Balancing of a water and air system

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A well performed testing, adjusting and balancing (TAB) of a HVAC system is essential for the proper performance of that system and can enhance indoor air quality and efficiency.

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A well performed testing, adjusting and balancing (TAB) of a HVAC system is essential for the proper performance of that system and can enhance indoor air quality and efficiency. Chapter 37 of the ASHRAE 2003 HVAC Applications Handbook gives the following definition of TAB:

HVAC system testing, adjusting, and balancing (TAB) is the process of checking and adjusting all environmental systems in a building to produce the design objectives. This process includes.

(1) Balancing air and water distribution systems

(2) Adjusting the total system to provide design quantities

(3) Electrical measurement

(4) Establishing quantitative performance of all equipment

(5) Verifying automatic control system operation and sequences of operation

(6) Sound and vibration measurement

These procedures are accomplished by checking installations for conformity to design, measuring and establishing the fluid quantities of the system as required meeting design specifications, and recording and reporting the results.

The following definitions used in this chapter. Refer to ASHRAE *Terminology of HVAC&R* (1991) for additional definitions.

Test. **Determine quantitative performance of equipment.**

Adjust. **Regulate the specified fluid flow rate and air patterns at the terminal equipment (e.g., reduce fan speed, adjust a damper).**

Balance. **Proportion flows in the distribution system (submains, branches, and terminals) according to specified design quantities.**

Balanced System. **A system designed to deliver heat transfer required for occupant comfort or process load at design conditions. A minimum heat transfer of 97% should be provided to the space or load served at design flow. The flow required for minimum heat transfer establishes the system's flow tolerance. The fluid distribution system should be designed to allow flow to maintain the required tolerance and verify its performance.**

Procedure. **An approach to and execution of a sequence of work operations to yield repeatable results.**

Report forms. **Test data sheets arranged in logical order for submission and review. They should also form the permanent record to be used as the basis for any future TAB work.**

Terminal. **A point where the controlled medium (fluid or energy) enters or leaves the distribution system. In air systems, these may be variable- or constant-volume boxes, registers, grilles, diffusers, louvers, and hoods. In water systems, these may be heat transfer coils, fan-coil units, convectors, or finned-tube radiation or radiant panels.**

Testing and Balancing HVAC

A systematic approach is called for when balancing a HVAC air and water systems. The following is a general procedure that can be applied to all systems.

1. Preliminary office work.

a. Gather and prepare report forms.

b. Gather plans and specifications.

Gather all applicable plans and specifications to include contract drawings, shop drawings, "as-built" drawings, schematics and Manufacturers' catalogs showing equipment data such as the description, capacities, and recommendations on testing of their equipment and equipment performance curves. All items will not be available, the more information that can be gathered. Better understanding of a system and its components is needed to do a proper balancing. Study the plans and specifications to become familiar with the system and the *Design Intent*. Color and label the system and all of its components. This will make plans easier to reed.

Design Intent as defined in ASHRAE Guideline 1 as a detailed explanation of the ideas, concepts, and criteria that defined work before starting. The design intent document utilized to provide a written record of these ideas, concepts, and criteria. Well-documented design intent benefits the engineer during the design process. When objectives that are poorly defined at the beginning of the project, changes in the design may occur after the design is complete or worse. It helps the engineer to communicate his or her intentions to the design team. This early communication with the team as to there expectations helps the engineer better define scope of the work and avoid making costly changes later. The definition of Design Intent has served the industry with varying degrees of success.

The basis of design intent is that all information necessary to accomplish the design including weather data, interior and exterior environmental criteria, other pertinent design assumption, cost goals, and references to applicable codes, standard, regulations, and guidelines be gather first.

Make out the report forms for each system being tested. Report forms will include equipment test sheets and balancing sheets for the distribution system. It is also recommend that a schematic drawing of each system be made. Schematics may be very detailed showing the central system with pressure and temperature drops across components, duct and pipe sizes, terminal devices, balancing valves and dampers, required and actual flow quantities, etc. Do as much paperwork in the office as possible.

2. Preliminary field inspection.

  c. Inspect the job site

  d. Inspect the distribution system.

  e. Inspect the equipment.

Preliminary Procedures

1. Review contract documents and plans for all of the HVAC system.
2. Review approved shop drawing and equipment submittals.
3. Prepare system schematics.
4. Insert preliminary data on test report forms.
5. Review electrical characteristics of equipment and assure that safety controls are operating and that all motor starters have the proper heater coils or overload protection.
6. Review completely the systems ready to be balance and verify that all balancing devices have been installed.
7. Confirm that all HVAC and temperature control system have been tested, strainers cleaned, systems flushed, etc., and are ready to be balance. Clean or temporary air filters should be in place as specified.
8. Confirm that all building components, such as ceiling plenums that affect system balance, are in place and sealed, and that all windows, door, etc., are installed and closed.
9. Confirm that all instruments are in good order.

