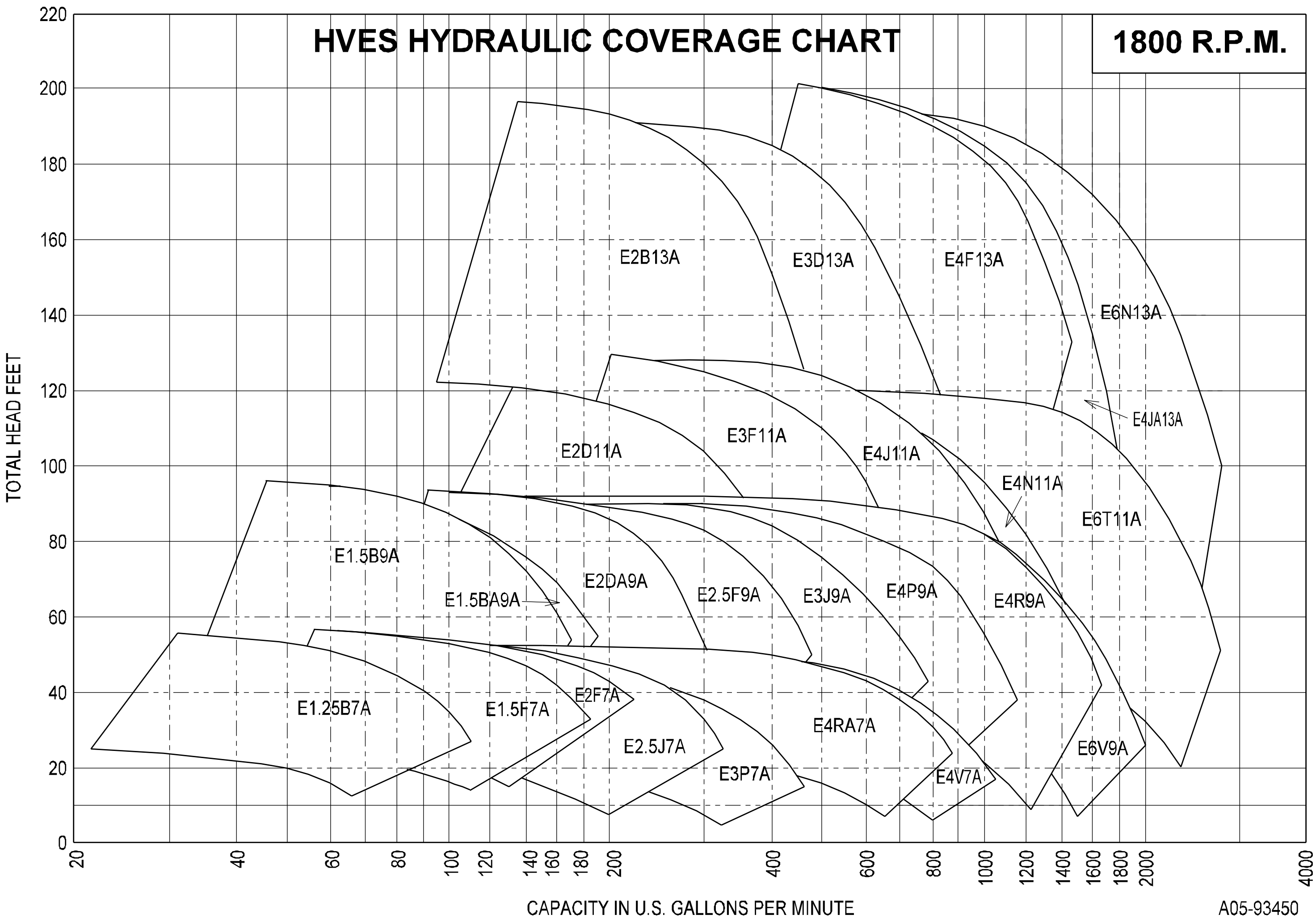


FIGURE 5-3. Family of pump curves.

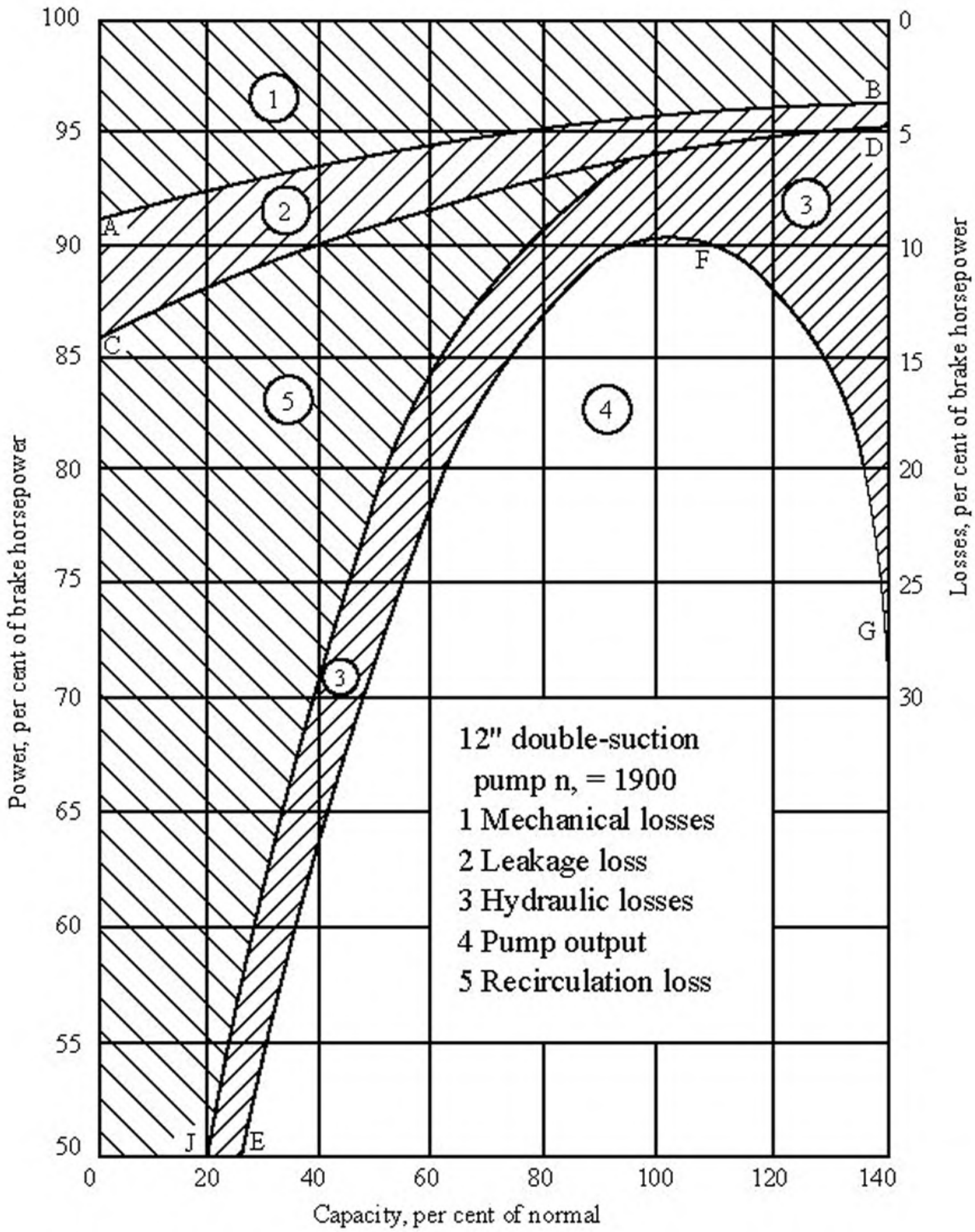
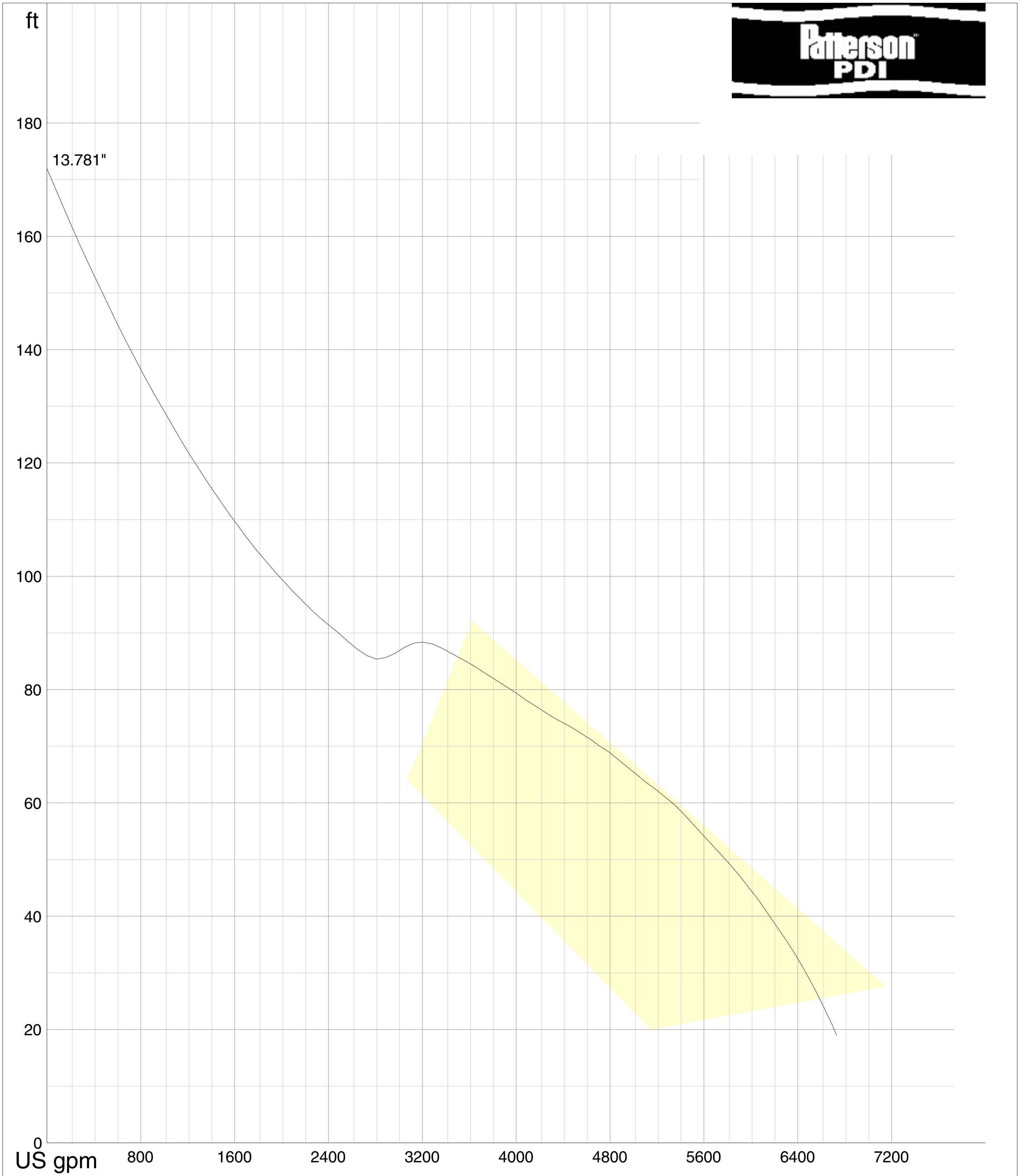


Figure 5-4 Power balance at constant speed. (From Centrifugal and Axial Flow Pumps, "A.J.Stepanoff, Ph.D, John Wiley and sons, New York, NY 1957.)



Curve No.: 12SAF-7-B

Patterson Pump Company	Patterson 60 Hz Vertical Pumps	Size: 12 SAFV-7-B
	Catalog: Patterson Pumps Vertical Turbine	Speed: 1770 rpm
07/19/04	A&MFlow - 1800	Dia: 13.781 in
.ufs		

FIGURE 5-5a

High specific speed pumps like these require care in their application, particularly for variable speed installations. Only an experienced pump engineer should develop these pump selections. Also, only the right-hand part of these curves is used for pump selection, Figure 5-5b.

Figure 5.1 has another set of curves that helps determine the value of variable speed pumps, and that is the parabolic path of the pump efficiency curves, as the speed is reduced to zero. This is one of the most misunderstood facts about variable speed operation. The peak efficiency (BEP) shifts to the left, as the speed is reduced.

PUMP SPEEDS:

There are various definitions for pump speed that should be explained.

Specific Speed:

The description of centrifugal pumps is difficult without knowing how a pump designer does his work. It is hoped that the following will help provide an understanding of the process leading to the design of a successful pump model.

The experienced pump designer uses a formula called specific speed, N_s to develop a pump impeller for a specific range of flows and heads, knowing that a certain specific speed will produce the desirable ratio between pump head and pump flow. This is a dimensionless number. The specific speed for a pump is calculated at the best efficiency point (BEP) for each pump. The experienced water system designer will develop a relationship of specific speed to the optimum pump selections for certain types of HVAC water systems.

$$\mathbf{N_s = \frac{S \cdot Q^{0.5}}{h^{0.75}}} \qquad \mathbf{(5.2)}$$

Where: S = Pump speed in rpm

Q = Pump flow in gpm

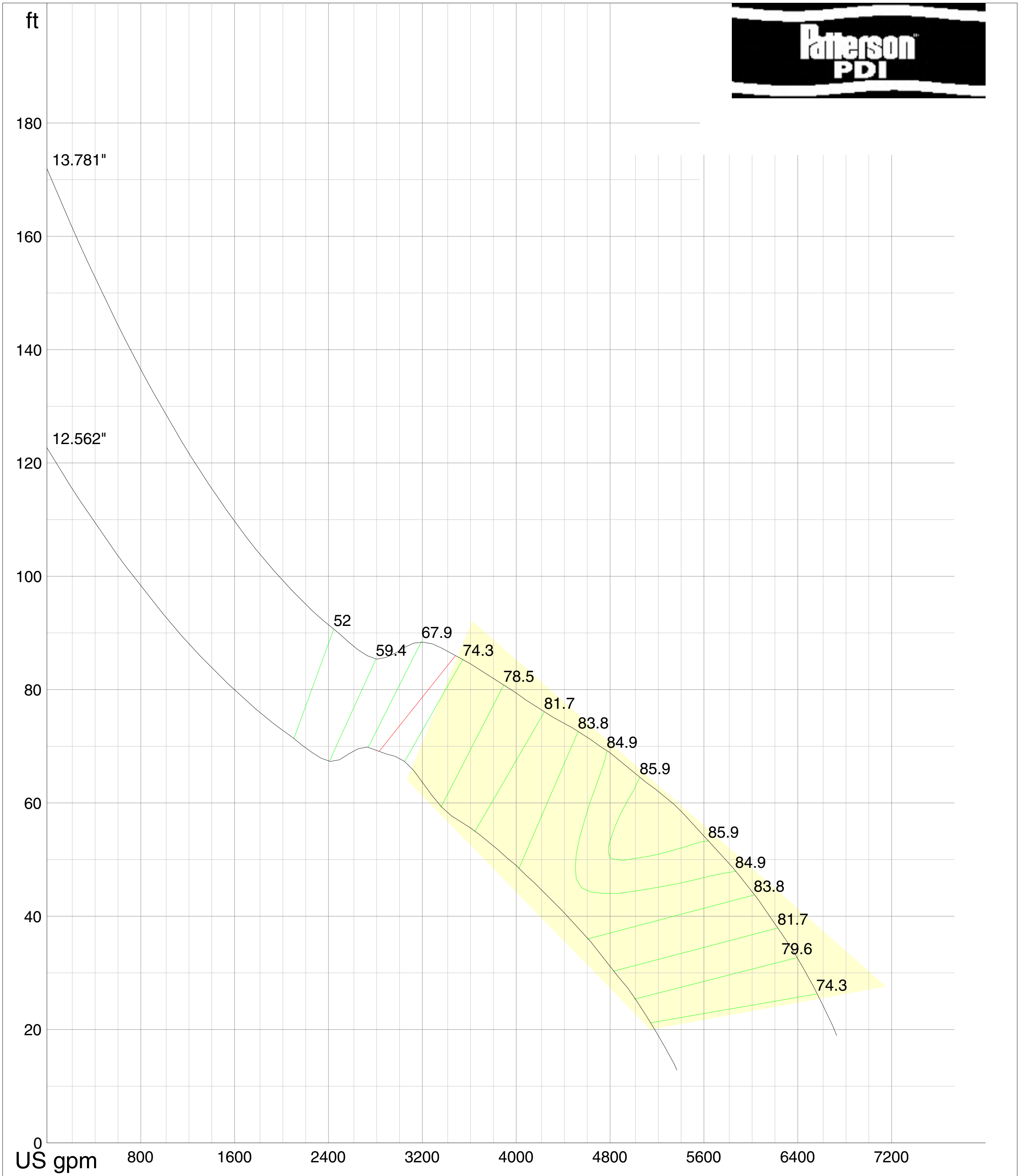
h = Pump head, ft

For example, assume that a pump has a capacity of 2,000 gpm at 120 ft. head and is rotating at 1,780 rpm:

The specific speed, $N_s = \dfrac{1{,}780 \cdot 2000^{0.5}}{120^{3/4}} = 2196$

With the aid of this formula and a great amount of experience and testing, the pump designer can develop the above-mentioned family of pump curves.

Specific speeds from 400 to 2,500 are used in most HVAC pumps. Diffuser



Curve No.: 12SAF-7-B

Patterson Pump Company	Patterson 60 Hz Vertical Pumps	Size: 12 SAFV-7-B
	Catalog: Patterson Pumps Vertical Turbine	Speed: 1770 rpm
07/19/04	A&MFlow - 1800	Dia: 13.781 in
.ufs		

FIGURE 5-5b

pumps for large cooling towers will have higher specific speeds. This is interesting background information for the HVAC system designer, but it is not used in the design of these water systems. It does help this designer to understand various pumps that may be applicable to a specific installation.

Critical Speed:

Every rotating assembly has a natural frequency at which vibrations become pronounced; they can increase until noise becomes objectionable, and there is danger in a pump that the rotating equipment will become damaged. The speed at which this occurs is called the **critical speed** of that rotating element. Some pumps will actually have more than one critical speed. Usually, the critical speed occurs at speeds beyond those in the normal range of operation. For example, a pump running at a normal maximum speed of 1780 rpm will have its critical speed at some speed such as 2,600 rpm. All pump designers must take critical speed into consideration in the structural design of their pumps, but it is of little concern to the HVAC system designer in the selection of centrifugal pumps.

Minimum Speed:

There is no minimum speed for pumps and their motors. Minimum speed is not a great concern in HVAC systems since there always is a factor about the system design that caused the pumps to run most of the time above 30% of design speed. Maintaining a minimum differential pressure at the ends of the loops in chilled and hot water systems requires around 30% speed. Variable primary pumping operations requires similar speeds to maintain minimum flow in the chiller evaporators.

AFFINITY LAWS OF PUMP OPERATION:

The basic laws that govern centrifugal pump performance with variable speed and impeller diameter are called the “affinity laws”. These are:

For a **Fixed Impeller Diameter:**

1. The pump flow varies directly with the speed.

$$\frac{Q_1}{Q_2} = \frac{S_1}{S_2} \qquad \textbf{(5.3)}$$

2. The pump head varies as the square of the speed.

$$\frac{h_1}{h_2} = \frac{S_1^2}{S_2^2} \qquad \textbf{(5.4)}$$

3. The pump brake horsepower required varies as the cube of the speed.

$$\frac{bhp_1}{bhp_2} = \frac{S_1^3}{S_2^3} \quad (5.5)$$

Where:

Q = Pump flow in gpm

S = Pump speed in rpm

h = Pump head in feet

bhp = Pump brake horsepower

Figure 5.6 describes these affinity laws for variation in pump speed. It must be reiterated that this describes the pump itself, not for a pump and the piping system that it serves. If there is static head in the water system that requires a head at the no-flow condition, the pump flow, head, and brake horsepower curves will change radically, Figure 5.7. This is a typical case for chilled and hot water, variable speed systems that utilize a remotely located differential pressure transmitter to control the speed of the pumps. In this case, the remote differential pressure was established at 24 feet, and it was maintained at all times regardless of the flow in the system.

For a **Constant Pump Speed:**

1. The pump flow varies directly with the impeller diameter

$$\frac{Q_1}{Q_2} = \frac{D_1}{D_2} \quad (5.6)$$

2. The pump head varies as the square of the impeller diameter

$$\frac{h_1}{h_2} = \frac{D_1^2}{D_2^2} \quad (5.7)$$

3. The pump brake horsepower required varies as the cube of the impeller diameter.

$$\frac{bhp_1}{bhp_2} = \frac{D_1^3}{D_2^3} \quad (5.8)$$

Where:

D = Pump impeller diameter in Inches

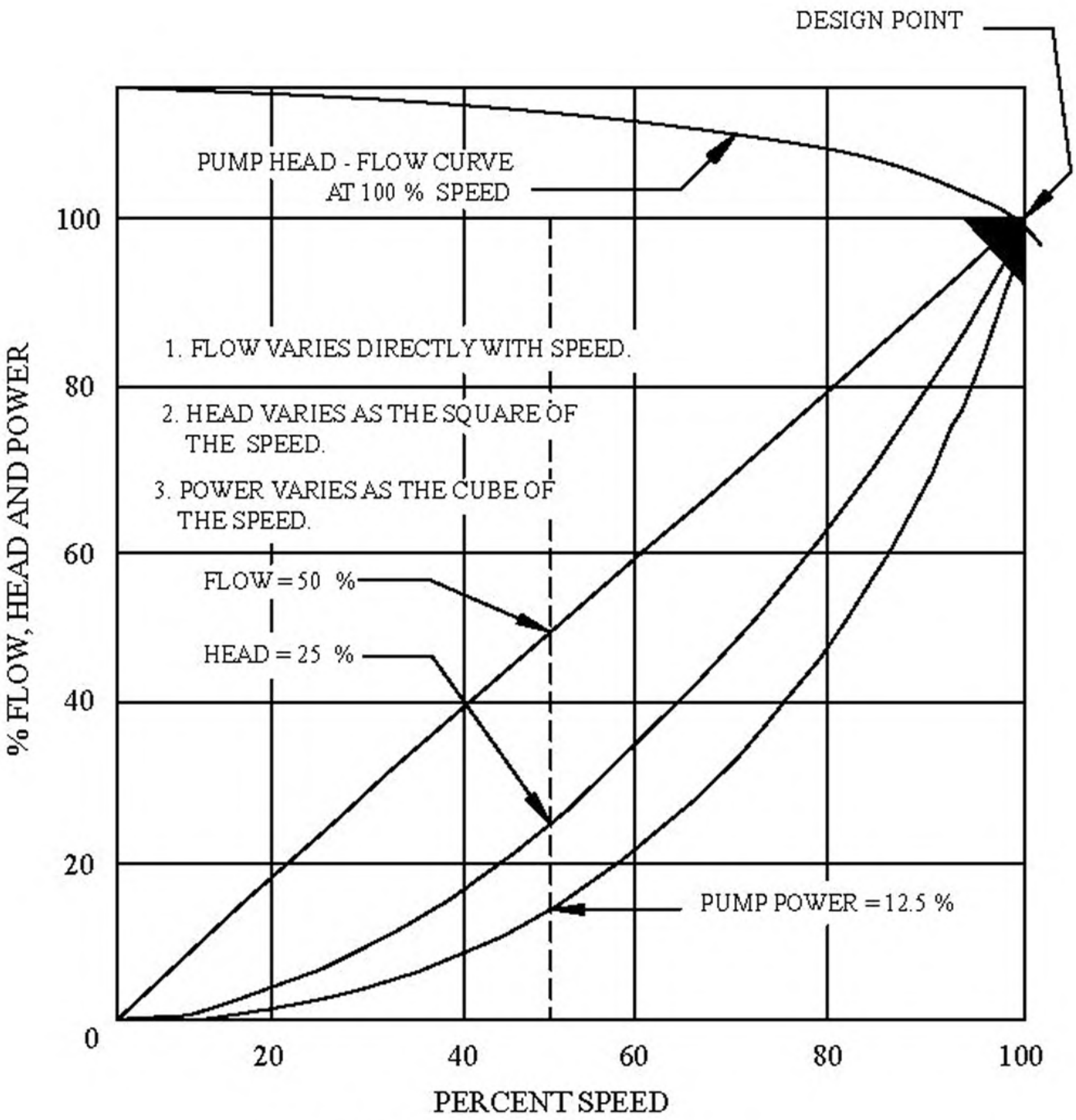
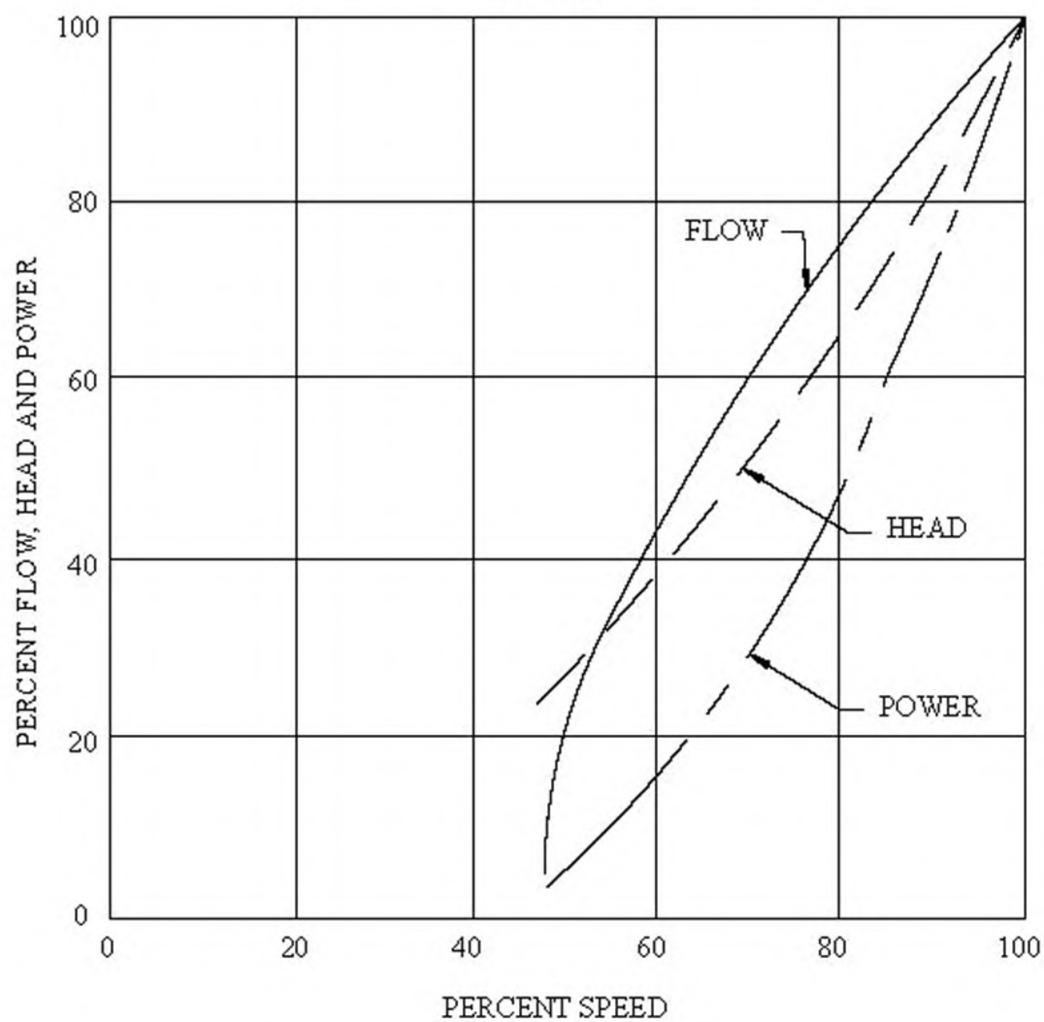


FIGURE 5- 6 Classical affinity law curves.

From: "HVAC Pump Handbook", James Rishel, PE, McGraw-Hill, New York, 1996.

FIGURE 5-7

AFFINITY LAWS FOR A PUMP
OPERATING WITH STATIC HEAD
OF 24 FT.



The pump manufacturer will show the allowable change in pump diameter (see Figure 5-2) for each pump impeller. The laws for the variation of pump performance with impeller diameter are for constant specific speed. In some impellers, the specific speed changes from maximum allowable to minimum allowable pump diameter.

MINIMUM FLOW IN A PUMP:

At design speed such as 1,780 rpm, the pump manufacturer will specify **a minimum flow.** The pump has been designed to accommodate a certain radial thrust throughout its allowable operating flow range (OFR). Operating the pump at lower flows may exceed this allowable radial thrust. (See Figure 4-13).

At reduced speeds, radial thrust may not be a problem, but temperature will be if the pump operates continuously at a low flow condition that causes heating in the pump.

The question often arises as to the minimum flow that is required through a pump to prevent it from overheating. The following equation, 5.9, provides this flow.

$$\textbf{Minimum Flow, } Q_M = \frac{bhp \cdot 2544}{\Delta T \cdot 8.33 \cdot 60}$$

$$= \frac{5.1 \cdot bhp}{\Delta T} \qquad (5.9)$$

Where: bhp is the brake horsepower of the pump near the shutoff or no flow condition.

ΔT is the allowable rise in temperature in °F of the water in the pump.

This equation is hard to use since it is difficult to acquire the actual brake horsepower consumed by the pump at very low flows. A pump company may publish pump brake horsepower curves down to zero flow or just specify a minimum flow for a particular pump. Other equations that use pump efficiency have the same problem, since the efficiency of a centrifugal pump approaches zero at very low flows. If this information is not available, the pump company should provide the minimum flow that will hold the temperature rise to the desired maximum. The obvious advantage of the variable speed pump over the constant speed pump appears here, since there is much less energy imparted to the water at minimum flow and speeds.

For most HVAC water applications, there should not be temperature rises in a pump higher than 10°F. **Do not bypass water to a pump suction! This continues to elevate the suction temperature and reduce the NPSHA.**

ENERGY CONSUMPTION OF AN HVAC PUMP:

Since most HVAC pumps are motor driven, this discussion will be limited to the energy consumed by such pumps. It is important to understand that the energy required by a water system is the **water horsepower,** and the energy required by a pump is the **brake horsepower.**

Water horsepower is denoted by "whp", and its equation is:

$$\mathbf{whp} = \frac{\mathbf{Q \cdot h \cdot s}}{\mathbf{3{,}960}} \qquad \mathbf{(5.10)}$$

Where:

Q = Flow in gpm

h = Head in Feet

s = Specific Gravity

3,960 = 33,000 ft. lb/minute ÷ 8.33 lb. per gallon

Excepting for glycol solutions and high temperature water, all HVAC systems are assumed to contain water with a specific gravity of one, so specific gravity is not shown in most energy calculations.

Following is an example of water horsepower:

If a pump is delivering 2,000 gpm at a total head of 120 ft, the water horsepower is:

$$\text{whp} = \frac{2{,}000 \cdot 120}{3{,}960} = 60.6$$

Pump horsepower, "bhp", is the water horsepower divided by the efficiency of the pump. The pump brake horsepower equation is therefore:

$$\mathbf{bhp} = \frac{\mathbf{whp}}{\boldsymbol{\eta_P}} \qquad \mathbf{(5.11)}$$

Where $\boldsymbol{\eta_P}$ = Pump efficiency as a decimal

For example:

If the pump in the above example is operating at an efficiency of 85%:

$$\text{bhp} = \frac{60.6}{0.85} = 71.3 \text{ hp}$$

For **constant speed pumps,** the energy consumption of the pump and electric motor is the input kW to the motor and can be called the “pkW” or just “kW input”. For constant speed motors, the pkW is merely the pump bhp divided by the motor efficiency and multiplied by the kW equivalent of a bhp, 0.746, and the equation is:

$$\mathbf{pkW = \frac{bhp \cdot 0.746}{\eta_E}} \qquad \textbf{(5.12)}$$

Where:

One kW = 0.746 hp

η_E = motor efficiency as a decimal

For variable speed pumps, the wire-to-shaft efficiency, η_{WS}, of the motor and drive must be determined; it is often difficult to secure accurate information on it. The drive manufacturer should provide this efficiency once the motor size, type, and manufacturer are determined.

$$\mathbf{pkW = \frac{bhp \cdot 0.746}{\eta_{WS}}} \qquad \textbf{(5.13)}$$

Where: η_{WS} is the wire-to-shaft efficiency of the motor and variable speed drive as a decimal.

For example, If the above pump is variable speed with a wire-to-shaft efficiency of 91% for the variable speed drive and high efficiency motor:

$$pkW = \frac{71.3 \times 0.746}{0.91} = 58.5 \text{ kW}$$

Combining the above equations:

$$\mathbf{pkW = \frac{Q \cdot h \cdot s}{5308 \cdot \eta_P \cdot \eta_E \text{ or } \eta_{WS}}} \qquad \textbf{(5.14)}$$

SERIES/PARALLEL PUMP OPERATION:

Almost all HVAC pump installations are with multiple pumps operating in parallel. Seldom are the pumps at one installation installed in series. Therefore, the head of the pump installation is that of any one pump at the design condition, and the flow is the sum of all of the pumps in operation. Also, multiple pumps at one installation are usually of the same size. With variable speed, so-called jockey pumps are not required to handle light loads. If the system operates for extended hours at minimum load, consideration should be given to determine the energy savings that could be achieved by installing a small pump for this situation. There are multiple pump installations that operate in series. (See Figure 6-21)

If there is a need for pumps of different sizes operating in parallel, it is imperative that all of the pumps have approximately the same shutoff head. Otherwise, one of the pumps may be driven into a no-flow condition at reduced speeds.

ENERGY CONSUMPTION OF HVAC PUMPS IN PARALLEL:

Almost all HVAC pump installations have two or more pumps operating in parallel to provide standby capacity and to account for the broad range of flows in these systems. As many as eight pumps operate in parallel on large systems. It is important to understand how the combined efficiency for all of the pumps is determined for these pumping systems. First, we should go back to Figure 5-1 to remember how a pump changes its head-flow curve as its speed is reduced. The best efficiency point (BEP) follows a parabolic curve to zero.

Recognizing this fact, the calculation of the energy consumption of a multiple pump system must also take in consideration 1) the friction loss through the pumping system, 2) the friction loss across a boiler, chiller or heat exchanger, 3) the friction loss in the water system, and 4) the differential pressure that is maintained in most HVAC variable speed pumping systems. Figure 5-8 describes these four friction losses that make up the design pump head.

Along with these losses, the efficiency of the pump and the wire-to-shaft efficiency of the motor and the variable speed drive must be determined, as the operating pumps vary their speed. The result of these calculations is a **kW Input Program** that computes the energy consumption from minimum flow with one pump operating to the design condition with the optimum number of pumps in operation. This will be explained in detail in Chapter 8 on Operating HVAC Pumps.

Pump Selection and Number of Pumps:

The actual selection and number of pumps for a specific application will be discussed in Chapter 8, *Operating HVAC Pumps*.

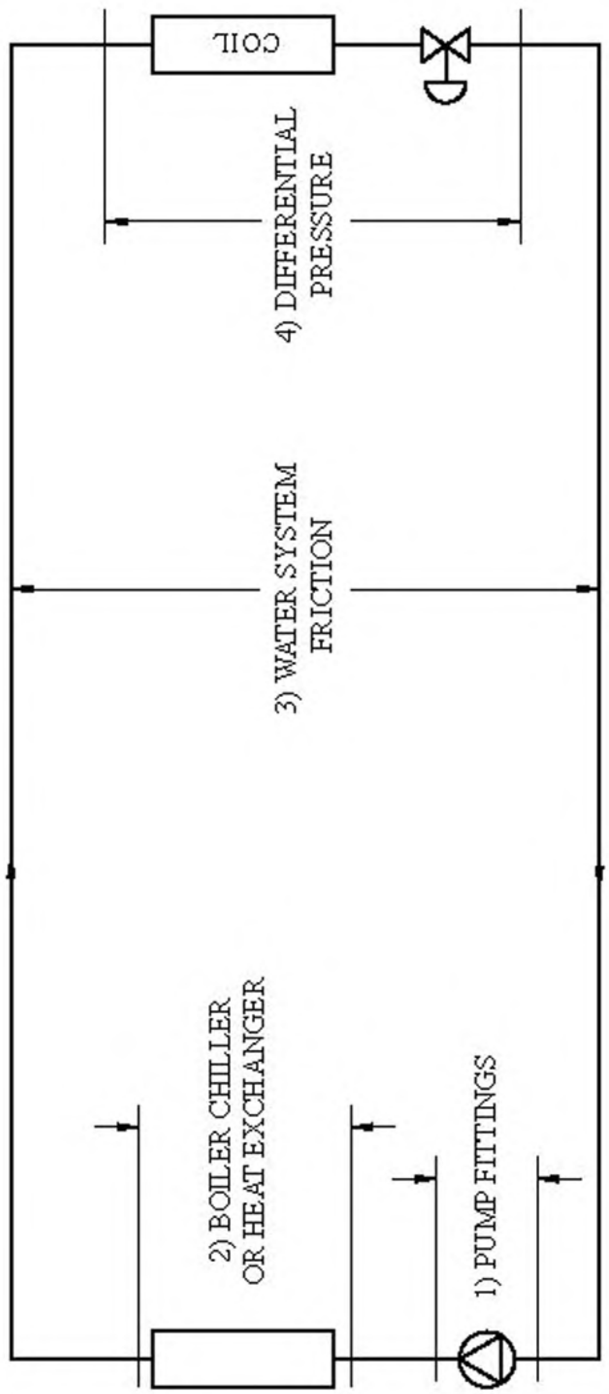
COMMON PUMP TERMS:

The words, **pump duty**, are often used when describing a pump's performance. Pump duty is normally defined as a certain flow in gpm at a specified head in feet. This is the point at which the pump has been selected for a particular application. Usually, this point is selected as closely as possible to the best efficiency of a pump impeller, which, for Figure 5-2, is 88%.

Other common terms that are encountered in the pump industry are:

1. **"Carry out" or "run out"**: This term is for a pump that is operating at the far right of its curve with poor efficiency.

FIGURE 5-8
FOUR FRICTION LOSSES
HVAC WATER SYSTEM



2. **“Shut off head”**: The head produced by a pump when it is running at the “no flow” or zero capacity point.

3. **“Churn”**: A pump is in churn when it is operating at shut off head or no flow condition.

4. **“Clear Service”**. These are clean water pumps as are most HVAC pumps.

5. **“Preferred Operating Region” (POR):** Hydraulic Institute has established the Preferred Operating Range for most HVAC pumps to be from 70 to 120% of the flow at the Best Efficiency Point. This is for operation at full speed such as 1,780 rpm.

6. **“Allowable Operating Region” (AOR):** A pump manufacturer provides an allowable operating region that may be greater than the POR. This is based upon the actual conditions of service.

This concludes the basic performance of HVAC pumps. Additional information on pump operation will be included in Chapters 6, 7, and 8.

Chapter 6

HVAC WATER SYSTEM DESIGN

HVAC PUMPS AND SYSTEMS
DESIGN MANUAL

PATTERSON PUMP COMPANY

INTRODUCTION:

The past twenty years have brought great changes in the design of HVAC water systems. Most of them are due to the advent of the variable speed drive and the development of digital control. We now can make an accurate accounting for the use of pump head in these water systems and eliminate many of the old energy-wasting devices that were used with constant speed pumps. The result is simple piping systems with very low energy consumption. This chapter will describe system head curves and how we control the use of pump head and eliminate energy waste.

SYSTEM HEAD CURVES:

The system head curves for HVAC water systems are similar to those for other water systems in that they have both constant head and variable head as shown in Figure 6-1. Most HVAC water systems are closed, meaning that the water is returned to its source such as a chiller, cooling tower, or boiler. The ***constant*** head consists of 1) the constant pressure maintained in a chilled or hot water distribution system or 2) the lift over a cooling tower. There will be a number of figures in this manual describing both conditions. The system head curve is generated by plotting the design flow for the entire water system on the "x" axis and the design system head on the "y" axis.

Unfortunately, most of the HVAC industry has ignored the fact that many water systems do not follow a simple system head curve due to the variation in the actual loads on these systems. Sometimes, the cooling or heating loads near the pumps are active while at other times, the loads far from the pumps are at 100% of design. The result is a system head area as shown in Figure 6-2. In many of the diagrams that follow, it will be apparent the actual system friction will be dependent upon the location of the active loads, whether they are near or far from the pumping system... This system head area does not affect the selection of pumps for such a system, but it is imperative that the pump control is designed to accommodate it.

