Chiller, Boiler and Distribution System Control Applications

SECTION OF ENGINEERING MANUAL OF AUTOMATIC CONTROL 77-1100

Contents

INTRODUCTION

This section provides descriptions of and control information about water chillers, cooling towers, hot water boilers, steam boilers, and water, steam, and district heating distribution systems. It discusses various methods of controlling and distributing steam or water to heating and cooling coils, and methods of ensuring safe and proper control of boilers, chillers, and converters. Sample system solutions are based upon digital control systems and networks. These digital control strategies may not be practical with other forms of control, and may not be possible with all digital controllers or by all digital controller programmers. Many solutions are portrayed as they may be specified to be displayed on a PC based BMCS color graphic monitor. The data points shown on the color graphics are still recommended for systems without color graphics, and may be specified via a points list or by publishing the graphic picture. The values (setpoints, timings, parameters, etc.) shown in the examples are arbitrary. Values for any given project must be specific to the project.

ABBREVIATIONS

AHU — Air Handling Unit
BMCS — A Building Management and Control System, including digital controllers and communication networks.
CHW — CHilled Water
DB — Dry Bulb
DP — Differential Pressure
Delta T (ΔT) — Differential Temperature
EPID — Enhanced PID. A PID algorithm which includes integral component windup prevention, user defined initial value, and user defined start-up ramp time. See CONTROL FUNDAMENTALS section.
HW — Hot Water
HX — Heat eXchanger
NPSH — Net Positive Suction Head
OA — Outside air
PID — Proportional-Integral-Derivative
SG — Specific Gravity
VAV — Variable Air Volume
VSD — Variable Speed Drive
WB — Wet Bulb

DEFINITIONS

Approach–
1. The temperature difference, in a cooling tower, between outdoor wet bulb and condenser water leaving the tower.
2. The temperature difference in an air cooled condenser between the liquid refrigerant leaving the condenser and the entering air dry bulb.
3. The temperature difference in a conduction heat exchanger between the entering working fluid and the leaving treated fluid.

Central plant–An area or building where the chillers and boilers for a building or group of buildings are located.

Compressor–A mechanical device for increasing a gas pressure.

- **Centrifugal compressor**–A device which uses centrifugal force to compress refrigerant vapors in a vapor-compression cycle chiller. This is not a positive displacement compressor.

- **Positive displacement compressor**–A compressor that reduces the volume of a compression chamber to compress a gas.

- **Reciprocating compressor**–A positive displacement compressor which uses the reciprocating motion of one or more pistons to compress a gas.

- **Screw compressor**–A positive displacement compressor which uses the rotary motion of two meshed helical rotors to compress a gas.

Constant speed pumping–A pumping system where the system pressure is maintained by a constant speed pump.

Deadband–A range of the controlled variable in which no corrective action is taken by the system and no energy is used.

Diversity–A design method where building elements (such as chillers and pumps) are sized for the instantaneous building peak load requirements (rather than the sum of all the loads, which may individually peak at times other than when the building peaks). Diversity does not allow the use of three-way AHU control valves.

Double bundle Condenser–A chiller condenser having two coils in the shell to allow the chiller to dissipate heat either to the cooling tower or to a heating load.

Head–Pressure of a fluid

- **Head pressure**–The pressure measured at the discharge of an operating pump or compressor.

- **Refrigerant head**–The pressure difference between compressor suction and discharge pressures or the temperature difference between condensing and evaporating temperatures.

- **Static head**–The pressure of a static fluid expressed as the height of the fluid or the height of another fluid it would support.

Non-symmetrical loading–Diversity in multiple load systems, where individual loads operate at different times or loading from the others.

Primary - Thermal production water elements such as chillers and boilers. Thermal production air elements such as air handlers.

Rangeability–The ratio of maximum to minimum flow of a valve within which the deviation from the inherent flow characteristic does not exceed some stated limits.

Secondary - Thermal consumption water elements such as AHU coils. Thermal consumption air elements such as VAV boxes.

Superheat–The additional heat contained in a vapor at a temperature higher than the saturation (boiling) temperature corresponding to the pressure of the vapor.

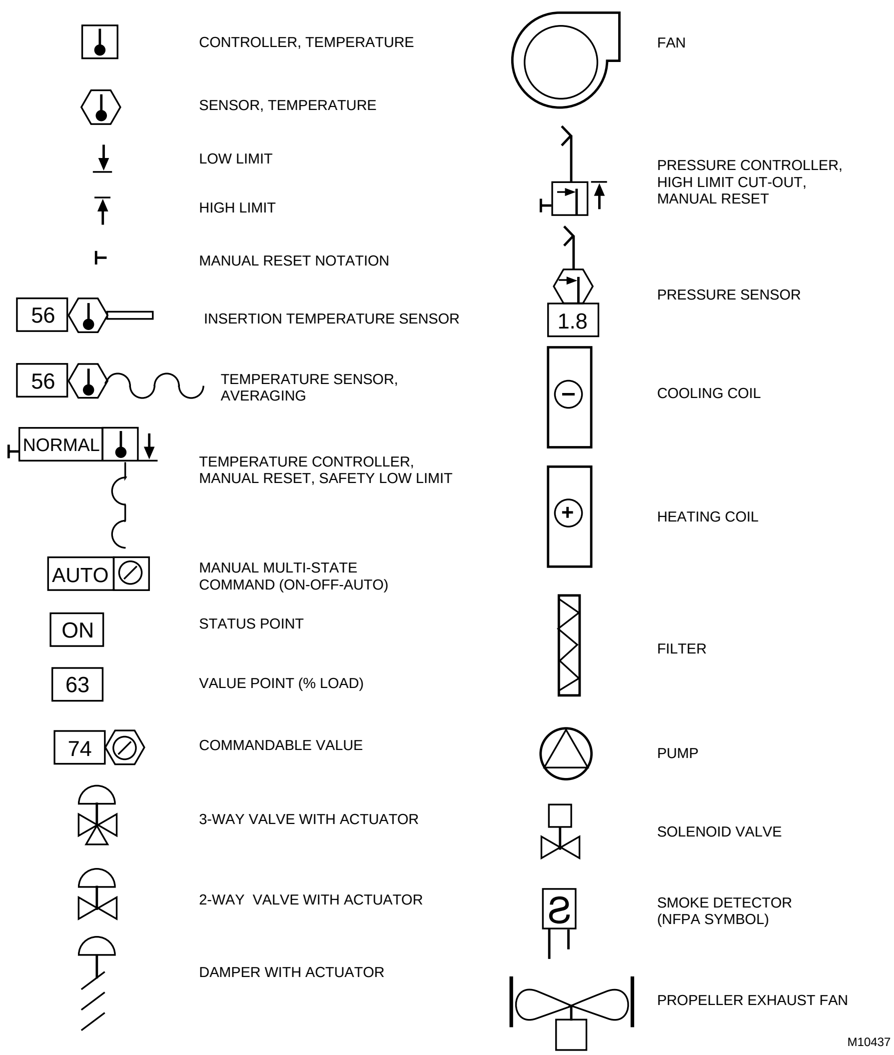
Surge–A condition where refrigerant reverses flow approximately every two seconds creating excessive noise, vibration, and heat. Surge is caused by insufficient pumping head to meet the rise in pressure from evaporator to condenser.

Symmetrical loading–Diversity in multiple load systems, where individual loads operate at the same time and same percentage loading as the others. Diversity factor = 1.

Variable speed pumping (VSP)–A pumping system where the flow/pressure is varied by changing the pump speed.

SYMBOLS

The following symbols are used in the system schematics following. These symbols denote the nature of the device, such as a thermometer for temperature sensing.



CHILLER SYSTEM CONTROL

INTRODUCTION

A chilled water system consists of a refrigeration system (water chiller), a chilled water distribution system, loads cooled by the chilled water and a means of dissipating the heat collected by the system. The refrigeration system cools water pumped through it by a chilled water pump. The chilled water flows through the distribution system to coils in air handling units or terminal units. Heat is removed from the refrigeration system using water or air. For chilled water control within AHU systems, see the Air Handling System Control Applications section.

Chilled water systems are used in many buildings for cooling because of their flexibility and operating cost compared with direct expansion (DX) cooling coil systems. Typically chilled water is generated at a central location by one or more chillers and distributed to coils in air handling system (Fig. 1). The quantity and temperature of the water supplied must be sufficient to meet the needs of all fan systems. Since the chilled water system is the major user of energy in many buildings, energy costs should be a consideration in chilled water plant configuration.

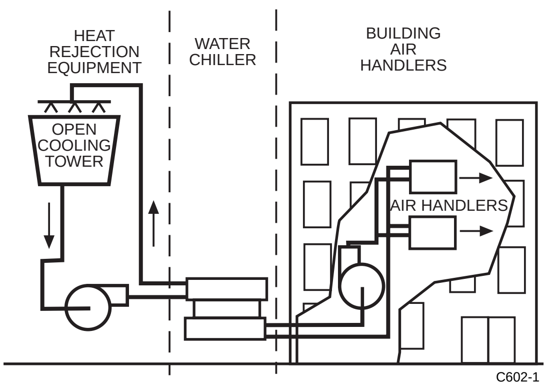


Fig. 1. Typical Water Chilling System.

A chilled water system can provide hot water for a heating load when a simultaneous heating and cooling load exists. It can be used with a chilled water, ice tank, or phase change material thermal storage system to lower the peak load demand and allow use of a smaller chiller. It can use the system cooling tower during light load conditions to supply cool water to the system without running the chiller, if the outside air WB temperature is low enough.

Chiller capacity controls are usually factory installed by the chiller manufacturer. The BMCS usually stages chillers on and off, provides chiller controls with a chilled water temperature setpoint, and controls the condenser water system. Chillers are usually controlled from their leaving water temperature; except that chillers using reciprocating compressors are often controlled from their entering water temperature, since staging and loading in steps causes steps in the leaving water temperature.

Chiller types are classified by type of refrigeration cycle: vapor-compression or absorption. In addition, those using the vapor-compression cycle are referred to by the type of compressor: centrifugal or positive displacement. A positive displacement compressor can be either reciprocating or screw for this discussion. See related ASHRAE and chiller manufacturers manuals for detailed information of chiller cycles.

VAPOR-COMPRESSION REFRIGERATION

VAPOR-COMPRESSION CYCLE

The vapor-compression cycle is the most common type of refrigeration system. When the compressor (Fig. 2) starts, the increased pressure on the high side and the decreased pressure on the low side causes liquid refrigerant to flow from the receiver to the expansion valve. The expansion valve is a restriction in the liquid line which meters the refrigerant into the evaporator. It establishes a boundary between the low (pressure) side, including the evaporator and the high (pressure) side, including the condenser and the receiver. The compressor is the other boundary. The liquid refrigerant in the evaporator boils as it absorbs heat from the chilled water. The refrigerant leaves the evaporator and enters the compressor as a cold low-pressure gas. The refrigerant leaves the compressor as a hot high-pressure gas and passes through the condenser where it is cooled by the condenser water until it condenses and returns to the receiver as a liquid. The cycle is the same regardless of the compressor type or refrigerant used.

Two common types of expansion valves are constant pressure and thermostatic. The constant pressure valve is suitable only when the load is constant. It is essentially a pressure regulator which maintains a constant pressure in the evaporator.

The thermostatic expansion valve is used for varying cooling loads, such as those found in HVAC systems. It has a remote temperature sensing element which is usually installed on the suction line between the evaporator and the compressor. It is set to adjust the expansion valve so there is a small amount of superheat in the suction line refrigerant. Superheat means that all of the liquid has evaporated and the vapor has been heated above the evaporation temperature by the water or air being cooled. This prevents liquid from entering the compressor.

A flooded shell and tube chiller evaporator (Fig. 3) is usually used with centrifugal compressors while, a direct expansion chiller evaporator (Fig. 4) is used with positive displacement compressors. In both cases the condenser is a large pressure cylinder (shell) with tubes connected to inlet and outlet headers. In the flooded shell and tube type evaporator, the shell is about 80 percent filled with refrigerant and the chilled water flows through the tubes. Heat from the water evaporates the refrigerant surrounding the tubes which cools the water. The refrigerant vapor rises to the top of the shell and into the refrigerant suction line.

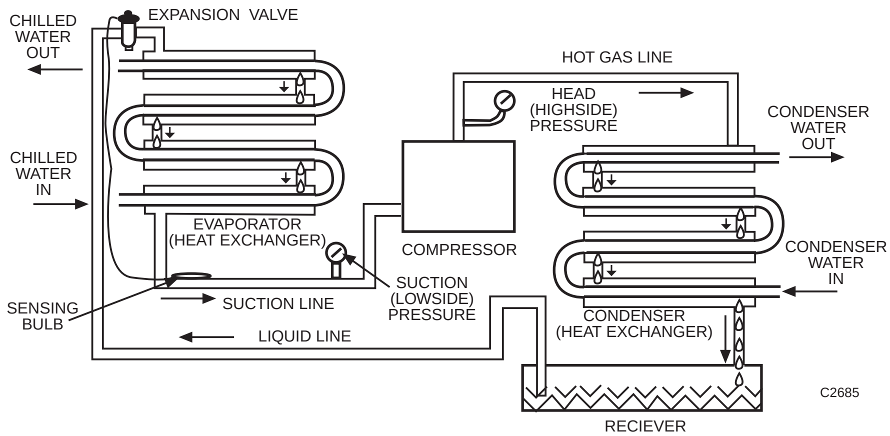


Fig. 2. Typical Vapor-Compression Cycle Water Chiller.

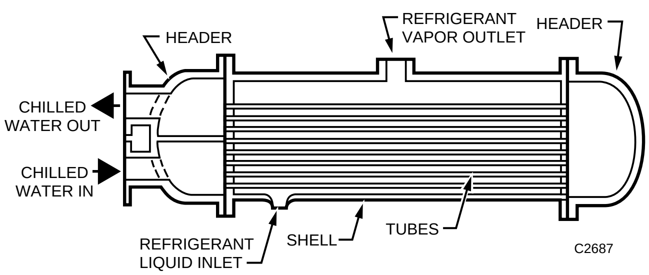


Fig. 3. Flooded Shell and Tube Chiller Evaporator.

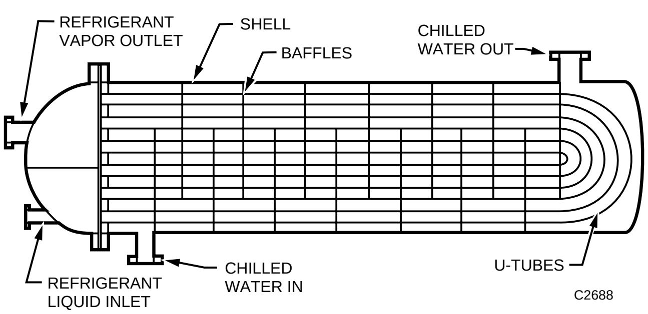


Fig. 4. Direct Expansion Chiller Evaporator.

The direct expansion chiller evaporator is the reverse of the flooded shell and tube chiller evaporator, water in the shell and the refrigerant in the tubes.

The compressor can be reciprocating, centrifugal, or screw type. The centrifugal and screw types are generally found on the larger systems.

The chiller condenser is usually water cooled but may be air cooled or evaporative cooled. The most common water cooled condenser is the shell and tube type (similar to Figure 3). The cooling (condenser) water flows through the tubes and the refrigerant vapor condenses on the cool tube surface and drops to the bottom of the shell where it flows into the liquid line to the receiver or evaporator.

An air cooled condenser is a series of finned tubes (coils) through which the refrigerant vapor flows. Air is blown over the coils to cool and condense the refrigerant vapor.

An evaporative condenser is similar to the air cooled condenser where the refrigerant flows through a coil. Water is sprayed over the coil and then air is blown over the coil to evaporate the water and condense the refrigerant. Evaporative condensers are rarely used because of the additional maintenance compared with an air cooled condenser.

CENTRIFUGAL COMPRESSOR

Centrifugal compressors are available in a wide range of sizes. Compressor capacity can be modulated from maximum to relatively low values. Centrifugal chiller systems can be designed to meet a wide range of chilled liquid (evaporator) and cooling fluid (condenser) temperatures.

Operation of the compressor is similar to a centrifugal fan or pump. Gaseous refrigerant enters the inlet (Fig. 5) and passes through inlet vanes into the chambers or blades radiating from the center of the impeller. The impeller, rotating at a high rate of speed, throws the gas to the outer circumference of the impeller by centrifugal force. This increases the velocity and pressure of the gas. The gas is then thrown from the impeller into the volute where most of the velocity (kinetic energy) is converted to pressure.

Use of a larger evaporator and condenser decreases the energy needed by the compressor for a given cooling load. Typical single stage high speed compressor construction is shown in Figure 5. The prerotation vanes (inlet guide vanes), located in the suction side of the compressor, control the gaseous refrigerant flow by restricting flow. As the vanes vary the flow, the compressor pumping capacity varies. In the open position the vanes give a rotating motion to the refrigerant in a direction opposite to the impeller rotation. This allows the chambers or blades to pick up a larger amount of gas.

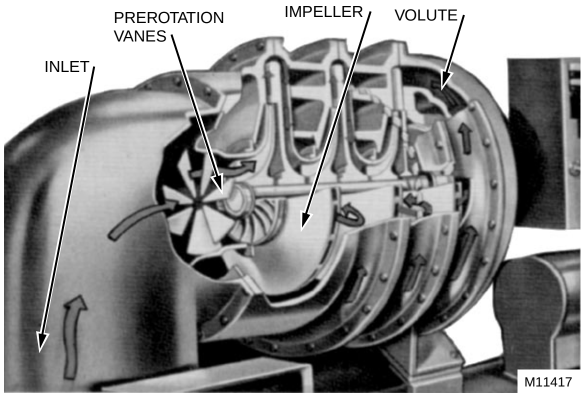


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Fig. 5. Cutaway of Single Stage Centrifugal Compressor.

Centrifugal compressors are driven by turbines, electric motors, or internal combustion engines. Inlet vane control or speed control varies the capacity. Each method has different performance characteristics. A combination of speed and inlet vane control provides the highest operating efficiency. Multiple stage direct drive type compressors are available in many configurations.

Refrigerant head is the pressure difference between the compressor inlet and outlet and is the primary factor affecting chiller efficiency. For a given load, reducing refrigerant head improves efficiency. Evaporation and condensation temperatures establish these pressures and are determined by chilled water temperature and condenser water temperature. Refrigerant head is reduced by the following:

– Reducing condenser water temperature.
– Raising chilled water temperature.
– Reducing load.
– Decreasing design differential temperature of evaporator and condenser heat exchangers by increasing the size of the heat exchangers.

The load for maximum chiller efficiency varies with chillers and chiller manufacturers, but is often 70 to 80 percent.

RECIPROCATING COMPRESSOR

The reciprocating compressor is a positive displacement device consisting of several cylinders and pistons. The crankshaft is driven by a motor or engine. Spring loaded valves allow low pressure refrigerant vapor to enter the cylinder on the downstroke and high pressure refrigerant vapor to exit on the upstroke. Because the compressor is a positive displacement device its capacity is not greatly influenced by refrigerant head. However, power required per unit of cooling is directly related to refrigerant head. Keeping condenser temperature as low as possible also reduces energy requirements, therefore, compressors with water cooled condensers use less power than air cooled condensers. However, condenser water temperature must not be allowed to go too low or there will not be enough pressure difference to circulate the refrigerant.

Reciprocating chiller capacity is controlled in stages (steps). Methods of capacity control include the following:

– Unloading cylinders
– On-off cycling of multiple compressors
– Hot-gas bypass
– Hot-gas through evaporator

Cylinder unloading or multiple compressor on-off cycling is sequenced by automatic controls. The cylinder inlet valves are held open so no compression takes place during cylinder unloading. Capacity control mechanisms and controls are usually packaged with the chiller.

Step control of refrigeration must provide a compromise between to frequent cycling and to wide temperature swings. Use of chilled water return temperature as controlling variable lengthens the compressor on and off cycles. When cylinder unloading is used, the minimum off time after the compressor is cycled off on low load, is normally less than for multiple compressors. Off time is critical because the refrigeration system must have time to equalize the pressure between high and low sides so that the starting load will not be too great for the motor.

SCREW COMPRESSOR

A screw compressor is a positive displacement device which uses two meshed helical rotors to provide compression. It is also known as a helical rotary compressor. Basic construction of a helical rotary twin screw compressor is shown in Figure 6. The capacity of a screw compressor can be modulated by speed control or a sliding valve that varies the length of compression area of the helical screws and bypasses some gas back to the inlet of the compressor.

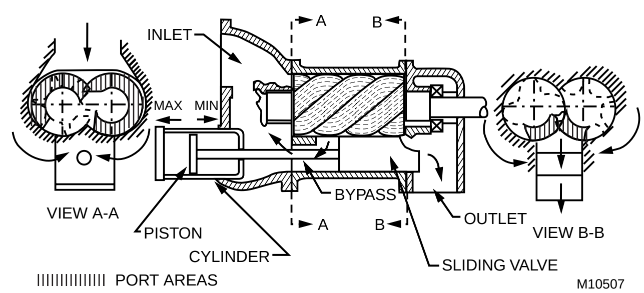

Fig. 6. Helical Rotary Twin Screw Compressor.

ABSORPTION REFRIGERATION

ABSORPTION CYCLE

The absorption cycle uses a fluid called an absorbent to absorb evaporated refrigerant vapor in an “absorber” section. The resulting combination of fluid and refrigerant is moved into a “generator” section where heat is used to evaporate the refrigerant from the absorbent.

In the absorber (Fig. 7) the absorbent, also called strong absorbent at this point, assimilates the refrigerant vapor when sprayed through it. The resulting weak absorbent is pumped by the generator pump through the heat exchanger, where it picks up some of the heat of the strong absorbent, then into the generator. In the generator the weak absorbent is heated to drive (evaporate) the refrigerant out of the absorbent and restore the strong absorbent. The strong absorbent then passes through the heat exchanger, where it gives up some heat to the weak absorbent, and then returns to the spray heads in the absorber completing the cycle for the absorbent.

NOTE: Industry standards reverse the definitions of strong absorbent and weak absorbent when ammonia is the refrigerant and water the absorbent.

The refrigerant vapor migrates from the generator to the condenser where it is cooled until it condenses to a liquid. The liquid refrigerant flows to the evaporator where the refrigerant pump sprays the liquid over the chilled water coils. The heat from the chilled water evaporates the liquid. The resulting vapor migrates to the absorber where it is absorbed by the strong absorbent and pumped to the generator to complete the refrigerant cycle.

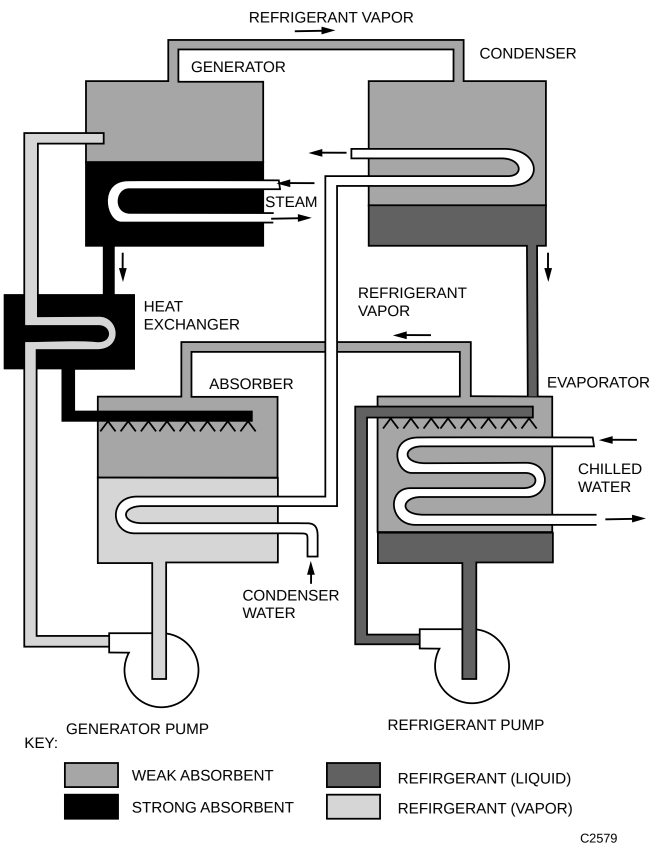

Fig. 7. Absorption Chiller Operating Cycle Schematic.

Figure 8 is a typical water-lithium bromide absorption cycle chiller. Lithium bromide is the absorbent and water is the refrigerant. Use of water as a refrigerant requires that the system be sealed, all the air removed, and an absolute pressure of 0.25 in. Hg be maintained. Under these conditions the refrigerant (water) boils at 40F which allows the refrigerant to cool the chilled water to 44F.

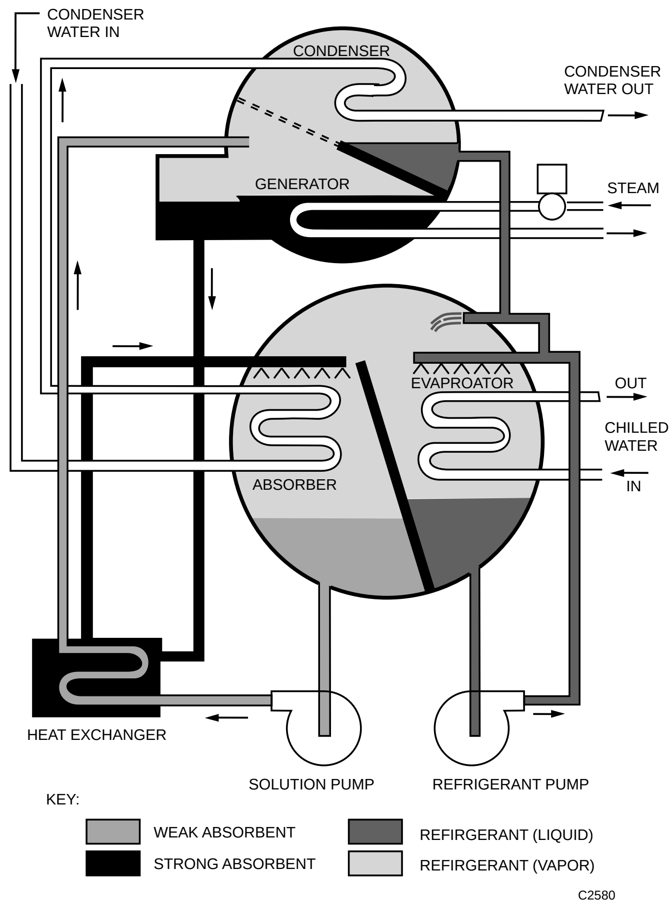

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Fig. 8. Diagram of Two-Shell Lithium Bromide Cycle Water Chiller.

ABSORPTION CHILLER

Capacity control of a water-lithium bromide absorption chiller is modulated by changing the concentration of the strong absorbent by varying the heat input to the process in the generator, controlling condenser water flow, or controlling flow of the strong absorber. Heat sources may be hot water, high temperature hot water, steam, or a gas flame (direct fired). Light loads require a reduced concentration of strong absorbent (absorbent retains more refrigerant) or less flow of the strong absorbent. The amount of heat required for a given cooling load is proportional to the temperature difference between condensing water and chilled water (refrigerant head). It is also proportional to temperature lift (chilled water temperature difference).

Some absorption chillers require the condensing water be kept constant at the design temperature. To improve seasonal operating efficiency some designs accept condensing water temperatures below design down to 45F. This requires an internal control that transfers liquid from refrigerant circuit to absorbent circuit, transfers liquid from absorbent circuit to refrigerant circuit, limits heat input, or a combination. Low condenser water temperature decreases energy usage and increases unit capacity.

When the condenser water temperature is too low, the absorbent contains too much refrigerant and the solution crystallizes. A safety control used by some absorption units senses when the lithium bromide concentration is getting too high with a low cooling water temperature and takes action to avoid crystallization.

Absorption chillers are normally used where heat energy is available at very low cost such as the exhaust from a steam turbine. They also are used to reduce electric load and therefore peak electric demand.

CHILLER CONTROL REQUIREMENTS

BASIC CHILLER CONTROL

Basic chiller control is a sensor in the chilled water supply or return and a controller to provide a control signal to a capacity control actuator. Capacity control is unique to each compressor type. Summarized, the controls for each compressor type are:

1. Centrifugal–Controller output operates a pneumatic or electric actuator to position inlet vanes as a function of the controlled temperature. If speed control is available, the controller sequences motor or rotor speed with inlet vanes.
2. Reciprocating–Controller provides a stepped output to sequence refrigerant solenoid valves, valve unloading, hot gas bypass, or multiple compressors as a function of controlled temperature.
3. Screw–Controller operates speed control or a pneumatic or electric actuator to position sliding bypass valve in response to temperature input.
4. Absorption–Controller output operates a valve to modulate the steam, hot water, or gas supply to maintain controlled temperature.

Capacity high limit controls are used on centrifugal and screw compressors to limit electrical demand during high load periods such as morning cool-down. A load limiting control reduces motor current draw to an adjustable maximum. Capacity of some chillers can be reduced to as low as 10 percent.

Most chillers with modulating capacity control use proportional-integral control, and often receive their chilled water setpoint from a BMCS to optimize building energy efficiency.

When chilled water discharge temperature control is used, offset can be reduced and control improved by using return water temperature to reset the supply setpoint upward at light loads to reduce the supply to return chilled water temperature difference.

Proportional-integral (PI) control improves the accuracy of control. When integral control is used, provisions to prevent integral wind-up must be made. Integral wind-up during system shutdown causes overshoot on start-up. For more information on PI control, refer to Control Fundamentals section.

SYSTEM CONTROLS WHICH INFLUENCE THE CHILLER

Whatever the configuration of a chilled water system, proper control is necessary to meet the overall system requirements. Condenser and chilled water temperatures establish refrigerant head and energy needed per unit of cooling. Minimum condenser temperature limits vary for different chiller designs. Condenser temperatures should be maintained as close to the minimum limits as possible to minimize refrigerant head. Actual condenser water temperature is dependent on outdoor wet bulb temperatures. Chilled water temperature is dependent on system design and building load.

SAFETY CONTROLS

When an unsafe condition exists, the compressor should stop automatically. Safety cutout controls may have automatic or manual reset and include the following:

1. High condenser pressure.
2. Low refrigerant pressure or temperature.
3. Backup for the low chilled water temperature controller (on some reciprocating chillers).
4. High motor temperature.
5. Motor overload.
6. Low oil pressure.
7. Low oil sump temperature.
8. High oil sump temperature.
9. Chilled water flow interlock.
10. Condenser water flow interlock.

The preceding are all two-position (on-off) controls. In addition, modulating limit controls sensing high condenser pressure or low evaporator pressure-temperature reduce compressor capacity to avoid the safety cutout conditions of Items 1 and 2.

CHILLER/BMCS INTERFACE

Most chillers are supplied with microprocessor controllers with a significant database of safety, operating, monitoring, and setpoint status and values. The BMCS normally provides control of the chilled water pump, the cooling tower fans, and the chiller system (AUTO/OFF commands to the chiller controller). The chilled water temperature setpoint and on occasions the maximum load setpoint (for capacity control) are also dictated by the BMCS.

It is desirable for the BMCS to have access to the chiller-controller database, but due to the cost and complexity of a custom interface to convert the data to a format acceptable to the BMCS, it is seldom done. Adoption of open communication standard protocols as the ASHRAE BACnet and the Echelon LonMark™ will replace the expensive interfaces with direct interfaces.

CHILLED WATER SYSTEMS

CENTRAL COOLING PLANTS

The central cooling system generates chilled water for distribution to a building or group of buildings. It consists of one or more chillers. Multiple chillers may all be the same or different capacities and/or different types. The energy may be provided by electricity or a fuel-combustion source. Central chiller system optimization is an important control function to minimize energy use, especially in multiple chiller plants. The control program must be dynamic and constantly check current conditions and adjust chiller system operations accordingly. A control program must select the most efficient loading and chiller combinations then, sequence pumps and control cooling towers to match the current load condition. Built-in safeguards prevent short cycling and exceeding demand limits. Strategies for total chiller system optimization include:

1. Supplying chilled water at a temperature that minimizes chiller and pump energy while satisfying the current demand load.
2. Selecting the chiller or chiller combination in multiple chiller plants to satisfy the current load at minimum operating cost. The influence of refrigerant head pressures and chiller efficiency curves must be considered.
3. Using rejected heat when a heating load exists at the same time as a cooling load.
4. Using thermal storage to store day time rejected heat and/or night time cooling. Thermal storage can also reduce the size of chiller equipment.

SINGLE CENTRIFUGAL CHILLER CONTROL

Capacity control is the primary method used to control a single chiller to meet the cooling load. Typically centrifugal chiller capacity control is accomplished by a chiller discharge water temperature controller. Discharge control responds quickly to load changes to maintain the chilled water temperature. The chilled water supply temperature may be reset from chilled water return temperature or from the zone with the greatest load. To ensure that all loads are met, resetting based on zone demand requires monitoring all the chilled water valves on the fan systems. Resetting from return water temperature recognizes the average temperature only and not the individual loads.

Where chilled water constant speed pumping horsepower is more than 25 to 33 percent of the compressor horsepower, increases in chilled water temperature could force the use of more pumping energy than can be saved in reduced compressor energy. This is because chilled water control valves open wider due to the increased water temperature. The increased flow requires the pump(s) to use more power. In these cases, chilled water reset should not be used or should be limited to reset only when flow is below the break even point of compressor versus pump energy.

SINGLE CENTRIFUGAL CHILLER CONTROL APPLICATION

FUNCTIONAL DESCRIPTION

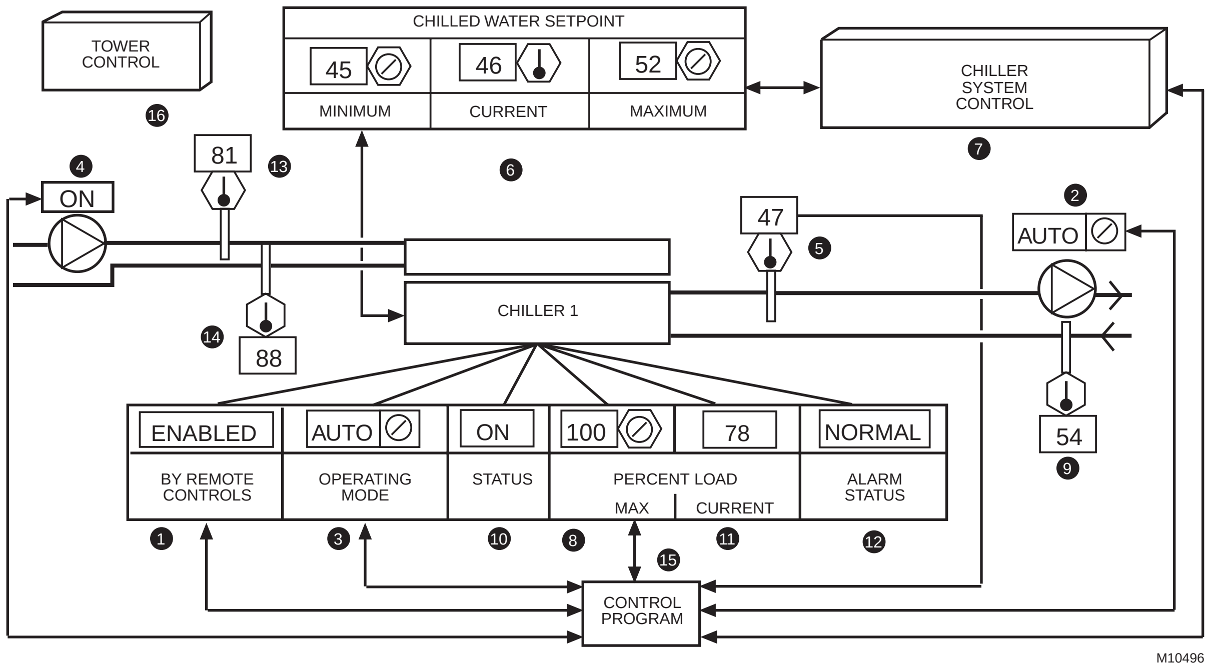


Fig. 9. Single Chiller Control Graphic.

Item No.	Function
1	Indicates when chiller system is required by fan system.
2	Chilled water pump ON-OFF-AUTO function. In AUTO pump runs when fan systems need chilled water.
3	Chiller ON-OFF-AUTO function (In ON and AUTO chilled water flow required for chiller to run).
4	Condenser pump status. Pump started by chiller controls when chilled water needed.
5,6	Chiller leaving water temperature and setpoint.
7	Icon to select chiller control dynamic sequence display (Fig. 10).
8	BMCS commandable load limiting function.
9-14	Operator information.
15	Control program coordinates chiller control.
16	Icon to select cooling tower control displays.

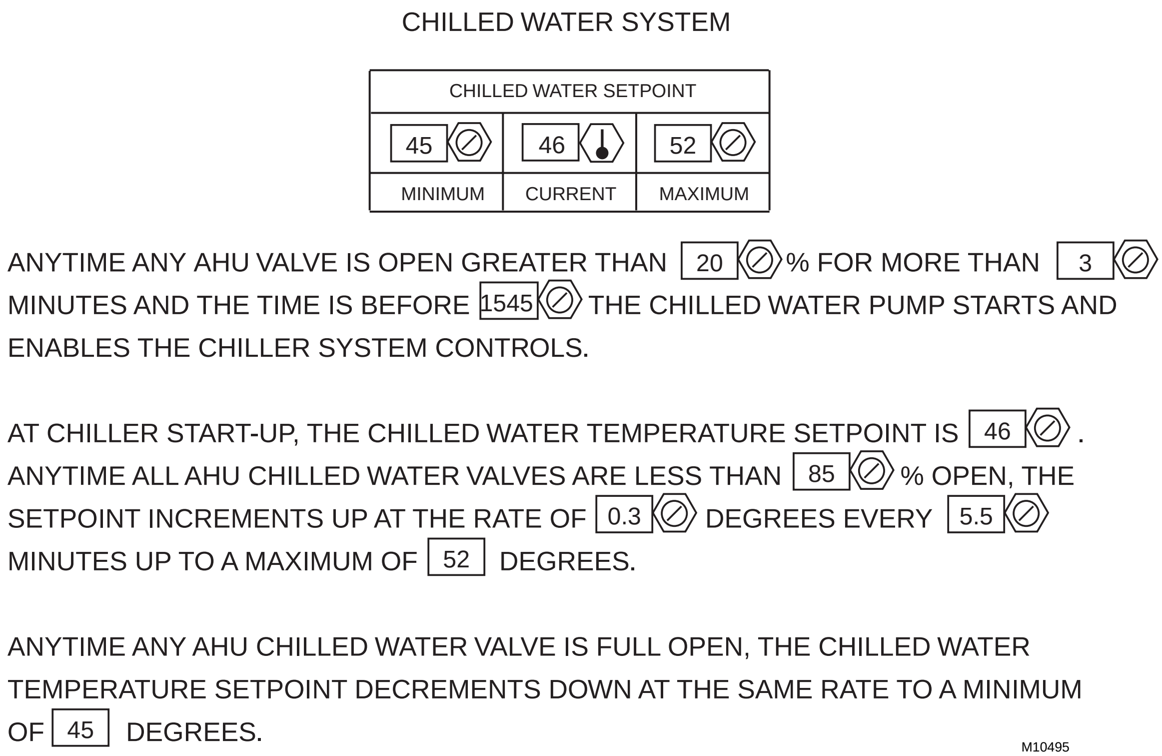


Fig. 10. Single Chiller Control Dynamic Sequence Display.

Features

1. Automatic start-stop and setpoint optimization of chiller.
2. User friendly monitoring and adjustment.
3. Optimized unoccupied operation.
4. Chiller cannot start late shortly before the unoccupied period begins.

Conditions For Successful Operation

1. Control network, software, and programming advises chiller controller of AHU chilled water demands.
2. Interlock and control wiring coordinated with the chiller manufacturer.
3. Appropriate cooling tower and control.
4. For single-chiller systems without primary-secondary pumping, three-way air handling unit valves may be used for 80 to 85 percent of the chilled water flow (Small valves, up to 15 to 20 percent total flow, may be two-way, which are simpler to pipe.) If all two-way valves are provided on single pump systems, chilled water flow or pressure controls (See DUAL CENTRIFUGAL CHILLERS) are provided to maintain the required flow (varies with chiller manufacturers) through the chiller. Do not use three-way valves when diversity is used in the chiller system design.

   NOTE: Little pumping energy can be saved on a single-pump single-chiller system by using two-way AHU control valves since the chiller usually requires high flow anyway.

5. During the unoccupied period the 80 percent load limiting parameter (see SPECIFICATION following) is based on the assumption that AHUs are VAV and are operating under a reduced maximum cfm setpoint during all unoccupied cooling modes (see the Air Handling System Control Applications) to save fan energy and place the chiller operation in the maximum efficiency range.
6. Chilled water temperature reset from AHU chilled water valve position requires:
   a. No valve always full open.
   b. Maximum of 30 to 40 valves. With too many valves the probability that one valve will always be open is very great.
   c. Zone setpoint discipline is maintained. Lowering setpoints to resolve complaints may result in one or more valves being always open.
7. Chilled water temperature reset timing increments are compatible with valve control response. If the temperature reset is too fast, the valve cannot adjust to the new temperature, resulting in instability.

Specification

The chiller system operation shall be enabled anytime the time of day is less than 1545 and any AHU chilled water valve is open greater than twenty percent for more than three minutes. Anytime the chiller system is enabled, the chilled water pump shall run.

Anytime chilled water flow is proven via a chilled water pump current sensing relay, the chiller controls shall be enabled to operate under factory controls, subject to a chiller software ON-OFF-AUTO function (chilled water flow must still be proven in the “ON” mode). Provide control and interlock wiring per the chiller manufacturers recommendation.

Upon a call for chilled water, the chiller controls shall start the condenser water pump and energize the cooling tower fan controls.

When condenser water flow is proven via a condenser water pump current sensing relay, the chiller shall start, operate, and load under chiller factory controls to maintain the chilled water temperature setpoint, 46F at start-up.

Anytime all chilled water valves are less than 85 percent open, the chilled water temperature setpoint shall be incremented at a rate of 0.3F every 10 minutes up to a maximum of 52F.

Anytime any chilled water valve is full open, the chilled water temperature setpoint shall be decremented at a rate of 0.3F degrees every 10 minutes down to a minimum of 45F.

The maximum allowable percentage of chiller full load electrical current shall be commandable from the BMCS, and shall be 80 percent during all unoccupied periods of operation.

MULTIPLE CHILLER SYSTEM CONTROL APPLICATIONS

Multiple chiller systems offer standby capacity and improved economy at partial loads. Multiple chiller systems may be piped for either parallel or series chilled water flow.

In the parallel piped arrangement (Fig. 11), return chilled water is divided among the chillers then recombined after chilling. Two methods of operation at light loads are depicted. One uses a pump and a check valve for each chiller. The other uses a common pump with an isolation valve for each chiller. Multiple pumps with check valves allow one chiller and the associated pump to be shut down during light load conditions to save energy and require that the system be able to operate with the reduced flow. The check valves prevent reverse flow through the shut down chiller. Use of a common pump and isolation valves require that the operating chiller be able to withstand full system flow. The isolation valves allow the operating chiller to supply only the chilled water temperature required to meet system demands. Without the isolation valves, half of the water flows through the chiller which is shut down and is not cooled. When the uncooled water is mixed with the cooled water, the temperature will be the average of the water temperatures. As a result, the on-line chiller must supply water cool enough so that the average will satisfy the primary sensor and thus the system. To meet this requirement the on-line chiller may need to supply water close to the freezing point.

The temperature sensor in the common chilled water supply is the primary capacity control. The temperature low limit control prevents the outlet temperature of each chiller from going too low. A return water temperature sensor can be used in conjunction with a supply water temperature sensor to turn off one chiller in light load conditions.

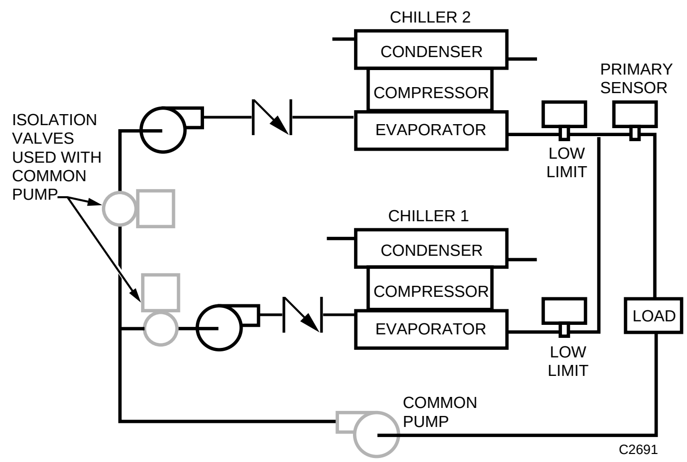


Fig. 11. Parallel Piped Chillers.

In the series arrangement (Fig. 12) chilled water pressure drop is higher if the chillers are not designed for higher flow. At partial loads, compressor power consumption is lower than for the parallel arrangement.

When the condensers of series units are water cooled, they are piped in series counterflow to balance loading. When piped series-counterflow, Chiller 1 receives warmer condenser and chilled water while Chiller 2 receives colder entering condenser and chilled water. This makes refrigerant head approximately the same for each chiller. The controls may be set to shutdown either chiller at partial loads.

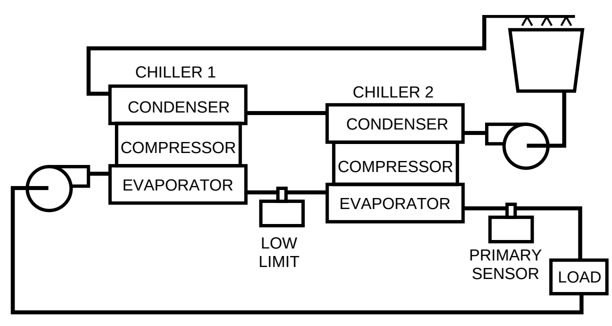


Fig. 12. Series Piped Chillers.

When two chillers of equal size and similar characteristics are used, the point at which the second chiller is activated is usually when the first chiller reaches 100 percent load. This is demonstrated in Figure 13 which plots kW per ton versus percent load for one chiller and two chillers at various temperature differences between condenser and chilled water temperatures. Curves vary slightly for different temperature differences so a microprocessor-based control system is used for maximum efficiency. The microprocessor checks chilled and condenser water temperature, looks up chiller efficiency at those temperatures, and calculates the optimum changeover point.

Curves A and B in Figure 13 illustrate that for the chillers operating at design condition with a 43F temperature differential (ΔT) between chilled and condenser water the second chiller must be added when the first chiller reaches 100 percent load (50 percent of chiller system capacity). The next set of curves (C and D) show the chiller is more efficient because of the smaller ΔT (31F) and that the first chiller can be loaded to 110 percent (55 percent of system load) before the second chiller is added. The third set of curves shows the extremely efficient operation with a 19F ΔT.

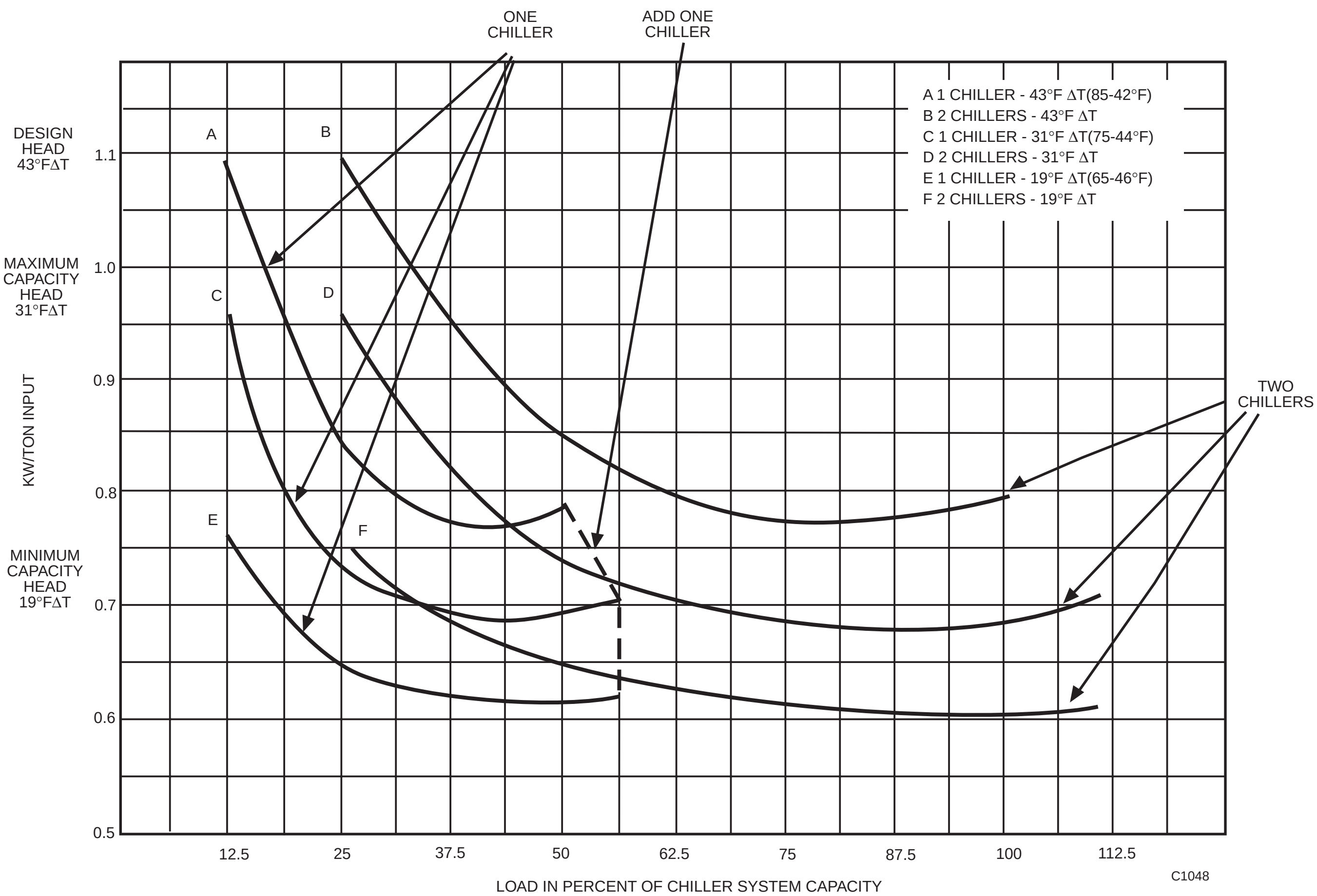


Fig. 13. Efficiency–Two Equal Size Chillers.

DUAL CENTRIFUGAL CHILLERS CONTROL APPLICATION

FUNCTIONAL DESCRIPTION

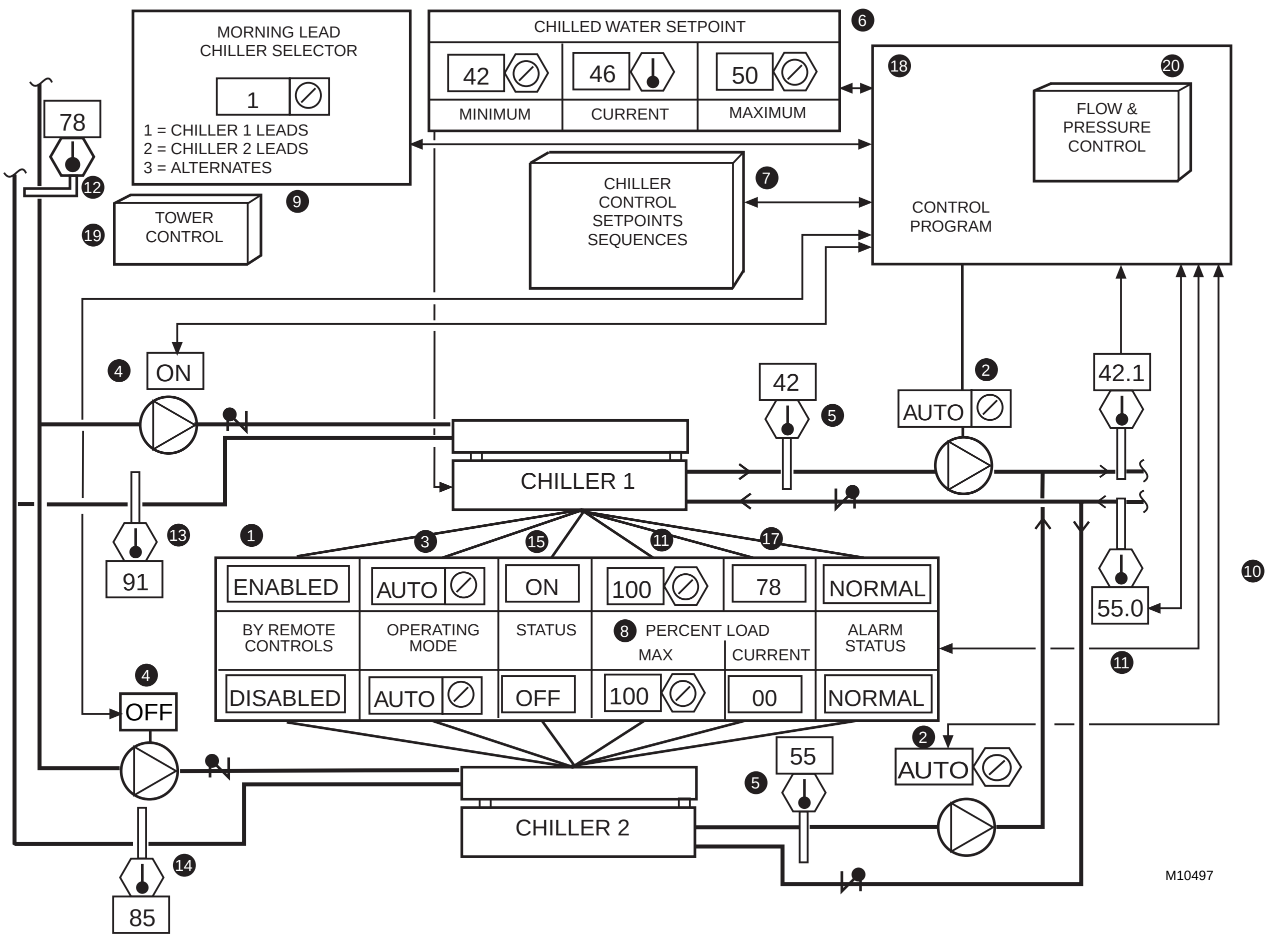


Fig. 14. Dual Centrifugal Chiller Control Graphic.

Item No.	Function
1	Indicates when chilled water is required by fan system.
2	Chilled water pump ON-OFF-AUTO function. In AUTO pump runs when system needs chilled water.
3	Chiller ON-OFF-AUTO function (In ON and AUTO chilled water flow required for chiller to run).
4	Condenser pumps status. Pumps started by chiller controls when chiller needed.
5,6	Chiller leaving water temperature and setpoint.
7	Icon to select chilled water setpoint reset dynamic sequence display (Fig. 15).
8	BMCS commandable load limiting functions.
9	Lead chiller selector function.
10-17	Operator information.
18	Control program coordinates chiller staging and control.
19	Icon to select cooling tower control displays.
20	Icon to select chilled water flow and pressure control displays.