Inspect the job site to see that the building is architecturally ready for balancing. For instance, are all the walls, windows, door, and ceilings installed? If the conditioned space isn't architecturally sealed, abnormal pressures and temperatures will adversely affect the system balance. Next, walk the air and water distribution system to ensure that they are intact, and aren't missing components such as dampers, valves, pressure and temperature taps, coils, terminal boxes, diffusers, grilles, etc. Lastly, inspect the equipment. Check that motors, fans, pumps, chillers, compressors, boilers, drives, etc. are mechanically and electrically ready. The following is a general list of items to be checked.

The Preliminary Procedures

Air Side

1. Ductwork intact and properly sealed.
2. Set all volume control damper and variable air volume boxes to the full open position, unless system diversity requires balancing in zones.
3. Set outside air dampers to the minimum position.
4. Ductwork leak tested.
5. Access doors installed and properly secured.
6. Ductwork installed according to the drawings and specification.
7. Ductwork free from debris.
8. Dampers, including fire and smoke dampers, installed and accessible.
9. Terminal boxes, reheat coils, electrical reheat, etc., installed, functional and accessible.
10. All other air distribution devices such as diffusers, etc., installed and functional.
11. Return air has an unobstructed path from each conditioned space back to the unit.
12. Filters clean and correctly installed.

13. Filter frame properly installed and sealed.
14. Coils cleaned and properly installed.
15. Drive components installed.
16. Sheaves properly aligned and tight on their shafts.
17. Verify correct fan rotations and speeds.
18. Belts adjusted for the correct tension.
19. Belt guard properly installed.
20. Automatic control dampers installed and functional.
21. Fan vortex dampers functional.
22. Fan housings installed and properly sealed according to the drawings and specifications.
23. Flexible connections installed properly.
24. Fan wheel aligned properly and adequate clearance in the housing.
25. Fan bearing lubricated
26. Check motor amperages and voltages; make necessary adjustments.

Water Side

1. Strainers and piping free from debris, cleaned, and flushed.
2. Set all balancing devices to full open position.
3. Construction strainer baskets replaced with permanent baskets.
4. Set mixing valves and control valves to full coil flow; close coil by-pass valves.
5. System filled to the proper level and the pressure-reducing valve set.
6. Automatic and manual air vents properly installed and functional
7. All air purged from the system.
8. Water in the expansion tanks at the proper level.
9. All valves, flow meters, and temperature/pressure tap installed correctly, accessible and functional.
10. Terminal coils installed, piped correctly, and accessible.
11. Pumps properly aligned, grouted and anchored.
12. Verify correct pump rotations and proper drive alignments.
13. Vibration isolators properly installed and adjusted.
14. Flexible connections installed properly.
15. Measure and record pump motor amperages and voltages.
16. Check pump flow before balancing coils.
17. Proceed with air systems balancing.

Boiler

1. All operating and safety setting for temperature and pressure are correct.
2. Pressure relief valve functional.
3. Boiler started and operating properly.

Chiller and Condenser

1. **All operating and safety settings for temperature and pressure are correct.**
2. **Chiller, condenser started, and operating correctly.**

Electrical

1. **Motors wired and energized.**
2. **Proper starter and overload protection installed.**
3. **Correct fuses installed.**
4. **Motors properly secured on their frames.**
5. **Motor bearings lubricated.**

Controls

1. **Controls complete and functional.**
2. **Make initial tests on all fans and pump applicable to the system being balanced.**
3. **Balance and adjust the distribution system.**
4. **Adjust the fan or pump as needed.**
5. **Take final reading.**
6. **Complete reports.**

The Balancing Procedures

Airside procedures

1. **All related HVAC and exhaust air system should be operating.**
2. **Determine whether any other HVAC or exhaust air system could affect the system ready to be balanced.**
3. **Make Pitot tube traverses on all main supply and major branch ducts where possible to determine the air distribution. Take extreme care that there is no damage to hepa filters by debris being pulled or pushed into them.**
4. **Adjust balancing dampers of each major branch duct that is high on airflow. A minimum of one branch duct-balancing damper shall remain fully open.**
5. **Measure and record the airflow of each terminal device in the system without adjusting any terminal outlet. Flow measuring hoods are the preferred airflow-measuring device.**
6. **The total airflow for the terminal outlets should be close to the Pitot tube traverse air measurement of that branch, and the main duct traverse air measurement should be within 10% of the total of all terminal outlet air measurements.**
7. **Check for excessive duct leakage if total terminal outlet air measurements are less than 95% of the main duct traverse air measurements.**
8. **Adjust the terminals that are highest on airflow to about 10% under design airflow.**
9. **Next, adjust each terminal outlet throughout the zone or system to design airflow and record measurements and make any necessary branch damper adjustments.**
10. **An additional adjusting pass throughout the system may be necessary. Make final adjustments to the fan drive where required. Record all data.**
11. **Adjust terminal device vanes to minimize drafts and for proper air distribution.**
12. **Measure and record system static pressures.**

13. Measure and record all required outdoor air, return air, mixed air, and supply air dry bulb a wet bulb temperatures. Measure and record all plenum static pressures.

14. Measure and record all coil entering air and leaving air dry bulb and wet bulb temperatures. Measure and record all coil pressure differentials.