Another fact that is often ignored is the effect of the pump fittings on the system head curve. These fittings include suction strainers, suction and discharge shut-off valves, discharge check valve, piping, and fittings. Well designed HVAC

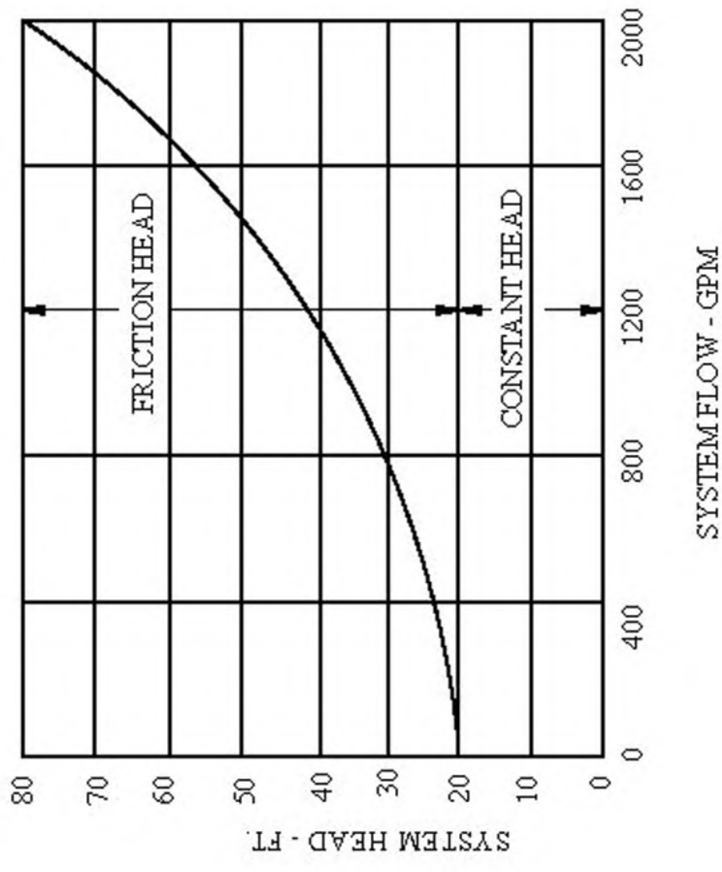


FIGURE 6-1
UNIFORM SYSTEM HEAD CURVE
WITHOUT PUMP FITTING EFFECT

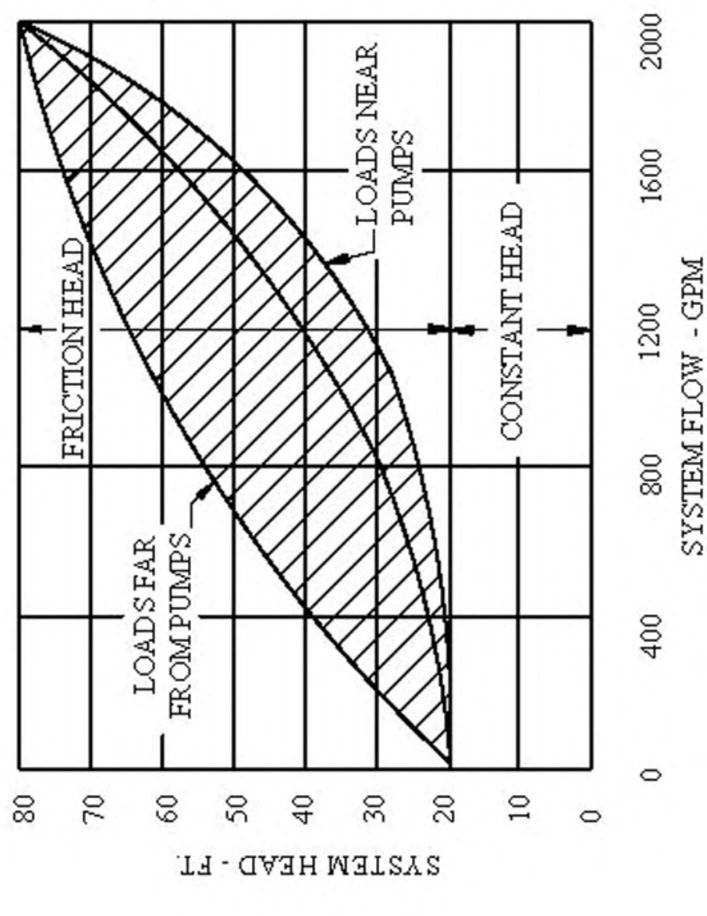


FIGURE 6-2
SYSTEM HEAD AREA

systems have pump fitting friction losses of 6 to 8 ft. at the pump design condition. Poorly designed systems may have losses as high as 15 ft. Most HVAC water systems have pumps operating in parallel. Therefore, the losses through the fittings of an individual pump are determined by the flow through that pump, not the total system flow. The result is a jagged system head curve, Figure 6-3. Any calculation for the overall efficiency of a pumping system must take in account the effect of the pump fittings on the system curve and the energy consumption.

Figure 6-3 also demonstrates some of the problems involved in sequencing pumps in parallel. This figure includes the parabolic paths of best efficiency for each number of operating pumps as pump speed is reduced. When adding and subtracting variable speed pumps operating in parallel, this figure shows that the best efficiency of the pumps as well as the friction loss through the pump fittings must be taken into consideration. Figure 6-4 is a larger scale version of part of Figure 6-3. The points 1 and 2 describe the transition points which should be equidistant from the two parabolic paths of best efficiency. This is one of the most important figures in this manual. It will be used in several places to illustrate parallel pump performance. A pump sequencing program based upon this figure will be included in Chapter 8.

CONTROL OF THE USE OF PUMP HEAD:

It was shown in Chapter 5 that pump energy is directly dependent upon the flow and head of a water system. We should, therefore, reduce the flow and head of an HVAC water system wherever possible to lower pump energy. The flow is determined by the maximum heating or cooling load on the system. The pump head on the system is more dependent on the actual design of the piping system and the means used to distribute the heating or cooling. The use of variable speed drives on the pumps enabled us to develop very simple water systems devoid of devices that destroy pump head. The hydraulic gradient has given us a tool that enables us to check the calculations of the pump head for a particular water system.

HYDRAULIC GRADIENTS:

The hydraulic gradient is a graphical representation of the use of pump head in a water system. It differs from the energy gradient in that the velocity head of the water system, $v^2/2g$, is not included. The velocity head is a small part of the total head and does not vary much throughout the water system. The hydraulic gradient does include the pump head, friction losses, and changes in the static head. All of the following diagrams, excepting the one for a high rise building, will assume that the water systems are operating at a constant elevation.

Before evaluating hydraulic gradients, it is important to remember that air and, therefore, oxygen retention in water increases with the pressure of that water

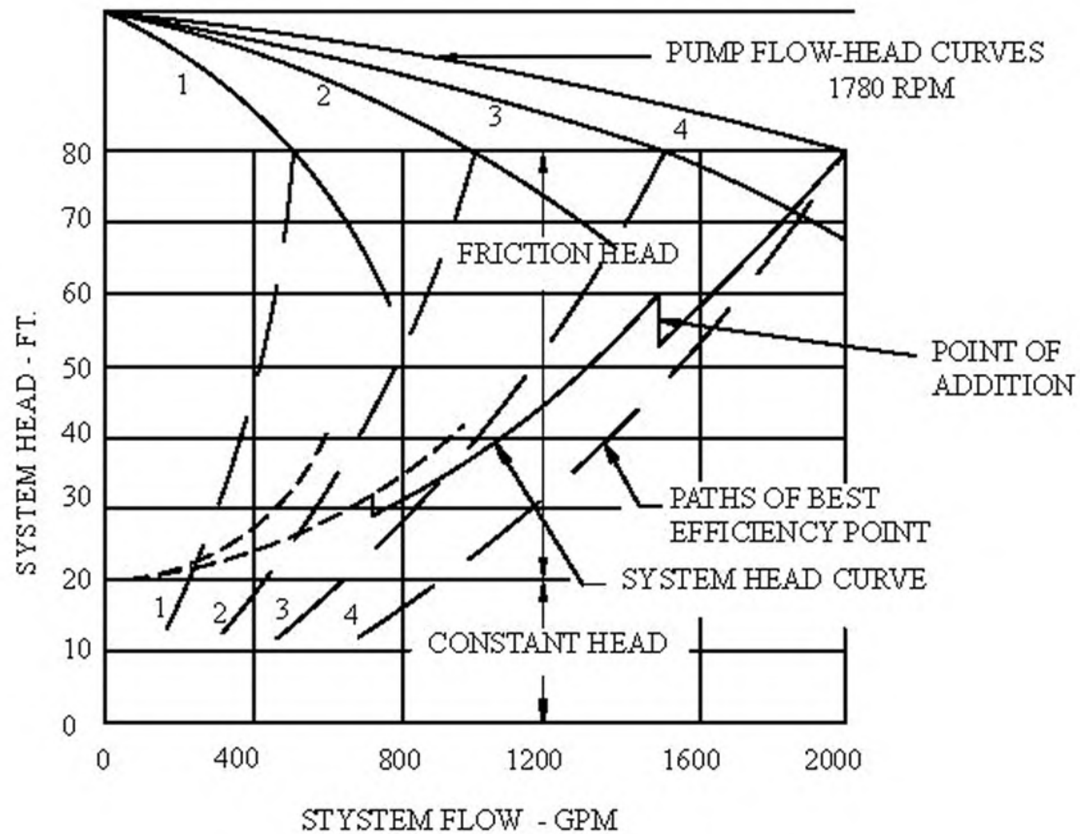


FIGURE 6-3

UNIFORM SYSTEM HEAD CURVE
WITH PUMP FITTING EFFECT

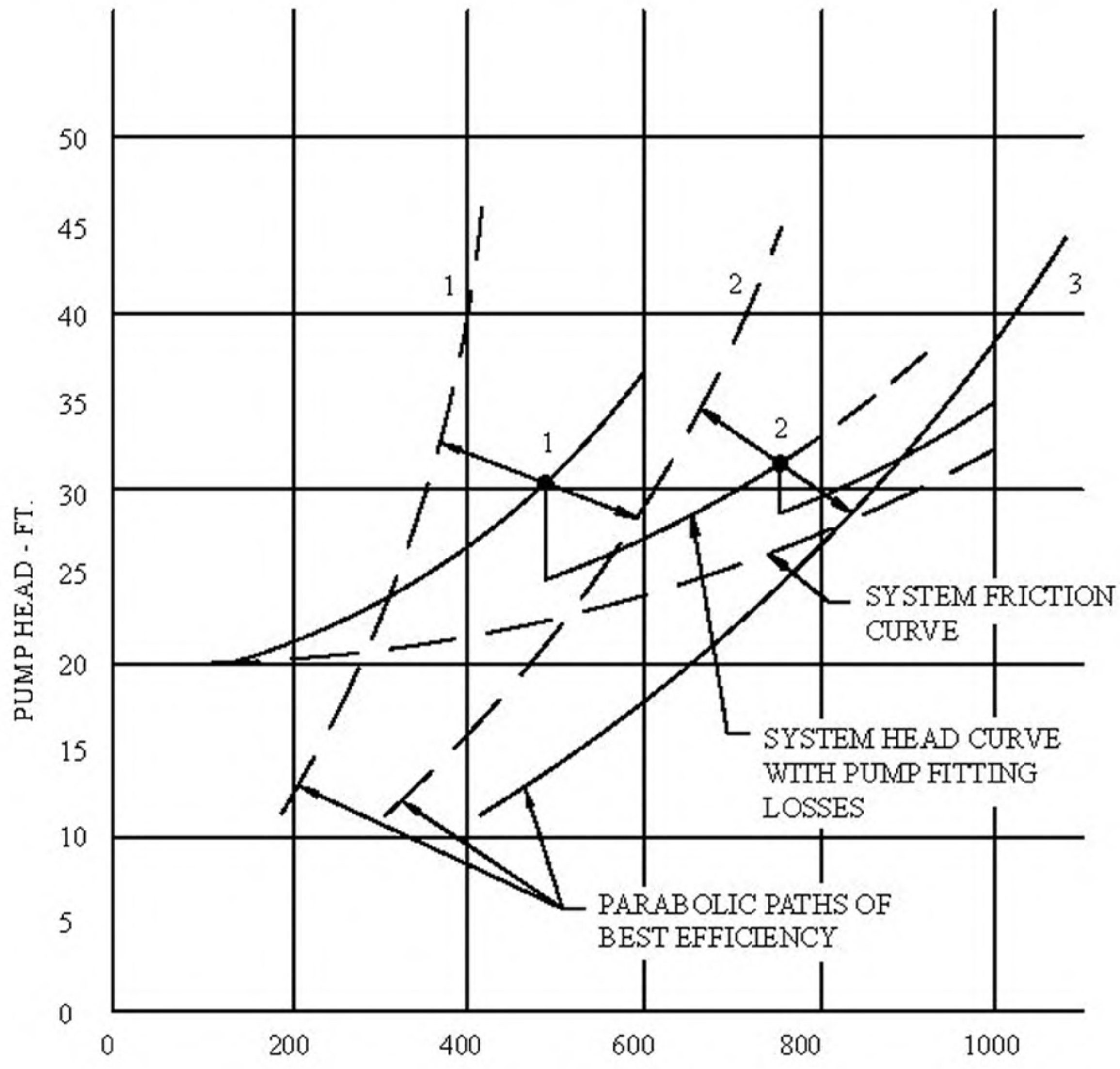


FIGURE 6-4
POINTS OF TRANSITION

system (see Figure 2-3). Therefore, all water system design should be done recognizing the value of achieving the minimum possible pressure in that system

Figure 6-5 describes a simple variable primary chilled water system while Figure 6-6 provides the hydraulic gradient for this water system. Pump head is always shown vertically upward while pressure losses can be vertically downward or diagonally. The vertical or "y" axis is scaled in feet of head while the horizontal or "x" axis generally has no scale. On large campus systems, the "x" axis can be in hundreds of feet.

The construction of the diagram begins by establishing the expansion tank pressure or static pressure; in Figure 6-6, this is 30 psig or 69 ft. The static pressure in feet is determined by selecting the heating or cooling coil that is at the highest location in the HVAC water system. The difference in elevation between this coil and the operating floor in the central plant should be determined, and a safety factor of around 10 psig or 23 feet should be added to it. If the height of the coil above the central floor is 100 feet, then 23 feet should be added to it for a total of 123 feet or 53 psig. The expansion tank, if it is located at the pump suction, should be maintained at 53 to 55 psig.

Next, the pump suction losses of 4 ft. are drawn downward diagonally followed by the pump head of 90 ft. head that is drawn vertically. The various losses are drawn downward or diagonally until the diagram closes at the expansion tank or fill pressure. Remember, these diagrams may look complicated, but really, they are quite simple.

Horizontal Building Gradient:

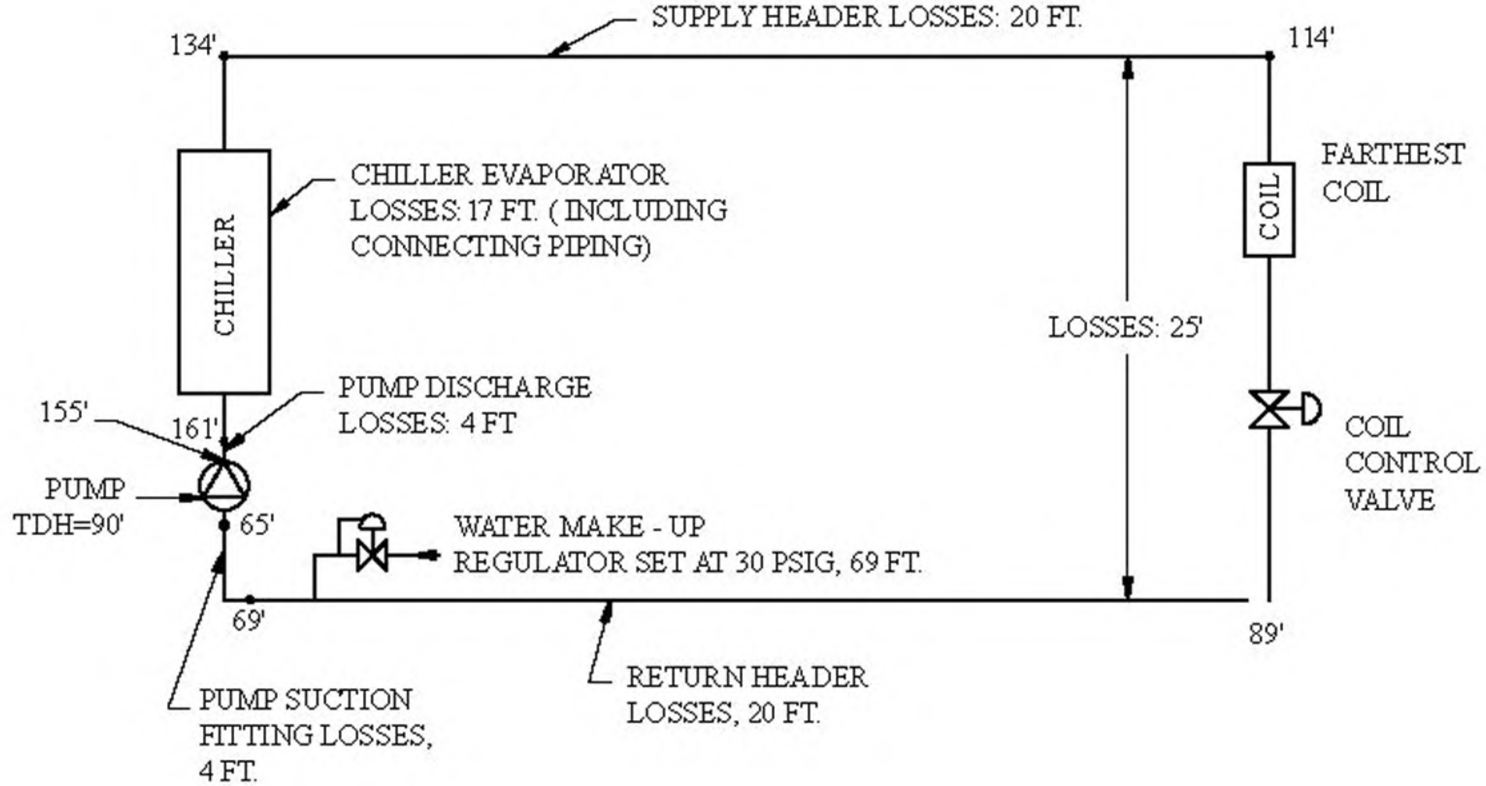
A horizontal building has a hydraulic gradient that looks much like the previous figures. The shut-off head of the pumps has not been shown on these diagrams, since it is not a significant value. It is important on tall buildings where the working pressure of the chillers or pumps can be exceeded easily. **The shut-off head of the pumps, at maximum speed, plus the expansion tank or fill pressure determines the maximum pressure that can be imposed on a water system.**

Vertical Building Gradient:

Figure 6-7 describes a typical high rise building with the chiller in the basement. Note that the chilled water pump is on the discharge of the chiller. Figure 6-8 provides the hydraulic gradient for this system. The shut-off head of the pump results in a discharge pressure of 365 ft. or 158 psig. If the pump had been located on the suction of the chiller, high pressure water boxes would have been required for the chiller.

FIGURE 6-5

SIMPLE SYSTEM TO DEMONSTRATE HYDRAULIC GRADIENT



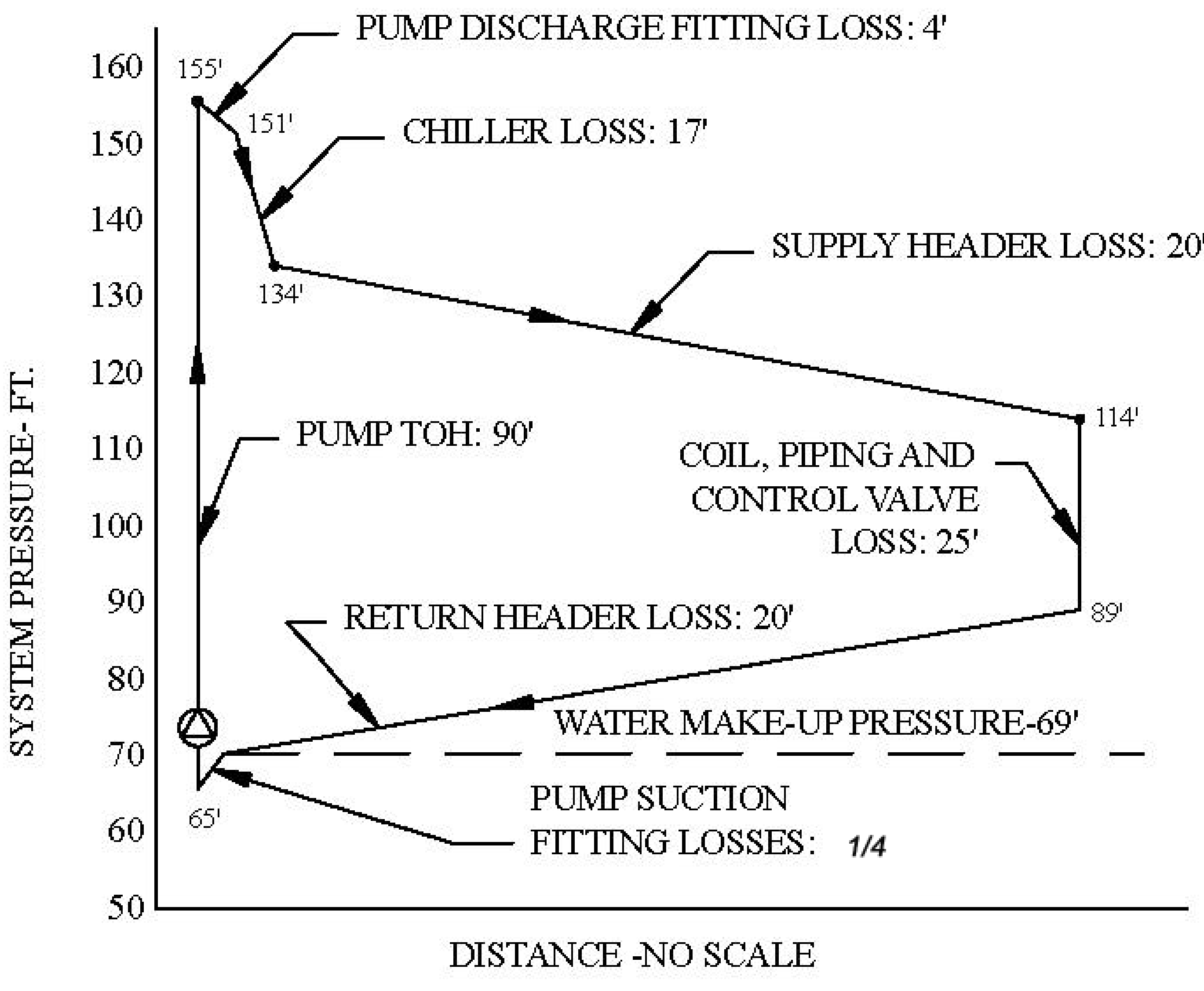


FIGURE 6-6
HYDRAULIC GRADIENT
FOR FIGURE 6-5

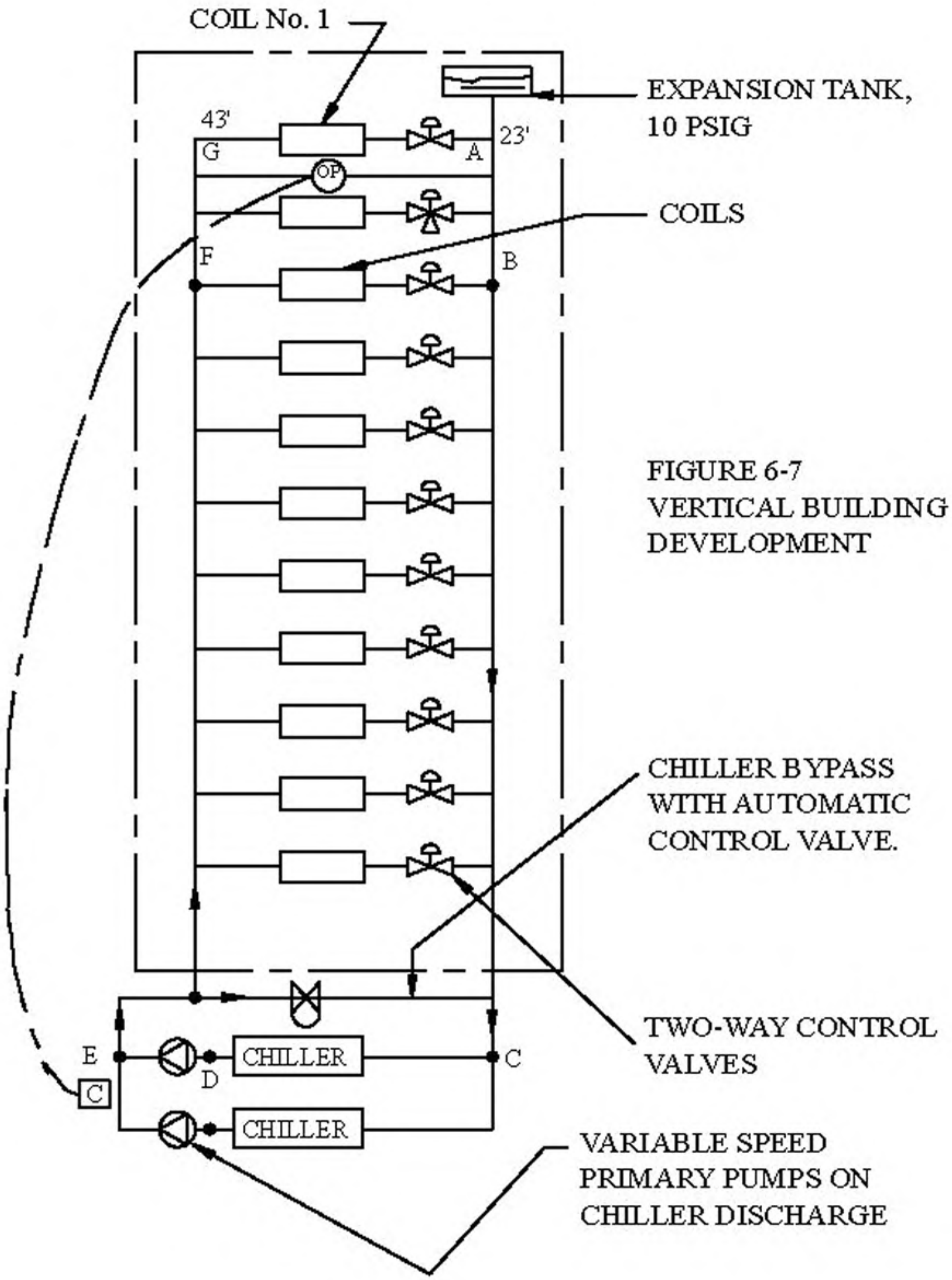
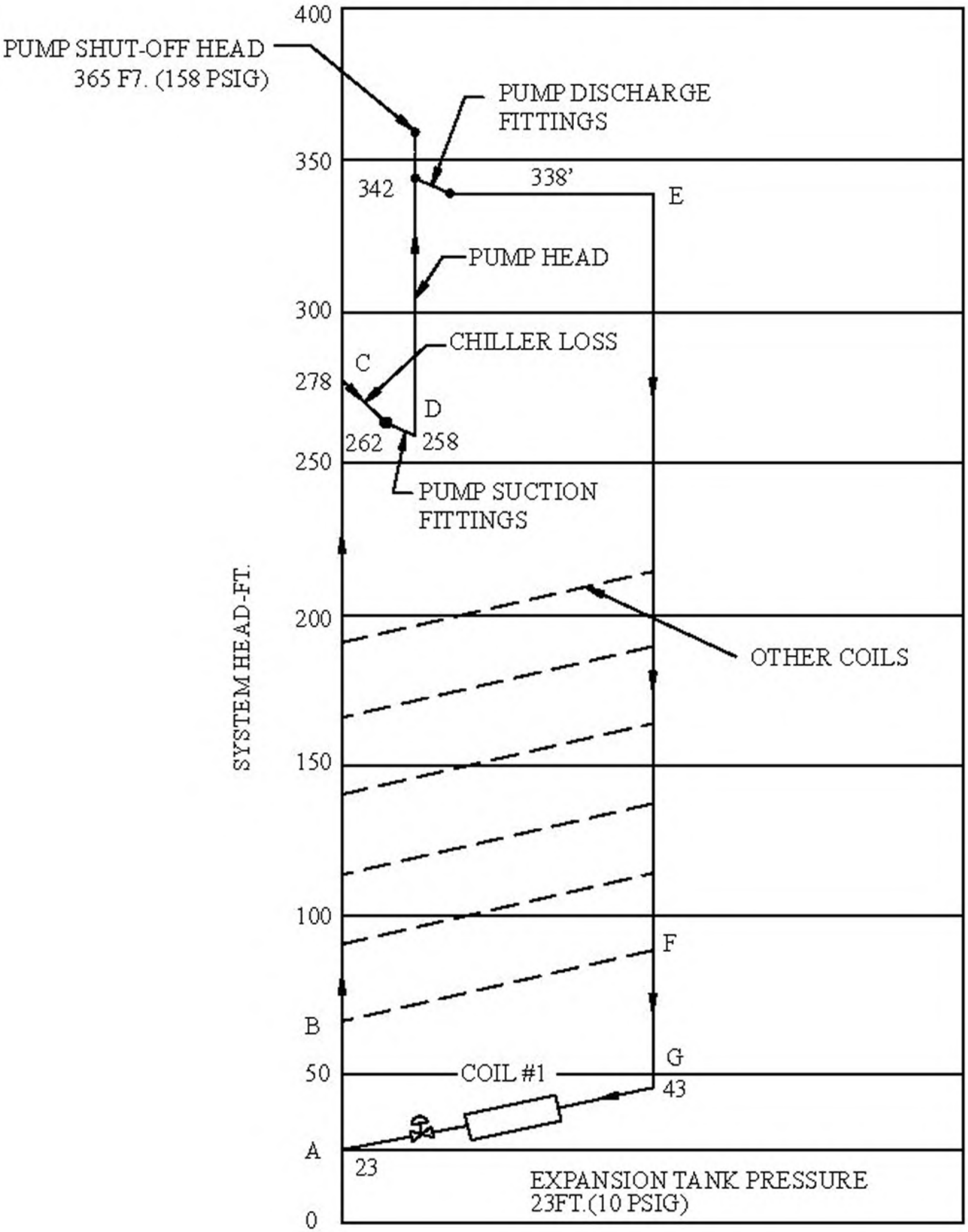


FIGURE 6-7
VERTICAL BUILDING
DEVELOPMENT

FIGURE 6-8
HYDRAULIC GRADIENT
VERTICAL DEVELOPMENT



Direct and Reverse Return Piping:

All of the water systems and diagrams shown above are called direct return systems. The water from each coil flows directly back to the chillers or boilers. In the earlier years of constant flow water systems when energy was relatively cheap, reverse return piping was used to balance the pressure losses across all of the cooling or heating coils. Figure 6-9 shows a reverse return diagram where the discharges of the coils are connected together reversely before returning to the chiller or boiler. Figure 6-10 is the hydraulic gradient for Figure 6-9 that indicates the same loss across all of the coils, and, therefore, the system was easy to balance. Reverse return piping is not used often today unless there is a large system with a great many fan coil units or water source heat pumps where it is imperative that the system distribution friction not be imposed upon the coil valves, Figure 6-11. Two differential pressure transmitters are needed to control such a distribution system.

A very good example of a hydraulic gradient for direct return piping is shown in Figure 6-12. Two cooling coils demonstrate the problem with mechanically balancing direct return systems through the use of balance valves. Coil No. 1 has no load on and yet has 96 ft of head on its control valve, while Coil No. 2 is fully loaded with 88.9 ft of head on its coil and control valve. This demonstrates the great amount of head that can be imposed upon the coil valves where the coils are located near the central plant. If you try to manually balance out this high head at full flow in the system, you will starve the coils when they operate at full load with reduced total load on the system. As for a variable volume system, there is no possible way to balance such a system with balance valves. This demonstrates that the coil control valve must be selected correctly to maintain the desired flow in the coils, whether there is near full load or little load on the chilled water system.

ELIMINATION OF PUMP HEAD-WASTING DEVICES:

Whenever we add a mechanical device to a water system, we should ask why do we need it and what energy does it cost in the form of lost pump head?

Balance Valves, manual and automatic:

As we saw above, variable volume water systems cannot be balancewith any kind of a balance valve. Since most, sizable HVAC water systems are variable volume with variable speed pumps, there is no reason to use any balance valves on them. There are thousands of efficient HVAC water systems in operation that have no balance valves whatsoever. Small constant volume water systems can be balanced using manual balance valves.

FIGURE 6-9

REVERSE RETURN SYSTEM

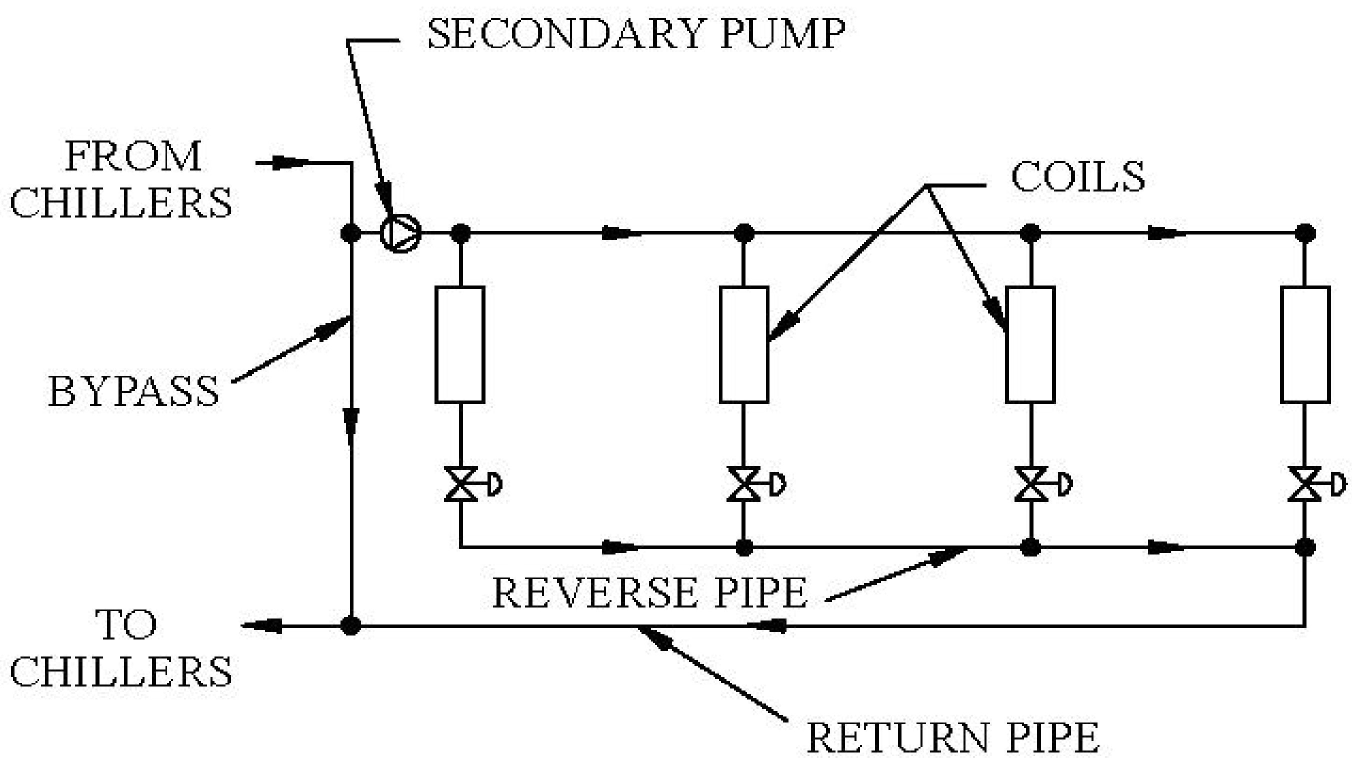
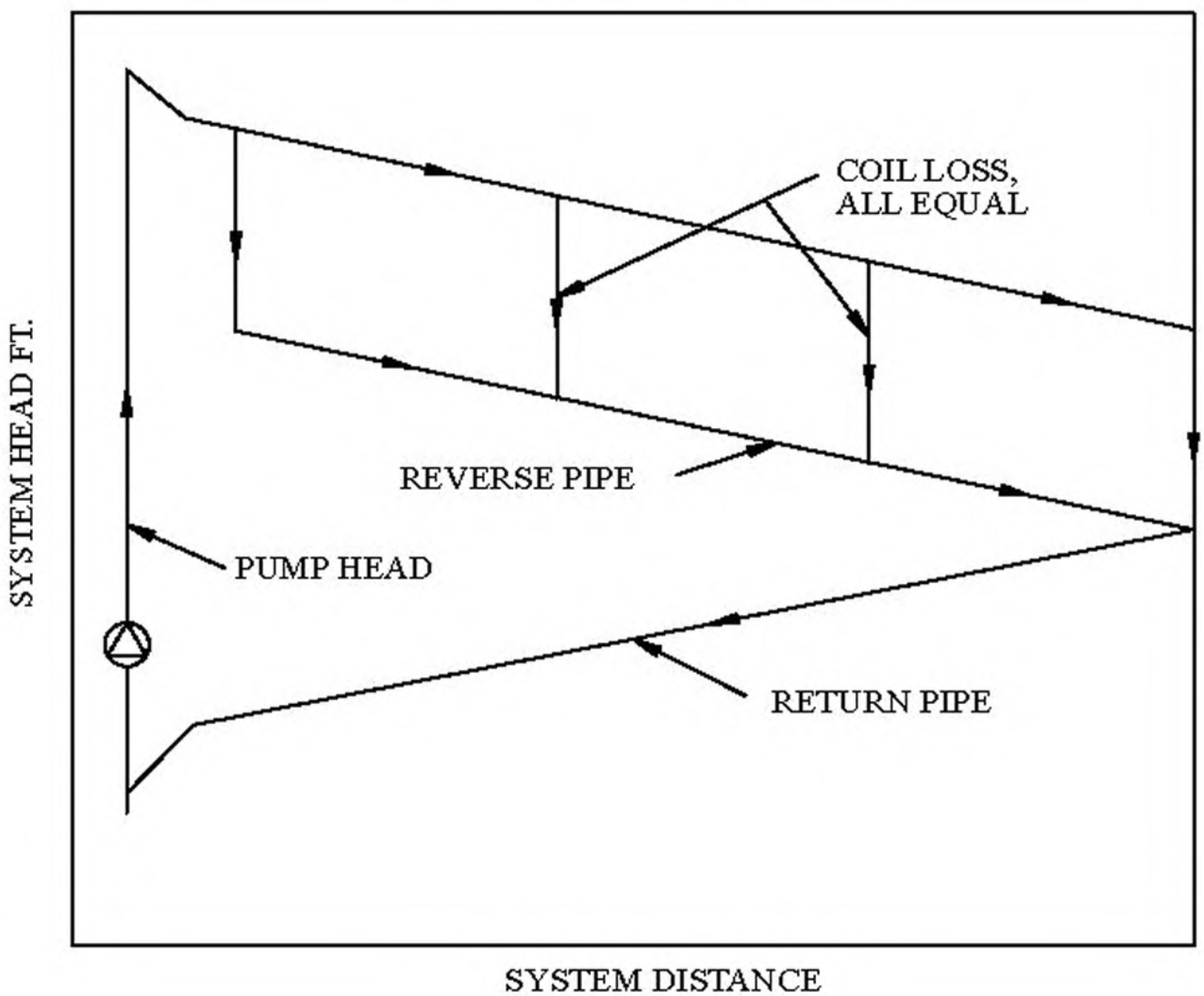


FIGURE 6-10

HYDRAULIC GRADIENT
FOR
REVERSE RETURN SYSTEM



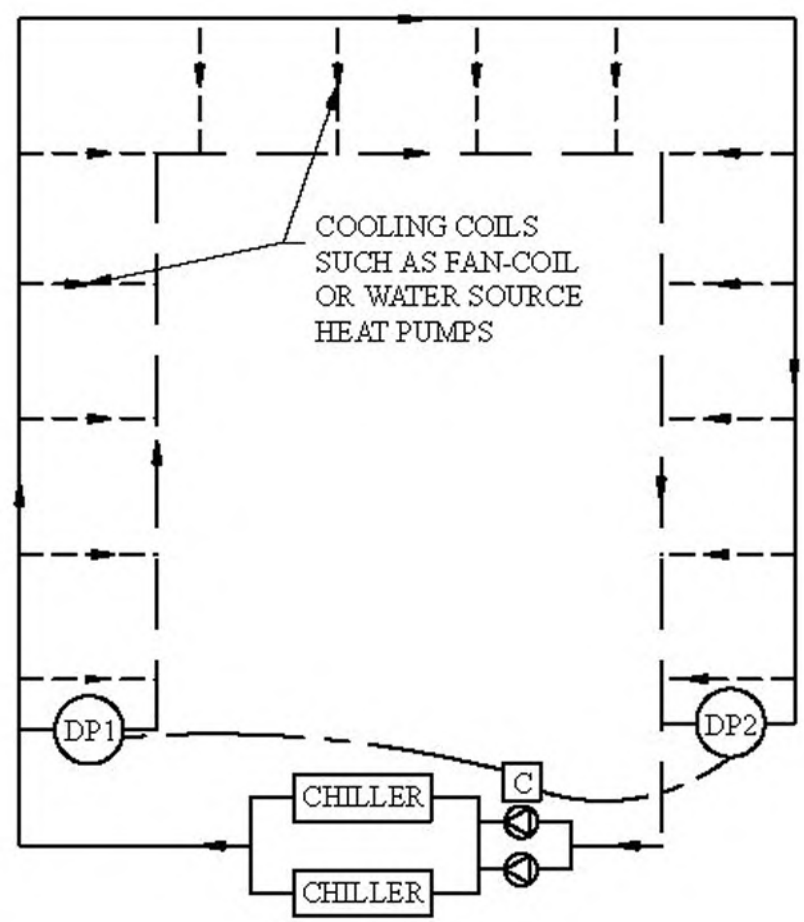
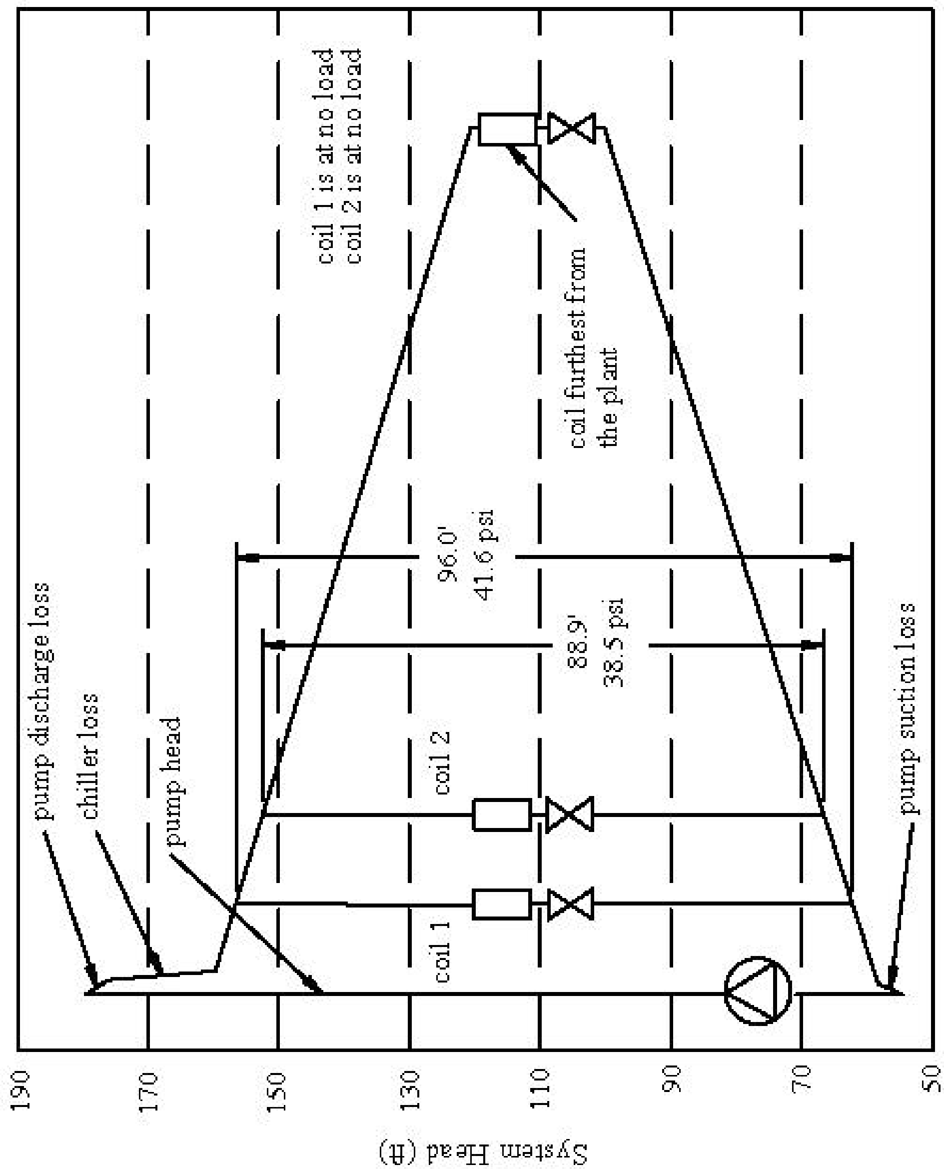


FIGURE 6-11

REVERSE RETURN
FOR LOW CONTROL VALVE
DIFFERENTIAL PRESSURES

FIGURE 6-12, HYDRAULIC GRADIENT FULL LOAD



Chilled Water Distribution System (no scale)

Three-Way Coil Control Valves:

These valves are a holdover from the days of constant volume water systems. They should be eliminated when these systems are converted to variable volume water systems. There are special and useful applications for three-way valves, such as in diverting water on cooling towers, changing water flows in chilled water and ice storage systems, and for blending water temperatures.