CHILLED WATER SYSTEM

CHILLED WATER SETPOINT		
42	46	50
MINIMUM	CURRENT	MAXIMUM

ANYTIME ANY AHU VALVE IS OPEN GREATER THAN 20 % FOR MORE THAN 3 MINUTES AND THE TIME IS BEFORE 1545, THE LEAD CHILLED WATER PUMP STARTS.

ANYTIME THE LEAD CHILLER HAS RUN LONGER THAN 90 MINUTES, THE CHILLED WATER TEMPERATURE IS GREATER THAN 1.0 DEGREES ABOVE SETPOINT FOR MORE THAN 4 MINUTES, THE BYPASS VALVE IS CLOSED, AND THE TIME IS BEFORE 1545, THE OFF CHILLED WATER PUMP STARTS.

ANYTIME BOTH CHILLERS ARE RUNNING AND THE WATER DIFFERENTIAL TEMPERATURE IS LESS THAN 4.4 DEGREES FOR MORE THAN 4.0 MINUTES, THE CHILLER WHICH HAS RUN LONGEST STOPS AND REMAINS OFF AT LEAST 30 MINUTES.

ANYTIME ALL AHU CHILLED WATER VALVES ARE LESS THAN 80 % OPEN THE CHW TEMP. SETPOINT INCREMENTS UP AT THE RATE OF 0.3 DEGREES EVERY 5 MINUTES TO A MAXIMUM OF 52 DEGREES.

ANYTIME ANY AHU CHILLED WATER VALVE IS FULL OPEN, THE CHW TEMPERATURE SETPOINT DECREMENTS DOWN AT THE SAME RATE TO A MINIMUM OF 45 DEGREES.

ANYTIME ANY CHILLER IS RUNNING AND ALL AHU VALVES ARE OPEN LESS THAN 20 % FOR MORE THAN 3 MINUTES, AND THE CHILLER HAS RUN GREATER THAN 30 MINUTES, THE CHILLED WATER PUMP STOPS.

M10498

Fig. 15. Dynamic Chiller Control Display.

Features

1. Automatic start-stop, staging, and setpoint optimization of chillers.
2. User friendly monitoring and adjustment.
3. Optimized unoccupied operation.
4. Chiller that has run longest since last start is first to stop.
5. Chillers cannot start late (shortly before the unoccupied period begins).

Conditions For Successful Operation

1. Control network, software, and programming advises chiller controller of AHU chilled water demands.
2. Interlock and control wiring coordinated with the chiller manufacturer.
3. Appropriate cooling tower and control is required. See COOLING TOWER AND CONDENSER WATER CONTROL.
4. Two-way AHU control valves. This allows good single-chiller, single pump operation.
5. Appropriate chilled water flow and differential pressure controlled bypass valve to keep the minimum required flow through the chillers (varies with chiller manufacturers).
6. The unoccupied period 80 percent load limiting parameters are based on the assumption that VAV AHUs are operating under a reduced maximum cfm setpoint during all unoccupied cooling modes (see the Air Handling System Control Applications section) to save fan energy while simultaneously placing the chiller operation in its maximum efficiency range.

NOTE: When using two-way AHU valves with this coupled chiller configuration, exercise care in optimizing the chilled water temperature. With both chillers running, raising the chilled water temperature results in greater flow and a smaller ΔT which must be considered in the chiller shedding strategy. For an example of a pressure bypass system, see DUAL PUMPS, DUAL CHILLERS, PRESSURE BYPASS, 90 PERCENT CHILLER FLOW, DIRECT RETURN.

Specification

The chiller system operation shall be enabled anytime the time of day is before 1545 and any AHU chilled water valve is open greater than twenty percent for greater than three minutes. Anytime the chiller system is enabled, the lead chilled water pump shall run.

Anytime the lead chiller has run longer than 90 minutes, the chilled water temperature has been greater than 1 degree above the chilled water temperature setpoint for greater than 4 minutes, and the time is less than 1545, the off chilled water pump shall start.

Anytime both chillers are running and the chiller plant water differential temperature has been less than 4.4F for greater than 4 minutes, the chilled water pump with the longest “on” duration since the last start shall stop and remain off at least 30 minutes.

Anytime chilled water flow is proven via a chilled water pump current sensing relay, the respective chiller controls shall be enabled to operate under factory controls, subject to a chiller software ON-OFF-AUTO function (chilled water flow must still be proven in the “ON” mode). Provide control and interlock wiring per the chiller manufacturers recommendation.

Upon a call for chilled water, the chiller controls shall start the condenser water pump and energize the cooling tower fan controls.

When condenser water flow is proven via a condenser water pump current sensing relay, the chiller shall start, operate, and load under chiller factory controls to maintain the chilled water temperature setpoint, 44F at start-up.

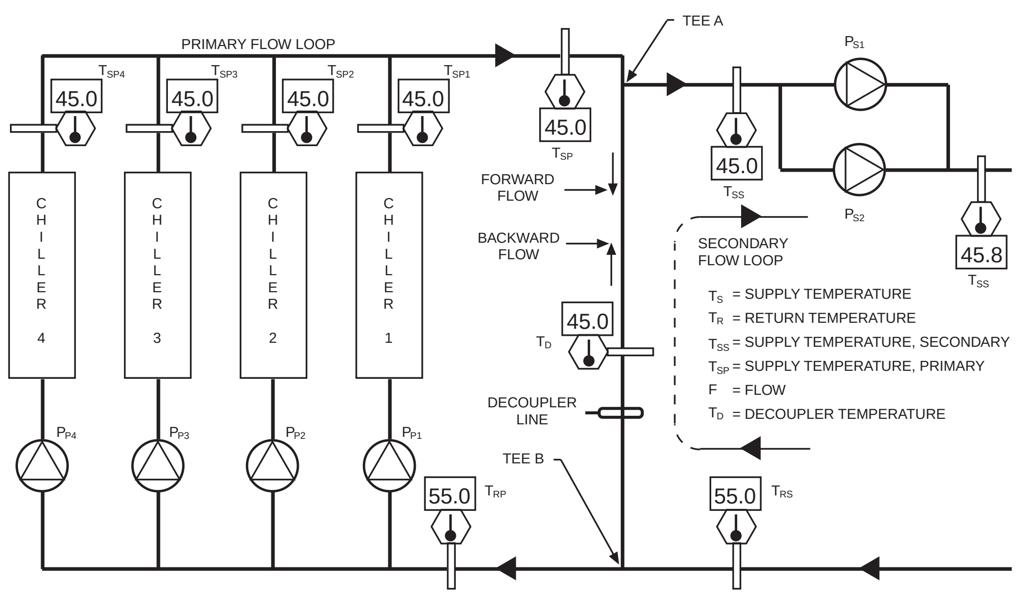
Anytime all chilled water valves are less than 80 percent open, the chilled water temperature setpoint shall be incremented at a rate of 0.3F every 5 minutes up to a maximum of 50F.

Anytime any chilled water valve is full open, the chilled water temperature setpoint shall be decremented at a rate of 0.3F every 5 minutes down to a minimum of 42F.

The maximum allowable percentage of chiller full load electrical current shall be commandable from the BMCS, and shall be 80 percent during all unoccupied periods of operation.

SIMILAR MULTIPLE CENTRIFUGAL CHILLERS CONTROL APPLICATIONS

Equal Sized Centrifugal Chillers Control



$$F_S = F_P \frac{\Delta T_P}{\Delta T_S}$$

WHERE F_S = SECONDARY FLOW
F_P = PRIMARY FLOW,
ΔT_P = DIFFERENTIAL TEMPERATURE, PRIMARY
ΔT_S = DIFFERENTIAL TEMPERATURE, SECONDARY

NOTE: F_P = SUM OF PRIMARY PUMP FLOWS BASED UPON TEST-AND-BALANCE DATA AND PUMP ON-OFF STATUS.

M10499

Fig. 16. Multiple Equal Sized Chillers Control Graphic.

SYSTEM DESCRIPTION

Figure 16 shows a typical “decoupled” multiple chiller system. Each chiller has a (primary) dedicated constant speed pump selected to produce the chiller design flow through the primary loop, including the “decoupler line”. The decoupler line isolates the primary and secondary pumping systems and handles any imbalance between the two flow loops. The decoupler line is typically sized to handle the flow of the largest primary pump at a negligible pressure drop, should be at least 6 to 10 pipe diameters in length, and the tees at each end should be configured as shown to oppose any undesirable induced flow forces. Decoupler flow should always be forward, not to exceed the flow of one chiller. Any backward decoupler flow will dilute the secondary chilled water supply with secondary return water thus raising the secondary supply temperature above design.

The secondary pumping system is variable volume and may contain many varieties of pumping loops.

Control, the staging of chillers on and off, is normally:

- Start a chiller anytime the decoupler has a backward flow.
- Stop a chiller anytime the decoupler forward flow exceeds that of the next chiller to be shed.

SOFTWARE PARTITIONING

From an operational and control perspective, the physical configuration of chiller plant digital controllers is usually transparent. The configuration varies, depending upon:

- Chiller staging algorithm.
- Redundant/backup control requirements.
- Condenser water system configuration.

NOTE: Where water leaving cooling towers becomes common before being extended to the chiller plant, a single cooling tower isolating, staging, and loading algorithm is usually preferred.

- Chiller monitoring requirements.
- Controller capacity for monitoring and control.
- Other project-unique requirements.

Figure 17 is a schematic of a digital system configuration. Each chiller has a dedicated cooling tower and a dedicated controller for chiller, cooling tower, and condenser water monitoring and control. Figure 18 shows a variation of Figure 17 where condenser water is common to all chillers and the cooling towers are staged in response to condenser water demand.

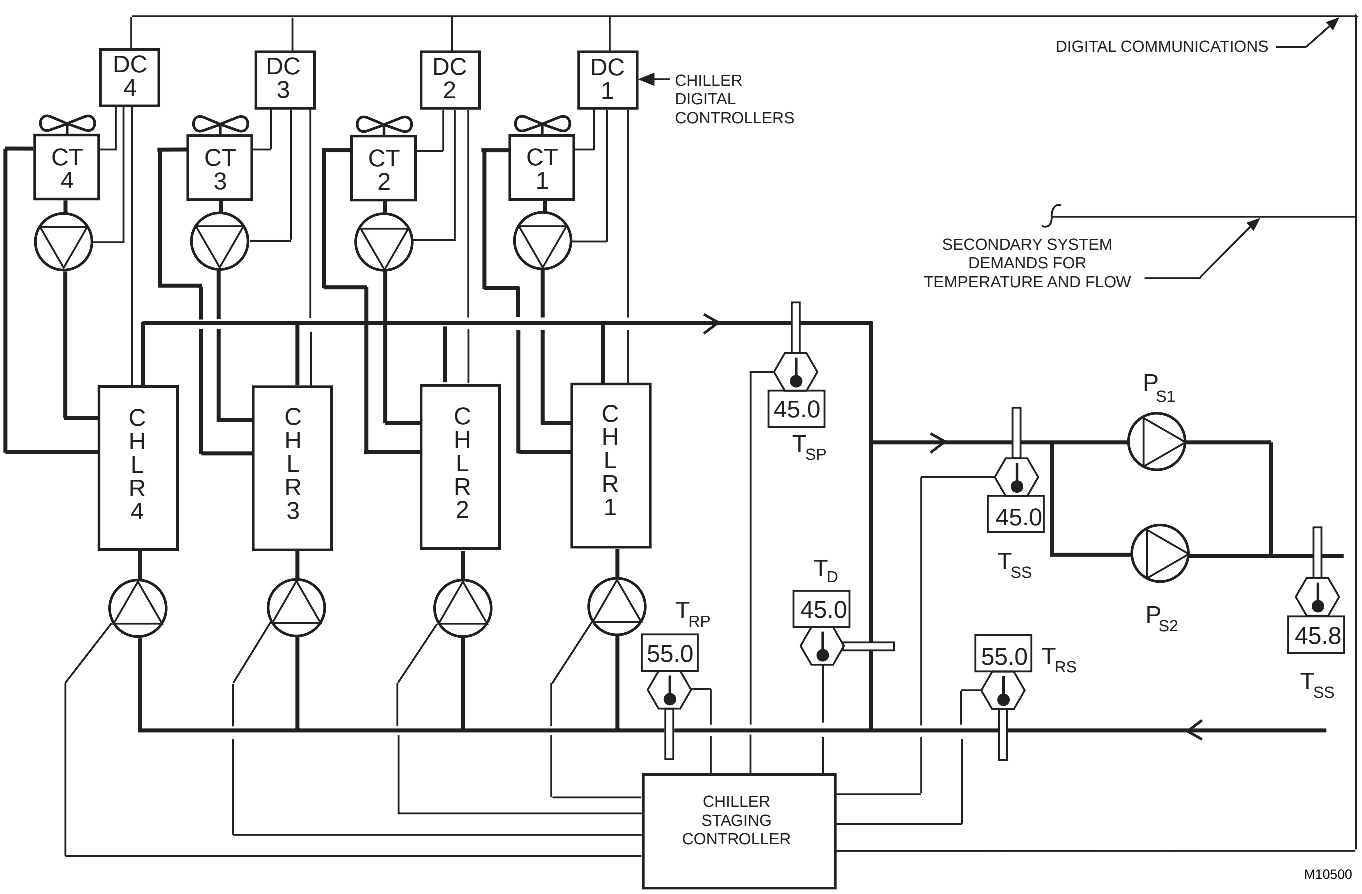


Fig. 17. Typical Digital Controller Configuration for Multiple Chillers.

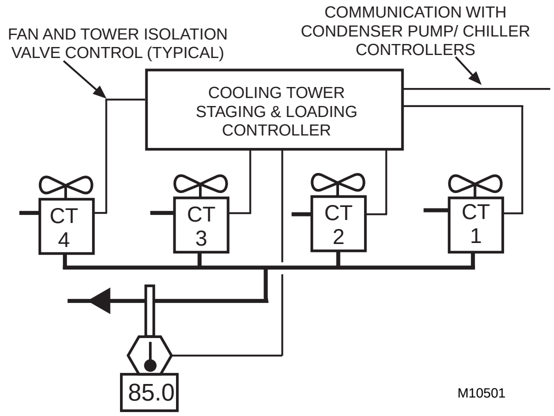


Fig. 18. Digital control of Sequenced Cooling Towers.

Multiple Centrifugal Chiller Sequencing

FUNCTIONAL DESCRIPTION

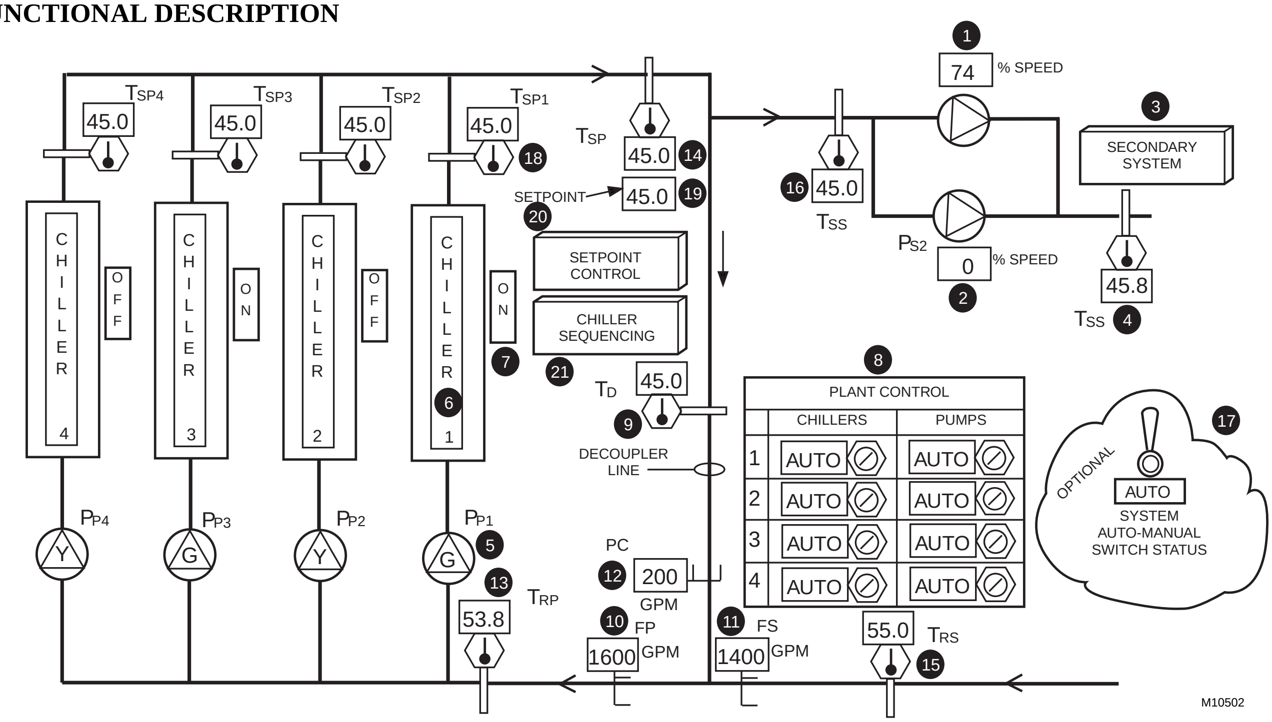


Fig. 19. Control Graphic for Multiple Similar Chillers.

Item No.	Function
1,2	Secondary pump speeds.
3	Icon for selection of secondary control details.
4	Secondary pump leaving water temperature (operator information).
5	Four chiller pump status indicators (green = on, yellow = off, red = alarm) (typical).
6	Four Icons for selection of chiller detail graphic.
7	Four chiller status indicators (typical), operator information.
8	BMCS commandable AUTO-OFF functions for each chiller and ON-OFF-AUTO functions for each chiller pump.
9	Decoupler temperature—indicates direction of flow.
10	Primary flow—indicates primary loop loading.
11	Secondary flow—indicates secondary loop loading.
12	Decoupler flow the difference between primary and secondary flows.
13-16	Temperatures for calculating secondary flow.
17	Status of optional AUTO-MANUAL toggle switch.
18	Four chiller CHWS temperature indicators.
19	Operator information (from secondary system).
20	Icon for selection of chilled water setpoint details display.
21	Icon for selection of chiller sequencing display (Fig. 20).

CHILLER SEQUENCING

CHILLED WATER PUMP OFF LONGEST IS STARTED UPON SECONDARY SYSTEM DEMAND FOR CHILLED WATER.

CHILLED WATER PUMPS ARE SEQUENTIALLY STAGED ON ANYTIME THE DECOUPLER TEMPERATURE EXCEEDS THE PRIMARY SUPPLY WATER TEMPERATURE GREATER THAN 2.0 DEGREES FOR GREATER THAN 3.0 MINUTES, BUT WITH NO LESS THAN 7 MINUTES BETWEEN STARTS.

CHILLED WATER PUMPS ARE STAGED OFF ANYTIME THE FORWARD DECOUPLER FLOW EXCEEDS 820 GPM FOR GREATER THAN 4.0 MINUTES, BUT WITH NO LESS THAN 4.0 MINUTES BETWEEN STOPS.

M10503

Fig. 20. Multiple Chiller Sequencing.

Features

1. Automatic start-stop sequencing of multiple decoupled chillers.
2. User friendly monitoring and adjustment.
3. Flow calculations without costly and maintenance-prone flow meters (see SPECIFICATION).
4. Chiller that has been off longest is next to start.
5. Constant flow through chillers with a variable flow secondary water system.

Conditions For Successful Operation

1. Control network, software, and programming to advise chiller plant controller of AHU chilled water demands.
2. Interlock and control wiring coordinated with the chiller manufacturer.
3. Appropriate cooling tower and control.
4. Two way AHU control valves provide variable flow secondary operation.
5. Precise and matched temperature sensors for accurate flow calculation. Refer to the flow equation in Figure 16.
6. Proper and precise positioning of primary return water well and sensor to get accurate measurement of mixed water temperature.
7. Digital controller configuration to suit cost and reliability requirements.

Specification

Chiller Plant Start-Up:

Anytime any secondary pump starts, the chiller plant controls shall be enabled, and the chiller pump that has been off longest shall start, subject to its software ON-OFF-AUTO function and its respective chiller software AUTO–OFF function. Pump/chiller combinations with either function OFF shall be removed from the control sequence.

When any chiller pump flow is proven, its respective chiller controls shall be energized. Upon a call for cooling by the chiller controls, the chiller controls shall enable the condenser water system controls and, upon proof of condenser water flow, the chiller shall start. Starting, loading, and interlock wiring shall be as recommended by the chiller manufacturer.

Chiller On-staging:

Anytime a chiller has operated greater than 50 minutes and the decoupler line temperature is greater than the chiller leaving water temperature setpoint by greater than 1.0F for greater than 5 minutes, the off chiller pump that has been off longest shall start.

Chiller Off-staging:

Anytime more than one chiller is operating and the decoupler has a supply chilled water flow in excess of the capacity of one chilled water pump for greater than 3.0 minutes, the chiller that has been running longest shall stop.

Chilled water flow calculations:

The primary supply water flow shall be calculated by summing the design water flow for all operating chiller pumps (each pump shall have a commandable value for its design flow).

$$F_P = \sum Flow(P_1, P_2, ...P_n)$$

The secondary water flow shall be calculated by dividing the product of the primary flow times the primary water differential temperature by the secondary water differential temperature.

$$F_S = \frac{F_P(T_{RP} - T_{SP})}{(T_{RS} - T_{SS})}$$

The decoupler flow shall be calculated by subtracting the secondary return water flow from the primary supply water flow.

$$F_D = F_P - F_S$$

The temperature sensors for flow calculation shall be platinum and software field-matched to within 0.1 degree at 50F. The primary return water sensor shall be in a stainless steel well extended at least 50 percent of the distance across the pipe and positioned as far away from the decoupler/secondary return mixing tee as possible.

DISSIMILAR MULTIPLE CENTRIFUGAL CHILLERS CONTROL

When a multiple chiller system consists of chillers that are different in capacity and/or characteristics, the control problem becomes more difficult. The more dissimilarities between chillers, the greater the potential error in using a fixed sequence and changeover schedule. On-line computer analysis which takes into account the many variations in chiller conditions and different loading conditions can significantly improve efficiency.

For optimization of a multiple chiller system, the efficiency curves (kW per ton vs load) for each chiller at all temperature differentials (condenser water temperature vs chilled water temperature) from minimum load to design must be known. Condenser water, chilled water and return water temperatures and flows, when appropriate, are used to calculate the optimum chiller combination and loading for each chiller. If the decision is to add or drop a chiller, minimum off and on time restrictions for each chiller must be considered. If the on/off restriction prevents adding the desired chiller, a second decision is made to pick the next most efficient combination or wait for the on/off time restriction to expire.

The program checks water temperatures and flow constantly, recalculates the combinations and loading, and turns chillers on and off accordingly. If a peak power demand limitation applies, demand is checked and the demand limit avoided.

Starting at minimum load, a typical calculation sequence selects and starts the most efficient chiller for that load. Data is constantly checked and as the load increases to the optimum load for adding a second chiller (based on current load, temperature differences, and efficiency curves for that condition), a second chiller is selected and started. The loading of each chiller is then adjusted for maximum system efficiency. As conditions change and the load increases or decreases, the loading of each chiller is adjusted and chillers are added or dropped within the limitations of the parameters input into the computer.

ALTERNATIVE MULTIPLE DECOUPLED CHILLER CONTROL

Another chiller staging approach measures the secondary load (via flow and differential temperature) and stage the chillers on and off to match the demand. Since the leaving chilled water temperature and the entering condenser water temperature are frequently optimized and chiller capacity varies with changes in either temperature, the per-chiller load expected should be dynamically modified based upon manufacturers data regarding these variations. This complicates an otherwise simple strategy.

COMBINATION ABSORPTION AND CENTRIFUGAL CHILLER SYSTEM CONTROL

In large buildings or campus installations using medium or high pressure steam, a combination of a turbine driven centrifugal chiller and an absorption chiller may be economical. The steam is routed first to the turbine driving the centrifugal chiller (Fig. 21). The exhaust from the turbine is then routed to the absorption chiller. The system is symmetrical so that the absorption chiller uses all of the exhaust steam from the turbine. System water piping is shown in Figure 22.

With the ABSORPTION/COMBINATION SELECTOR switch (Fig. 21) in the combination position, temperature controller T1, with its sensor in the chilled water supply, controls the centrifugal compressor and pressure controller P1 controls the absorption chiller capacity control valve so that all of the exhaust steam from the centrifugal chiller turbine is used by the absorption chiller. This also maintains a constant back pressure on the turbine. When the cooling load is reduced, less exhaust steam is available from the turbine and P1 positions the capacity control valve to reduce the capacity of the absorption chiller until the steam balance is achieved. Whenever the absorption chiller condenser water pump is running, pressure controller P2 allows temperature controller T2 to control the cooling tower bypass valve to maintain the condenser water temperature required by the absorption chiller. The centrifugal chiller is normally turned off when the system load is 15 to 35 percent. The cooling tower fans are controlled by a temperature sensor in the tower leaving water.

With the ABSORPTION/COMBINATION SELECTOR switch in the absorption position, the centrifugal chiller is off and temperature controller T1 modulates the absorption chiller capacity control valve to maintain the required chilled water temperature. The pressure reducing valve (PRV) supplies steam to the absorption chiller.

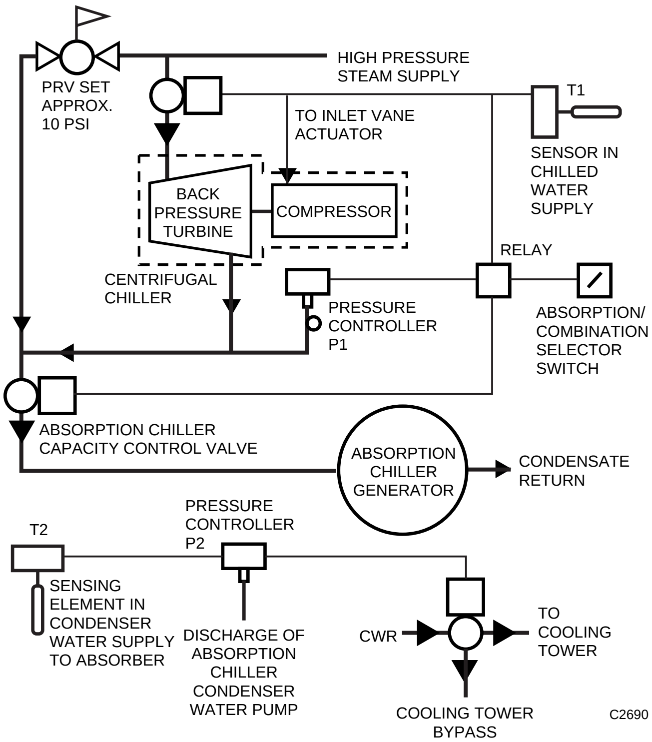


Fig. 21. Schematic of Combination Absorption/ Centrifugal Chiller System Steam Piping and Control Circuit.

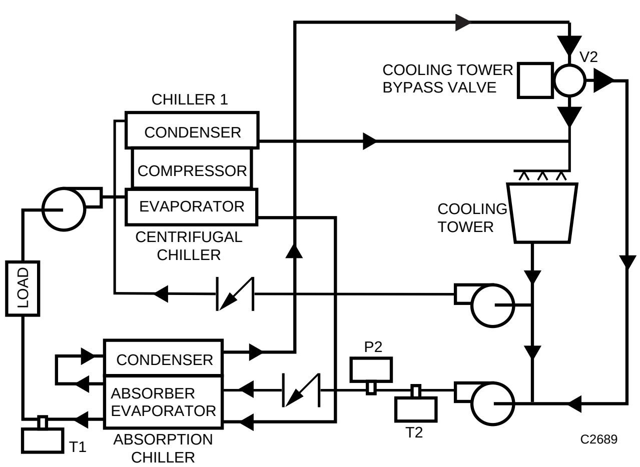


Fig. 22. Schematic of Combination Absorption/ Centrifugal Chiller System Water Piping.

Figure 23 compares the steam consumption of individual chillers to a combination system showing the savings using the combination system. The range of centrifugal cutout points indicates where most centrifugal chillers are shut down and the absorption chiller provides all cooling required by the system.

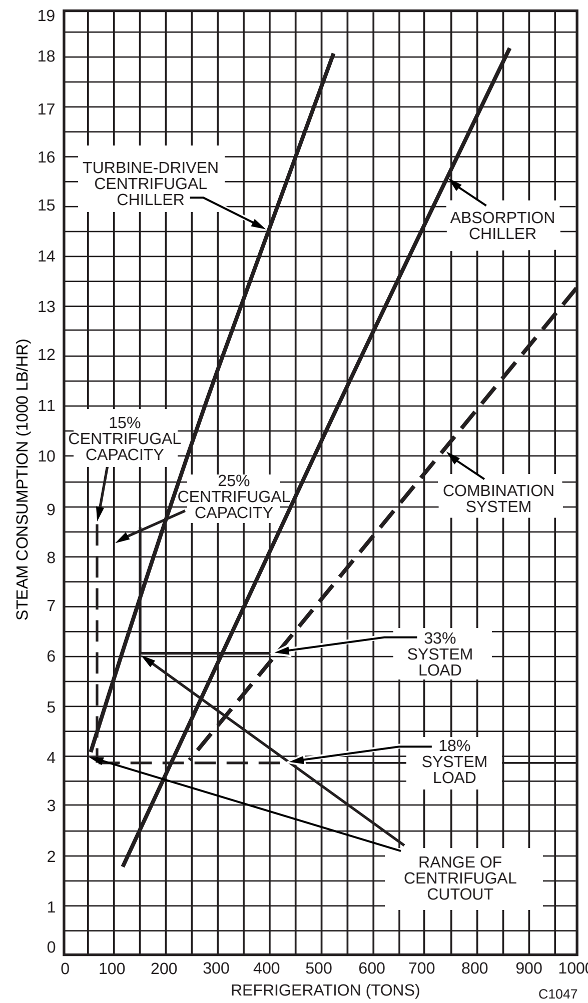


Fig. 23. Typical Steam Consumption, Individual and Combination Chillers.

CHILLER PUMP OPTIMIZATION

Pump configuration (variable water flow versus constant water flow) also affects chiller system efficiency. Constant water flow causes the largest change in efficiency from system no load to design because the pump energy becomes a greater percentage of the total energy input at light load conditions. Variable water flow, using two-way load valves or variable speed pumps, maintains the pump load at nearly a constant percentage of the total energy input at all loads. If a chiller system has constant water flow, chiller optimization must include pump energy usage as part of the power calculation to account for energy cost variations over the range of loads. In a chiller system using variable water flow, it is not necessary to account for pump energy cost. Refer to HOT AND CHILLED WATER DISTRIBUTION SYSTEMS AND CONTROL.

THERMAL STORAGE CONTROL

GENERAL

Thermal storage is used to save heating or cooling for future use. Typically enough storage is provided to meet the load requirements for a 24-hour period. During the cooling season, storage of low cost night time cooling can save energy and demand charges and reduce the chilled water generating equipment design size. During the heating season, storage of rejected day time excess refrigeration heat or solar heat can be used for night time heating loads. Storage may use a water tank or ice bin for cooling and a water tank or thermal conducting fluid for heat. The storage efficiency depends on the amount of insulation and, in the case of water storage, on minimizing the mixing of return water with storage water. Mixing in water storage can be minimized by use of a segmented container for storage (Fig. 24).

Primary control requirements are storage charge and storage discharge at the proper times and rates. Storage charge, storage discharge, and instantaneous cooling are the three basic control modes. Combinations of the basic control modes may be required to meet the load or predicted load. When the predicted load is high, the chiller provides chilled water for both the current load and for storage. When the predicted load occurs, the stored cooling is added to the chiller output.

CONTROL MODES

The appropriate control mode depends on the load predicted for next day, relative storage size, and relative costs of stored and instantaneous cooling, including electrical demand rate structures. The storage charging cycle is normally activated when cooling generation cost is lowest, such as at night when lower condenser water temperatures provide lower refrigeration head and when lower time-of-day electric rates may be applicable. The rate of charge should satisfy storage quantity requirements within the limits of the time available.

Use of stored energy verses instantaneous cooling energy is prioritized. When enough stored energy is available to satisfy the load through the peak demand period of the next day, only stored energy is used (storage priority). In this case, the charging cycle is scheduled to start when low cost cooling is available. The charging cycle is stopped when storage is sufficient for the next days load or when the storage capacity is filled. The storage discharge cycle is controlled to meet load conditions. If storage capacity is not large enough for the next day, the chiller supplements the use of stored cooling, as necessary. The control sequences for segmented chilled water storage (Fig. 24) are:

– Instantaneous Cooling Cycle: Pump P_1 is on and Pump P_2 is off. Valves V_C and V_D are closed. Valve V_I is controlled by ΔP and T_1 controls chiller.
– Charging Cycle: Valves V_D and V_I are closed and V_C is controlled by T_6 to maintain the flow rate at F_1. Pump P_2 is off. T_1 controls chiller capacity to maintain 40F CHWS temperature. When the T_5 (A through E) location, representing the needs for the next day, reaches 40F, the charging cycle is stopped.
– Charging plus Instantaneous Cooling Cycle: Charging cycle continues while V_I is controlled from ΔP. Pump P_2 is off. The flow through the storage is T_{5E} to T_{5A}.
– Discharge Cycle (enough storage for anticipated load): Valves V_C and V_I are closed, chiller and Pump P_1 are off, and Pump P_2 is on. Valve V_D is controlled from ΔP.
– Discharge plus Instantaneous Cooling Cycle (not enough storage for anticipated load): Pumps P_1 and P_2 are on, V_C is closed. Chiller demand D_1 limits chiller capacity while T_2 positions V_I to maintain chilled water supply temperature. ΔP throttles valve V_D to provide flow from storage.

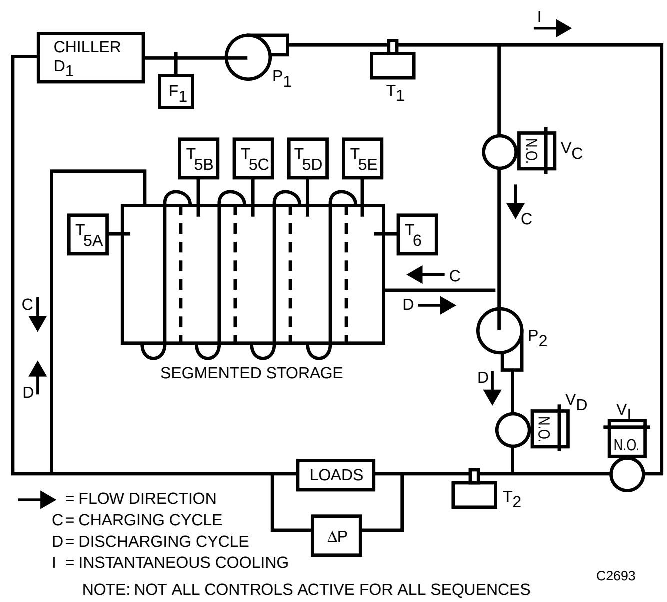


Fig. 24. Segmented Chilled Water Storage.

The chiller takes priority when stored energy costs are larger than instantaneous energy costs. Under these conditions the only cooling stored is that required to reduce the anticipated demand limit for the next day. Storage size and energy cost, including demand charges, establish the control strategies for this situation. The control objectives are to limit demand for the billing period and to minimize energy costs. Chiller discharge is controlled to meet load. The sequence of the control is to run the charging plus instantaneous cooling cycle, then stop charging when the quantity necessary to meet the anticipated demand limit for the next day is stored.

COOLING TOWER AND CONDENSER WATER CONTROL

COOLING TOWER PERFORMANCE CHARACTERISTICS

The cooling tower dissipates the heat collected from the building by the chiller and chilled water system by cooling the condenser water. Evaporatively cooled condenser water will cool refrigerant to within 7F of the outdoor air wet-bulb temperature. Air cooled condensers will cool refrigerant to within 20F of the outdoor air dry-bulb temperature. A cooling tower normally provides a refrigerant head about 30F lower than an air cooled condenser. This means an evaporative cooling tower provides a significantly lower cooling cost than an air cooled condenser.

Figure 25 shows water-air counterflow in a cooling tower. The fill increases the time that the water and air are in contact making the cooling tower more efficient. Fill is generally one of two types, either splash or film type. A splash type fill is a series of slats over which the water cascades. Film type fill causes the water to flow down thin closely spaced vertical sheets of plastic.

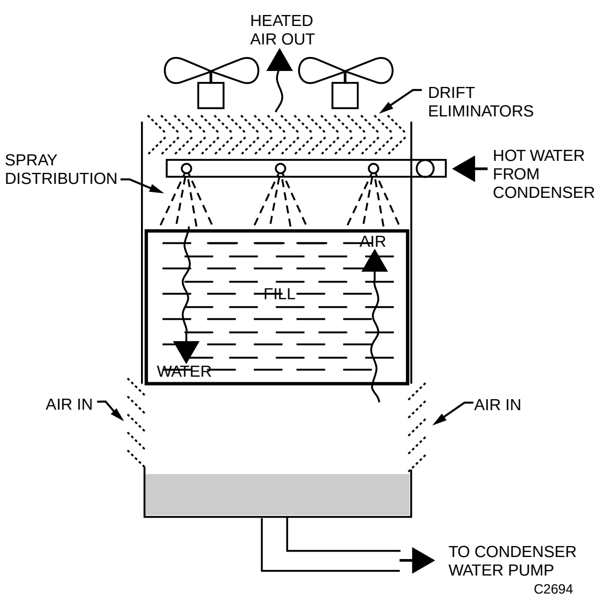


Fig. 25. Cooling Tower, Showing Water-Air Counterflow.

The range (inlet water temperature minus outlet water temperature) at design load of a cooling tower is determined by the amount of heat to be rejected and the water flow used to carry this heat to the tower. The cooling capability is then expressed as the design approach (approach specifies how close to the OA WB a cooling tower can cool water) at design range. Since most operation is at less than design load and/or design outdoor air temperatures, partial load operating characteristics have a strong influence on operating costs. Partial load operating characteristics are also used to establish the cooling capacity and capability for free cooling cycles at low outdoor air temperatures. Partial load characteristics for a tower at design flow rate are shown in Figure 26.

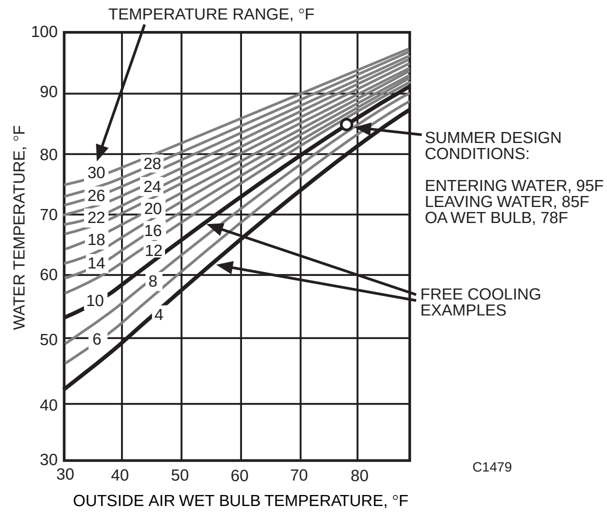


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Fig. 26. Typical Cooling Tower Partial Load Characteristics.

In Figure 26 summer design conditions are approximately 85F cold water leaving the tower, 95F water entering, and OA wet-bulb of 78F. This is 7F approach at 10F range. Notice that the point is plotted on the 10F range line. The same tower used for free cooling at an OA wet-bulb of 50F would provide 65F leaving water (15F approach) at full load (10F range) or 58F water (8F approach) at 40 percent load (4F range).

COOLING TOWER CAPACITY CONTROL

Fan control is the usual method of reducing tower capacity to maintain condenser water supply temperature in mild weather. A tower bypass valve is used to further limit cooling when the fans are off (Fig. 27). On-off fan control is very difficult at light loads because at light load the OA WB is usually well below design, which increases the tower capacity, producing short fan "on" periods. Controlling the air volume with dampers, blade pitch, or fan speed provides much closer and more stable control. A variable speed fan is more efficient than a two-speed fan when varying tower capacity.

Modulating tower water bypass for capacity control at low ambient temperatures can cause freeze-up in some tower designs.

Since, use of a tower bypass mixing valve in the tower leaving water can lower the pump suction pressure below the pumps minimum NPSH, a diverting valve in the tower entering water is generally used.

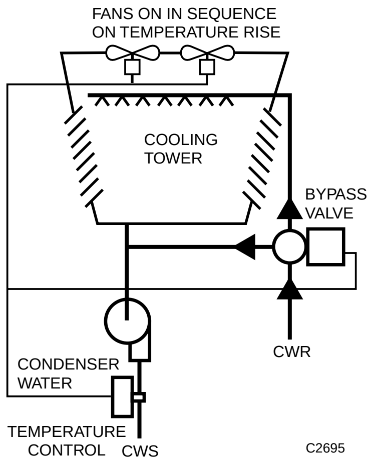

Fig. 27. Cooling Tower Control.

For minimum chiller energy usage, the condenser water temperature setpoint should be as low as can be safely used by the refrigeration system or as low as can be provided by outdoor air conditions. The approach value of the evaporative cooling tower indicates how close the cooled water temperature can come to outdoor air wet-bulb temperature at design conditions. When outdoor air is at a lower wet-bulb temperature than design, the tower can cool water to a lower temperature, but never to the wet-bulb temperature. Therefore, the controller setpoint should be at the lowest temperature attainable by the cooling tower to save chiller energy and not waste fan energy trying to reach an unobtainable value. Figure 28 is a reset schedule of condenser water setpoint as a function of outdoor air wet-bulb temperature with the minimum safe temperature indicated.

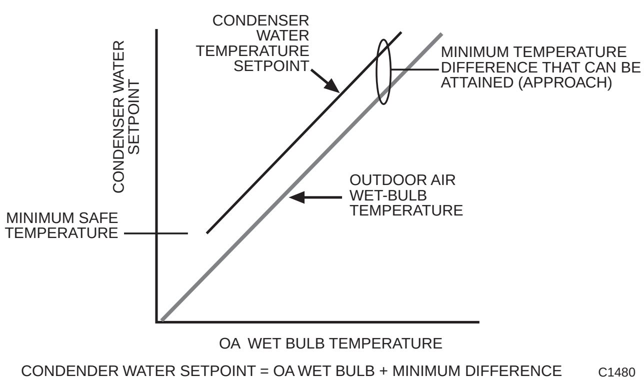

Fig. 28. Condenser Water Temperature Reset from Outdoor Wet-Bulb.

When wet-bulb reset is not used, a condenser water temperature setpoint around midpoint between the design (usually 85F) and the minimum value acceptable to the chiller is satisfactory.

On-Off Cooling Tower Fan Control

On-off fan control is satisfactory where the load is always high and where several towers are banked together for multistage control.

On-off cooling tower control with a single setpoint for a PI control function produces questionable performance. A preferred solution starts the fan on a rise in condenser water temperature to 80F, stops the fan on a drop to 73F, and limits on periods to a minimum of 5 minutes. The 5 minute minimum on time can still produce water too cool for the chiller during light loads and low OA WB temperatures. The use of a modulating valve can prevent condenser water temperature from dropping below the lowest that the chiller can tolerate (valve control would be PI with a single setpoint). During below freezing OA conditions throttling condenser water into the tower can cause icing, cycling the valve open and closed can reduce freezing. The specification following provides warmer ice-melting water during colder weather.

Specification:
On a rise in condenser water temperature to 80F the cooling tower fan shall start and on a drop in condenser water temperature to 73F the cooling tower fan shall stop. When started, the cooling tower fan shall run a minimum of five minutes. When the OA temperature is above freezing, the bypass valve shall modulate if required to prevent the condenser water from falling below 71F. When the OA temperature drops below 32F, the bypass valve shall open to the bypass on a drop in condenser water temperature to a minimum acceptable temperature and open to the tower upon a rise in temperature. The temperature rise shall vary between 5 and 12F as the OA temperature varies from 32F to 0F.

NOTE: The variable temperature rise allows warmer water to enter the tower during extreme weather to encourage melting and reduce icing.

Two-Speed Cooling Tower Fan Control

For two-speed cooling tower fans, a single setpoint for a PI control function also produces unpredictable performance. A better solution would be one of absolute setpoints similar to that noted for on-off fans. If wet-bulb reset is used, the control objective is to turn the fan off at the optimum setpoint but at no less than the minimum temperature acceptable to the chiller plus 1 degree, and cycle to and from low and high speeds above this value with minimum on times for each speed and with no more than 7 speed changes per hour for fans under 10 feet in diameter or as required by the tower manufacturer. The specification following is typical.

Specification:
Condenser water temperature setpoint shall be equal to the OA WB plus 7F, or the minimum temperature acceptable to the chiller, whichever is higher. Fan shall stop when condenser water temperature drops to setpoint. Fan shall start at low speed when the temperature rises to the setpoint plus 6F, and shall not stop for a minimum of 9 minutes. Fan shall transition from low to high speed anytime it has run at least 3 minutes at low speed and the temperature is above the low-speed start point plus 2F or 85F, whichever is lower. Fan shall run a minimum of 9 minutes on high speed, and shall transition from high speed to low speed if the temperature drops to a value equal to the switch-to-high-speed value minus 6F.

If required, use a valve control method similar to the one used for single speed fan control.

Variable Speed Cooling Tower Fan Control

For variable speed cooling tower fans, an EPID control function with WB optimized setpoint or fixed setpoint works well until the fan drops to a minimum safe speed, below which results in motor overheating. If the tower capacity is still excessive at minimum fan speed, the fan control mode reverts to on-off. During this mode, disable the PI function or when the fan is restarted it will start at a speed greater than the minimum, although the minimum speed was too great. This produces an undesirable short on time. The following is a possible application and specification:

SINGLE COOLING TOWER VARIABLE SPEED FAN CONTROL

Functional Description

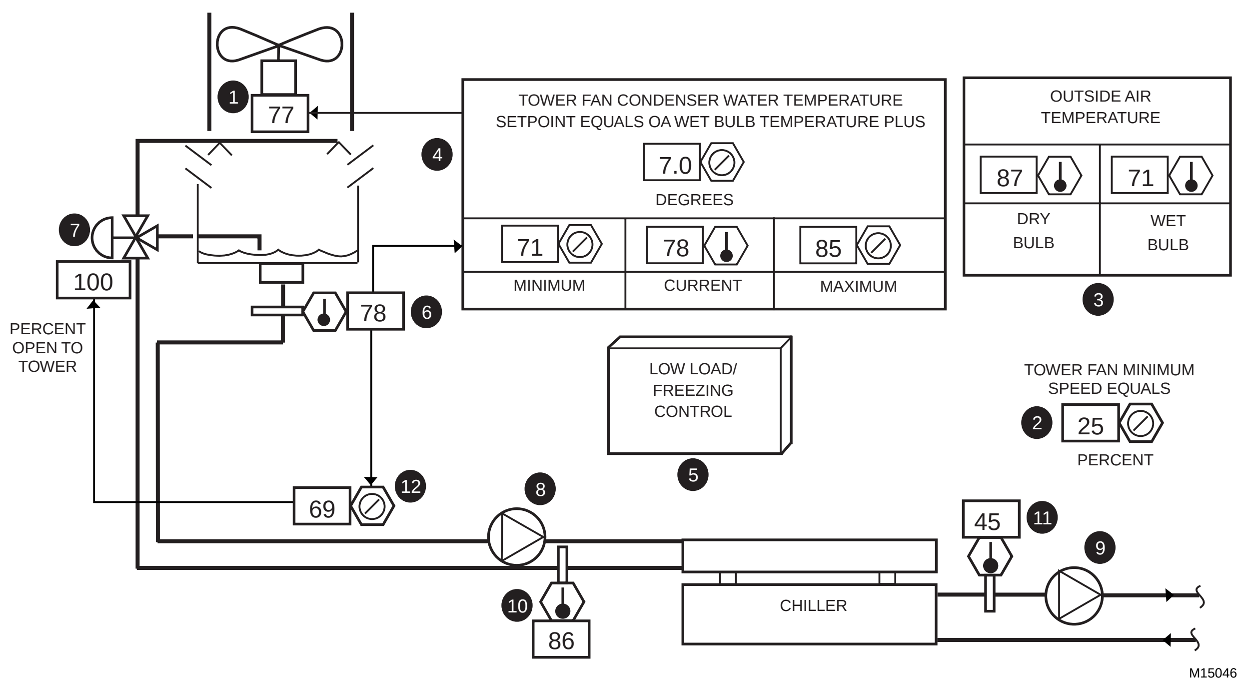

Fig. 29. Single Cooling Tower Variable Speed Fan Control Graphic

Item No.	Function
1	Indicates percent maximum tower fan speed.
2	Sets tower minimum speed.
3	Current OA conditions.
4	Sets minimum and maximum condenser water temperature setpoints and WB approach.
5	Icon to select tower low load control dynamic sequence display (Fig. 30).
6	Condenser water control sensor.
7	Displays valve position.
8,9	Dynamic pump symbols denote pump operation.
10,11	Operator information.
12	Sets valve control setpoint.

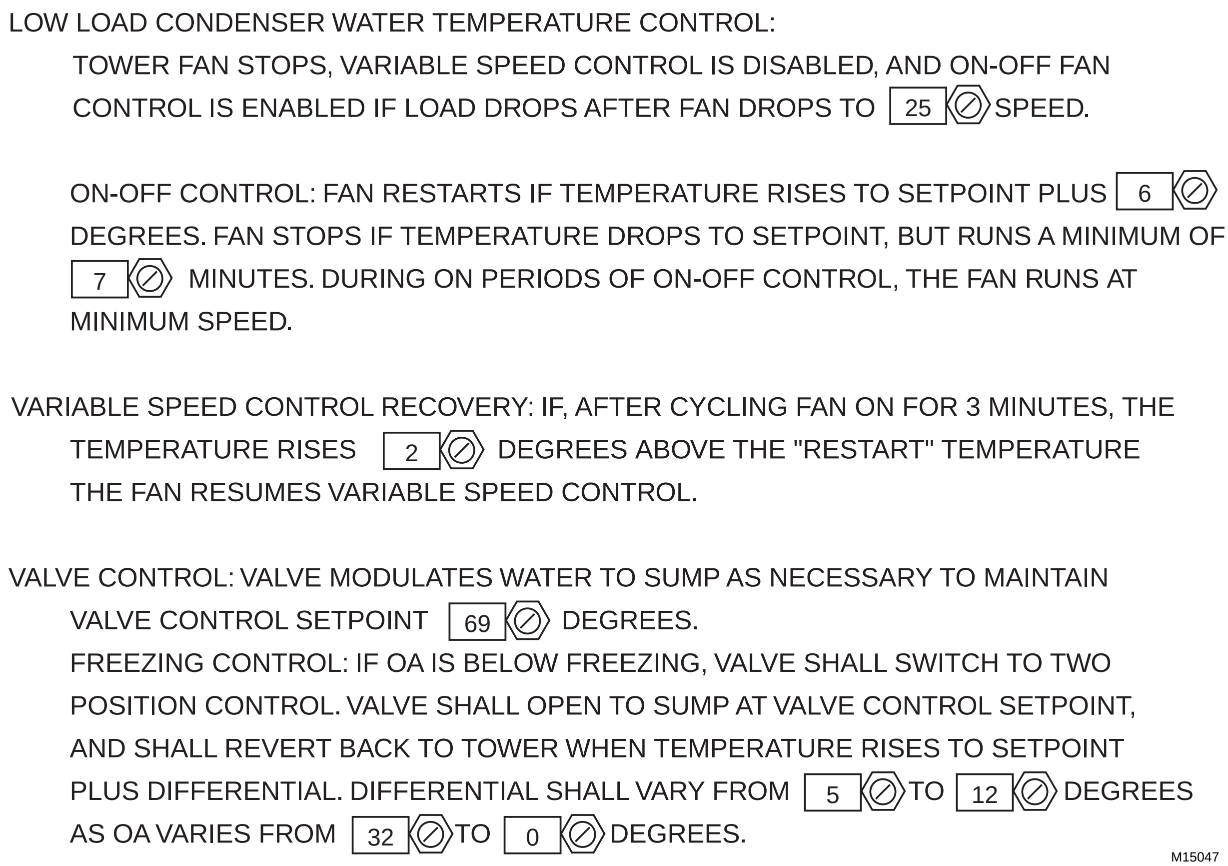

Fig. 30. Cooling Tower Load Control Dynamic Display.

Features

1. Precise PI condenser water control above minimum load.
2. User friendly monitoring and adjustment.
3. Optimized condenser water temperature.
4. Freeze prevention cycle.
5. Fan motor light-load overheating protection.

Conditions For Successful Operation

1. Control network, software, and programming to advise tower controller of chiller demands.
2. Interlock and control wiring coordinated with the chiller manufacturer.
3. Good OA RH sensor and WB calculation.
4. Proper project-specific setpoint and parameter settings.
5. If tower must operate with OA temperature below freezing, the diverting valve must be controlled two position, during freezing conditions.

Specification

Cooling tower controls shall be enabled anytime the condenser water pump runs. The cooling tower fan speed shall be modulated under EPID control down to a minimum 25% speed to maintain a condenser water temperature setpoint equal to the OA WB temperature plus 7F, or a value equal to the minimum temperature acceptable to the chiller plus 2F, whichever is higher, but not to exceed 85F. If the tower capacity is still excessive at minimum speed, control shall revert to an on-off algorithm, cycling the fan off at the minimum speed at a setpoint equal to the above setpoint, and back on at the minimum speed at a temperature equal to the off temperature plus 6F, with on cycles of no less than 7 minutes duration. If, after cycling the fan on for 3 minutes, the temperature rises above the “back on” setpoint by 2F, the EPID function shall be reinstated with a start value of 25% and a ramp duration of 90 seconds.

The tower diverting valve shall modulate to divert tower water to the sump if required to prevent the condenser water temperature from dropping below the minimum temperature acceptable to the chiller. When the OA temperature drops below freezing, the valve shall switch to two-position control, wherein the valve shall open to the sump when the water temperature drops to the setpoint and close when the temperature rises a differential which shall vary from 5 to 12F degrees as the OA temperature varies from 32 to 0F.