15. Measure and record final fan motor full load amperages and voltages.

16. If the airflow is low, proceed with a proportional balancing. Proportional balancing is purporting the air equally by dividing the total airflow by the design airflow to get the percentage of airflow. Then multiplying the percentage of airflow to the design airflow for the percentage of air to set to.

Waterside procedures

1. Continually check system and vent air from high points and circuits with lower flows during hydronic balancing. Periodically check and clean strainers.

2. Using "pump shutoff head," verify each pump head, operating curve and impeller size.

3. Adjust pump to design flow and record data.

4. Adjust boilers and or chillers to design flows and temperatures and record data. If the Pump is pumping water to coils first start there, always fallow the piping.

5. If flow-measuring devices are used, record the flow data throughout the system before adjusting the system.

6. Measure and record pressure drops through all coils and or units. Compare with submittal data for high and low flows.

7. Adjust high flows to near design.

8. Adjust pump flow to design and check pressures, amperages and voltages.

9. Set bypass balancing cocks to 90% of maximum flow through coils that have three-way control valves.

10. Repeat the above procedures until all coils and units are operating within 10% of design. When coils in parallel are above five coils, balance the coil nearest to the pump to the coil at the end of the piping. The first coil is set at 80% to 85% of design flow depending on the number of coils; increase the percentage equally for each coil to the end of piping, then check the first coil flow.

11. Measure and record final pump pressure, amperages, and voltages.

12. Measure and record all coil and or unit pressure drops entering and leaving water temperatures.

13. Measure and record all data from all flow measuring devices.

Airflow Measurement

CFM = cubic feet per minute

FPM = velocity in feet per minute

FT^2 = area in foot square

The basic airflow equation for any free area is

{airflow (cfm) = area (ft^2) X velocity (fpm)}

2000 cfm = $2ft^2$ X 1000 fpm

{area (ft^2) = airflow (cfm) / velocity (fpm)}

$2ft^2$= 2000 cfm / 1000 fpm

{velocity (fpm) = airflow (cfm) / area (ft^2)}

1000 fpm = 2000 cfm / $2ft^2$

{Velocity fpm = 4005 X √ Velocity pressure}

997.2 fpm = 4005 X √ 0.062 in. w.g.

Free area defined as the total minimum area of openings in an air outlet or inlet device through which air can pass. Free area of return air or supply air grilles may be as low as 50% of the duct connection size. The free cross-sectional area of a duct normally is 100%. If other data are not available, it may be assumed that all similar return air or supply air grilles would have similar free areas when measured with the same instrument.

Airflow equation for a return air or supply air grilles

{Airflow (cfm) = area (ft^2) X velocity (fpm) X % of free area}

2250 cfm = 3 ft^2 X 1000 fpm X 75% free area

{% of free area = airflow (cfm) / area (ft^2) X velocity (fpm)}

75% = 2250 cfm / 3 ft^2 X 1000 fpm

Example 1: find the cfm of a duct of 24"X12" with a velocity of 1000 fpm. Find the cfm of the duct.

Solution

24"X12"/144= 2 ft^2

2 ft^2 X 1000 fpm = 2000 cfm

Example 2: A 48"X36" return air grille has a measured average velocity of 370 fpm. A Pitot tube traverse of the connecting duct indicates airflow of 2975 cfm. Find the return air grille free area percentage.

Solution

Free area is equal to; duct airflow in cfm divided by grille velocity in fpm, Divided by the area of the grill times 100

2975 cfm / 370 fpm = 8.04

48"X36"/144= 12 ft^2 grill

8.04 / 12 ft^2 = 0.67

0.67 X 100 = 67% free area

15

Example 3: Find the nearest standard size round duct to handle 4600 cfm at a velocity of 1000 fpm.

Solution

Area of the duct in ft² is equal to: Airflow cfm / Velocity fpm

Food square to inch square is equal to: ft² X 144 = in²

In² to duct radius is equal to: √ square root of; area in² / π(3.14)

Radius to diameter is equal to: radius X 2 = diameter

4600 cfm / 1000 fpm = 4.6 ft²

4.6 ft² X 144 = 662.4 in²

662.4 in² / 3.14 π = 210.95

√ of 210.95 = 14.52

14.52 X 2 = 29.05

29.05" diameter of a duct for 4600 cfm at 1000 fpm

Duct Flow

The preferred method of measuring duct volumetric flow is the pitot-tube traverse average. The maximum straight run should be obtained before and after the traverse station. To obtain the best duct velocity profile, measuring points should be located as shown in Chapter 14 of the 2001 *ASHRAE Handbook—Fundamentals* and ASHRAE *Standard* 111. When using factory-fabricated volume-measuring stations, the measurements should be checked against a pitot-tube traverse.

Power input to a fan's driver should be used as only a guide to indicate its delivery; it may also be used to verify performance determined by a reliable method (e.g., pitot-tube traverse of system's main) that considers possible system effects. For some fans, the flow rate is not proportional to the power needed to drive them. In some cases, as with forward-curved-blade fans, the same power is required for two or more flow rates. The backward-curved-blade centrifugal fan is the only type with a flow rate that varies directly with the power input.