Multiple-Duty Valves:

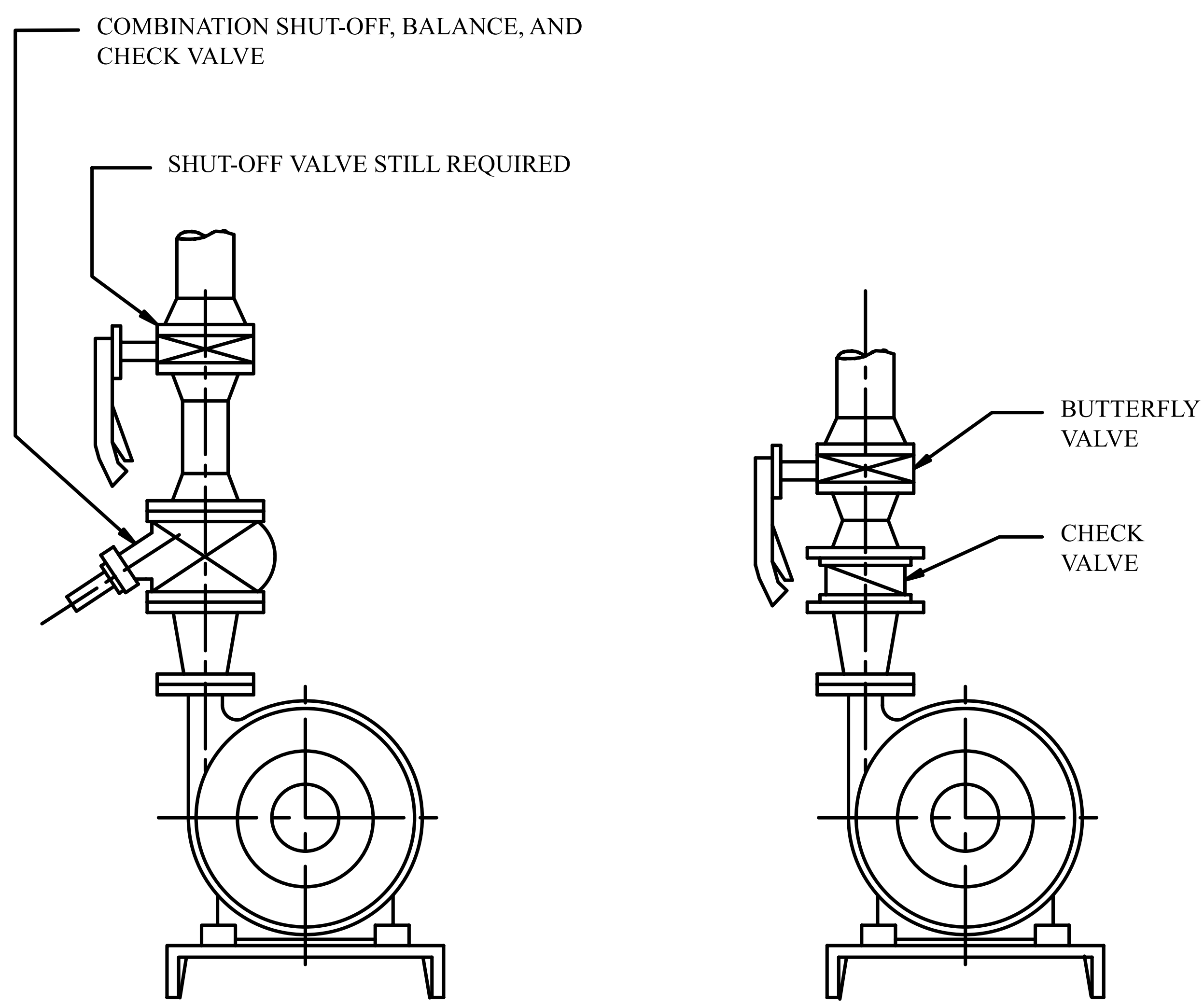
These valves are also holdovers from the days of constant volume systems. They are great energy wasters and should be removed or opened fully when found on variable volume systems. Figure 6-13 describes the multiple-duty valve that is supposed to be a check, balance, and shut-off valve. Unfortunately, most of them must have a shut-off valve downstream, as they cannot be repaired without draining the water system. All that is needed on the discharge of any HVAC pump is a check valve and a shut-off valve as shown in Figure 6-13b. On special occasions, the shut off valve may be a motorized, two position control valve.

On small constant volume systems, the multiple-duty valve should not be used where the pump head is too large. Instead, just the shut-off valve and the check valve should be provided as shown in Figure 6-13b. The shut-off valve should be closed partially until the pump head is acceptable. The valve handle sticking out at an angle is a reminder that the pump impeller should be evaluated for trimming.

Cross-Over Bridges:

Here again, we have a device from the constant speed days that is unnecessary today and should be avoided due to its energy waste and the increased pressure that it adds to the water system. Figure 6-14 describes a typical installation of a cross-over bridge to a variable primary pumping system, and Figure 6-15 is its hydraulic gradient. This is the addition of a building and tertiary pump to the system shown in Figure 6-12 in the place of Coil No.1 of that figure.

Figure 6-15 demonstrates the great energy waste of 96 ft. across the return valve. Also, since the main system pump and the building pump are in series, the overall system pressure is increased to 227 ft. or 98 psig. Both of these unsatisfactory conditions can be eliminated by changing the building pump to a Booster Pump which will be described later in this chapter. A typical cross-over bridge is not necessary when blending water temperatures. This will also be discussed later in this chapter.



NOT RECOMMENDED
a.

RECOMMENDED
b.

FIGURE 6-13

Discharge valves for HVAC pumps.

From: "HVAC Pump Handbook", James Rishel, PE, McGraw-Hill, New York, 1996.

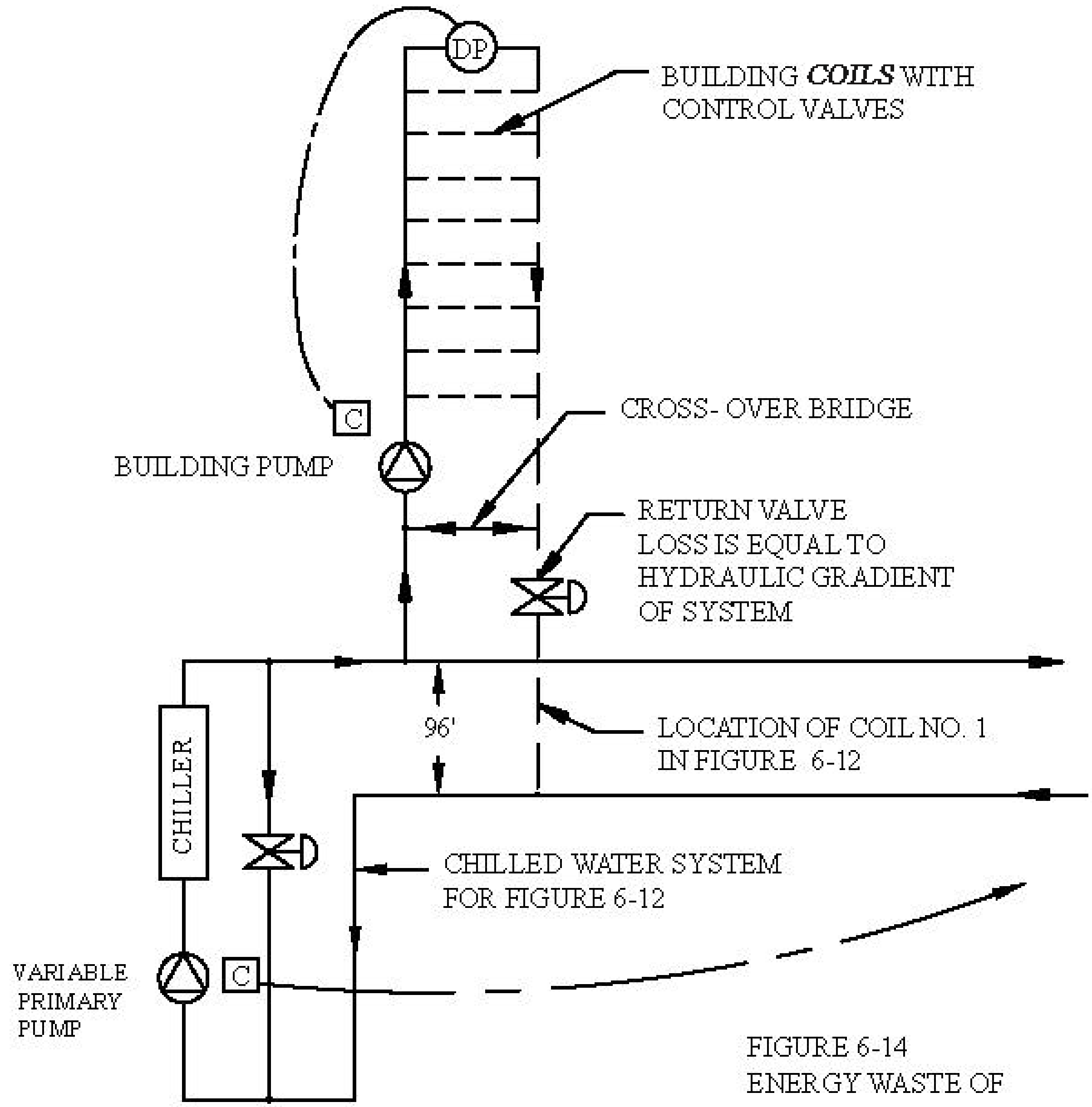
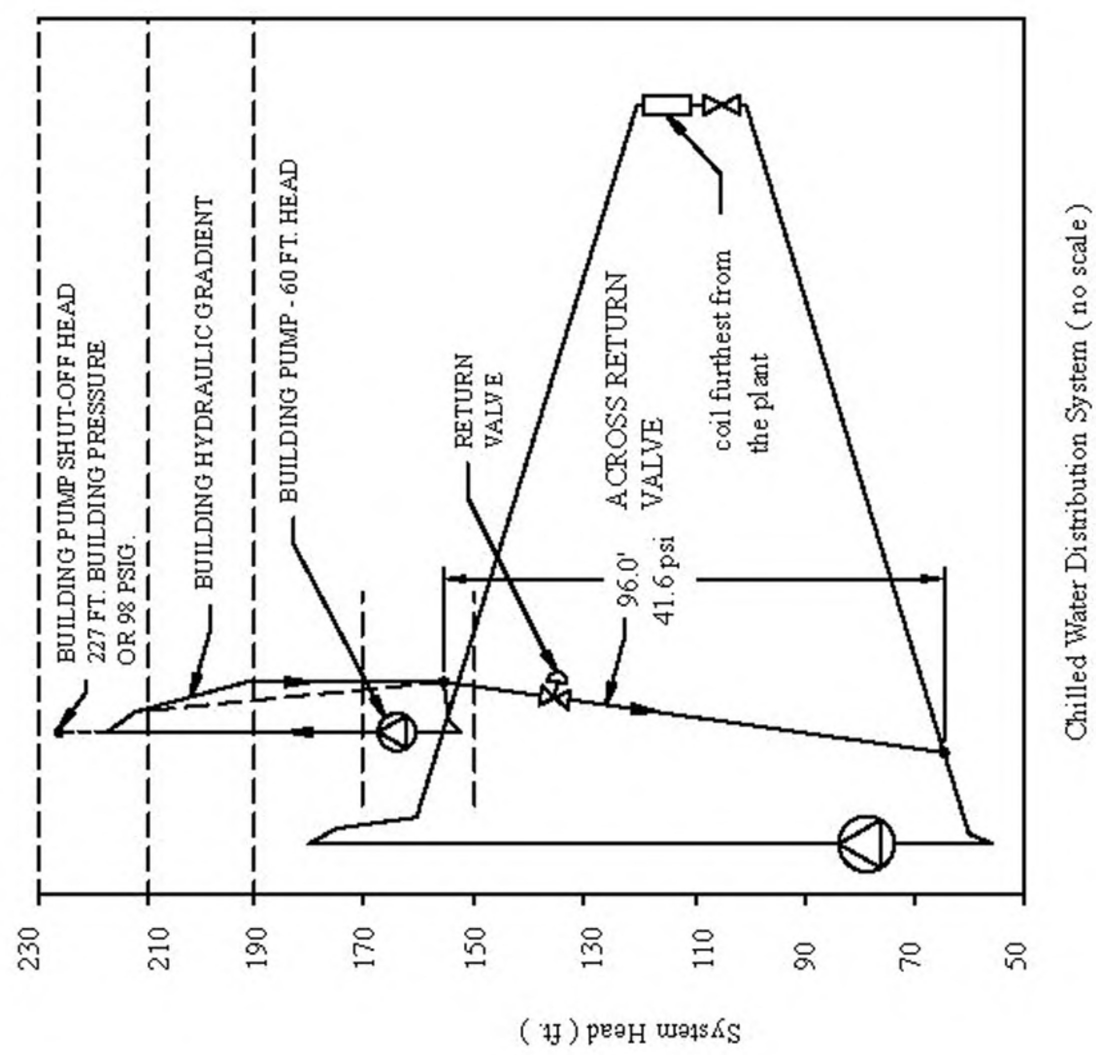


FIGURE 6-14
ENERGY WASTE OF
CROSS OVER BRIDGES
ON VARIABLE SPEED SYSTEMS

FIGURE 6-15
HYDRAULIC GRADIENT
FOR
FIGURE 6-14



Throughout this manual will be examples of how to reduce pump head by eliminating these four types of mechanical devices that waste pump head.

PUMP ARRANGEMENTS FOR CHILLED AND HOT WATER SYSTEMS:

Over the years with the advent of variable speed and digital control, pump arrangements for HVAC systems have changed dramatically. In the days of cheap energy and constant speed pumps, the hot and chilled water systems were **constant flow with three-way valves** on the coils as shown in Figure 6-16 with continuous flow through all of the chillers. This created an unsatisfactory condition, since the chillers had to be maintained on the line whether there was any cooling load on them.

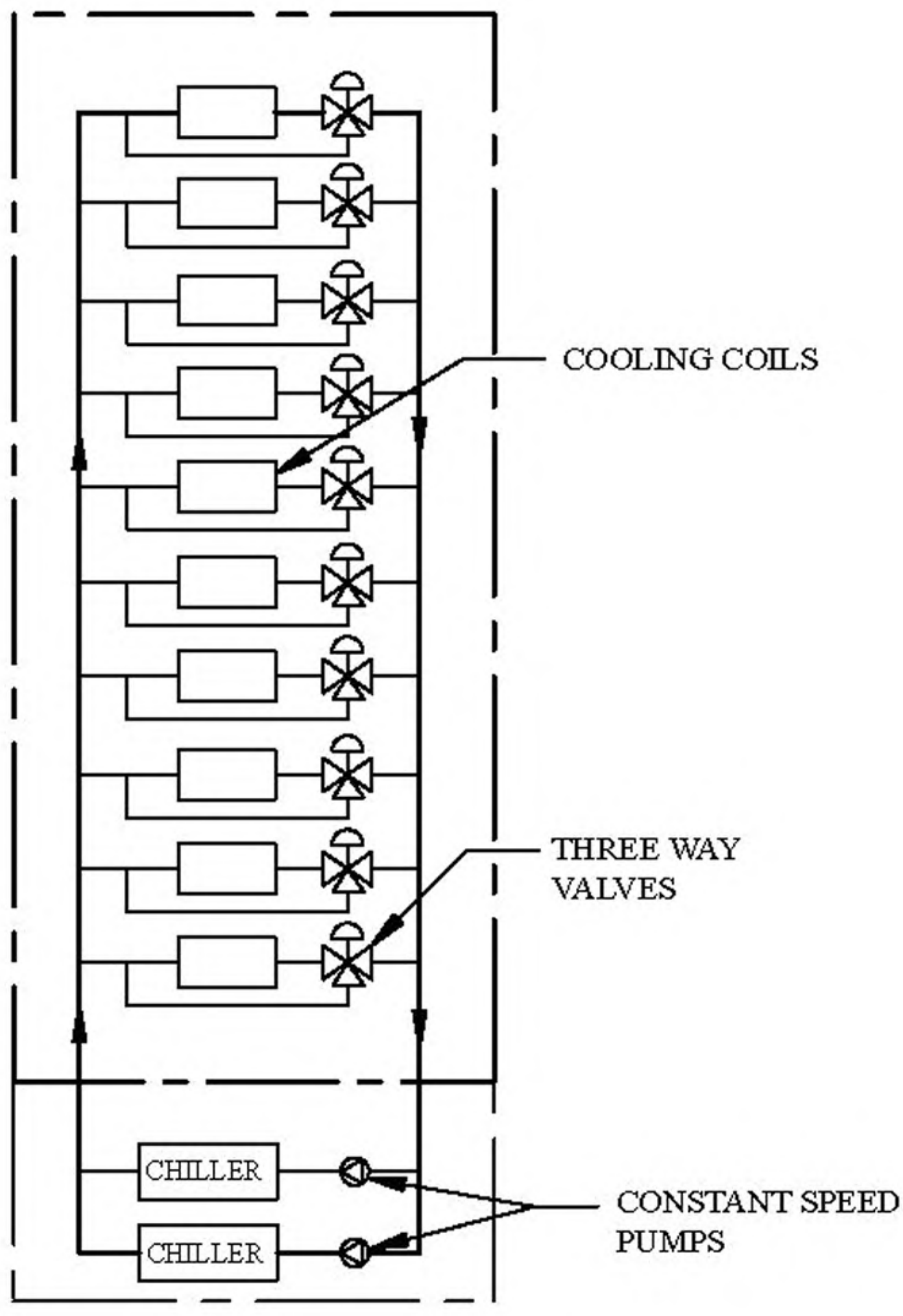
This was solved by changing the pumping to **primary/secondary pumping** as shown in Figure 6-17. Now the chillers could be added or subtracted as needed, since system flow is either through the bypass or the chillers. Initially, primary/secondary systems used three-way valves on the coils, but with the arrival of variable speed pumps, the coils were equipped with two-way valves and the day of variable volume flow arrived. Now, serious evaluation of pumping energy could begin, and all the mechanical devices of constant volume pumping could be eliminated.

In the early 1990's, most chillers were converted to digital control, and variable flow was allowed in the chiller evaporators. This resulted in **Variable Primary Pumping** that eliminated the need for the secondary pumps of primary/secondary pumping. Figure 6-18 shows the usual arrangement of a variable primary pumping system with the bypass valve that opens when the flow drops below the minimum allowable flow in the chiller evaporator. At present, most chiller manufacturers allow flows as low as 3 feet per second in their evaporators.

This should always be verified by the manufacturer of the chiller that is or will be on a particular chilled water system. The bypass valve is controlled by a differential pressure transmitter across the chiller evaporators, or it can be controlled by the flow meter that measures flows through the chillers. Elimination of the secondary pumps of primary/secondary pumping saves equipment room space and, usually, provides increased efficiency through the elimination of the friction loss in the secondary pumping system. Today, Variable Primary Pumping is the pumping arrangement of choice on most variable volume chilled water systems. Properly designed variable primary systems result in much simpler water systems than primary/secondary systems.

Primary/Secondary/Tertiary pumping systems are still touted by some pump companies, but they are no longer acceptable, as they generally use cross-over bridges. It has been demonstrated above that the use of cross-over bridges waste energy and can increase the operating pressure of a water system.

FIGURE 6-16
CONSTANT VOLUME
PRIMARY SYSTEM



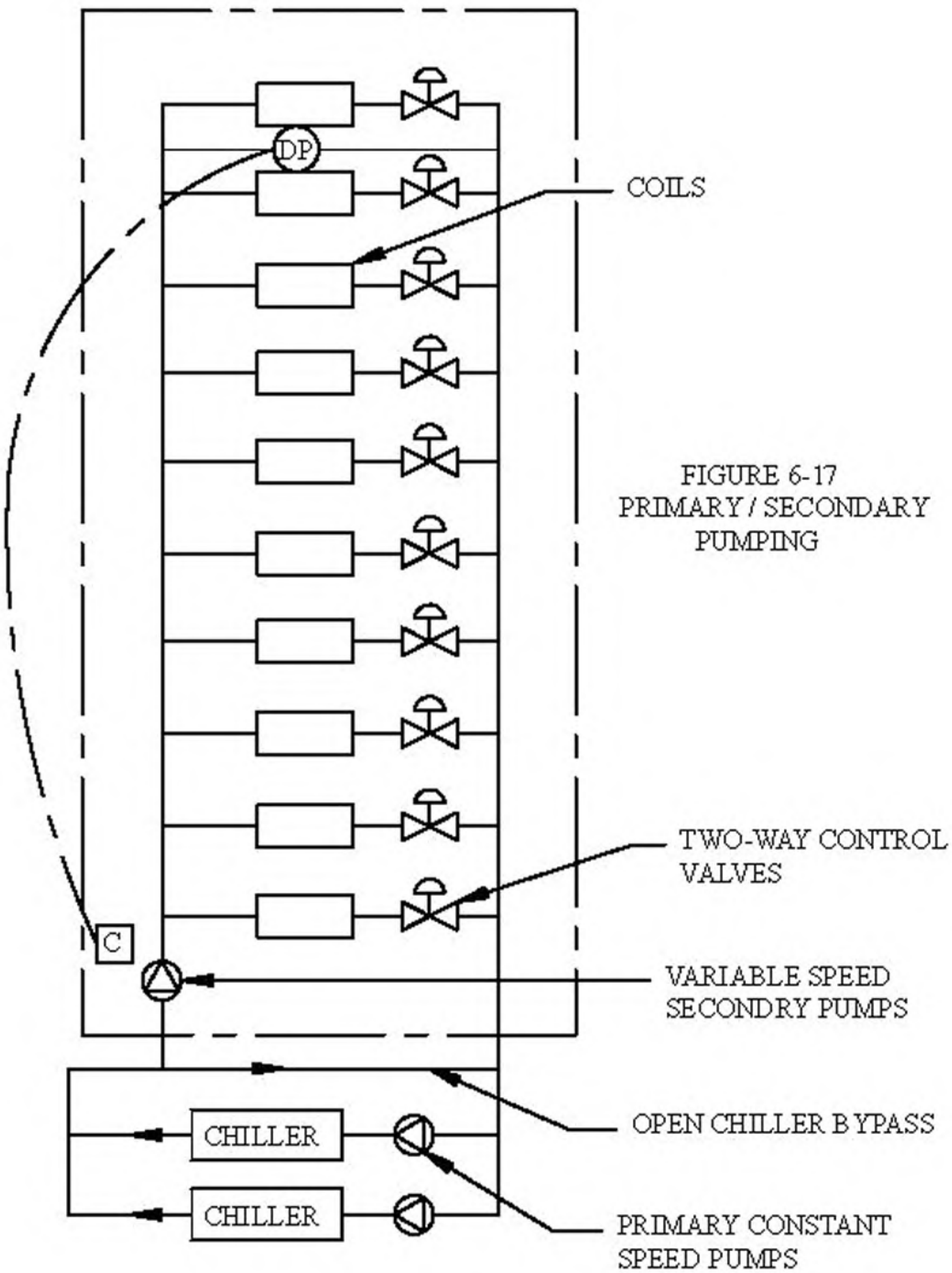


FIGURE 6-17
PRIMARY / SECONDARY
PUMPING

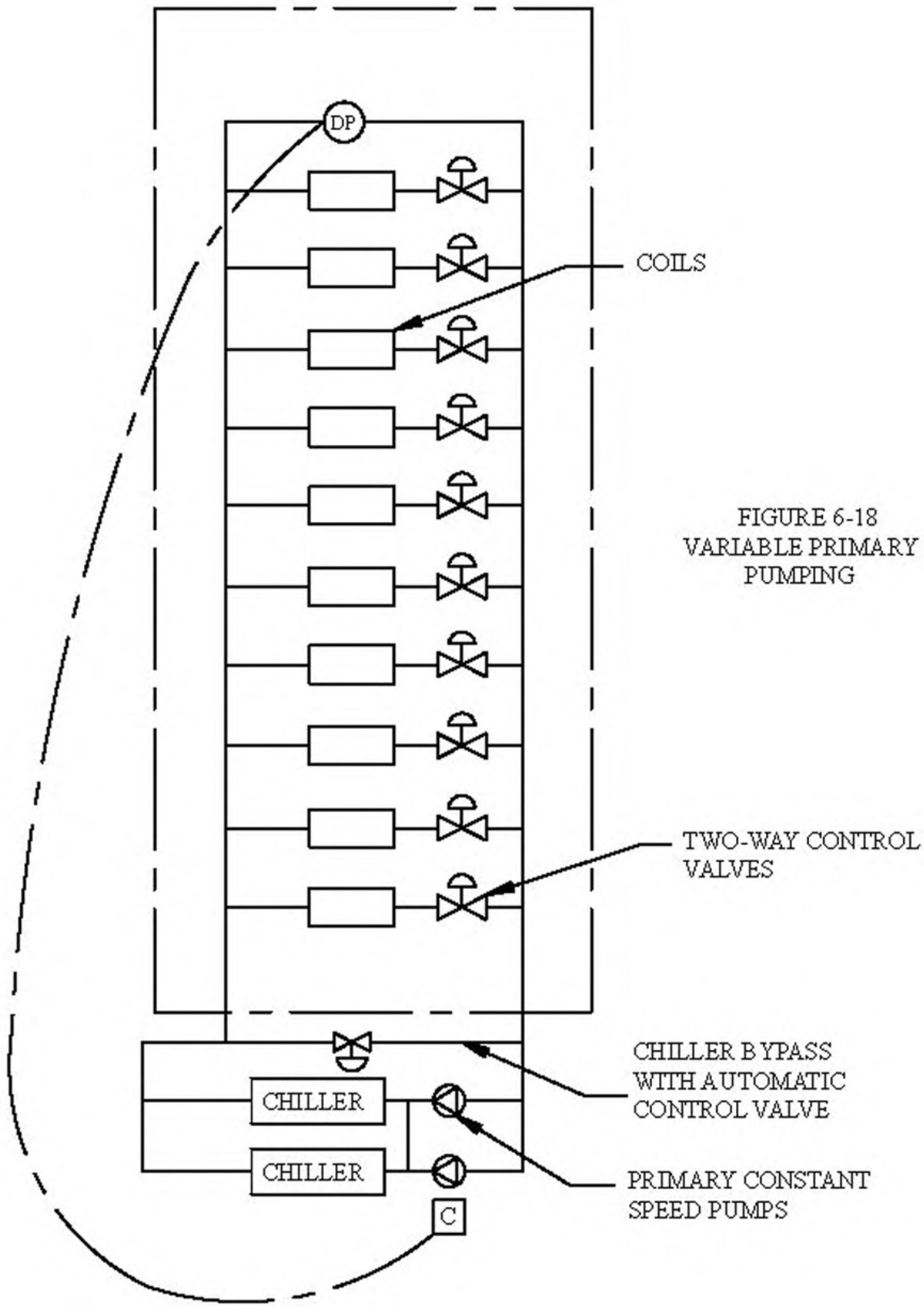


FIGURE 6-18
VARIABLE PRIMARY
PUMPING

Booster Pumping is often confused with primary/secondary/tertiary pumping. It does not use a cross-over bridge and is usually variable speed. It is used to boost chilled water into outlying buildings. A hydraulic gradient for a typical booster pumping system is shown in Figure 6 -19. This diagram describes the reduction in pump head that is achieved by using a booster pump instead of increasing the pump head at the central plant. Often, the central plant pumps can provide adequate pump head so that the booster pump can be stopped. When this is possible, the booster pump should be equipped with a bypass that allows the water to flow around it, Figure 6-20. If the central plant pumps are constant speed and of high head that could overpressure the building, a modulating valve that is controlled by the building differential pressure transmitter should be installed downstream from the building pump and by-pass check valve.

It is apparent that the efficient use of pump head can be a problem on large, multiple building installations. Several pumping procedures should be evaluated before settling on one particular design. **Distributed Pumping** offers another efficient method of distributing water to a campus type installation. Figure 6-21 describes a large airport (Denver International) with distributed pumping. Primary/secondary pumping at the chiller plant was planned originally, but the secondary pumps become very large (1,000 hp). Figure 6-22 was the hydraulic gradient for them, and it demonstrates the problem with large secondary pumps. Too much pump head is applied at the beginning of the distribution system. Air retainment and operating pressures would increase the overall maintenance of this water system. Pressure reducing valves or cross-over bridges, with their energy waste, will be required on the buildings near the central plant to prevent over pressuring of the buildings.

Compare this to **Distributed Pumping** where there are no secondary pumps, Figure 6-23 that demonstrates that there is no excessive pressure at the beginning of the system. Now there is no over pressuring of any of the buildings, and the system runs most of the time at very low systems pressures, Figure 6-24. The variable speed pumps in each building are controlled by a building differential pressure transmitter. The overall system pressure is greatly reduced, so the maintenance of the piping will be much less than that for secondary pumping. Energy consumption for distributed pumping has been reduced appreciably over what is required for secondary pumping. In this case, the water hp for distributed pumping was 74% of that for secondary pumping. Distributed Pumping is a distribution system that is difficult to use on most existing systems. It should be considered on new, campus type systems.