DUAL COOLING TOWER VARIABLE SPEED FAN CONTROL

Functional Description

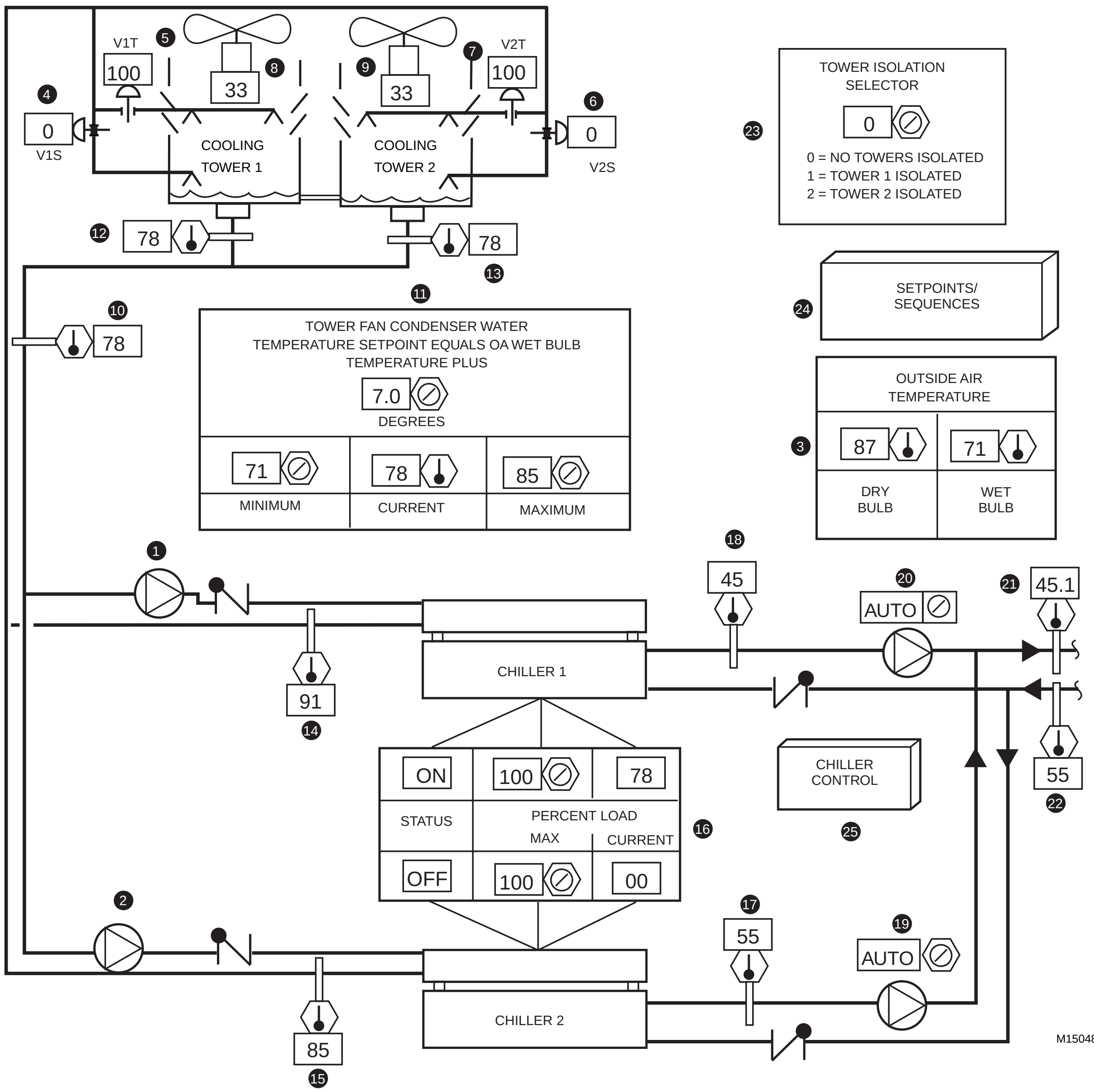

Fig. 31. Dual Cooling Tower Variable Speed Fan Control Graphic

Item No.	Function
1,2	Dual dynamic pump symbols indicate pump operational status.
3	OA Conditions dictate freeze-protection and optimized setpoint control strategies.
4-7	Valves provide tower isolation, modulating low limit condenser water control when OA temperature is above freezing, and two position low limit operation when OA temperature is below freezing.
8,9	Indicates tower fans speed (percent).
10	Input for common condenser water temperature control.
11	Optimum condenser water temperature calculation based upon OA WB.
12-22	Operator information.
23	Isolates towers if required.
24	Icon, for dynamic control and setpoint display.
25	Icon to select tower low load control dynamic sequence display.

Features

1. Precise PI condenser water control above minimum load.
2. User friendly monitoring and adjustment.
3. Optimized condenser water temperature.
4. Freeze prevention cycle.
5. Fan motor light-load overheating protection.

Conditions For Successful Operation

1. Control network, software, and programming to advise tower controller of chiller demands.
2. Interlock and control wiring coordinated with the chiller manufacturer.
3. Good OA RH sensor and WB calculation.
4. Proper project specific setpoint and parameter settings
5. Electrical heaters and insulation on tower piping where water cannot drain when the tower is isolated or when the pump stops, if required.
6. Towers operate satisfactorily down to 50% water flow.
7. The common tower water temperature sensor has a high responsiveness and is in a well with heat conductive compound to rapidly detect temperature variations.
8. Chiller or free cooling HX tolerates variations in condenser water temperature when OA temperature is below freezing and fans and valves are cycling under temperature differential control and valves are not modulated because of freezing danger.
9. Common piping from chillers to cooling towers.

Specification

The dual variable speed fan cooling tower control configuration requires consideration of four control strategies:

- Single pump operation, above freezing OA.
- Single pump operation, below freezing OA.
- Dual pump operation, above freezing OA.
- Dual pump operation, below freezing OA.

NOTES:

1. If the towers are not required to operate when OA is suitable for air-side economizer cooling (usually below 57 or 58F), leave the fans running at minimum speed (rather than switching to an on-off algorithm) and modulate the sump valves when necessary to prevent cooling to a temperature unacceptable to the chiller.
2. With both towers activated, the differentials used for cycling fans and valves are half of the differentials used when only one pump and tower are activated because only half the water is affected by the change.

Tower valves:

Each tower supply piping shall be provided with two modulating butterfly control valves; one normally open valve in the supply to the tower header, and one normally closed valve in a line to the tower sump. The valves shall operate inversely during temperature control modes. When both pumps are off, both valves shall open (to drain as much water as possible to tower sump), and both shall close during respective periods of tower isolation.

Condenser water set point calculation:

During chiller operation, the common condenser water temperature setpoint shall be reset to a value equal to the OA WB plus 7F, but to no lower than the chiller can tolerate and no higher than 85F. During tower free cooling mode, the setpoint shall be determined by the tower free cooling control module. The OA WB shall be calculated from OA RH (the RH sensor shall be of no less than 2% accuracy) and OA DB. The fans shall operate at the setpoint but no cooler than the minimum temperature the chiller can tolerate plus 2F. The valves shall operate to maintain a fixed minimum temperature setpoint of the minimum value the chiller can tolerate or the tower free cooling setpoint in the tower free cooling mode of operation.

Single pump operation with above freezing OA temperature:

For both towers; valves shall position open to the tower and closed to the sump and the tower fans shall modulate in unison under EPID control down to a minimum fan speed of 25% to maintain common condenser water temperature. If the condenser water load drops below the capacity of both fans operating at minimum speed, the fan that has operated longest shall stop and both its valves shall close, and the remaining tower fan shall assume the full load. If the condenser water load drops below the capacity of the remaining fan operating at minimum speed, the fan shall stop and revert to an on-off algorithm. In the on-off mode the fan shall stop at the fan control setpoint noted, the tower fan that has been off the longest shall restart after a rise in temperature of 6F, but to no higher than 85F and shall run no less than 7 minutes. The operating tower valves shall modulate to maintain the common condenser water setpoint at anytime regardless of fan operation. If, after running greater than 4 minutes in the on-off mode of operation, the water temperature rises to a value equal to the “fan on” setpoint plus 2F, the fan shall revert back to EPID control with a start value of 25% and a ramp duration of 120 seconds. If only one fan is running in the EPID control mode and its speed rises to 80 percent, the off tower valves shall position open to the tower and the off tower fan shall start at 25 percent speed and ramp into unison control over a 240 second duration.

Single pump operation with below freezing OA temperature:

This mode of fan operation shall be the same as for above freezing, except that the valve control algorithm must be two position (because modulating the tower water at below freezing OA temperatures can cause ice formation). Thus the valves shall position to full flow to the tower sump anytime the water temperature drops to the valve control setpoint, and shall position back to full flow to the tower after the water temperature has risen a value that resets from 6 to 12F as the OA temperature varies from 32F to 0F. The tower fan shall not be allowed to start until the valves have been positioned open to the tower greater than 90 seconds and the water temperature is rising.

Dual pump operation with above freezing OA temperature:

For both towers; valves shall position closed to the sump and the tower fans shall modulate in unison, to maintain a common condenser water temperature setpoint, under EPID control down to a minimum speed of 25%. If the condenser water load drops below the capacity of both fans operating at minimum speed, the fan that has operated longest shall stop, and the remaining tower fan shall assume the full load. If the condenser water load drops below the capacity of the remaining fan operating at minimum speed, the fan shall stop and revert to an on-off algorithm alternating the fans operating at the minimum speed in the “on” mode. In the on-off mode the fan shall stop at the fan control setpoint noted above, and shall restart after a rise in temperature of 4F, but to no higher than 85F, and shall run no less than 7 minutes. The operating tower valves shall modulate to maintain the above noted valve control setpoint at anytime regardless of fan operation. If, after a fan is running greater than 4 minutes in the on-off mode of operation, the water temperature rises to a value equal to the "fan on" setpoint plus 2F, the fan shall revert back to EPID control with a start value of 25% and a ramp duration of 120 seconds. If the single fan operates to a speed exceeding 60%, the "off" fan shall start at an initial speed of 25% and shall ramp into unison control over a 240 second duration (during which time the operating fan will slow down as the starting fan slowly assumes part of the load).

Dual pump operation with below freezing OA temperature:

For both towers; valves shall position closed to the sump and the tower fans shall modulate in unison, to maintain common condenser water temperature, under EPID control down to a minimum fan speed of 25%. If the condenser water load drops below the capacity of both fans operating at minimum speed, the fan that has operated longest shall stop, and the remaining tower fan shall assume the full load. If the condenser water load drops below the capacity of the remaining fan operating at minimum speed, the fan shall stop and revert to an on-off algorithm alternating the two tower fans operating at minimum speed in the "on" mode. In the on-off mode the fan shall stop at the fan control setpoint noted above, and shall restart after a rise in temperature of 4F, but to no higher than 85F, and shall run no less than 7 minutes. If either fan is on and the water temperature drops to the valve control setpoint plus 1 degree, the valves of the tower in which the fan is on shall position open to the sump until the fan stops, at that time the valves shall position back to the tower.

If both fans are off and the water temperature drops to the valve control setpoint plus 1 degree F, the valves of Tower 1 shall position open to the sump, and the Tower 2 valves shall alternately cycle open to the sump with a 3 degree temperature differential. If, while one valve is open to the sump, the water temperature drops to the valve control setpoint, the other valve shall position open to the sump, and the valves shall alternate open to the tower with a 3 degree differential. If, after one valve is open to the tower for 2 minutes, and the water temperature rises above the valve control setpoint plus 4F, the remaining valve shall position open to the tower and the valves shall revert to alternately opening to the sump at the valve control setpoint plus 1 degree, and opening to the tower after a 3 degree differential. If, after both valves position open to the tower for 2 minutes, the temperature rises to the valve control setpoint plus 4F, control shall revert back to alternately cycling the fans on at minimum speed as before. If, after a fan is running greater than 2 minutes at minimum speed in the on-off mode of operation, the water temperature rises to a value equal to the "fan on" setpoint plus 2F, the fan shall revert back to EPID control with a start value of 25% and a ramp duration of 120 seconds. If the single fan operates to a speed exceeding 60% for greater than 3 minutes, the "off" fan shall start at an initial speed of 25% and shall ramp into unison control over a 120 second.

NOTE: This "Dual pump below freezing OA temperature" sequence gets quite complex as the condenser water load increases and decreases. Table 1 may help understand the shifting sequence.

Table 1. Temperature Control Sequence for Dual Pumps Running in Below Freezing OA.

Load:	Load Sequence	
	Decreasing	**Increasing**
Between both fans running at minimum speed and full load.	Seq. 1: Modulates both fans in unison to control at Setpoint + 2F.	—
Less than both fans at minimum speed, greater than one fan at minimum speed.	Seq. 2: Fan running longest stops. Remaining fan modulates to maintain Setpoint +2F.	Seq. 10: If one fan runs greater than 60% speed greater than 2 minutes: Remaining fan ramps on in unison from 25% speed over 240 seconds. Revert to Seq. 1.
Less than one fan at minimum speed, greater than both fans Off.	Seq. 3: Fans alternately cycled: At Setpoint + 5F first fan On at minimum speed. At Setpoint + 2F first fan Off. Fans run 6 minutes minimum. Repeat cycle for second fan.	Seq. 9: If either fan is on at minimum speed greater than 2 minutes & water temperature is greater than Setpoint + 4F: Revert to Seq. 2 via EPID with start value of 25% and ramp duration of 120 seconds.
One fan running at minimum speed greater than 2 minutes and water temperature less than Setpoint + 1 degree.	Seq. 4: Positions valve set of the operating tower to sump until fan stops. Repositions valve set back to tower.	—
Both fans Off greater than 2 minutes & water temperature less than Setpoint + 1 degree.	Seq. 5: Alternately positions valve sets: At Setpoint + 1 degree to sump. At Setpoint +3F back to tower (one set of valves always open to tower). Fans locked Off.	Seq. 8: If both valve sets position open to the tower greater than 2 minutes & water temperature is greater than Setpoint + 4F: Revert to Seq. 3.
One set of valves open to tower (Seq. 5) greater than 2 minutes, & water temperature less than Setpoint.	Seq. 6: At setpoint positions valve sets of both towers to sump. At Setpoint + 3F alternately positions valve sets to tower (one set of valves always closed to sump).	Seq. 7: If one valve set positions open to the tower greater than 2 minutes & water temperature greater than Setpoint + 4F: Revert to Seq. 5.

CHILLER HEAT RECOVERY SYSTEM

A chiller heat recovery system uses heat rejected from a chiller to satisfy a simultaneous heating load. To accomplish this the heating water design temperature becomes the condenser water design temperature for the chiller. A typical simultaneous cooling and heating load includes an interior zone cooling load and a perimeter zone heating load. When the cooling load is greater than the heating load, the excess heat is dissipated in a cooling tower. The control strategy is to use as much heat as needed from the condenser and to reject excess heat to the cooling tower when necessary.

Figure 32 shows a heat recovery system and control sequence for a cooling system with one chiller and a double bundle condenser where chiller capacity is controlled from the chilled water supply temperature. The double bundle condenser uses heat recovery water or cooling tower water or both to cool refrigerant. Cooling tower water cannot be mixed with heat recovery water so the two systems must be isolated. In a single chiller system, the control system checks the water temperature coming out of the heat recovery bundles (T3). If it is warmer than the water returning from the heating loads (T2), valve V1 opens to circulate condenser water through the system. A hot water generator (boiler or converter) provides additional heat as required. If the heat recovery system is not cooling the condenser sufficiently, the cooling tower is used to dissipate excess heat. In multiple chiller systems, the heat recovery chiller is controlled by the heating load (T1). The other chillers provide additional cooling as required.

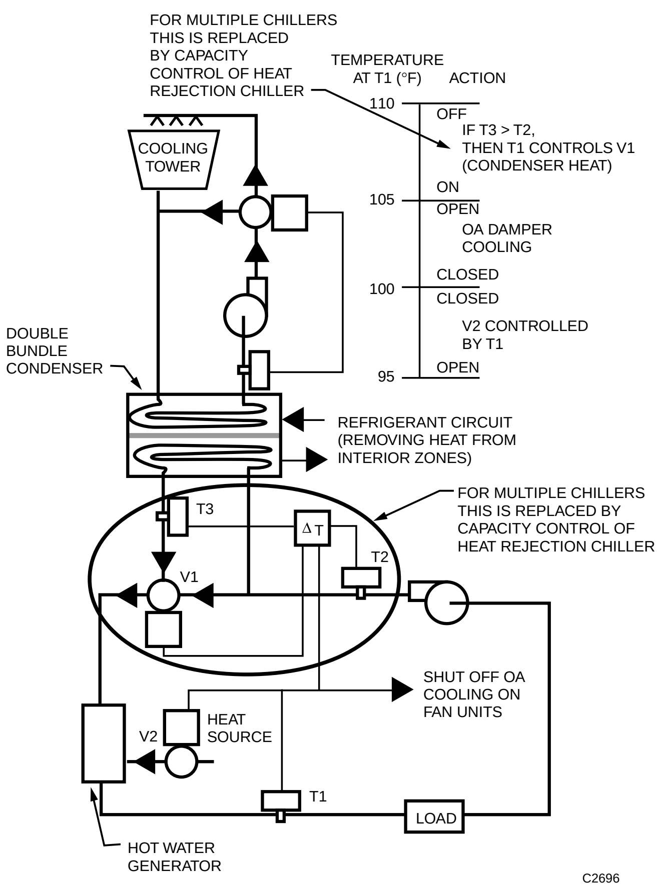

Fig. 32. Heat Pump Cycle Controls and Sequence.

If chilled water reset is used, stop the reset action when the cooling tower is off. This provides recovery system heat to a lower outdoor temperature before it is necessary to use fuel for heating.

FREE COOLING-TOWER COOLING

When the condenser water temperature from an evaporative cooling tower is equal to or lower than chilled water load requirements, mechanical refrigeration is not needed. During these times “free cooling” from the evaporative cooling tower is available without having to supply energy to the compressor. Figures 33, 34, and 35 show three methods of providing tower cooling to the chilled water circuit.

In refrigerant vapor migration (Fig. 33) two refrigerant paths with migration valves are added between the condenser and the evaporator. The paths bypass the compressor and allow gravity flow of liquid refrigerant from condenser to evaporator and vapor flow from evaporator to condenser. Refrigerant vaporizes in the evaporator and migrates to the condenser where it is condensed by cooling tower water. The liquid then flows back to the evaporator by gravity. The chiller must be designed so the flow paths are unrestricted and the evaporator is lower than the condenser. The bypass valves are normally included as a chiller package option by the manufacturer.

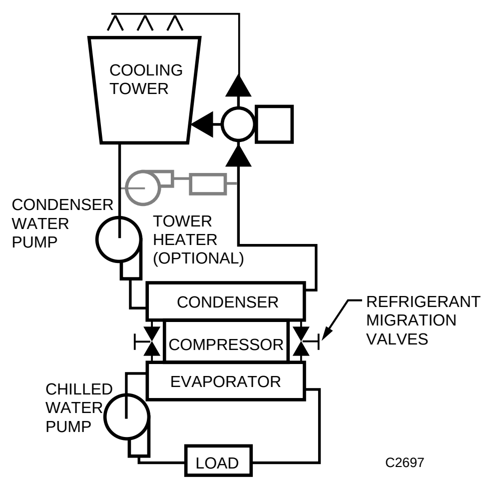

Fig. 33. Free Cooling Using Refrigerant Vapor Migration.

In Figure 34, condenser and chilled water flows are diverted from the chiller to a heat exchanger.

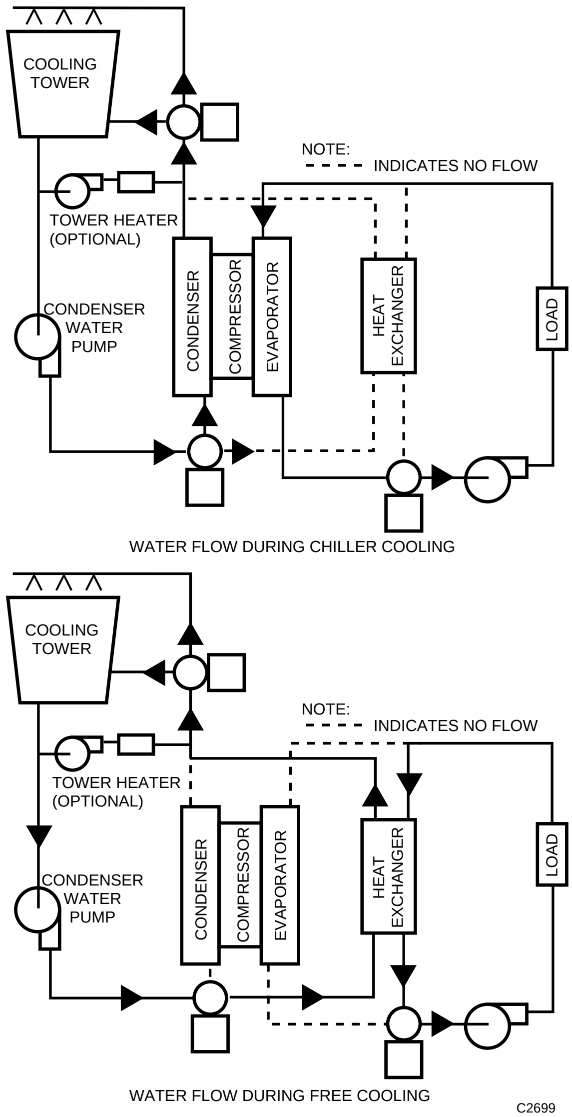

Fig. 34. Free Cooling Using Auxiliary Heat Exchanger.

In Figure 35, condenser and chilled water circuits are interconnected bypassing the chiller. In this configuration providing tower water treatment and filtering and pumping capacity for proper cooling tower flow are major considerations.

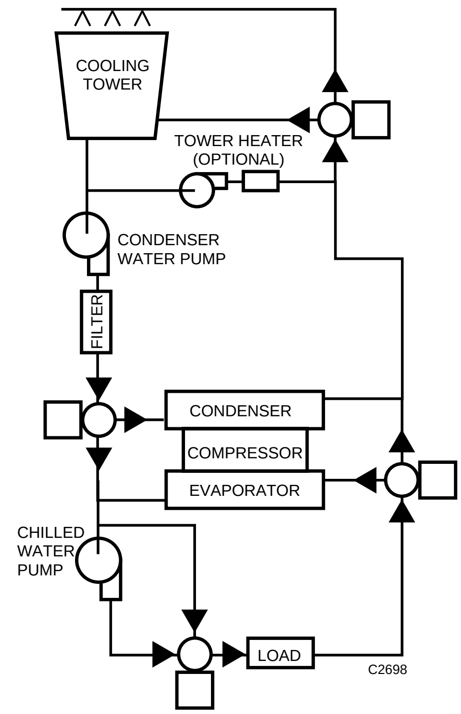

Fig. 35. Free Cooling by Interconnecting Water Circuits.

When the chiller is shut down during free cooling operation, the normal chilled water supply temperature control is bypassed and the chilled water supply temperature is determined by the tower water temperature. The normal cooling tower water control cycles tower fans and modulates the tower bypass valve to maintain a setpoint in the 65 to 85F range but cooling tower water is controlled at 35 to 45F when used directly for cooling.

During free cooling the fans may be manually turned on if there is no danger of supply water being too cold or too much energy being used by a winterized cooling tower. However, if either condition exists, the tower water temperature control should be in operation at all times to maintain a free cooling temperature of 35 to 45F.

The changeover from free cooling to chiller operation may be manual or automatic. Changeover sequence should change the tower water controller setpoint first, then change the heat exchanger valves when temperatures are proper for the new mode of operation.

Following is a tower-free-cooling example using plate-frame heat exchangers for the dual chiller system shown in the DUAL CENTRIFUGAL CHILLER example. The DUAL COOLING TOWER VARIABLE SPEED FAN CONTROL example is also relevant.

TOWER FREE COOLING, DUAL CHILLERS

FUNCTIONAL DESCRIPTION

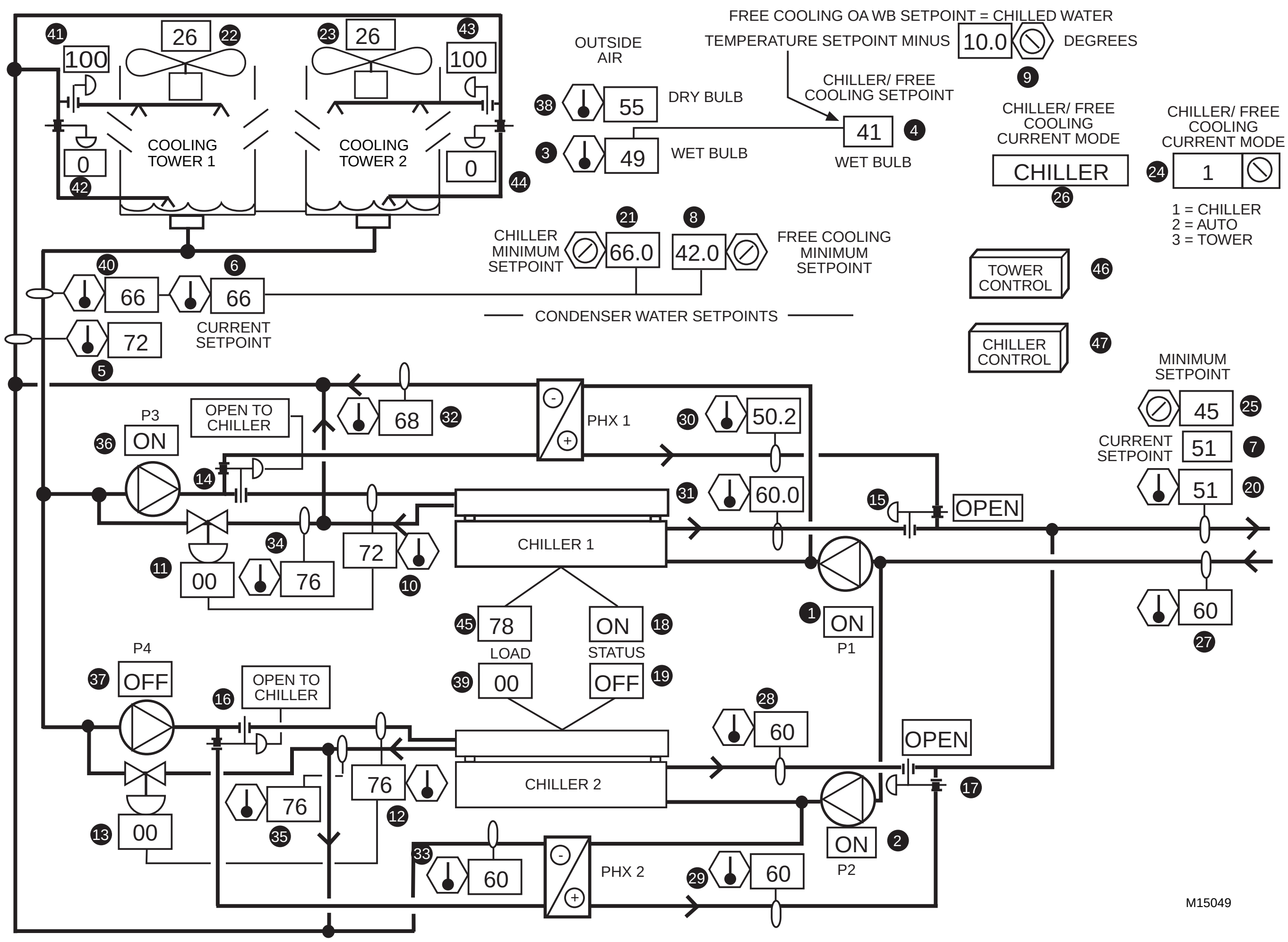

Fig. 36. Graphic for Tower Free Cooling System with Dual Chillers.

Item No.	Function
1,2	Chilled water pumps produce flow through chillers or plate/frame heat exchangers.
3,4	Free cooling mode initiated when OA WB drops to setpoint.
5	Tower water entering temperature, operator information.
6,7	Setpoints indicate need for cooling tower and chilled water.
8	Minimum condenser water temperature setpoint provides free cooling low limit.
9	In free cooling mode, tower will produce water about 7F above OA wet-bulb temperature, and HX will cool chilled water to within 2 to 3F of condenser water temperature, so the sum of these dictates when to activate free cooling.
10-13	Maintains condenser water temperature entering chiller during period when tower is pulling water temperature down to free cooling setpoint value and the temperature is too low for chiller.
14-17	Valves direct water from chiller/condenser to HX when condenser water reaches setpoint.
18,19	Chiller Status.
20	Operator information.
21	Prevents tower from cooling water below minimum value that chiller can tolerate (during chiller cooling mode).
22,23	Tower fans modulate to maintain condenser water setpoint.
24	CHILLER-AUTO-TOWER software mode selection function.
25	Prevents chilled water temperature setpoint from dropping below the design minimum.
26-45	Operator information (see chiller control and cooling tower control examples).
46	Icon, to display cooling tower control data.
47	Icon, to display chiller control data.

Features

1. Production of chilled water via condenser water and a HX in lieu of chiller operation.
2. User friendly monitoring and adjustment.
3. Automatic selection of free cooling mode.
4. Maintains full flow and minimum temperature condenser water during free cooling pull-down period.

Conditions For Successful Operation

1. Control network, software, and programming to advise chilled water plant of AHU chilled water demands.
2. Interlock and control wiring coordinated with the chiller manufacturer.
3. Appropriate cooling tower and chiller control.
4. Accurate calculation of OA WB from OA DB and OA RH.
5. Pumps selected for HX pressure drops and chiller/ condenser water flow balancing.
6. Flow switches and starter auxiliary contacts provided to prove flow in lieu of current sensing relays for the chiller and condenser pumps.

Specification

NOTE: This control scheme assumes that the chilled water plant must continue to operate while the condenser water temperature is dropped from the minimum temperature acceptable to the chiller to the temperature required to produce chilled water. It is also assumed that full condenser water flow is required at all times (otherwise, chiller head pressure could be maintained by modulating the tower-side changeover valves to reduce condenser flow).

Chilled water pump:

Anytime at least one chilled water pump is operating and the OA WB drops below the free cooling setpoint for greater than ten minutes, the condenser water temperature setpoint shall be lowered to a value equal to the chilled water setpoint minus 3F, but no lower than 42F. The OA WB free cooling changeover setpoint shall be the chilled water temperature setpoint minus 10F.

Condenser water:

During the condenser water temperature pull-down period, the condenser entering water temperature shall be prevented from dropping below the minimum acceptable value by modulating open a valve between the condenser leaving water and the pump inlet.

When the condenser water temperature reaches the free cooling setpoint, the changeover valves for the chiller that has operated longest shall position open to the HX and the chiller shall stop. If the free cooling condenser water temperature setpoint is not reached within eight minutes, the condenser water setpoint shall be returned to the chiller cooling value and a descriptive system alarm shall be issued.

Five minutes after the first chiller changes over to free cooling, the second chiller shall changeover as noted for the first chiller.

In the free cooling mode, the tower controls shall operate to maintain the condenser water temperature setpoint, which shall be reset from 42F to 56F by the chilled water temperature setpoint demand for cooling.

Anytime the system is running in the free cooling mode and the chilled water temperature is greater than 1 degree above setpoint for greater than 5 minutes, the changeover valves for the chiller that has been off longest shall position open to the chiller, the condenser water temperature setpoint shall be raised to the value specified for chiller operation, and the chiller controls shall be enabled. If both chilled water pumps are operating, the remaining chiller shall changeover and start 2 minutes later. As the chillers start, their respective low-limit condenser water temperature valves shall be placed under EPID control with a start value of 75% open and a ramp duration of 60 seconds.

After reverting back to the chiller cooling mode, during the condenser water warm-up period the cooling tower fans shall remain on at no less than minimum speed, to prevent short cycling, and the condensers entering water temperature shall be controlled by the low-limit condenser water temperature pump inlet valves.

A software CHILLER-AUTO-TOWER selection function shall be provided to lock the system in either mode if desired.

BOILER SYSTEM CONTROL

INTRODUCTION

A boiler is a closed vessel intended to heat water and produce hot water or steam through combustion of a fuel or through the action of electrodes or electric resistance elements. Steam and hot water boilers are available in standard sizes from very small boilers for apartments and residences to very large boilers for commercial and industrial uses.

BOILER TYPES

Boilers are classified by water temperature or steam pressure. They are further classified by type of metal used in construction (cast iron, steel, or copper), by type of fuel (oil, gas, or electricity), or by relationship of fire or water to the tubes (firetube or watertube).

– Low-pressure boilers are those designed to produce steam up to 15 psig or hot water up to 250F with pressures up to 160 psig.
– Medium and high pressure boilers produce steam above 15 psig or hot water above 160 psig or 250F or both.

CAST IRON AND STEEL BOILERS

Boilers are typically constructed of cast iron or welded steel. Cast iron boilers (Fig. 37) are made of individually cast sections and are joined using screws or nuts and tie rods or screwed nipples to join sections. The number of sections can be varied to provide a range of capacities.

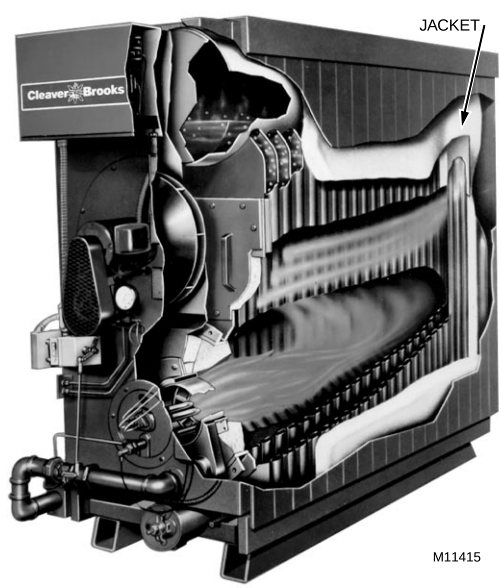

Photo Courtesy of Cleaver Brooks

Fig. 37. Typical Cast Iron Boiler (Watertube).

Steel boilers come in a large variety of configurations. They are factory assembled and welded and shipped as a unit. Figure 38 illustrates a firetube boiler. The fire and flue gases are substantially surrounded by water. The products of combustion pass through tubes to the back then to the front and once more to the back before finally exiting at the front. This makes it a four-pass boiler. Firetube boilers are manufactured in many other configurations such as:

– External firebox, firebox not surrounded by water.
– Dry back, firetubes directly available from clean-out doors at back of boiler.
– Scotch-Marine, low water volume and fast response.

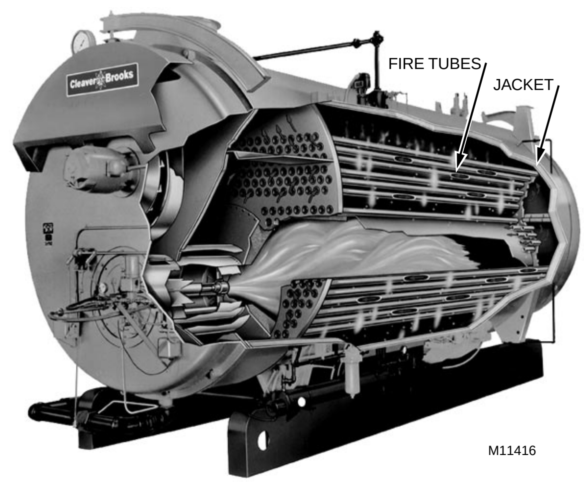

Photo Courtesy of Cleaver Brooks

Fig. 38. Typical Firetube Boiler.

Watertube boilers are steel body boilers used for high capacity requirements of more than two million Btuh. Watertube boilers use a water-cooled firebox which prolongs the life of furnace walls and refractories.

MODULAR BOILERS

Modular boilers are small hot water boilers rated at 200,000 Btuh to 900,000 Btuh input. These boilers are available with 85 percent or higher gross efficiency. Figure 39 shows features of a typical modular boiler. These boilers are often used in tandem to provide hot water for space heating and/or domestic hot water. For example, if the design heating load were 2,000,000 Btuh, four 600,000 Btuh (input) modular boilers might be used. If the load were 25 percent or less on a particular day, only one boiler would fire and cycle on and off to supply the load. The other three boilers would remain off with no water flow. This reduces flue and jacket (covering of the boiler) heat losses.

Some modular boilers have very small storage capacity and very rapid heat transfer so water flow must be proven before the burner is started.

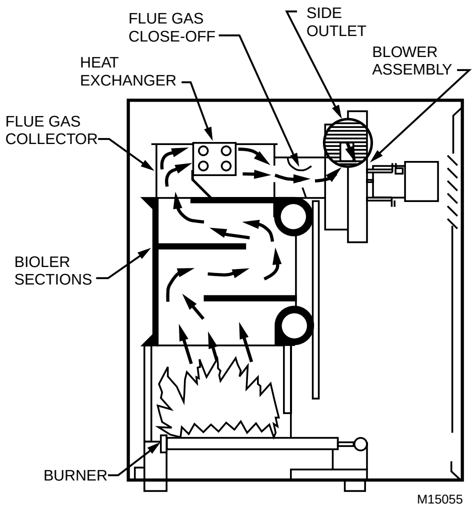

Fig. 39. High Efficiency Modular Boiler.

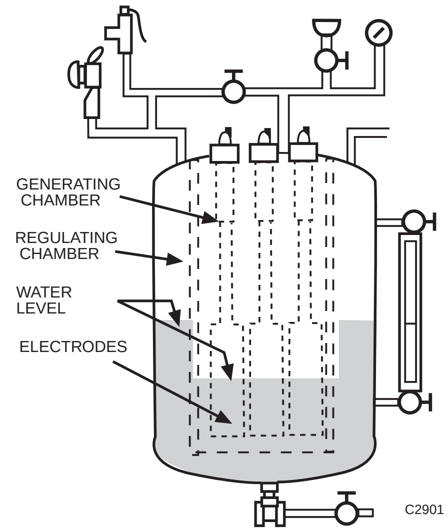

Fig. 40. Electrode Steam boiler.

ELECTRIC BOILERS

Electric boilers heat water or produce steam by converting electrical energy to heat using either resistance elements or electrodes. Electric boilers are considered to be 100 percent efficient since all power consumed, directly produces hot water or steam. Heat losses through the jacket and insulation are negligible and there is no flue.

Electrode boilers (Fig. 40) have electrodes immersed in the water. Current passes through the water between electrodes and the resistance of the water generates heat. Electrode boilers are available in sizes up to 11,000 kW. Resistance boilers have the resistance (heating) elements immersed in but electrically insulated from the water and are manufactured in sizes up to 3000 kW.

Electric elements and electrodes are grouped to provide four or more stages of heating. A step controller responding to steam pressure or hot water temperature activates each stage of heating as required to heat the building.

BOILER RATINGS AND EFFICIENCY

Boilers can be rated in several ways. Figure 41 shows commonly used ratings and terms. The terms Btuh (Btu per hour) and MBtuh or MBH (thousand Btuh) indicate boiler input rate. Input ratings are usually shown on the boiler (or burner) nameplate. The terms hp (boiler horse power), EDR (equivalent direct radiation), and pounds per hour (steam) indicate boiler output rate.

Gross efficiency is output (steam or water heat content and volume) divided by fuel input (measured by a fuel meter at steady-state firing conditions). The efficiency as indicated by flue gas conditions does not take into account jacket and piping losses so is usually higher than the gross efficiency.

A testing procedure issued by the US Department of Energy in 1978 measures both on cycle and off cycle losses based on a laboratory procedure involving cyclic conditions. This result is called the AFUE (Annual Fuel Utilization Efficiency) rating or seasonal efficiency, which is lower than gross efficiency.

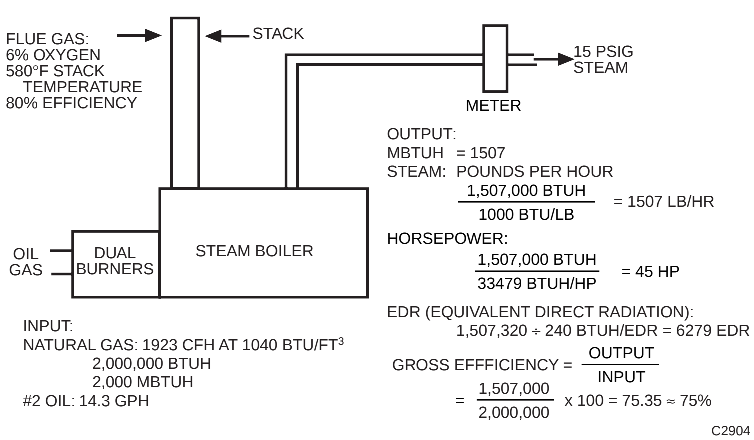

Fig. 41. Boiler Ratings and Efficiency.

COMBUSTION IN BOILERS

PRINCIPLES OF COMBUSTION

When gas, oil, or other fuels are burned, several factors must be considered if the burning process is to be safe, efficient, and not harmful to the environment. The burning process must:

1. Provide enough air so that combustion is complete and undesirable amounts of carbon monoxide or other pollutants are not generated.
2. Avoid excess air in the fuel-air mixture which would result in low efficiency.
3. Provide complete mixing of air with fuel before introducing the mixture into the firebox.
4. Provide safety controls so that fuel is not introduced without the presence of an ignition flame or spark and that flame is not introduced in the presence of unburned fuel.
5. Avoid water temperatures below the dewpoint of the flue gas to prevent condensation on the fireside of the boiler.

FLUE GAS ANALYSIS

Combustion can be monitored by flue gas analysis. For large boilers, over 1,000,000 Btuh, the analysis is typically continuous. For small boilers, flue gas is analyzed periodically using portable instruments.

Flue gas composition analysis routinely measures the percent of CO_2 (carbon dioxide) or O_2 (oxygen), but not both. Ideal CO_2 is in the 10 to 12 percent range. Percent oxygen is the most reliable indication of complete combustion. The ideal O_2 concentration is in the three to five percent range. Lower concentrations are impractical and often unsafe. Higher O_2 concentrations mean that an excessive quantity of air is admitted to the combustion chamber and must be heated by the fuel. This excess air passes through the boiler too quickly for the heat to be efficiently transferred to the water or steam. Carbon dioxide measuring instruments are simpler or lower cost than O_2 measuring instruments.

The CO_2 or O_2 concentration plus stack temperature provide a burner efficiency in percent either directly or by means of charts. This efficiency indicates only the amount of heat extracted from the fuel. It does not account for excess heating of combustion air or losses from leaks or the boiler jacket.

OIL BURNERS

Oil burners are usually of the atomizing variety, that is, they provide a fine spray of oil. Several types exist:
– Gun type burners spray oil into a swirling air supply.
– Horizontal rotary burners use a spinning cup to whirl oil and air into the furnace.
– Steam or air atomizing burners use high pressure air or 25 psig steam to break up the oil into fine droplets.

For modulating or high/low flame control applications the rotary or steam/air atomizing burners are most common.

GAS BURNERS

Two typical types of gas burners are the atmospheric injection burner and the power type burner. The atmospheric injection burner uses a jet of gas to aspirate combustion air and is commonly used in home gas furnaces and boilers. The raw-gas ring burner (Fig. 42) is an atmospheric injection burner.

Power burners (Fig. 43) use a forced-draft fan to thoroughly mix air and gas as they enter the furnace. They are common in commercial and industrial applications.

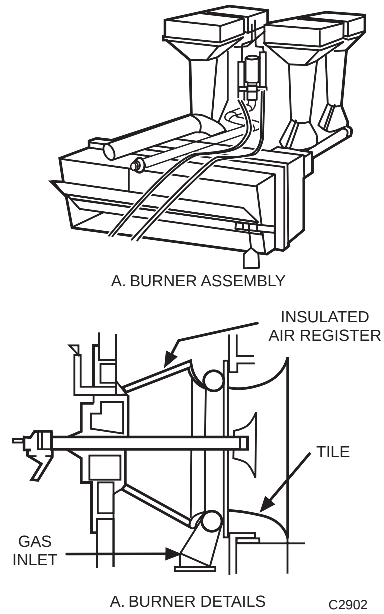

Fig. 42. Raw Gas Ring Burner.

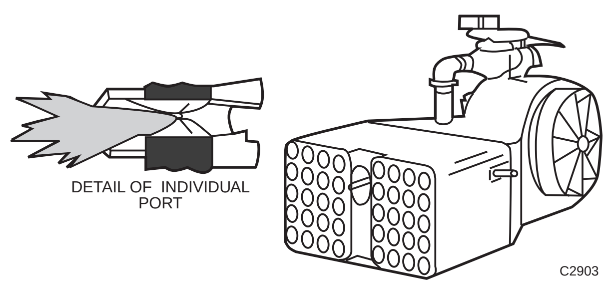

Fig. 43. Multiport Forced-Draft Gas Burner.

BOILER CONTROLS

BOILER OUTPUT CONTROL

There are three ways to control the output of a commercial boiler:

1. On-off (cycling) control.
2. High-fire, low-fire control.
3. Modulating control.

On-off (cycling) control is most common for small boilers up to 1,000,000 Btuh capacity. The oil or gas burner cycles on and off to maintain steam pressure or water temperature. Cycling control causes losses in efficiency because of the cooling of the fireside surfaces by the natural draft from the stack during the off, prepurge, and postpurge cycles necessary for safety.

High-fire, low-fire burners provide fewer off cycle losses since the burner shuts off only when loads are below the low-fire rate of fuel input.

Modulating control is used on most large boilers because it adjusts the output to match the load whenever the load is greater than the low-fire limit, which is usually not less than 15 percent of the full load capacity. Steam pressure or hot water temperature is measured to control the volume of gas or oil admitted to the burner.

Boiler firing and safety controls are boiler manufacturer furnished and code approved. A BMCS usually enables a boiler to fire, provides a setpoint, controls pumps and blending valves, and monitors alarms and operation.

COMBUSTION CONTROL

Combustion control regulates the air supplied to a burner to maintain a high gross efficiency in the combustion process. More sophisticated systems use an oxygen sensor in the stack to control the amount of combustion air supplied. Smoke density detection devices can be used in the stack to limit the reduction of air so stack gases stay within smoke density limits. A continuous reading and/or recording of flue gas conditions (percent O_2, stack temperature) is usually included in the control package of large boilers.

A simple combustion control system contains a linkage that readjusts the air supply from the same modulating motor that adjusts fuel supply (Fig. 44). There may be a provision to stop flow of air through the fluebox during the off cycles.

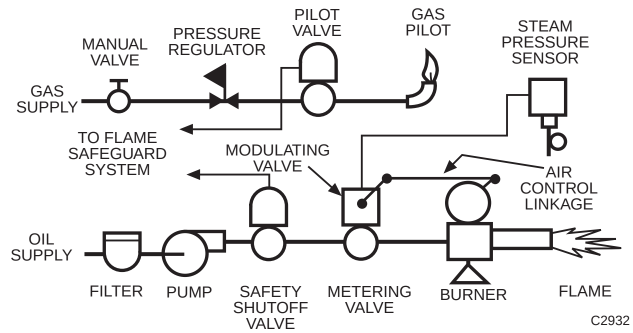

Fig. 44. Combustion Control for Rotary Oil Burner.

FLAME SAFEGUARD CONTROL

Flame safeguard controls are required on all burners. Controls for large burners can be very complicated while controls for small burners such as a residential furnace are relatively simple. The controls must provide foolproof operation, that is, they must make it difficult or impossible to override any of the safety features of the system. The controls also should be continuous self check. For commercial and industrial burners, the flame safeguard control goes through a series of operations. The following sequence is an example:

- Purge firebox of unburned fuel vapor (prepurge).
- Light pilot.
- Verify that pilot is lit.
- Open main fuel valve.
- Verify that flame is present as soon as fuel is introduced.
- Cut off fuel supply promptly if the flame fails.
- Purge firebox of any unburned fuel after each on cycle (postpurge).

The key to any flame safeguard system is a reliable and fast means of detecting the presence or absence of a flame. Methods of detection are:

- Response of bimetal sensor to heat (slow response).
- Response of thermocouple to heat (slow response).
- Flame conductivity (fast but can be fooled).
- Flame rectification (fast, reliable).
- Ultraviolet flame detection (fast, reliable).
- Lead sulfide (photo) cells (fast, reliable if flame frequency check included).

Some sensors can potentially cause improper operation because of shorts, hot refractories, or external light sources. Other sensors, like flame rectification and ultraviolet, respond to flame only. Flame safeguard systems must be approved by UL or Factory Mutual for specific applications.

Figure 45 shows a flame safeguard system often applied to small gas boilers or furnaces. The flame of the gas pilot impinges on a thermocouple which supplies an electric current to keep the pilotstat gas valve open. If the pilot goes out or thermocouple fails, the pilotstat valve closes or remains closed preventing gas flow to the main burner and pilot. The pilotstat must be manually reset.

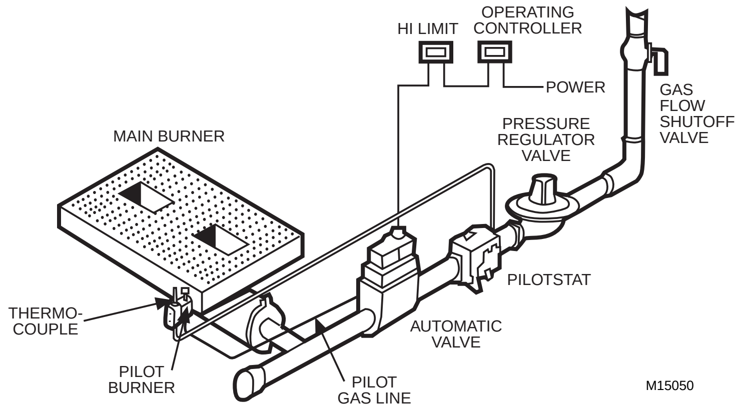

Fig. 45. Simple Flame Safeguard for a Gas Furnace.

Figure 46 shows how flame safeguard controls are integrated with combustion controls on a small oil fired steam boiler. The ultraviolet (UV) flame detector is located where it can see the flame and will shutdown the burner when no flame is present.

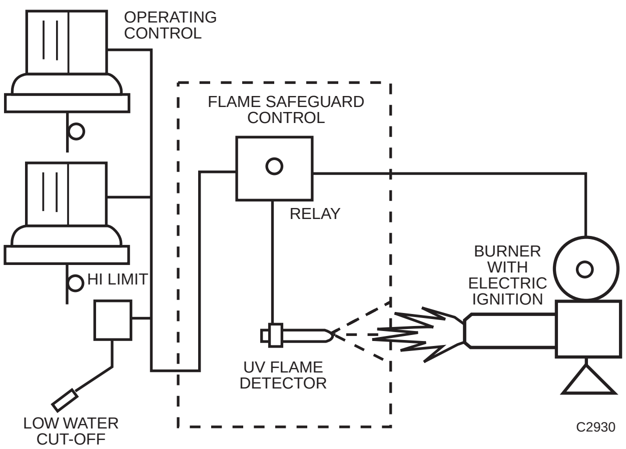

Fig. 46. Combustion Controls with Flame Safeguard Circuit.

FLAME SAFEGUARD INSTRUMENTATION

In addition to the combustion, safety, and flame safeguard controls shown in Figure 46, additional instrumentation often provided on larger burners measures:

– Percent of O_2 or CO_2 in flue gas (to monitor combustion efficiency).
– Flue gas temperature.
– Furnace draft in inches of water column.
– Steam flow with totalizer or hot water Btu with totalizer.
– Oil and/or gas flow with totalizer.
– Stack smoke density.

APPLICATION OF BOILER CONTROLS

Boilers have to provide steam or hot water whenever heat is needed. A conventional BMCS is often set to provide a continuous hot water or steam supply between October and May at anytime the OA temperature drops to 60F for more than 30 minutes and an AHU is calling for heat. The BCMS should include a software ON-OFF-AUTO function. Unlike chillers, boilers can be left enabled at no-load conditions, during which time the water temperature will be held at the design temperature. Frequent warm-up and shutdown of boilers causes stress buildup. Boiler manufacturers recommendations provide specific guidelines.

Unless a water temperature low limit is used, hot water boiler burners are not controlled to provide water temperatures based on outdoor temperatures because the reset schedules require water temperatures to be supplied below the dew point temperature of the flue gas. Some boilers require entering water temperatures to be above 140F before going to high fire. In this case, if a building is using hot water and the boiler is locked into low-fire because the entering water is too cold, the system may never recover.

MULTIPLE BOILER SYSTEMS

GENERAL

Basic boiler connections for a three-zone hot water system are shown in Figure 47. In this system, two boilers are connected in parallel. Hot water from the top of the boilers moves to the air separator which removes any entrapped air from the water. The expansion tank connected to the separator maintains pressure on the system. The tank is about half full of water under normal operating conditions. Air pressure in the tank keeps the system pressurized and allows the water to expand and contract as system water temperature varies. Water from the boiler moves through the separator to the three zone pumps, each of which is controlled by its own zone thermostat. In some systems, each zone may have a central pump and a valve. Return water from the zones returns to the boiler in the return line. There several variations are possible with this type system but the process is the same. There is no minimum boiler water flow limit in this example.

The Dual Boiler Plant Control example following is a dual boiler plant with high-fire, low-fire controlled boilers, 145F minimum entering water temperature required prior to high-fire, water flow must be maintained when the boiler is enabled, and a secondary hot water reset schedule of 110F water at 55F OA temperature and 180F water at 5F OA temperature. The concepts adapt well for single or multiple boiler systems.

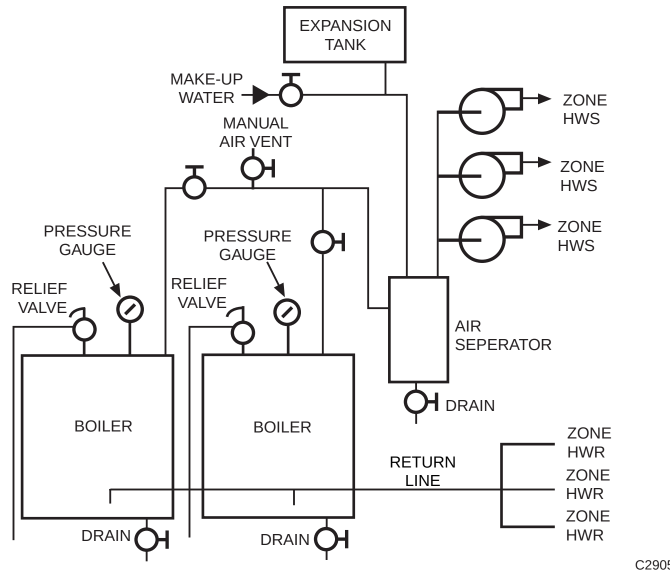

Fig. 47. Typical Piping for Multiple-Zone Heating System.

NOTE: The primary/secondary decoupler is sized for the full secondary flow, and like the chiller plant decoupler, should be a minimum of 6 pipe diameters in length. Unlike the chiller decoupler, normal flow may be in either direction.