Pitot Tube Traverses

Procedures

- **To accomplish repeatable traverse measurements, take the measurements in a specific, measured pattern, as indicated in 3.2. below.**
- **Duct size must not change in a traversed section.**
- **Face the Pitot tube into the air stream, parallel to the ductwork at each measurement point, and measure the velocity.**
- **Convert velocity pressure to fpm velocity before averaging if the traverse is taken at other than standard conditions.**
- **Take traverse measurements at actual conditions and actual cubic feet per minute [Actual CFM]. Correct Actual CFM to standard CFM [Standard CFM] when specified by**

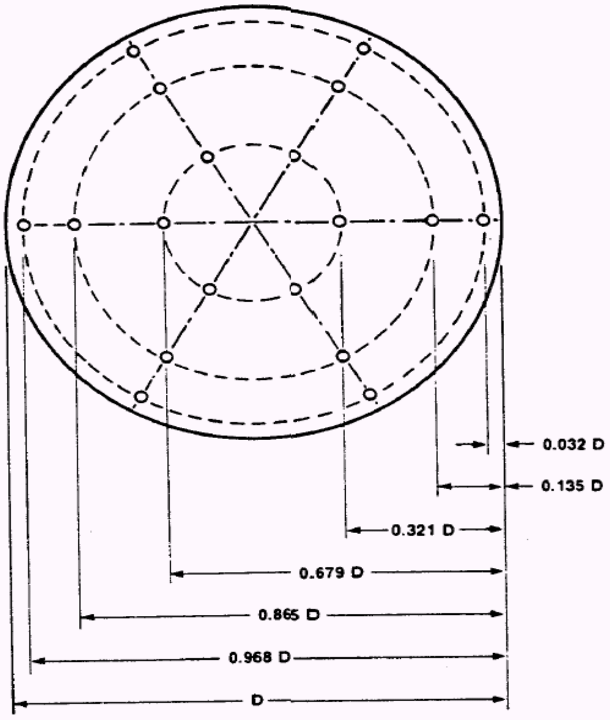
using the density correction.

- **With figure 3.2 Verify that velocity measurements are acceptable. A traverse plane is suitable for flow measurements if more than 75% of the velocity pressure readings are greater than 1/10 of the maximum velocity measurement and are not negative.**

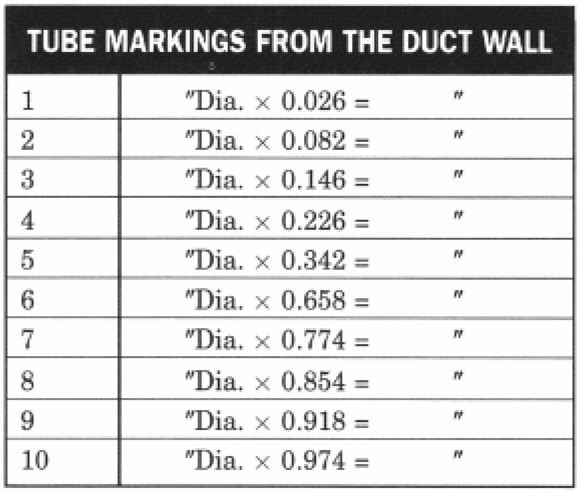
- **Show all traverses in the final report which will show duct size, static pressure at the traverse, velocity pressure and corresponding velocity, duct area, and the airflow. If the traverse is taken in other than standard conditions, show barometric pressure and temperature. Show density corrections for each traverse.**

Round Duct Traverse

For field traverses any duct that are greater than 10" (250 mm) diameter, use the standard Pitot tube. Two holes located 90" apart in the same plane are required to make a Pitot tube traverse of a round duct. The Pitot tube must be marked so that velocity pressure readings (10 for each hole) can be taken at centers of equal concentric areas. Pitot tube locations for traversing round ducts with 20 readings are as follows in Figure 3.1.



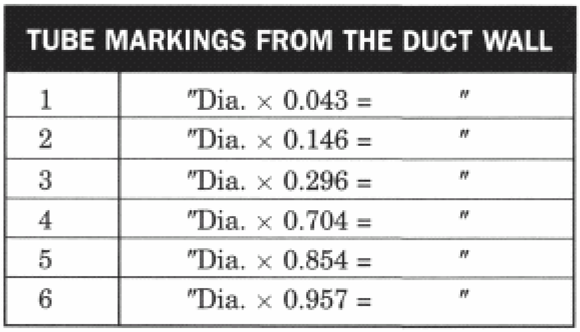
NO. OF MEASURING POINTS PER DIAMETER	POSITION RELATIVE TO INNER WALL
6	0.032, 0.135, 0.321, 0.679, 0.865, 0.968
8	0.021, 0.117, 0.184, 0.345, 0.655, 0.816, 0.883, 0.981
10	0.019, 0.077, 0.153, 0.217, 0.361, 0.639, 0.783, 0.847, 0.923, 0.981