This covers most of the chilled water distribution systems that are used for chilled and hot water. It must be emphasized that the old rule about avoiding the installation of pumps in series still applies to constant speed pumps, but not to variable speed pumps. The secret to controlling variable speed pumps in series is to insure that proper control exists for each set of pumps. This is shown in Figure 6-25. It should be remembered that maximum operating pressure is

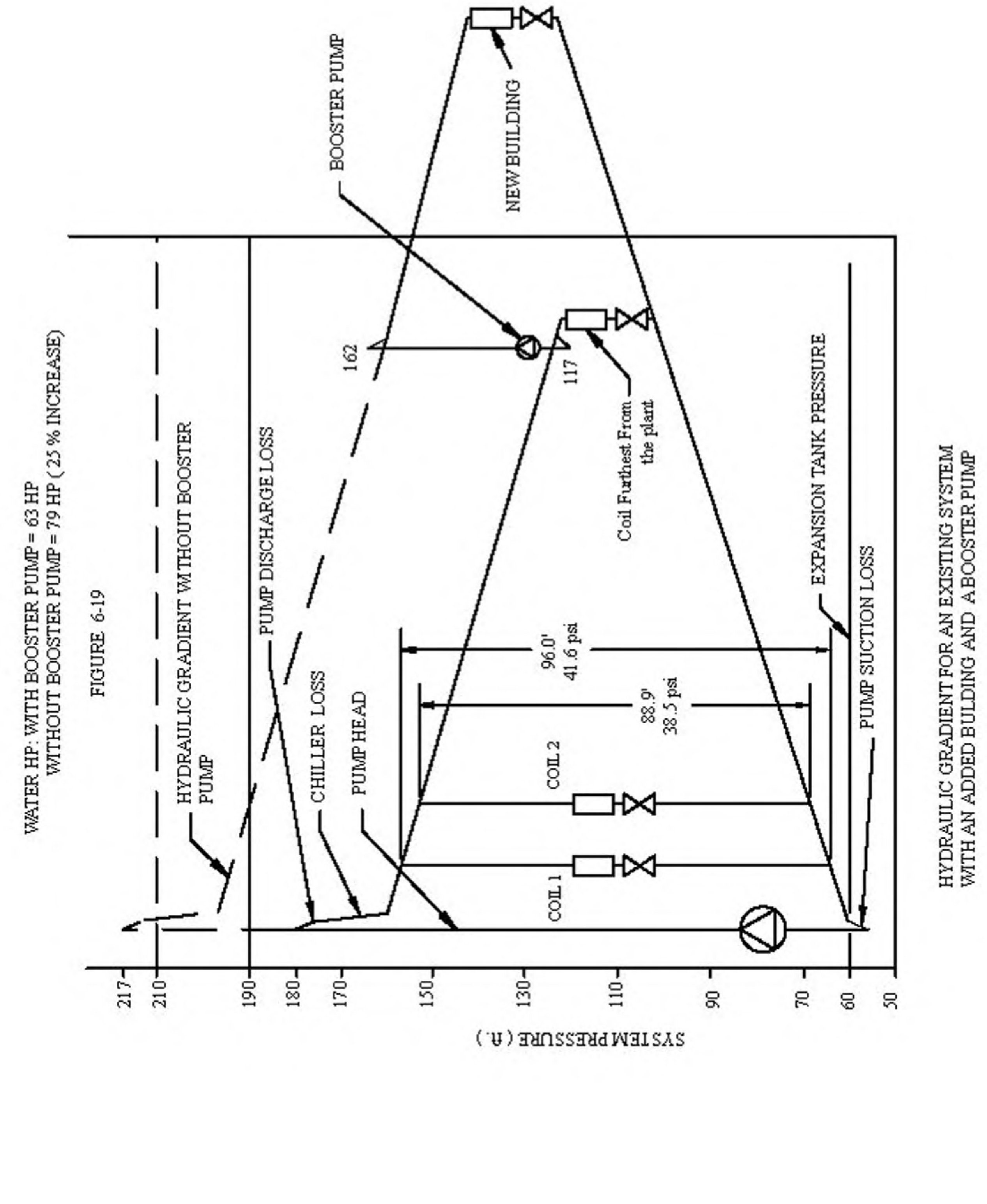


FIGURE 6-19

HYDRAULIC GRADIENT FOR AN EXISTING SYSTEM WITH AN ADDED BUILDING AND A BOOSTER PUMP

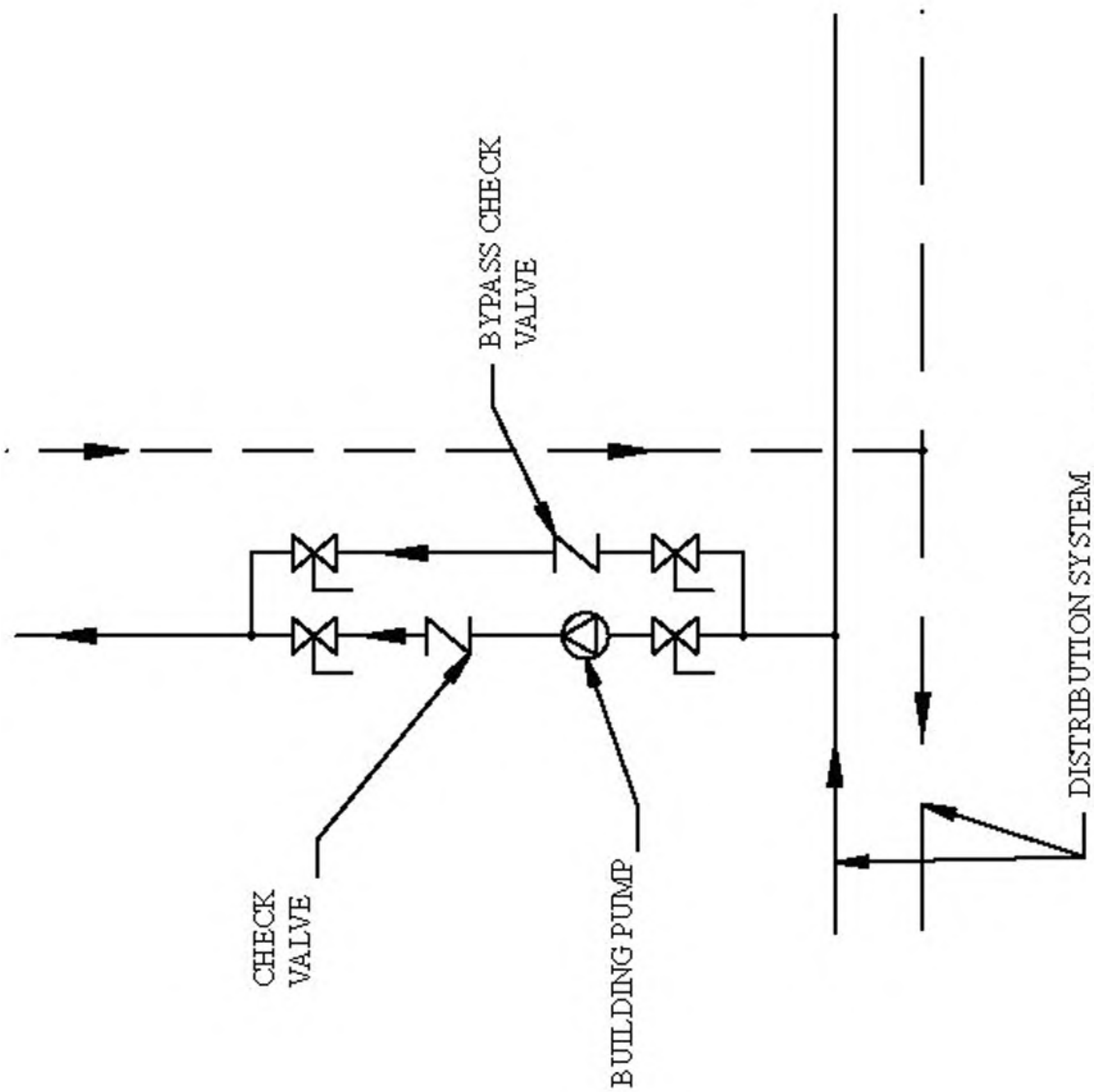


FIGURE 6-20
BYPASS CHECK VALVE
FOR
BUILDING PUMP

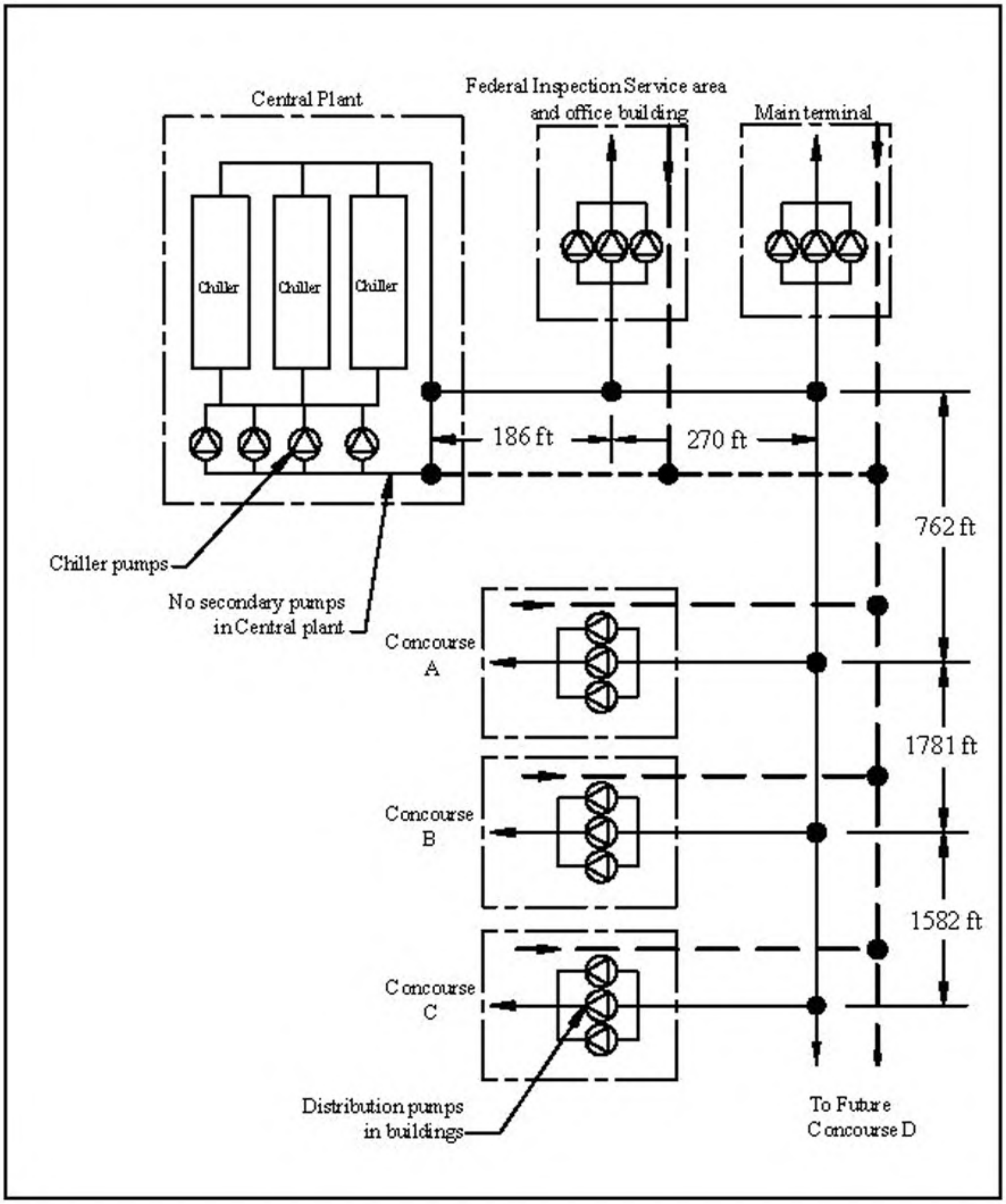


FIGURE 6-21

Distributed pumping system at the airport. Secondary pumping is accomplished by distribution pumps in each building rather than by secondary pumps back at the central plant.

"Distributed Pumping Lowers Horsepower Needs at Denver International", James Ottmer, PE and James Rishel, PE, HPAC Magazine, October 1993.

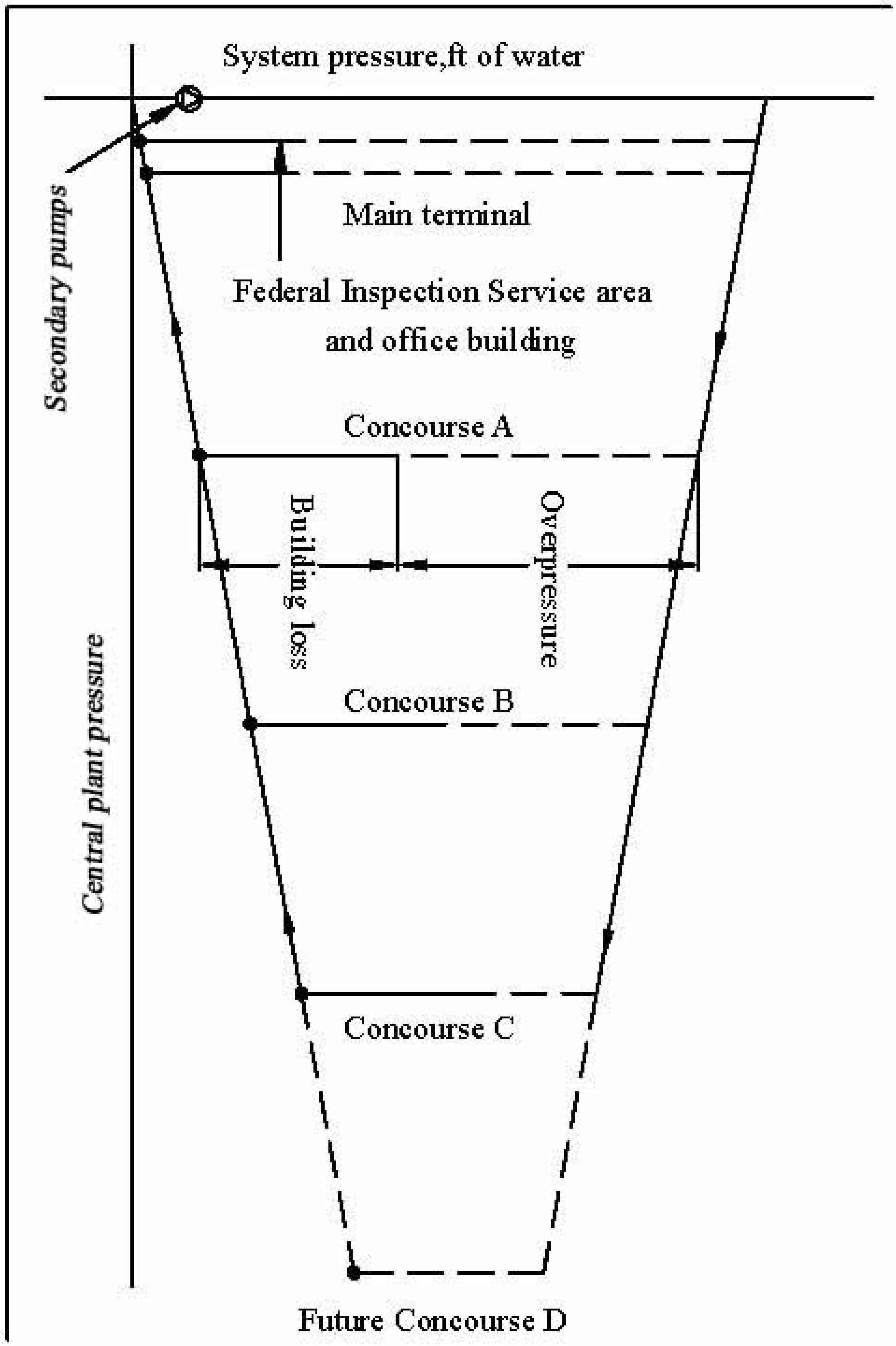


FIGURE 6-22

Typical pressure gradient for diagram for a conventional secondary pump loop.

"Distributed Pumping Lowers Horsepower Needs at Denver International", James Ottmer, PE and James Rishel, PE, HPAC Magazine, October 1993.

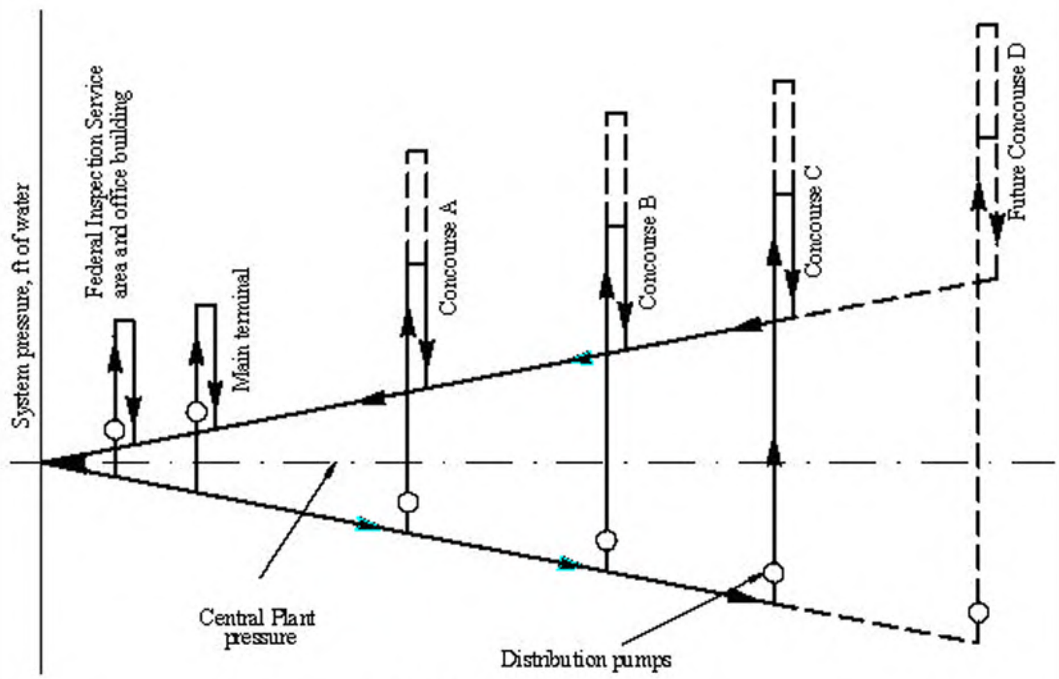


FIGURE 6-23

Typical pressure gradient diagram for disributed pumping at full load.
dotted lines represent pressure gradient after future concourse expansion.

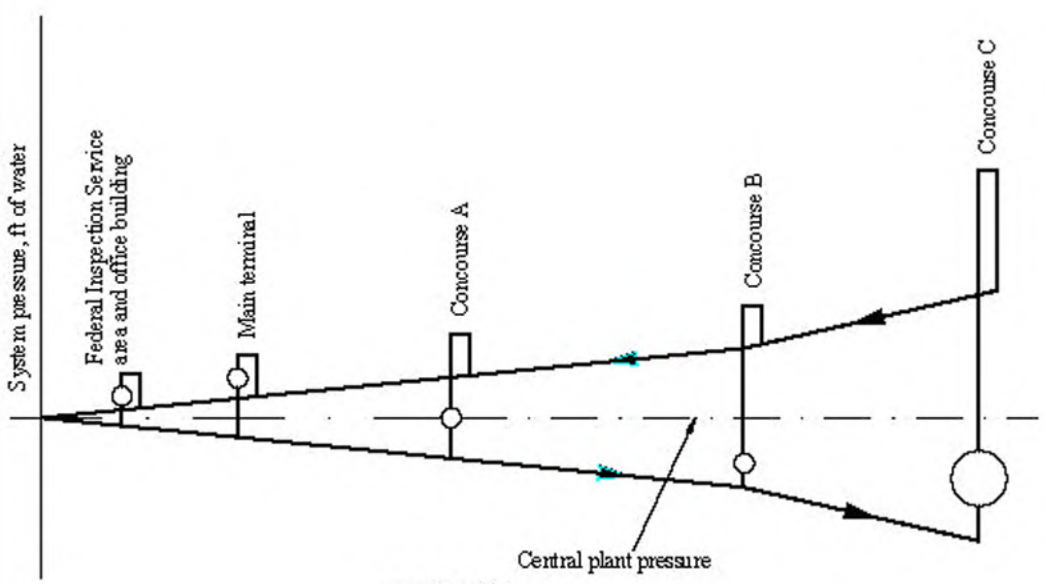


FIGURE 6-24

Typical pressure gradient diagram for distributed pumping at partial load.

"Distributed Pumping Lowers Horsepower Needs at Denver International", James Ottmer, PE and James Rishel, PE, HPAC Magazine, October 1993.

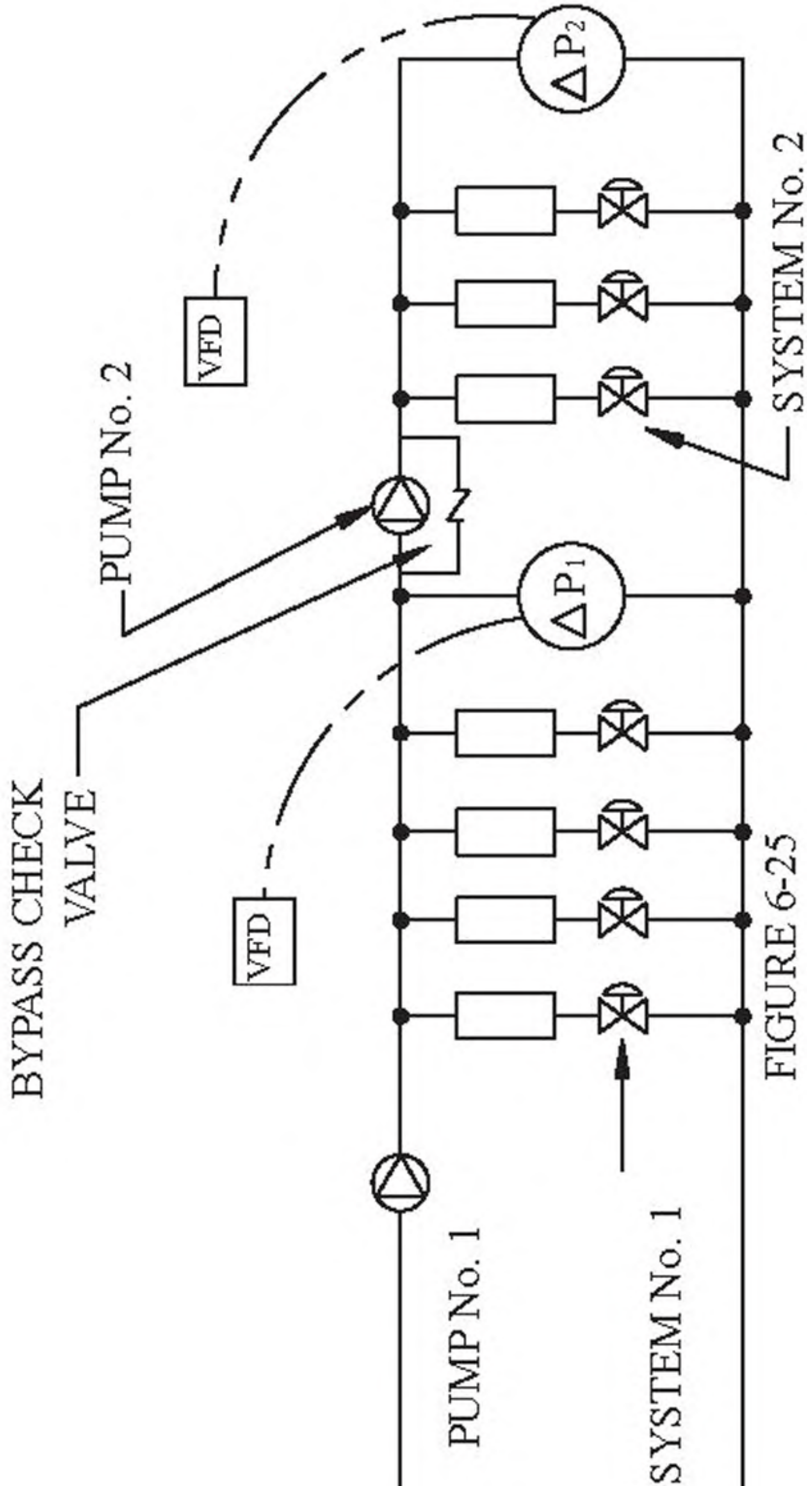


FIGURE 6-25

VARIABLE SPEED PUMPS IN SERIES

always calculated by adding the suction pressure to the shutoff heads of all of the pumps in series.

HEAT EXCHANGERS:

For High Rise Buildings:

As indicated above, the design working pressure for a chilled or hot water system is the shut-off head of the pumps added to the expansion tank pressure or fill pressure. If the expansion tank is at the top of a tall building as shown in Figure 6-4, the static height of the building must be added to the fill pressure and pump shutoff head. In the case of Figure 6-4, the maximum pressure is 365 ft. or 158 psig. This is caused by the vertical height of the building. If a tall building like this is added to a group of buildings of only two or three stories, the overall system pressure is increased by this tall building. Often heat exchangers are used to reduce the operating pressure as described in Figure 6-26. In this figure, the heat exchanger is taking water from a central plant where the pumps are variable speed. If this heat exchanger is the farthest load hydraulically from the central plant, a differential pressure transmitter will be located across the supply side of the heat exchanger as shown. Most of the heat exchanges for this service are now of the plate and frame type.

For Steam Heated Exchangers:

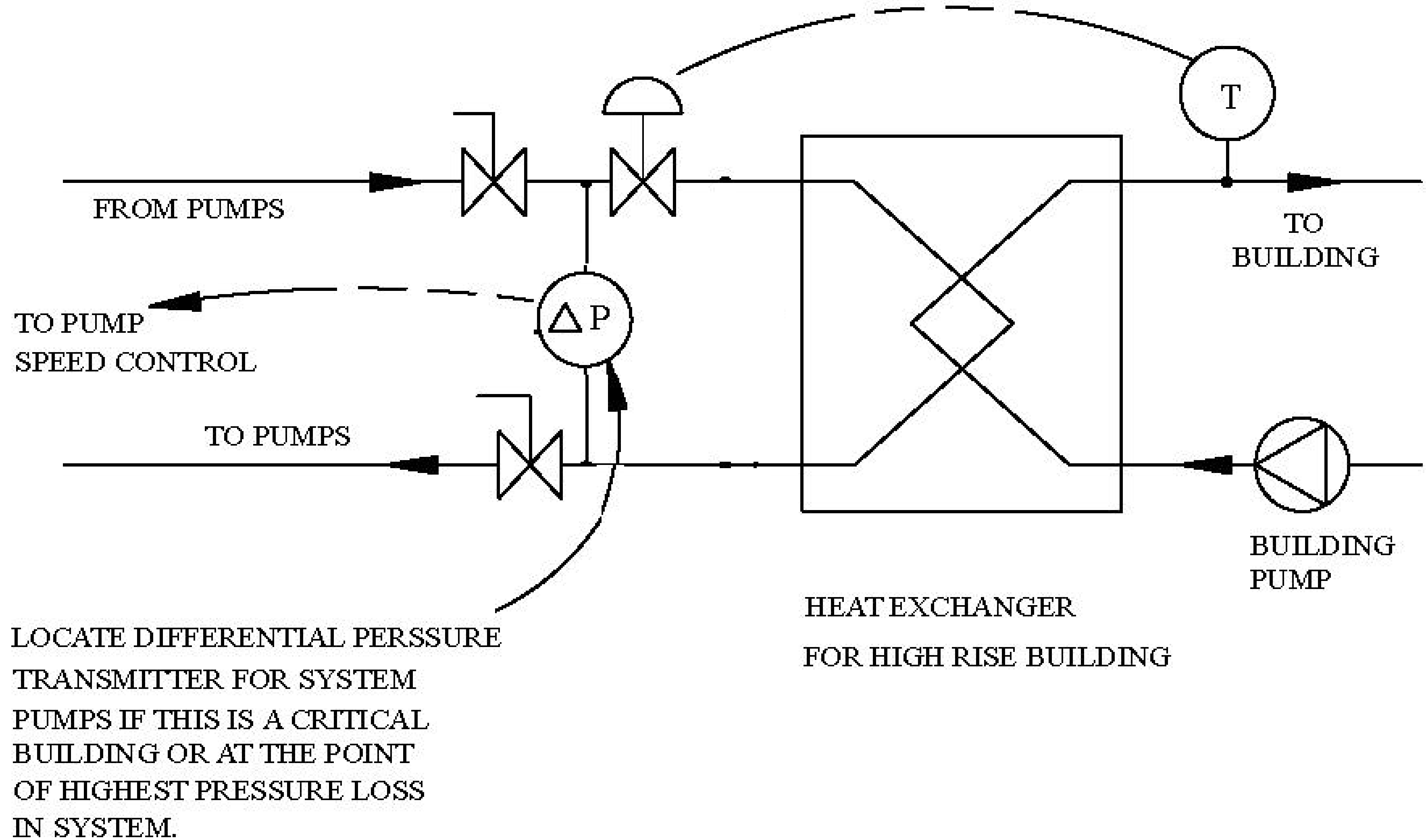
A source of heat for hot water systems can be steam on large installations with steam boilers. The heat exchangers for most of these services are still of the shell and tube type. The heat exchanger piping can be of the traditional arrangement, Figure 6-27, or it can be of the three-way control valve type, Figure 6-28. The latter configuration offers more precise temperature control and eliminates the vacuum problems in the shell that can occur with the piping of Figure 6-27. Also, air is not sucked back into the heat exchanger shell through the vacuum breaker. Care should be taken to insure that the steam controls are selected for the steam pressure that is be used in the heat exchanger. Low pressure steam in the range of 2 to 5 psig is preferable. (See also heat exchangers for heat recovery in Chapter 7).

HEATING AND COOLING COIL CONNECTIONS:

The point of use of the chilled and hot water of HVAC water systems is the finned coil. It is normally controlled by an automatic valve that regulates the leaving air temperature from the coil. Some coils may have humidity control as well. In the days of constant volume systems, coils were not equipped with control valves and were called “wild” coils using face and bypass dampers to control the leaving air temperature. Such construction is now considered to be unacceptable since the return water temperature is uncontrolled, causing inefficient central plant operation.

FIGURE 6-26

HEAT EXCHANGER FOR HIGH RISE BUILDING



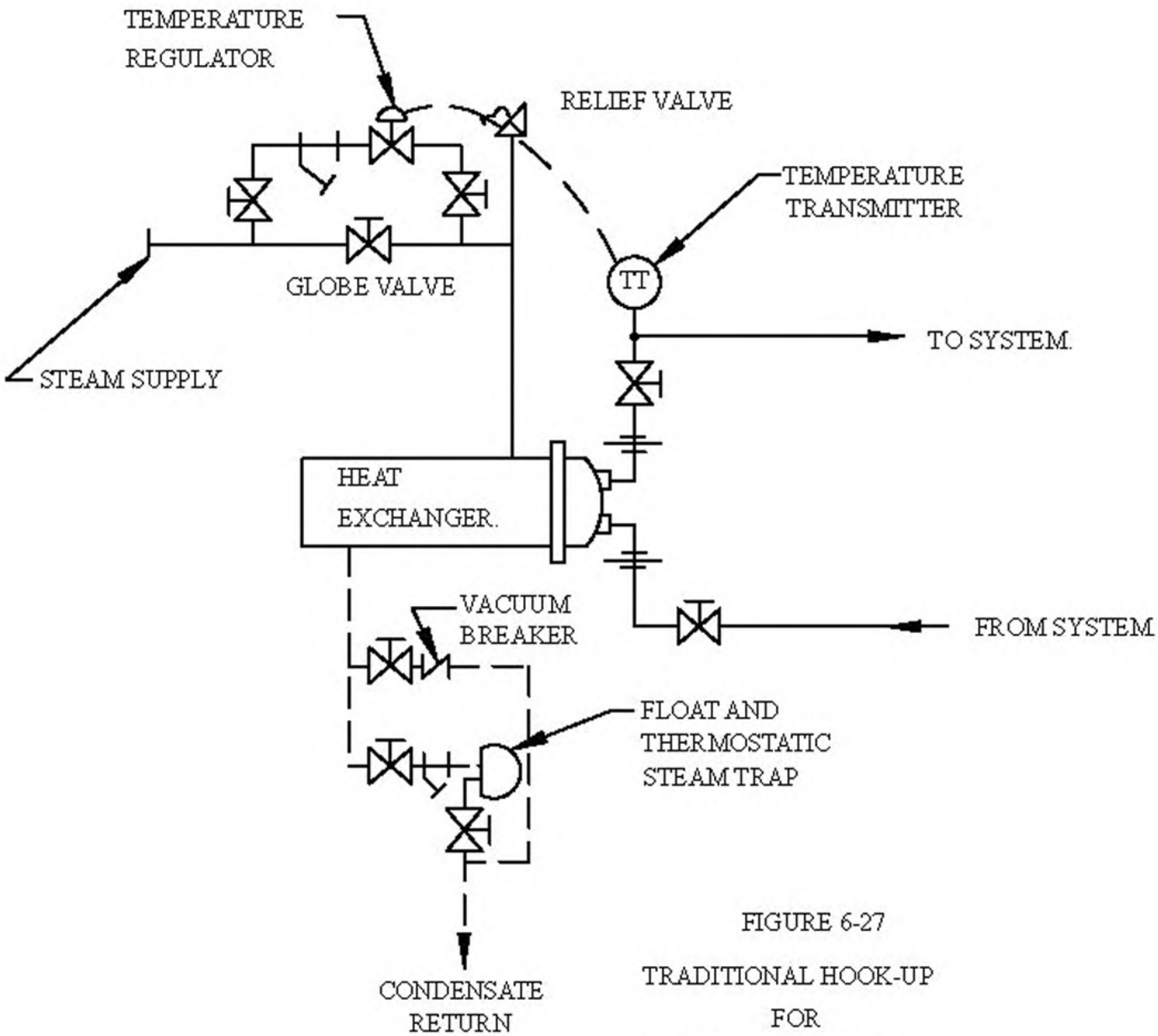


FIGURE 6-27

TRADITIONAL HOOK-UP
FOR
STEAM TO HOT WATER
HEAT EXCHANGER

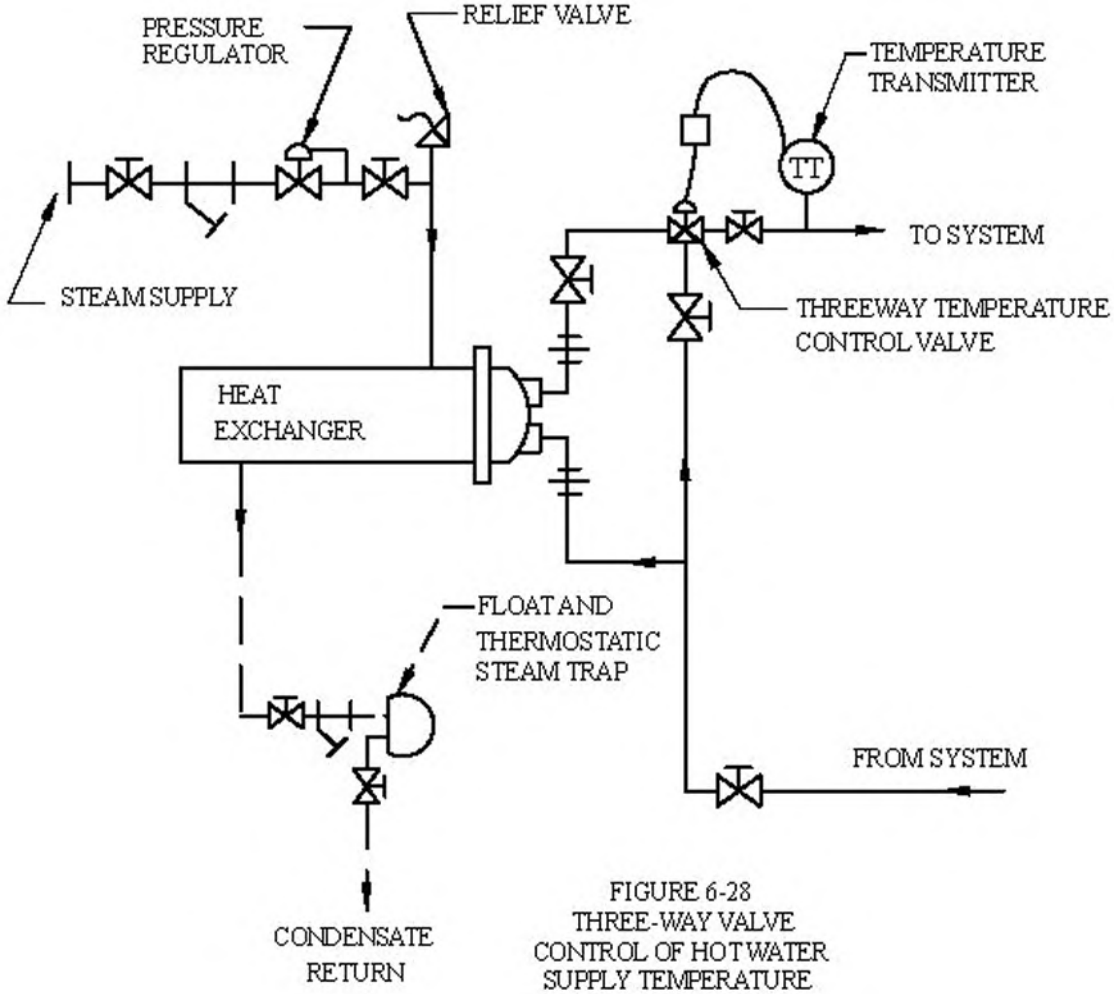


FIGURE 6-28
THREE-WAY VALVE
CONTROL OF HOT WATER
SUPPLY TEMPERATURE

Figure 6-29 describes the recommended connection of most of these coils. With the variable volume control that we have today, there is no need for any kind of balance valve on such coils. On small, constant volume systems, balance valves may be needed to balance the flow through the coils.

HEAT PUMPS:

Heat pumps offer an economical answer for many commercial and large residential loads. Water pumping is quite simple since all of these installations are closed circuits. Variable speed pumping can offer sizeable energy savings for them due to the great load variations that can occur. There are really no special requirements for the pumping system. Two 60 or 75% capacity pumps usually offer an economic installation with adequate standby capacity.

Heat Pumps with Closed Circuit Cooler:

Figure 6-30 describes the most common type of heat pump installation with reverse return piping to reduce the pressure drop across the coil control valves. Since there can be as many as 500 or more units on these projects, the control valves are usually of light construction and cannot have high differential pressures. The closed circuit cooler is usually of the evaporative type due to the relatively low temperature of the condenser water.

Geothermal Applications:

The advent of geothermal installations has proved to be energy efficient for large high schools and similar installations where there are a number of large central heat pumps, Figure 6-31. As shown, wells can be the geothermal source; other installations use lakes and rivers. Contrary to some sources, there is no need for special pumps for pumping the geothermal field. Only one set of pumps is needed with speed control by the remote differential pressure transmitters out in the building. The pumps can be located on the return to the well field as shown or on the supply from the well field. The actual physical pressures exerted on the system will determine the best location for the pumps. Again, due to the cost restraints or such projects, two 75% capacity pumps should be adequate for most installations.

WATER AND ICE STORAGE:

Energy storage is achieved through water or ice storage in the HVAC industry. Their basic uses are to shave peak energy capacity or avoidance of high daytime electrical demand charges. Peak shaving avoids the need to add additional chiller capacity, while use of this storage during the daytime hours reduces the demand charges that may be high during heavy demand hours such as from 12:00 to 6:00 PM.

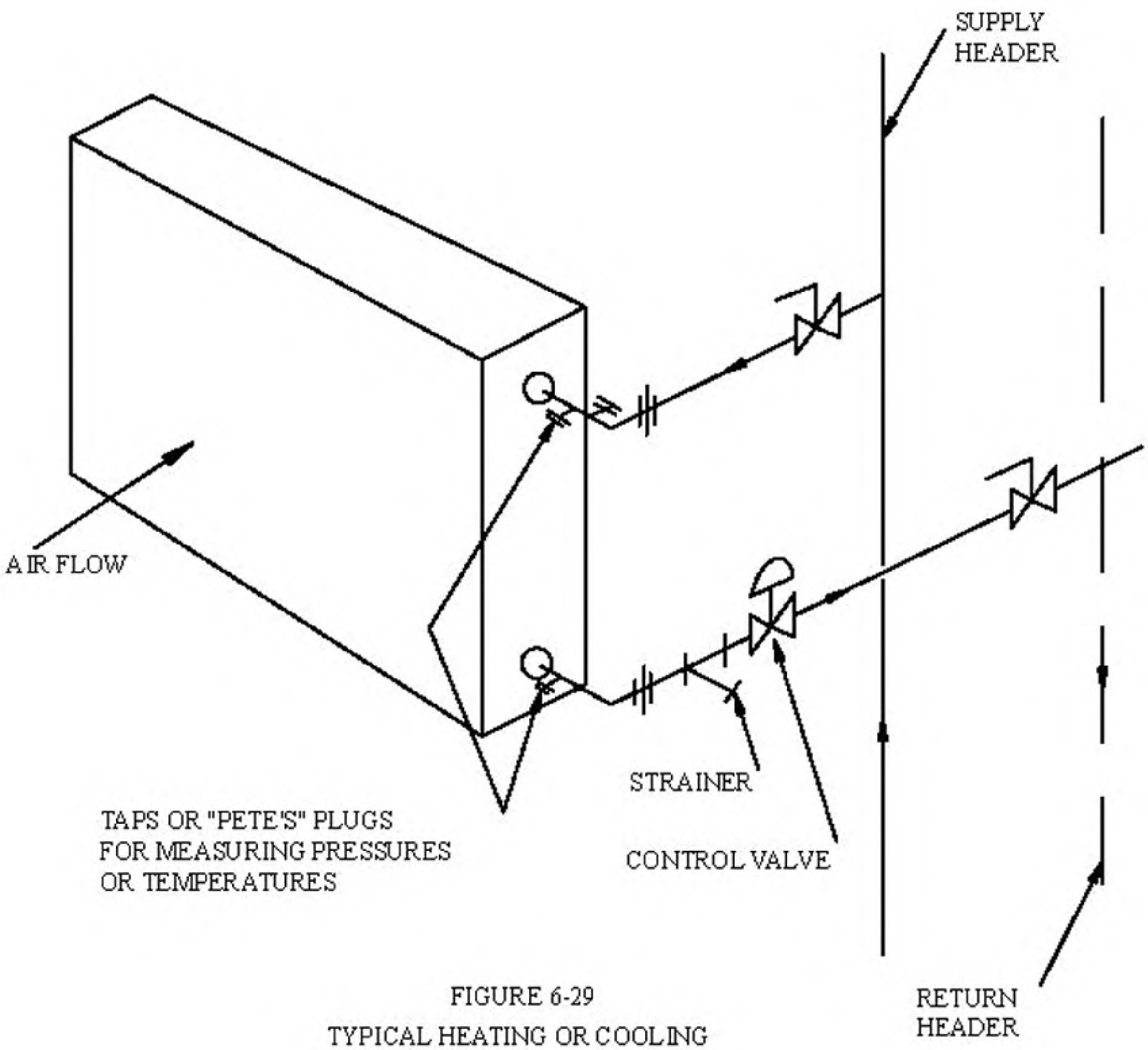


FIGURE 6-29
TYPICAL HEATING OR COOLING
COIL CONNECTIONS

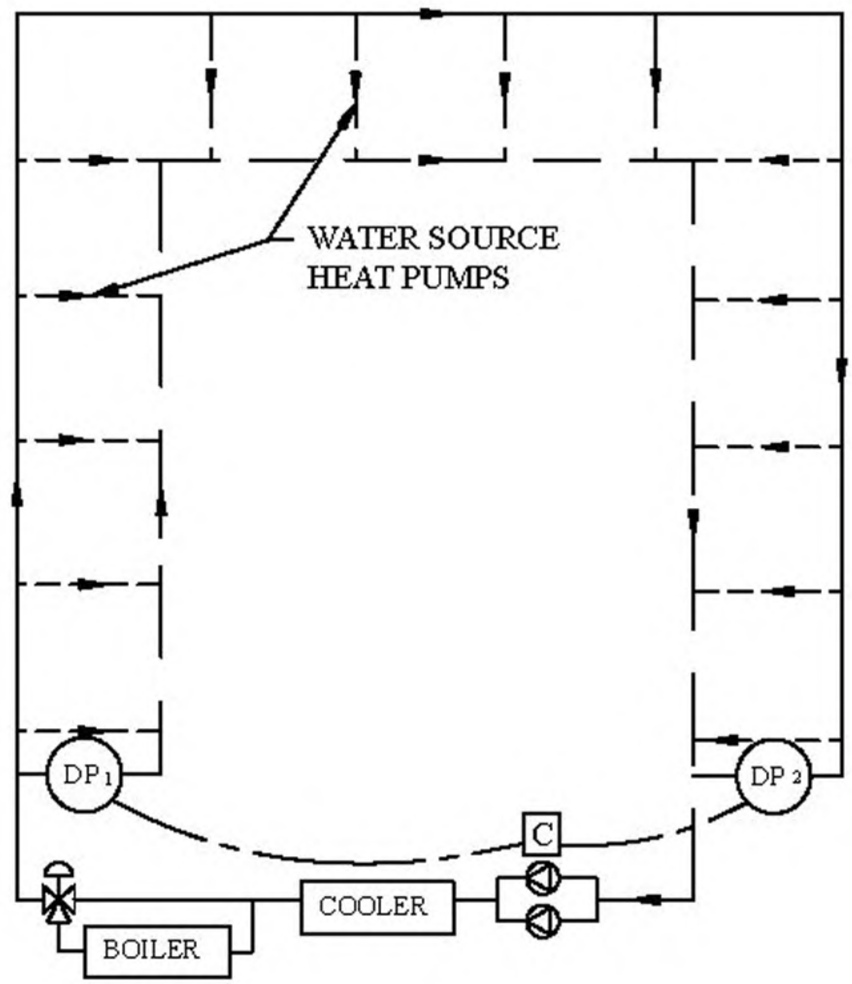


FIGURE 6-30

WATER SOURCE HEAT
PUMPS WITH CLOSED
CIRCUIT COOLER

The basic unit of measurement for water or ice storage is the **ton-hour** which is equal to 12,000 btu/hr stored or used. These storage systems vary from less than 100 up to tens of thousands of ton-hours.