DUAL BOILER PLANT CONTROL

FUNCTIONAL DESCRIPTION-

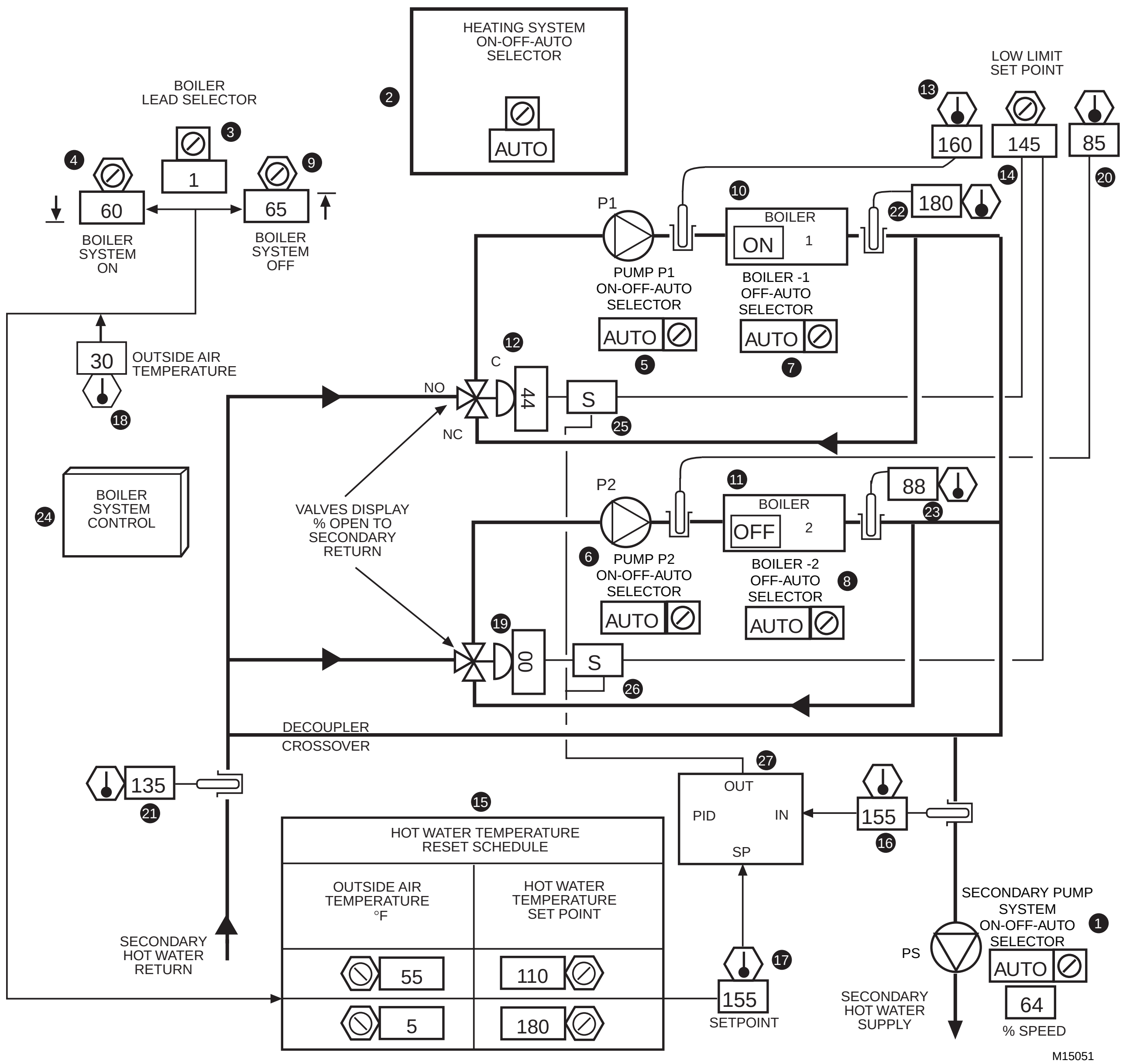

Fig. 48. Dual Boiler Plant Control Graphic

Item No.	Function
1	ON-OFF-AUTO function for secondary pumping system
2	ON-OFF-AUTO function for heating system
3	Selects lead boiler.
4	Heating system start point (Outside air temperature).
5,6	ON-OFF-AUTO function for primary pumps.
7,8	OFF-AUTO function for boilers.
9	Heating system stop point (Outside air temperature).
10,11	Operator information.
12-14	Valve modulates to prevent entering water from dropping below low limit setpoint (145F).
15-18	Secondary water setpoint reset from OA.
19,20	Valve modulates to prevent entering water from dropping below low limit setpoint (145F).
21-23	Operator information.
24	Icon, selects the Boiler System Control dynamic display (Fig. 49).
25,26	Software signal selection functions, allows valve to control secondary HW temperature, subject to boiler low limits.
27	OA reset valve control PID.

Features

1. Full flow through operating boilers.
2. Minimum temperature limit on boiler entering water.
3. Variable flow secondary system with full boiler flow.
4. Automatic boiler staging
5. User friendly monitoring and adjustment.

Conditions For Successful Operation

1. Control network, software, and programming to advise heating plant controller of secondary fan and water flow demands.
2. Interlock and control wiring coordinated with the boiler manufacturer.
3. Control in accord with boiler manufacturers recommendations.
4. Proper setpoint and parameter project specific settings.

Specification

The heating plant shall operate under automatic control anytime the secondary pump ON-OFF-AUTO function is not “OFF”, subject to a heating system ON-OFF-AUTO software function. The lead boiler, as determined by a software lead boiler selection function, shall be enabled anytime the date is between October first and May first, the OA temperature drops below 60F for greater than 30 minutes, and an AHU is calling for heat. Each boiler primary pump shall have a software ON-OFF-AUTO function, and each boiler shall have a software AUTO-OFF function. The heating plant shall be disabled anytime the OA temperature rises to 65F for greater than 1 minute and after May 1.

Anytime the boiler plant is enabled, the lead boiler primary pump shall start and as flow is proven, the boiler shall fire under its factory controls to maintain 180F. If the lead boiler status does not change to “on”, or if flow is not proven within 5 minutes, the lag boiler shall be enabled.

BOILER SYSTEM CONTROL

A BLENDING VALVE ON EACH BOILER MODULATES IN THE RECIRCULATING POSITION TO PREVENT THE BOILER ENTERING WATER TEMPERATURE FROM DROPPING BELOW 145 DEGREES.

LEAD BOILER (1) AND ITS ASSOCIATED PUMP START ANYTIME THE OUTSIDE AIR TEMPERATURE DROPS TO 60 AND SHUTS DOWN ANYTIME THE OUTSIDE AIR TEMPERATURE RISES TO 65 DEGREES. ANYTIME THE LEAD BOILER STARTS FROM THIS OUTSIDE AIR TEMPERATURE SETTING, THE OTHER BOILER IS LOCKED OUT FOR 60 MINUTES.

ANYTIME THE LEAD BOILER CONTROL VALVE IS COMMANDED FULL OPEN BY THE SECONDARY WATER TEMPERATURE CONTROL LOOP FOR 5 MINUTES AND THE SECONDARY HOT WATER SUPPLY TEMPERATURE IS MORE THAN 5 DEGREES BELOW IT'S SETPOINT, THE LAG BOILER AND ITS ASSOCIATED PUMP START. ANYTIME BOTH BOILERS ARE OPERATING AND THEIR CONTROL VALVES ARE LESS THAN 40 PERCENT OPEN TO THE SECONDARY RETURN, THE BOILER SYSTEM OPERATING LONGEST SHUTS DOWN.

THE BOILER BLENDING VALVES MODULATE (SUBJECT TO THEIR LOW LIMIT CONTROL) TO PRODUCE SECONDARY WATER TEMPERATURES FROM 110 TO 180 DEGREES AS THE OUTSIDE AIR TEMPERATURE DROPS FROM 55 TO 5 DEGREES.

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Fig. 49. Boiler System Control Dynamic Display.

During boiler operation, a three way blending valve shall position to place the boiler flow in a recirculating mode until the water entering the boiler exceeds a low limit value of 145F, at which time the blending valve shall modulate to maintain the secondary water temperature between 110F and 180F as the OA temperature varies from 55F to 5F.

The lag boiler shall be locked out from operation for 60 minutes after the lead boiler starts. Thereafter, anytime one boiler control valve is commanded full open by the secondary temperature control loop for greater than 5 minutes and the secondary water temperature is a temperature less than 5F below the secondary water temperature setpoint, the "off" (lag) boiler pump shall start, and upon proving flow, the "off" boiler shall be enabled to fire under its factory controls to maintain 180F. The just-started boiler blending valve shall be controlled by an entering water 145F temperature low limit sensor and setpoint similar to the lead boiler, and subsequently, in unison with the other boiler blending valve to maintain the reset secondary hot water temperature.

Anytime both boilers are operating and their control valves are less than 40% open to the secondary return line, the boiler and pump that has run longest shall shut down.

MODULAR BOILERS

Modular boilers provide heat over a large range of loads and avoid standby and other losses associated with operating large boilers at small loads. Figure 50 shows a primary-secondary piping arrangement where each modular boiler has its own pump. The boiler pump is on when the boiler is on.

Boilers that are off have no flow and are allowed to cool. Each boiler that is on operates at or near full capacity. Avoiding intermittent operation prevents losses up the stack or to the surrounding area when the boiler is off.

Normal control of modular boilers cycles one of the on-line boilers to maintain water temperature in the supply main to meet load requirements. The supply main control sensor cycles the boilers in sequence. If the load increases beyond the capacity of the boilers on-line, an additional boiler is started. The lead (cycling) boiler can be rotated on a daily or weekly basis to equalize wear among all boilers or when using digital control, the program can start the boiler that has been off the longest.

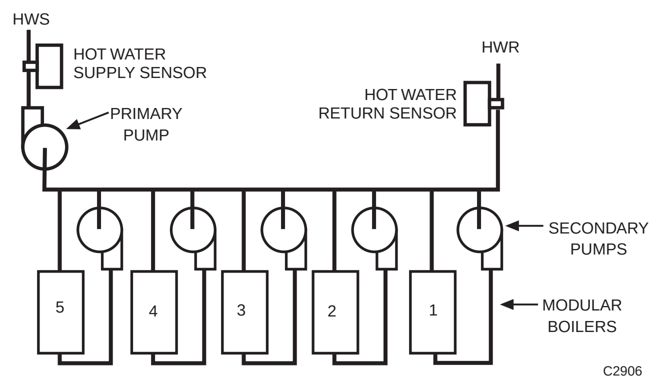


Fig. 50. Typical Primary-Secondary Piping for Modular Boilers.

HOT AND CHILLED WATER DISTRIBUTION SYSTEMS CONTROL

INTRODUCTION

Hot and chilled water pumping, distribution, and control systems have similar characteristics. A hot and/or chilled water system distributes heating or cooling energy through a building. The water is pumped from a boiler or chiller to coils or terminal units. Effective control of this energy requires understanding the control loops and related control valves and also an understanding of the pressure/flow relationships between the piping and pumping components of the system.

CLASSIFICATION OF WATER DISTRIBUTION SYSTEMS

Water distribution systems used in buildings include:

— **LTW**. Low temperature water systems supply water at temperatures up to 250F and working pressures up to 160 psi. Although, most LTW boilers have a maximum working pressure of 30 psi.

— **MTW**. Medium temperature water systems supply water at temperatures between 250 to 350F with pressures up to 160 psi. Maximum medium temperature boiler temperature is 350F.

— **HTW**. High temperature hot water systems supply water at temperatures over 350F, usually in the 400 to 450F range, with working pressures up to 300 psi.

— **CHW**. Chilled water systems supply water at temperatures from 40 to 55F with pressures up to 125 psi.

— **DTW**. Dual temperature water systems supply LTW during the heating season and CHW during the cooling season to the same terminal units.

TYPICAL WATER DISTRIBUTION SYSTEM

A typical system (Fig. 51) illustrates the principles of water distribution in a system. The system consists of a heating or cooling source, a pump, distribution piping, and valve controlled coils. The pump provides force to push the water through the system and valves control the flow through the individual coils. The air separator removes entrapped air from the system.

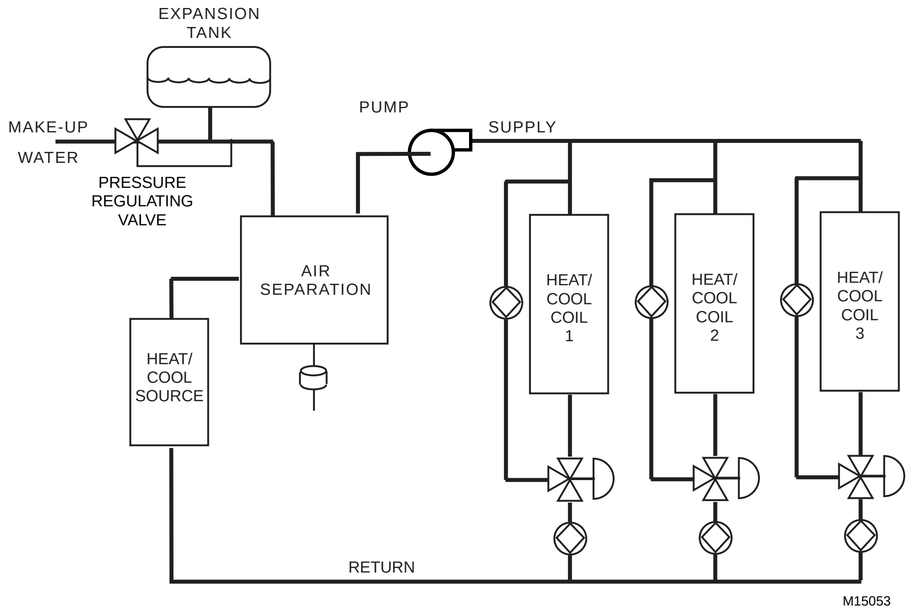


Fig. 51. Typical Water Distribution System.

The expansion tank is charged with compressed air to place the system under the minimum pressure required at the inlet to the pump to prevent pump cavitation and the resultant impeller erosion. The minimum inlet pressure required by a pump is referred to as the net positive suction head (NPSH). Figure 54 indicates the NPSH for a particular pump. The air volume in the tank is sized, based upon the volume of water in the system and the expected water temperature variations, to allow the water to expand and contract as water temperatures vary throughout the year. The expansion tank static pressure does not effect the closed system control valve differential close-off pressure, but must be considered, in addition to the pump head, for valve body and other piping system component pressure rating selection.

The air separator and expansion tank are omitted from the examples in this section for simplicity.

CONTROL REQUIREMENTS FOR WATER DISTRIBUTION SYSTEMS

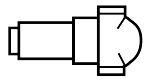
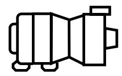
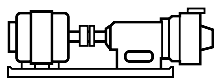
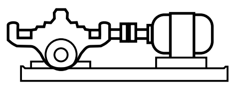
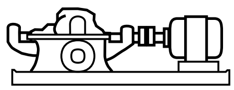
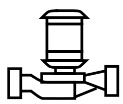
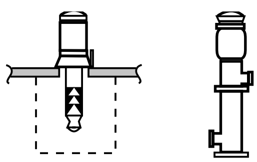
The requirements of a properly applied distribution system are:

1. Maintain controllable pressure drop across the control valves.
2. Maintain required flow through the heating (or cooling) source.
3. Maintain desired water temperature to the terminal units.
4. Maintain minimum flow through the pump(s) that are running.
5. Maintain NPSH.
6. Stage the pumps of a multipump system to satisfy conditions 1 through 5.

CENTRIFUGAL PUMPS USED IN HOT AND CHILLED WATER SYSTEMS

The pump is a key component of a water distribution system. It is essential to understand pump characteristics in order to understand and design distribution systems and to understand pumping system control solutions. Centrifugal pumps are commonly used to distribute hot and chilled water through commercial buildings. Many varieties of centrifugal pumps are available, as shown in Table 2. Figure 52 shows a typical base-mounted pump.

Table 2. Characteristics of Centrifugal Pump Types.

Type		Impeller Type	No. of Impellers	Casing	Motor Connection	Motor Mounting Position
Circulator		Single suction	One	Volute	Flexible-coupled	Horizontal
Close-coupled, end suction		Single suction	One or two	Volute	Close-coupled	Horizontal
Frame-mounted, end suction		Single suction	One or two	Volute	Flexible-coupled	Horizontal
Double suction, horizontal split case		Double suction	One	Volute	Flexible-coupled	Horizontal
Horizontal split case, multistage		Single suction	Two to five	Volute	Flexible-coupled	Horizontal
Vertical inline		Single suction	One	Volute	Flexible- or close-coupled	Vertical
Vertical turbine		Single suction	One to twenty	Diffuser	Flexible-coupled	Vertical

Source: ASHRAE Handbook—1996 Systems and Equipment

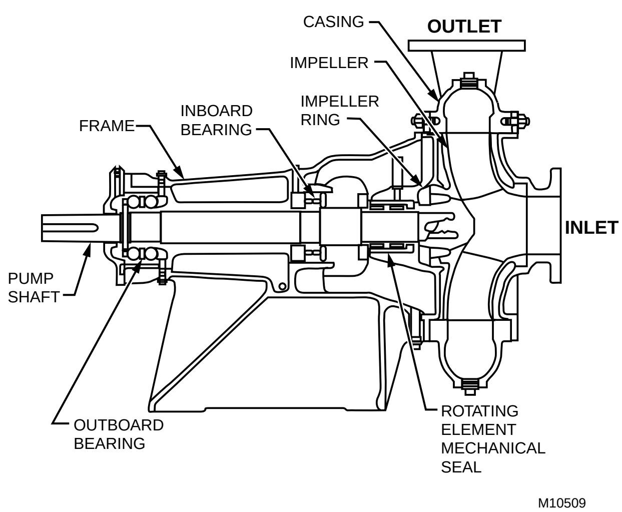


Fig. 52. Typical Cross-Section of an End Suction Pump.

PUMP PERFORMANCE

The performance of a given pump is expressed in a curve showing pump head in feet versus gallons per minute (gpm). Figure 53 shows a typical curve. The head is expressed in feet (of water column) which describes pump operation independent of water temperature or density. Pressure losses in piping and components used in HVAC systems are always calculated in feet.

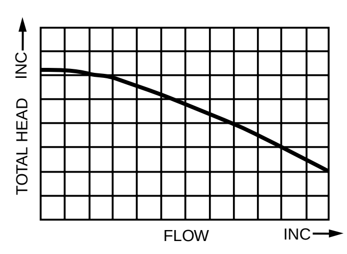


Fig. 53. Typical Pump Head Capacity Curve.

Pump Power Requirements

The pump curve in Figure 53 is part of a family of curves for a pump. Each curve of the family represents a different size impeller used with the pump at a specified rpm. It relates to the power input required just to move the water (water horsepower) as follows:

Water hp = (flow x head x SG) ÷ 3960

Where:

flow = gpm
head = ft
SG = specific gravity of the liquid (water = 1)

The motor driving the pump must have a horsepower rating in excess of water horsepower to take care of bearing and seal friction, recirculation within the housing, and impeller efficiency.

NOTE: Water horsepower increases with head and flow. If flow is allowed to increase, the motor may overload.

Pump Performance Curves

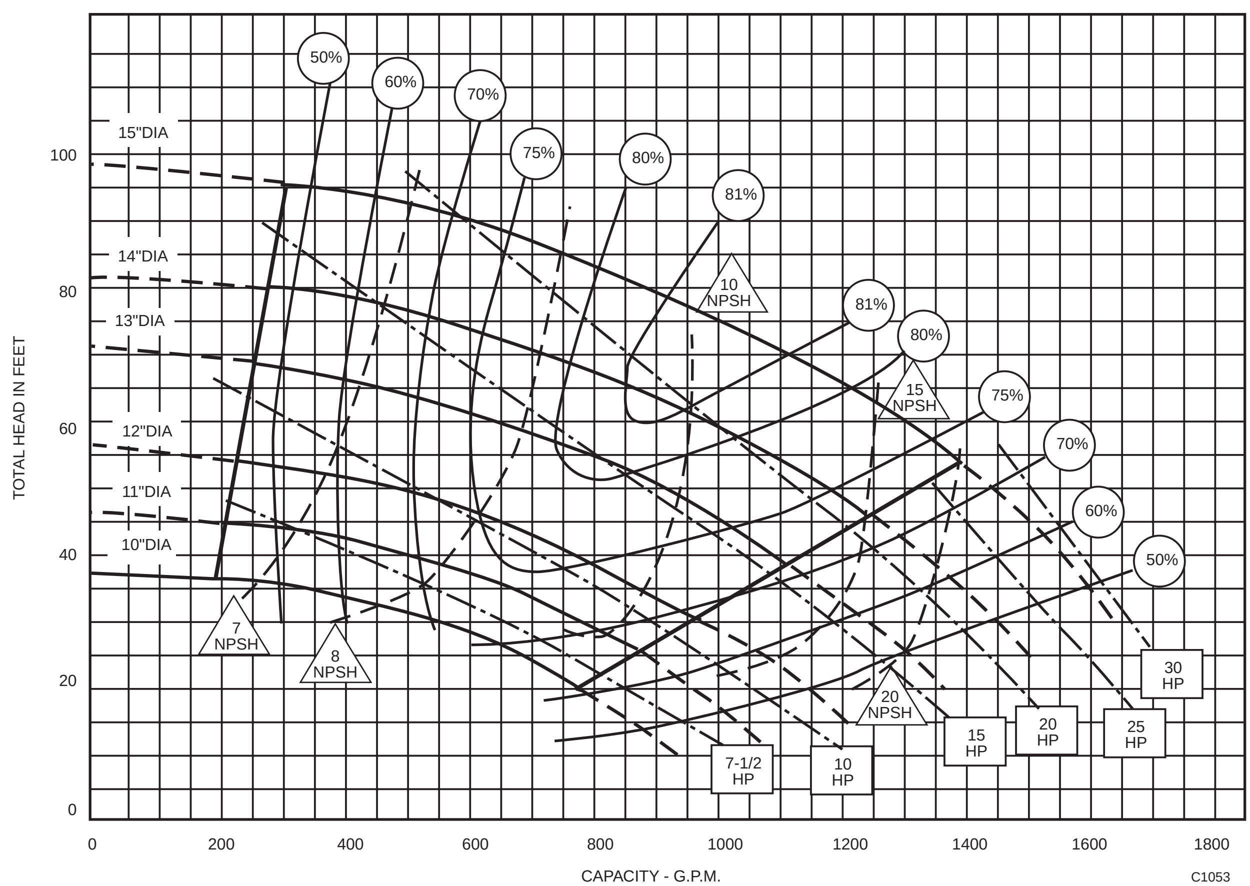
Commercial pumps have performance curves showing the following data for a given pump speed:

— Total head in ft versus flow in gpm
— Total head versus flow for various impeller diameters
— Pump efficiency at various operating points
— Brake horsepower (BHP) or motor horsepower required
— Net positive suction head (NPSH). NPSH is the absolute pressure (psia) required at the suction inlet to prevent cavitation due to “boiling” and formation of bubbles in the water.

Figure 54 is a typical performance curve showing some of the preceding data. Impeller diameters are shown on the left.

Pump Efficiency

Pump efficiency is a comparison of water horsepower developed in the pump and brake horsepower applied by the motor to the shaft and impeller.



Courtesy of Aurora Pump

Fig. 54. 1150 RPM Typical Pump Curve.

Figure 55 illustrates a pump fitted with a 6-1/2 inch impeller, operating at 45 ft of head, and delivering 65 gpm of water.

$$\text{Water hp} = 65 \text{ gpm} \times 45 \text{ ft} \times 1.0/3960 = 0.74 \text{ hp}$$

The curves show that the motor output is 1.17 horsepower. Efficiency = (water horsepower)/(motor horsepower) x 100 = (0.74/1.17) x 100 = 63 percent. This agrees with the efficiency curves shown in Figure 55.

Pump Affinity Laws

Pump affinity laws (Table 3) show how pump flow, head, and brake horsepower vary as impeller diameter or speed change. These laws help when adjusting an installed pump to changes in the system served. For example, if a pump with an 8-inch impeller delivers 80 ft of head, with a 7.21-inch impeller it would deliver 65 ft of head. It is calculated as follows:

$$\text{New head} = \text{Old head} \times \left(\frac{\text{New Diameter}}{\text{Old Diameter}}\right)^2 = 80\left(\frac{7.21}{8}\right)^2 = 64.98 \text{ ft}$$

Table 3. Pump Affinity Laws.

Impeller Diameter	Speed	Specific Gravity (SG)	To Correct for	Multiply by
Constant	Variable	Constant	Flow	$\left(\frac{\text{New Speed}}{\text{Old Speed}}\right)$
			Head	$\left(\frac{\text{New Speed}}{\text{Old Speed}}\right)^2$
			Bhp or kW	$\left(\frac{\text{New Speed}}{\text{Old Speed}}\right)^3$
Variable	Constant	Constant	Flow	$\left(\frac{\text{New Diameter}}{\text{Old Diameter}}\right)$
			Head	$\left(\frac{\text{New Diameter}}{\text{Old Diameter}}\right)^2$
			Bhp or kW	$\left(\frac{\text{New Diameter}}{\text{Old Diameter}}\right)^3$
Constant	Constant	Variable	Bhp or kW	$\frac{\text{New SG}}{\text{Old SG}}$

Source: ASHRAE Handbook--1996 System and Equipment

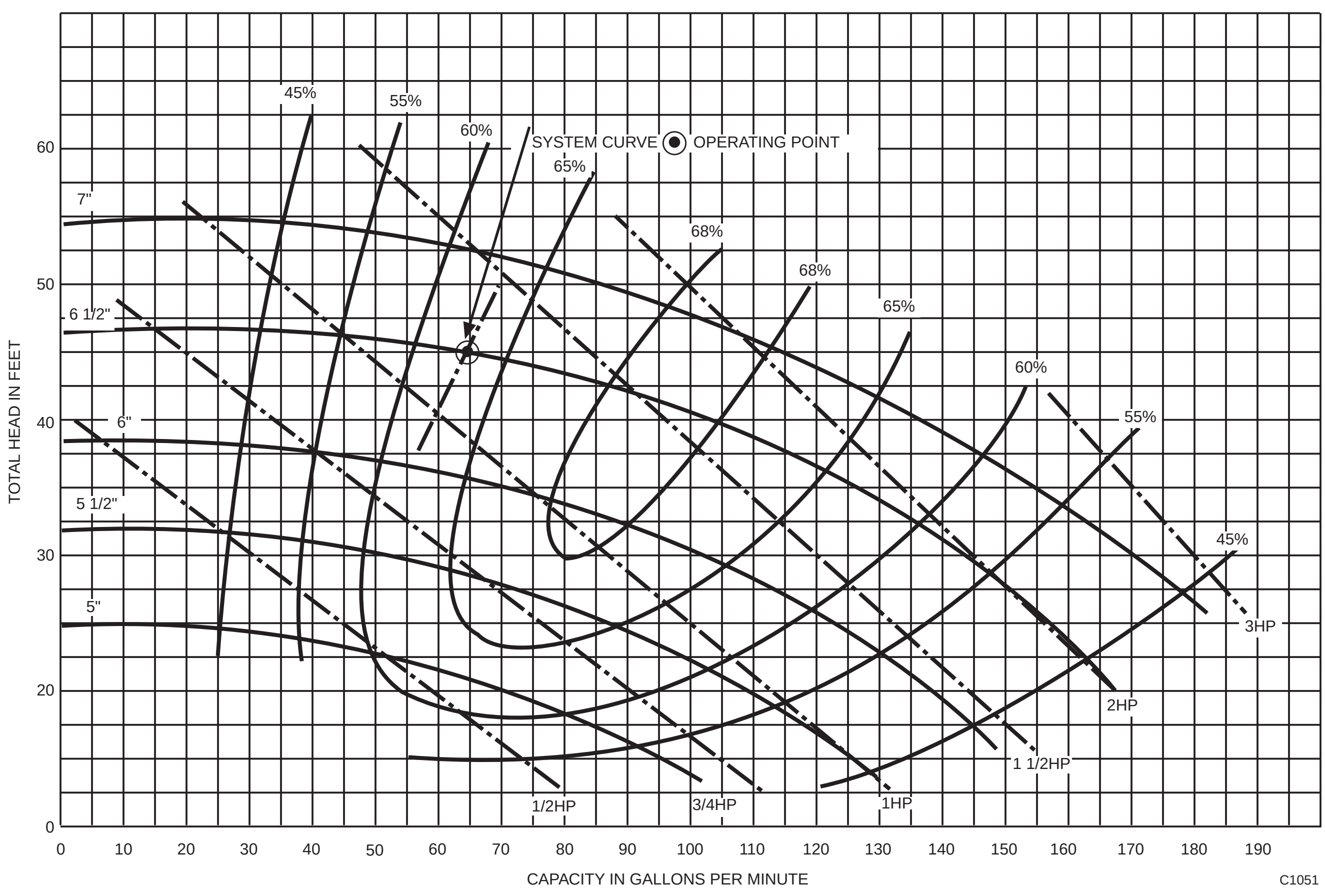


Fig. 55. Pump Curve for 1750 RPM Operation.

MATCHING PUMPS TO WATER DISTRIBUTION SYSTEMS

System Curves

The pump curves and affinity laws are used to select a pump or pumps for a particular application. The first step is to establish a system head curve. This is calculated from design flow and head loss tables for all the piping, coils, control valves, and other components of the system.

Plotting A System Curve

An example is shown in Figure 56. The design point is 65 ft of head at a 515 gpm flow. A system curve (a simple square root curve) can be plotted once the flow and head are known at any particular point, since:

$$\left(\frac{gpm_2}{gpm_1}\right)^2 = \frac{h_2}{h_1}$$

Where:
gpm_1 = flow at h_1 in ft of head
gpm_2 = flow at h_2 in ft of head

Plot the points (Fig. 56) for flows of 200, 400, and 600 gpm:

Point A: $\left(\frac{200}{515}\right)^2$ x 65 ft = 0.1508 x 65 ft = 9.8 ft of water column

Point B: $\left(\frac{400}{515}\right)^2$ x 65 ft = 0.6032 x 65 ft = 39.2 ft of water column

Point C: $\left(\frac{600}{515}\right)^2$ x 65 ft = 1.357 x 65 ft = 88.2 ft of water column

The system curve assumes all balancing valves are set for design conditions, that all controls valves are fully open, and that flow through all loads is proportional. The system curve is always the same, even if loading is not proportional, at no load and full (100 percent) load. If the loads are not proportional (such as, some loads off or some valves throttling), the curve rises above that shown for values between full and no load. The system curve in Figure 56 is used in Figure 57 to select the single speed pump. Using this system curve to determine the switching setpoint for dual parallel pumps when the load flow is not proportional can result in damaging pump cycling.

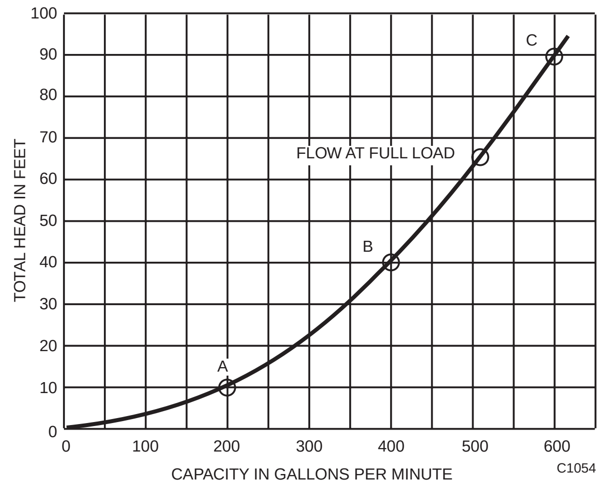


Fig. 56. System Curve for Pump Application.

Combining System And Pump Curves

The design system flow is 515 gpm. Piping, control valve, and equipment losses are calculated at 65 ft. An impeller size and a motor horsepower are selected by imposing the system curve on the pump curve (Fig. 57). The designer has the option of selecting a pump with a 9 1/2 in. impeller (515 gpm at a 76 foot head) or a 8 3/4 in. impeller (500 gpm at a 60 foot head). The smaller impeller requires a 10 horsepower motor and the larger impeller requires a 15 horsepower motor. Selection of the 9 1/2 in. impeller requires system balancing valves to reduce the system pressure differentials to those matching the design flow of 515 gpm.

When selecting a pump, it is important to remember that:

— With constant speed pumps (and two-way AHU control valves), flow rides the pump curve. The system curve is plotted assuming that the control valves are full open, which in any system, only occurs at the full load. As control valves throttle and loads are turned off the system becomes non-proportional and the system curve rises between no load and design.

— With variable speed pumps, the system control objective is to have the pump curve ride the system curve by keeping at least one control valve open and reducing the pump speed to reduce flow with the diminishing system drop.

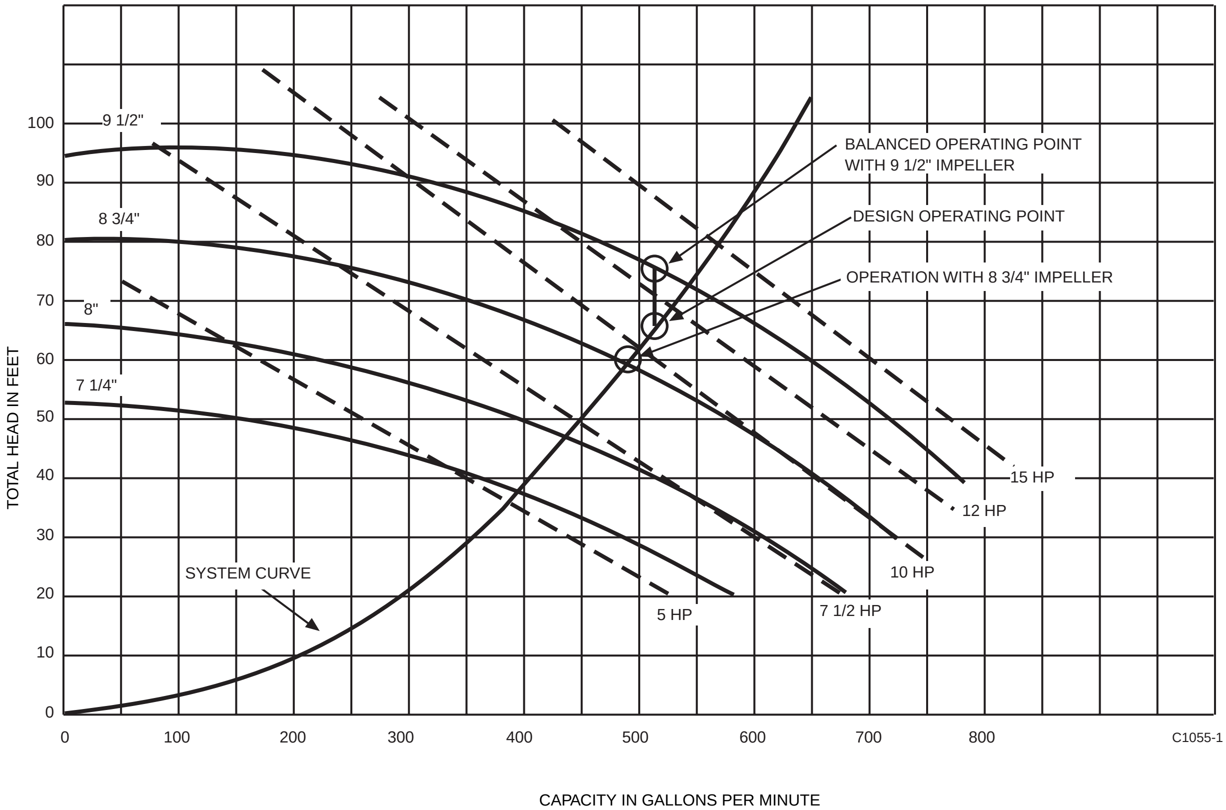


Fig. 57. Matching Pump to System.

To better understand system curves, pump curves, and flow control, Figure 58 shows the control valve(s) (the only variable element of a typical system curve) separately from the rest of the system elements. Lines are shown for each of three valve positions; full, 80 percent, and 50 percent flow. These lines, when added to the curve for all other elements of the system intersect the pump curve at the corresponding operating point(s).

Figure 58 shows a system with 500 gpm and 70 foot head at design, 10 ft of which is a full open control valve at the end of the piping run. Line "A" represents the control valve and connects the pump curve to the static-element system curve. If all control valves positioned to 80 percent flow, the pump head rises, the static-element system resistance drops, and the control valve, represented by line "B", makes the difference; about 36 ft. Similarly, at 50 percent flow, the valve drop, represented by line "C", accounts for about 63 ft.

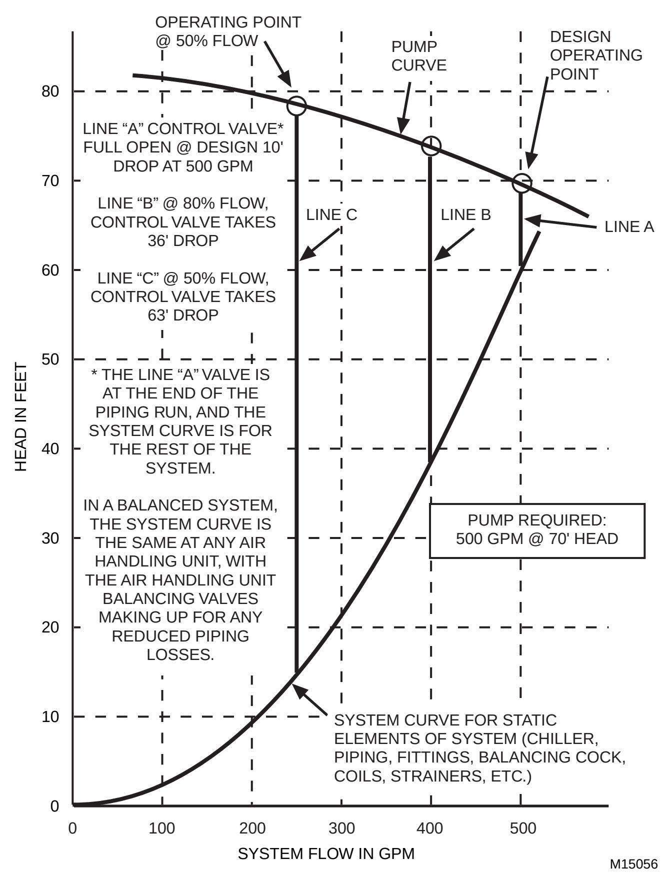


Fig. 58. Pump and System Curves and Control Valves

VARIABLE SPEED PUMPS

From the pump affinity laws (Table 3), pump horsepower decreases by the cube of the decreased speed, and flow decreases linearly with speed; so at 80 percent flow, the horsepower is down to nearly 50 percent (80 percent cubed). Since many systems have sharply reduced flow requirements at medium or low loads, pump speed control can provide economical operation for most of the heating (or cooling) season. Figure 59 shows typical performance at reduced speeds. The shaded area of Figure 60 shows the wide range of heads and flows available from a variable speed pump. Variable speed pumps are usually controlled from a differential pressure sensor with either fixed or load reset setpoints.

Control objectives of variable speed pumping systems in networked digital control systems, is to keep the most demanding load control valve full open by varying the pump speed.

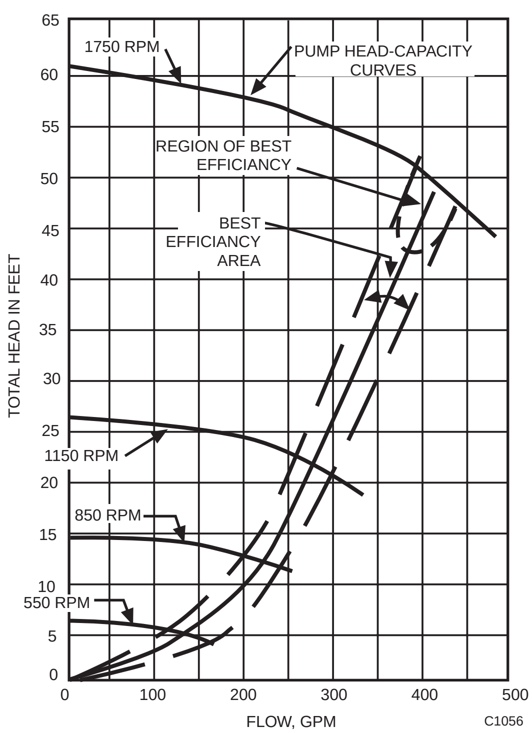


Fig. 59. Pump Performance and Efficiency at Various Speeds.

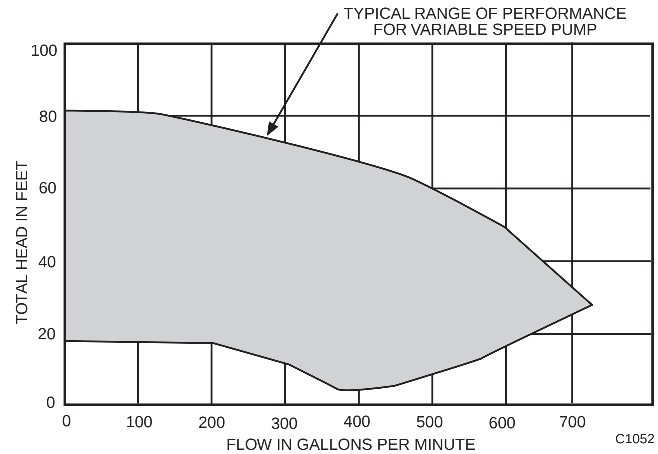


Fig. 60. Typical Variable Speed Pump Performance Range.

Table 4 and Figure 61 show the Figure 58 system with all the control valves remaining full open and the load controlled by varying the pump speed. This is the ideal system wherein the loads of all AHUs vary in unison and the pump speed is controlled to satisfy the valve with the greatest demand. This is usually accomplished via differential pressure control, automatically reset.

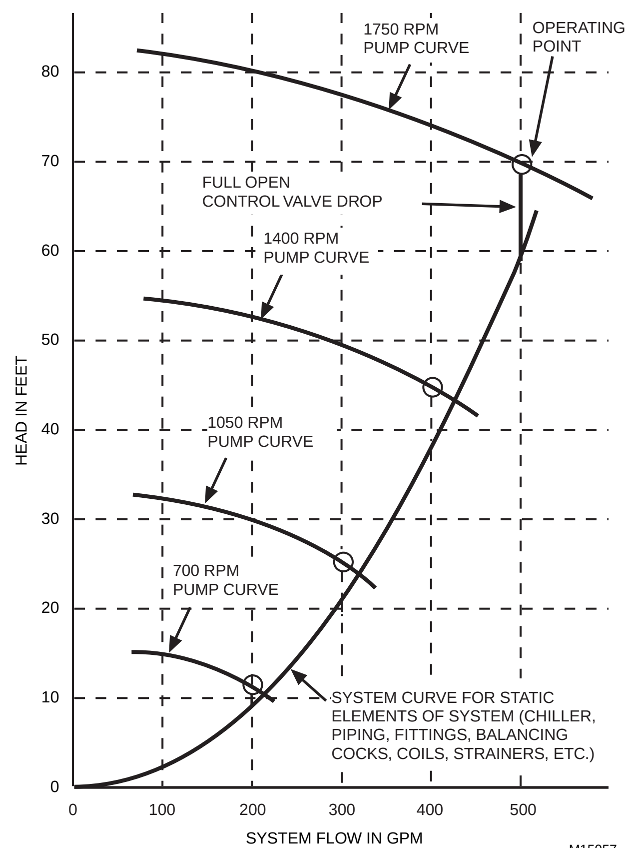


Fig. 61. Ideal Variable Speed Pump Control.

Table 4. Variable Speed Pump Relationships

Condition	Load (percent)			
	100	80	60	40
Flow (gpm)	500	0.80 x 500 = 400	0.60 x 500 = 300	0.40 x 500 = 200
Speed (rpm)	1750	0.80 x 1750 = 1400	0.60 x 1750 = 1050	0.40 x 1750 = 700
Total Head (ft)	70	$70 \times \left(\frac{1400}{1750}\right)^2 = 44.8$	$70 \times \left(\frac{1050}{1750}\right)^2 = 25.2$	$70 \times \left(\frac{700}{1750}\right)^2 = 11.2$
Fully Open Valve Drop (ft)	10	$10 \times 0.80^2 = 6.4$	$10 \times 0.60^2 = 3.6$	$10 \times 0.40^2 = 1.6$
Brake Horsepower	13.6	$13.6 \times \left(\frac{1400}{1750}\right)^3 = 6.96$	$13.6 \times \left(\frac{1050}{1750}\right)^3 = 2.94$	$13.6 \times \left(\frac{700}{1750}\right)^3 = 0.87$

PUMPS APPLIED TO OPEN SYSTEMS

In cooling towers (Fig. 62) and other open systems, static head must be considered when establishing system curves and selecting a pump. Notice that the 90 ft of vertical pipe (static discharge head) is partially offset by the 80 ft of vertical pipe (static suction head) in the suction line. When a system curve is drawn for such a system, static head of the tower must be added to the system curve. The system is designed to operate at 200 gpm against 30 ft of head for piping and valves. The system curve in Figure 63 is drawn through zero head (ignoring the static head of 10 ft) which leads to choosing Pump A.

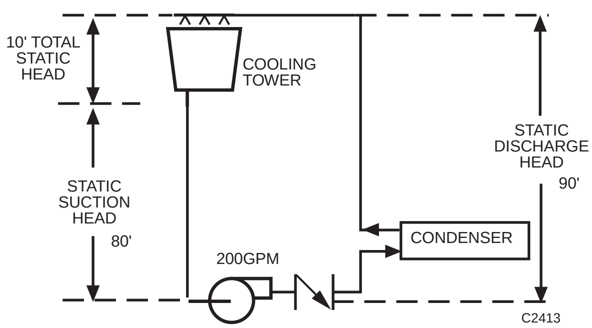


Fig. 62. Typical Cooling Tower Application.

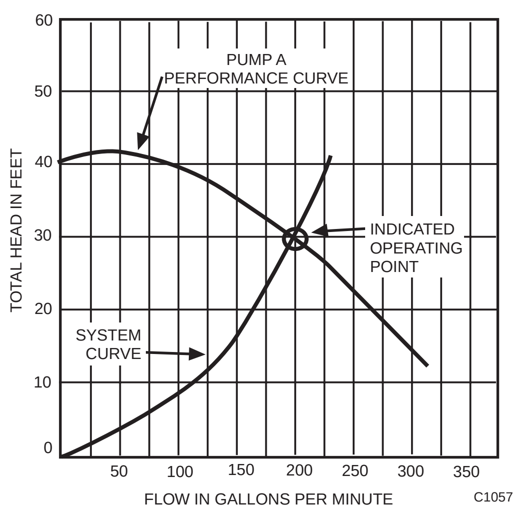


Fig. 63. System Curve for Open Circuit without Static Head.

Figure 64 shows the system curve adjusted for the 10 ft of static head and the actual operating points of Pump A (selected in Figure 63) and Pump B. Notice Pump A supplies only 175 gpm at 32 ft of head.

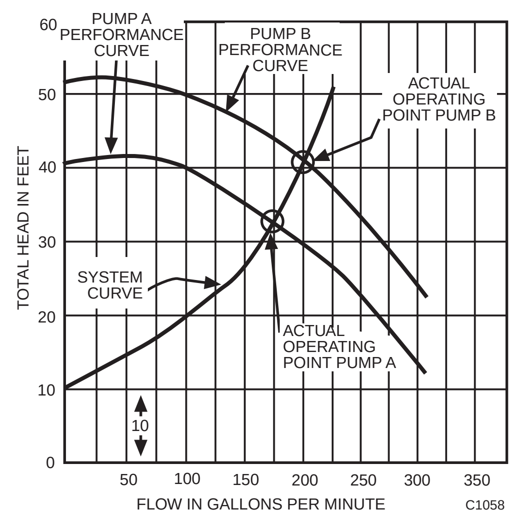


Fig. 64. System Curve for Open Circuit with Static Head.

MULTIPLE PUMPS

Multiple pumps are used when light load conditions could overload a single pump. These conditions normally occur when two-way control valves are used in the control system. Two-way control valves sharply reduce flow when they begin to close. Figure 65 shows that in a single-pump system, over pressure can result at low flow. At one-third flow, the pump head has increased, the source and piping drop is reduced to one-ninth of the design drop, and the control valve drop has increased greatly. Bypass, variable speed, or throttling valve pressure relief should be used with a single pump. Where the heat exchanger (such as a chiller) requires a high minimum flow rate, a single pump is used, and diversity is not used, three-way load control valves should generally be used.

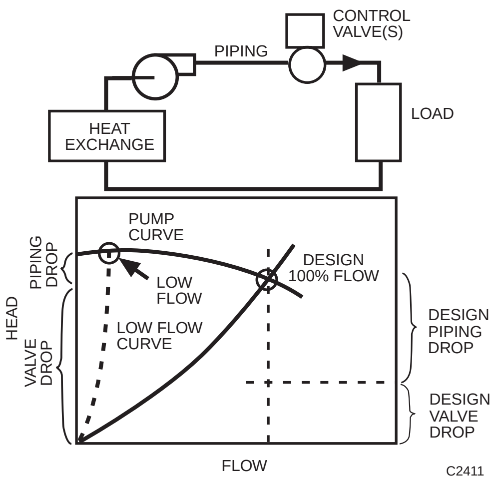


Fig. 65. System Operation with One Pump, Design and Low Flow Condition.

Operation

Multiple pumps may be connected either in parallel or in series into the system. In the dual parallel pump configuration of Figure 66 a single pump can usually handle 75 to 80 percent of the total flow. The system curves show that at design conditions the control valve drop is ten feet (from A to B). At 75 percent flow (375 gpm), the valve drop with both pumps operating increases to over 41 ft (C to E). With one pump and 75 percent flow the valve drop is about 16 ft (C to D). When flow is reduced to 50 Percent, the valve drop is about 55 ft for one pump (F to G) or 63 ft for two pumps (F to H). Dual parallel pumps save energy and provide redundancy for 75 to 80 percent of the flow. They do not provide much relief for high valve pressure drops at low flow.

The pump curves and the system curves indicate possible pump start/stop setpoints. One scenario on a pumping differential fall to 42 ft, energizes the second pump and on a pumping differential rise to 77 ft, switches back to one pump. The 42 ft pumping differential corresponds to a point just before the 1-pump curve intersects the system curve (I), the point at which a single pump no longer can support the system. When the second pump is started, the operating point moves to the 2-pump curve and when the control valves have settled out will be at about Point J. It will vary along the 2-pump curve down to B or up to K. When the operating point reaches K (about 77 ft) the system switches back to a single pump and the operating point is now on the 1-pump curve until the differential pump pressure drops to I, at which time the cycle repeats. See PLOTTING A SYSTEM CURVE for statement on use of ideal system curve for determining setpoints when coil loading may not be proportional.

Again a reminder to exercise caution when using the ideal system curves for switching pumps on and off. The ideal curves are valid only at full and no load conditions, the rest of the time the actual curve is somewhere above the ideal. Since setpoint determination is not possible without the actual system curve, the lag pump stop setpoint should have a significant margin of safety incorporated. The lag pump start setpoint should be controlled by a differential pressure controller and have the software requirement that one control valve be full open for four minutes before starting.

Time delays must be built in to the control sequence to prevent rapid switching between one pump and two pump operation. With each change in pump operation, all control valves must adjust to new steady-state conditions. The adjustment process often causes overshoot or undershoot until temperature stability returns and no switching should take place during this time. Depending upon the type of temperature control loops, switch-lockout period can vary from 5 minutes for relatively fast discharge air control to over 30 minutes for relatively slow space control.

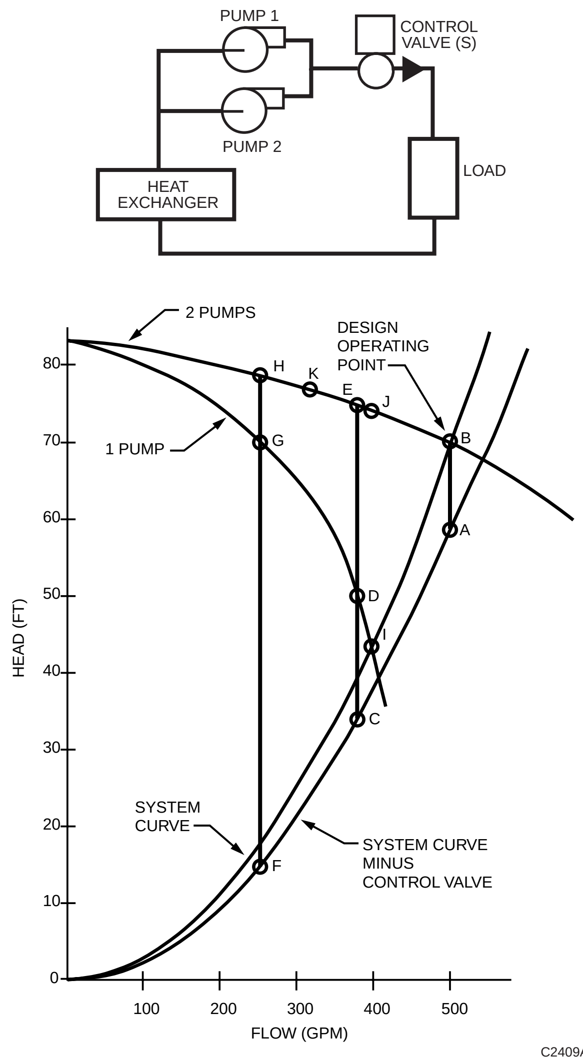


Fig. 66. System Operation with Two Pumps in Parallel.

Series pumps (Fig. 67), though rarely used in HVAC systems, are useful where both flow and head are sharply reduced at light loads.

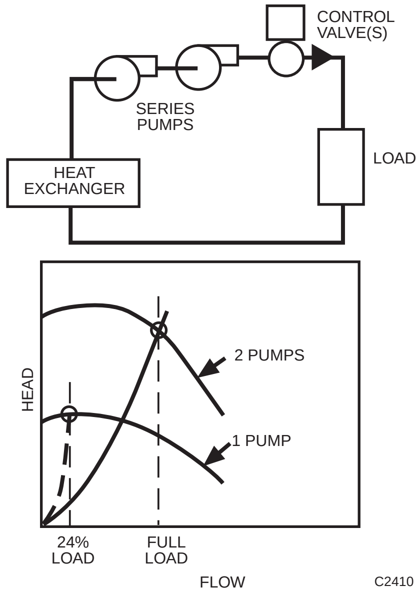


Fig. 67. System Operation for Series Pumps.

Dual Pump Curves

For pumps in parallel (Fig. 66), assuming two identical pumps, the curve is developed using the following formula:

$$gpm_3 = (gpm_1) \times 2 \text{ for any } h_1$$

Where:

gpm_3 = Total flow for both pumps
gpm_1 = gpm of one pump
h_1 = Head in ft for Pump 1 at gpm_1 for any point on pump curve

For pumps in series (Fig. 67), assuming two identical pumps, the curve is developed using the following formula:

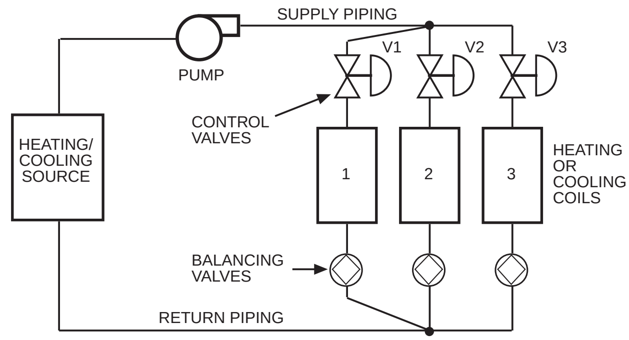
$$h_3 = (h_1) \times 2 \text{ for any } gpm_1$$

Where:

h_3 = Total head in ft for both pumps
h_1 = Head in ft for one pump at gpm_1 (for any point on Pump 1 curve)

DISTRIBUTION SYSTEM FUNDAMENTALS

Figure 68 illustrates a closed system where static head (pressure within the system with pump off) does not need to be considered as long as all components are rated for the static head encountered. The pump provides force to overcome the pressure drop through the system and valves control the flow and pressure through the system. Figure 69 shows a graph of the system and pump curves for design load and reduced load conditions. The system curve indicates the pressure drop through the system (with the control valves full open) at various flow rates. The pump curve shows the pump output pressure at various flow rates. Flow always follows the pump curve.