TUBE MARKINGS FROM THE DUCT WALL	
1	"Dia. × 0.026 = "
2	"Dia. × 0.082 = "
3	"Dia. × 0.146 = "
4	"Dia. × 0.226 = "
5	"Dia. × 0.342 = "
6	"Dia. × 0.658 = "
7	"Dia. × 0.774 = "
8	"Dia. × 0.854 = "
9	"Dia. × 0.918 = "
10	"Dia. × 0.974 = "

Fig. 3.1 Pitot Tube markings for round duct using ten readings for each hole

For traverses taken in round duct equal to or less than 10" (250 mm) in diameter, a l/8" (3 mm) diameter Pitot tube shall be used. The 1/8" (3 mm) diameter Pitot tube must be marked so that 12 velocity pressures (six for each hole) can be taken at the center of equal concentric areas. Pitot tube locations for traversing the round duct with 12 readings are given in Figure 3.2:



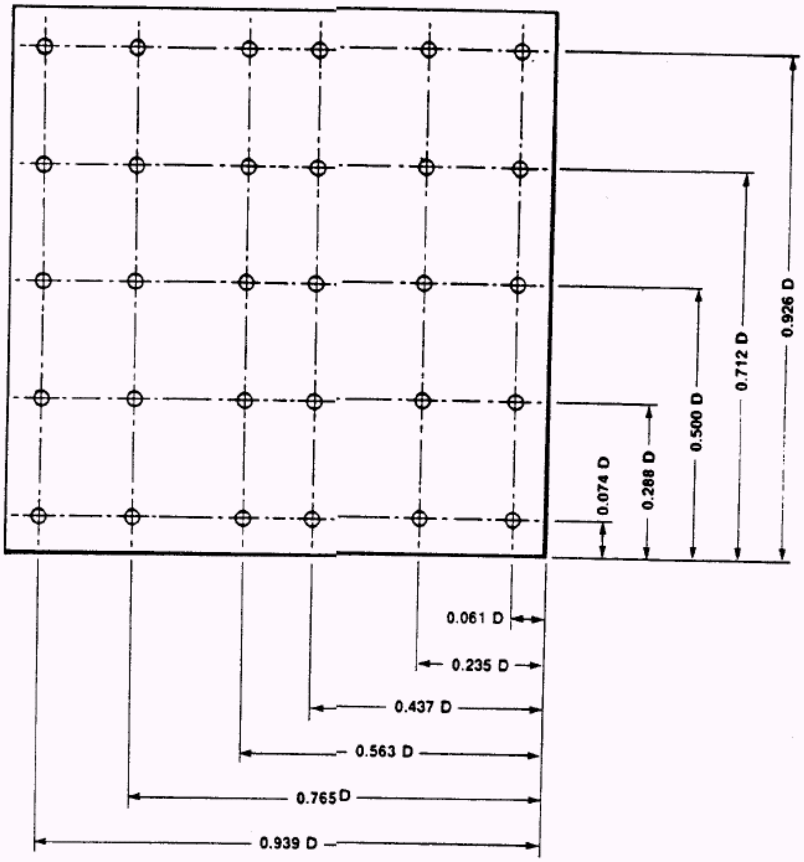
TUBE MARKINGS FROM THE DUCT WALL	
1	"Dia. × 0.043 = "
2	"Dia. × 0.146 = "
3	"Dia. × 0.296 = "
4	"Dia. × 0.704 = "
5	"Dia. × 0.854 = "
6	"Dia. × 0.957 = "

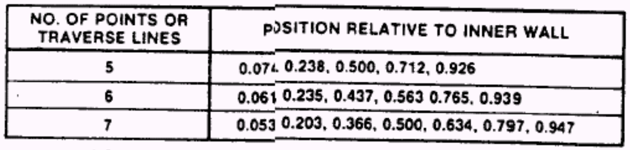
Fig. 3.2 Pitot tube markings for round duct using six readings for each hole

Square or Rectangular Duct Traverse

Performing a Pitot tube traverse of a square or rectangular duct, the number and spacing of the holes in the duct and the markings on the Pitot tube are determined using the following method:

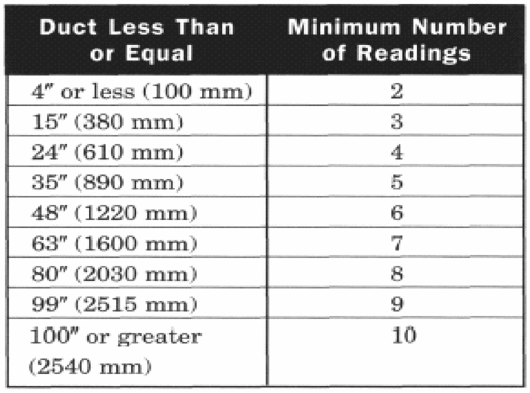
The minimum number of readings taken in a square or rectangle duct is 4. This would be for a duct with height and width 4" (100 m) or less. The number of readings for each side will be based on Figure 3.3.