The decision as to whether to use ice or water storage depends upon a number of economic factors. Water requires around seven to ten times as much volume as ice, so this is one factor that must be included. Others are first cost and operating expenses. A detailed evaluation of the value of energy storage must be made by engineers experienced in this field.

One of the problems with open storage tanks is the installation of control valves on the pump suction, Figure 6-32. The storage system designer must understand that the control valve has to be on the pump discharge, not the suction where it can reduce the NPSHA.

Water Storage:

From a pumping standpoint, water storage is quite simple. There are no special requirements, since, in most cases, the storage is above the pumps with no NPSH problems. There may be a need for heat exchangers if high rise buildings are involved, Figure 6-33. Back pressure valves on the return have been used with some success with the elimination of the heat exchanger.

The storage tank pumps can be constant speed with the temperature control valve modulating as shown; on larger systems, they may be variable speed with pumps for tank temperature control and other pumps for building temperature control.

Ice Storage:

Ice storage can be with open tanks or closed loops in open tanks. Open, underground tanks are used with ice harvester systems, Figure 6-34. The tank pumps should be self-priming or ordinary volute type pumps equipped with a self-primer. It may be more efficient to use low head tank pumps with secondary, higher head system pumps. With the priming system, the tank and system pumps can be combined.

Closed loop pumping systems are found in the larger ice storage systems, Figure 6-35. Ice pumps actually circulate a cold, glycol solution from the tank to a heat exchanger for the chilled water system. There are many different chiller designs for ice generation that really do not affect the pumping system.

Control of Return Water Temperature:

A subject that become of great concern is return water temperature in chilled and hot water systems. This has been generated by poor distribution and control of

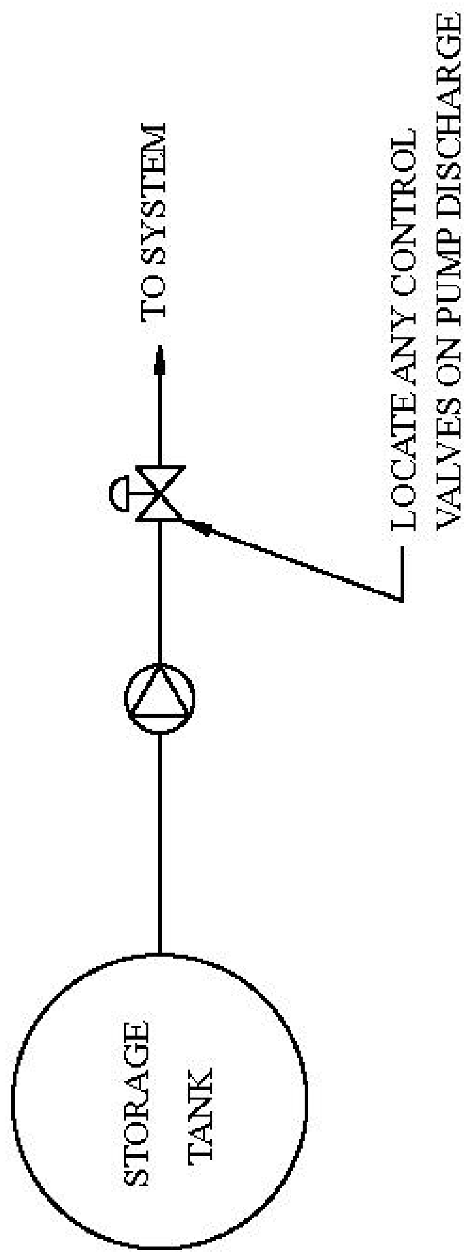


FIGURE 6-32

LOCATION OF CONTROL
VALVES.

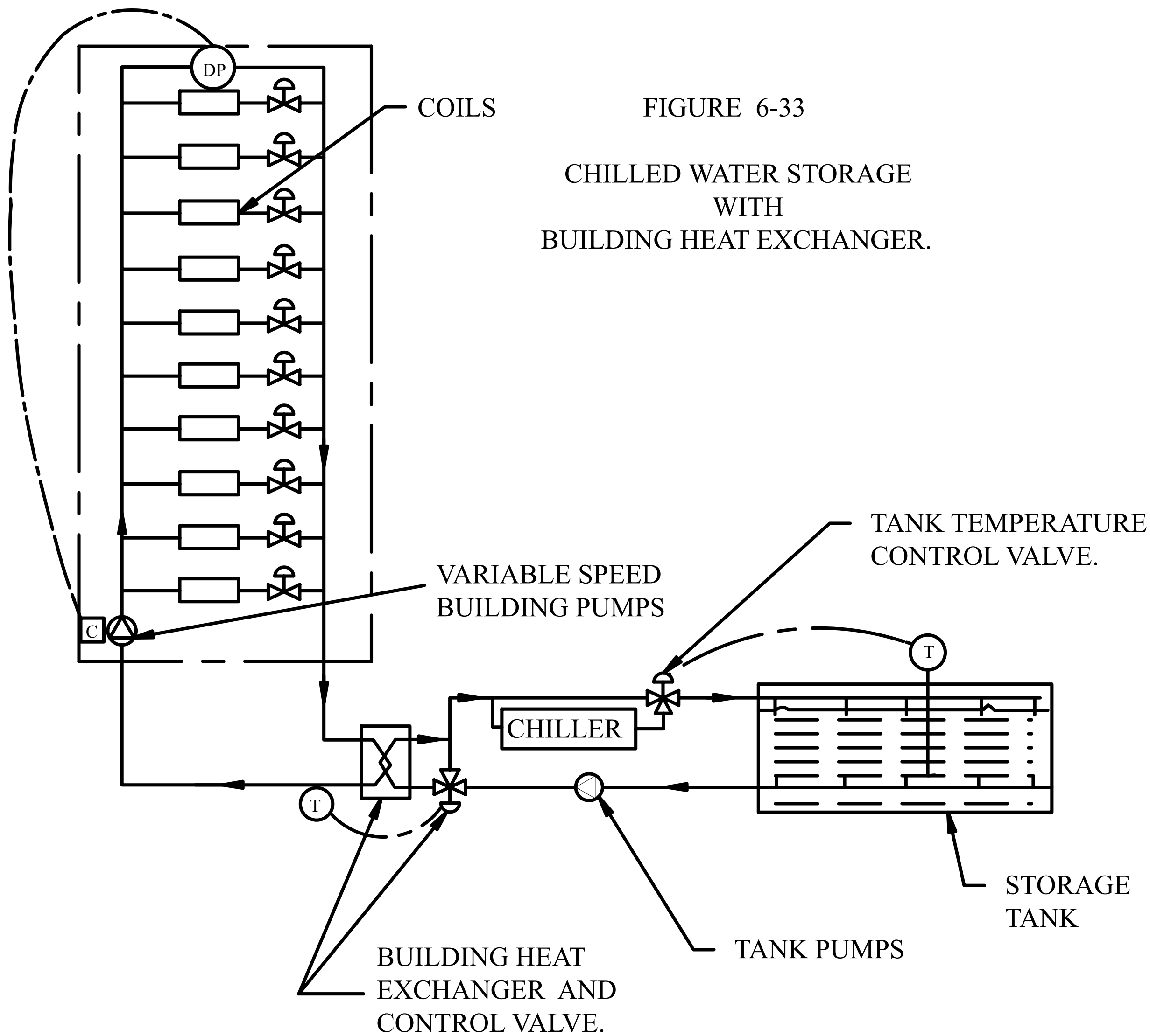


FIGURE 6-33

CHILLED WATER STORAGE
WITH
BUILDING HEAT EXCHANGER.

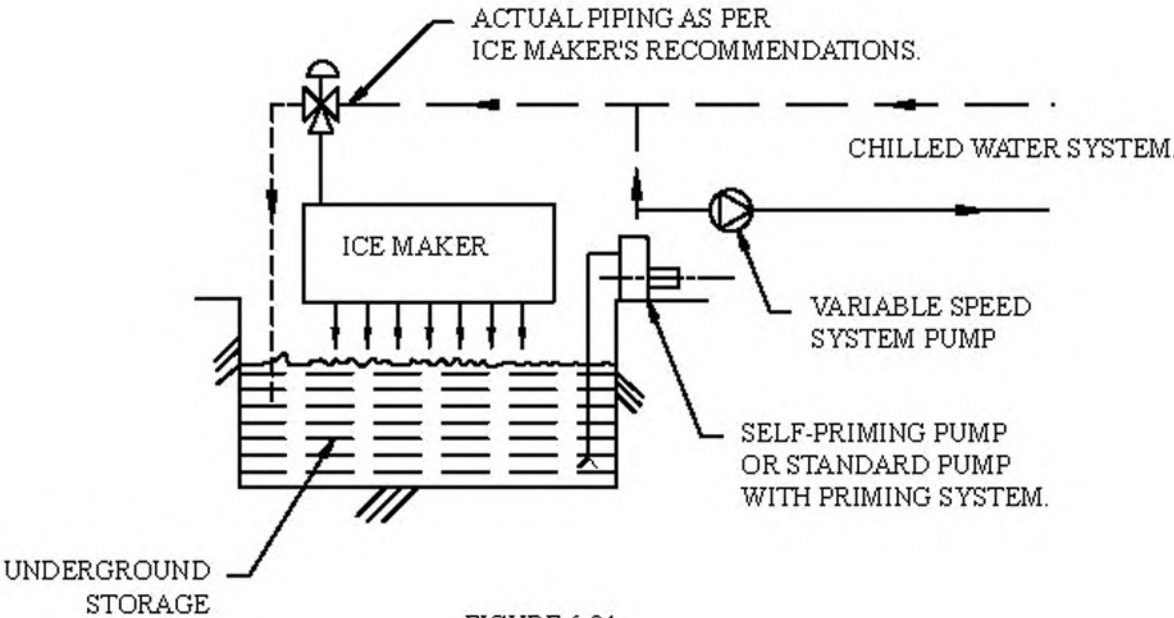


FIGURE 6-34

ICE HARVESTER SYSTEM.

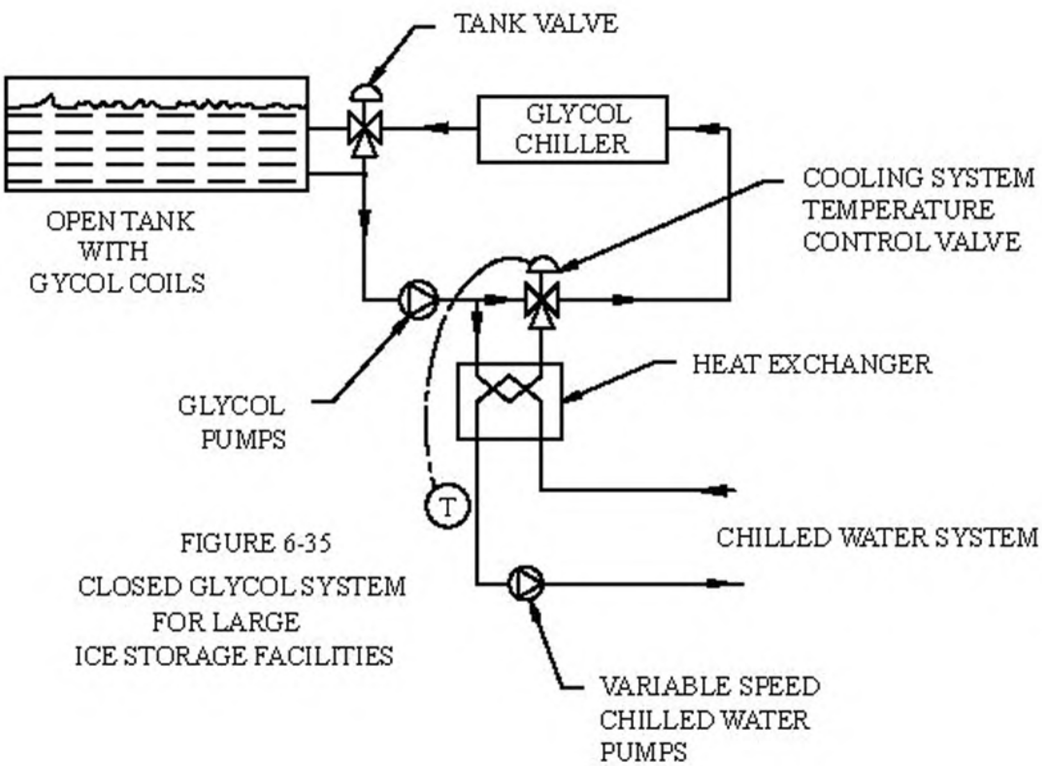


FIGURE 6-35

CLOSED GLYCOL SYSTEM
FOR LARGE
ICE STORAGE FACILITIES

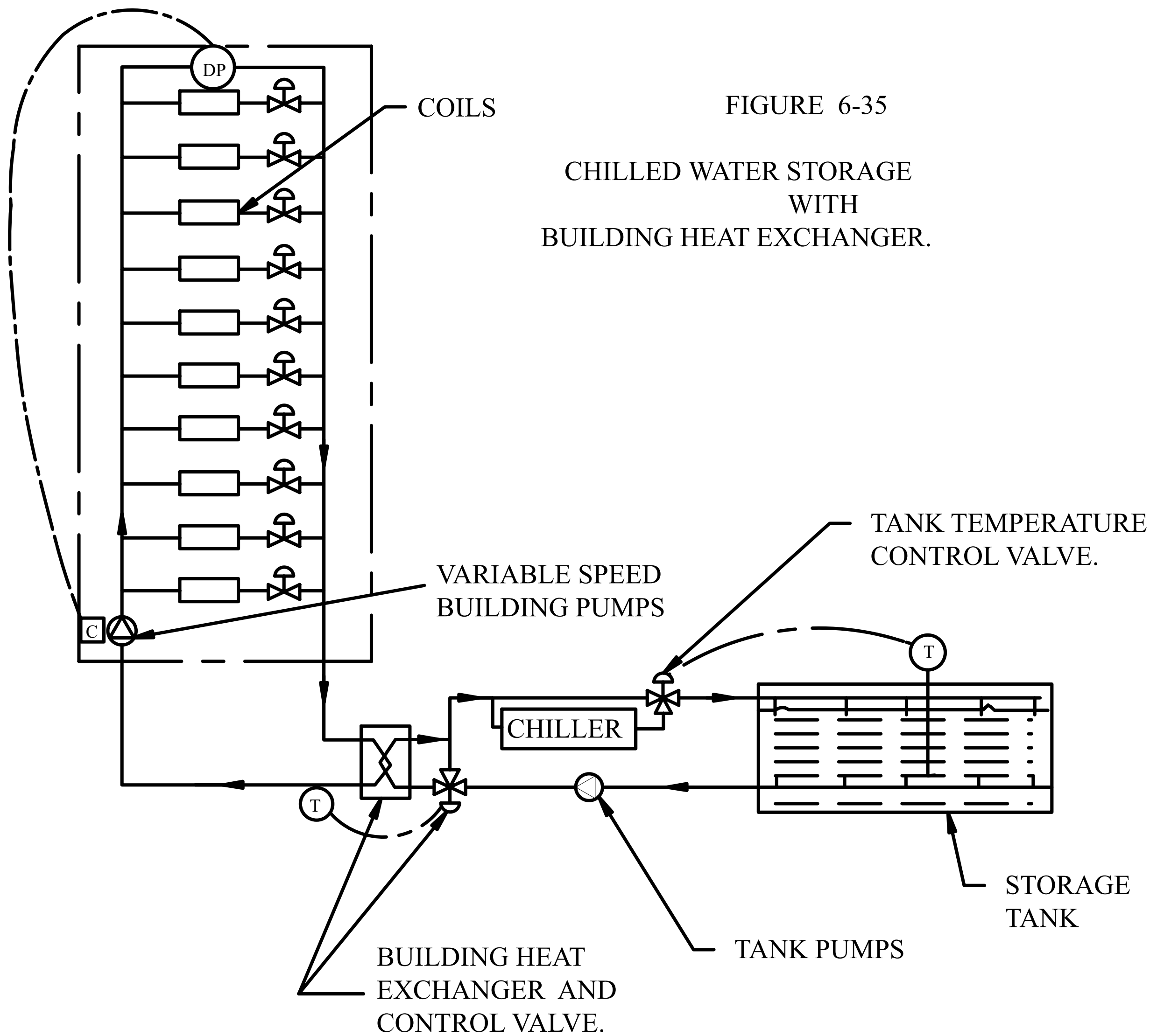


FIGURE 6-35

CHILLED WATER STORAGE
WITH
BUILDING HEAT EXCHANGER.

the water in theses systems. Chilled water systems have been designed for 12°F or 44°F supply and 56°F return water temperature. Poor control of the chilled water has resulted in return water temperatures as low 48°F. The result is that excessive amounts of water have to be pumped to achieve the required cooling. Load. How do we eliminate this problem: by correct system design and proper selection of the size and type of cooling coil control valves. We have emphasized good design in this manual by reducing operating pressures and eliminating energy-wasting devices and procedures. Selection of cooling coil control valves is beyond the scope of this manual. We should discourage methods of artificially controlling the return water temperature, such as the use of return temperature control valves on cross-over bridges as shown in Figure

We should include how we calculate the gpm per ton of cooling or gpm per mbh of heating. Following are the formula:

$$\text{gpm/ ton of cooling} = 12{,}000 \text{ btu/ton} \div (8.34 \text{ lb/gal.} \cdot 60 \text{ min/hr.} \cdot \Delta T)$$

$$= \frac{24}{\Delta T} \tag{6.1}$$

$$\text{gpm/ mbh of heating} = 1{,}000 \div (8.34 \text{ lb/gal.} \cdot 60 \text{ min/hr.} \cdot \Delta T)$$

$$= \frac{2}{\Delta T} \tag{6.2}$$

Where ΔT is the temperature difference, in °F, between the system supply and return temperatures.

Artificial return temperature control can be accomplished by using a return control valve as shown in Figure 6-14. This can insure the design return temperature to the chiller plant, but it creates havoc in the buildings. The supply temperature rises with loss of cooling capacity and unacceptable humidity in the buildings. Loss of differential temperature in these systems must be corrected by positive actions, not by a patch on the system by using the return temperature control valve.

Location of Expansion Tanks and Air Separators:

As shown on the various drawings, in most cases, these tanks should be located on the suction side of the pumps. As shown in Figure 2-3, solubility of air in water increases with system pressure, so the air separator should always be at the point of lowest system pressure.

The exception to this rule is the high rise building, Figure 6-7. The expansion tank should be at the top of the building to insure that adequate pressure always exists at the top heating or cooling coil. Also, the design pressure of the tank will not have to include the static pressure of the building. The air separator should still be at the pump suction with air vents at the top of the building.

Location of Differential Pressure Transmitters:

The differential pressure transmitter is still the best transmitter for controlling pump speed. Since its inception around 1970, it has been used on thousands of buildings. It has worked so well, as it eliminates the distribution friction loss from the control signal. There are a number of typical applications of the transmitters in the figures for this chapter.

Large chilled water systems can require as many as ten differential pressure transmitters for adequate control. Reverse return systems often require differential pressure transmitters at the beginning and ending of the chilled water system, Figure 6-11.

Summary:

This is a synopsis of HVAC water systems that provides general design of these systems. There is much additional detailed information that is needed for each specific installation. An effort has been made to describe equipment and designs that increase the energy consumption of HVAC water systems. It is hoped that designs have been provided that result in lower energy use for chilled and hot water systems.

Chapter 7

CHILLERS, COOLING TOWERS, AND BOILERS

HVAC PUMPS AND SYSTEMS
DESIGN MANUAL

PATTERSON PUMP COMPANY

INTRODUCTION:

Chillers and boilers are two of the principal sources of cooling and heating for the HVAC industry. Cooling towers can also provide evaporative cooling without chillers. Cooling towers along with evaporative and dry coolers are the principal means used to eliminate excess heat.

CHILLERS:

Chillers are those sources of cooling that utilize water as the means of distributing the cooling. Mechanical chillers are manufactured in reciprocating, screw, and centrifugal types; other chillers are absorption and gas fired units. Most of the larger chilled water plants utilize the mechanical types with absorption machines providing electrical peak-shaving capacity. Almost all of the mechanical chillers are electric motor driven; very large, field assembled centrifugal chillers may be diesel engine driven.

Mechanical chillers can be 1) direct expansion (DX) air cooled or 2) water cooled with cooling towers. Most of the larger chiller plants are water cooled due to higher efficiency than DX. The mechanical chiller that utilizes water cooling requires both chiller water pumps for distributing the cooling and condenser water pumps for transferring the rejected heat from the chiller to the cooling tower, Figure 7-1.

Rating of Chillers:

Chillers are rated in tons of cooling. One ton of cooling is the time rate of cooling of 12,000 btu/hr. The performance of a chiller is measured by several technical expressions. The most popular in the U. S. is **kW/ton** which defines the energy consumed by an electric motor driven chiller. The most efficient contemporary chillers have kW/ton values of around 0.50 at design conditions. kW/ton is derived by Equation (1)

$$\text{kW/ton} = \frac{24 \cdot \text{kW}}{\text{gpm} \cdot \Delta T} \qquad (7.1)$$

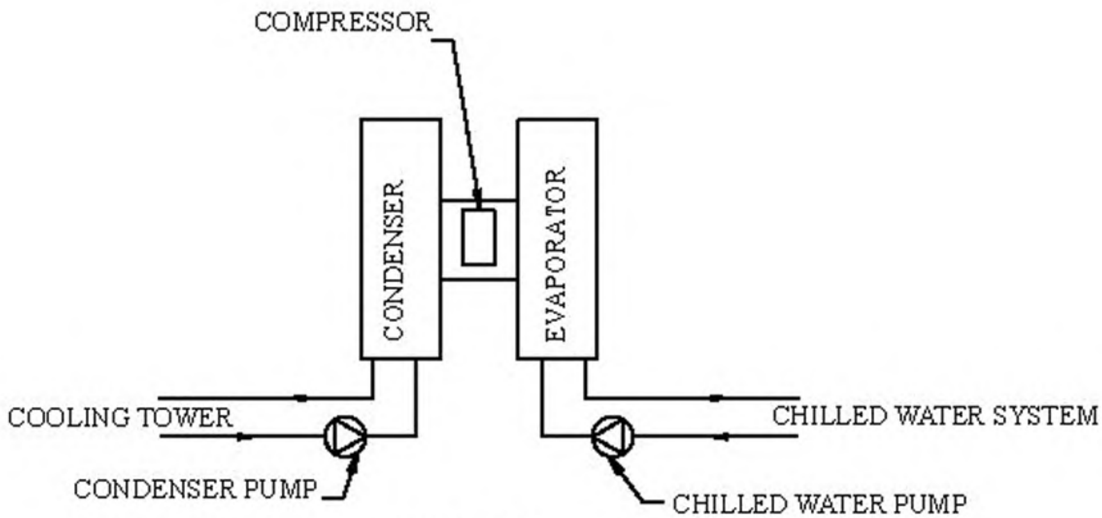


FIGURE 7-1
WATER COOLED CHILLER

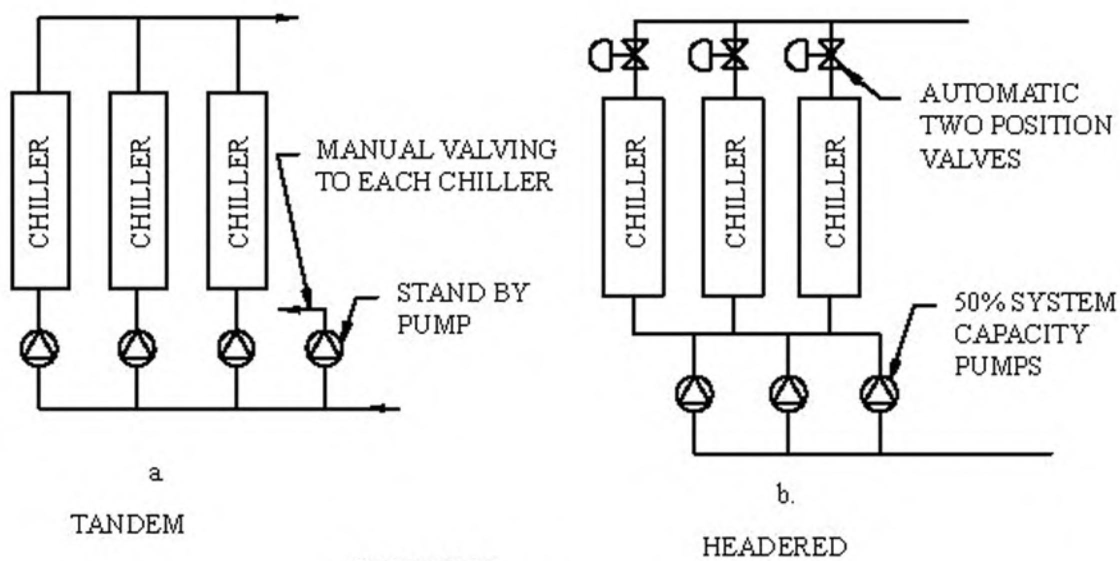


FIGURE 7-2
CHILLED WATER PUMP

CONNECTIONS

From "HVAC Pump Handbook", James Rishel, PE, McGraw-Hill, New York, 1996.

Where: gpm is the chilled water flow rate and ΔT is the temperature difference between the chilled water supply and return temperatures.

Two other values that will be encountered are **COP (Coefficient of Performance)** and **EER (Energy Efficiency Ratio)**.

$$\textbf{COP = Cooling Achieved} \div \textbf{Energy Applied} \qquad \textbf{(7.2)}$$

If a chiller has a rating of 0.55kW/ton, Its COP would be 12,000 ÷ (3,412 · 0.55) or 6.4. The value, 3,412 is the btu equivalent of a kW. This is a dimensionless ratio.

The EER is infrequently used but is similar and measured in btu/watt-hour; it is equal to the cooling achieved in btu/hr ÷ watts. In the above example the EER would be 12,000 btu/hr ÷ 550 watts or 21.8 btu/watt.

Chilled Water Pumps:

Chilled water pumps at one time were sized to a particular chiller and connected in tandem with them, Figure 7-2a. Chillers utilized mechanical control and could not handle any variation in flow through the evaporator. This produced a need for constant volume (flow) systems, Figure 6-16, or primary/secondary pumping, Figure 6-17. Usually, a standby pump was necessary with rather complicated valving. In the early 1990's, chiller control became digital, and the flow through the evaporator could be varied from a minimum such as 3 ft/sec. to a maximum of around 10 ft/sec. Today, some chiller manufacturers allow a minimum flow of 1½ ft/sec with part load on the chiller. This has resulted in variable flow and the development of variable primary pumping, Figure 6-18. Now it is advantageous for the pumps to be headered with automatic shut-off valves on the chillers, Figure 7-2b. The pumps can be selected to find the most efficient number of pumps regardless of the number and size of the chillers. The result has been that many chilled water systems contain three pumps, each with 50% system capacity. This provides the highest pumping efficiency from minimum to maximum chilled water flow.

Assume that a prospective chiller plant is estimated to have a maximum cooling load of 10,000 tons, and five 2,000 ton chillers are contemplated. The cooling coils are to be designed for a differential temperature of 16°F. From Equation 6.1, the system design flow would be 24 ÷ ΔT or 24 ÷ 16 = 1.5 gpm/ton. The maximum flow would be 10,000 tons · 1.5 gpm/ton = 15,000 gpm. Using three 50% capacity pumps, the design flow for each pump would be 7,500 gpm.

The predominant method of pumping chillers today is by use of variable primary pumping, Figure 6-18. The bypass for maintaining minimum flow can be located on either side of the chillers. The important point is the method of controlling the automatic bypass valve. It can be by differential pressure transmitters on the chillers, or it can be by flow meter in the chiller piping, Figure 7-3. Each method has its proponents among the HVAC engineering firms.

A point that is often overlooked is that the flow must be uniform into all of the chillers, Figure 7-4. Otherwise, one chiller will receive more flow than the others, and it will be difficult to pump the chiller plant efficiently. Likewise, connecting chillers together with different friction losses will require modulating valves on the chillers to balance the flow. The modulating valves can be full-open when a chiller with low friction loss is operating alone. Other valve positions can be set for operation with other chillers.

Although primary/secondary pumping is being used less frequently, we must still insure that the chillers are connected properly when it is used. Figure 7-4 describes the correct connection of the open bypass for the chillers. The velocity head in the piping must be used to prevent backflow of warm return water in the bypass. By connecting the piping as shown, there should be no bypassing unless the secondary pumps have more capacity than the primary pumps. This should not be allowed to happen; correct control should prevent this.

Chilled water pumps are almost always volute type. Smaller chiller plants with capacities of less than 500 tons can use end-suction or in-line pumps. The majority of larger chiller plants utilize split case pumps. The advent of the larger in-line pump provides another source for these chilled water pumps.

Condenser (Cooling Tower) Pumps:

Condenser pumps are designed to move the heat generated in the chiller condensers to the cooling towers. Usually, the condenser pumps are constant speed due to the fact that most of the pump head is the static rise across the cooling tower and the friction loss in the condenser. This results in fairly constant head due to little total friction head in the piping. Very large cooling towers remotely located from the chiller plant may justify variable speed.

It is the consensus of most chiller manufacturers that constant flow in the condenser aids the energy consumption of the chiller and reduces the fouling factor. The condenser pumps can, therefore, be tandem or headered with the chiller – similar to the chilled water pumps of Figure 7-1a or b.

It is still assumed that the total heat rejection per ton of cooling is 15,000 btu/hr even though the kW/ton of chillers has been reduced appreciably. Actually 14,000 btu/hr is closer to the current actual value for most chiller water cooled chiller plants. The equation for condenser flow is:

$$\text{Condenser flow – gpm/ton} = \frac{\text{Heat of Rejection/Ton}}{8.34\ \text{lb/gal} \cdot 60\ \text{min/hr} \cdot \Delta T}$$

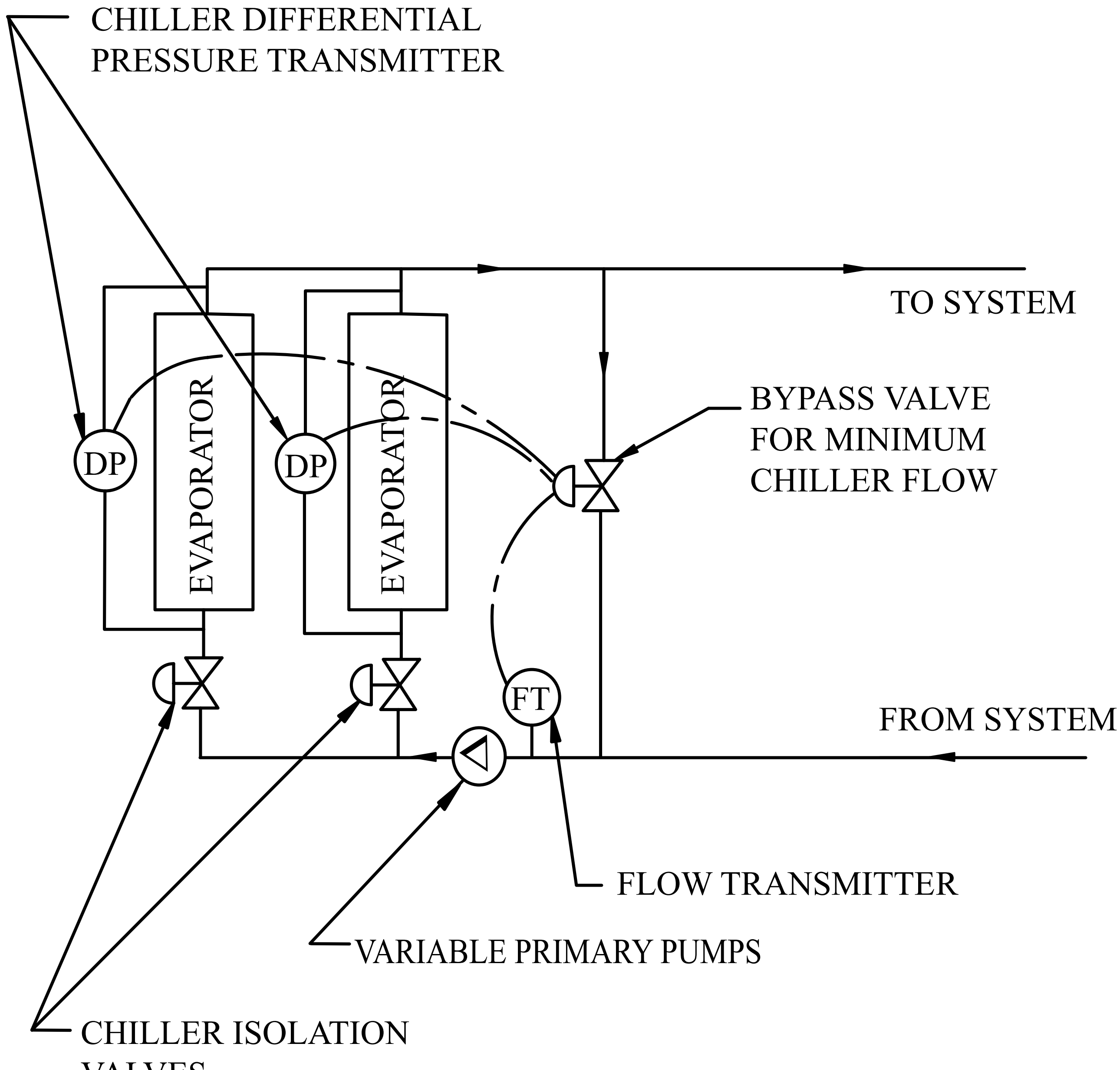
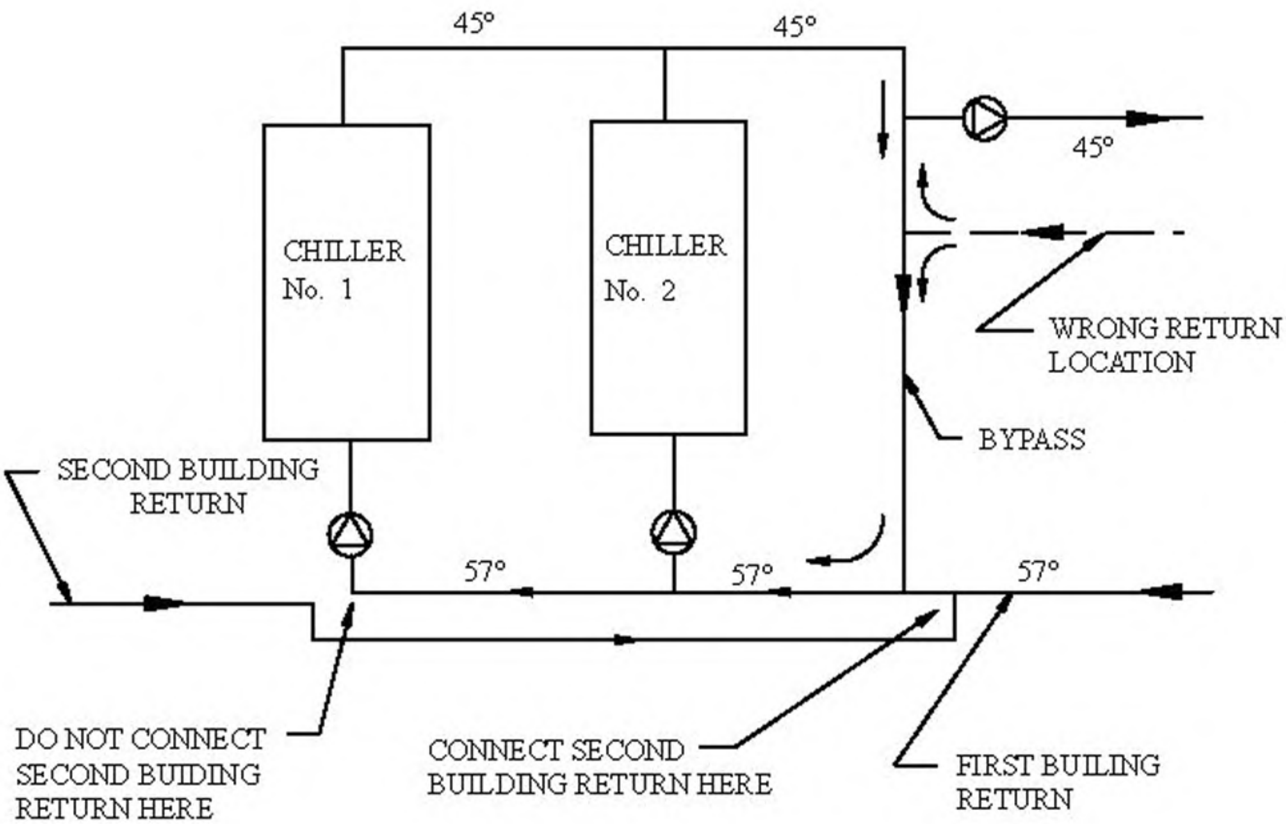


FIGURE 7-3

MINIMUM CHILLER
FLOW CONTROL



RETURN TEMPERATURE MUST BE UNIFORM TO EACH CHILLER REGARDLESS OF LOAD ON SYSTEM OR NUMBER OF CHILLERS RUNNING.

FIGURE 7-4
PROPER CONNECTION OF RETURNS AND BYPASS FOR PRIMARY/SECONDARY PUMPING

From "HVAC Pump Handbook", James Rishel, PE, McGraw-Hill, New York, 1996.

$$= \frac{\textbf{Heat Rejection/Ton}}{\textbf{500} \cdot \Delta \textbf{T}} \qquad \textbf{(7.3)}$$

The standard condenser temperatures have been 95°F return to and 85°F supply from the cooling towers. Recent attempts have been made to change the differential temperature to 15°F, from 100°F to 85°F. Care should be taken to insure that the differential temperature is 10 or 15°F. The temperature of 85°F is the design temperature in most cases. The actual temperature should be verified for each installation.

Condenser pumps are end suction or in-line on smaller chiller plants; larger plants use split case or in-line. The largest plants may use diffuser pumps installed in the sump below the cooling towers, Figure 7-5. As mentioned before, strainers should not be installed on the suctions of condenser pumps.

COOLING TOWERS:

Cooling towers are offered in many different configurations, but the type of cooling tower provided on a particular installation does not affect the performance of condenser pumps. Cooling towers are, basically, constant water flow devices. They function best with constant water flow; with the media full of water, there are fewer chances of blow-through and water spraying out of the tower. This lessens the possibility of Legionnaires disease. With constant fan speed, towers have a limited turn-down of around 10%.

With variable speed fans, the turn-down for a tower is much greater; the problem is coordinating the flow rate with the air flow rate. Since chiller condensers are generally considered to be constant flow devices, the practical solution is to design each tower cell to the chiller capacity. For example, if the condenser flow rate for a 1,000 ton chiller is 2,800 gpm, a tower cell should be designed for 2,800 gpm. If there are four chillers, the cooling tower should have four cells. This makes the simplest piping arrangement. A two position on-off valve should be provided for each cell.

In colder climates where freezing can occur, a bypass valve should be provided, Figure 7-6. During start-up, this valve can be in the bypass position to prevent very cold water from entering the chiller condenser. As shown, the tower piping should drain when the valve is in the bypass position: this is done by opening any cell valves.

Many building installations require remote sumps due to height or freezing. The additional head between the cooling tower and the sump must be included in the condenser pump head, Figure 7-7.