ITEM	FLOW GPM	DESIGN PRESSURE DROP IN FEET		
		COIL & PIPING	CONTROL VALVE	BALANCING VALVE
HEATING OR COOLING SOURCE AND DISTRIBUTION PIPING	40	23*	–	–
COIL 1 LOOP	10	8	11	2
COIL 2 LOOP	12	10*	11*	0
COIL 3 LOOP	18	7	11	3
TOTAL FLOW AND DROP	40	44*		

* SUM OF SOURCE AND PIPING (23 FT) AND LOOP 2 (21 FT) = 44

M15054

Fig. 68. Simplified Water Distribution System.

In Figure 68 the flow and pressure considerations are:

1. The flow through the heating or cooling source, the supply piping, and the return piping (40 gpm) is the same as the sum of the flows through the three coil circuits:
   10 + 12 + 18 = 40 gpm.
2. Design pressure drop (head loss) includes the drop through the heating or cooling source, supply piping, return piping, and the highest of the three coil circuits:
   23 + (10 + 11) = 44 ft.

NOTE: In this example, Coil 1 and 3 balancing valves balance each load loop at the 21 ft design for Loop 2. If the actual coil and control valve drops were less than the design maximum values, the actual balancing valve effects would be greater.

In this example the pump must handle 40 gpm against a total head of 44 ft (19 psi) as shown in Figure 69. (This curve is taken from actual pump tests). The design drop across the valve is 11 ft (4.5 psi) with the valve fully open.

If Figure 68 is a heating system, as the loads reduce valves V1, V2, and V3 start to close. Hot water flow must be reduced to about 15 percent of full flow (6 gpm) to reduce heat output to 50 percent. As flow through the coil is reduced the water takes longer to pass through the coil and, therefore, gives up more heat to the air.

As flow through the system is reduced, a new system curve is established. See the 6 gpm curve in Figure 69. When the flow is reduced, the new head loss in source and supply and return piping can be calculated using the formula:

$$\left(\frac{gpm_2}{gpm_1}\right)^2 = \frac{h_2}{h_1}$$

Where:

gpm_1 = present flow
gpm_2 = next flow
h_1 = present pressure drop
h_2 = next pressure drop

For example: If gpm_1 = 40, gpm_2 = 6, and h_1 = 33 ft, then h_2 = 0.74 ft.

$$\left(\frac{6}{40}\right)^2 = \frac{h_2}{33}$$

$$h_2 = 33 \times 0.0225 = 0.74 \text{ ft}$$

At low flow, the pressure drops through the coils, coil piping, and heat source tend to disappear and nearly all of the now elevated pump head appears across the partially closed valves V1, V2, and V3. This can cause valve noise, poor control, or failure of valves to seat. Control solutions are discussed in following sections.

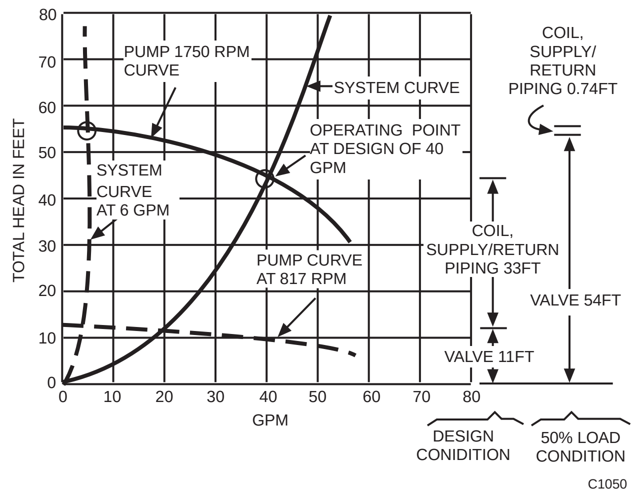


Fig. 69. System and Pump Curves for a Closed System at Various Loads.

DIRECT VS REVERSE RETURN PIPING SYSTEMS

Distribution system control solutions vary dependent upon whether the designer chose a direct or reverse return piping system. Systems are sometimes configured as a combination of both; a high-rise building could, for example be reverse return on the riser and direct return on the floor run-outs. Direct return systems are usually lower cost and used in smaller installations. Reverse return systems are used in both small and large installations.

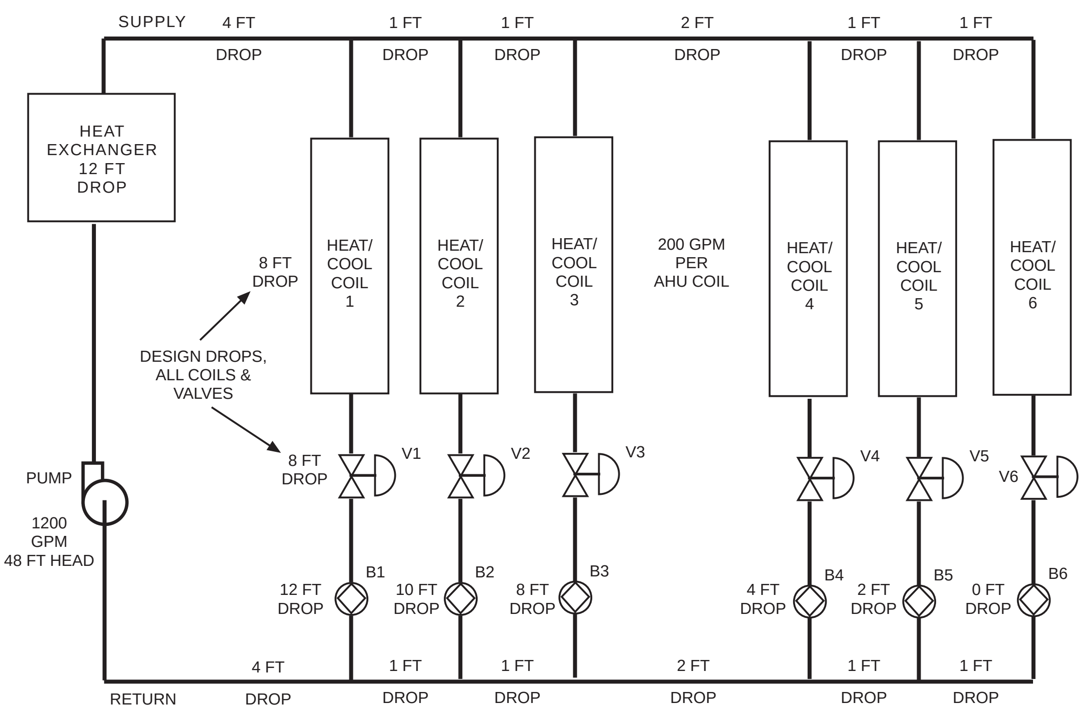


Fig. 70. Direct Return Piping System

The Figure 70 supply piping runs out to the coils decreasing in size between AHU 1, 2, 3, 4, 5, and 6. The return lines between each AHU are typically sized the same as the respective supply lines. The drop across AHU 6 must be 16 ft in order to get the 200 gpm through the valve (8 ft) and the coil (8 ft). To get the 200 gpm from AHU 5 to AHU 6, the drop across the piping at AHU 5 is 18 ft (16 ft required to get the flow through AHU 6 plus 2 ft to overcome the supply and return piping drops between AHU 5 and AHU 6). The AHU 5 balancing valve is set to prevent the 18 ft drop across AHU 5 from forcing more than 200 gpm to pass through the AHU 5 coil and control valve. The balancing valve B5 then is set to take a 2 ft drop at 200 gpm. Similarly, set a 4 ft, 8 ft, 10 ft, and 12 ft drop respectively across balancing valves B4, B3, B2, and B1.

If this is a variable flow loop with a variable speed pump and the pump is controlled to produce 16 ft across AHU 6, the control issue here is: When AHUs 2, 3, 4, 5, and 6 are off (no flow beyond AHU 1), then, the drop across AHU 1 is only 16 ft, 12 ft of which the AHU 1 balancing valve needs for design flow. Proper solutions are presented later in this section.

Supply piping is the same for a reverse return system (Fig. 71) as for the direct return system (Fig. 70). The return flow is reversed such that the return piping increases in size between AHUs 1, 2, 3, 4, 5, and 6. A full size return line is run back to the source room from AHU 6. In this example, the pump is the same as the direct return, since the return line from AHU 6 also takes a 6 ft drop. If the AHUs and source were positioned in a circular or hex pattern such that Coil 6 is closer to the pump, the run from AHU 6 back to the source would be shorter, and the reverse return piping head would be less than for the direct return, and the piping cost would be similar. In reverse return systems, balancing is usually only a trimming exercise.

COUPLED VS DECOUPLED PIPING SYSTEMS

Piping systems requiring constant flow through primary equipment (chillers, boilers) and variable flow to AHUs may be coupled or decoupled. See CHILLER SYSTEM CONTROL and BOILER SYSTEM CONTROL for examples. Primary flow control for coupled systems and secondary flow control for decoupled systems are discussed later.

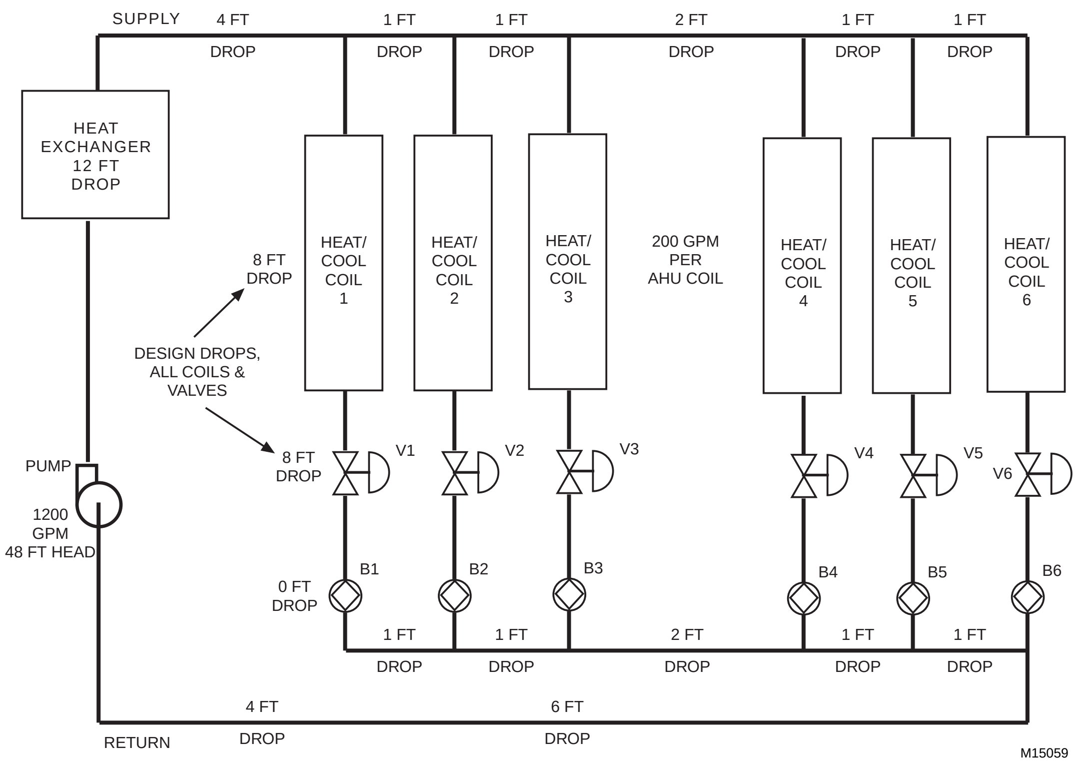


Fig. 71. Reverse Return Piping System

METHODS OF CONTROLLING DISTRIBUTION SYSTEMS

There are several methods for controlling pressure and flow in water distribution systems. The methods described in this section apply, in general, to both heating and cooling applications.

THREE-WAY COIL BYPASS AND TWO-WAY VALVE CONTROL

Coil bypass control uses three-way valves on terminal units and other coil loads in a water distribution system and satisfies the first four of the control system requirements (see CONTROL REQUIREMENTS FOR WATER DISTRIBUTION SYSTEMS). At reduced loads, the flow bypasses the coils and goes directly to the return main. Figure 72 illustrates the operation of this system as requirements change. Balancing valve (B) is adjusted for equal flow in the coil and the bypass

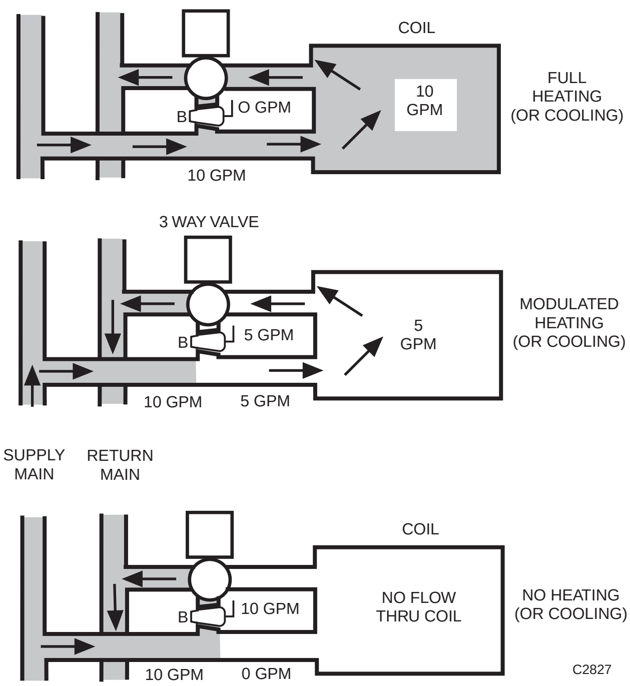


Fig. 72. Three-Way Valve Control—Coil Bypass.

Two way valves vary both the coil flow and the system flow, thus using less pumping energy at reduced flow.

VALVE SELECTION FACTORS

Consider the following factors when deciding on two-way or three-way control valves.

— **Piping Cost.** Costs are higher for three-way valves than two-way valves, especially where limited space is available for piping (such as in room air conditioning units and unit ventilators). In addition, balancing cocks must be installed and adjusted in the bypass line.
— **Three-Way Valve Cost.** A diverting valve is more expensive than a mixing valve and a mixing valve is more expensive than a two-way valve. A mixing valve installed in the leaving water from a coil provides the same control as a diverting valve installed on the inlet to the coil.
— **Diversity**. If chillers and pumps are selected based upon diversity, three-way valves are inappropriate.
— **Flow Characteristics**. Three-way valves have linear flow characteristics and two-way valves may be either linear or equal percentage. Obtaining close control with three-way valves requires use of scheduled (reset) hot water temperatures.
— **Capacity Index (C_V).** Three-way valves for C_Vs below 1.0 are often not available, therefore, small three-way valves tend to be oversized. Consider using two-way control valves for all applications of C_V = 4.0 and less where the quantity of two-way valves will have little effect on the total system flow.
— **Constant Flow in Mains.** Constant flow provides nearly constant pressure differential (drop) across a coil and valve.
— **Pumping Cost.** A three-way valve system uses full pump capacity even when the system load is very small.
— **Part Load Control.** Two-way valves allow better control on multiple pump systems during pump failure or part load periods.
— **Automatic Control.** Distribution control is a manual balancing task for flow loops employing three-way control valves. Automatic distribution controls are usually required to maintain flows and pressures (bypass valve, variable speed pump, pump staging control) for flow loops employing two-way control valves.

For further discussion on control valves, refer to the Valve Selection and Sizing section.

FLOW AND PRESSURE CONTROL SOLUTIONS

Control solutions for water distribution systems may vary based on:
- Direct return vs reverse return piping
- Pressure bypass valve control vs variable speed pumping systems
- Coupled secondary systems vs decoupled systems
- Evenly varying heating/cooling loads vs unevenly varying loads
- Variable flow vs constant flow systems
- Single pump vs multiple parallel pump systems
- Pressure bypass valve objectives of maintaining high flow rates through chillers vs maintaining a low drop across control valves

The examples in this section on Flow And Pressure Control Solutions use a distribution system that has six equal loads (coils) as shown in Figure 73. These control solutions are:

1. Single constant speed pump, single chiller system, two-way AHU control valves, and pressure bypass valve to control chiller flow to a minimum of 90 percent full flow.
   a. direct return.
   b. reverse return.
2. Dual constant speed pumps, dual chiller systems, and pressure bypass valve to control chiller flow to a minimum of 90 percent full flow.
   a. direct return.
   b. reverse return.
3. High control valve differential pressure control.
4. Decoupled variable speed secondary pumping system with two-way AHU control valves.
   a. direct return.
   b. reverse return.

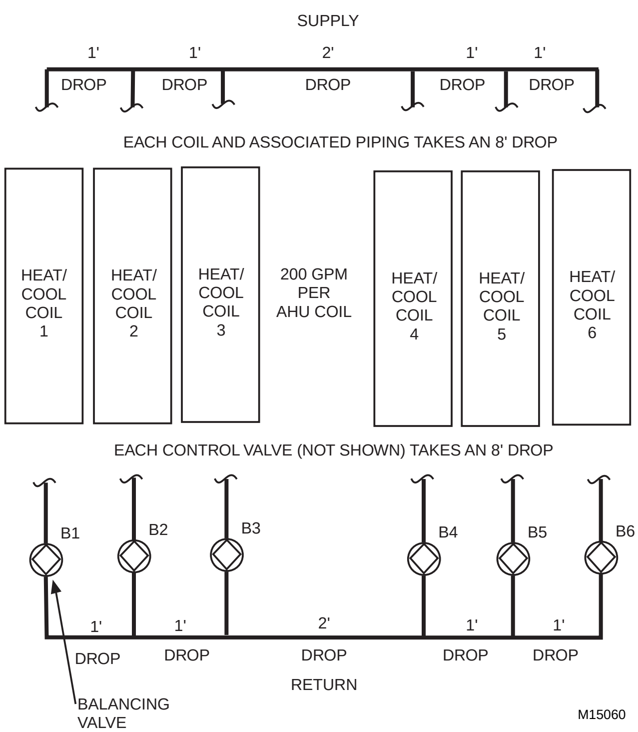


Fig. 73 Typical Example Loads.

Single Pump, Pressure Bypass, Direct Return

Figure 74 analyzes Figure 70 pumping system at full flow and at half flow. The flow reduction at half flow is taken evenly across each coil. At half flow with no pressure bypass the control valve pressure drops increase from 8 ft to 44 ft as system friction drops reduce to one-forth of the design values and the pump head rises from 48 to 54 ft.

Figure 75 shows a pressure controlled bypass valve set to maintain 90 percent minimum flow through the chiller to satisfy the chillers minimum flow requirement. At 90 percent flow through the chiller (1080 gpm), the chiller and equipment room piping drops are 81 percent of design (90 percent squared). The pump curve (not shown) indicates a pump head of 50 ft at 1080 gpm.

VALVE LOCATION AND SIZING

Since the system piping between Loads 1 and 2 is designed for only 1000 gpm and the low load bypass flow could exceed that, the bypass valve is located remotely before Load 1. If necessary to locate the bypass valve after Load 1, redesign the piping to carry the 90 percent flow.

If the differential pressure sensor is located across the main lines at Load 1 as shown in Figure 75 (see DIFFERENTIAL PRESSURE SENSOR LOCATION), the best place for the bypass valve is the same location. Because the differential pressure is lower than in the chiller room, valve wear is less.

The valve is sized for approximately 1000 gpm with a 34 foot drop. A double seated valve is appropriate to reduce actuator close off requirements and the inherent leakage will not be a significant factor.

DIFFERENTIAL PRESSURE SENSOR LOCATION

As previously stated the chiller design flow is 1200 gpm at 12 ft of head and requires a minimum flow of 1080 gpm. At 1080 gpm the pump curve shows a head of 50 ft.

From the formula:

$$\left(\frac{gpm_2}{gpm_1}\right)^2 = \frac{h_2}{h_1}$$

Calculate the drop across the chiller as 9.6 ft.

$$\left(\frac{1080}{1200}\right)^2 = \frac{h_2}{12}$$

$$12\,(0.90)^2 = h_2$$

$$h_2 = 12 \text{ x } 0.81 = 9.6 \text{ ft.}$$

Similarly calculate the reduced drop in the supply and return to Load 1 as 3.2 ft.

With the differential pressure controller located across Load 1, the setting is:

50 ft – 9.6 ft – 3.2 ft – 3.2 ft = 34 ft

This location provides a lower head across the load control valves at light loads than if located across the pump and chiller. To ensure the best sensing, be sure that the system strainer is located up stream from the differential pressure controller return pickup, so that a dirty strainer is not sensed as an increasing pressure drop (decreasing flow).

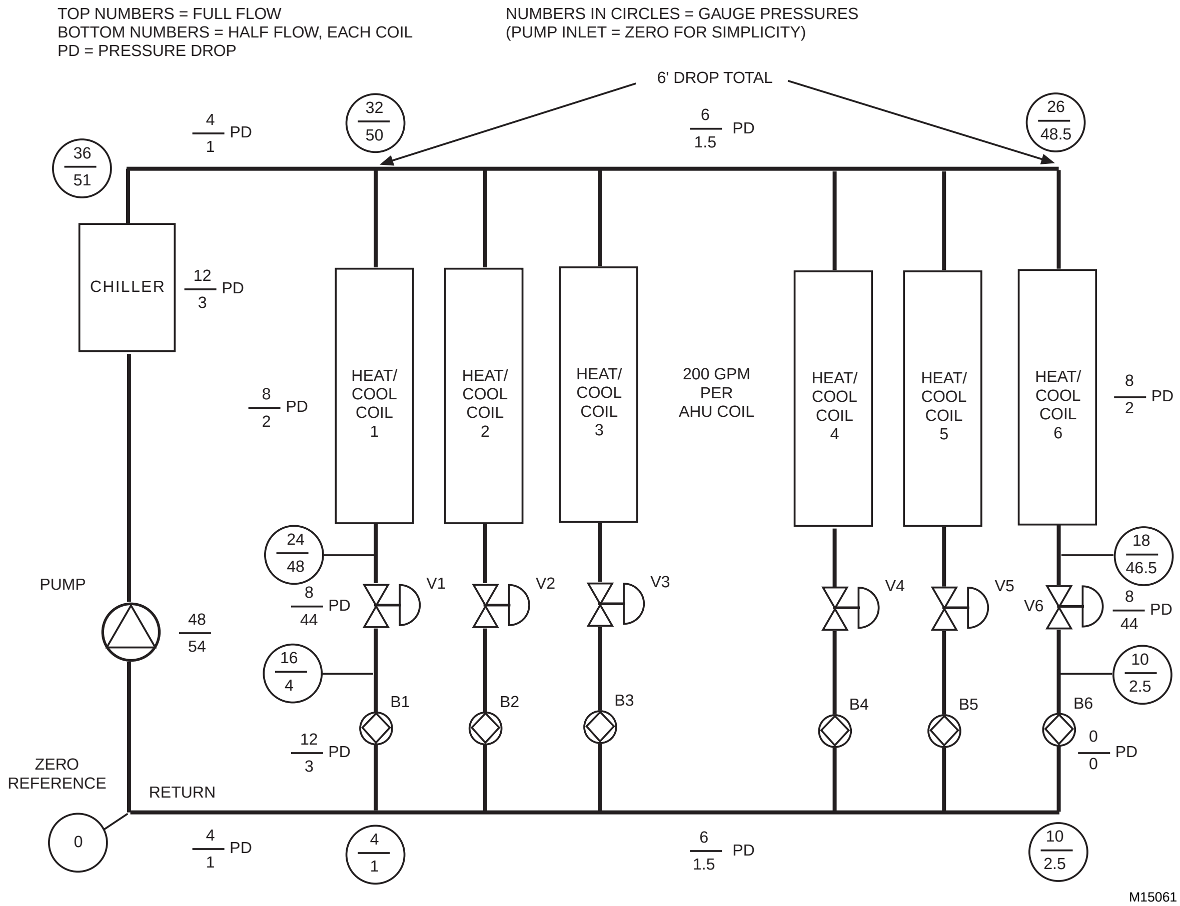


Fig. 74. Single Pump, Pressure Bypass, Direct Return at Full and Half Flow.

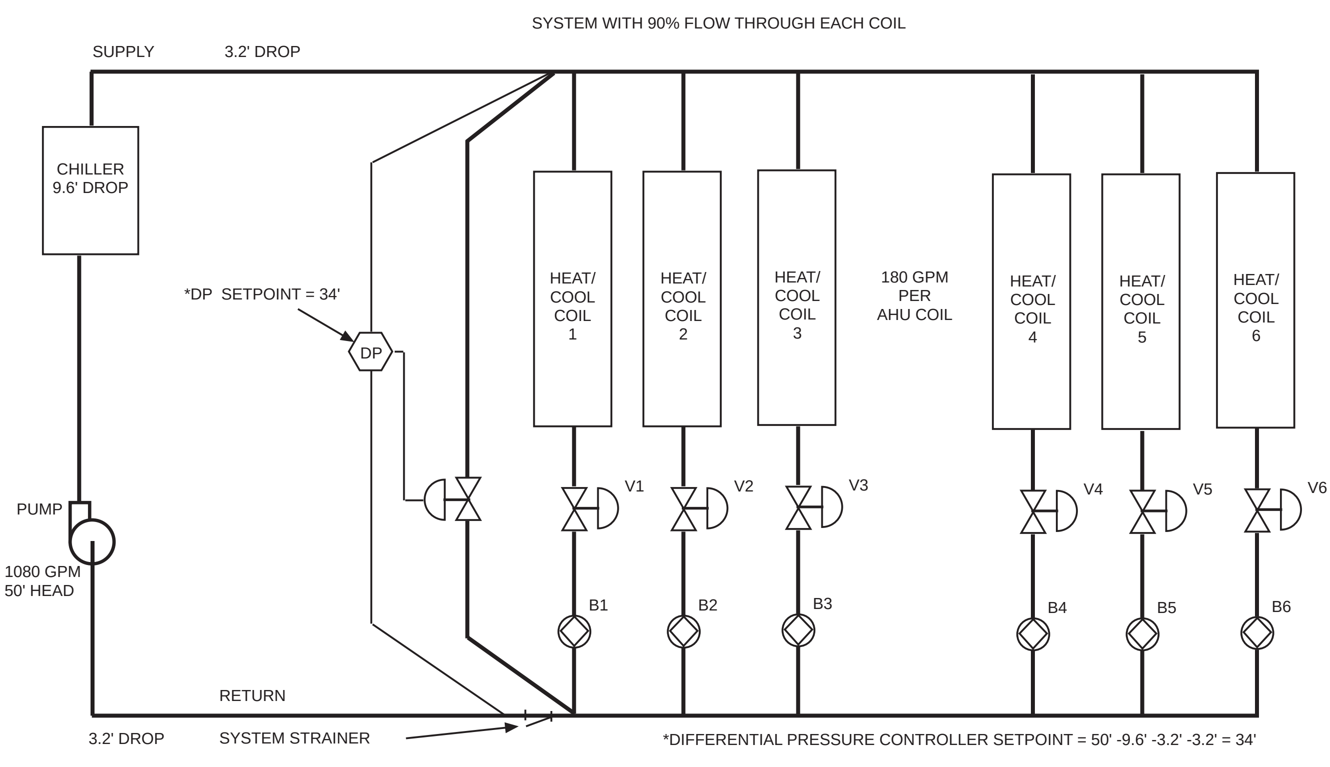


Fig. 75. Single Pump, Pressure Bypass System at 90 percent Flow.

The greatest change at the sensor provides the most tolerant and robust control. For this reason the sensor is located, not across the chiller with a setpoint of 9.6 ft, but across Coil 1 where the greatest differential pressure change exists (28 ft at design to 34 ft at 90 percent of design).

NOTE: With the controller pickups in these locations, it does not matter where in the system the load is located, what value it has, or whether it is symmetrical or non-symmetrical.

Also note that the Load 1 balancing valve takes a full flow 12 foot drop and even with the pressure bypass valve, the control valves will be subjected to drops of near 34 ft for light loads. For larger and more extended systems, both of these values must be considered when evaluating reverse return and control valve high differential pressure solutions.

These values lead to the conclusion that a differential pressure bypass solution may satisfy the light-load flow through a chiller, but may not adequately relieve control valve light-load differential pressures on larger systems. Also be aware that pressure drop changes due to scaled chiller tubes effect the bypass valve operation.

Single Pump, Pressure Bypass, Reverse Return

A reverse return system analysis equivalent to the direct return analysis of Figure 74 would show that at half flow and no bypass, the pump head still rises to 54 ft and the control valve drop still rises to 44 ft.

For 90 percent flow control, the pump still operates at a 50 foot head with the sensor located as far away from the pump as practical to take advantage of as many friction drop changes as possible.

The preferred sensor location (Fig. 76) is DP-1 if point A is near Load 1. Sensor location DP-2 is second best and again if point B is near Load 1. If location DP-3 is selected by default, with a setpoint of 40.4 ft as compared with the DP-3 full flow differential pressure of 36 ft. This small 4.4 foot change requires a higher quality sensor and more frequent calibration checks than for locations DP-1 and DP-2. Locate the pressure bypass valve and sensor as far from the chiller/pump as possible , but no closer than DP-3.

In all cases it is still preferable to locate the system strainer outside the control loop as shown in Figure 75.

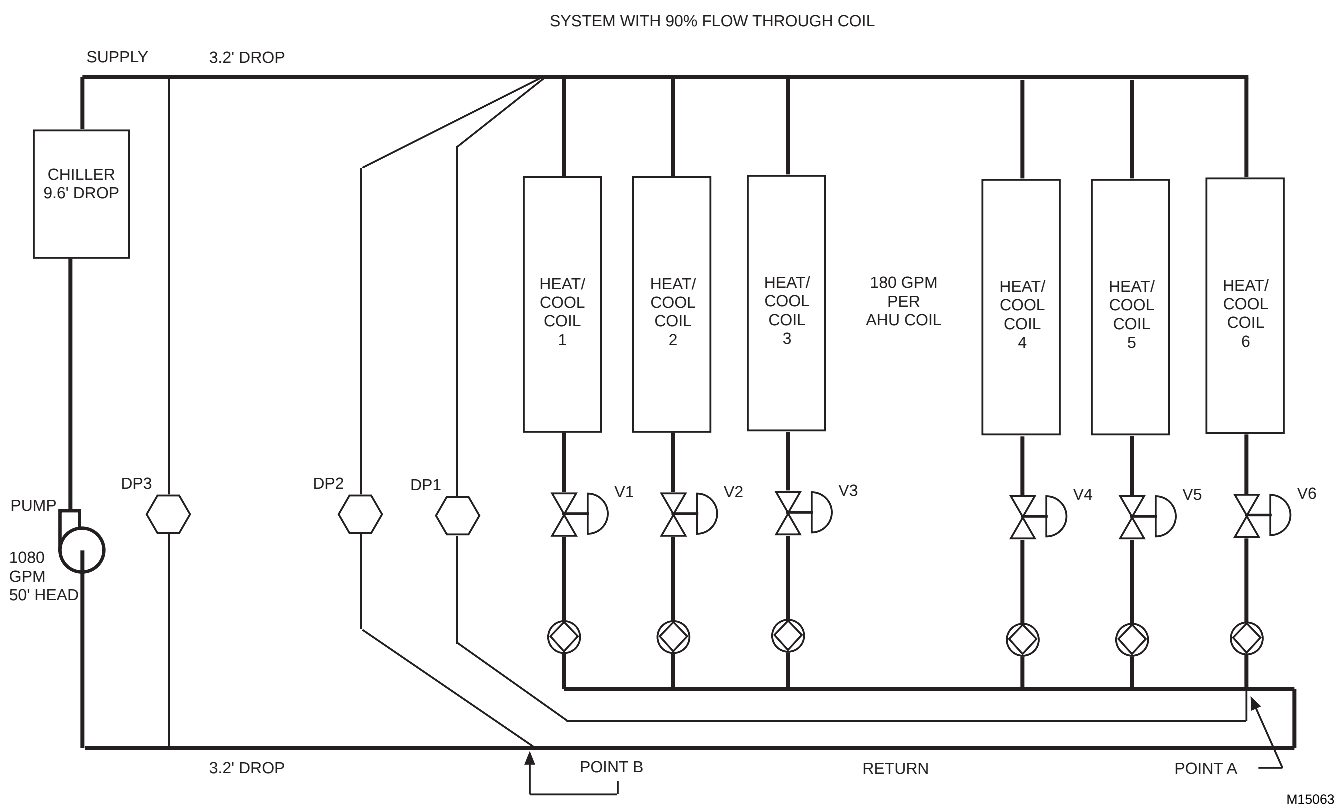


Fig. 76. Single Pump, Pressure Bypass, Reverse Return at 90 percent Flow.

Dual Pumps, Dual Chillers, Pressure Bypass, 90 Percent Chiller Flow, Direct Return

Dual chiller pressure bypass systems are popular because if a chiller, tower, or pump fails, there is part load redundancy and better part load efficiency.

In Figure 77, the bypass sensor and valve is in the same location as the single chiller system (Fig. 75). The valve is sized for approximately 500 gpm.

The differential pressure setpoint is the same as for the single pump/chiller system (34 ft) when both pumps are running. When only one pump/chiller is operating, the piping between the pump/chiller and differential pressure sensor carry a minimum of 540 gpm (90 percent flow for one chiller) and the piping friction drop (head) falls from 4 ft to 0.8 ft, thus:

$$4\left(\frac{540}{1200}\right)^2 = h_2 = 0.8 \text{ ft.}$$

With a single pump/chiller operating, the differential pressure setpoint for 90 percent flow is 38.8 ft.

Setpoint = 50 ft (pump drop) – 9.6 (chiller drop) – 0.8 (piping head) – 0.8 (piping drop) = 38.8 ft.

This is up from the 34 ft with both chillers operating.

With digital controls, the differential pressure setpoint offset adjustment when only one chiller/pump is operating is handled by a software routine (dual chiller/pump setpoint plus 4.8 ft.) invoked anytime only one pump and one chiller are operating.

One method using pneumatic controls uses two pressure controllers with separate setpoints. The primary controller is set at 38.8 ft with the secondary controller set at 34 foot setpoint and configured as a low limit device during periods of single pump/chiller operation. During periods of dual pump/chiller operation, the 34 foot setpoint controller is used alone.

AHU Valve High Differential Pressure Control

As noted in the discussion of Figure 75, the differential pressure controlled bypass valve in constant speed pumping systems is adequate to maintain a high flow through chillers, but does little to prevent high differential pressures across the AHU load control valves. In the previous example the load control valve differential pressure varied from 8 ft at design load to 38.8 ft at the single chiller, light-load mode of operation.

If it is expected that the light-load differential pressures will exceed the load control valves close-off rating, locate a throttling valve in the common load piping (valve V8 in Figure 78) to reduce the load pressures while still allowing adequate pressure to maintain chiller flows. Select either a double-seated or balanced cage type valve for the high differential pressure.

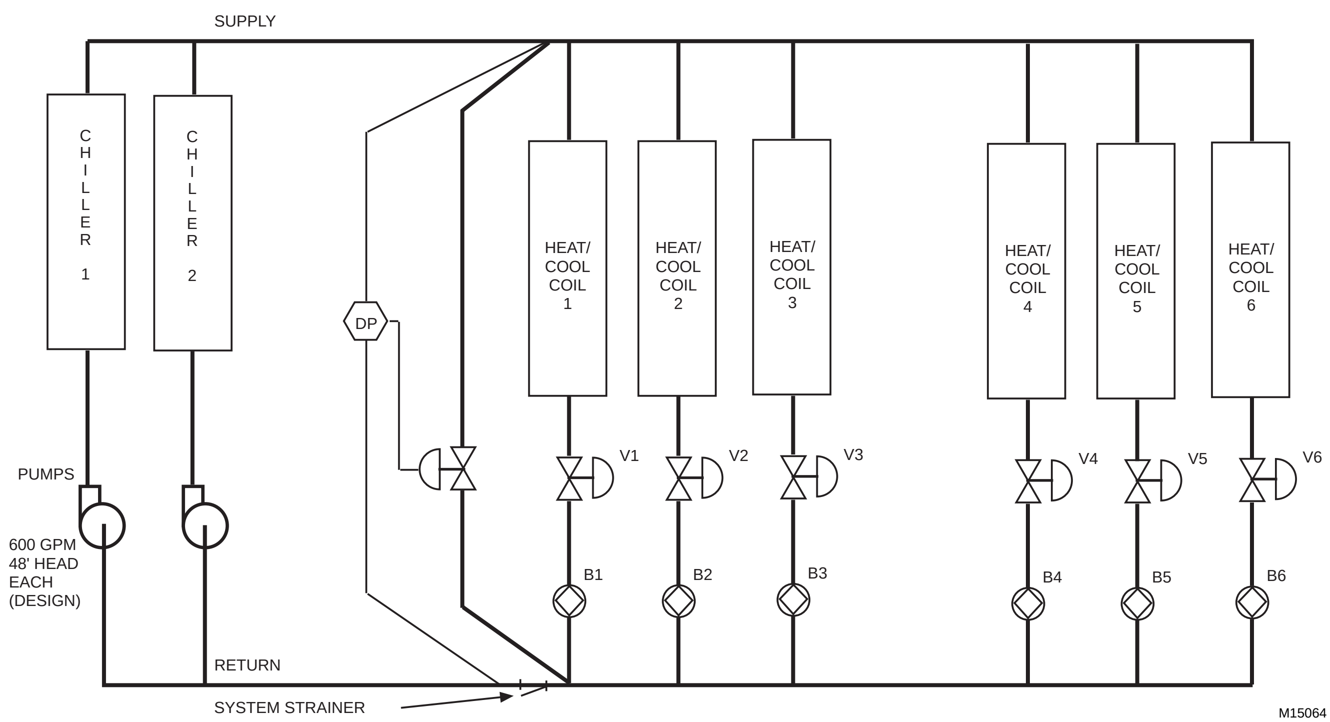


Fig. 77. Dual Pumps, Dual Chillers—Pressure Bypass

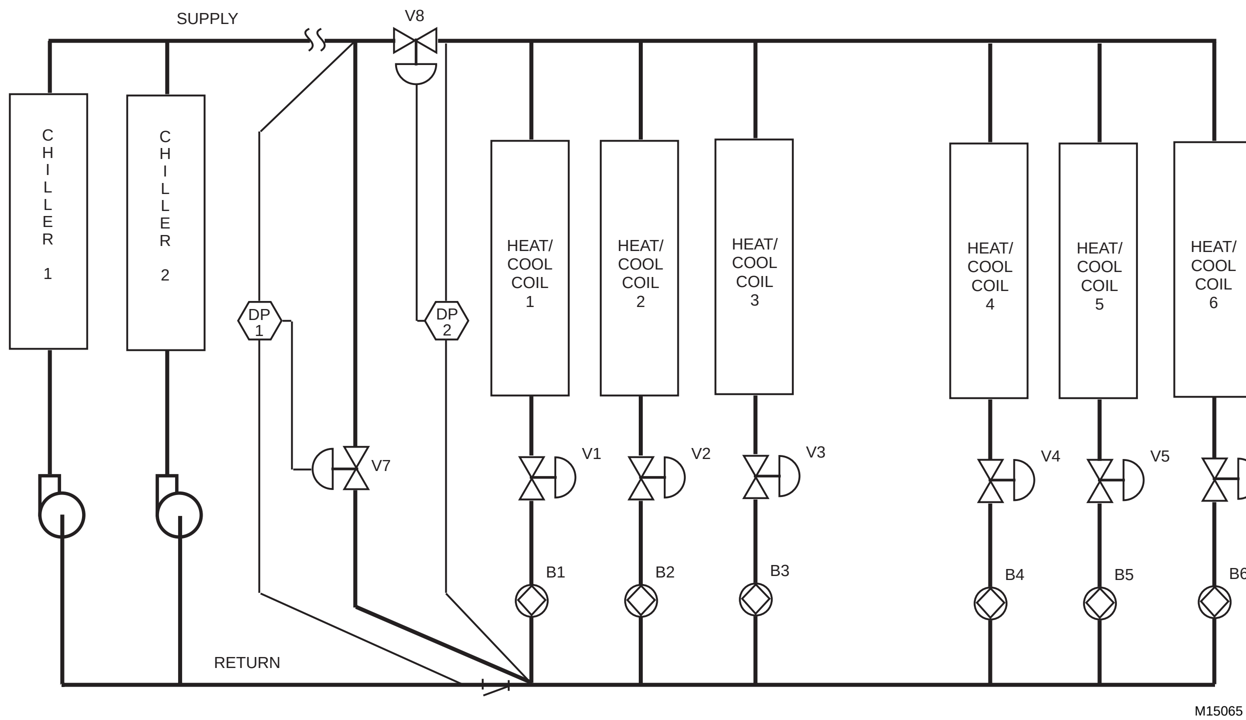


Fig. 78. High AHU Valve Differential Pressure Control

In these examples, the design differential pressure across Load 1 is 28 ft (Fig. 74), and pressure bypassing occurs at 34 ft. The maximum setpoint for DP-2 (Fig 78) should be about 29 or 30 ft. This initial setpoint is then slowly lowered based upon the percent-open values of load control valves V1 through V6, to a minimum value of 10 to 12 ft.

Specification:

Anytime either chiller pump starts, DP-2 shall be enabled to control pressure reduction valve V8 at an initial setpoint of 20 ft. Anytime any load control valve is greater than 95 percent open, the DP-2 setpoint shall be incremented at the rate of 0.5 ft every 2.0 minutes up to a maximum of 30 ft. Anytime all control valves are less than 80 percent open the DP-2 setpoint shall be decremented at the same rate to a minimum value of 12 ft. All values shall be user adjustable.

Specification Discussion:

The value of 2.0 minutes in the specification assumes that valves V1 through V6 are controlled from discharge air temperature and should recover in less than 2 minutes from V8-caused changes. If V1 through V6 were controlled from space temperature directly, the time rate for V8 adjustments may need to be extended to 12 to 15 minutes to allow incremental space temperature control recovery from the flow reductions brought about by V8. Valve V8 is normally open and line size to minimize its pressure drop during full load operation. This valve is applicable for direct or reverse return piping configurations where significant piping friction losses migrate out to the control valves upon low flow conditions.

In reverse return systems, locate Valve V8 in the main line piping just before the AHU 1 take-off and the DP-2 sensor across AHU 1 set for approximately 19 ft, this should be adequate for any variation in uneven loading. Allow 19 ft for close-off and good control. Resetting the DP-2 setpoint lower is unnecessary in this constant-speed pumping reverse return example.

VARIABLE SPEED PUMP CONTROL

Decoupled Variable Speed Pump Control, Direct Return

Similar to Figure 74, Figure 79 decouples the loads from the source pumps and heat exchangers and uses of a variable speed pump to provide significant energy savings at reduced loads and simpler control. Variable speed pump control matches pump speed to system flow demands. If source devices perform well with variable flow, the primary pumps may be replaced with variable speed pumps and controlled similarly to the decoupled example.

NOTE: The variable speed pump head is only 36 ft since the primary pumps account for the 12 ft chiller head.

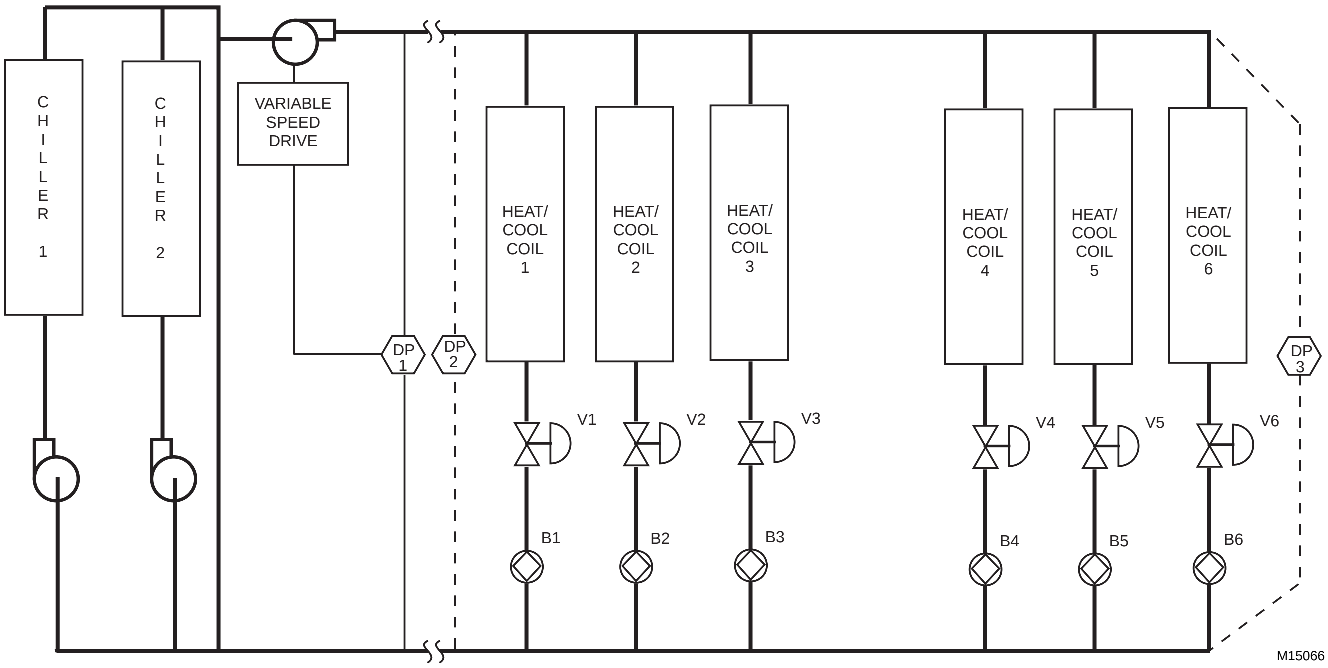


Fig. 79. Variable Speed Pump Control

AHU 1 requires a 28 foot differential pressure (8 feet for the coil, 8 feet for the valve, and 12 feet for the balancing valve), for full flow. If 28 feet is available at AHU 1, all other AHUs will have at least the required design differential pressure. Controlling the pump with a sensor positioned as shown for DP-2 set for 28 feet is acceptable.

Locating the sensor (DP-3) at AHU 6 and set for the 16 feet required by AHU 6 will not work when only AHU 1 is operating. With these conditions DP-3 maintains a maximum drop of only 16 feet across AHU 1 which needs a 28 ft differential pressure because of the balancing valve. If the sensor is positioned at AHU 6, the setpoint must still be 28 feet if the system is to operate satisfactorily with non-symmetrical loading.

DP-1 located at the variable speed drive (VSD) is the most convenient place. It requires the DP-2 setpoint plus the friction losses between the pump and AHU 1.

Figure 80 shows the operating curve of the system with the differential pressure sensor located in the DP-2 position and set for 28 feet. With each AHU at one-third flow, the speed is 1412 RPM, which produces a 28 foot differential pressure at AHU 1 with 400 gpm system flow. When the coils are equally loaded at one-third flow, each control valve takes a 26 foot drop. In this configuration the pump never operates much below the 1400 rpm speed because of the 28 ft head setpoint.

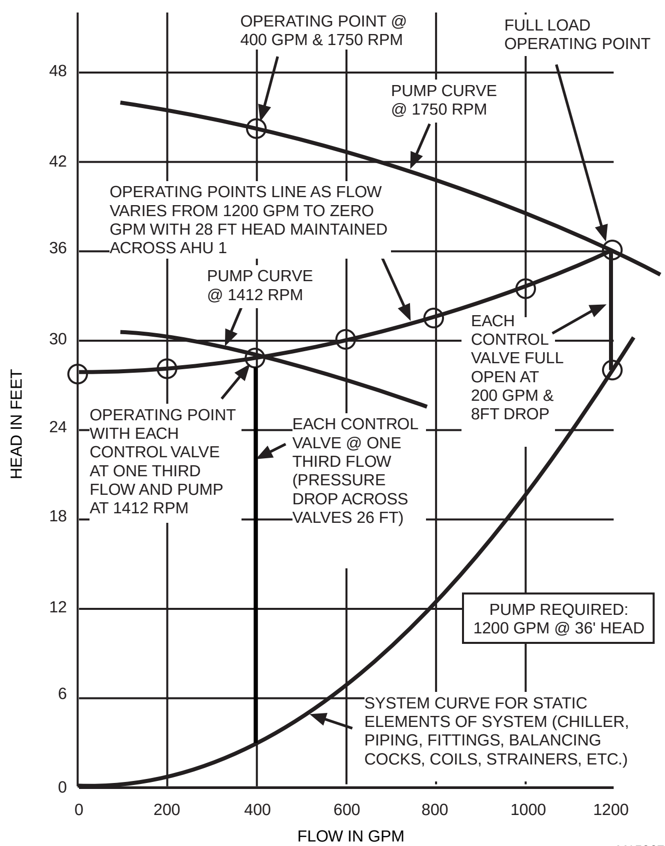


Fig. 80. Fixed Setpoint with PI, Variable Speed Pumping Control

No pump head control example, so far, takes advantage of both the variable speed pump and a digital control system. The digital control system VSD control algorithm adjusts the differential pressure setpoint based on the demands of all the valves (Fig. 81) and all loads are satisfied with significant savings over any of the three fixed setpoint options. Since, when using valve position load reset there is no difference in performance between the three locations, DP-1 is preferred because of initial cost. Valve position load reset provides adequate control performance whether the sensor is only proportional or is only a static pressure sensor as compared to a differential pressure sensor.

Specification:

Anytime any AHU chilled water valve is greater than 15% open for greater than one minute, the secondary pump shall be started under EPID control at 20% speed and with a ramp duration of 120 seconds. The pump VSD shall be controlled by a differential pressure sensor located between the supply line leaving the plant room, as far from the pump as practical to avoid hydronic noise that may be present at the immediate pump discharge, and the system return line. At start-up the differential pressure setpoint shall be 30 ft (See Note 1). Anytime any load control valve is greater than 95% open, the differential pressure setpoint shall be incremented at the rate of 0.5 ft every minute up to a maximum value of 38 ft. Anytime all load control valves are less than 80% open, the differential pressure setpoint shall be decremented at the same rate down to a minimum of 7 ft. After 12 minutes, the increment/decrement rate shall be changed from one minute to three minutes (See Note 2). All values shall be user adjustable.

NOTES:
1. From Figure 81, pump head is 36 feet if all AHUs require full flow, therefore, the 30 foot value is an arbitrary compromise.
2. This relaxes the control demands for smooth stability after the response to the initial load.

Figure 81 shows the ideal performance of the load reset setpoint control with each AHU demanding one-third flow. All control valves are full open and the differential pressure adjusted to produce a speed of 525 RPM. If the coil loading is non-symmetrical to the point that AHUs 1 and 2 are fully loaded while the others are off, the operating point for 1/3 system flow is the same as shown in Figure 80 for 1/3 system flow, since AHU 1 requires a differential pressure of 28 ft for full flow.

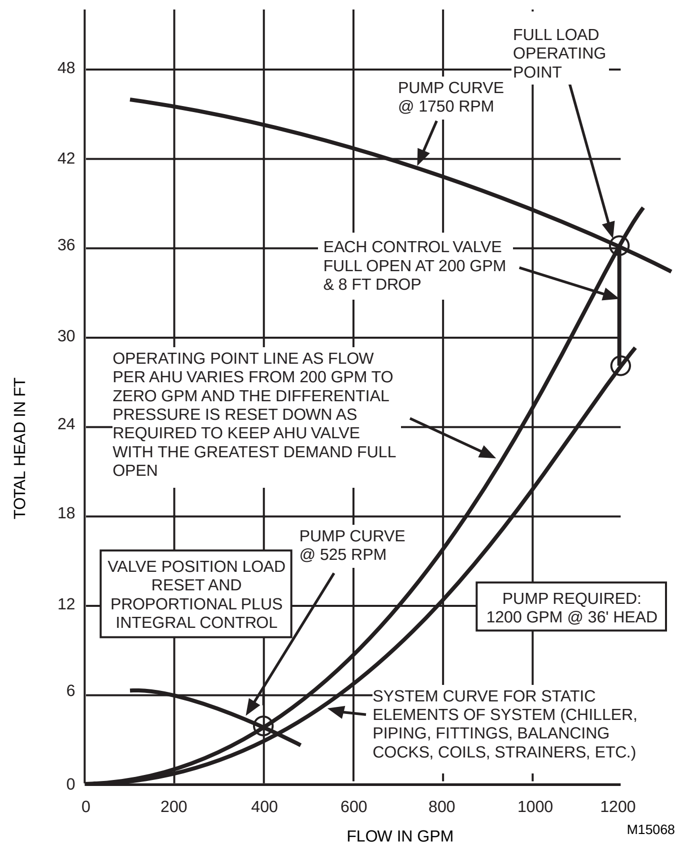


Fig. 81. Variable Setpoint, Variable Speed Pumping Control (Ideal Curve).

Pump Speed Valve Position Load Reset

A pump speed valve position load reset program with over 20 valves can become cumbersome. Also, if any one valve, for whatever reason, stays open most of the time, then the load reset program becomes ineffective. Figure 82 shows an example of the valve position load reset program concept applied to a multibuilding facility with varying differential pressures entering each building, due to varying distances from the pumping plant. The example address two issues, differential pressure control within each building to relieve control valves from extremely high differential pressures and pump speed load reset.