NO. OF POINTS OR TRAVERSE LINES	POSITION RELATIVE TO INNER WALL
5	0.074 0.238, 0.500, 0.712, 0.926
6	0.061 0.235, 0.437, 0.563 0.765, 0.939
7	0.053 0.203, 0.366, 0.500, 0.634, 0.797, 0.947

LOG TCHEBYCHEFF RULE FOR RECTANGULAR DUCTS



Duct Less Than or Equal	Minimum Number of Readings
4″ or less (100 mm)	2
15″ (380 mm)	3
24″ (610 mm)	4
35″ (890 mm)	5
48″ (1220 mm)	6
63″ (1600 mm)	7
80″ (2030 mm)	8
99″ (2515 mm)	9
100″ or greater (2540 mm)	10

Fig. 3.3 Minimum number of readings for rectangular duct

For any duct with a side greater than 100” (2400 mm), the maximum distance between holes shall not exceed 12” (300 mm). For all readings, the corner reading shall be located l/2 the distance between readings. For example, a 12” (300 mm) duct width will have 3 readings 4” (100 m) apart with the first reading taken 2“ (50 mm) from the duct wall.

Flat Oval Duct Traverses

For field readings, flat oval ducts may be traversed by two different methods depending on the dimensions of the duct.

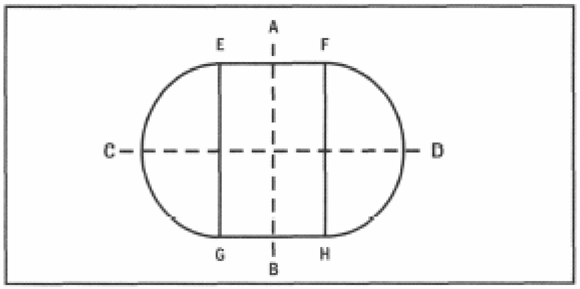


Fig. 3.4 Oval duct traversed as a circle

If the width of rectangle EF to GH as shown in Figure 3.4 is less than the diameter of the semi-circle (Line EG), then traverse as a circle using the Equal Area Method (Figure 3.1 or 3.2) at (A-B) and (C-D). The area of the oval will be computed by adding the area of the circle to the area of the rectangle.

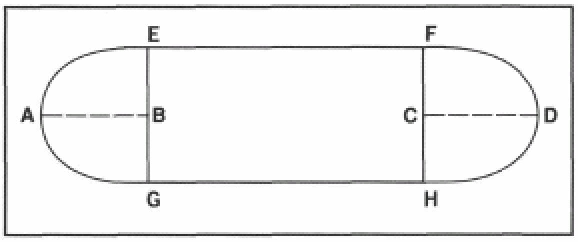


Fig. 3.5 Oval duct traversed as a rectangle and centerline radius AB and CD

If the width of rectangle EF to GH as shown in Figure 3.5 is greater than or equal to the diameter of the semi-circle (Line EG), then traverse as a rectangle (Figure 3.3). For the two semi-circles traverse the plane (A-B) and (C-D) using the spacing shown in round duct traverse (Figures 3.1 or 3.2). The area will be computed by adding the area of the circle to the area of the rectangle.

CFM of Ducts at a Friction Loss in inches of water

Per 100 feet of duct at 29.29”

Duct sizes in inches

	5”	6”	7”	8”	9”	10”	12”	14”	16”	18”	20”	22”
.06”	**52**	**85**	**123**	**175**	**240**	**320**	**505**	**775**	**1100**	**1500**	**1940**	**2450**
.08”	**60**	**97**	**143**	**210**	**270**	**365**	**600**	**900**	**1390**	**1725**	**2290**	**2950**
.10”	**68**	**109**	**162**	**240**	**315**	**420**	**690**	**1000**	**1460**	**1975**	**2610**	**3150**
.12”	**75**	**120**	**178**	**255**	**345**	**485**	**750**	**1110**	**1600**	**2200**	**2900**	**3600**
.14”	**84**	**130**	**195**	**270**	**387**	**510**	**830**	**1200**	**1750**	**2400**	**3180**	**3950**

CFM of Ducts at different Velocity

Velocity in Feet per minutes at 29.92”

Ducts	**5”**	**6”**	**7”**	**8”**	**9”**	**10”**	**12”**	**14”**	**16”**	**18”**	**20”**	**22”**
FPM												
600	**83**	**120**	**160**	**208**	**265**	**330**	**475**	**650**	**850**	**1040**	**1300**	**1600**
800	**110**	**160**	**210**	**280**	**350**	**440**	**625**	**850**	**1150**	**1600**	**1750**	**2100**
1000	**138**	**195**	**265**	**345**	**440**	**550**	**790**	**1090**	**1400**	**1780**	**2190**	**2420**

If an installation has an inadequate straight length of ductwork or no ductwork to allow a pitot-tube traverse, the procedure from Sauer and Howell (1990) can be followed: a vane anemometer reads air velocities at multiple points across the face of a coil to determine a loss coefficient.

Balancing Devices

Volume Dampers

Balancing dampers are a primary component in duct systems, and their importance should not be ignored. Balancing dampers are used to control the volume of airflow in the system by creating a resistance to flow. Balancing dampers can produce unnecessary resistance and noise problems, and will not control the air as intended if improperly selected,

located, installed, or adjusted.