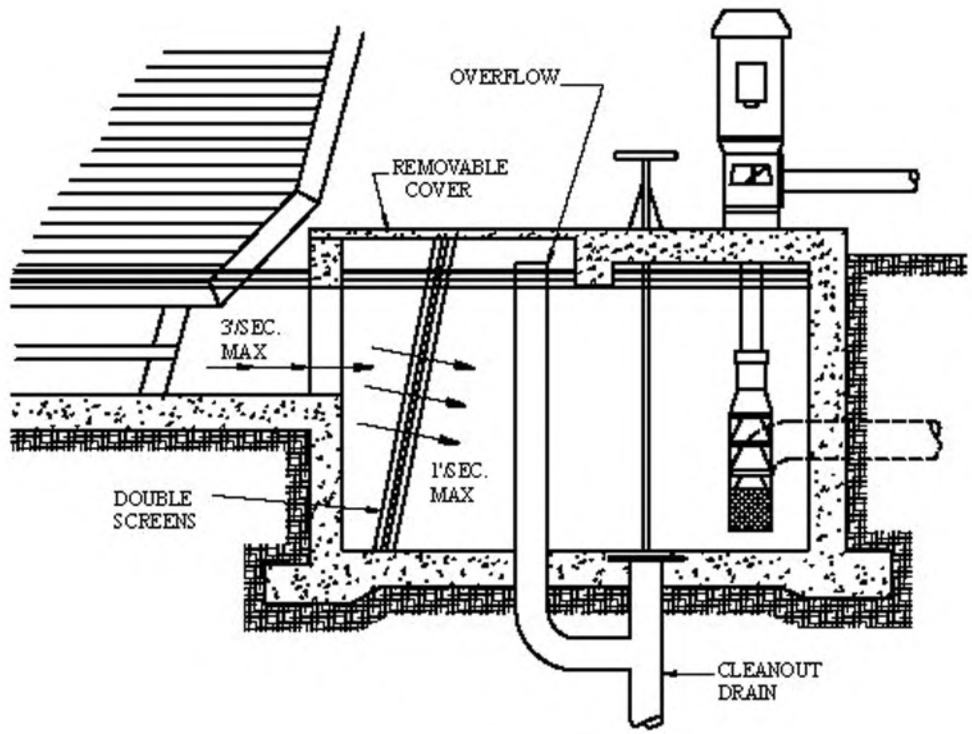
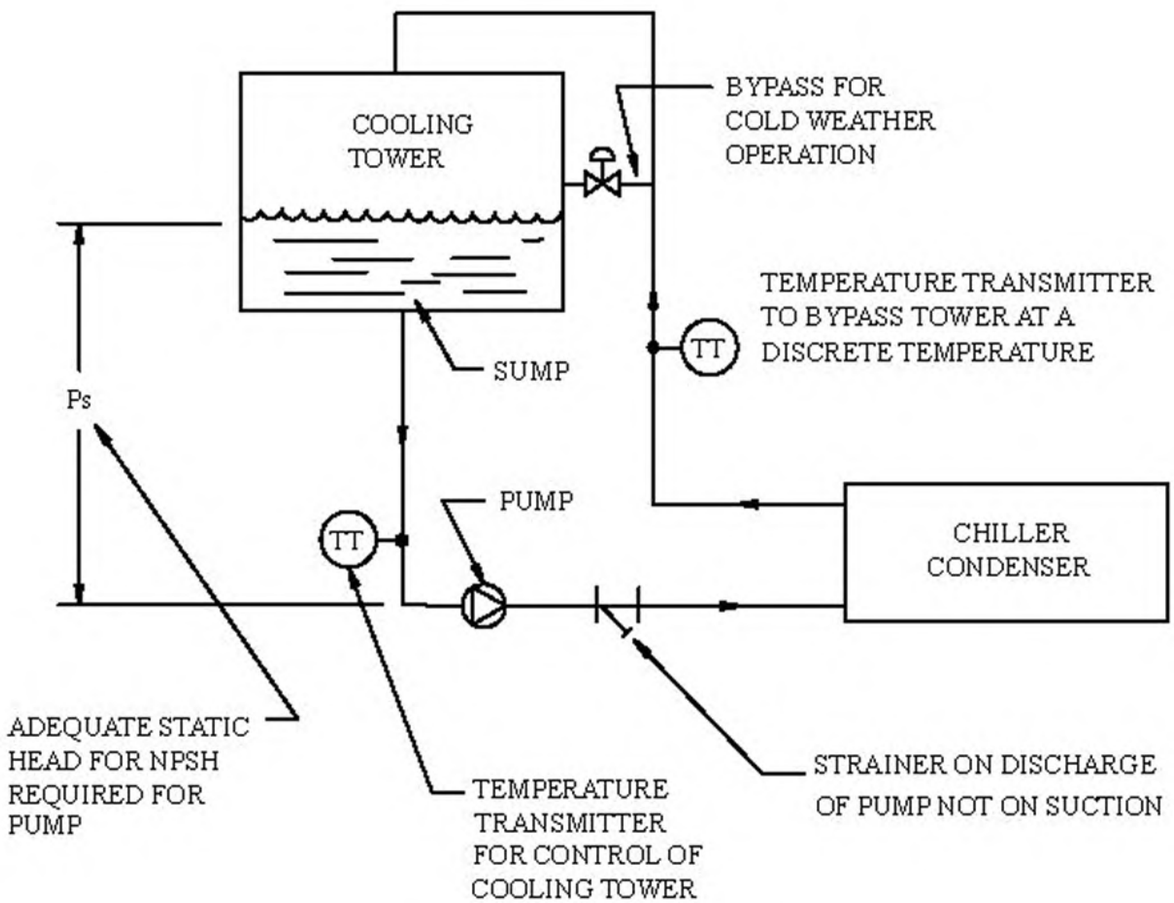


FIGURE 7-5
Typical cross section of Concrete Sump
Source: Marley Cooling Tower Co.
used with permission.

FIGURE 7-6

COOLING TOWER PIPING



NOTE: PROVIDE MANUAL DRAINS FOR REMOVING COOLING TOWER FROM OPERATION DURING COLD WEATHER

From "HVAC Pump Handbook", James Rishel, PE, McGraw-Hill, New York, 1996.

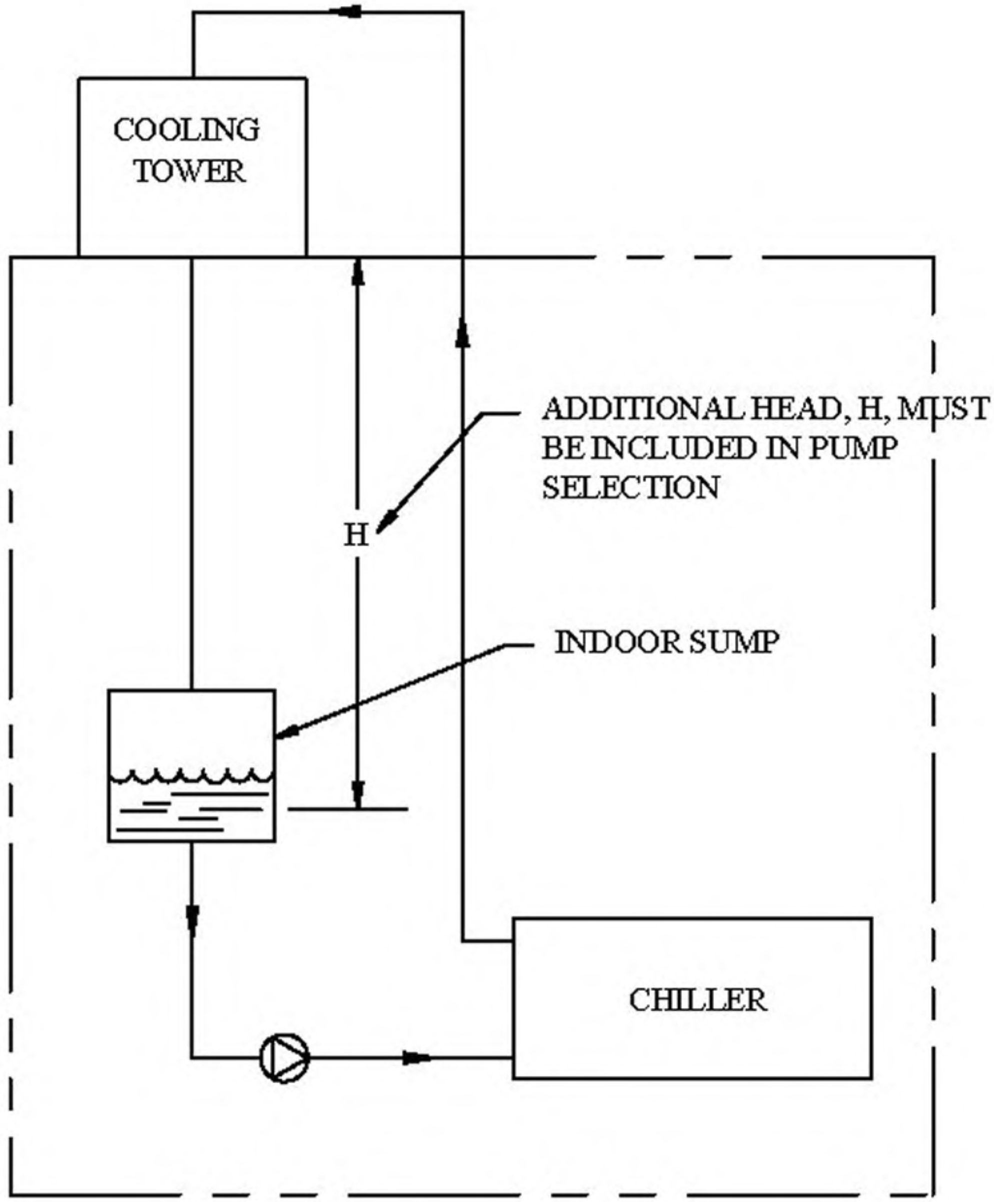


FIGURE 7-7

REMOTE SUMP HEAD

From "HVAC Pump Handbook", James Rishel, PE, McGraw-Hill, New York, 1996.

Larger cooling tower fans should be variable speed, not constant or two-speed. This makes adjustable condenser water temperature control much easier with appreciable energy savings in the chillers.

HEAT RECOVERY:

Realizing that as much heat as 14 to 15,000 btu/hr per ton of cooling is disposed of in cooling towers, efforts have been made to utilize this heat in heating domestic water or for reheat in air handling units. Also, cooling towers can be utilized in free cooling when the cooling tower water is colder than the temperature of the chilled water returning to the chiller plant.

Heat Recover Exchangers:

The most common type of condenser heat recovery is the heat exchanger, Figure 7-8. The condenser water is passed through the heat exchanger when that water is colder than the system return water. The control of this heat exchanger must fit the actual conditions to insure that it is effective and that the chiller condensers are not starved by this use of condenser water. This is an effective procedure for recovering heat when the chilled water return is above 54°F.

Heat Recovery Chillers:

Heat recovery chillers can consist of a double bundled condenser or a special heat recovery chiller. Much work is being done in this field now. It has been discovered that there is no reason to use hot water in the temperature range of 180 to 200°F. for space heating or for heat recovery in air handling units. In reality, water in the range of 110 to 130°F performs very well with increased boiler efficiency and the possibility of heat recovery from condenser water. This is a complex subject as indicated in Figure 7-9 for a heat recovery chiller providing heat to domestic water or the hot water heating system. The control of these chillers varies from job to job and is too detailed for discussion here. It is well that design engineers and pump sales people are aware of this application of centrifugal pumps. Usually, there are four to six pumps on these applications. It is simple duty for centrifugal pumps, as there are no problems with high temperatures or NPSHR.

BOILERS:

Hot water pumps can be a significant market for the end suction and in-line types of centrifugal pumps. The pumps are much smaller than the chilled and condenser water pumps. In selecting these pumps, the operating temperature and pressure must be determined first.

FIGURE 7-8

REVERSE RETURN SYSTEM

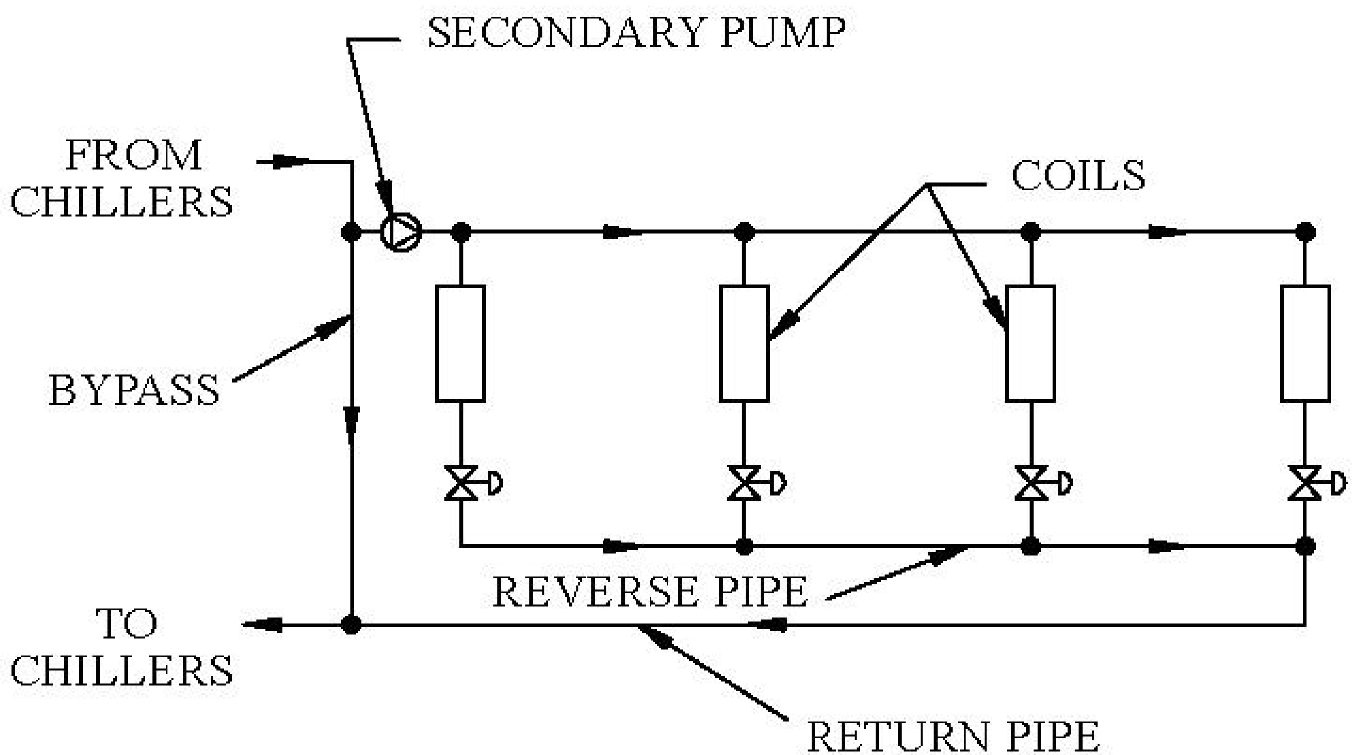
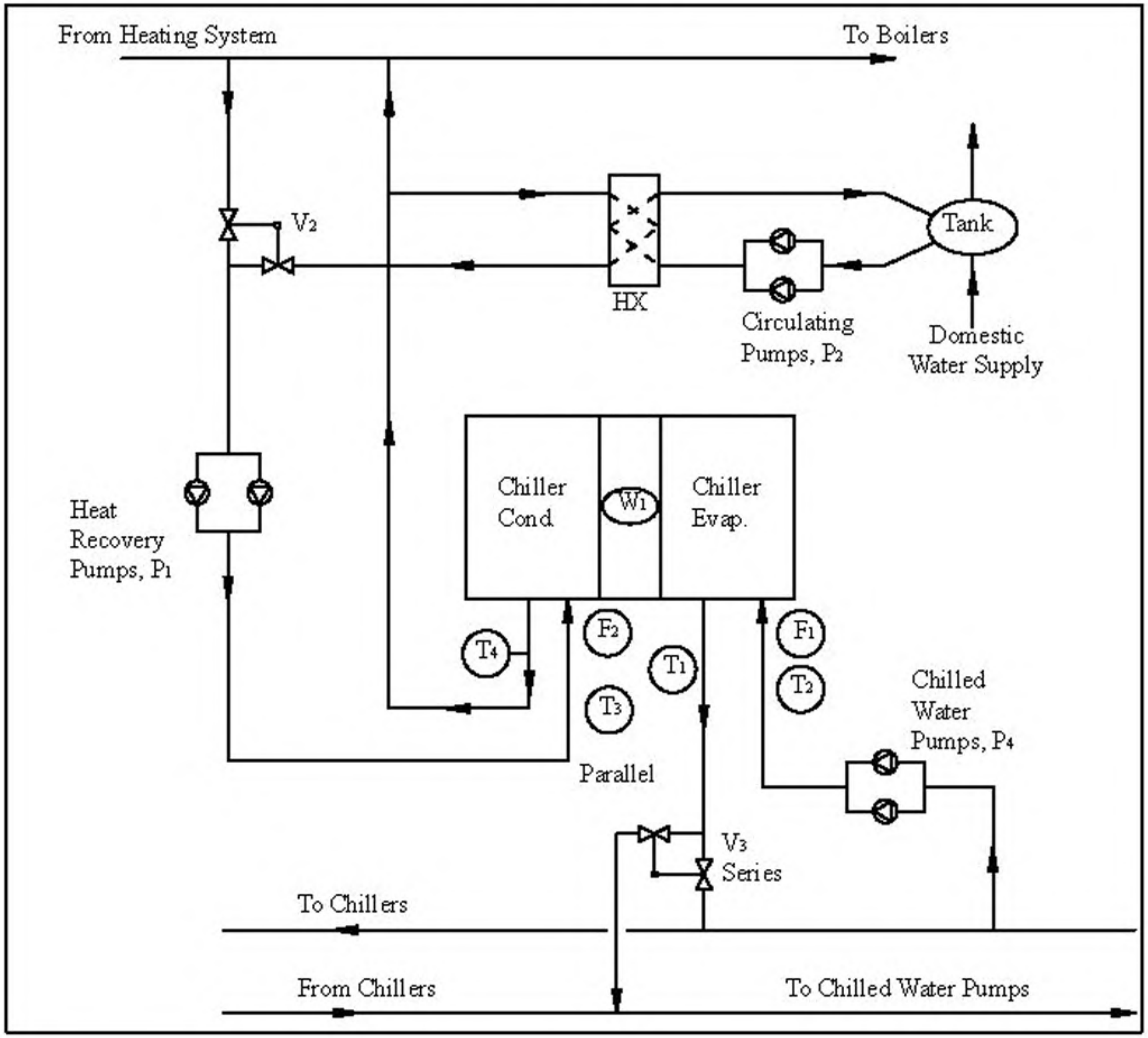


FIGURE 7-9
HEAT RECOVERY CHILLER



From: "Dedicated Heat Recovery", Thomas Durkin, PE and James Rishel, PE, ASHRAE Journal, October 2003.

Boilers are available in low pressure and high pressure as well as low and high temperature. Boilers are designed to the ASME Code for Low Pressure Heating Boilers or for High Pressure Power Boilers. Low pressure boilers are limited to 15 psig steam or 160 psig water and 250°F temperature. Boilers for pressures or temperatures higher than these must be designed to the Power Boiler Code.

Following is a rough selection table for centrifugal pumps for boilers.

Low Temperature Hot Water: End-suction and In-line centrifugal
High Temperature Hot water: Special end-suction centrifugal
Boiler Feedwater Service:
Low Pressure Steam: End suction and in-line centrifugal
High Pressure Steam: Multi-stage, vertical diffuser

There are many types of hot water boilers - water tube, fire tube, and cast iron. There are boilers that run non-condensing and others that run condensing. Condensing boilers operate with flue gas temperatures below 140°F so that the water vapor condenses and provides a much higher efficiency. Some boilers have minimum flow rates; others do not. It is the responsibility of the water system designer to specify the correct operating procedure for the pumps that will satisfy the requirements of the boilers.

NPSH is not a problem with hot water boilers, but it can be serious with boiler feed pumps for steam boilers. Feed water is taken from open tanks or deærators where the water may be near saturation temperature. Low NPSHR pumps are usually required. The static height of the tank water level and the friction in the pump suction piping must be checked to insure that the NPSHR is adequate.

Rating of Boilers:

Hot water boilers are rated in thousands of btu/hr, **mbh.** The gpm for boilers operating in the 130 to 180°F range is calculated using Equation 6-2. A boiler with a rating of 1,000 mbh and a differential temperature of 40°F would have a flow rate of $(1{,}000 \cdot 2) \div 40$ or 50 gpm. This assumes that the density of water is 62.3 lb/ft^3.

High temperature water boilers can be rated in pounds of water per hr. The pumps are often selected on cold water specific gravity since the water must be circulated initially before heating. With variable speed, the reduced specific gravity and viscosity will result in lower speed at design temperature. The pumps must be designed physically to operate at the design pressure and temperature. The optimum pump for this service is the end suction, center-mount, petroleum pump; Patterson Pumps may have special end-suction pumps available for this service.

Steam boilers are rated in pounds of steam at a specific temperature such as 212°F. In the past, boilers were rated in boiler horsepower. One horsepower was equal to 34.5 pounds of steam at 212°. The cold water boiler feed rate, in gpm is:

$$\text{Boiler feed, in gpm} = \frac{\text{Pounds of Steam/hr}}{8.34 \text{ lb/gal} \cdot 60 \text{ min/hr} \cdot 0.9}$$

$$= \frac{\textbf{Pounds of Steam/hr}}{\textbf{450}} \qquad \textbf{(7.4)}$$

If the boiler is rated in boiler horsepower:

$$\text{Boiler feed, in gpm} = \frac{\text{Boiler horsepower} \cdot 34.5}{8.34 \text{ lb/gal} \cdot 60 \text{ min/hr} \cdot 0.9}$$

$$= \frac{\textbf{Boiler Horsepower}}{\textbf{13.0}} \qquad \textbf{(7.5)}$$

Where: 0.9 is a contingency factor for steam quality and leakage.

There may be people who use a higher contingency factor. This should be checked with the designer of the boiler system. Also, high pressure boilers with superheated steam require very careful calculation of the boiler feed, but they are seldom found in the HVAC industry.

This is a resume of chillers, cooling towers, and boilers. There are many HVAC texts that can provide much greater information on this equipment.

Chapter 8

SELECTION, TESTING, AND OPERATION OF HVAC PUMPS

HVAC PUMPS AND SYSTEMS
DESIGN MANUAL

PATTERSON PUMP COMPANY

INTRODUCTION:

Proper pump selection and operation is important to achieve minimum energy consumption and long pump life with little maintenance. This chapter emphasizes this by summarizing material provided in the previous chapters and applying it to actual HVAC systems. This general review will include applying figures such as Figure 6-3 to chilled water distribution systems. It is so important that HVAC pumps be selected near the Best Efficiency Point. Even though we emphasize this, we still find many pumps being selected back on their curves at high radial thrust conditions.

SELECTION OF PUMPS:

HVAC water systems are generally high flow, low head systems. Typical are chilled water systems where the flow can be 10,000 gpm at a maximum head of 100 ft. Most HVAC water systems are variable flow where the flow can drop to less than 10% of the design flow. This causes HVAC pumps to be installed in parallel for turn-down capability as well as standby capacity. As many as eight or ten pumps can be installed in parallel on the larger installations. (Remember that installing pumps in parallel adds their capacities at the system design head.) For smaller installations, it has been found that three pumps, each with 50% capacity, often provide maximum security and higher operating efficiency than two 100 % capacity pumps. (Also remember that we no longer size chilled water pumps to the capacity of individual chillers.)

Pump Selection Points:

As mentioned before, it was traditional to select constant speed pumps to the left of the BEP to provide some room for run-out, Point 1 in Figure 8-1. Obviously, the wrong place to select a pump is at Point 2 where the pump operates noisily at high radial thrust and poor efficiency. With variable speed pumps, the pumps can be selected at Point 1 on Figure 8-2. The reason for this is the fact that today's control of variable speed pumps and variable frequency drives eliminates the possibility of run-out and overloading the motor. The running limit on the drive prevents the motor from taking more current than the allowable current for the drive. In this case, the motor horsepower at condition is 58.8 hp, so a 60 hp motor and drive can be used in lieu of the non-overloading motor of 75 hp. In the

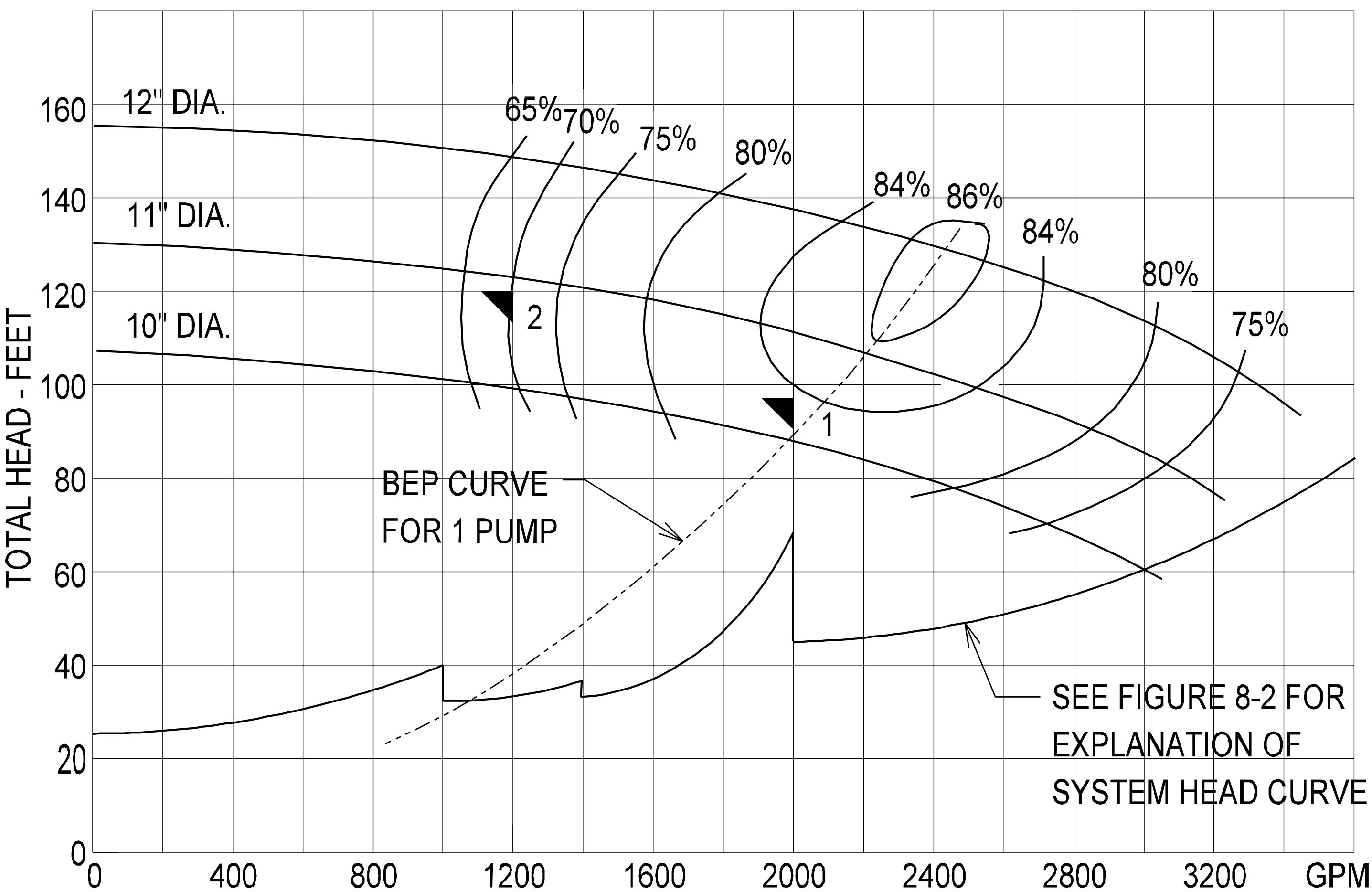
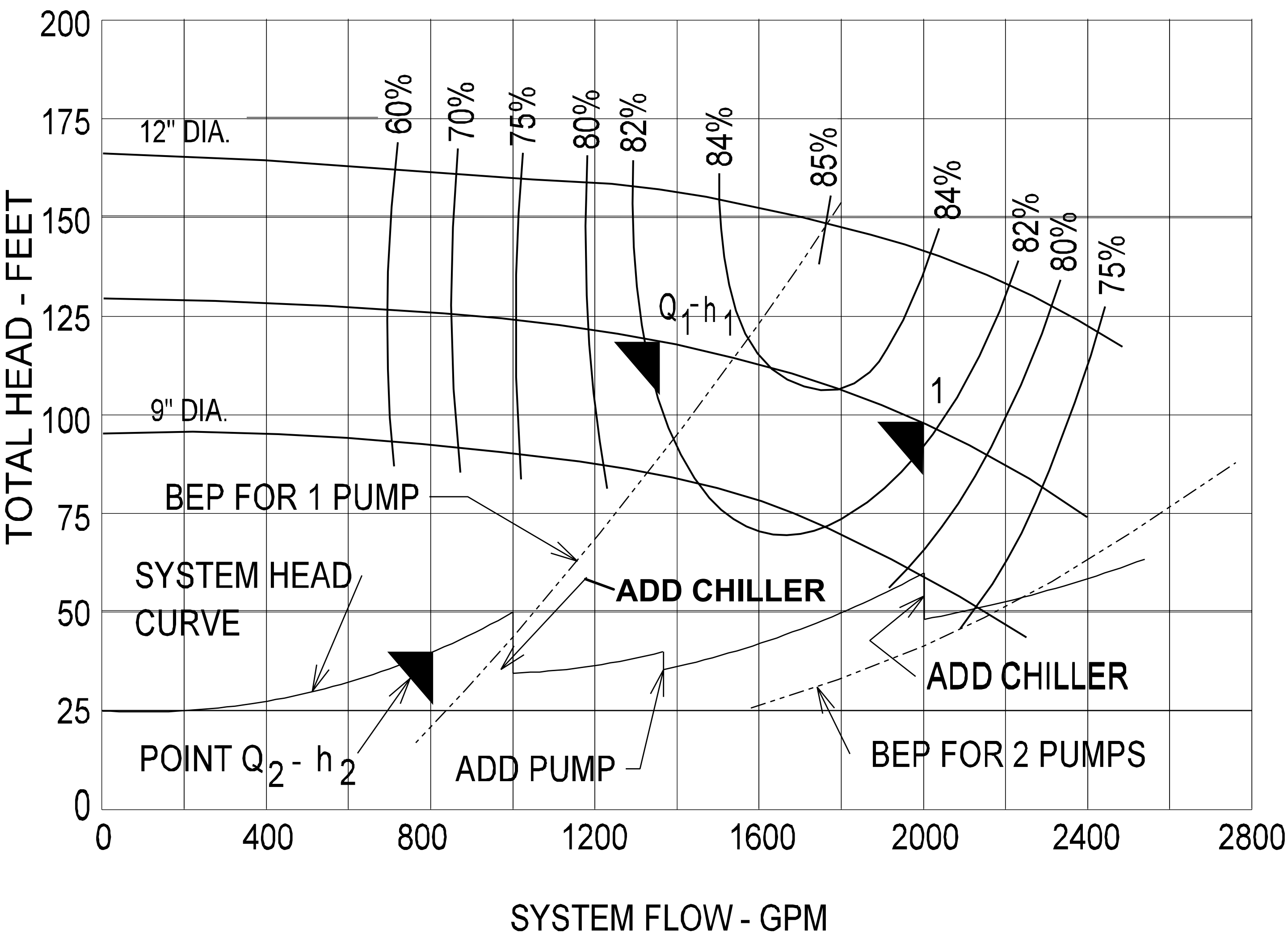


FIGURE 8-1. Traditional selection of pump



PATTERSON 8X6 MI-B
1770 RPM

FIGURE 8-2.

past, the pump with a rating of 2,000 gpm @ 96 ft. head would be picked to the left of the BEP as shown in Figure 8-1, and it would be equipped with a 75 hp motor. A 60 hp motor can also be used on this application, but the pump and its installation would cost more than the pump described in Figure 8-2; in this case the smaller pump is around 70% of the cost of the larger pump. The decision to use which pump would hinge upon the energy saving with the larger pump and its slightly greater efficiency over the additional cost. (See System Head Curve below for explanation of this curve shown on Figures 8-1 and 8-2.)

Generation of a kW input Program:

kW input calculations and control for multiple pumps operating in parallel can produce the lowest possible energy consumption. These calculations provide a guideline for the selection of the correct size of pumps. After selection of efficient pumps, they are sequenced automatically to insure that the correct number of pumps is in operation at all flows in the water system.

The kW Input program is based upon the pump affinity laws that state: 1) the flow varies directly with the speed and 2) the head varies with the square of the speed. So,

$\frac{Q_1}{Q_2} = \frac{S_1}{S_2}$ and $\frac{h_1}{h_2} = \frac{S_1^2}{S_2^2}$, and these two equation can be combined to to eliminate the speed, S.

The result is Equation 8.1:

$$\mathbf{Q_1 = (Q_2 \cdot h_2 \div h_1)^{0.5}} \qquad \mathbf{(8.1)}$$

Where:

Q_1 and h_1 define an unknown equivalent point of operation on a known pump curve operating at a known pump speed, Figure 8-2.

Q_2 and h_2 define a known point of operation on a system head curve at a speed other than the above known pump speed, Figure 8-2.

kW Input programs are preferable to wire-to-water efficiency programs, since a differential pressure transmitter is not required across the pumping system headers when using kW Input. The same energy results are achieved with greater instrument accuracy.

Information Required:

Patterson Pumps can provide a complete kW Input Report if the following information is provided them.

System Description: List type of system such as chilled water. If hot water, specify operating temperature.

Design Flow and Pressure: Provide minimum and maximum system flows and possible operating pressures.

Pumps: 1) No. of pumps; 2) pump duty, namely flow in gpm and head in ft.

Chillers or Boilers: Provide the number of chillers or boilers and the minimum and maximum flow for each.

System Friction Losses – in feet: 1) Water system distribution loss, 2) differential pressure set point

Equipment Losses – in feet: 1) Pump fitting losses at pump design flow, 2) chiller or boiler loss at design flow.

System Head Curve:

The system head curve is developed for the proposed installation. As discussed in Chapter 5, system head curves in the HVAC industry can be complicated. It is not the smooth system friction curve often seen in municipal pump installations. Sharp changes in the curve occur as pumps or chillers are added and subtracted. The system curve shown in Figures 8-1 and 8-2 is based upon the following information:

> Assume that the design flow for a chilled water system with four chillers is 4,000 gpm, and the pumping system has three equal pumps, each with a capacity of 2,000 gpm. Assume the system losses are 45 ft. for the distribution piping and 25 ft. differential pressure. The equipment losses are 8 ft. for the pump fitting losses at 2,000 gpm, and the chiller loss is 18 ft. at 1,000 gpm.
>
> The pump total dynamic head in this case is 45 + 25 + 8 + 18 = 96 ft.

Objectives: 1) Compute the kW input from minimum to maximum flow with one, two, and three pumps running at flow increments of 5% of maximum flow. 2) Make a comparison between the three pump kW inputs and select the optimum number of pumps for each system flow percentage.

Basis of Computations:

1. These programs utilize the affinity laws for pumps and Equation 8.1 above for computation of pump operating points.

2. For each increment of flow, by trial and error, assign values for h_1 until the point Q_1, h_1 lies on the known pump curve, Figure 8-2. Once this point is determined, the pump speed and efficiency can be secured for that point on the system head curve. The pump speed is the ratio of $(Q_1 \div Q_2)$ x S where S is the known curve speed, in this case 1770 rpm. The pump efficiency can be read directly from the pump curve - 82.5% in this case.

3. Remember that a chiller is added when the system flow reaches the maximum allowable flow through a chiller, 1,000 gpm.

This is the basic program for securing optimum operation for a number of pumps operating in parallel.

Patterson Pumps can provide a kW input evaluation for a prospective or existing HVAC water system upon receipt of the above information along with the description of the project and the design engineer's name.

TESTING HVAC PUMPS:

Testing of HVAC pumps accomplished two objectives. It proves that the pump is sound mechanically and that it meets certain performance criteria. Factory testing is far superior to field testing, as it is difficult to establish accurate instrumentation in the field. Almost all pump manufacturers are familiar with these testing standards established by the Hydraulic Institute and are capable of testing their pumps in accordance with them.

There are two basic Hydraulic Institute test standards for centrifugal pumps. ANSI/HI 1.6-2000 is titled “Centrifugal Pump Tests”, but it is basically for volute type pumps. ANSI/HI 2.6-2000 is titled “Vertical Pump Tests”, but it is for all diffuser pumps - turbine, mixed flow, and axial flow (propeller), whether installed horizontally or vertically. These standards establish criteria for the pumps themselves as well for the quality of the instrumentation and its installation.

The test stand should conform to the requirements listed in these standards. Since their publication in the year 2,000, an additional requirement has emerged for the instrumentation. NIST (National Institute of Standards and Technology) has developed a traceability program so that actual test stand instrumentation accuracy can be traced to the accuracy of instrumentation at NIST. This insures the accuracy of the test instrumentation.

Types of Tests:

The types of tests as listed in these standards are:

1. Performance test to demonstrate hydraulic and mechanical integrity

2. Hydrostatic test of pressure-containing components

3. NPSHR tests

4. Mechanical test

5. Priming time test

(For airborne sound testing, see HI 0.1-9.5-2000)

Of these, only performance and hydrostatic testing are needed for most HVAC pumps. The NPSHR and Priming tests should be conducted for self priming pumps which are used infrequently on HVAC water systems.

Performance Tests:

Volute type pumps have two levels of test, “A” and ”B”. Level A is basically for pumps custom designed for a specific application. Level B tests are for production type pumps normally encountered in the HVAC field. Level A test should be required for large pumps designed for a particular installation while Level B tests should be specified for most HVAC pumps.

Normally, an HVAC pump is specified at a particular flow and head with a specified efficiency. The factory test would be conducted at that flow and head with confirmation of the pump efficiency. Other specifications may call for more than one point in the test, but that is really unnecessary for HVAC pumps. Most large pumps in this industry are variable speed; the exception is condenser pumps which usually operate near their specified flow, head and speed.

The important thing about testing HVAC pumps is that the manufacturer’s test stand complies with the HI standards and that its instrumentation is NIST traceable for accuracy and repeatability

Hydrostatic Tests:

Hydrostatic tests confirm the structural integrity of the pump and insure that it will perform satisfactorily under the pressures of the water system. The hydrostatic test pressure must be not less than the greater of the following.:

150% of the pressure which would occur in that part when the pump is operating at rated speed for the given application of the pump.

125% of the pressure which would occur in that part when the pump is operating at rated speed for a given application, but with the pump discharge valve closed.

All Patterson HVAC pumps will be hydrostatically tested in accordance with this HI Standard.

OPERATION OF PUMPS:

The emphasis in this manual has been energy conservation in pumping HVAC systems. Our first step is to eliminate energy-consuming devices such as balance valves which were necessary in the days of constant speed pumping. Next, we must be sure that we are operating the pumps as efficiently as possible, The kW input program above helps us determine how to program the pumps to achieve the lowest possible kW input at all loads on the HVAC water system.

Once we have established the programming for the pumps, it is quite simple to maintain that program with contemporary digital control. There are some simple procedures to follow that will insure that that program Is maintained.

1. The failure program must be established correctly. If a pump fails, the next pump in sequence must start immediately, not waiting for the system pressure to fall and indicate a failure. This is easy to do with today's digital control.
2. Each pump should be equipped with a differential pressure switch that stops the pump when it proves the pump's inability to establish or maintain differential pressure across its suction and discharge.

3. Pumps should not be alternated automatically. It has been proved that alternating pumps for equal wear has proved dangerous, as all of the pumps wear uniformly and can wear out at the same time. Wear is not a concern with properly designed and selected HVAC pumps. As indicated elsewhere in this manual, these pumps run for over twenty years without any significant replacement of parts. Manual selection of lead and lag positions for pumps operating in parallel is adequate for HVAC pumps.

4. Each pump should be equipped with a hand-off-auto switch as well as a manual selection switch. The latter is used to select the lead-lag positions for the automatic additional and subtraction of the pumps.

5. Run and fail indication are useful devices for the operation of these pumps.

Unequal Pumps in Operation:

All of the examples of pump operating in parallel so far have been for equal size pumps. Usually, this is the case, as it is easy to program the pumps on and off. On retrofit situations, different sizes of pumps may be encountered. Also, where there are many hours of operation at low flows in the HVAC system, it may be economical to install a small pump for such operations.