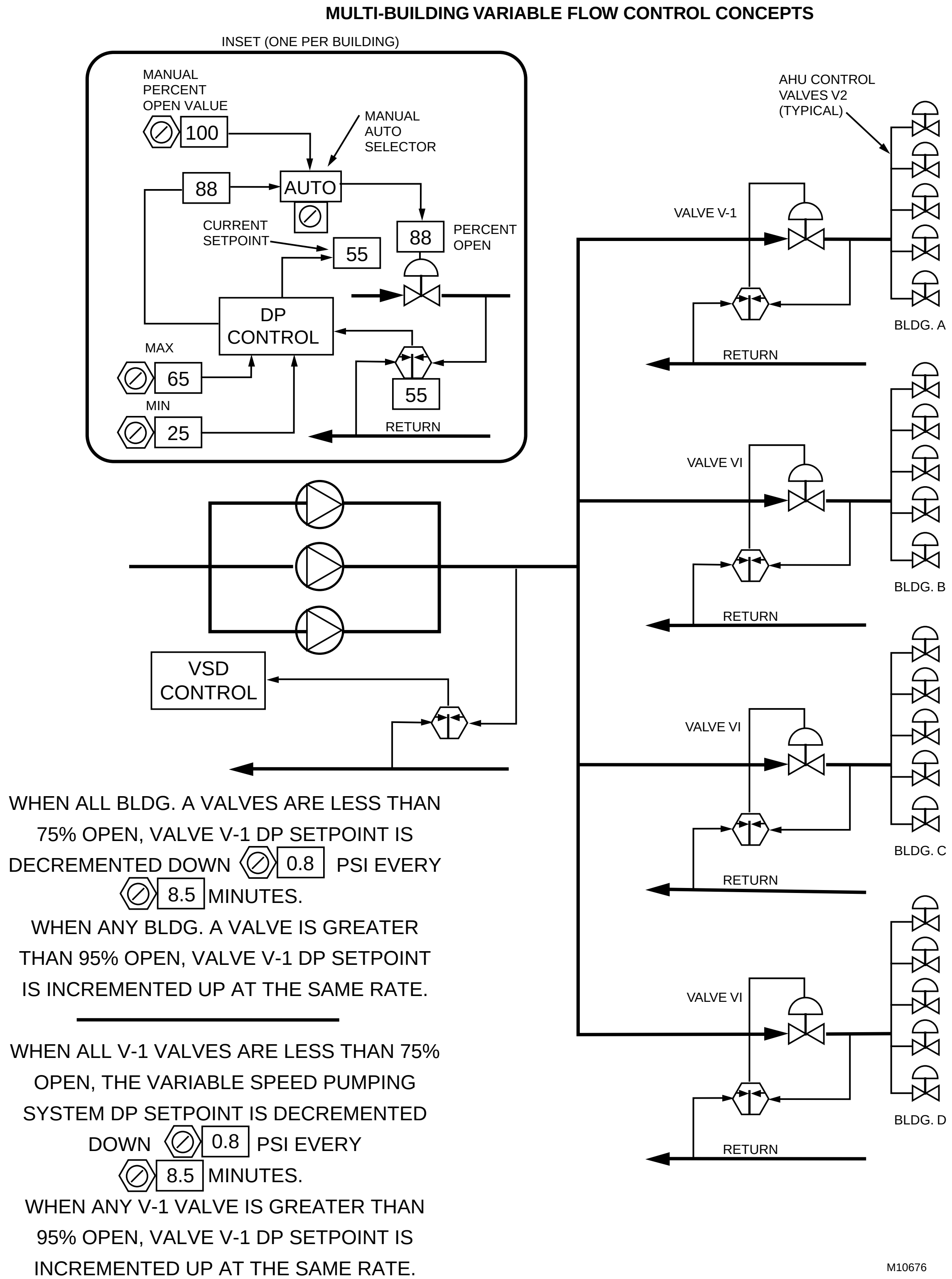


Fig. 82. Multibuilding, Variable Speed Pumping Control.

Each building is provided with a “choke” valve (V-1), and a load reset loop to maintain water pressure within the building, such that the most demanding AHU valve is always between 80 and 95 percent open. Each building requires a Valve V-1 Control Detail (inset) and a dynamic sequence description of the program. Each control detail includes minimum and maximum differential pressure setpoints, a software MANUAL - AUTO selector, and a setpoint value for the manual position. Ideally the control detail along with the current percent open for each valve within the building is provided graphically for each building.

NOTE: If the choke valve (V-1) is omitted on the most remote building, the choke valves need not be considered for pump sizing.

Pump speed is reset to keep the most demanding Building Valve V-1 between 80 and 95 percent open.

Adjust balancing valves in buildings close to the pumping plant with the choke valve in control, so that high balancing valve differential pressures are not set to negate low load value of the choke valve reset concept.

This concept can be combined with tertiary pumps in remote buildings to control the building differential pressure and choke valves in closer buildings utilizing central pumping.

Balancing Valve Considerations

BALANCING VALVE EFFECTS

Figure 80 assumes that all coils are equally loaded (1/3 flow), and that all friction losses are 1/9th [$(1/3)^2$] of their full load value. In symmetrical loading and with valve position reset, the balancing valves have no adverse effect. However, if at 400 gpm total flow, AHU 1 and 2 are operating at full flow (200 gpm) and all others are off, the required differential pressure across AHU 1 is 28 ft, 12 ft of which is wasted on the balancing valve.

BALANCING VALVE ELIMINATION

Elimination of all balancing valves allows the valve position load reset control strategy to satisfy the non-symmetrical loading described in BALANCING VALVE EFFECTS, by producing only 16 ft differential pressure at AHU 2 (slightly higher at AHU 1) and save significant pumping energy during most periods of non-symmetrical operation. Before eliminating balancing valves consider:

1. Load coil temperature control setpoints must be strictly maintained. In the example, lowering AHU 1 leaving air temperature 5 degrees below the design temperature causes AHU 1 water flow loop to draw significantly more than design flow because it is nearer the pump where the differential pressure is higher. This will slightly starve the other loads.
2. Cool-down periods for other than AHUs 1 and 2 will be extended. With all valves full open, until AHUs 1 and 2 are satisfied, the other AHUs will be starved.
3. Industrial valves may be required to maintain acceptable controllability. At properly controlled full load design conditions, a 28 ft differential pressure drop appears across the AHU 1 control valve. This is the 16 ft differential pressure required at AHU 6 plus the 12 foot piping drop from AHU 1 to AHU 6. With the high differential pressure, some piping configurations will require an industrial valve.

Elimination of or fully open balancing valves might work well in a continuously operating facility with operators who understand disciplined setpoint and self-balancing concepts.

If eliminating balancing valves in a fixed setpoint scheme, position the DP sensor across AHU 6, with a setpoint of 16 ft. If AHU 6 is very remote from the VSD pump controller, it is recommended to put an additional differential pressure sensor at the pump with a maximum set point of 36 ft then reset the setpoint down as required to prevent the AHU 6 differential pressure from exceeding 16 ft. Use of a DDC PID input and output in separate controllers is not recommended because of the communications system reliability.

If balancing valves are removed in a valve position load reset scheme, use the single differential pressure sensor at the pump with a max differential pressure setpoint of 36 ft.

Differential Pressure Sensor Location Summary

Refer to Figure 79, in summary:

1. If valve position load reset is employed, the DP sensor may be located in the pump room for simplicity (position DP-1)
2. If valve position load reset is not employed and balancing valves are provided, the DP sensor should be located at AHU-1 (position DP 2) and set for 28 ft
3. If valve position load reset is not employed and balancing valves are not provided, the DP sensor should be located at AHU-6 (position DP 3) and set for 16 ft.
4. If the sensor is located at AHU 6, resetting the setpoint of a sensor located at position DP 1 is recommended as noted in BALANCING VALVE CONSIDERATIONS.

Pump Minimum Flow Control

Pumps require a minimum flow to dissipate the heat generated by the pump impeller. A bypass around the pump located out in the system provides the required flow and prevents the heat from building up in the pump. The minimum flow is calculated from the equation:

$$\text{Minimum flow (gpm)} = \frac{\text{BHP x 0.746 kW/bhp x 3,412 Btu/kW}}{\text{8.33 lb/gal x 60 min/hr x } \Delta T}$$

Where:

BHP = Horsepower

ΔT = Low flow water temperature rise across the pump

The minimum flow for 10 horsepower with a 10 degree ΔT (attributed to the pump heating the water) is only 5 gpm. The bypass may be fixed or if there is a remote AHU in the 12 to 25 gpm size, using a three-way valve on that AHU with the bypass in the bypass leg of the three-way valve prevents bypass water from flowing during full-load periods. Another option is an automatic bypass valve programmed to open anytime all AHU control valves are less than 10% open.

Decoupled Variable Speed Pump Control, Reverse Return

In these examples, the design differential pressure for all AHUs is 16 ft with reverse return, since the balancing-valve drop becomes negligible. If approximately 18 ft is maintained at AHU 1, design flow is available to all AHUs during any non-symmetrical flow condition. With a reverse return system locate the differential pressure sensor across AHU 1 with a setpoint of 18 ft. Advantages of valve position load reset are much less with reverse return systems, however, resetting the setpoint from 18 ft to 8 or 10 ft based upon load is worthwhile on large systems.

HOT WATER DISTRIBUTION SYSTEMS

GENERAL

Advantages of hot water compared to steam for heat distribution within commercial buildings are:

- Water temperature can be varied to suit variable loads.
- Hot water boilers typically do not require a licensed operating engineer.
- Heat loss from piping is less than steam systems because fluid temperatures are usually lower.
- Temperature control is easier and more stable because water temperature is matched to the load.
- Fewer piping accessories to maintain than steam systems.
- Reduced air stratification at light loads with reset hot water.

The control objectives for a hot water distribution system providing space heating are:

1. Provide adequate hot water flow to all heating units.
2. Maintain a stable pressure difference between supply and return mains at all load conditions.
3. Match hot water temperature to the heating load based on outdoor air temperature and/or occupancy.
   a. Avoid heat exchanger flow velocities dropping into the laminar range.
   b. Keep water velocities up to prevent freezing.
   c. Reduce mixed air temperature stratification.
      – No stratified cold air to trip low temperature controllers or cause freeze-up.
      – Minimizes stratified air from having adverse effects on coil discharge temperature sensing.
      – Minimizes possibilities of hot air flowing out some duct outlets and cold air flowing out others.

Refer to METHODS OF CONTROLLING DISTRIBUTION SYSTEMS for additional information on control objectives.

HOT WATER CONVERTERS

A hot water converter uses either hot water or steam to raise the temperature of heating system water to a temperature which satisfies the heating requirements of the system. The most widely used hot water converters use steam.

Steam is often supplied to remote buildings or mechanical rooms from a central plant. Within the buildings or zones, hot water is often preferred because of its ease of control. Steam to water converters are used for these applications.

A converter consists of a shell and tubes. The water to be heated passes through the tubes. Steam or hot water circulates in the shell around the tubes. Heat transfers from the steam or hot water in the shell to the heating system water in the tubes.

If the pressure on the water in a converter drops below the vaporization pressure for the temperature of the water, the water can flash to steam. Pumps are usually located on the return side of the converter to provide a positive pressure and minimize flashing.

Figures 83 and 84 show commonly used controls for hot water converters. See HOT WATER RESET for converter control.

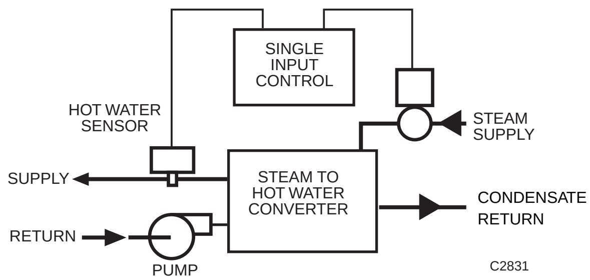


Fig. 83. Constant Temperature Hot Water Converter Control.

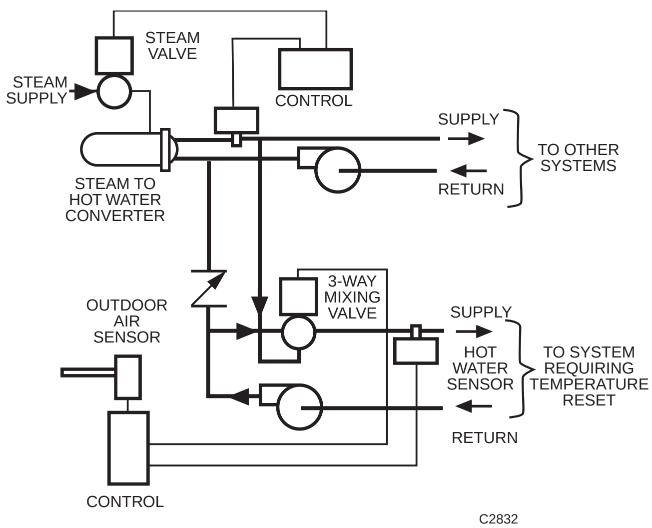


Fig. 84. Hot Water Converter Control with Constant Temperature and Reset Zones.

HOT WATER PIPING ARRANGEMENTS

General

Hot water piping systems use one- or two-pipes to deliver the water to the terminal units. The one-pipe system shown in Figure 85 uses flow diverting fittings to scoop water out of the single loop into each radiator or baseboard radiation section. Another fitting returns the water to the main where it is mixed with the supply water. The next unit will be supplied with slightly lower temperature water. One-pipe systems are typically used in small buildings with only one or two zones.

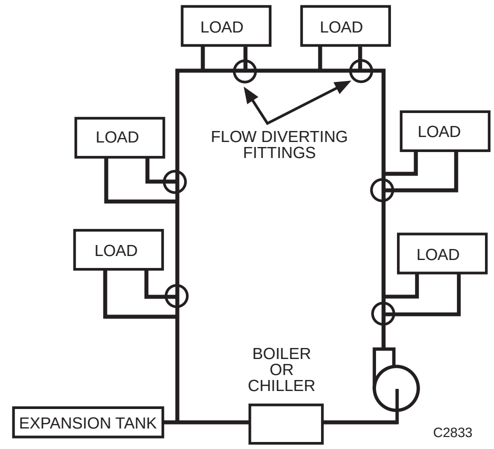


Fig. 85. One-Pipe System.

Figure 86 shows a two-pipe reverse-return system. The length of supply and return piping to each terminal unit is the same, making the system self-balancing. The reverse-return is used in large or small installations.

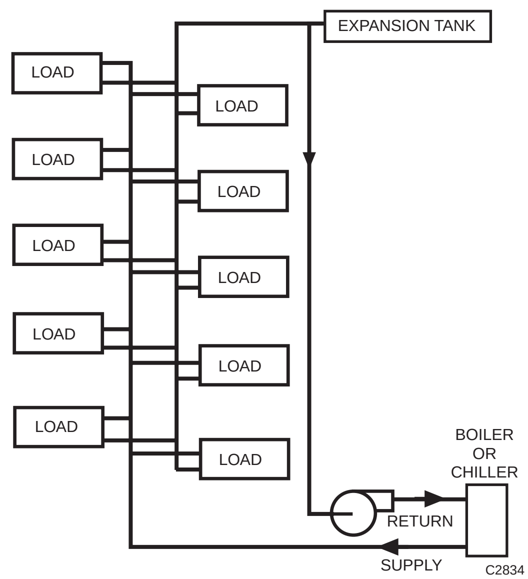


Fig. 86. Two-Pipe Reverse-Return System.

Figure 87 shows a two-pipe direct-return system. The direct-return system is used on small systems. The shorter return piping reduces installation cost but each load loop must be symmetrical.

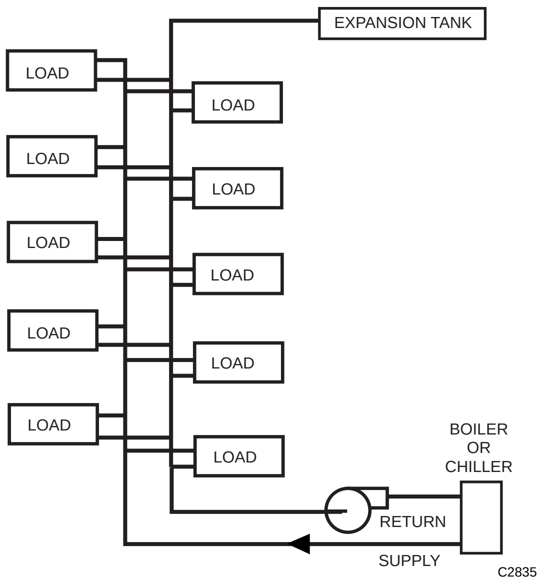


Fig. 87. Two-Pipe Direct-Return System.

Primary-Secondary Pumping

Primary-secondary pumping allows water temperature and on/off times of each secondary zone to be independently controlled. The main supply pump uses less power in this arrangement since it is sized to handle only main pressure losses. Secondary pumps handle only the zone piping losses.

Figure 88 shows common configurations for primary-secondary pumping. The three-way valve configuration provides more positive control than the two-way valve.

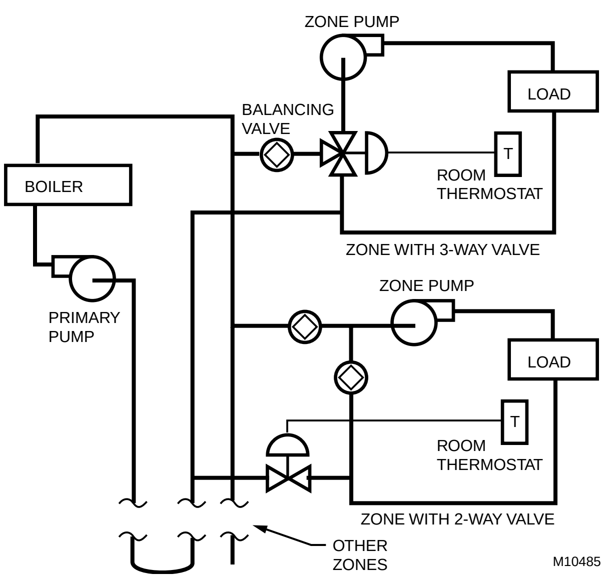


Fig. 88. Piping for Primary-Secondary Pumping.

A converter (heat exchanger) shown in Figure 89 can be used in large or high rise buildings to reduce the zone temperature/pressure requirements from those of the mains.

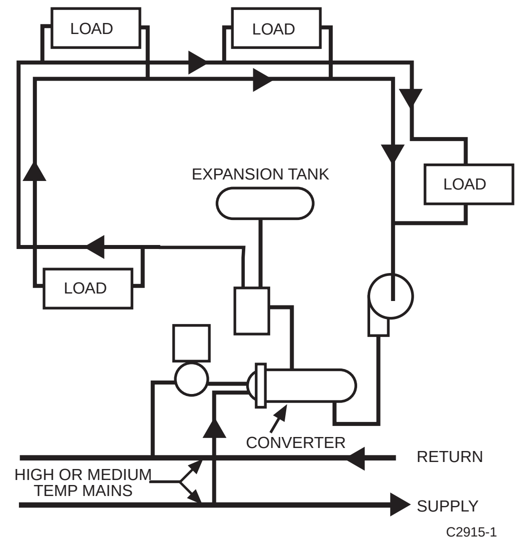


Fig. 89. Converter Used to Supply Zones from a Larger System.

CONTROL OF HOT WATER SYSTEMS

General

Heating (terminal) units used in hot water heating systems are:

— Radiant Panels
— Radiators or finned tubes
— Forced air heating coils

Control of heat output from a heating unit can use one or a combination of the following methods:

— On-off control, by starting and stopping a pump or opening and closing a valve
— Modulating flow control
— Supply water temperature control

Modulating Flow Control

Varying water flow to finned tube radiation or a heating coil, each supplied with constant temperature hot water, is shown graphically in Figures 90 and 91.

In both cases, reducing flow 75 percent reduces the heat output only 30 percent because as flow is reduced, more heat is extracted from each gallon of water. At low flows leaving water temperature decreases sharply (Fig. 91). At light loads modulating flow control by itself is not the best means of controlling heat output.

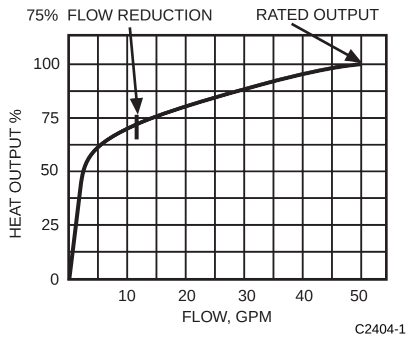


Fig. 90. Effect of Flow Control on Finned Tube Radiator Heat Output.

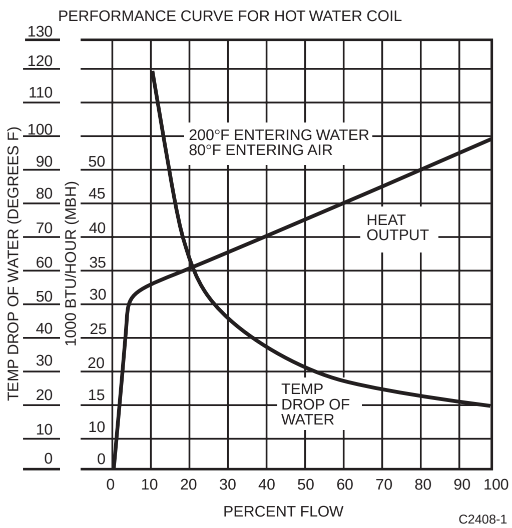


Fig. 91. Effect of Flow Control on Hot Water Coil Heat Output.

Supply Hot Water Temperature Control

If supply water temperature is varied (reset) in response to a change in heating load, heat output varies almost linearly (Fig. 92). This mode of control appears ideal, except that it is impractical to provide a different hot water temperature to each heating coil or piece of radiation in a building. Varying supply water temperature as a function of outdoor temperature provides a good compromise.

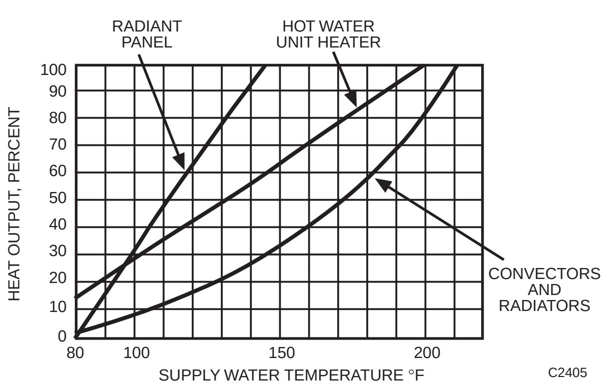


Fig. 92. Supply Water Temperature vs Heat Output at Constant Flow.

Supply Hot Water Temperature Control With Flow Control

Combining flow control with supply water temperature reset from outdoor air temperature or any other measurement of load results in effective control.

Figure 93 shows output of a typical air heating coil with flow and supply water temperature control. When 110F water is supplied during light load conditions, the maximum air temperature rise through the coil is only 10F. With the proper water temperature reset schedule, the control valve stays in the 30 to 100 percent open range where flow control output is linear. Thus, radical changes from desired output do not occur. This is true for radiators, finned tubes, and reheat coils.

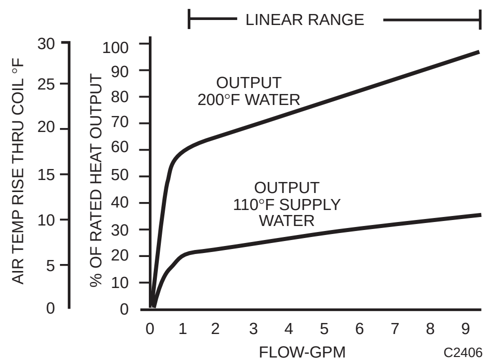


Fig. 93. Heat Output of Hot Water Coil at Two Supply Temperatures.

Hot Water Reset

As previously discussed, reset of hot water supply temperature is the most effective method of controlling Btu output of a water supplied heating coil, panel, radiator, or convector. Btu output of most water supplied heating is relatively linear with respect to supply water temperature. Water temperature reset can be provided by a steam to hot water converter, a three-way valve, or boiler water temperature reset. Reset of hot water using a steam to hot water converter is discussed in this section.

HOT WATER CONVERTER

The supply water temperature to the radiant panel (Fig. 94) is reset by the controller modulating a valve in the converter steam supply. The controller uses the temperatures at the outdoor air sensor and the supply water sensor to control the steam supply valve. The valve is positioned to maintain the converter discharge water temperature as determined by the reset schedule and outdoor air temperature. A typical reset schedule is shown in Table 5.

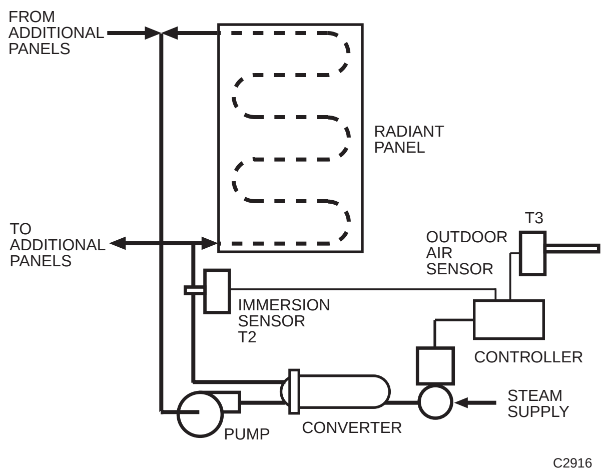


Fig. 94. Radiant Panel with Scheduled Water Temperature Using Converter.

Table 5. Typical Hot Water Reset Schedule.

OA Temperature	Water Temperature
70F	80F
35F	100F
0F	120F

Before determining a digital converter control scheme, two issues must be explored; the nature of the hot water load and the steam source.

If the load is affected by the OA temperature, then an appropriate reset schedule should be used. If the load has a step change, such as going from a warm-up mode (no sun, no people, no lights or internal heat load) in the early morning; to the occupied mode (lights, people, office equipment, sun) at a specific time, then the reset schedule should also shift from a high temperature reset schedule hot water during the warm-up mode, to a low temperature reset schedule as the occupied loads come into effect.

If the heating load includes 100 percent OA preheat coils, special care must be given to assuring high water flow rates through the coil when the OA temperature is below freezing. Preheat coil valve-position reset of hot water temperature could be applied such that anytime the OA temperature is below freezing, the hot water temperature setpoint can be lowered as required to keep the most demanding preheat coil hot water valve greater than 85 percent open, and raised incrementally anytime any valve is greater than 95 percent open. This can be accomplished via applying (incrementing) a multiplier of 0.75 to 1.25 to the OA reset schedule-derived setpoint. Setpoint calculations should complement the objectives of the AHU optimum start programs.

Another digital control function is steam valve demand limiting. Water systems are usually balanced such that if all water valves are full open, design flow is delivered to all load coils (unless diversity is used). Steam systems are usually not balanced. From size-to-size, control valve capacities typically increase about 35 percent, for example 1-1/2 in. valve, Cv = 25; to 2 in. valve, Cv = 40. When steam valves are sized and valve selections are rounded up to the next larger size, on the average they are about 17 percent oversized (and worse case more than 30 percent oversized). If a valve with a Cv of 40 is furnished for a load requiring a Cv of 30 and the oversizing has a negative impact on the steam source, software can ”destroke” the control valve. A destroked valve has the maximum stroke reduced, such that as the load varies from 0 to 100 percent, the oversized valve is positioned from 0 to 75 percent open. Destroked valves may be specified to display their actual stroke percent open or to display 0 to 100 percent open as the valve varies from 0 to the destroked maximum position (destroking is normally transparent to HVAC system operators).

When hot water converter loads are scheduled (shut down at night and started in the morning), the morning start-up is usually greater than 100 percent load every morning because the hot water has cooled down such that the steam valve starts full open. In mild weather, when the actual load may be only 15 to 20 percent, this momentary 100 percent plus start-up demand causes boilers to surge up to full capacity (or in a multiple boiler system to stage multiple boilers on) only to unload down to the operating load of 15 to 20 percent. Digital systems can be programmed such that as the OA temperature varies from 60F to 15F, the maximum start-up steam valve positions vary from 20 to 100 percent (this start-up destroking limit could be removed at 0900).

Common practice on large converters provides two parallel piped steam control valves (usually sized for 1/3 and 2/3 capacity) such that the small valve handles light loads (preventing a single large valve from throttling down to the point where the valve plug approaches the valve seat where noise and seat erosion occur) and the large valve is sequenced in after the small valve is full open. At the point where the 2/3 sized valve starts opening, the same noise and erosion is possible.

With digital systems, the valves may be staged (rather than sequenced) such that as the load varies 0 to 33 percent, the small valve modulates from 0 to full open, and as the load varies 33 to 66 percent, the small valve closes and large valve modulates from 50 percent to full open, and as the load varies 66 to 100 percent, both valves modulate in unison from 2/3 to full open.

The following converter control example is for a dual valve converter with demand limiting and a warm-up shifted temperature reset schedule.

Dual Valve Converter, Demand Limiting, Setpoint Shift

Functional Description

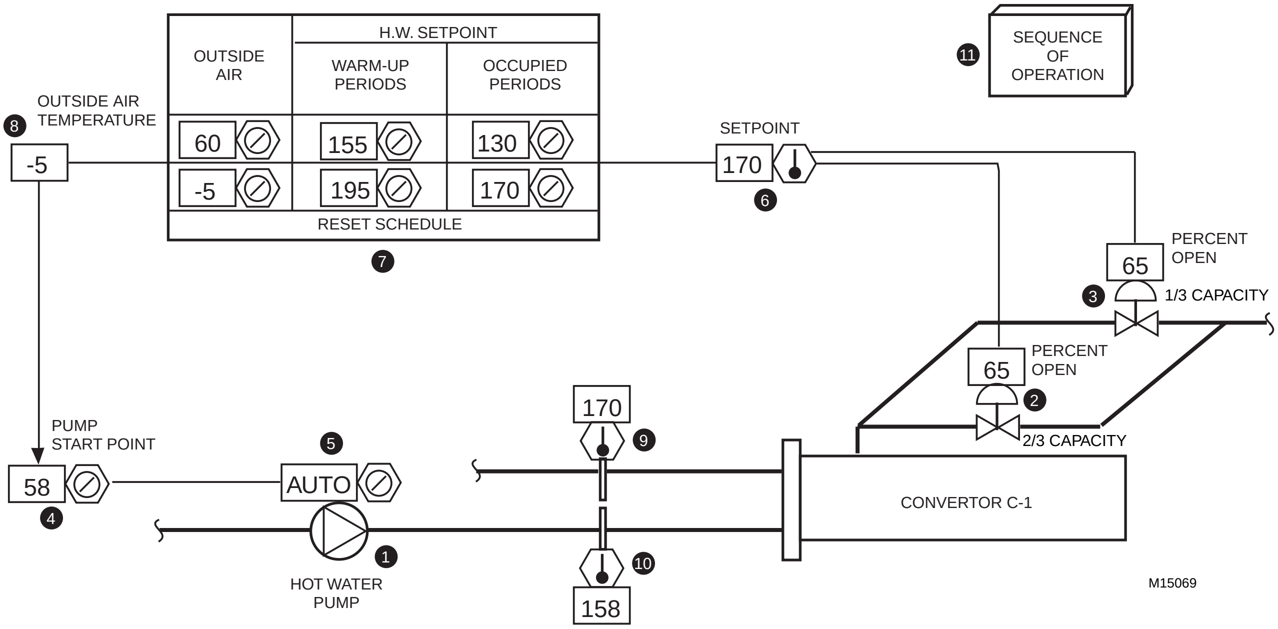


Fig. 95. Dual Valve Converter, Demand Limiting, Setpoint Shift Graphic.

Item No.	Function
1	Dynamic pump symbol. Pump starts on drop in OA temperature.
2,3	Steam valves staged for hot water temperature control.
4,5	Pump runs below OA temperature setpoint, subject to ON-OFF-AUTO selector.
6-8	Hot water temperature setpoint varied by dual reset schedule as OA temperature varies.
9	Controlled leaving water temperature.
10	Entering water temperature, operator information.
11	Icon, selects the sequence of operation text display.

Features

1. Staged 1/3 - 2/3 capacity control valves.
2. Dual reset schedule to accommodate the warm-up mode and occupied periods.
3. Demand limiting to prevent the converter from exceeding its required capacity when controls demand full load.

Conditions For Successful Operation

1. Technicians and users capable of understanding and tuning the control strategy.
2. Proper settings and timings of all control parameters.

Specification

Converter Control:

Anytime the pump does not prove operation, the converter valves shall close.

Control:

The control system shall be enabled anytime steam pressure is available. The hot water pump shall start anytime the OA is below 58F, subject to a software on-off-auto command.

Control:

During early morning periods (AHUs in warm-up mode) the hot water (HW) temperature setpoint shall vary from 155 to 195F as the OA varies from 60 to –5F. During occupied periods (any AHU in occupancy mode) the HW temperature setpoint shall vary from 130 to 170F as the OA varies from 60 to –5F. Setpoint warm-up to occupied period switching shall be ramped such that the change occurs over a 15-minute time duration. A PID function shall modulate the hot water valves as required to maintain the setpoint.

Staging:

Two normally closed valves shall be provided with Cvs of 25 and 63. On a demand for heating, the small valve shall modulate open. Upon demand for steam beyond the capacity of the small valve for a period of five minutes, the small valve shall close and the large valve shall assume the load. Upon a demand for steam beyond the capacity of the large valve for a period of five minutes, the small valve shall be re-enabled and both valves shall operate in unison, but with a combined capacity not to exceed the capacity noted in Demand Limiting.

With both valves operating, as the total demand drops below the capacity of the large valve for five minutes, the small valve shall close. With the large valve operating, as the demand drops below the capacity of the small valve for five minutes, the large valve shall close and the small valve shall assume the load.

Demand Limiting:
A large valve Cv limiting parameter shall be provided and set such that upon full demand, the valve Cv shall not exceed the design Cv of 51.

All valve timings and parameters shall be field adjustable by the owner.

THREE-WAY VALVE

The supply water temperature to the radiators (Fig. 96) is reset by the controller modulating the three-way valve. The controller modulates the three-way valve to maintain the reset schedule in response to temperature changes at the outdoor air sensor and the supply water sensor. Pump runs continuously during the heating season. As water temperature changes, the heat output changes linearly. This allows accurate changes in heating plant capacity as a function of outdoor air temperature or some other signal to match the load.

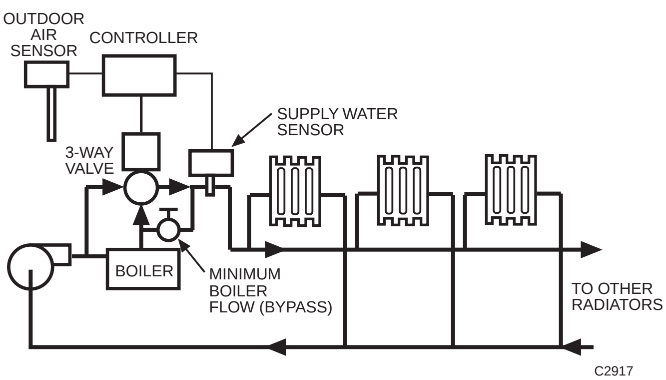


Fig. 96. Radiators with Scheduled Water Temperature Using Three-Way Valve.

Coordinating Valve Selection and System Design

A prerequisite to good modulating control of water systems is a coordinated design of the entire water system. All control valves must be sized so that the system will deliver design flow at full load and not generate uncontrollable conditions at minimum load.

Control valve selection is based on pressure differentials at the valve location, full load flow conditions, valve close-off, and valve controllability at minimum load conditions.

Generally, for smooth modulating control, the no-flow pressure differential should not exceed the full flow differential by more than 50 percent.

Hot Water Control Method Selection

Supply water temperature control is suitable for controlling the heat delivery from a heat exchanger or a secondary water pump.

Flow control is acceptable for controlling individual terminal units such as convectors, fan coils, or induction units. An equal percentage characteristic valve complements the water to air heat exchanger characteristic of heat output versus flow. The result is a relatively linear heat output as a function of stem position (Fig. 97). The heat output is relatively constant, providing optimized controllability of heat exchangers such as converters, fan coils, and induction units.

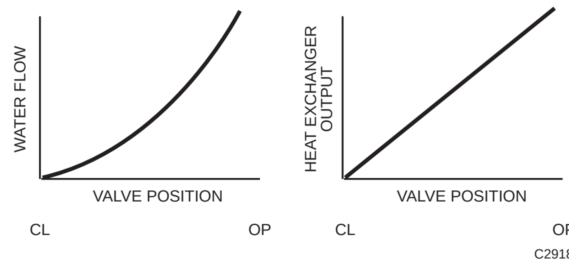


Fig. 97. Control of a Water Heat Exchanger Using an Equal Percentage Characteristic Valve.

CHILLED WATER DISTRIBUTION SYSTEMS

GENERAL

Chilled water systems for cooling of commercial buildings usually provide water between 40 and 50F to finned coils in room units or air handlers.

The amount of water delivered to the cooling coils and/or produced by chillers is related to temperature difference (TD_W) across the coil or chiller and energy (tons of refrigeration) by the equation:

$$gpm = \frac{Tons \times 24}{TD_W}$$

Where:
1 Ton = 12000 Btuh
24 = a constant
TD_W = water temperature difference in degrees F

EXAMPLE:
Given: A 100 ton chiller with a TD_W of 10F.

$$gpm = \frac{100 \text{ Tons} \times 24}{10} = 240 \text{ gpm}$$

The same equation expressed in MMBtuh (million Btu per hour) is:

$$gpm = MMBtuh \times \frac{2000}{TD_W}$$

EXAMPLE:

Given: A 100 ton chiller with a TD_W of 10F.

$$100 \text{ Tons} \times \frac{12{,}000 \text{ Btuh}}{\text{Ton}} = 1.2 \text{ MMBtuh}$$

$$gpm = \frac{1.2 \text{ MMBtuh} \times 2000}{10} = 240 \text{ gpm}$$

System Objectives

The chilled water control and distribution system should:

1. Provide the minimum flow of chilled water through the chiller as specified by the manufacturer.
2. Provide a stable pressure difference across supply and return mains.
3. Prevent freeze-up in chiller and/or coils exposed to outdoor air.
4. Control system pumps and bypass valves to prevent short-cycling of pumps or radical pressure changes across control valves at the terminal units.

In addition, for systems with two or more chillers operating at once, the chilled water control and distribution system should:

1. Prevent return water from mixing with chilled water before leaving the chiller plant.
2. Shut off flow through idle chiller(s).

Control of Terminal Units

The heat transfer characteristic is close to linear for a cooling coil (Fig. 98) because the air to water temperature difference is lower than that for hot water coils. This means a valve with either a linear characteristic or an equal percentage characteristic plug is satisfactory.

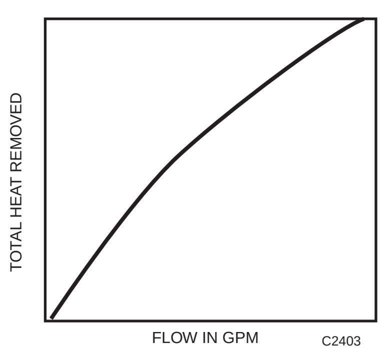


Fig. 98. Typical Heat Transfer Characteristic of a Cooling Coil Supplied with Chilled Water.

DUAL TEMPERATURE SYSTEMS

Figure 99 shows a typical arrangement where the same pipes carry hot water for heating or chilled water for cooling to the same terminal units. These are usually fan coil units.

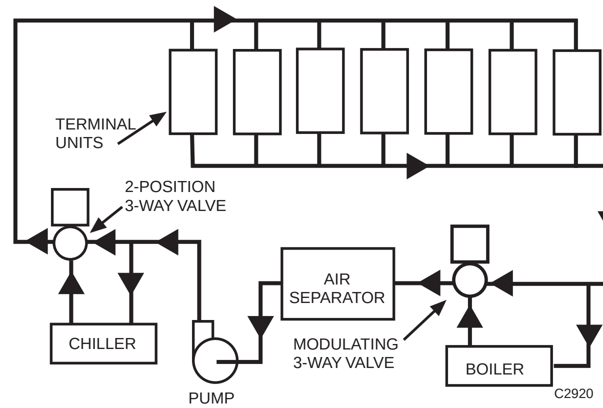


Fig. 99. Two-Pipe Dual-Temperature System.

The three-way valves on the chiller and boiler are used for changeover. The valve on the chiller is controlled two-position. The three-way boiler valve is controlled modulating for heating so that the hot water supply temperature may be reset from the outside temperature. Boiler minimum flow or temperature may require system modifications.

Changeover from heating to cooling can be based on outdoor air temperature, solar gain, outdoor wet-bulb temperature, or a combination. System bypass or other pressure control method may also be required for such systems.

Changeover Precautions

A time delay in changing the operation from hot to chilled water or vice versa is required to avoid putting hot water into the chiller or chilled water into the boiler. A deadband between heating and cooling will usually provide enough delay. Hot water to the chiller can cause the compressor to cut out from high head pressure and/ or damage the compressor. Chilled water to the boiler can cause thermal shock to the boiler and/or flue gases to condense on the fireside. Protection for boilers is covered in DUAL BOILER PLANT CONTROL. For chillers a maximum entering water of 80F is required to avoid excessive refrigerant head (high head pressure). A thermostatic interlock can provide this safety feature.

MULTIPLE-ZONE DUAL-TEMPERATURE SYSTEMS

Since different zones often have different changeover requirements, it is often necessary to furnish hot water to one zone and chilled water to another, as in a curtain wall building with high solar loads.

Figure 100 uses three-way valves V1 through V4 operating two position to accomplish zone changeover. In large systems, two-way valves may offer a tighter isolation of the hot and chilled water circuits than three-way valves.

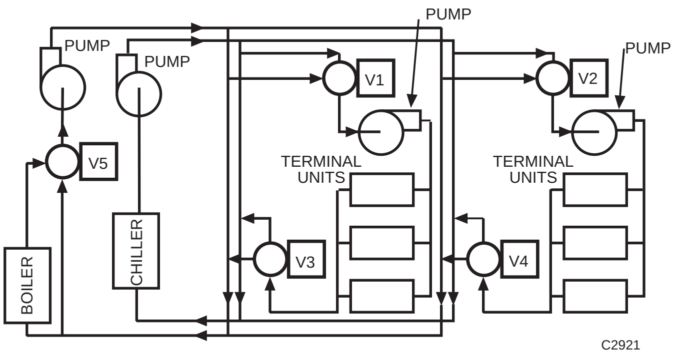


Fig. 100. Two-Pipe Multiple-Zone Dual-Temperature System Using Zone Pumps.

STEAM DISTRIBUTION SYSTEMS AND CONTROL

INTRODUCTION

Steam distribution systems are classified as either low pressure (15 psig and less) or high pressure (above 15 psig). Low pressure systems have many subclasses such as one-pipe, two-pipe, gravity, vacuum, and variable vacuum. See HOT WATER DISTRIBUTION SYSTEMS for steam-to-hot water converter configurations and control.

ADVANTAGES OF STEAM SYSTEMS VS HOT WATER SYSTEMS

The principle reasons for the use of steam to distribute heat in commercial buildings or in groups of buildings separated from the heating plant are:

- Steam is light weight (27 cubic feet per pound).
- Steam has high heat content (1000 Btu per pound).
- Steam flows through pipes unaided by pumps.
- Steam does not create excessive static pressure on piping in tall buildings.
- Terminal units can be added or removed without basic design changes.
- Draining and filling are not necessary to make repairs as with hot water systems.
- Steam can be distributed through a large system with little change in heating capacity due to heat loss.

STEAM SYSTEM OBJECTIVES

Carefully consider distribution system objectives when applying controls either to the boiler or to the distribution system itself. Control and/or piping of the boiler or steam generator are not considered in this section. Not all of Objectives 1 through 7 may apply to any given distribution system.

1. Steam mains must provide adequate capacity so steam velocity is between 8,000 and 12,000 feet per minute (133 to 200 feet per second).
2. Water must not be allowed to accumulate in the mains. Provisions must be made for the use of traps or superheated steam to reduce or eliminate water in mains.
3. Pockets of water must not be allowed to accumulate. Steam traveling at 200 feet per second (135 mph) can propel the water causing water hammer, which can damage or destroy piping.
4. Condensate must be returned to the boiler at the same rate as steam leaves the boiler. Otherwise, the boiler will be shut down by low water cutoff control or be damaged from lack of water covering heated metal.
5. Provision must be made to expel the air when steam is again supplied. When any part of the system is not supplied with steam, that part of the system fills up with air from the atmosphere. If air is present with the steam, the air plates the heat exchanger surfaces and reduces capacity. The oxygen in the air causes pitting of iron and steel surfaces and the carbon dioxide (CO_2) in the air forms an extremely corrosive carbonic acid solution.
6. The return condensate piping system must be sized for a low-pressure loss to eliminate flashing. For example, if 15 psi steam condenses in a heating coil, the condensate is still near the boiling point, say 240F; and if the return main is at atmospheric pressure, the condensate can flash into steam. This wastes heat and can block the return of condensate to the boiler.
7. If necessary, the flow of steam must be accurately measured to account for steam usage. When steam is used in a closed system (none is vented to atmosphere), the steam flow to a building or zone can be measured by measuring condensate flow.

PROPERTIES OF STEAM

One Btu added to one pound of water raises the water temperature one degree Fahrenheit. When water temperature reaches 212F at sea level (14.7 psia) it contains 180 Btu/lb (212 – 32) = 180. However, it takes another 970 Btu to convert the one pound of water to a vapor (steam). The total heat of the vapor is: 180 Btu/lb + 970 Btu/lb = 1150 Btu per pound. The 970 Btu/lb is the latent heat required to vaporize water.

One pound of water in the liquid state occupies about 28 cubic inches at 32F. When converted to vapor at 212F, it occupies 27 cubic feet or 1672 times as much space as the liquid.

One pound of steam (water vapor) when cooled and condensed in a radiator or other heating device gives up 970 Btu to the device and returns to its liquid state. If the liquid (water) leaves the radiator at 180F, it gives up another 32 Btu, so the total heating value of low pressure steam is said to be 1000 Btu per pound (actually 32 + 970 or 1002 Btu).

STEAM SYSTEM HEAT CHARACTERISTICS

Figure 101 shows the characteristics of one pound of steam as it travels through a steam heating system.

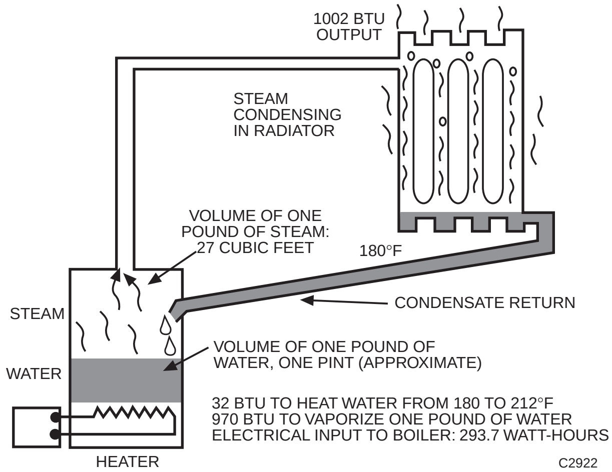


Fig. 101. One Pound of Water in a Steam Heating System.

STEAM PRESSURE, TEMPERATURE, AND DENSITY

Steam temperature is directly related to pressure, see Table 6. For a more extensive table refer to the General Engineering Data section. Note that density, which is the reciprocal of specific volume, increases sharply with pressure, while total heat per pound remains relatively constant.

Table 6. Approximate Values for Properties of Saturated Steam.

Pressure in PSIG	Temperature Saturated	Density, Pounds per 1000 Cubic Feet	Latent Heat of Vaporization, Btu/Lb	Total Heat Btu/Lb
1	215F	40	968	1152
10	239F	61	953	1160
100	338F	257	881	1190
200	406F	469	838	1199

This table illustrates that in a given size pipe more than four times as much steam can be carried with steam at 100 psig as with steam at 10 psig. However, a 100 psi steam main is 99F hotter than 10 psi steam.

EXAMPLE:
The heat in 1000 ft^3 of steam:

at 10 psi is 61 lb x 1,160 Btu/lb = 70,760 Btu
at 100 psi is 257 lb x 1,190 Btu/lb = 305,830 Btu

STEAM QUALITY

Steam tables generally show properties of dry-saturated steam. Dry-saturated steam has no entrained moisture and is at the boiling point for the given pressure. Dry-saturated steam is said to have 100 percent quality. Steam produced in a boiler usually has some water droplets entrained in the steam and is called wet-saturated steam. Condensation collecting within steam mains can also become entrained and lessen steam quality. If 10 percent of the steam weight is liquid, the steam has 90 percent quality.

If steam has 85 percent quality and if 1000 lb/hr is needed through a pipe or valve, then 1000/0.85 = 1176 lb/hr of the 85 percent quality steam is required.

Superheated steam is steam at a temperature above the boiling point for the indicated pressure. It can be produced by passing the saturated steam through a superheater. A superheater consists of steam tubes exposed to hot gases from the firebox. This steam is hotter than the temperature listed in steam tables. Superheat is expressed in degrees Fahrenheit or Celsius. Since superheated steam has a higher heat content per pound, the steam quantity needed through a pipe or valve is reduced.

Pressure reducing valves can also produce superheated steam. For example: 100 psi steam at 338F passing through a pressure reducing valve gives up no heat as it expands to 10 psi, so the 10 psi steam downstream will be at 338F not 239F. This is 99 degrees Fahrenheit of superheat and downstream valves and piping will be exposed to the higher temperature. To correct for superheated steam, 1 Btu/lb is added for each Fahrenheit degree of superheat.

EXAMPLE:
1000 MBtuh of 10 psi steam is required. From Table 6, 10 psi steam has a latent heat of 953 Btu/lb. If condensate leaves at 180F, the steam gives up 1012 Btu/lb (953 Btu/lb condensing the steam and 59 Btu/lb cooling the condensate from 239F to 180F). If the 10 psi steam has 90 degrees of superheat, the added heat is 90 Btu/lb. Thus, heat available from 10 psi steam with 90F of superheat is:

1012 Btu/lb + 90 Btu/lb = 1102 Btu/lb.

Steam quantity needed is:

$$\frac{1{,}000{,}000 \text{ Btuh}}{1102 \text{ Btuh}} = 907.4 \text{ lb/hr}$$

STEAM DISTRIBUTION SYSTEMS

STEAM PIPING MAINS

Steam piping mains must carry steam to all parts of the system allowing a minimum amount of condensate (heat loss) to form, must provide adequate means to collect the condensate that does form, and must prevent water pockets that can cause water hammer.

Figure 102 shows typical connections at the boiler. The system shown uses a condensate (receiver) tank and pump. Condensate returns must be properly pitched for gravity condensate flow and properly insulated so heat contained in the condensate is not wasted.

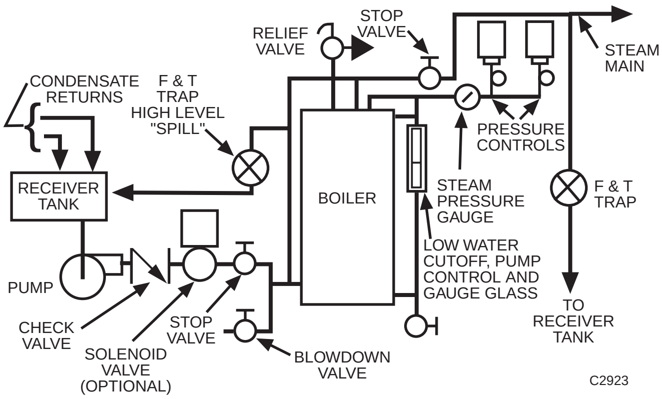


Fig. 102. Typical Boiler Connections.

STEAM TRAPS

Traps remove condensate from the steam mains and all steam-using equipment without allowing steam to enter the return mains. The thermostatic trap (Fig. 103) is most common for trapping condensate from radiators and convectors in low pressure systems. When the thermostatic element senses steam temperature, it closes the valve. The valve remains closed until the element cools.

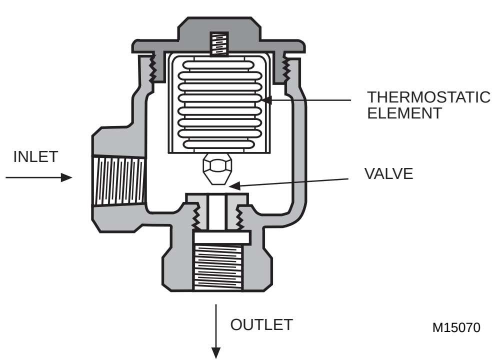


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Fig. 103. Thermostatic Trap.

The float and thermostatic trap (Fig. 104) can handle large amounts of air and condensate and is commonly used on steam coils in air handling systems. In this trap, the thermostatic element passes air until it senses steam at which time it closes the valve. As condensate water builds up in the trap, the float valve opens to discharge condensate into the return line.

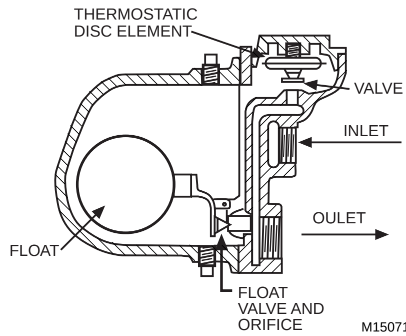

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Fig. 104. Float and Thermostatic Trap.

Another common trap is the inverted bucket (Fig. 105) which has a large capacity (up to 55,500 lb/hr) for discharging condensate. In the bucket trap, the bucket is normally down so the valve is open. The bucket is normally about two-thirds full of condensate so, when condensate enters the trap, it flows around the bucket and through the trap. As air or steam enters the bucket, the bucket rises, closing the valve when the condensate level drops to about one-third full. Air escapes slowly through the air vent and steam condenses so the bucket drops, opening the outlet.

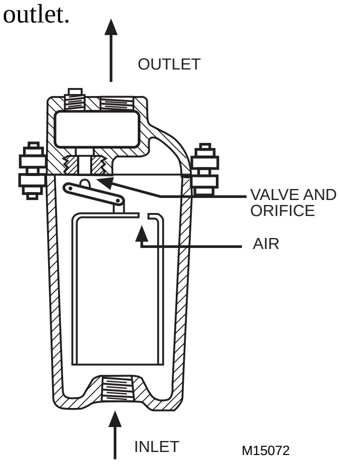

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Fig. 105. Inverted Bucket Trap.

Kinetic traps, shown in Figures 106, 107, and 108, are used in high pressure systems. They rely on flow characteristics of steam and condensate and on the fact that condensate discharging to a lower pressure contains more heat than necessary to maintain its liquid state.

The disc trap (Fig. 106) is a device with only one moving part. As steam in the chamber above the disc cools and condenses, the disc snaps open releasing condensate. These traps cycle independent of condensate load. Systems using these traps usually have a means of recovering heat from the condensate. They are used primarily on high pressure steam systems.

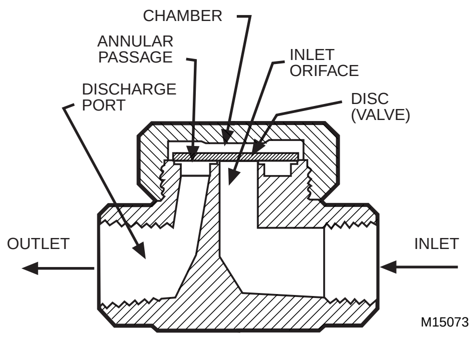

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Fig. 106. Disc Trap.

The orifice trap (Fig. 107) has no moving parts. It relies on only a little steam flowing through the orifice because of the low density of steam compared to condensate. It must have constant supply pressure, constant demand, and constant ambient temperature to operate.

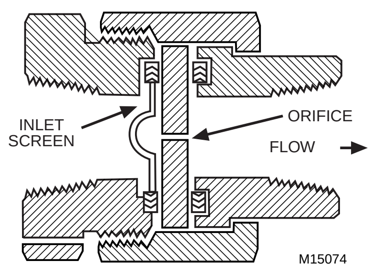

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Fig. 107. Orifice Trap.