The relationship between the position of a damper and its percentage of airflow with respect to the airflow through a fully open damper is termed its “flow characteristic.”

Opposed-blade dampers are generally recommended for large ducts because they provide better control of the airflow, and therefore have better flow characteristics than parallel blade dampers. The actual effect of closing a damper can only be determined from field measurements. Proper location of balancing dampers not only permits efficient air distribution but also equalizes the pressure drops in the different airflow paths within the system (Figure 6.1). Manually operated opposed-blade or single-blade, quadrant-type volume dampers must be installed in every zone duct of a multi-zone system and each branch duct takeoff to control the amount of air entering the branch.

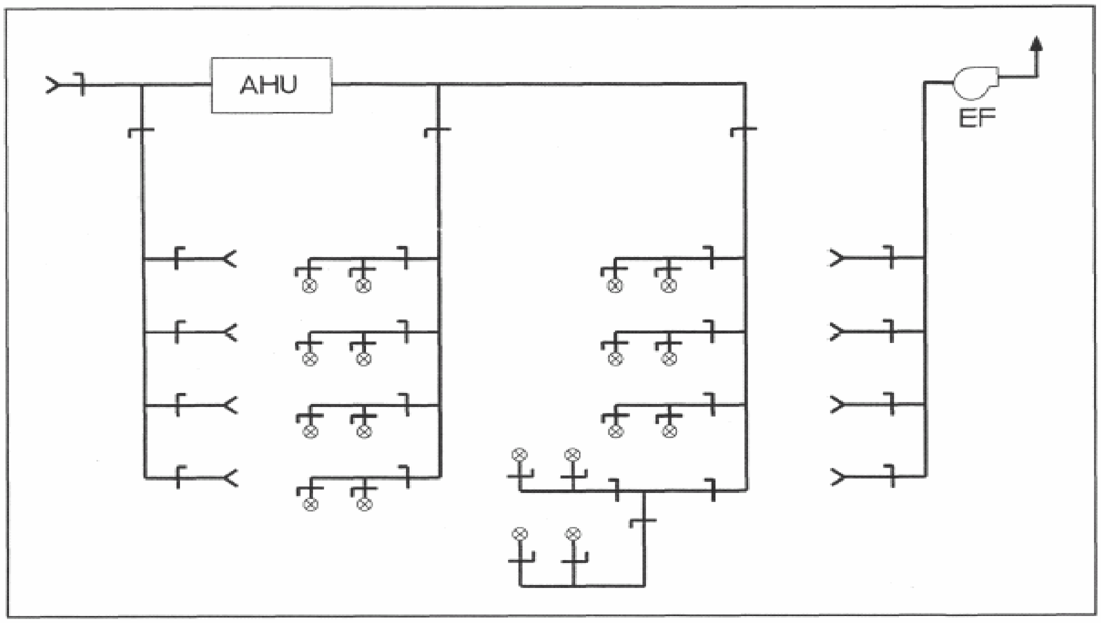


Fig. 6.1 Balancing Damper Location

Manual dampers should be provided in each branch takeoff to control the airflow to grilles and diffusers. Single-blade or opposed-blade volume dampers located immediately behind diffusers and grilles should not be used for balancing because when throttled they create noise, change the flow factor, can be easily tampered with and are an unnecessary expense. Proper installation and location of balancing dampers in the takeoffs eliminates the need for volume controls at grilles and diffusers. Each damper should be adjustable with a locking quadrant handle or regulator and end bearings that have sufficient strength and rigidity for the pressures being controlled. The locking quadrant must be located outside the vapor seal of the insulation.

Generally, parallel-blade dampers are used in mixing plenum applications, and opposed-blade dampers are used in a volume-control application. It should be noted that only a slight opening of an opposed blade volume damper might generate a relatively high noise level as the air passes through. To minimize generated duct noise at volume dampers, dampers should be located at least two duct diameters from a fitting and as far as possible from an outlet/inlet.

A spin-in collar with a volume damper is used to make a run out to a grille or diffuser. They can be conical, bell mouth, flanged or notched. They should be used for ducts with velocities below 1000 fpm (5.1 *m/s)* and under 0.50” static pressure. The round damper must have a square rod with end bearings and standoff for externally insulated ducts and locking quadrants. The square rod must be continuous for all dampers 12” (300 mm) and above. The damper should also be stiffened to prevent movement in the airstreams.

Splitter devices offer little or no control of air volume in ducts. They should be regarded as air diverters only with maximum effectiveness when present in duct systems exhibiting low resistance to airflow. When used as a diverter, splitter devices must have at least one push rod with a locking device. The push rod must be connected to the leading edge of the splitter at 90 degrees to the blade, not controlled by the quadrant handle on the pivot shaft. Splitter devices should be long enough to divert the airflow in the duct. As a minimum, the length should be twice the size of the smallest throat of the nested connection.

The use of adjustable extractors (Figure 6.2) is not generally recommended because they can cause turbulence in the main trunk line, thereby increasing the system total pressure and affecting the performance of other branch outlets downstream. Extractors are not balancing devices and should not be used at branch or main duct takeoffs to provide volume control. Extractors are principally used to divert air to branch ducts when they are the full height of the duct.