There are two cases that must be reviewed for unequal pumps: 1) the smaller pump must operate with the main pumps, and 2) the small pump never operates with the main pumps. It is important that this decision be made before any pumps are selected.

Consider first the case where a small pump must operate with large pumps. If there is an existing pump with a capacity of 500 gpm and it is desirous to add a new pump with a capacity of 750 gpm, the most important characteristic that must be addressed is the shut-off head of the existing pump. The new pump must have a shut-off head approximately the same as the existing pump, Figure 8-3. This enables the pumps to run together without any concern that one of the pumps could be run to the shut off condition with the resulting inefficiency and possible damage. There may be some sacrifice in efficiency for the new pump to insure compatible operation. To illustrate this point, no consideration was given to a possible increase in friction in the system due to the addition of the larger pump. Just adding a pump may not provide much new capacity. The system head curve must be computed for the two pumps' capacity to see if there is added friction loss to the system.

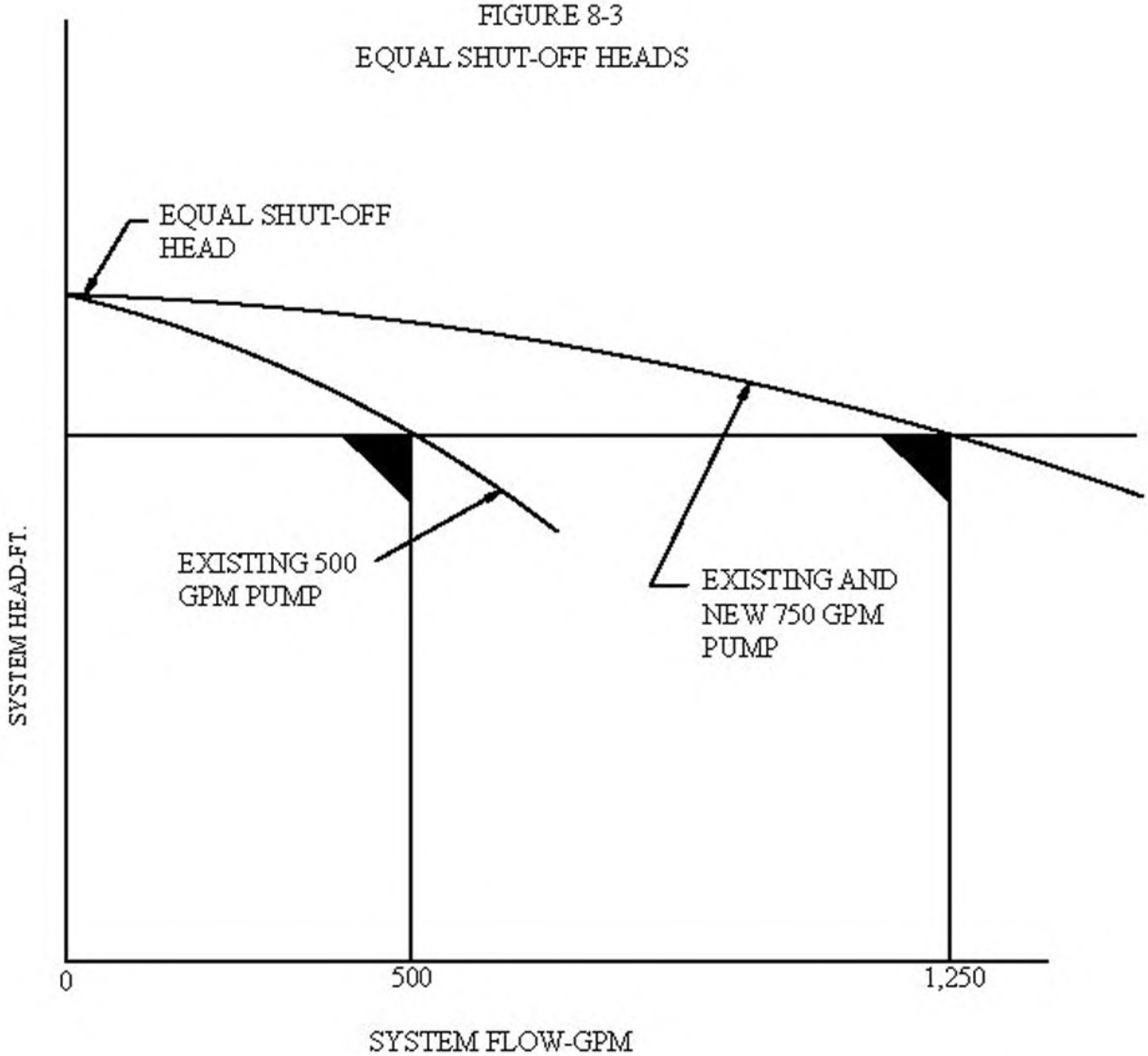
The second case involves the use of a small pump that runs alone to satisfy system operation at very low flows. This does not occur often, but it should be reviewed. For example, the system head curve of Figures 8-1 and 8-2 indicated possible operation around 500 to 800 gpm for extended periods of time. No longer is it necessary for the pumps to have the same shut-off head. Figure 8-4 describes a smaller pump with a capacity of 800 gpm at 40 ft. head requiring only a 10 hp motor. This pump would offer efficient operation at the minimum flow conditions of the water system. Generally, only one small pump is installed. If it needed servicing, one of the larger pumps could operate the system until the small pump is repaired.

GENERAL COMMENTS:

Constant and Variable Speed Operation:

The use of variable speed pumps on HVAC systems has become so common now that we no longer have to worry about warning people not to try to run constant and variable speed pumps together. In the past, this was tried with disastrous results. Pumps would run at shut–off head and be damaged. The

FIGURE 8-3
EQUAL SHUT-OFF HEADS



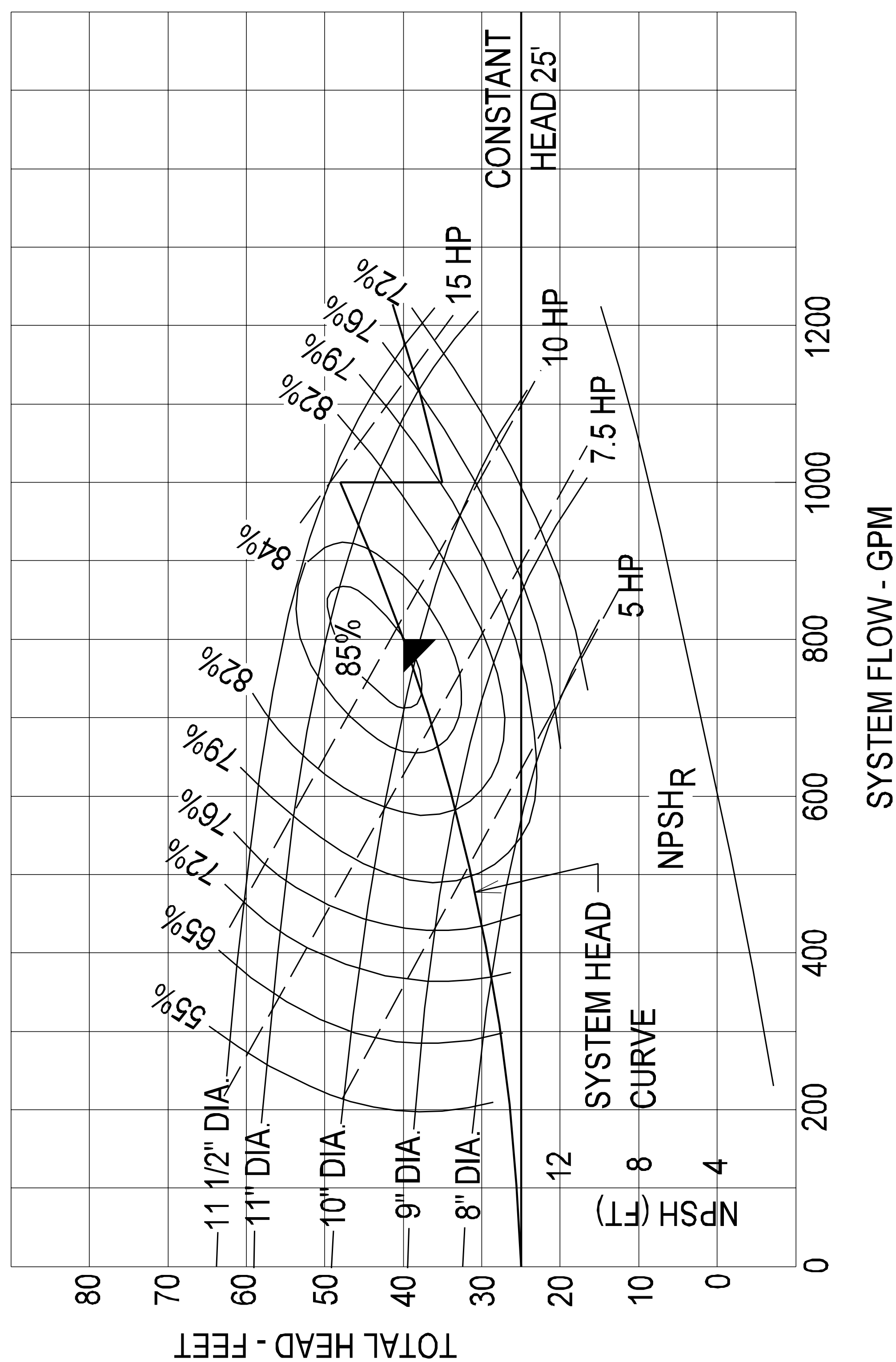

TOTAL HEAD - FEET
SYSTEM FLOW - GPM
0
10
20
30
40
50
60
70
80
0
200
400
600
800
1000
1200
11 1/2" DIA.
11" DIA.
10" DIA.
9" DIA.
8" DIA.
55%
65%
72%
76%
79%
82%
84%
85%
82%
79%
76%
72%
15 HP
10 HP
7.5 HP
5 HP
CONSTANT
HEAD 25'
SYSTEM HEAD
CURVE
NPSH (FT)
12
8
4
$NPSH_R$

FIGURE 8-4

overall efficiency of the pumping system was very poor. Today, it is taken for granted that each pump has to have its own variable speed drive and that all of them are variable speed. Attempts to run more than one pump motor off a single drive are no longer encountered.

Frequency of Repair:

In the past, HVAC pumps were subjected to needless repair. This was due to constant speed, constant volume systems where the pumps operated continuously at full speed and flow. Also, pumps were often selected at points of high radial thrust. Further, water treatment in HVAC systems was from non-existent to minimal.

With today's water treatment and the wise selection and operation of pumps, HVAC pumps should last for long periods of time with little repair. There are many instances of variable speed HVAC pumps lasting beyond 20 years service with little repair.

Chapter 9

PACKAGED PUMPING SYSTEMS

HVAC PUMPS AND SYSTEMS
DESIGN MANUAL

PATTERSON PUMP COMPANY

INTRODUCTION:

Packaged pumping systems have been in use in the HVAC field for the past thirty years. They have proved to be an economic source for pumping equipment on new buildings or for existing buildings where access is available to the chiller plant. These pumping systems can be provided in sizes up to those limited by highway transportation. Some units are built in sections and assembled at the jobsite.

TYPES OF PUMPING SYSTEMS:

Packaged pumping systems have been built for hot, chilled, and condenser water. These systems are available in small two pump units with end suction pumps, Figure 9-1, up to large systems with split case pumps, Figure 9-2. Other systems are available with in-line pumps, Figure 9-3. Pumping systems can be furnished with the expansion tank and air separator, Figure 9-4. Pump houses, Figure 9-5, can be provided where room is not available in existing plants. Complete chiller plants are available, Figure 9-6, where added chiller capacity is needed, and no room exists for new chillers.

Special systems have been built for high temperature water as well as steam heated exchangers for hot water, Figure 9-7. Other packaged pumping systems have been built for cooling tower installations that do not utilize chillers; packaged pumping systems provide compact pumping installations for geothermal projects.

ADVANTAGES FOR PACKAGED PUMPING SYSTEMS:

There are a number of advantages for factory assembly of pumping systems. Obviously, these vary from job to job. Here are the most significant.

1. First cost. Careful evaluation of actual job costs has proved that the savings due to factory assembly vary from 10 to 15% over field construction. Here are the reasons:

a. Labor. Fewer man hours are required due to experience of the workers with pumps, fittings, valves, etc. Shop material handling equipment speeds up the

FIGURE 9.1

FLO-PAK
Atlanta Georgia
A BUSINESS UNIT OF PATTERSON PUMP

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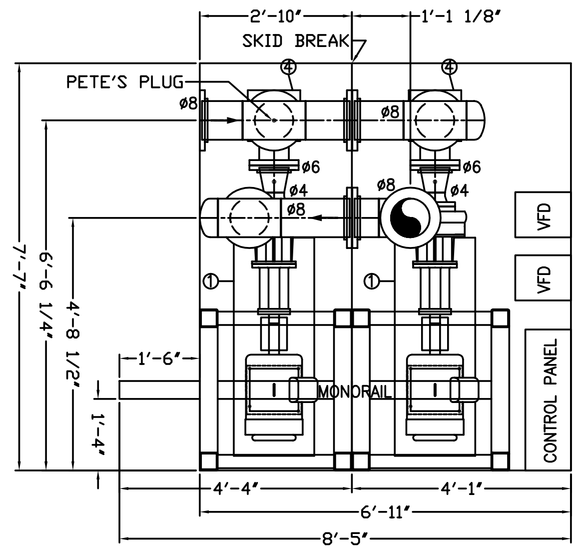
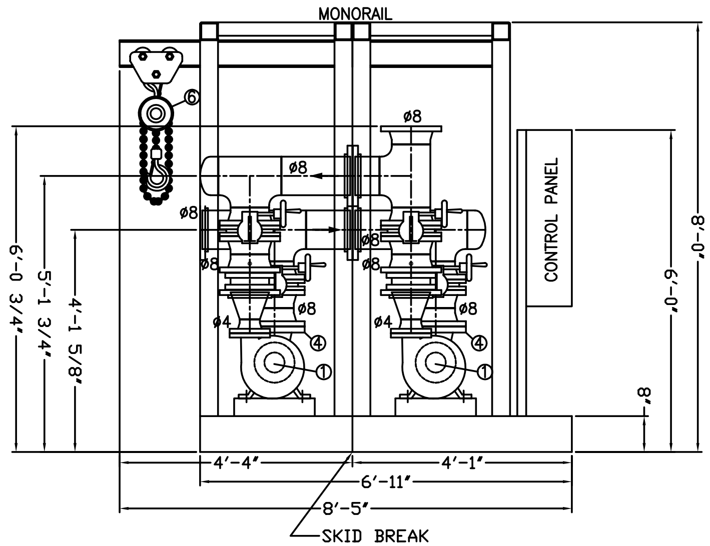
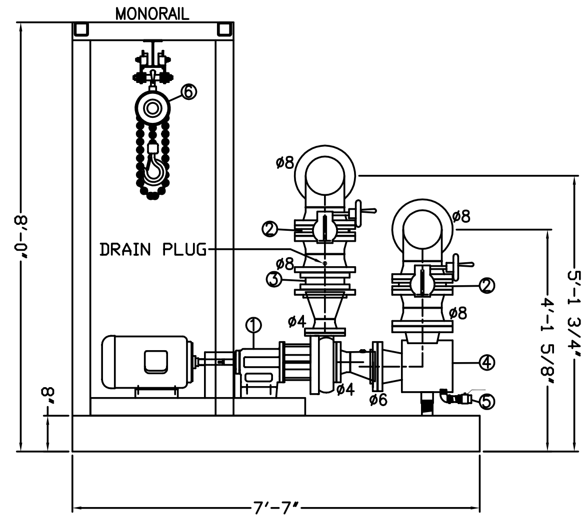


PLAN VIEW



FRONT ELEVATION



RIGHT ELEVATION

KEY LEGEND

KEY	DESCRIPTION
1	PATTERSON 4x4x7 40HP 1750RPM
2	8" LUG B'FLY VALVE
3	8" CHECK VALVE
4	8" x 6" PATTERSON SUCTION DIFFUSER
5	1" NIBCO #T-580 BALL VALVE
6	1/2 TON TROLLEY & HOIST

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FIGURE 9.2

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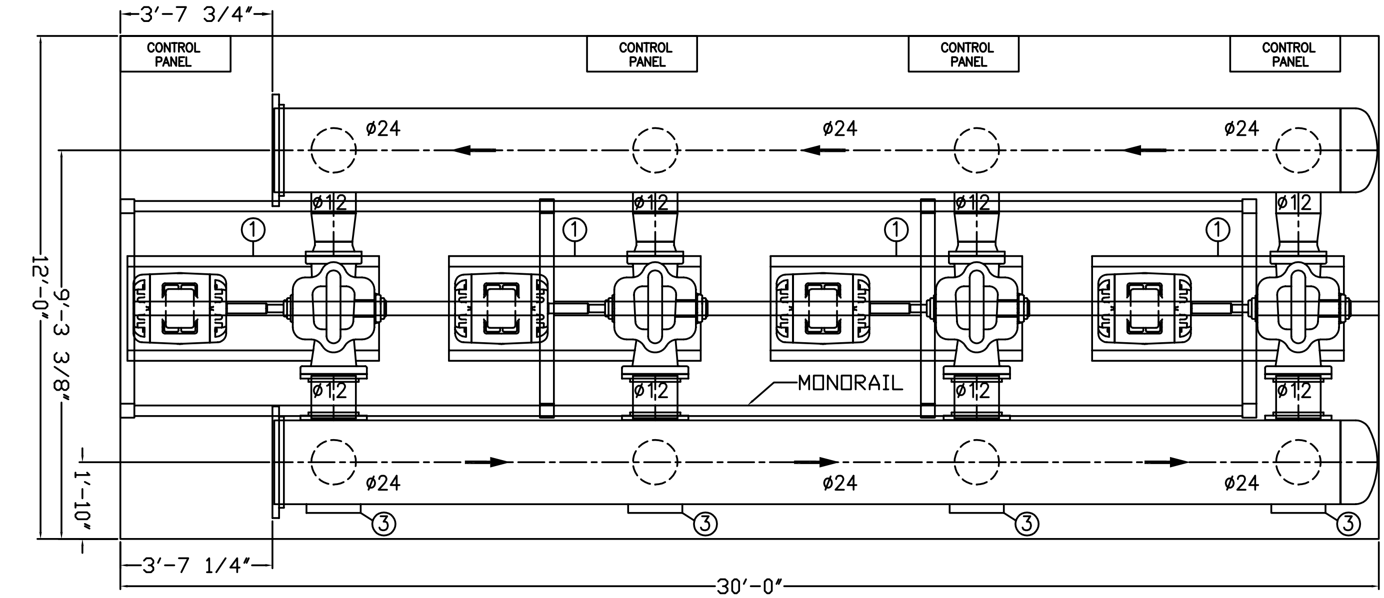

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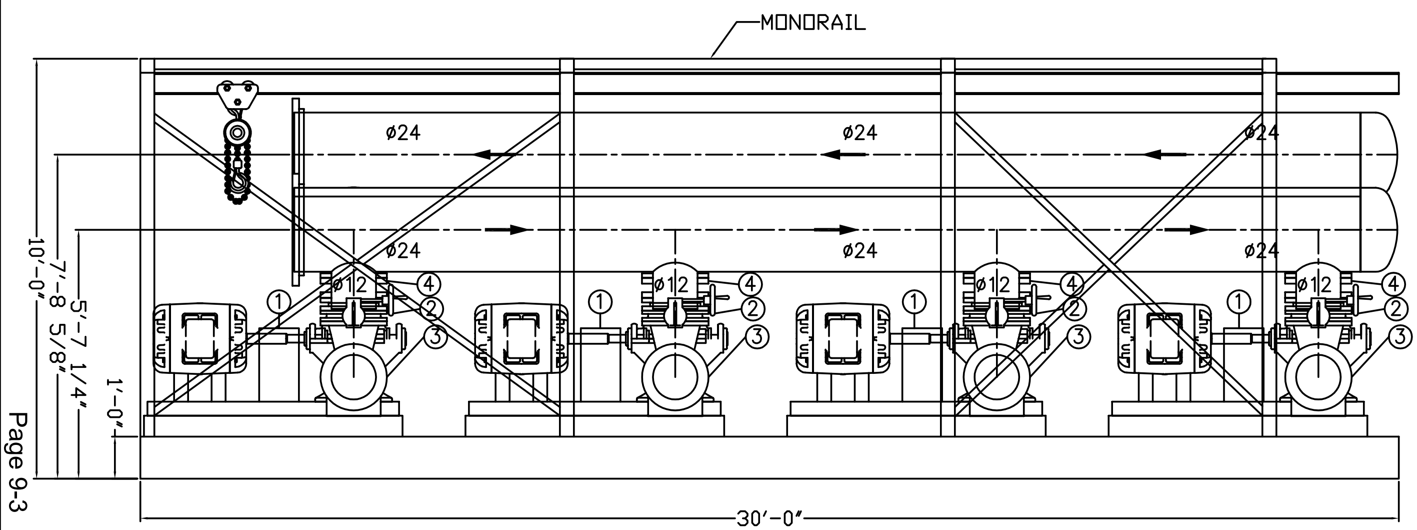

FRONT ELEVATION

KEY LEGEND

KEY	DESCRIPTION
1	PATTERSON PUMP 125HP–1750RPM
2	12" LUG B'FLY VALVE – GEAR OPER.
3	PATTERSON 12x12 SUCTION DIFFUSER
4	12" WAFER CHECK VALVE

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FIGURE 9.3

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KEY LEGEND

KEY	DESCRIPTION
1	PATTERSON 10x10x13 150HP
2	12" BFLY VALVE
3	PATTERSON 12"x10" SUCTION DIFFUSER
4	12" GLOBE CHECK
5	24" NEOPRENE FLEX CONNECTOR
6	12" NEOPRENE FLEX CONNECTOR

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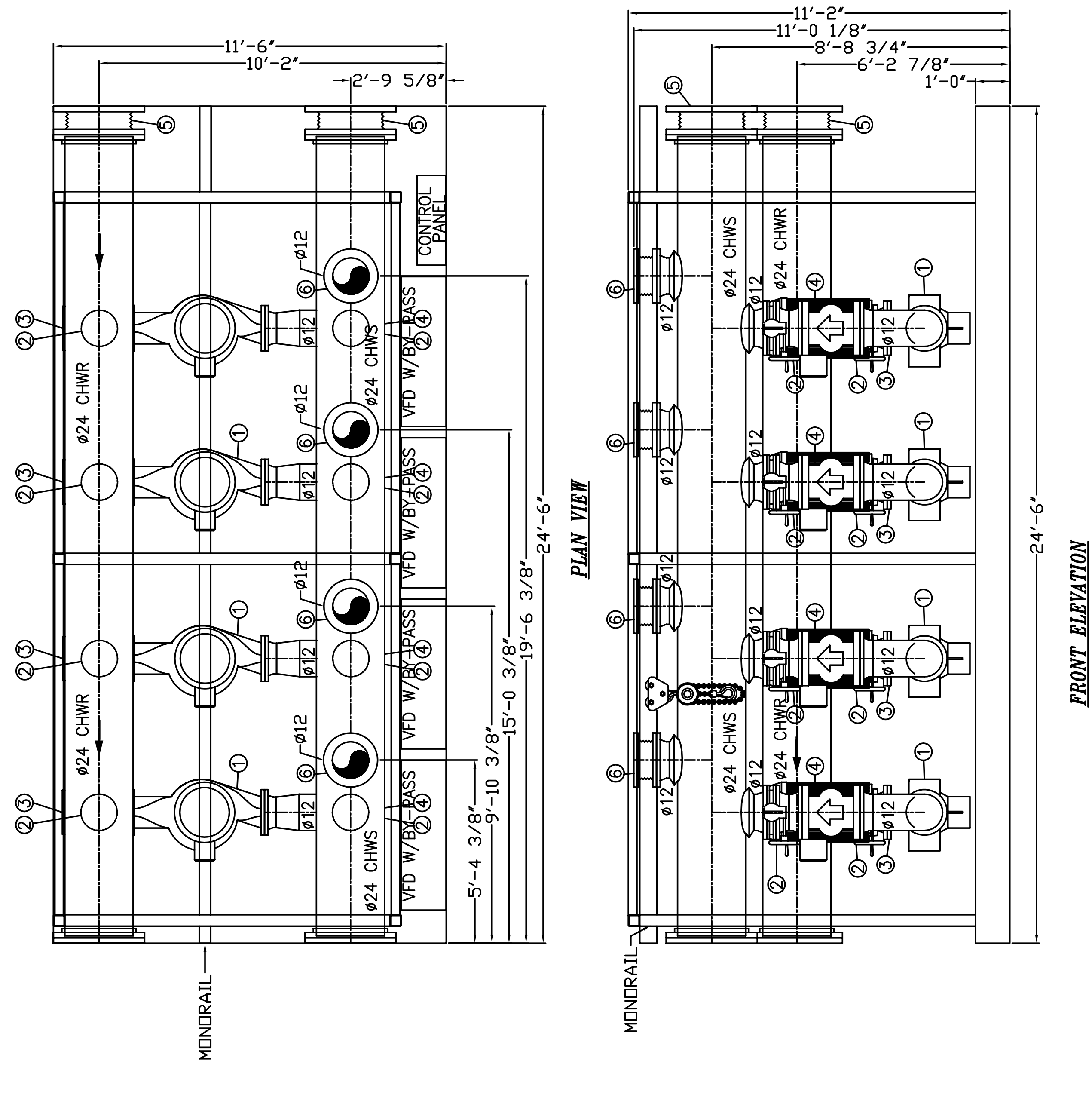

FIGURE 9.4

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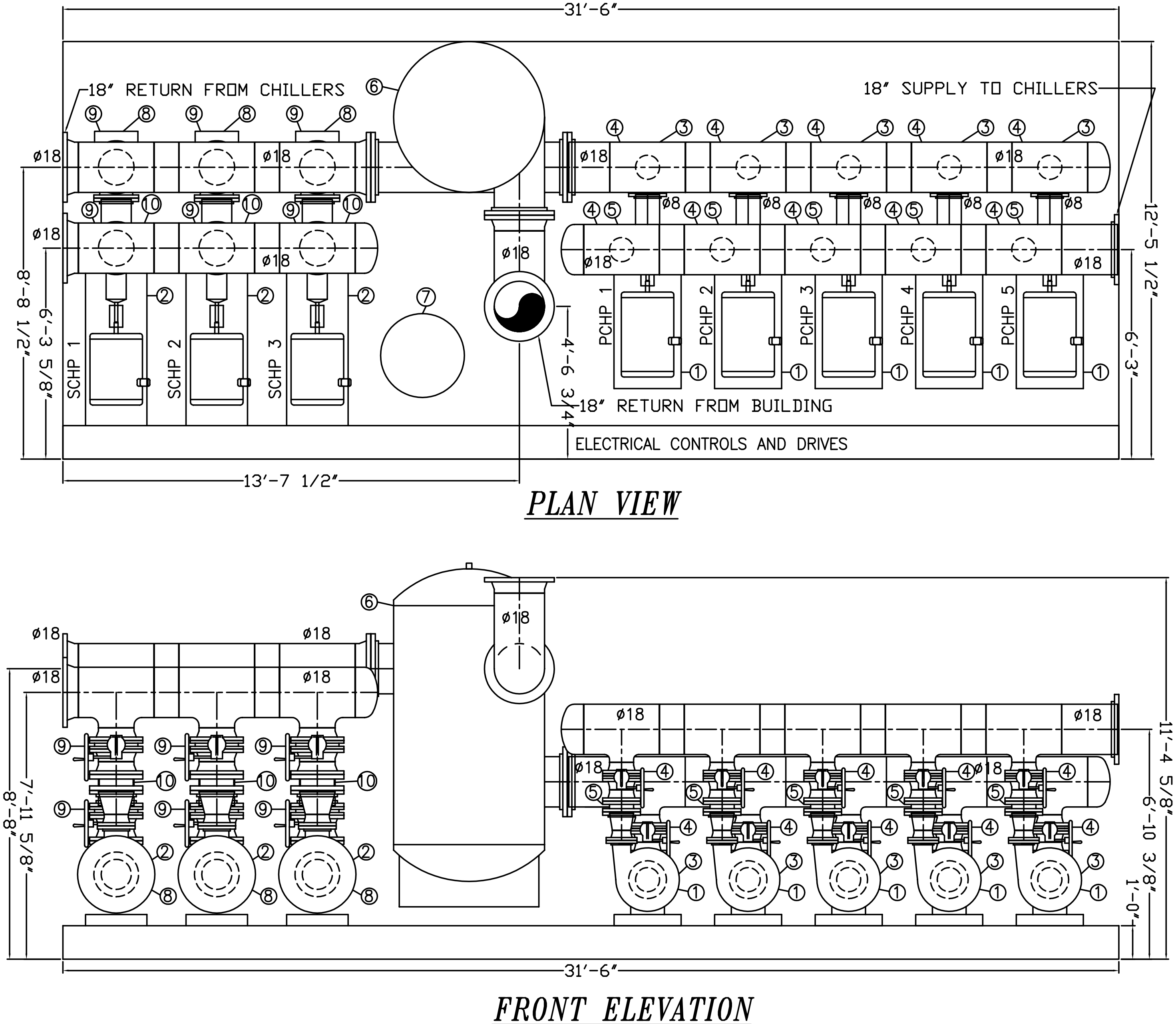

KEY LEGEND

KEY	DESCRIPTION
1	PATTERSON 20HP 1150RPM
2	PATTERSON 75HP 1750RPM
3	PATTERSON 8" x 8" SUCTION DIFFUSER
4	8" LUG B'FLY VALVE – LEVER OPER.
5	8" WAFER CHECK VALVE
6	18" AIR SEPARATOR W/16" TANGENTIAL
7	EXPANSION TANK
8	12" x 10" SUCTION DIFFUSER
9	12" LUG B'FLY VALVE – LEVER OPER.
10	12" WAFER CHECK VALVE

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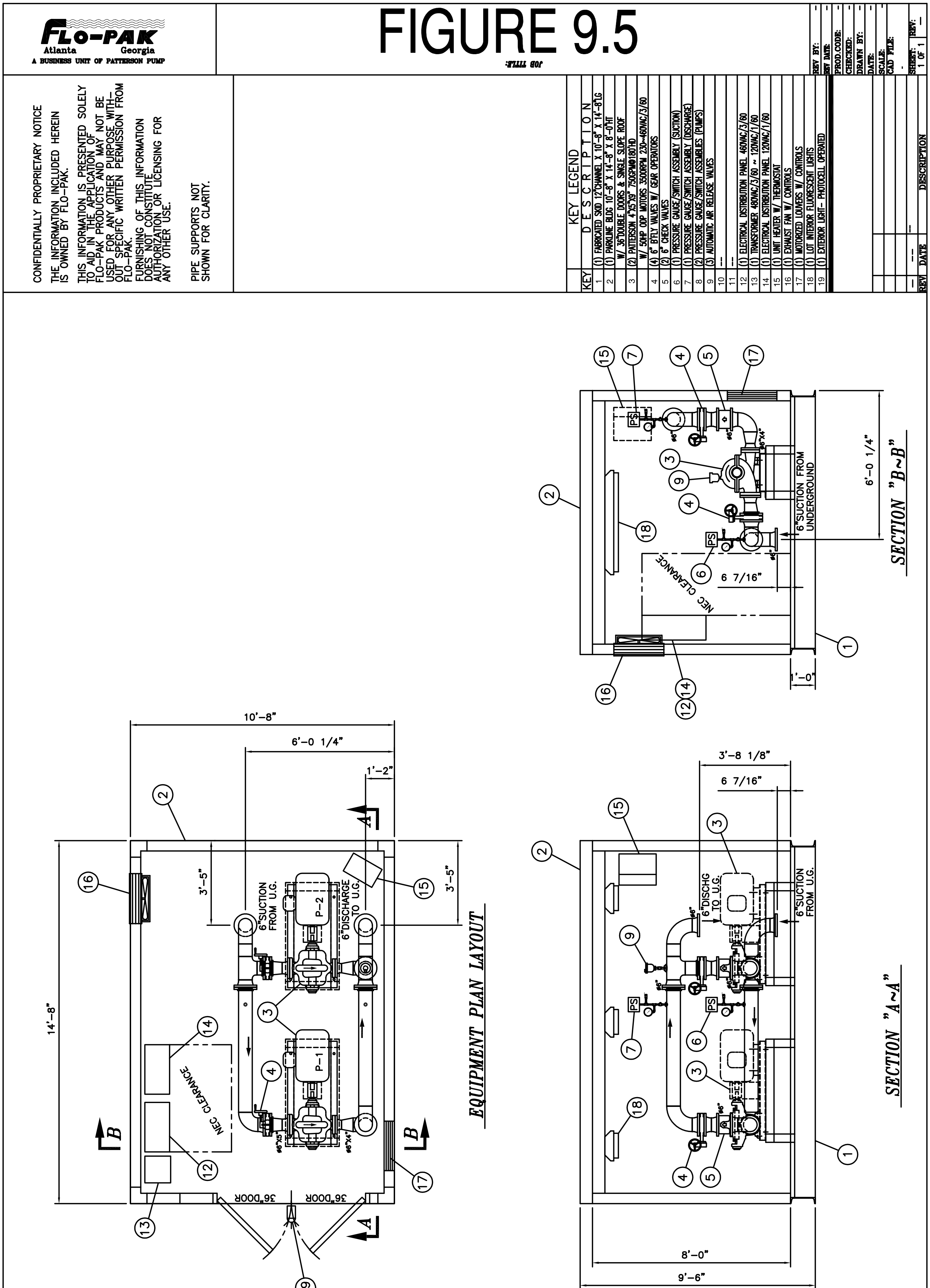
FIGURE 9.5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PIPE SUPPORTS NOT SHOWN FOR CLARITY.
KEY LEGEND
KEY DESCRIPTION
1 (1) FABRICATED SKID 12"CHANNEL X 10'-8" X 14'-8"LG
2 (1) PARKLINE BLDG 10'-8" X 14'-8" X 8'-0"HT W/ 36"DOUBLE DOORS & SINGLE SLOPE ROOF
3 (2) PATTERSON 4"X5"X9" 750GPM@180'HD W/ 50HP ODP MOTORS 3500RPM 230-460VAC/3/60
4 (4) 6" B'FLY VALVES W/ GEAR OPERATORS
5 (2) 6" CHECK VALVES
6 (1) PRESSURE GAUGE/SWITCH ASSEMBLY (SUCTION)
7 (1) PRESSURE GAUGE/SWITCH ASSEMBLY (DISCHARGE)
8 (2) PRESSURE GAUGE/SWITCH ASSEMBLIES (PUMPS)
9 (3) AUTOMATIC AIR RELEASE VALVES
10 ---
11 ---
12 (1) ELECTRICAL DISTRIBUTION PANEL 460VAC/3/60
13 (1) TRANSFORMER 480VAC/3/60 - 120VAC/1/60
14 (1) ELECTRICAL DISTRIBUTION PANEL 120VAC/1/60
15 (1) UNIT HEATER W/ THERMOSTAT
16 (1) EXHAUST FAN W/ CONTROLS
17 (1) MOTORIZED LOUVERS W/ CONTROLS
18 (1) LOT INTERIOR FLUORESCENT LIGHTS
19 (1) EXTERIOR LIGHT- PHOTOCELL OPERATED
REV DATE DESCRIPTION
REV BY:
REV DATE:
PROD.CODE:
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DATE:
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EQUIPMENT PLAN LAYOUT
10'-8"
6'-0 1/4"
1'-2"
14'-8"
3'-5"
6"SUCTION FROM U.G.
6"DISCHARGE TO U.G.
P-1
P-2
NEC CLEARANCE
36"DOOR
SECTION "B~B"
6'-0 1/4"
6"SUCTION FROM UNDERGROUND
6 7/16"
1'-0"
SECTION "A~A"
3'-8 1/8"
6"DISCHG TO U.G.
6"SUCTION FROM U.G.
8'-0"
9'-6"

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A BUSINESS UNIT OF PATTERSON PUMP

FIGURE 9.6.1

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PIPE SUPPORTS NOT SHOWN FOR CLARITY.