The piston trap (Fig. 108) relies on a pressure change caused by steam or condensate entering the trap to open or close the piston or port. A small amount of condensate bleeds through an orifice in the control disc into the control chamber. When steam or condensate that flashes to steam enters the chamber, the port is closed until the pressure in the control chamber falls. The port then opens allowing condensate to pass.

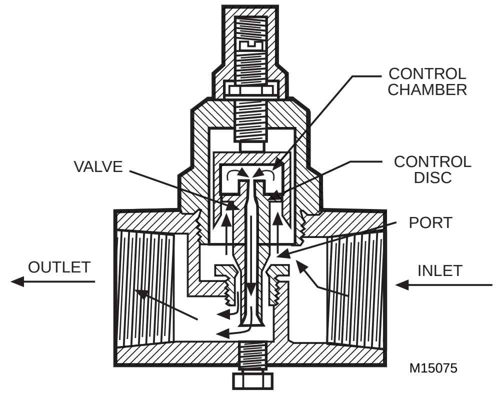

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Fig. 108. Piston Trap.

PRESSURE REDUCING VALVE STATION

Pressure reducing valve (PRV) stations are used in steam systems where high pressure steam, usually at 50 to 125 psig, is distributed for low pressure requirements (Fig. 109).

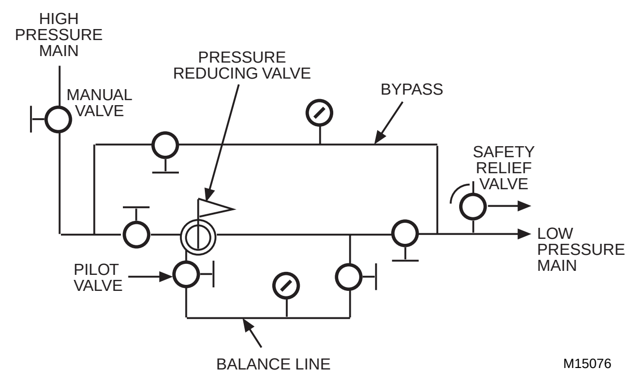


Fig. 109. PRV Station.

If steam is supplied at less than 50 psig, a single stage PRV (Fig. 109) is required. For 50 psig or higher main pressure, a second stage is added, in part because critical pressure drop (50 percent of the absolute inlet pressure) through a single stage PRV will cause noise and wear problems.

Since steam at 125 psig has a temperature of more than 350F and little heat energy is dissipated through a PRV, the 5 psi steam leaving the PRV station will be superheated (temperature is more than 227F) for some distance downstream from the station.

Selection of valve materials used to regulate the 5 psi steam must take this superheat into account, particularly for a device in close proximity to the PRV.

TYPES OF LOW PRESSURE STEAM DISTRIBUTION SYSTEMS

One-Pipe Steam Systems

One-pipe systems were often used in small buildings and apartments because of their simplicity and low cost. The same pipe carried both steam and condensate (Fig. 110). They used gravity returns and, therefore, required no traps.

Control valves on individual radiators are a potential problem in one-pipe systems. As valves close, water is trapped in the radiators resulting in low boiler water and water hammer noise at the radiators when the valves reopen. Oversized valves controlled two-position are required to allow condensate to flow out of radiators while steam is entering. Thermostatic air vents allow steam to completely fill the radiator. Many one-pipe systems have no valves and temperature is controlled by cycling the boiler.

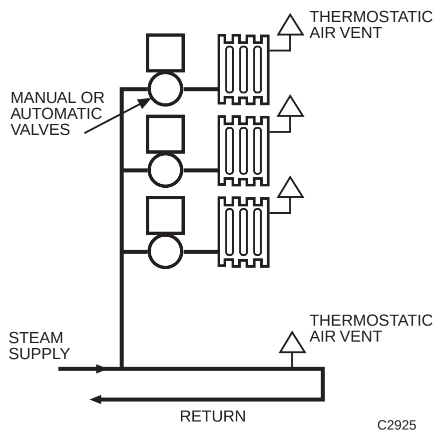


Fig. 110. One-Pipe System.

Two-Pipe Steam Systems

Two-pipe systems (Fig. 111) are used for all systems using automatic control of terminal units. A trap at each unit removes condensate.

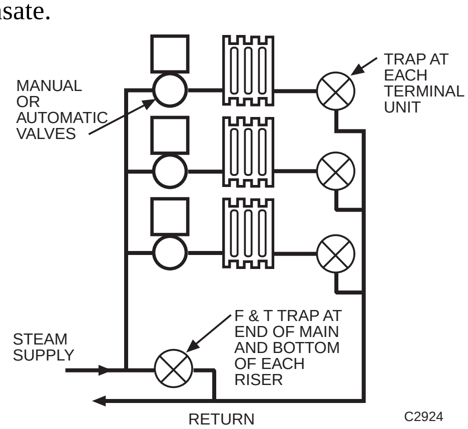


Fig. 111. Two-Pipe System.

TYPES OF CONDENSATE RETURN SYSTEMS

In addition to gravity return used for small systems, either open or vacuum return systems can be used.

In open return systems (Fig. 112) condensate flows by gravity to one or more receivers where it is pumped back to the boiler. Pumps are started and stopped by float controls in the receiver.

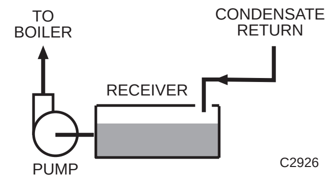


Fig. 112. Open Return System.

A vacuum return system requires a vacuum pump assembly to discharge noncondensable gases to the atmosphere and to force the condensate into the boiler (Fig. 113). A vacuum pump assembly consists of a vacuum pump, a separator tank, and an air vent.

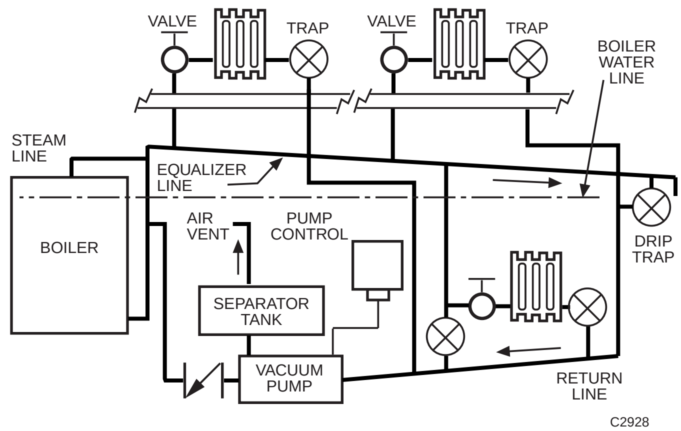


Fig. 113. Typical Two-Pipe Vacuum Pump System.

The vacuum in the return line draws a condensate/air mixture into the receiver. This mixture is forced into a separating tank where the air is vented to the atmosphere and the condensate returned to the boiler. The air is discharged through a check valve or a nonreturn vent to maintain a vacuum in the return line under operating conditions. The vent on the vacuum pump is usually the only vent in the system.

Typically, a selector switch allows the vacuum pump to be operated automatically or manually. In the automatic position a vacuum switch or float switch cycles the vacuum pump.

A steam system may be either open or closed. In a closed system all condensate is returned to the boiler. In an open system the condensate is expelled or used for other purposes, such as humidifiers, dishwashers, and autoclaves and must be replaced by makeup water.

Low pressure steam systems operate on pressures ranging from 0 psig to 15 psig. The steam main pressures are usually under 10 psig and the returns are vented to the atmosphere. To maintain the return main pressures at 0 psig, an air vent releases the air from the system when steam is present and allows it to return to the system when steam is not present. These vents do not allow steam to escape. A sealed chamber (bellows) within the vent contains a volatile liquid which vaporizes or condenses when temperature changes. This causes the valve to close when steam is present preventing the discharge of steam from the system.

Since all of the air in a steam system may not be forced out through the radiator vents, it is necessary to provide auxiliary vents at the ends of the steam mains and in the returns to ensure air removal from the system. These vents are larger than radiator vents and are referred to as quick vents.

VARIABLE VACUUM RETURN SYSTEMS

In a variable vacuum steam system, also called a subatmospheric steam system, the operating pressures range from 20 in. Hg vacuum to 15 psig. The principle of water boiling at reduced temperatures in a vacuum is the basis of a variable vacuum steam system. In an atmospheric-return low-pressure steam system, when the thermostat turns the burner off, boiling ceases as soon as the temperature of the water drops to 212F. In a variable vacuum system, boiling continues until the water temperature drops to the boiling point of the vacuum pressure created by the condensation of steam in the system. This means that steam is continuously supplied to the radiator at a decreasing temperature until the limit of induced vacuum is reached. Table 7 shows reduced heat output from radiators in a vacuum system. To operate successfully, the system must be air tight. Variable vacuum systems use nonreturn vents which allow air to leave the system but none to return. A nonreturn vent is similar to the low pressure steam vent with a small check disc that closes to prevent inflow of air to the system when the pressure drops below atmospheric.

The period of time that a variable vacuum system can operate in the vacuum range is dependent upon the tightness of the connections of the system. Once a variable vacuum system has built up a vacuum, it is possible for the burner to operate at frequent enough intervals to keep the system within the vacuum range during mild and intermediate load conditions. Since a low steam temperature is maintained, the radiator temperature is low. By keeping the radiator warm for longer periods of time, temperature stratification in the space is reduced.

Table 7. Heat Output versus Steam Pressure for Radiators, Convectors, and Coils.

Saturated Steam Pressure PSIG	Steam Temp, °F	BTU/HR/EDR Radiators [b]	Typical Steam Coil [a] With 40F Entering Air	
			Convectors [c]	MBH [d]
10	239.4	286	288	111.0
5	227.1	262	259	104.5
4	224.4	254	250	103.2
3	221.5	247	242	101.6
2	218.5	240	235	100.0
1	215.3	233	228	98.2
0	212.0	226	221	96.2
2 in. Hg	208.5	216	211	94.5
4 in. Hg	204.8	209	204	92.0
6 in. Hg	201.0	199	194	90.2
10 in. Hg	192.2	182	178	—
15 in. Hg	178.9	158	151	—
20 in. Hg	161.2	125	111	—

[a] 2 row steam distributing coil, 7 fins per inch, 500 fpm face velocity, and 2.4 sq ft face area
[b] 75F Room temperature
[c] 70F Inlet temperature
[d] 1000 Btu/hr

CONTROL PRINCIPLES FOR STEAM HEATING DEVICES

GENERAL

To control a steam supplied device, the system design should include valves and other equipment required to produce the amount of heat needed at design conditions. In addition, the system should be capable of controlling a steady flow of heat directly related to the demands of the thermostat or other controller at load conditions from 0 to 100 percent.

To design a steam system that is capable of controlling the various radiators and coils in a building, the pressure relationships between the various elements of a steam heating system must be analyzed under various load and system conditions.

MODULATING STEAM COIL PERFORMANCE

Figures 114 and 115 show a steam coil supplied from a 5 psig steam main, controlled by an oversized modulating valve (Fig. 114) or a correctly sized modulating valve (Fig. 115), and discharging condensate to the return through a trap. The figures demonstrate the importance of proper sizing of the modulating valve and the ability of the valve to control the output of the coil. Refer to the Valve Selection and Sizing section for valve sizing procedures.

Oversized Valve

In Figure 114A a large valve is used to ensure 4 psi in the coil. When the valve is full open, the coil is full of steam and the output is 100,000 Btuh. The return header and trap are also full of steam keeping the trap closed much of the time. The trap opens slightly when water accumulates around the thermostatic element allowing the condensate into the return mains. The pressure drop through the valve and coil is small, but large across the trap.

Figure 114B shows the pressure relationship and heat output as the modulating valve closes to the half-open position. The smaller opening in the valve requires 4 psi to get the steam through the valve, called a 4 psi drop across the valve. This leaves 1 psi in the steam coil header. Although the thermostat has signaled to the valve to cut heat output in half, the coil is still full of steam and the heat output is only reduced by 5 percent. The reduction is because of the difference in temperature of 4 psi steam compared to 1 psi steam, assuming 50F air temperature entering the coil. The trap drop is 0.9 psi. Most of the pressure drop between the supply and the return mains is now across the steam valve. The portion of the modulating valve stroke between half open and full open reduces the heat output only 5 percent.

Figure 114C shows the quarter-open valve position. Half of the coil surface is starved of steam, the heat output is reduced to about half of the original value and the trap is full open. All of the steam has been condensed in the coil before reaching the trap. Virtually all of the drop between the supply and return mains is dissipated through the control valve.

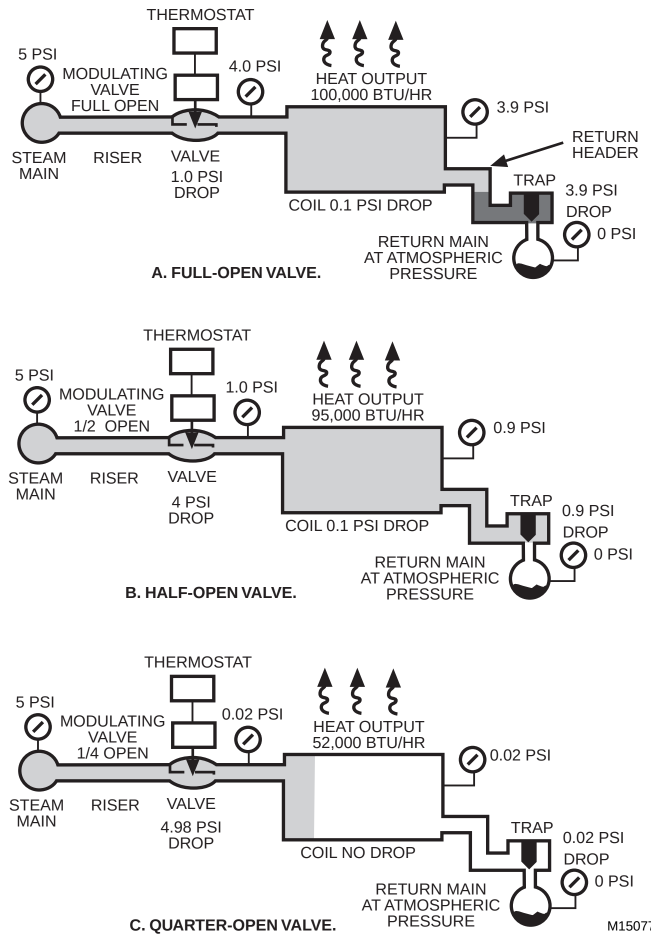


Fig. 114. Control Results with Oversized Valve.

The conclusions reached from Figure 114 are:

1. The sum of the individual pressure drops across the valve, coil, and trap equals the pressure difference between the supply and return mains.
2. Heat output changes little due to pressure change within the coil, as long as the coil is full of steam.
3. Heat output changes little until the valve assumes most of the pressure drop between supply and return mains.
4. Heat output from the coil is reduced by starving a part of the coil surface of steam and allowing the surface to cool off.

NOTE: Steam distributing coils allow reduced output without the return end becoming cold.

Correctly Sized Valve

The valve in Figure 115A is sized for a 4 psi pressure drop when it is full open leaving 1 psi of steam for the coil and the trap. Full heat output of the coil is 95,000 Btuh and most of the drop between supply and return mains occurs across the valve.

Figure 115B shows the valve in the half-open position. The output of the coil is cut approximately in half (54,000 Btuh). This is in contrast to the oversized valve application (Fig. 100B) where the heat was cut only 5 percent.

Figure 115C shows the output cut to 27,000 Btuh with the valve in the quarter-open position. In contrast, the oversized valve in the quarter-open position produced 52,000 Btuh.

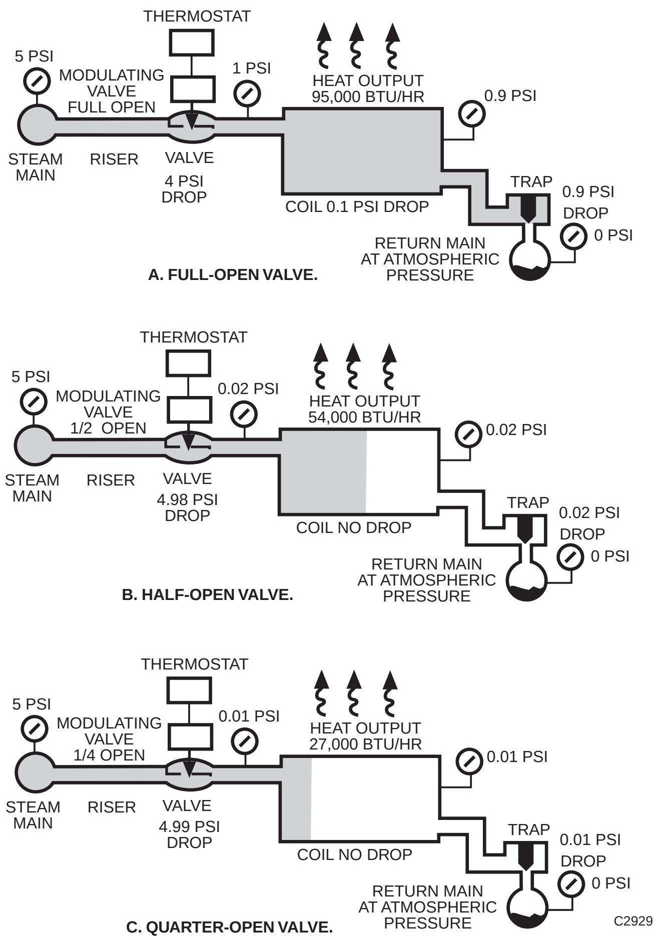


Fig. 115. Control Results with Correctly Sized Valve.

The conclusions reached from Figure 115 are:

1. A valve with a large pressure drop will be effective in controlling heat output over its entire stroke.
2. The valve, not the trap, takes up most of the pressure drop between supply and return mains.

SUPPLY AND RETURN MAIN PRESSURES

The supply main pressure should be constant and sufficient to allow an 80 percent drop through the control valve and still leave enough steam pressure downstream from the valve to produce the desired heat output. If boiler pressure is not constant, install a pressure reducing valve ahead of all steam supplied devices where output temperatures may vary rapidly with steam pressure fluctuations.

Even though the control valves do not change position, variations in return main pressure causes fluctuations in steam flow through control valves. From a control standpoint, an atmospheric return with a condensate pump is more effective than a vacuum return with a vacuum pump that can cycle over a range of several inches of vacuum.

As an example of the effect of fluctuating supply and return main pressures, assume a system where the boiler cycles so that it shuts off at 6 psi and cuts in at 2 psi. On the same system assume that a vacuum pump is used which cuts in at 4 in. Hg and shuts off at 8 in. Hg of vacuum. The pressure difference between supply and return mains can vary from a minimum of 4 psi to a maximum of 10 psi as the boiler and vacuum pump cycle. This means a 60 percent variation in capacity of the control valves in the building as the pressure fluctuates. Control valves correctly sized for 4 psi are 60 percent too large during periods when a 10 psi difference exists across the supply and return mains.

SYSTEM DESIGN CONSIDERATIONS FOR STEAM COILS

Figure 116 shows the optimum design conditions and piping arrangement for a steam supplied heating coil. Considerations for effective control are:

1. Steam mains held close to design pressures. Refer to SUPPLY AND RETURN MAIN PRESSURES.
2. Returns at atmospheric pressure, unless lifts (condensate pumps) are required in the returns.
3. Traps sized to pass design condensate flow at 1 psi drop.
4. An equalizer line to prevent formation of a vacuum within coil.
5. A control valve pressure drop of 80 percent of the difference between supply and return main pressures.

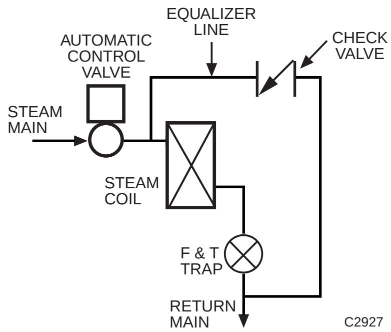


Fig. 116. Steam Supplied Air Heating Coil.

High vacuum systems are an exception to these considerations since they lower the steam temperature and pressure as the heating load decreases. Vacuum systems are adaptable to automatic control valves, since usual practice is to maintain a controlled difference between supply and return main pressures while varying supply main pressure with heating load.

LOW TEMPERATURE PROTECTION FOR STEAM COILS

Any steam coil exposed to outdoor air is in danger of freeze up in cold climates. A coil begins to freeze in the 30 to 32F temperature range. Steam coils designed for heating cold air contain internal distributing tubes to ensure that steam reaches all parts of the coil as the control valve modulates.

Another approach to freeze-up control is to design coils with dampers so that the control valve does not modulate but remains open when air entering the coil is below freezing. If too much heat is being delivered, face and bypass dampers control the airflow across the coil. Above freezing, the valve can be modulated.

In all cases, a low limit temperature controller, which responds to the coldest portion of the capillary sensing element, should be part of the design. For addition examples of control with freezing air conditions entering a coil see Air Handling Systems Control Applications section.

HIGH TEMPERATURE WATER HEATING SYSTEM CONTROL

INTRODUCTION

HIGH TEMPERATURE WATER (HTW) HEATING

High temperature water systems operate with supply water temperatures of 250 to 450F and pressures from 55 to 450 psig.

HTW is typically generated by boilers; however, experimental systems utilizing geothermal or solar energy have been built. First costs are similar for steam and high temperature water systems, however, maintenance and operating costs are generally lower for HTW systems. The use of the same boiler for both HTW and steam generation is not recommended because feed water requirements for steam eliminate some of the advantages of HTW.

When relatively small amounts of steam are required, steam can be produced from HTW at the required location. A steam generator using 350F HTW (120 psig) will produce 15 psig steam and using 380 to 410F HTW (200 to 275 psig) will produce 100 psig steam allowing a HTW temperature drop of 50 to 60F.

A HTW system offers several advantages over a steam system:

— Boiler sizes can be smaller than low pressure boilers because of the high heat capacity in a HTW system.
— Diameter of distribution piping can be reduced.
— The piping system requires no grading for return of water to boiler.
— Feedwater requirements are minimal, eliminating treatment costs and introduction of air which is a source of corrosion. HTW systems tend to remain clean.
— Steam traps and condensate receivers, sources of heat loss and maintenance costs are eliminated.
— Heat not used in terminal heat transfer units is returned to the HTW generator.

Several major design characteristics are typical of HTW systems:

1. The HTW boiler is controlled by pressure rather than temperature to eliminate flashing if heating load fluctuates.
2. Multiple boiler systems must be designed so that loads are divided between the boilers. Generally it is less costly to operate two boilers at part load than one at full load.
3. HTW systems can be pressurized by steam or air in the expansion tank but typically an inert gas such as nitrogen is used because it absorbs no heat energy and excludes oxygen.
4. All piping is welded except at mechanical equipment which must be maintained. Connections at equipment are flanged, including provision for removal of small threaded control valves.
5. Terminal units are rated for the high temperature and pressure.

Figure 117 illustrates the elements of a typical HTW system.

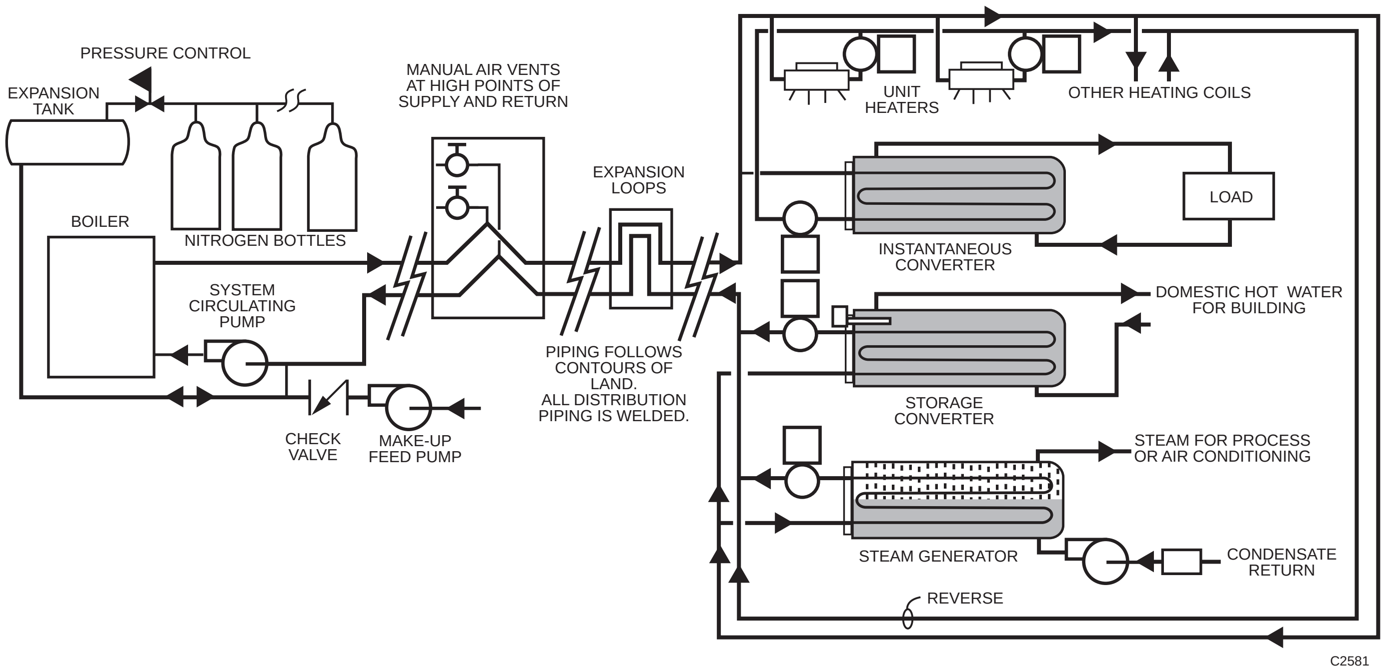


Fig. 117 Typical Nitrogen Pressurized High Temperature Water System.

HIGH TEMPERATURE WATER SAFETY

A well designed HTW system with proper installation of piping to prevent undue stress rarely fails.

HTW drops in temperature very rapidly with a minor, low mass, flow leak into space. In low or medium temperature water the escaping fluid remains at nearly the same temperature. HTW flashes to steam and the turbulent mixing of liquid and vapor with the room air rapidly drops the fluid temperature below 212F and within a short distance to 125 to 140F.

Minor leakage at valves is usually imperceptible except for deposits of scale at the point of leakage. Valves should be inspected periodically and scale removed from valve stems to prevent damage to packing and to allow free valve movement.

HTW CONTROL SELECTION

The features of good temperature control for low and medium temperature water heating systems also apply to high temperature water heating. Special considerations are needed for controls to withstand the temperatures and pressures encountered in a HTW system. The large temperature difference between HTW and the heated fluid (air, water, or steam) means that most of the control selection precautions are to provide fail safe operation. Careful consideration must be given to what happens to the secondary fluid of a converter and its controls when flow is reduced or stopped, or equipment is shutdown. Secondary fluid can overheat immediately after or during prolonged shutdown. Controls near a HTW supplied coil can be damaged by high temperature if airflow stops.

Controls can be pneumatic, electric, or digital. The low mass of electronic temperature sensors (thermocouple or resistance bulb) provides faster response than fluid-filled or bimetal elements. Pneumatic actuators generally have the faster response, higher temperature ratings, and greater reliability required for HTW applications.

Standard commercial quality controls with integral (automatic reset) and derivative (rate) action are satisfactory for most applications when used with industrial quality valves. An industrial controller or a digital system may be required where local recording of a control value is specified. The control system selected should:

1. Function from zero to full load.
2. Close the HTW valve if there is failure in the control system, loss of power or air, or shutdown of the system being controlled.

HTW VALVE SELECTION

Valves for HTW must be selected to ensure suitability for high temperature and pressure conditions. A control valve must have wide rangeability (50 to 1 is desirable) and the ability to position accurately and hold that position. Rangeability is the ratio of maximum flow for a valve to the minimum controllable flow. These requirements dictate use of industrial quality valves. For additional information on rangeability see the Valve Selection and Sizing section.

VALVE STYLE

Single seated valves are recommended because of their tight shut-off and availability with equal percentage and reduced capacity (Cv) trim.

Double seated valves are not recommended because they do not provide tight shut off. High velocity leakage past the plug of a double-seated valve causes greater erosion to the seats than a single-seated valve.

Three-way valves are not recommended because they are not available with equal percentage flow characteristics and therefore do not provide the required control stability. They also are not available with reduced Cv trim for accurate sizing.

Valve bodies should be flanged although threaded bodies can be used for sizes of 1-1/4 inch or less. Weeping and flashing at threads is likely but usually not visible except for deposits left behind. If threaded valves are used in the HTW lines, use adjacent bolted flanges not union fittings.

VALVE BODY MATERIALS

Cast steel bodies with 300 psi or 600 psi body ratings are recommended. These have temperature ratings above 750F, the limit for packing materials. Manufacturers literature lists actual pressure and temperature ratings.

Bronze bodies are not recommended because the maximum temperature limit is 400F. In addition, since HTW piping is steel, galvanic action between bronze and steel can be serious at the elevated temperatures encountered.

Cast iron bodies are not recommended because maximum operating conditions are limited to 406F at 250 psig, which are too close to operating conditions.

VALVE TRIM

Use stainless steel valve stems and trim. Standard trim for industrial valves is usually 316 stainless steel. Composition discs leak and corrode at the seat and disc and are not used.

VALVE PACKING

Shredded Teflon, Teflon asbestos, graphite asbestos, and spring loaded Teflon V-rings, are available in industrial valves and acceptable to 450F. A valve with a deep packing gland is normally required.

VALVE FLOW CHARACTERISTICS

The flow characteristics of a valve is the relationship between fluid flow expressed in percent of flow and valve stem travel expressed in percent of travel. Always use equal percentage characteristic valves for HTW control. Equal percentage valves compensate for the nonlinear heat output of HW heat exchangers so percent of heat output change is equal to percent of valve travel. These valves provide the best control at low flows. An industrial valve with an equal percentage plug has the required rangeability of 50 to 1.

ACTUATORS

Valve actuators must close the valve on loss of power or control air.

If the close-off rating is less than twice the expected pressure difference across the closed valve, positioners on pneumatic valves are used only, otherwise, sequencing of multiple valves is required. Electro-pneumatic positioners or separate electro-pneumatic transducers can be used to receive an electric signal (4 to 20 mA) and operate the pneumatic valve.

Electric actuators are not satisfactory for HTW valves because they are not available with a safe ambient temperature rating and the operating speed is too slow.

VALVE SIZE

Accurate valve sizing is critical for satisfactory HTW control since an oversized valve controls over only a small fraction of its total stem travel and loses sensitivity. Safety factors should be avoided but if used should be very small. See the Valve Selection and Sizing section.

The pressure drop across a control valve should be between 50 and 80 percent of the drop across that part of the piping system in which flow will vary as a result of valve action. Always use a 5 psig or greater pressure drop across the valve.

VALVE LOCATION

Always locate a HTW valve on the return side of the converter, coil, steam generator, or other equipment because it allows the valve to operate at lower pressures and temperatures. HTW is less likely to flash when the valve is located on the return side of the equipment.

INSTANTANEOUS CONVERTER CONTROL

An instantaneous converter is the standard type of converter used in nearly all HTW heating installations because of its extremely fast response. It is compact and available as standard equipment.

The HTW flows through several passes of tubing within a shell just large enough to accommodate the tube bundle. Secondary water is supplied into the shell and surrounds the tubes (Fig. 118). Only a small amount of HTW is needed to heat a large volume of secondary water and the change is nearly instantaneous. There is no room within the shell for a temperature sensing element to quickly detect the change, therefore, the temperature sensor must be located as close to the converter as possible.

There must be a provision in the control loop to close the HTW valve as soon as the secondary flow ceases. This requires proving flow in case the flow stops for a reason other than the pump is stopped by outdoor temperature, another signal, or power failure. Even if the HTW valve closes as soon as secondary water flow stops, flashing is likely to occur as there is enough HTW in the tubes to overheat secondary water. Flashing causes water hammer and possible equipment damage. If shutdown is controlled, closing the HTW valve immediately and delaying the secondary pump shutdown allows excess heat in the converter to dissipate.

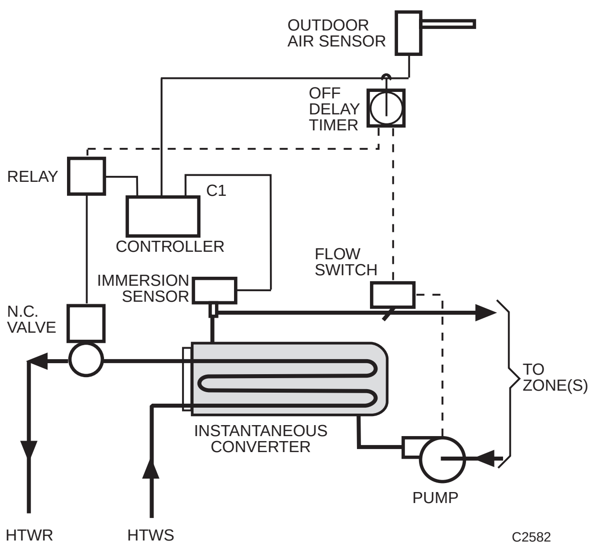


Fig. 118. Control of HTW to Constant Temperature Hot Water Converter.

A constant temperature hot water supply is used where a minimum temperature is required for all or part of the converter load. Normally a converter for space heating does not require fast response as the load changes only as fast as occupancy and outdoor air conditions change. Because of the inherent fast converter response there are several control requirements:

1. The primary sensing element must be located in outlet water as close to the converter as possible.
2. A stainless steel well that matches the element with heat conductive compound in the well must be used.
3. A primary control must be used that has integral action and integral cutout to allow a wider throttling range for stability with minimum deviation from setpoint and to eliminate integral windup.

A control arrangement for a typical zone supplied with HW from a HTW to constant temperature HW converter is shown in Figure 119. Water temperature in the converter shell must be reset from the outdoor air temperature for best control. Zones typically are also reset from outdoor temperature.

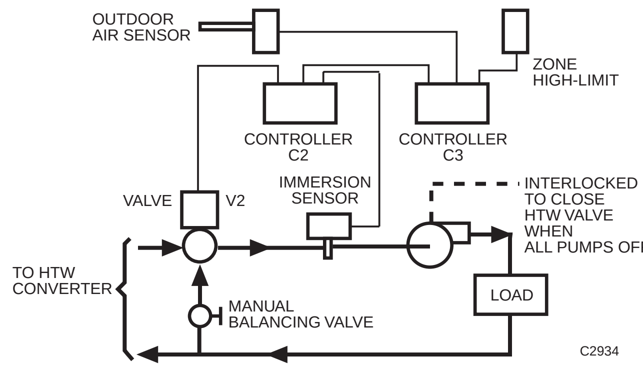


Fig. 119. Typical Zone Control of Constant Temperature HW from HTW Instantaneous Converter.

For modulating control, instantaneous converters must operate with at least half design flow. To accomplish this, reset schedules of controller C1 (Fig. 118) and C2 are arranged so that valve V2 will be between half and full open to the converter under normal conditions of operation. The reset schedule for the converter is slightly higher than the temperature normally required in the zones so that most of the water to supply the needs of the zones must come from the converter rather than the bypass.

C2 as the primary control prevents C3 from positioning V2 so the return water bypasses the converter. If flow is cut off through the converter, the sensor located in the outlet piping rather than within the shell cannot control V1 in a stable manner.

The use of an instantaneous converter (Fig. 120) for heating water supplied directly to an air heating coil provides fast response. Control system selection problems are the same as for the control of a coil supplied with low or medium temperature water except that a high quality valve with fast control response is required for HTW.

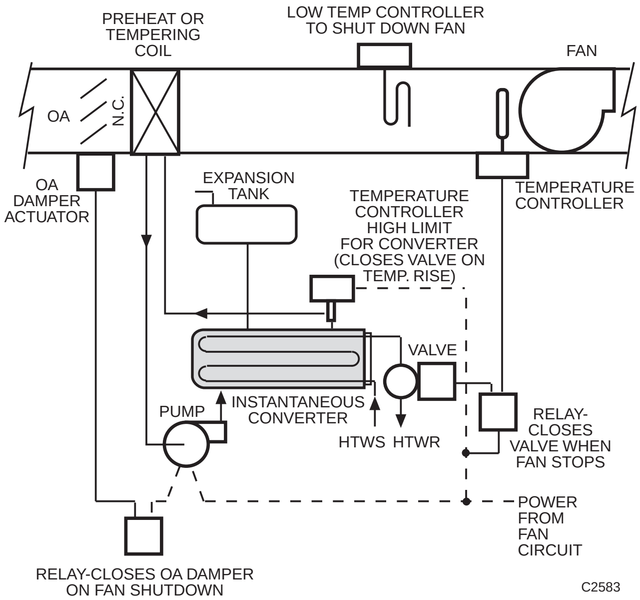


Fig. 120. Instantaneous Converter Control for HTW to Glycol Solution for Outdoor Air Preheat.

Where coils are exposed to freezing temperatures, as in a 100 percent outdoor air supply unit, the cost of installing a foolproof control system plus necessary duct work and dampers make the use of a water supplied coil questionable. Flow to the coil should be constant with variable temperature. Better protection against freezing is obtained by use of a heat transfer fluid such as a glycol solution.

Glycol solutions used with HTW heating must be closely monitored. Solutions of ethylene or propylene glycol assume the corrosivity of the water from which they are prepared and are oxidized by air into acidic products. The deterioration rate of inhibitors and stabilizers added to commercial glycol is influenced by temperature, amount of air in the solution, and, to some extent, the metals in the piping system. Heat exchanger surfaces should be kept below 285F as temperatures above 300F accelerate deterioration of inhibitors.

If HTW flow is stopped while the air handling unit continues to operate with below freezing entering air temperature, heat transfer solution returning to the converter could cause the converter to freeze. A low-limit duct temperature controller is recommended to stop the fan if a low air temperature leaving the coil is sensed.

HTW COILS

Figure 121 shows an acceptable method of mixed air control when there is a possibility of freezing air passing through the heating coil. Face and bypass dampers control the heat output of the coil during normal operation. When the outdoor temperature rises so that no heat is needed from the coil, valve V1 closes preventing overheating from the down stream side of the coil. The low-temperature limit controller senses the air temperature leaving the coil and opens the valve on sensing a freezing temperature. Use of a glycol solution as shown in Figure 120 is recommended.

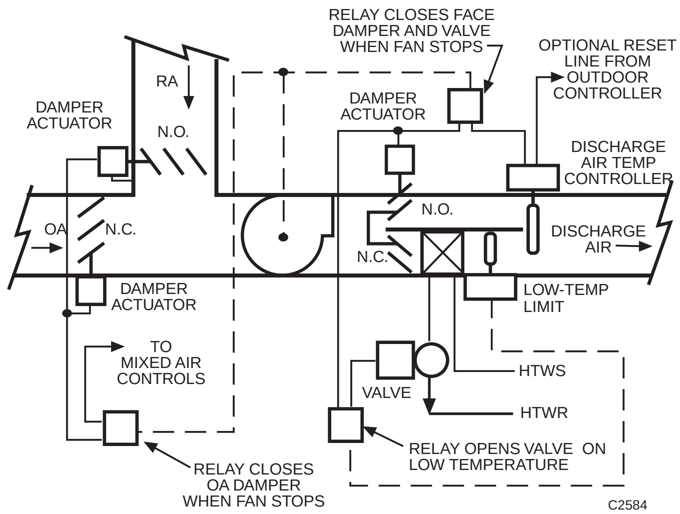


Fig. 121. HTW for Mixed Air Reheat.

HTW SPACE HEATING UNITS

Unit heaters, convectors, radiation, and radiant panels can be supplied directly with HTW. This allows use of smaller units and smaller piping and eliminates intermediate heat exchangers. Most of the additional control cost is in the valves suitable for HTW as the small unitary control valves normally used on low and medium temperature heating systems are not able to stand the pressures and temperatures of HTW. Actuator design, body ratings, disc materials, packing gland design, and lack of low flow ports are all likely limitations for this type of valve. Flanged connections are required as union connections do not seal well enough for HTW applications.

These units are most often used in storage areas, shops, hangers, or other areas where people do not work in close proximity to the units. Where people do work near HTW supplied unitary equipment, some type of limiting control is required to reduce surface and/or discharge temperatures.

STORAGE CONVERTERS

A storage converter is most often used for heating domestic water (Fig. 122). It can also be used for building heating. Separate converters are required for domestic water and building heating (Fig. 123). Heating system water is treated and recirculated while domestic water is used and replenished.

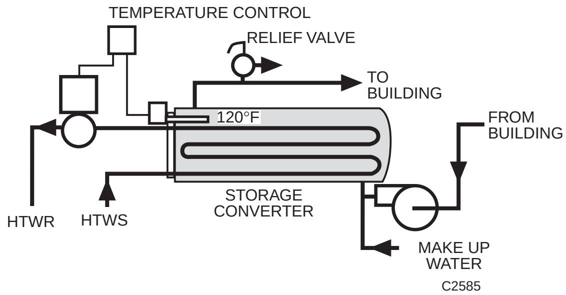


Fig. 122. Direct HTW to Domestic Hot Water.

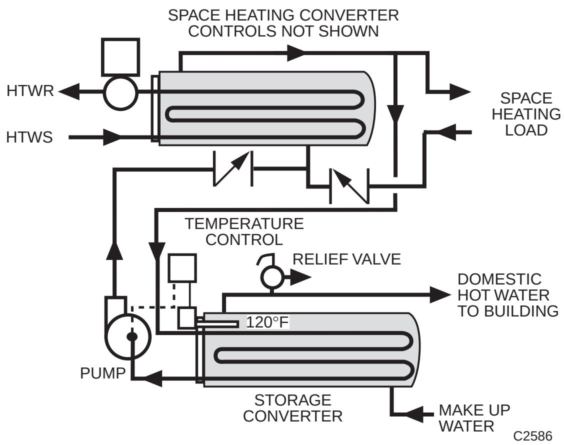


Fig. 123. Indirect HTW to Domestic Hot Water.

In a storage converter, the tube bundle is immersed in the bottom of a storage tank (shell) with capacity to provide for large intermittent demands. There is little lag in transferring heat from the tube bundle but because of the large capacity of secondary water in the shell, the system provides more stable control. Rate of change might be 20 to 50F per hour. Large installations frequently employ a system circulating pump to assure quick availability of hot water at remote usage points and help eliminate stratification in the tank.

MULTIZONE SPACE-HEATING HTW WATER CONVERTERS

Figure 124 shows a storage converter application. It is controlled as a conventional converter with only the HTW valve and pump interlock having special requirements because of the heating medium.

Because the response rate of a storage converter is slow, the storage capacity must be sufficient to supply any sudden demands. Warm-up is the demand period where the converter recovery rate becomes the determining factor for length of time required for warm-up. Normally all other changes in a building heating system are gradual.

HTW STEAM GENERATORS

Using the central HTW boilers for steam production is not recommended. The HTW system is best used to produce steam by locating steam generators at the point of need.

The steam generator (Fig. 125) is designed so that the minimum water level covering the tubes takes up 60 percent or less of the volume of the shell. The remaining 40 percent for steam is sufficient to avoid water carry over. A water eliminator at the steam exit removes most water from the steam.

Flash converters convert HTW to steam by reducing the pressure. They are not satisfactory steam generators because water is carried with the steam and control is less stable.

Control of a steam generator is simpler because pressure changes are sensed immediately and corrections in valve position are made quickly to maintain the desired steam pressure.

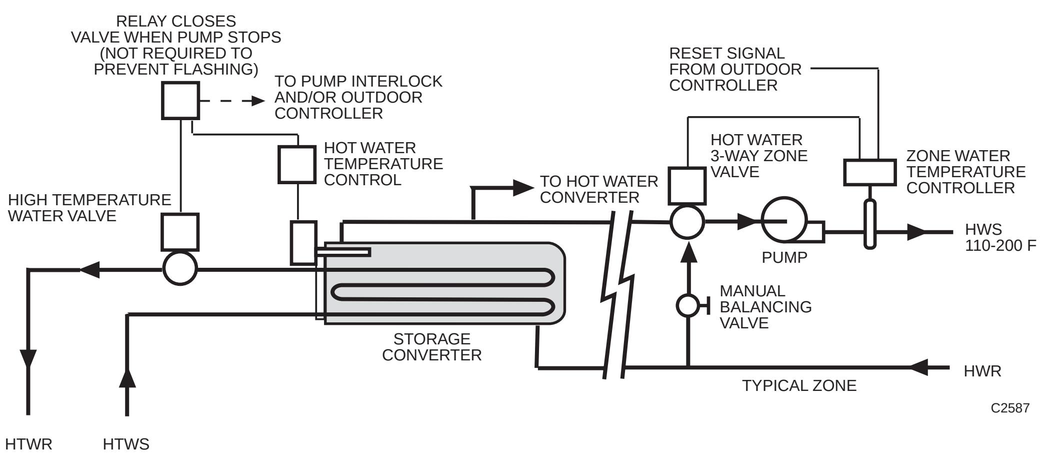


Fig. 124. HTW to HW Multizone Storage Converter.

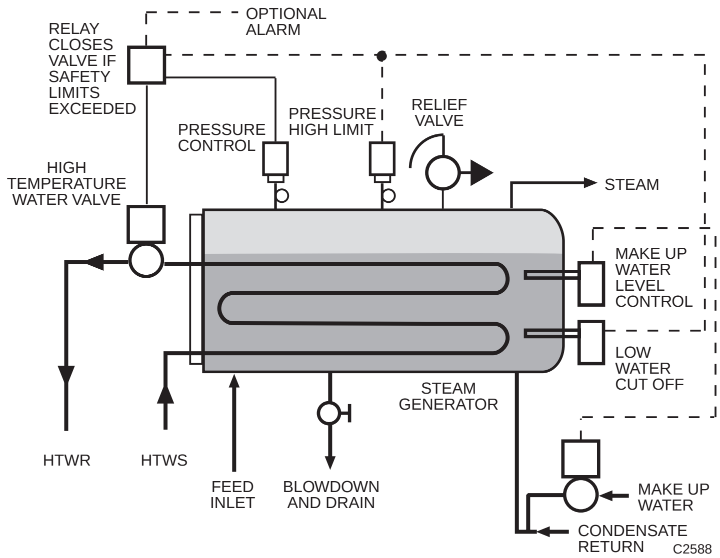


Fig. 125. Steam Generator Using HTW.

DISTRICT HEATING APPLICATIONS

INTRODUCTION

District Heating (DH), refers to a method of supplying heat to buildings, factories, and other facilities from a central source. The basic structure of District Heating Systems is shown in Figure 126. A DH System consists of one or more heat generating facilities, a widely spread heat distribution network, and substations between the distribution network and the consumer. Heat is generated by the heat source(s) and transferred to an appropriate heating medium such as hot water or steam. The heating medium is transported via pipelines to a substation. In the substation heat is transferred to the individual heating systems of the end-users.

Space heating, space cooling, and domestic hot water supply represents the largest part of energy consumption (75%) in buildings and industries. This demand is met mainly by fossil fuels and electrical energy. At the same time a vast amount of waste heat is discharged into the atmosphere by power and waste incineration plants and other industrial processes. The efficiency of current power plants does not exceed 50%.

District heating brings the waste heat directly to the customers as a salable product. This makes individual furnaces redundant. Additional advantages result from higher efficiency of central heat generation, lower emissions, and the capability of fuel diversification by using the fuel with the lowest price and the best availability. A central energy supply, based on combined heat and power generation has an overall efficiency of up to 80%. Additionally it shows a considerable emissions reduction from reduced fuel consumption.

HEAT SOURCES

Sources of heat to supply the network include waste incineration plants, boiler houses, heat pumps, and waste heat from electric power generating plants, steel foundries, or similar industrial processes.

A combined heat and power plant (CHP) which generates electricity using a steam turbine or an engine is probably the most common heat source. It heats the heating medium in the distribution network using the exhaust gases leaving the turbine.

Because these systems are parts of an industrial processes, control components and systems are typically industrial standard. Control components and systems are chosen to meet the requirements of the heat source and the specifications.

THE DISTRIBUTION NETWORK

Pipelines transfer heat from where it is generated to the consumer. Depending on the heat source, the distribution network generally consists of pairs of supply and return pipes. Hot water flows continuously through the supply pipes to the substations, heats the secondary fluid in heat exchangers, and returns to the heat source through the return flow pipes.

The distribution network facilities also include booster pumps and pressure reducing stations, mixing (temperature reducing) stations, key-points with valves to turn on or shut off branch lines and a large number of measuring points distributed over the entire network.

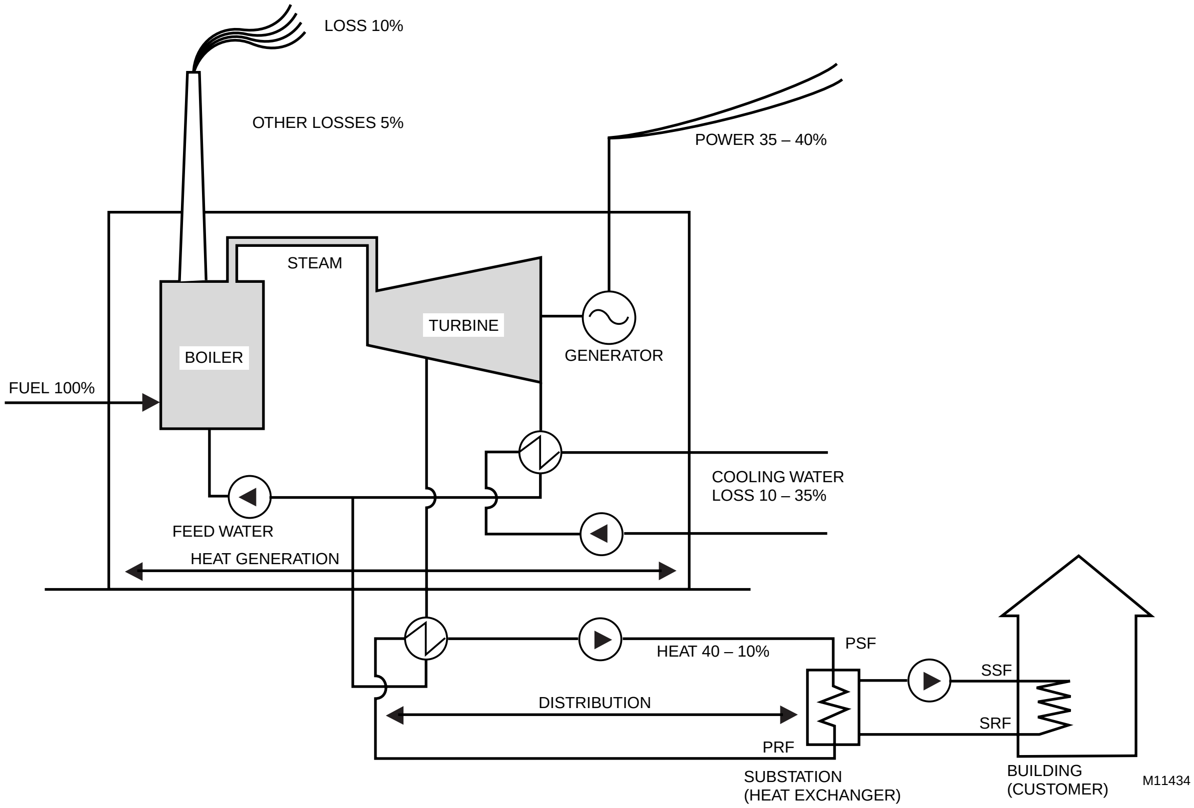


Fig. 126. District Heating Network.

The transfer medium is generally hot water at temperatures up to 400F and pressures up to 290 psi. The optimum operating conditions, temperature and pressure, depend on the structure and dimensions of the network and the heat source. The water temperature is generally limited to 265F on the supply side. In fact, many networks now keep the supply temperature under 180F allowing use of lower cost equipment and fewer safety devices. See HOT WATER DISTRIBUTION SYSTEMS and HIGH TEMPERATURE WATER HEATING SYSTEM CONTROL.

Outdoor Air Temperature Control reduces the supply temperature in a network with increasing outdoor temperature. The Return Water Temperature in a DH Network is controlled to not exceed either a fixed or variable temperature level.

THE SUBSTATION

Substations house heat exchangers which transfer the required heat from the distribution network to individual building heating networks (customer). In some cases, a large heat exchanger station (HES) is required between different types of networks. The HES transfers heat from high power primary networks to smaller secondary networks with lower temperatures and pressures.

A DH system only works efficiently if all components are matched to one another. The heat source must deliver the required heat at the lowest cost level, 365 days a year. Heat demand as a function of the day of the week, time of day, and weather conditions must be predicted in order to manage the heat supply in the most efficient manor. In combined heat and power (CHP) plants the electrical power generation must also be considered.

Managing a system requires highly sophisticated dedicated software and highly reliable control equipment. The software, which includes demand prediction and total energy management functions, plays an important role in the efficiency of CHP systems.

ABBREVIATIONS

CHP	—	Combined Heat & Power.
DH	—	District Heating.
DHW	—	Domestic Hot Water.
ESD	—	Emergency Shut Down systems
HES	—	Heat Exchanger Substation.
HEX	—	Heat Exchanger.
HTS	—	Heat Transfer Station.
OAT	—	Outdoor Air Temperature.
PRF	—	Primary Network Return Flow.
PSF	—	Primary Network Supply Flow.
RT	—	Return Water Temperature.
SRF	—	Secondary Network Return Flow.
SSF	—	Secondary Network Supply Flow.

DEFINITIONS

Booster Pump station: Maintains sections of a network (return or supply pipe) in required pressure conditions. Used in long networks or highly elevated sections.

Combined Heat & Power: Combines production of electricity and heat for space (buildings) and processes (industrial)

Domestic Hot Water: Water controlled at a constant temperature suitable for use in showers and hand washing stations.

Differential Pressure Controller: Prevents too low or too high hydraulic pressure at the building heat exchanger substation. This improves the control performance of the station.

Heat Exchanger Substation: Compact Station, controls the radiator and DHW loop in a building.

Heat Exchanger: Also called a convertor, transfers heat from the primary network to a secondary network.

Heat Surface Factor: A value corresponding to the effectiveness of radiating surface. The value depends upon the shape, size, and color.

Heat Transfer Station: Controls flow, pressure and temperature in sections of a primary network.