KEY LEGEND

KEY	DESCRIPTION
1	PATTERSON 3x4 30HP 1750RPM
2	PATTERSON 4x5 50HP 1750RPM
3	PATTERSON 4x5 20HP 1750RPM
4	PATTERSON 6x6 40HP 1750RPM
5	8" AIR SCOOP
6	McQUAY 300 TON CHILLER
7	McQUAY 500 TON CHILLER
8	COOLING TOWER #2
9	COOLING TOWER #1
10	PATTERSON 8X8 SUCTION DIFFUSER
11	PATTERSON 6X5 SUCTION DIFFUSER
12	PATTERSON 6X4 SUCTION DIFFUSER
13	8" B'FLY VALVE
14	PATTERSON 8X6 SUCTION DIFFUSER
15	LIGHT FIXTURE
16	HOUSE ELECTRIC PANEL
17	PUMP STARTER
18	CHILLER #1 STARTER
19	CHILLER #2 STARTER
20	6" B'FLY VALVE
21	8" TRIPLE DUTY VALVE
22	6" TRIPLE DUTY VALVE
23	8" S.S. FLEX CONNECTOR
24	6" S.S. FLEX CONNECTOR
25	8" 3-WAY VALVE
26	8" 2-WAY VALVE W/ACTUATOR
27	CHEMICAL TREATMENT SYSTEM

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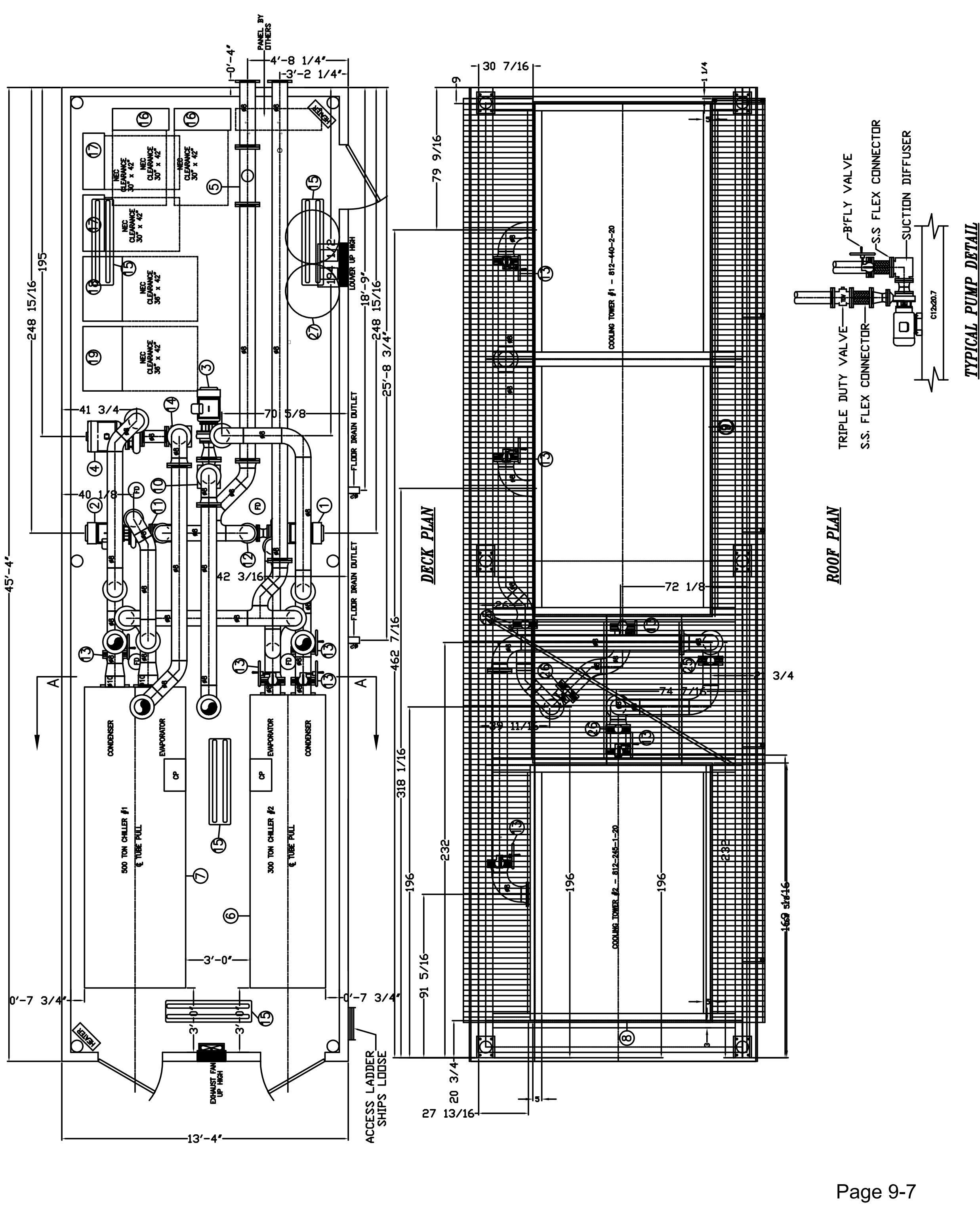

FIGURE 9.6.2

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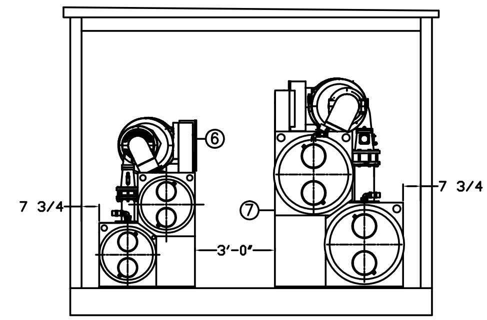
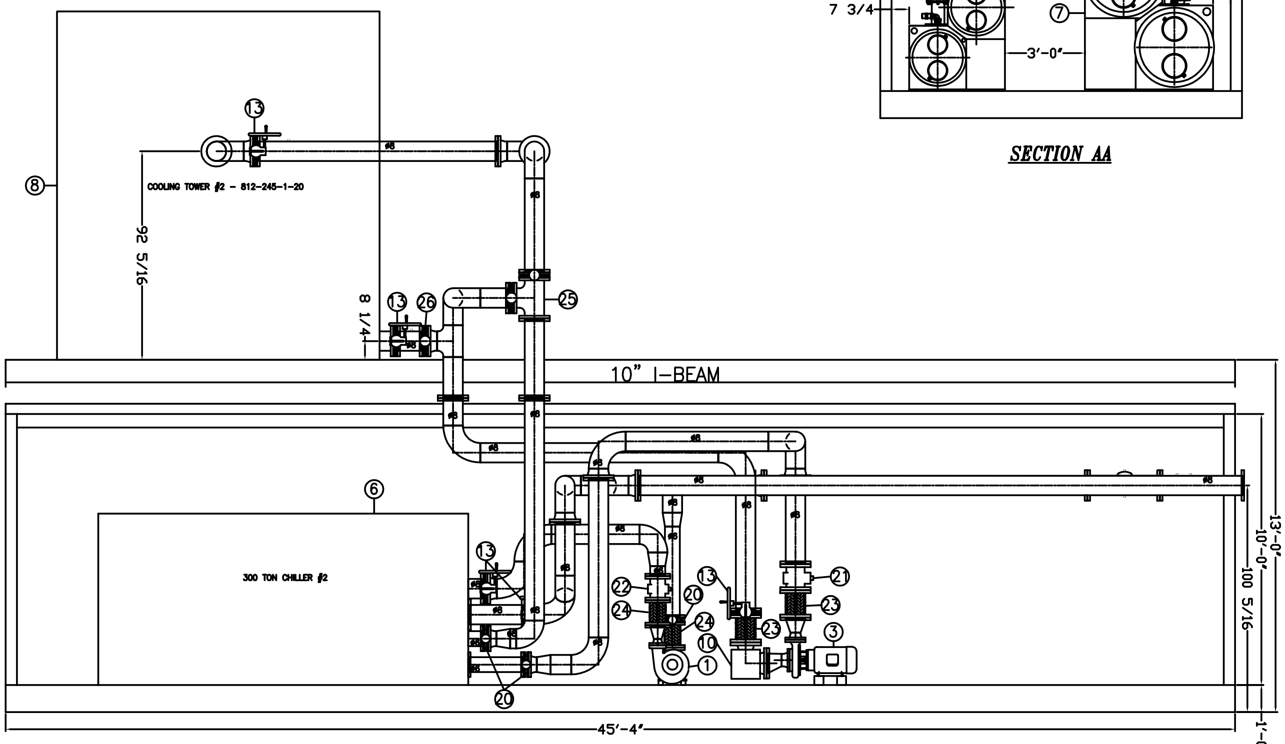
PIPE SUPPORTS NOT SHOWN FOR CLARITY.



SECTION AA



FRONT ELEVATION @ COOLING TOWER & CHILLER #2

KEY LEGEND

KEY	DESCRIPTION
1	PATTERSON 3x4 30HP 1750RPM
2	PATTERSON 4x5 50HP 1750RPM
3	PATTERSON 4x5 20HP 1750RPM
4	PATTERSON 6x6 40HP 1750RPM
5	8" AIR SCOOP
6	McQUAY 300 TON CHILLER
7	McQUAY 500 TON CHILLER
8	COOLING TOWER #2
9	COOLING TOWER #1
10	PATTERSON 8X8 SUCTION DIFFUSER
11	PATTERSON 6X5 SUCTION DIFFUSER
12	PATTERSON 6X4 SUCTION DIFFUSER
13	8" B'FLY VALVE
14	PATTERSON 8X6 SUCTION DIFFUSER
15	LIGHT FIXTURE
16	HOUSE ELECTRIC PANEL
17	PUMP STARTER
18	CHILLER #1 STARTER
19	CHILLER #2 STARTER
20	6" B'FLY VALVE
21	8" TRIPLE DUTY VALVE
22	6" TRIPLE DUTY VALVE
23	8" S.S. FLEX CONNECTOR
24	6" S.S. FLEX CONNECTOR
25	8" 3-WAY VALVE
26	8" 2-WAY VALVE W/ACTUATOR
27	CHEMICAL TREATMENT SYSTEM

JOB TITLE:

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FLO-PAK
Atlanta Georgia
A BUSINESS UNIT OF PATTERSON PUMP

FIGURE 9.6.3

JOB TITLE:

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PIPE SUPPORTS NOT SHOWN FOR CLARITY.

KEY LEGEND

KEY	DESCRIPTION
1	PATTERSON 3x4 30HP 1750RPM
2	PATTERSON 4x5 50HP 1750RPM
3	PATTERSON 4x5 20HP 1750RPM
4	PATTERSON 6x6 40HP 1750RPM
5	8" AIR SCOOP
6	McQUAY 300 TON CHILLER
7	McQUAY 500 TON CHILLER
8	COOLING TOWER #2
9	COOLING TOWER #1
10	PATTERSON 8X8 SUCTION DIFFUSER
11	PATTERSON 6X5 SUCTION DIFFUSER
12	PATTERSON 6X4 SUCTION DIFFUSER
13	8" B'FLY VALVE
14	PATTERSON 8X6 SUCTION DIFFUSER
15	LIGHT FIXTURE
16	HOUSE ELECTRIC PANEL
17	PUMP STARTER
18	CHILLER #1 STARTER
19	CHILLER #2 STARTER
20	6" B'FLY VALVE
21	8" TRIPLE DUTY VALVE
22	6" TRIPLE DUTY VALVE
23	8" S.S. FLEX CONNECTOR
24	6" S.S. FLEX CONNECTOR
25	8" 3-WAY VALVE
26	8" 2-WAY VALVE W/ACTUATOR
27	CHEMICAL TREATMENT SYSTEM

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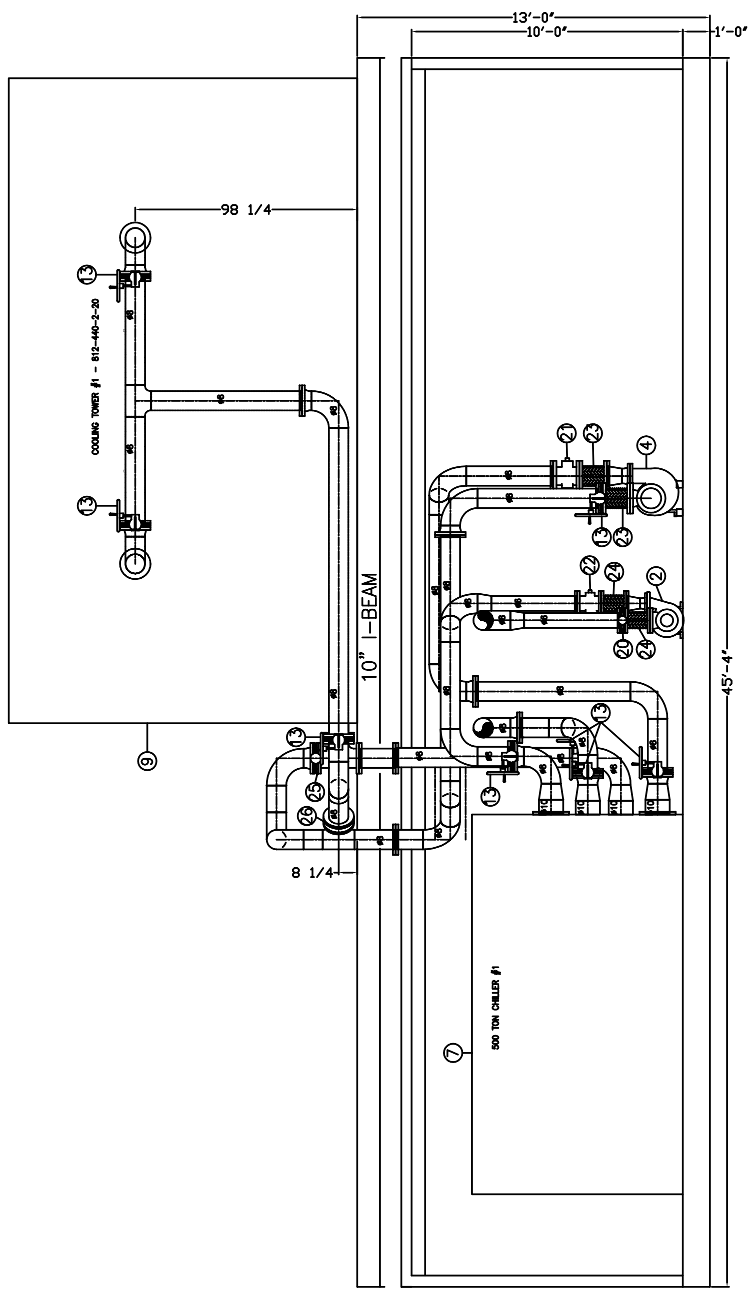
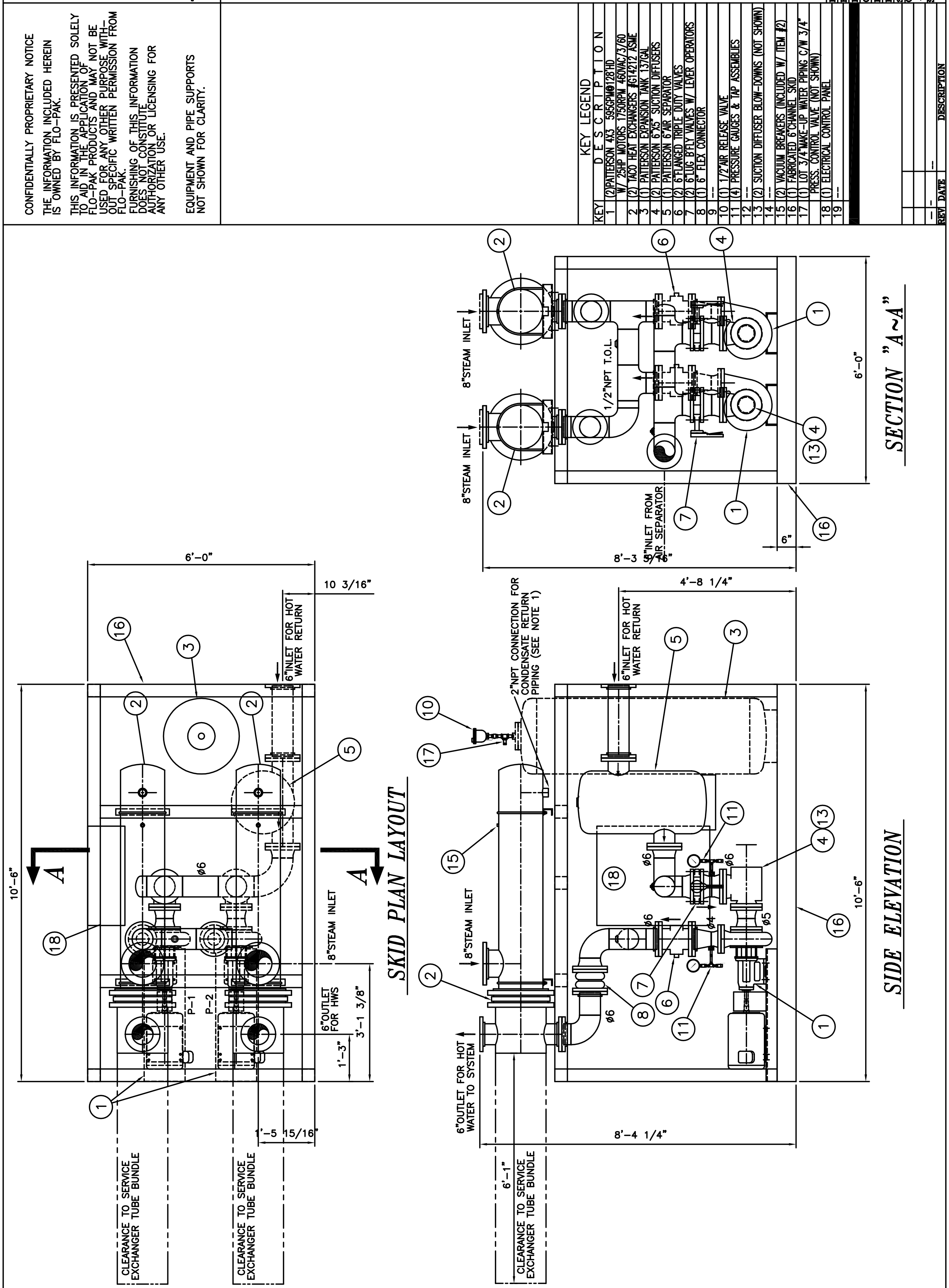


FRONT ELEVATION @ COOLING TOWER & CHILLER #1

FIGURE 9.7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EQUIPMENT AND PIPE SUPPORTS NOT SHOWN FOR CLARITY.
KEY LEGEND
KEY DESCRIPTION
1 (2) PATTERSON 4X3 595GPM@128'HD W/ 25HP MOTORS 1750RPM 460VAC/3/60
2 (2) TACO HEAT EXCHANGERS #614212 ASME
3 (1) PATTERSON EXPANSION TANK 137GAL
4 (2) PATTERSON 6"X5" SUCTION DIFFUSERS
5 (1) PATTERSON 6"AIR SEPARATOR
6 (2) 6"FLANGED TRIPLE DUTY VALVES
7 (2) 6"LUG B'FLY VALVES W/ LEVER OPERATORS
8 (1) 6" FLEX CONNECTOR
9 ---
10 (1) 1/2"AIR RELEASE VALVE
11 (4) PRESSURE GAUGES & TAP ASSEMBLIES
12 ---
13 (2) SUCTION DIFFUSER BLOW-DOWNS (NOT SHOWN)
14 ---
15 (2) VACUUM BREAKERS (INCLUDED W/ ITEM #2)
16 (1) FABRICATED 6"CHANNEL SKID
17 (1) LOT 3/4"MAKE-UP WATER PIPING C/W 3/4" PRESS. CONTROL VALVE (NOT SHOWN)
18 (1) ELECTRICAL CONTROL PANEL
19 ---
REV DATE DESCRIPTION
REV BY: REV.DATE: PROD.CODE: CHECKED: DRAWN BY: DATE: SCALE: CAD FILE: SHEET: REV:
SKID PLAN LAYOUT
SIDE ELEVATION
SECTION "A~A"
8"STEAM INLET
1/2"NPT T.O.L.
6"INLET FROM AIR SEPARATOR
6"INLET FOR HOT WATER RETURN
6"OUTLET FOR HWS
6"OUTLET FOR HOT WATER TO SYSTEM
2"NPT CONNECTION FOR CONDENSATE RETURN PIPING (SEE NOTE 1)
CLEARANCE TO SERVICE EXCHANGER TUBE BUNDLE
P-1
P-2
6'-0"
10'-6"
10 3/16"
3'-1 3/8"
1'-3"
1'-5 15/16"
8'-3 5/16"
4'-8 1/4"
8'-4 1/4"
6'-1"
6"

assembly. Similar equipment is difficult to acquire in the field. This is particularly evident when the actual installation is remote from pipe and equipment warehouses.

b. Factory assembled units usually reduce actual equipment room floor space by as much as 30%. The system manufacturer has actual data for the design engineer on the space required for pump removal and repair.

c. The design engineer can use manufacturer's data on the actual size of the pumping system. Submittals are much simpler since there is one source for all of the equipment, not three or four manufacturers.

d. The contractor can place one order for a sizeable equipment assembly, not many orders for pumps, drives, piping, and controls.

e. Materials are purchased at OEM pricing and in large quantities.

2. Scheduling: The complete pumping system can be scheduled for a specific delivery date. The pumping system can be assembled in the factory and delivered to the job site when the equipment room is completed. Just in time delivery is possible with packaged pumping systems. The time to set and connect the piping and power to the pumping system is far less than the time to assemble the system.

3. Single Responsibility:

One of the unique advantages of factory assembly is single responsibility. The pumping system manufacturer must guarantee the design flow and head within the energy constraints of the specifications. It is the responsibility of the manufacturer to insure that the system is installed in accordance with the requirements of the system.

4. Factory Testing:

Not enough can be said for the advantages of factory testing. So often, a pumping system is specified with capacity for future expansion of the water system. There is no possible way that the system can be tested in the field for this capacity since it will not be available for years to come. The pumping system can be tested in the factory at the design flow and head. The design engineer and the owner are, therefore, assured that the capacity will be there when the future expansions occur.

5. Reduced Friction Loss:

It is rather amusing that some of the opponents of factory assembly claimed that the friction loss would be greater for the closely assembled fittings of the factory assembled system. They insisted upon a test to prove this. ASHRAE Research Test No. RP1035 was conducted with the results as seen in Figure 9-8. Much to their chagrin, it was proved that the total friction loss of the fittings could be 71 to 95% of that for field spaced fittings. The manufacturer of the system is usually required to establish a specific friction loss for the system; this can be verified in the factory testing.

6. Reduced Maintenance:

The manufacturer of the pumping system, particularly those who manufacturer their own pumps, can insure that the pumps operate at optimum flows and avoid the high pump maintenance that occurs with high radial thrust. The pumps can operate near the BEP and avoid extremely low or high flows.

The pumping system manufacturer can minimize pump repair by installing the correct suction and discharge fittings and headers, as well as proper pipe supports.

PACKAGED PUMPING SYSTEM EQUIPMENT:

The equipment included in a pumping system can include all that is required for its safe and reliable operation as it delivers the specified head and flow.

1. **Pumps:** Almost all of the pump types used on HVAC water systems have been provided with packaged pumping systems. Systems with one to fifteen pumps have been assembled. One advantage that is occurring now is the ability of the pump manufacturers to provide their own pumps with their packaged pumping systems. This offers an additional advantage to the designer and owner. Verification of performance and repair parts can be provided by a single manufacturer.

2. Motors:

Electric motors are normally 460 V/ 3 PH/ 60 Hz. Other voltages such as 208, 2300 and 4100 have been furnished. Since most of these systems are installed in central plants, open frame drip proof construction is generally adequate. There is little advantage in requiring TEFC for most installations. The largest motors that have been furnished on these systems are in the range of 500 to 750 hp.

3. Switchgear and Drives:

All necessary switchgear required for the operation of a packaged pumping system is usually furnished mounted on the base. This includes variable speed drives and transfer switches where necessary. Power may be designed for individual motors, or there may be one power supply for the entire system. Emergency back-up equipment can be provided such as generator sets.

4. General equipment:

a. Pump piping and valving include shut-off valves, check valves, suction strainers or diffusers, and branch piping. Motorized valves may be required in lieu of check valves on some applications.

b. Pumping system suction and discharge headers.

c. Bypasses for chillers or boilers, including control valves as required on variable primary pumping.

d. Structural steel supports for all piping and accessories.

e. Movable rails and hoists for removing all components of the pumping system.

f. Expansion tanks with water makeup valves, including back-flow preventers.

g. Air eliminators of the tank or in-line type.

h. Chemical feeders

j. Heat exchangers – plate and frame or shell and tube types.

k. Instrumentation can include just suction and discharge pressure gauges, or it can also include temperature indicators or transmitters, watt transmitters, and flow meters. Special instrumentation is provided for particular applications

l. Control supplied with a packaged pumping system can vary from simple on-off control of the pumps only to complete chiller plant control.

m. Pump houses can be furnished complete with all necessary equipment such as the structural base, the pumping system, Air conditioning and ventilating equipment, electrical switch gear, sump pumps, and any other equipment required for a complete pumping station.

COMPLETE CHILLER PLANTS:

The expansion of buildings has often resulted in the need for additional chilled water capacity. In many cases, the most economical answer to this need is a complete packaged chiller plant. These plants include all of the packaged pumping system equipment plus the chillers and provisions for mounting the cooling towers on top or to the side of the chiller plants. Flo-Pak Patterson has produced a number of these plants with capacities up to 1,200 tons.

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Chapter 10

INSTALLATION AND MAINTENANCE OF HVAC PUMPS

HVAC PUMPS and SYSTEMS
DESIGN MANUAL

PATTERSON PUMP COMPANY

INTRODUCTION:

The proper installation and operation of HVAC pumps will provide years of service with very little maintenance and repair. More damage can be done to these pumps during the installation period than during years of operation. HVAC water systems offer relatively easy service when compared to pumps in the industrial and municipal fields. Need of maintenance by these pumps indicates improper installation or operation.

BEFORE INSTALLATION:

All HVAC pumps should be tagged by project number or use before delivery to the jobsite. Failure to do this can cause a great amount of confusion and lost material and man-hours. There can be a great many pumps on large projects. Packaged pumping systems should be checked for orientation to insure that the system is not installed backwards.

The pumps and packages should be kept in a dry place before installation to avoid damage from freezing. If it is necessary that this equipment be stored where it can freeze, the pumps and any piping should have drain plugs removed to prevent accumulation of any water.

Pumps should be checked for size, capacity, motor horsepower, and voltage. Split case pumps should be checked for proper rotation.

Pumps and packages should be lifted in place by specified rigging points, not by the piping or the pumps themselves.

The power supply for the motors or total package should be checked for leg-to-leg voltages to avoid any imbalance.

PUMP AND PACKAGE SYSTEM BASES AND FOUNDATIONS:

Much of the success of a pump or packaged system is the quality of the pump base itself. Although HVAC pumps are not high head generally, frame mounted end suction and split case pumps should be equipped with structural steel bases

that prevent misalignment during transportation and installation. Stamped and “erector set” bases do not have the rigidity of structural steel.
Basically, there are two classes of pump bases, rigid or floating. On rigid bases, the pump base is bolted directly to the concrete pump foundation, Figure 1. The height and structural content of the pump foundation is dependent upon the size of the pump and the physical requirements of each installation. A rigid packaged pumping system is often furnished with open structural steel that is filled with concrete after installation, Figure 10-2. A 1” housekeeping pad should be poured on the floor to keep water away from the system base itself as shown in this figure.

Floating bases are required when no vibration is allowed into the equipment room floor by pumps or systems, Figure 10-3. A vibration base is almost always required when this equipment is installed on a mezzanine or upper floor of a building. Also, in critical environments like laboratories, a vibration base may still be required even though the pumps or packages are located on the basement floor of the building. There are a number of different types of vibration bases; they can be a total sub-base or an individual vibration base under each pump on a pumping system. Various manufacturers offer this equipment for specific applications.

Seismic restraints are being included in many specifications today. The Uniform Building Code has developed lateral force levels for various zones of the United States. The July 2001 issue of the ASHRAE Journal contained an article by Patrick Lama, PE on seismic restraints including an explanation of the requirements of Boca, UBC 97, 2000 IBC and CNBC. Figures 10-4 and 5 are from this article.

PIPING FOR PUMPS:

The first principle of pump piping is that it imposes no thrust or pressure on the pump connections. On chilled and condenser water, there is little expansion, so the pipe supports must do this. After the piping installation is completed, it is imperative that flanges near the pump be disconnected to see if there is any movement such as slippage or gapping. If either occurs; the supports should be adjusted to eliminate these dangerous conditions. On hot water where there is expansion, flexible connectors should be provided to account for both axial and lateral expansion.

There can be as many as a dozen different valves, fittings, strainers, and other accessories around a pump. Each of these incurs a friction loss. On well designed HVAC pump instillations, the total of these losses should not exceed 8 feet of pump head. Unfortunately, some packagers will cut the pipe size of these fittings to reduce the cost, and this loss will exceed 15 feet. It is imperative that specifications include the size of the piping around the pumps.

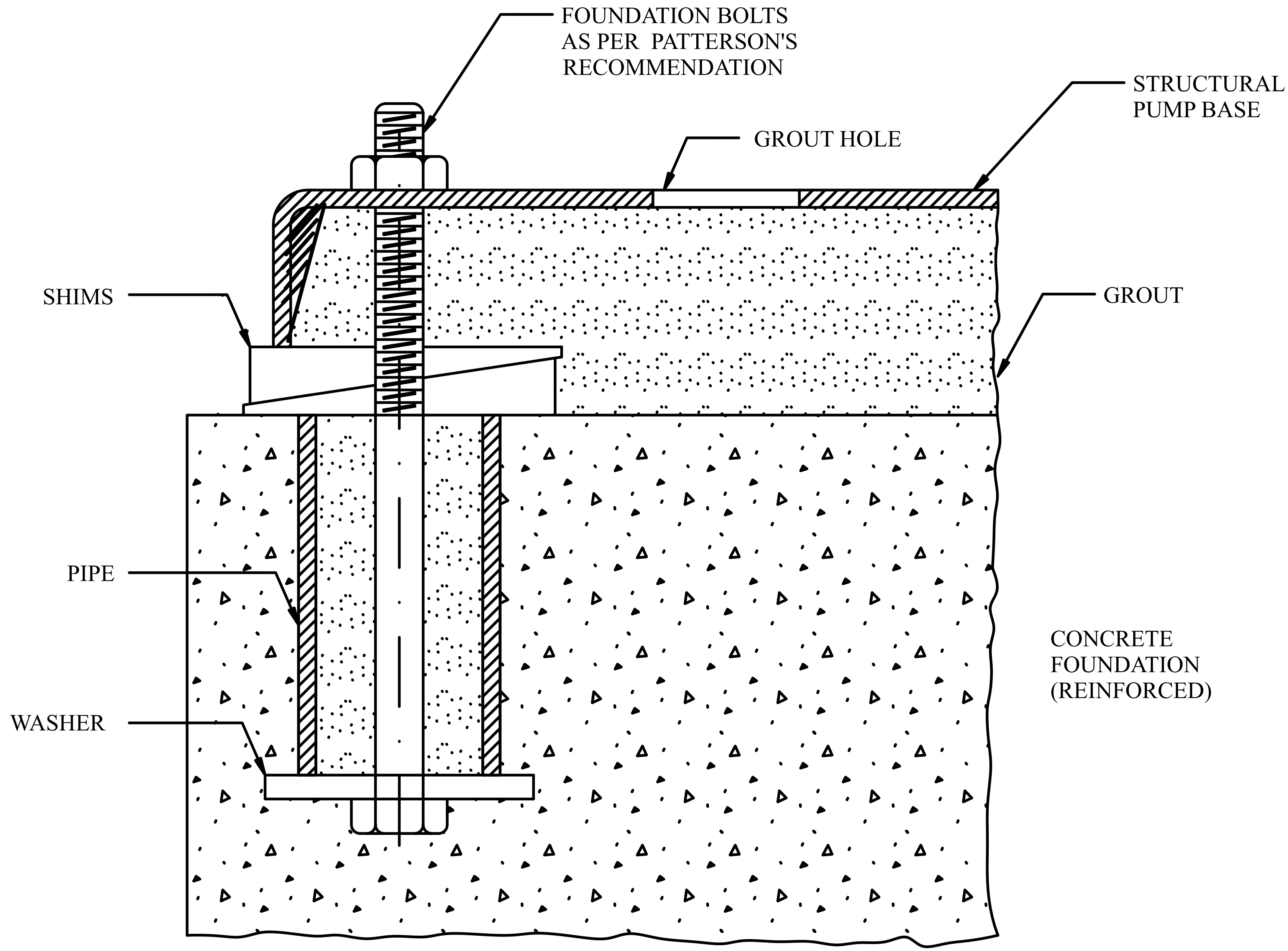
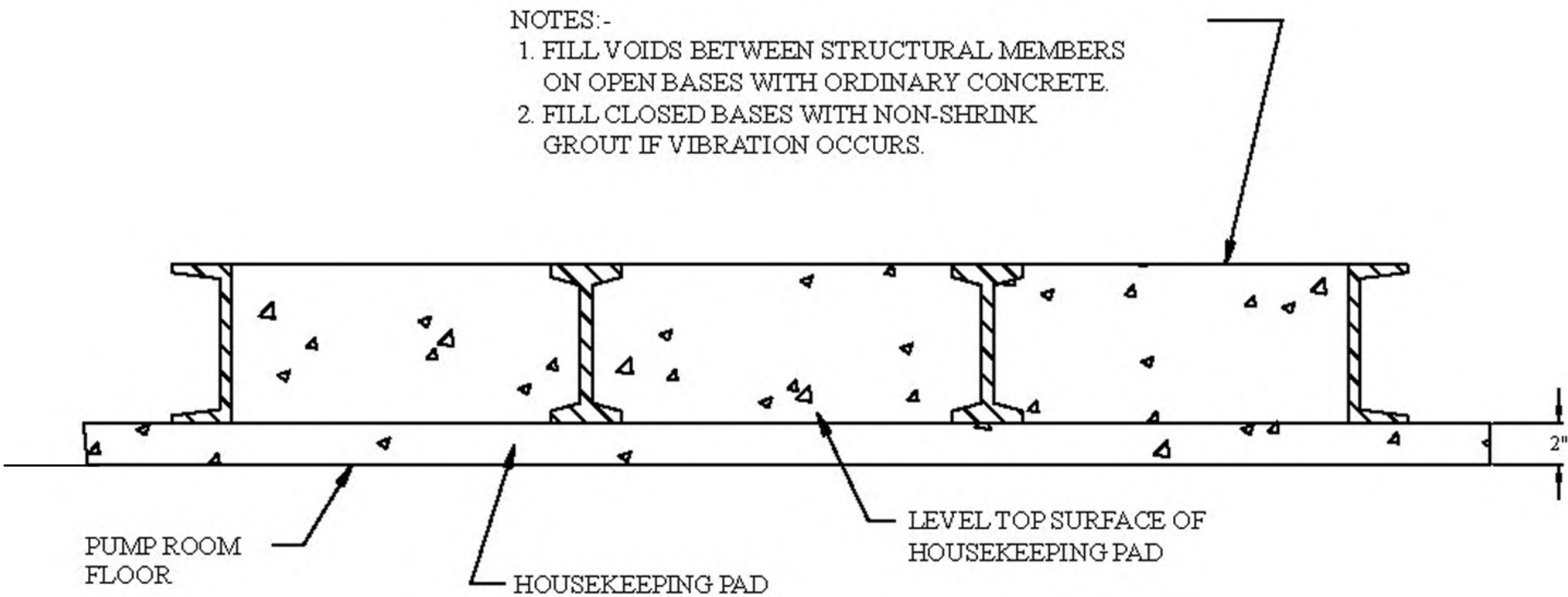


FIGURE 10-1

INSTALLATION OF PUMPS WITH BASES

FIGURE 10-2
INSTALLATION OF PUMPING
SYSTEM BASE



From: "Water Pumps and Pumping Systems:, James Rishel, PE, McGraw-Hill, New York, 2002.

Figure 10-3: Vibration Base with Isolators

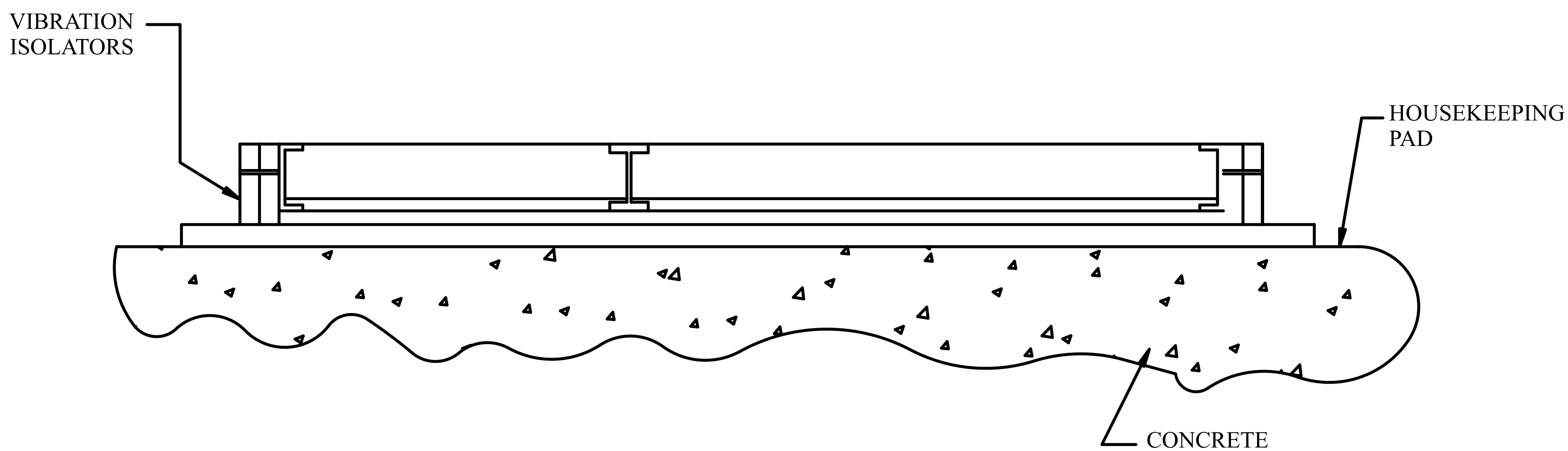
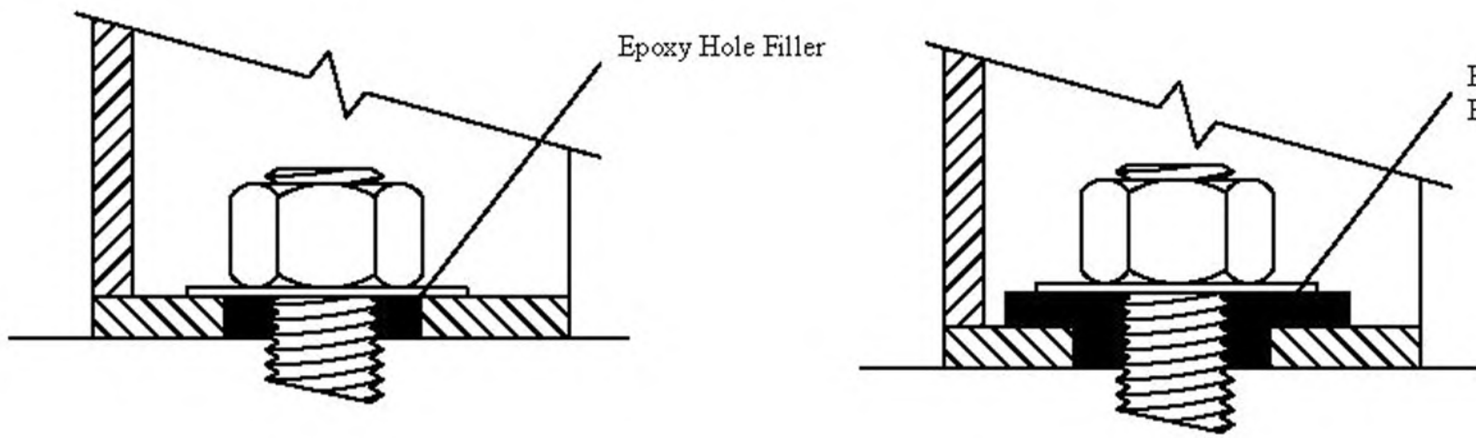
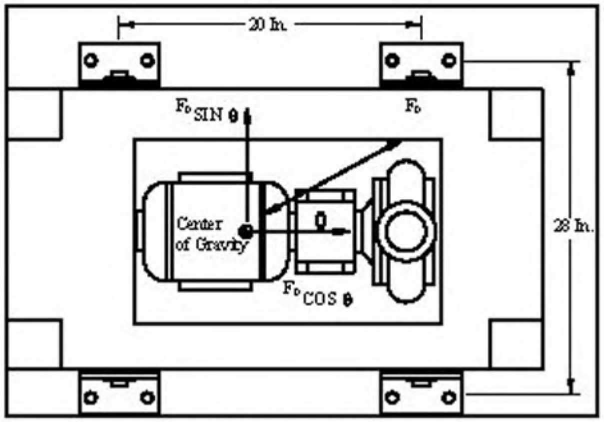


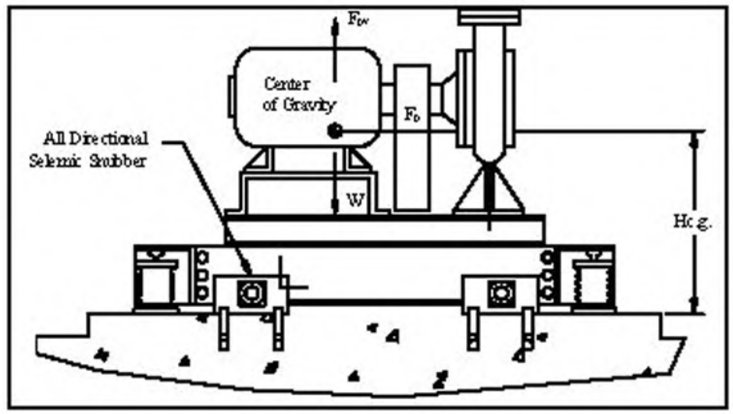
FIGURE 10-4
SEISMIC INSTALLATION
OF FLOOR BOLTS



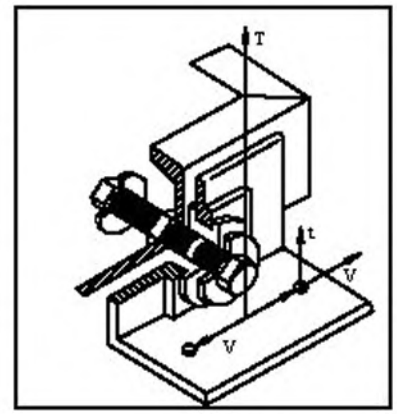
From: "Seismic Installation of Floor Bolts", Copyright 2001, American Society of Refrigeration and Air-Conditioning Engineers, Inc., www.ashrae.org, ASHRAE Journal, July 2001.



Plan view of seismic mounts.



Elevation view of equipment across length.



Snubber detail.

FIGURE 10-5
SEISMIC INSTALLATION OF PUMPS

ELECTRICAL INSTALLATIONS FOR PUMPS:

Electrical installations for pumps should result in safe and efficient operation. Basically, the latest version of the National electrical Code should be available to the designer. Also, there may be local codes that have specific requirements that are necessary because of environmental conditions.

Some simple requirements should be followed, such as bolting the electrical leads instead of using wire nuts, and always providing a flexible connection but not over 5 feet long. A disconnect switch must be installed at the motor if it is out of sight of or 50 feet from the disconnecting means.

OPERATION OF PUMPS:

Refer to Patterson manuals for practical pump operation.

Once the pumps are installed correctly, the initial start up can begin. Be sure that the pump is filled with water and the system pressure is that specified. Open any air vents to insure that most of the air is out of the system.

Be sure that the isolation valves on the suction and discharge are open!

The rotation of the pumps should be checked next by bumping the manual start and insuring that the rotation is correct. If it is backwards, switch any two leads on three phase power, and the rotation will be reversed.

If the pump is variable speed, put the VFD on manual control and start the pump at minimum speed. Check for noise or vibration. Gradually increase the speed to the maximum speed allowable by system conditions.

Find the cause of any noise, heat, or vibration before putting the pump into service.

SUMMARY:

An extra hour devoted to start up of a pump will give back thousands of hours of satisfactory service!

The Pump People®

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