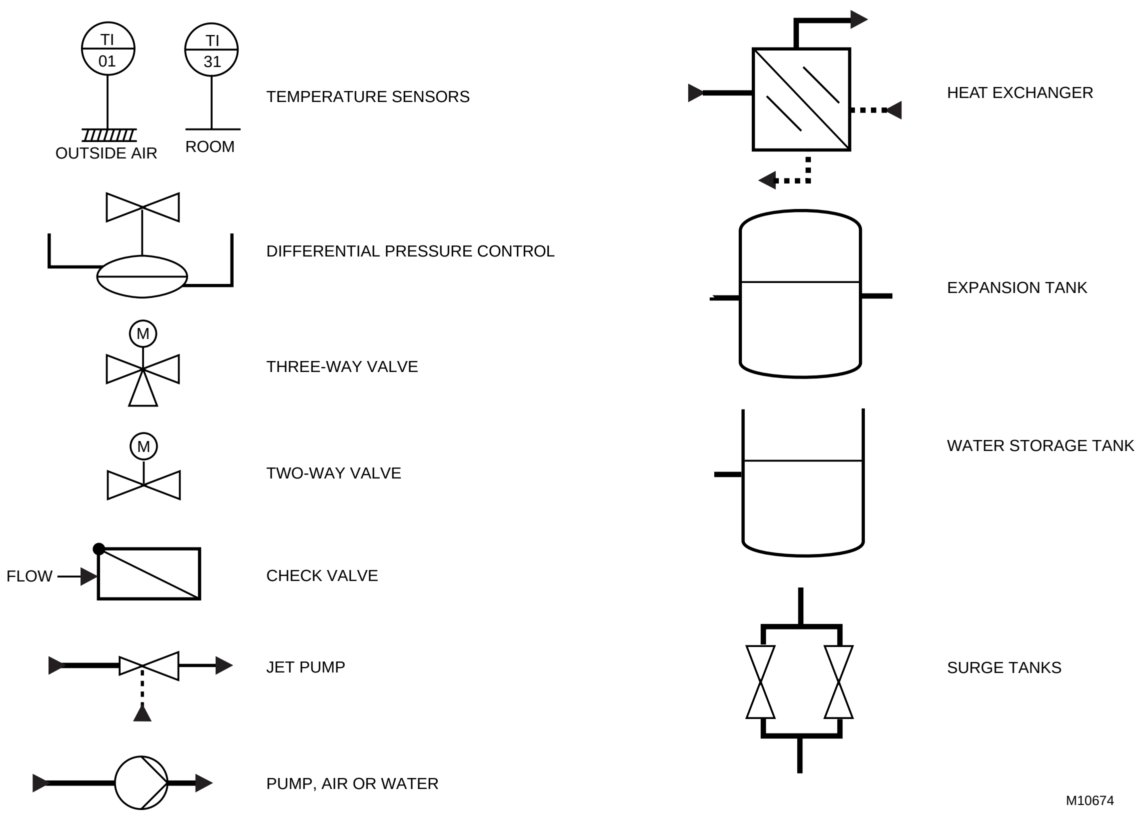
Primary Network: Supply and return pipe for heating medium (hot water or steam) from the heat/power plant to the consumer heat exchanger.

Secondary Network: Supply and return pipe for heating medium (actuator) from heat exchanger to radiators or tabs.

SYMBOLS

Since District Heating is more common in Europe, District Heating is based on information supplied by personnel in Europe.

The symbols used in Figures 126 through 143 are those commonly used in Europe and are supplied here for convenience.



SYSTEM CONFIGURATION

HEAT GENERATION

Excess heat is produced in steam electrical power generating stations which can be reclaimed as in Figure 126 and sold to provide heat for other uses. Other industrial processes and waste incineration may also provide a source of excess heat which can be reclaimed. Geothermal sources and boilers are other heat sources.

HEAT DISTRIBUTION NETWORK

The distribution network is one part of a district heating system. It transports the heat from the heat source to the consumer. Heat is absorbed by the heat transfer medium, hot water or steam, at the source and delivered to the consumer.

Primary networks are one- or two-pipe distribution systems. In a two-pipe system the hot pipe, or supply line, transports the water or steam to the substation, heat is drawn from the network at the substation and transferred to the consumer's side, then the cooled water or condensate flows through the return line back to the heat source to be reheated.

A one-pipe system transports the water to the consumer, heat is drawn from the network at the substation and transferred to the consumer's side, then the cooled water is discharged to a drain. This system is used with a geothermal hot water source.

HOT WATER SYSTEM CONTROL

Variable speed circulating pumps move the water through the primary network providing the needed differential pressure in hot water systems. An additional pressure holding system is installed to keep the absolute pressure at the required level.

There are two control strategies based on outdoor air temperature for transferring a requested amount of heat to the consumer:

1. Constant Flow Control maintains a constant supply flow rate and varies supply flow temperature.
2. Variable Flow Control maintains a constant supply flow temperatures and varies supply flow rate.

The optimal temperature/pressure ratio depends on the length and structure of the network, actual load, outdoor air temperature, and pipe insulation. Because of the many variables, a combination of both Outdoor Air Temperature Control and Variable Flow Control or Variable Temperature Control is often used.

STEAM SYSTEM VS HOT WATER SYSTEM

Steam networks differ mainly in the following points from hot water systems:

– No pumps are required, the pressure difference between boiler and consumer drives the movement of the steam.
– Condensate traps are required approximately every 0.3 miles.
– The return line diameter is much smaller because condensate takes up less space than steam.
– Heat losses are significantly higher than hot water systems.
– The heat storage capacity of steam is lower than hot water.
– Maintenance costs are higher than hot water.

HOT WATER PIPELINE SYSTEM

Hot water systems must be protected against high pressure peaks which could damage the pumps and pressure drops below the evaporation point which results in the water changing to steam. Common types of preinsulated pipelines must be protected against temperatures exceeding 265F.

Factors affecting efficiency include optimal temperature/ pressure ratio with respect to the length and capacity of the network, temperature differences between supply and return flow, heat and water losses, as well as friction between the water and pipe wall. Higher temperatures cause greater heat loss by radiation and conduction while greater differential pressure in the network produces heavier pump loads. Every network has a different optimum value for the supply and return pipelines. Use of friction reducing chemicals to decrease the friction losses in the pipeline can reduce the pump power required. Extracting as much heat at the substations as possible also reduces pumping costs.

Additionally the water in the entire district heating system pipelines also serves as a large heat accumulator and helps compensate for peak loads or short periods of low heat generation.

BOOSTER PUMP STATION

In large pipeline systems, using a single main pump requires a large differential pressure (pump head) to overcome the friction in the network. Figure 127 shows a profile of a single pump system. Decentralized pumps (booster pump stations) avoid this and keep the pressure in every pipeline section within the required levels.

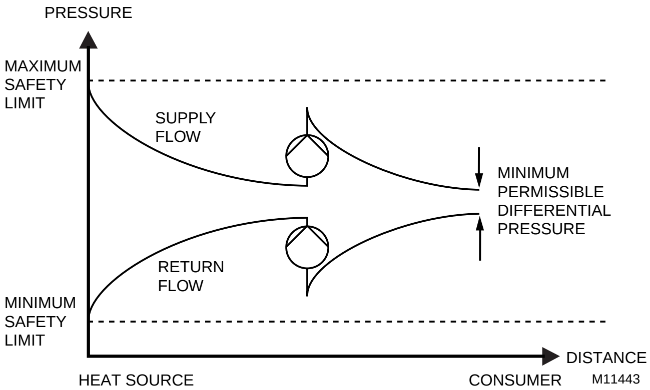


Fig. 127. Network Pressure Profile.

A booster pump station overcomes the pressure drop in areas with considerable differences in altitude (to overcome 165 ft difference in altitude a pressure of 71.5 psi is required). They are often applied where going under or over obstacles is necessary. Equip the pipeline with an emergency shut down system (ESD) in areas with considerable differences in altitude to protect the system from high pressures in case of power failure.

PRESSURE REDUCING STATIONS

A pressure reducing station is the counterpart to the booster pump station. A pressure reducing station is used in lines located in mountainous areas to protect the pipeline from over pressure and to keep the pressure in the return line lower than the supply line. For this application pressure reducing valves are control valves.

MIXING STATION

A mixing station (Fig. 128) is used in hot water networks. It is a variable speed (mixing) pump which mixes cooled return flow directly into the supply flow to reduce the supply flow line temperature to the required level. These facilities are used to provide different maximum temperatures in the network pipeline.

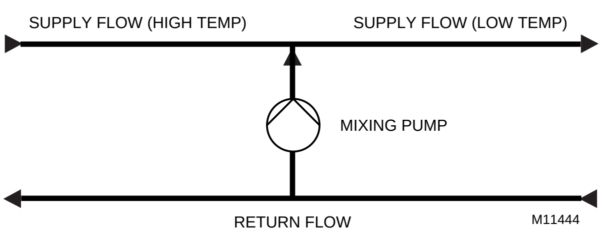


Fig. 128. Principle Of A Mixing Station.

KEY POINTS

Key points are locations where pipelines branch off (Fig. 129). They consist of valves for supply and return flow which can separate the branch line from the main line. If a branch line is shut down, the pressure ratio in the entire network is affected. To prevent pressure spikes in the main system, the shutdown must be performed slowly and carefully. Often two different sized surge tanks in parallel are used to damp the pressure peak during shutdown. Usually key point temperatures and pressures are monitored.

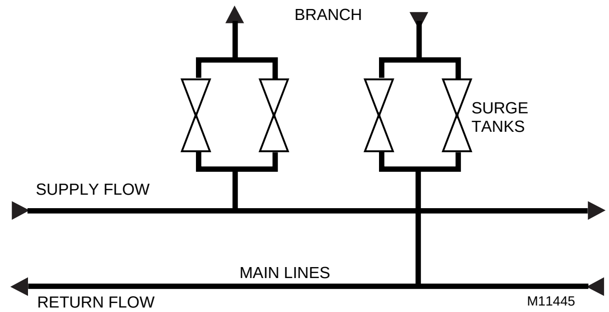


Fig. 129. Typical Key Point.

HEAT TRANSFER SUBSTATIONS

In general heat transfer substations link district heating networks with the consumer. The consumer side can be either another network or the end user.

Heat transfer can be either direct and indirect. Direct transfer uses mixing valves, jet pumps or two way valves to supply the heating medium directly to the consumer. Indirect substations use heat exchangers and physically decoupled or independent heating circuits.

Direct Heat Transfer Substations

The main parts of a direct substation are:

- Primary side.
- Heat flow regulation unit.
- Circulating pumps.
- Secondary side.

Direct heat transfer substations:

- Transfer the required heat from the supply (primary) side to the consumer (secondary) side.
- Meter heat.
- Provide safety functions to protect consumer and equipment against overheating, frost, and harmful agents in hot tap water.
- Provide optimization functions to reduce energy consumption to the lowest possible level.

Functional principles:

The primary side consists of the supply and return lines, plus necessary pressure reducing, regulating, and safety equipment.

This is self-regulating equipment which provides a given differential pressure, absolute pressure reducing, and safety close off functions.

The heat regulation unit (Fig. 130) provides the required temperature by controlling the primary flow (A) or by mixing the cooled return water with supply water (B). Different configurations with two and three way valves can be used. In the flow control configuration a fixed speed circulating pump increases the pressure in the return line above the supply flow pressure. In the temperature control configuration either a jet pump or a three way valve is used.

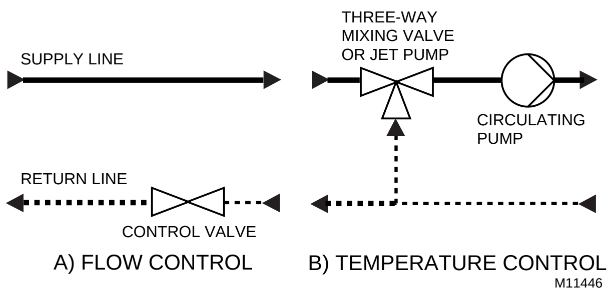


Fig. 130. Two- And Three-Way Valve Configurations for a Heat regulation Unit.

Jet pumps use the effect of injection, making a mechanical pump unnecessary, thereby, saving electrical energy. However, adapting and dimensioning jet pump applications to fit operating conditions is difficult.

Control loops used in a direct substation:

- Supply flow temperature reset on outdoor air temperature.
- Return temperature limit.
- Time schedule functions.
- Night setback and frost protection.

Small Substation For Multiple Family Buildings

Figure 131 shows a typical direct heat transfer substation.

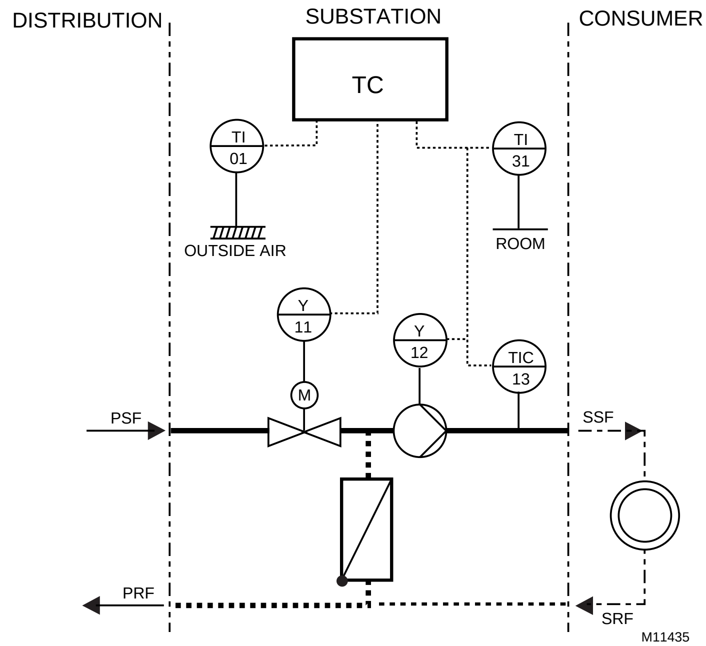


Fig. 131. Small Direct Heat Transfer Substation for Multiple Family Buildings

Table 9. Description of Figure 131 Reference Points.

Reference	Description
TC	Controller
TI 01	Outdoor air temperature sensor
TI 31	Room temperature
TIC 13	SSF temperature
Y 11	Actuator control valve PSF
Y 12	Circulating pump

Control Strategies:

1. Control valve Y 11 maintains the SSF temperature as dictated by TIC 13 which is reset by the outdoor air temperature. Room temperature sensor TI 31 shifts the reset schedule up or down. Controller TC provides night setback and other unoccupied programs.
2. Pump Y 12 provides constant SSF flow through the check valve.
3. Summer operation shuts down the entire station. However, it is recommended that the summer function include a function to exercise the pump and therefore all the devices once a week.

Indirect Heat Transfer Substations

Indirect substations use heat exchangers and physically decoupled or independent heating circuits. Applications range from small substations for a one family house to large substations for industrial types of networks. Three applications for heat transfer substations with different heat exchanger configurations, primary flow and/or the differential pressure is controlled by control valves, heat flow controlled by modulating control valve to vary the primary flow, and circulating pumps to provide the secondary supply and return flow, follow:

Indirect heat transfer substations:

– Transfer the required heat from the supply (primary) side to the consumer (secondary) side.
– Meter heat.
– Provide safety functions to protect consumer and equipment against overheating, frost, and harmful agents in hot tap water.
– Provide optimization functions to reduce energy consumption to the lowest possible level.
– Provide hydraulic separation between the high-temperature high-pressure system and the low-temperature low-pressure system.

The main parts of an indirect substation are:

– Primary side.
– Heat exchanger.
– Circulating pumps.
– Expansion and storage tanks.
– Feed water facilities.
– Secondary side.

Functional principle:

The primary side of the substation (Fig. 132) contains the differential pressure control and a normally closed temperature control valve with safety spring return, if the secondary medium temperature exceeds 212F. The differential pressure control is used in large networks where over time significant pressure differences exist.

The primary supply flow (from the district heating network) enters the heat exchanger, transfers the heat to the secondary supply flow, and returns to the heat source through the primary return.

In large networks with distributed pressure and temperature parameters the primary flow and/or the differential pressure is controlled by electric or self regulating control valves. This ensures a constant inlet pressure differential regardless of the actual pressure differential pressure in the network. Separate control valves or valves with combined functionality in the return or supply flow ensure this.

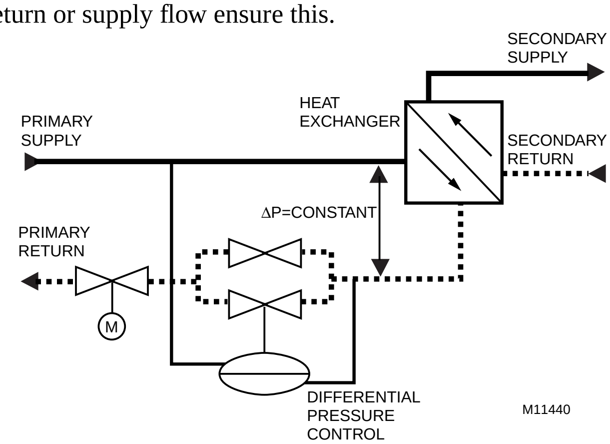


Fig. 132. Primary Side of Indirect Heat Transfer Substation.

The heat exchanger (HEX) separates both circuits and maintains the heat flow between primary and secondary side (Fig. 133). Both plate or tube type heat exchangers are used in either parallel or series configurations. The heat flow is controlled by modulating the control valve position varying the primary flow.

The secondary supply flow temperature setpoint is usually set from the outdoor air temperature during the day and set back during the night. A high limit safety device reduces the supply flow temperature, if the return flow temperature exceeds high limit setpoint.

In addition HEXs are used for domestic hot water either directly or via a charge pump and hot water storage tanks.

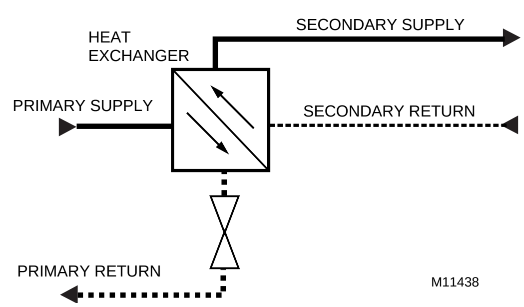


Fig. 133. Heat Exchanger Portion of Indirect Heat Transfer Substation.

Control loops:
Circulating pumps provide the differential pressure between secondary supply and return flow. In facilities where the load varies over a wide range, variable speed pumps are used to meet the pressure requirements and save electrical energy.

The absolute pressure in the secondary circuit must be kept within the safety limits (upper limit to avoid pipe and pump damage, lower limit to avoid water flashing to steam and cavitation).

In general three different configurations are used:

– Open systems using water column static pressure and expansion tanks.
– Closed systems (Fig. 134) using pressurized air and expansion tanks (static pressure holding system).
– Dynamic systems using a combination of pressure holding pumps and vent-valves.

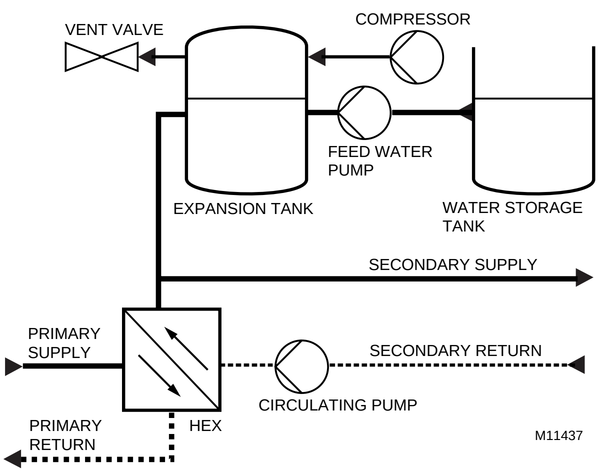


Fig. 134. Feed Water System in Secondary of Indirect Heat Transfer Substation.

Feed water facilities maintain the water level in a heating system. Two ways of feeding water exist depending on the requirements placed on the water quality.

The first method takes water from the primary circuit. This water is of sufficient quality for use in the secondary circuit. Pressure reducing, cooling, and safety (close off) equipment may be necessary depending upon the pressure and temperature level in each circuit.

The second method uses treated tap water. Use of untreated water is not recommended as it can coat system with minerals precipitated from the water and dissolved oxygen can corrode the piping. This method can also be used as a back up to the first method.

Hybrid Heat Transfer Substations

This combines direct heating with indirect domestic hot water supply (DHW) as shown in Figure 135. Sizes range from small substations to large heat exchanger substations supplying several blocks of buildings.

As described before, there are usually three main parts to the substation:

– Primary side including pressure/flow regulating equipment.
– Heat exchanger or hot water storage tank (chargeable) for DHW and heat flow regulating equipment for heating.
– Circulating pumps for secondary side (heating) and for DHW.

The primary side includes shut off valves, differential pressure, and flow control equipment provide safety functions and a required differential pressure or flow. Heat metering is also included. The water flow here is divided into two parts.

The HEX transfers the heat from the primary system to the DHW loop. A valve placed in series with the HEX controls the flow. The other part of the primary flow enters the secondary circuit and transfers the heat to the consumer directly. The supply flow temperature is controlled by one three-way valve, two two-way valves, or one jet pump making an additional circulation pump redundant.

In cases without a jet pump, circulation pumps provide the needed differential pressure.

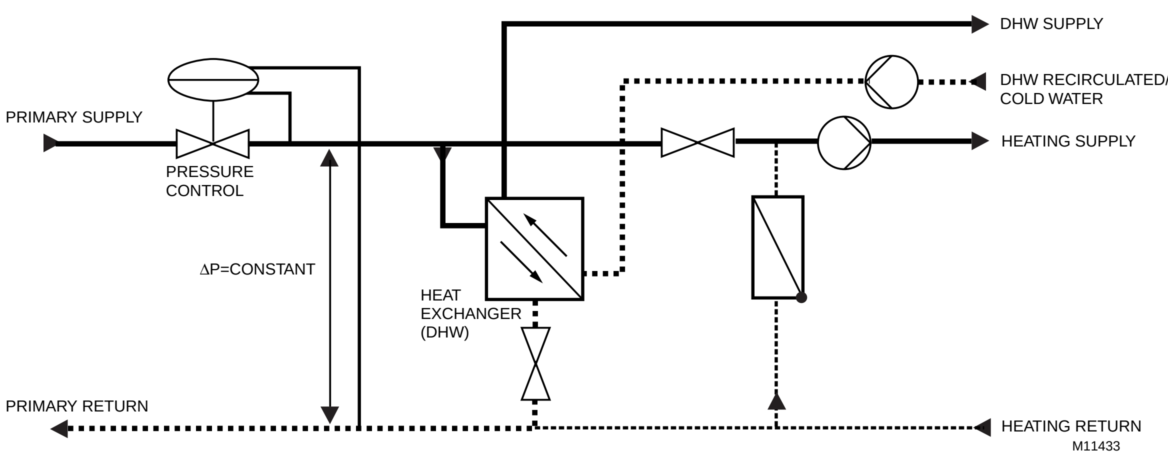


Fig. 135. Hybrid Heat Transfer Substation.

CONTROL APPLICATIONS

DDC CONTROL OF HIGH PERFORMANCE SUBSTATION

Table 10. Description of Figure 136 Reference Points.

Reference	Description	Reference	Description
LIC 56	Water level supervision & control	TI 31	PRF temperature (HEX 12)
LIC 63	Water level supervision & control	TI 47	SRF temperature
PDIC 14	Primary differential pressure	TIC 28	SSF temperature (HEX 1)
PDIC 45	Secondary differential pressure	TIC 38	SSF temperature (HEX 12)
PI 13	PSF pressure	TIC 44	SSF temperature
PI 16	PRF pressure (internal)	TSA+ 22	High temperature limit Primary Return Flow Heat exchanger 1 (PRF HEX 1)
PI 46	SRF pressure	TSA+ 32	High temperature limit (PRF HEX 12)
PI 110	PRF pressure (outdoor)	TSA+ 25	High temperature limit (SSF HEX 1)
PIC 57	SSF pressure	TSA+ 35	High temperature limit (SSF HEX 12)
PLC	S9000 Supervisory Control	TZA+ 26	Safety temperature limit (SSF HEX 1, locked)
PSA+ 52	Safety pressure limit (refill pipe)	TZA+36	High temperature limit (SSF HEX 12, locked)
PSA+ 58	Safety pressure limit (SSF)	TZA+ 41	Safety temperature limit (SSF total, high)
PZA+ 24	Safety pressure limit (high)	Y 11	Actuator valve Primary Supply Flow (PSF)
PZA+ 34	Safety pressure limit (high)	Y 17, Y 18	Actuator control valve (PDPC)
PZA+ 43	Safety pressure limit (SSF high)	Y 19	Safety close off valve
PZA-27	Safety pressure limit (low)	Y 23	Actuator control valve
PZA-37	Safety pressure limit (low)	Y 29	Damper (secondary flow, HEX 1)
PZA–42	Safety pressure limit (SSF total low)	Y 33	Actuator control valve
QIR 53	Metering refill quantity	Y 39	Damper (secondary flow, HEX 12)
QIR 62	Cold water metering	Y 48, Y 49	Secondary flow circulating pumps
QIR 111	Heat metering	Y 51	Actuator valve (water refill from PRF)
TI 01	Outdoor air temperature	Y 54	Vent valve
TI 12	PSF temperature	Y 55	Compressor
TI 15	PRF temperature (total flow)	Y 61	Actuator valve (cold water refill)
TI 21	PRF temperature (HEX 1)	Y 64	Feed water pump

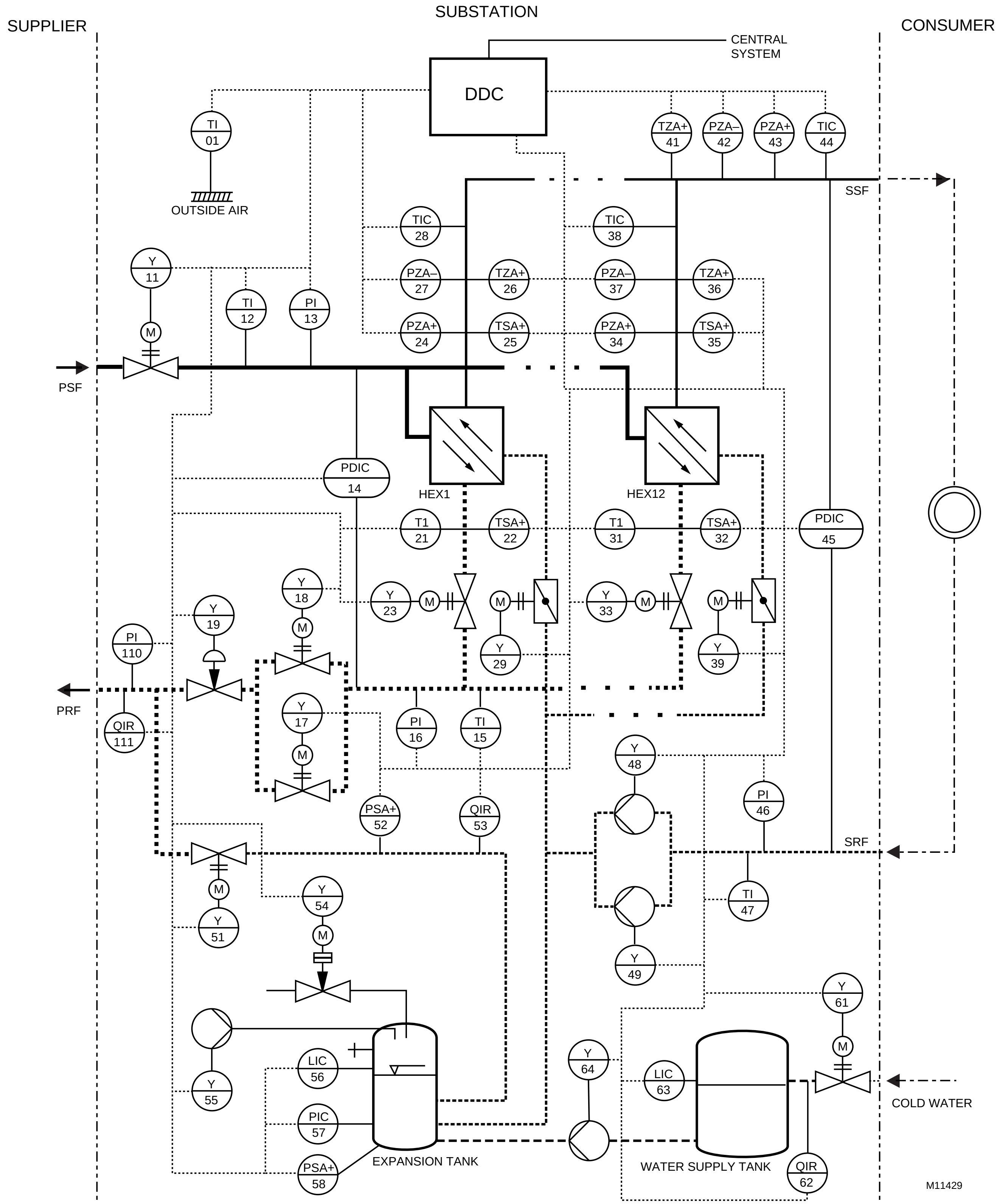


Fig. 136. DDC Control of Large High Performance Indirect Heat Transfer Substation.

Control Strategies

1. Primary Flow Loop Differential Pressure Control: Provides a constant value pressure drop across the heat exchangers. Actuator control valves (Y17, Y18) open or close sequentially to maintain the pressure drop measured by PDIC 14.
2. Secondary Loop Supply Flow Temperature Control: The SSF temperature is reset from outdoor air temperature (Fig. 137). The actual SSF temperature is measured by sensors TIC 28 and TIC 38. The regulation of the heat flow through the HEX is provided by the actuator control valves Y 23 and Y 33 in the primary return flow line of each heat exchanger.
3. Secondary Flow Loop Differential Pressure Control: Circulating pumps Y 48 and Y 49 are switched off or on and the speed varied to maintain the differential pressure setpoint. A frequency converter (fc) controls pump speed.
4. SSF Loop Pressure Control: Maintains the pressure sensed by transmitter PIC 57 within a defined range to avoid too low pressure (vaporization) and too high pressure (pump overload). Vent valve Y 54 maintains the high limit and compressor Y 55 maintains the low limit.
5. Expansion Tank Water Level Control (WLC) Loop: LIC 56 maintains the expansion tank water level. Normally PRF water through Valve Y 51 provides make-up water, in cases of a high quantity water loss, Pump Y 64 pumps make-up water from the water supply tank. The WLC protects the circulating pumps against running without water.
6. Water Supply Tank Water Level Control: LIC 63) controls Valve Y 61, providing make-up water from the cold water.
7. Secondary Return Flow (SRF) and the Primary Return Flow (PRF) temperatures are limited to provide a high temperature drop between supply and return flow.

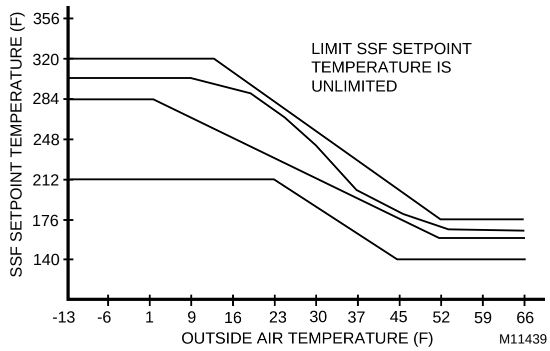


Fig. 137. Adjustment Of Temperature Curves.

DDC CONTROL OF A SMALL HOUSE SUBSTATION

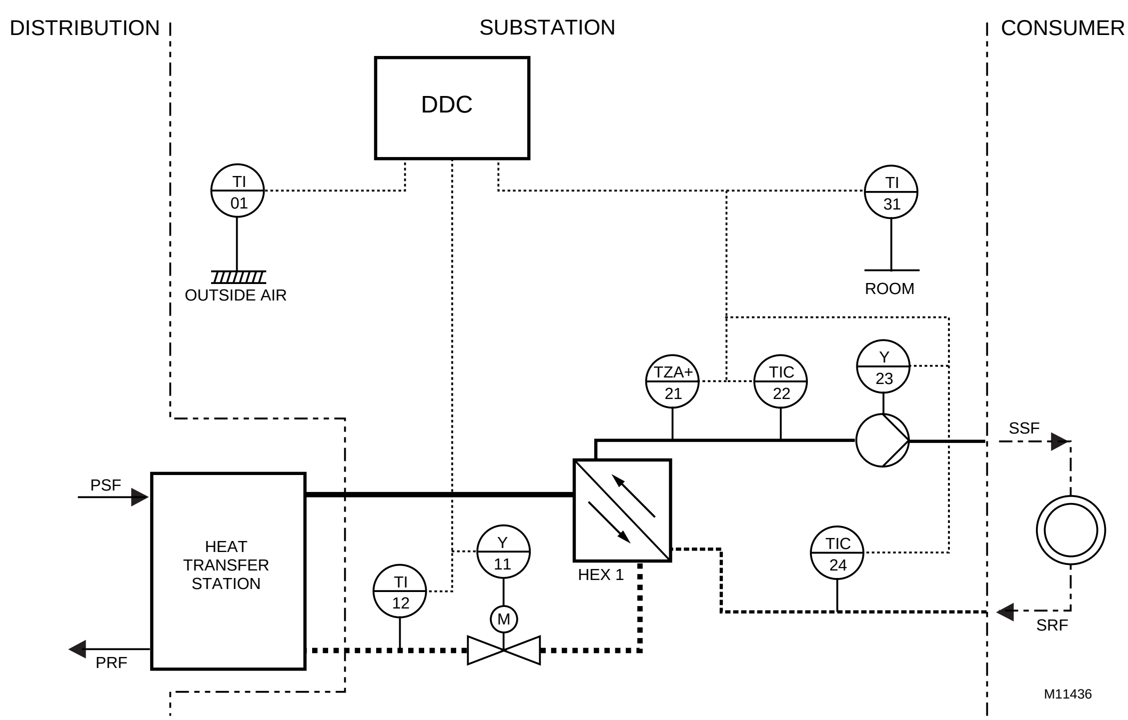


Fig. 138. DDC Control of a Small House Heat Transfer Substation.

Table 11. Description of Figure 138 Reference Points.

Reference	Description	Reference	Description
DDC	Controller	TIC 24	SRF temperature
TI 01	Outdoor air temperature	TZA+ 21	Safety temperature limit (SSF)
TI 31	Room temperature sensor & setpoint setting	Y 11	Control valve PRF
TIC 12	PRF temperature	Y 23	Circulating pump
TIC 22	SSF temperature		

Control Strategies:

1. Primary temperatures: supply up to 265F, return up to 195F.
2. Secondary temperatures: supply up to 195F, return up to 160F.
3. System provides a SSF temperature depending on the outdoor temperature according to a reset schedule (Fig. 139).
4. Heat demand dependent pump switching, with one full cycle of the pump per day to prevent seizing.
5. Diagnostic functions via connector.
6. PRF temperature limitation with fixed and variable value available. The control valve reduces the flow until the PRF temperature is below the parameter value. The setpoint can be fixed or floating with a certain ΔT below the PSF.
7. SSF temperature limitation available (parameter).

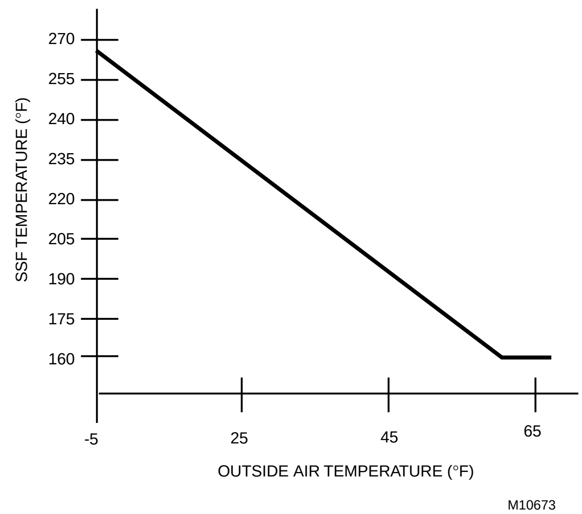


Fig. 139. SSF Temperature Reset Schedule.

CONTROL OF APARTMENT BUILDING SUBSTATIONS WITH DOMESTIC HOT WATER

DDC Control of Two-Stage Heat Exchanger for Heating and Domestic Hot Water.

Table 12. Description of Figure 140 Controls.

Reference	Description	Reference	Description
DDC	Controller	TIC 16	PRF–temperature
PI 11	PSF pressure	TIC 21	DHW temperature (supply flow)
PI 17	PRF Pressure	TIC 31	SSF temperature
PI 22	DHW pressure (supply flow)	TIC 34	SRF temperature
PI 24	DHW pressure (return flow)	Y 13	Actuator valve (SSF temperature control)
PI 28	Cold water pressure	Y 14	Actuator valve (DHW temperature control)
PI 213	DHW network pressure	Y 23	Actuator valve (DHW supply flow)
PIC 32	SSF pressure	Y 25, 26, 27	DHW circulating pumps
PIC 35	SRF pressure	Y 29, 210	DHW pressure booster pumps
QIR 02	Elec. Power metering	Y 31	Actuator valve (SSF)
QIR 15	Heat metering	Y 36, 37	Circulating pumps (secondary fl.)
QIR 214	Drinking water metering	Y 38, 39	Refill pumps (PRF to SRF)
TI 01	Outdoor air temperature	Y 211, 212	DHW pressure booster pumps
TI 12	PSF temperature	Y 310	Actuator valve
TI 215	DHW circulation, return flow temperature		

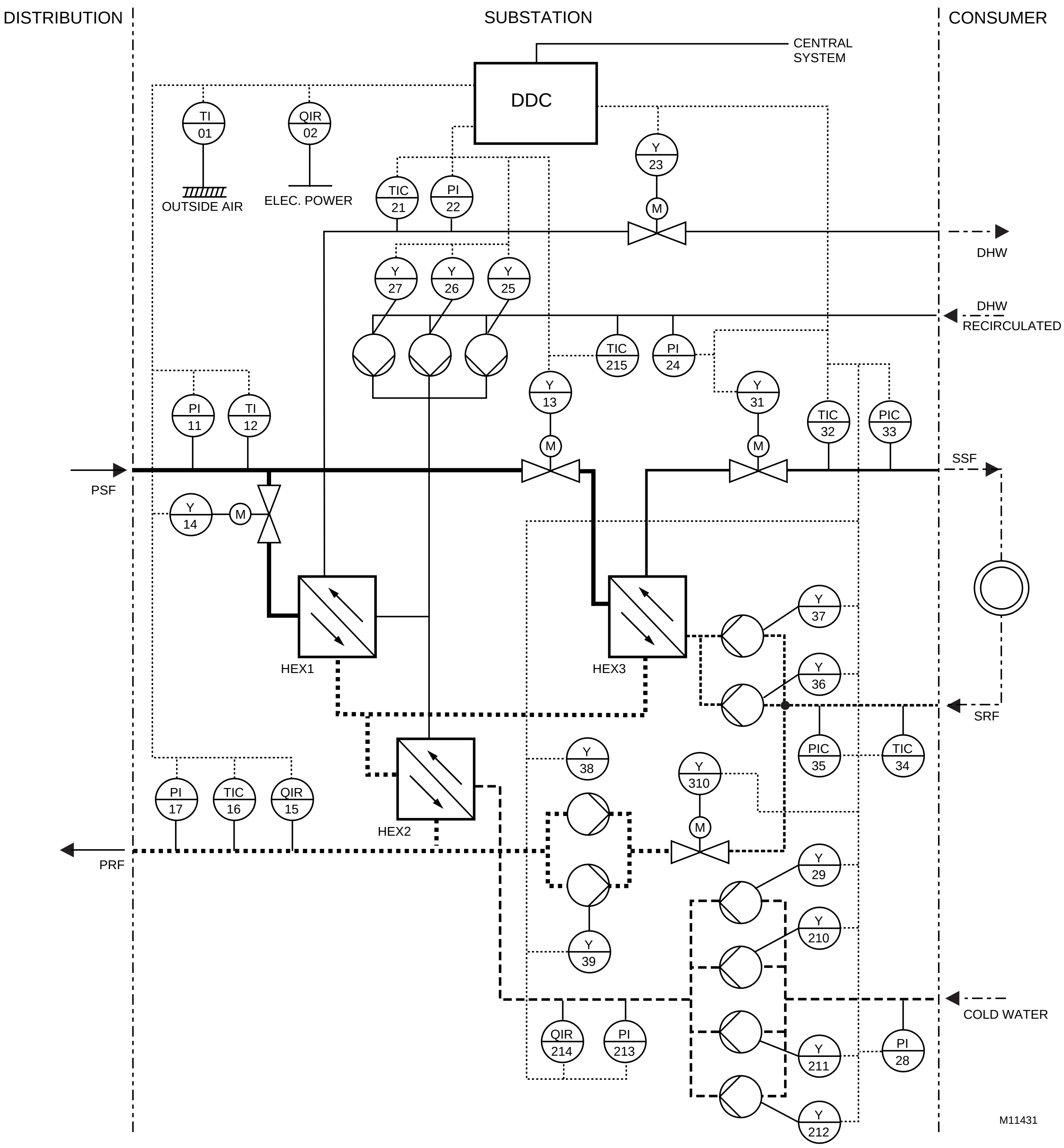


Fig. 140. DDC Control of Two-Stage Heat Exchanger for Domestic Hot Water and Heating.

Control Strategies

1. Secondary Supply Flow Temperature Control: The SSF temperature is reset from outdoor air temperature. TIC 32 provides the SSF temperature to the DDC controller. Valve Y 13 regulates flow through HEX3 primary .
2. Secondary Flow Loop Differential Pressure Control: Circulating pumps Y 37 and Y 36 are switched off or on and the speed varied to maintain the differential pressure setpoint. The pump speed is controlled by a frequency converter (fc).
3. Secondary Supply Flow Temperature Control: The SSF temperature is reset from outdoor air temperature (Fig. 141). The actual SSF temperature is measured by sensor TIC 32. The regulation of the heat flow through the HEX is provided by the actuator control valves Y 13 in the primary supply flow line of HEX3.
4. Secondary Loop Make-up Water: Pumps Y 38 and Y39 through Valve Y 310 provide make-up water from the PRF.
5. DHW Pressure Control: Booster Pumps Y29, Y210, Y211, and Y212 boost the cold water pressure to the pressure at PI 213.
6. DHW Circulating Pumps Control: Circulating Pumps Y 25, Y26, and Y27 circulate DHW through the pipes to maintain hot water temperature at the end of the DHW system.
7. DHW Supply Temperature: Valve Y 14 modulates the PSF water to HEX 1 and HEX 2 to maintain the DHW supply temperature at Y 27.

Functional Diagram

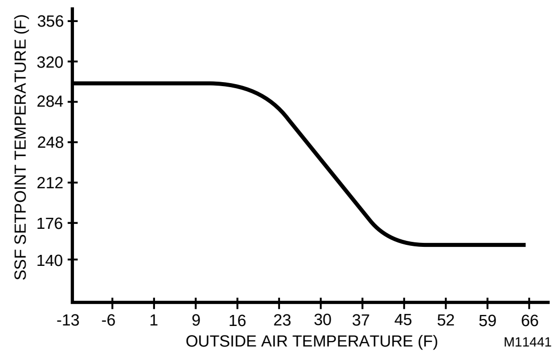


Fig. 141. Secondary Supply Flow Temperature Reset schedule.

DDC Control of Combination Heat Exchanger and Domestic Hot Water Storage Tank

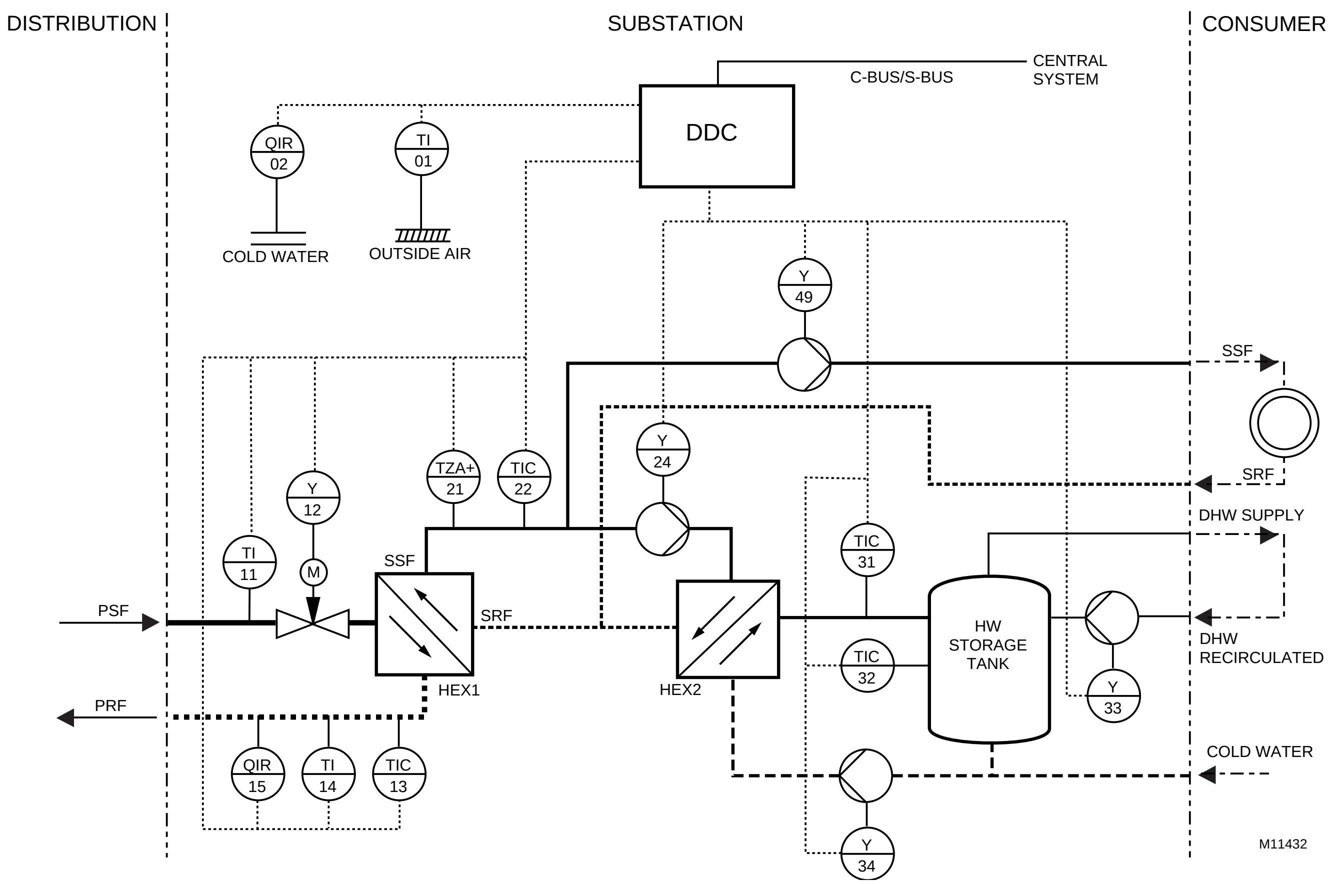


Fig. 142. DDC Control of Dual Heat Exchangers for SSF and DHW Storage.

Table 13. Description of Figure 142 Controls.

Reference	Description
DDC	Controller
QIR 02	Cold water metering
QIR 15	Flow meter (heat meter)
TI 01	Outdoor air temperature
TI 11	PSF temperature
TI 14	PRF temperature (heat meter)
TIC 13	PRF temperature
TIC 22	SSF temperature
TIC 31	Loading circuit temperature
TIC 32	HW storage tank temperature
TZA+ 21	High temperature limit (SSF)
Y 12	Control valve primary flow
Y 23	Circulating pump (SSF)
Y 24	Circulating pump (SSF)
Y 33	Circulating pump
Y 34	Charging pump (HW storage tank)

Control Strategies

1. Secondary Supply Flow (SSF) temperature control: Modulating normally-closed Valve Y 12 in the primary supply flow (PSF) and TIC 22 control the SSF temperature. The SSF setpoint is reset based on outdoor air temperature (Fig. 143). If the primary return flow (PRF) exceeds the TI 14 temperature limit, the SSF 1 temperature reset schedule is adjusted to a lower temperature schedule. If the SSF temperature exceeds the high limit setpoint of hard wired thermostat (TZA+ 21), power to valve Y 12 is shut off closing the valve.
2. DHW temperature control: Valve Y 12 in conjunction with TC 32 maintains the HW storage tank water temperature. Thermostat TIC 32 supplies the HW storage tank loading cycle start point. During the loading cycle, SSF water heats HEX 2, other loads are switched off, and the pumps Y 24, Y 34 load the HW storage tank. Temperature sensor TIC 31 turns loading operation off.
3. DHW Circulating pump control: Circulating pump Y 33 is time schedule controlled to maintain the DHW temperature at the end of the line.

Functional Diagram

Variations of temperatures based on a heat surface factor (HSF) of radiators in a building (equivalent to a building heat curve variation) ranging from 0.8 to 1.6 (Fig. 143).

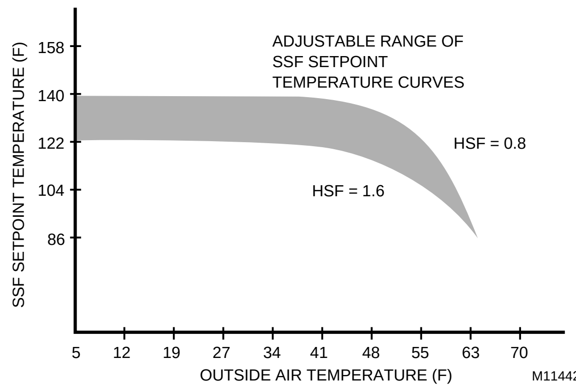


Fig. 143. SSF Temperature Reset Range.

HYBRID BUILDING SUBSTATION

Combination Jet Pump and Heat Exchanger.

The Heating System is mainly used in Eastern Europe or the CIS and consists of a Hydro-elevator Mixing Valve (jet pump). Hydro-elevators are mostly uncontrolled devices, mixing the building return water with supply water according to the hydro-mechanical configuration in the valve and requires no pump for heating water circulation in the building heating system (Fig. 144).

Table 14. Description of Figure 144 Controls.

Reference	Description	Reference	Description
DDC	Controller	TI 16	SSF temperature
HEX 1	Heat exchanger	TI 18	SRF temperature
QIR 12	Primary flow meter	TI 19	PRF temperature
QIR 17	Secondary flow meter	TI 32	Heating return temperature
TI 01	Outdoor temperature	TIC 21	DHW supply temperature
TI 11	PSF temperature	TIC 31	Heating supply temperature
TI 13	PSF temperature	Y 14	DHW control valve
TI 15	PRF temperature	Y 33	Heating control valve

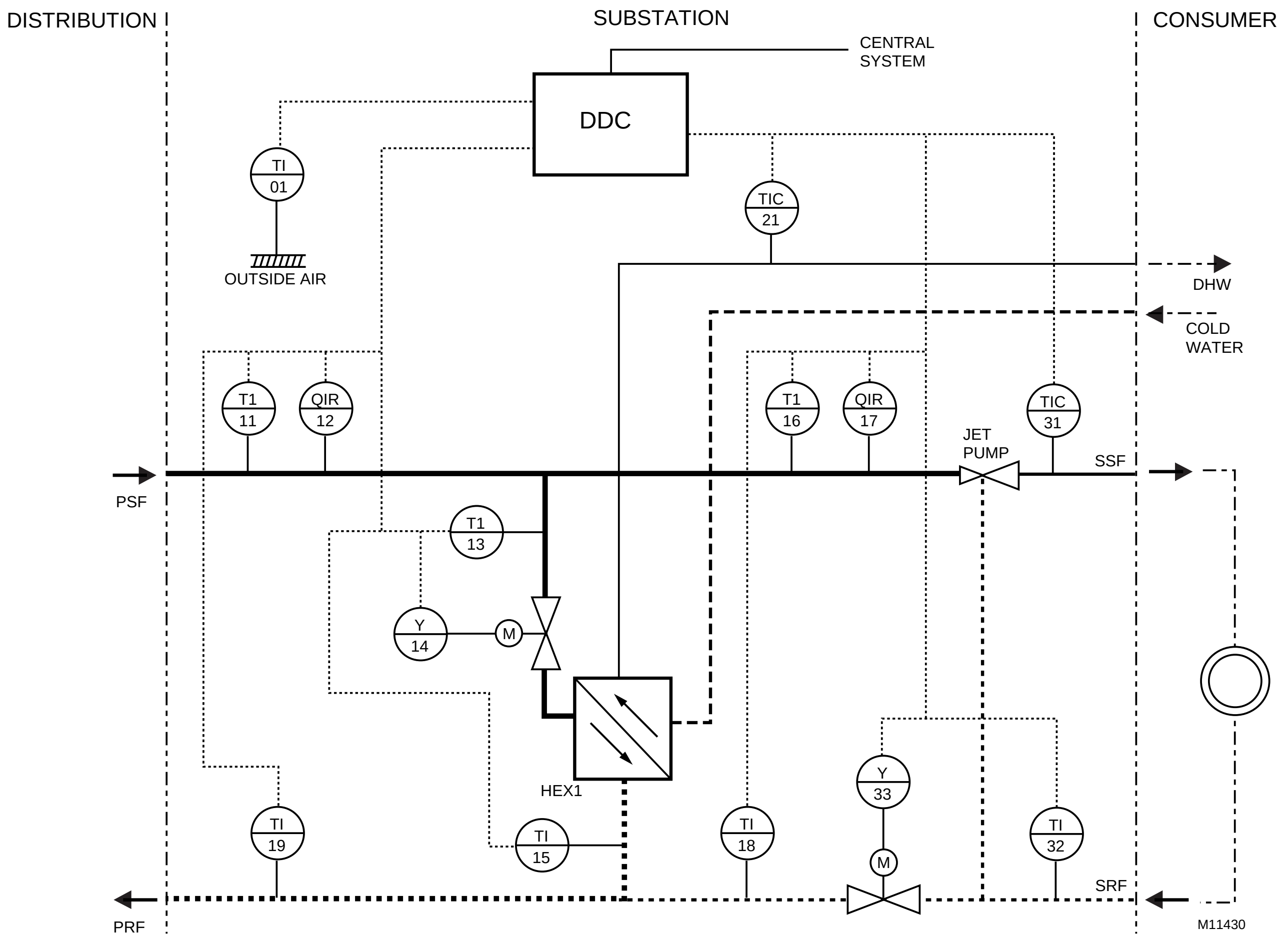


Fig. 144. Control of Heat Exchanger for DHW and Jet Pump for SSF.

Control Strategies

1. The SSF temperature depends on the mixing ratio of the jet pump. Jet pumps do generally not require secondary circulation pumps. Valve Y 33 on the return pipe provides a limited amount of flow control. As the valve varies from full open to 30%, flow through the jet pump varies from full flow. When the valve is less than 30% open, SSF flow cannot be guaranteed. TI 32 controls Y33 to maintain the SRF temperature as reset from outdoor air.

2. Heat exchanger HEX 1 Transfers heat to the isolated domestic hot water system (DHW). Control valve Y 14 in conjunction with temperature sensor TIC 21 control the DHW temperature. Because the amount of DHW in the heat exchanger is small, a fast control system response time is required. The response times should be:

Controller	cycle time	< 1 sec.
Actuator	on/off time	< 1 sec.
Sensor	rise time	< 2 sec.

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