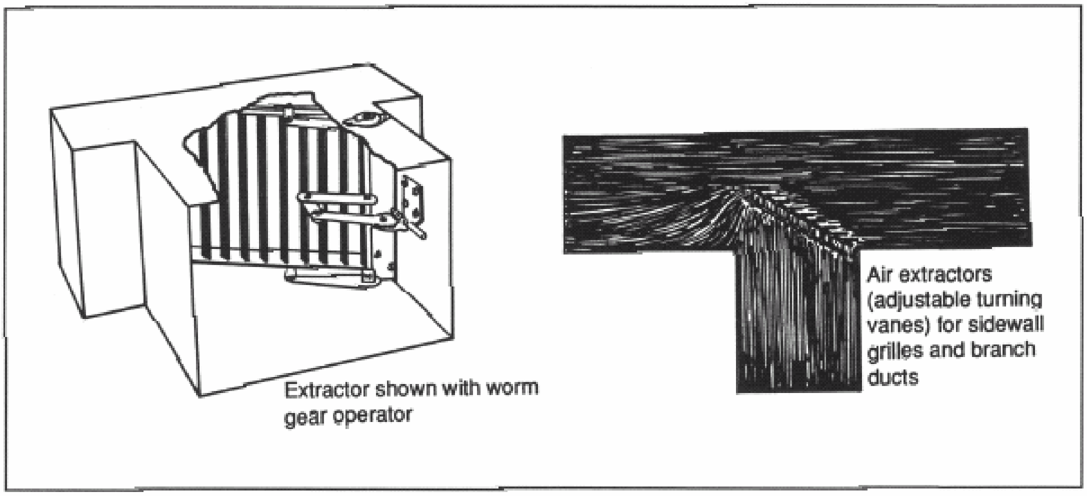
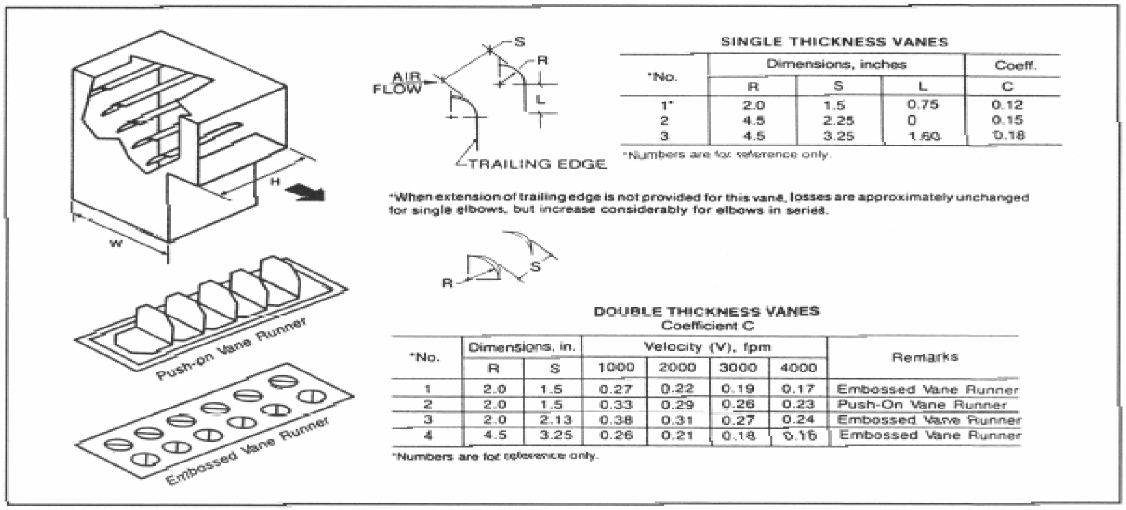


Fig. 6.2 Extractors.

Turning Vanes

Turning vanes (Figure 6.3) reduce the pressure loss and provide a uniform velocity distribution downstream from mitered rectangular elbows when smooth radius rectangular elbows are not used. Turning vanes should be installed so that the air entering and leaving the vanes is parallel to the duct walls. Double thickness or single thickness with a trailing (extended) edge turning vanes should be used in all rectangular elbows.



SINGLE THICKNESS VANES

*No.	Dimensions, inches			Coeff.
	R	S	L	C
1*	2.0	1.5	0.75	0.12
2	4.5	2.25	0	0.15
3	4.5	3.25	1.60	0.18

*Numbers are for reference only.

*When extension of trailing edge is not provided for this vane, losses are approximately unchanged for single elbows, but increase considerably for elbows in series.

DOUBLE THICKNESS VANES
Coefficient C

*No.	Dimensions, in.		Velocity (V), fpm				Remarks
	R	S	1000	2000	3000	4000	
1	2.0	1.5	0.27	0.22	0.19	0.17	Embossed Vane Runner
2	2.0	1.5	0.33	0.29	0.26	0.23	Push-On Vane Runner
3	2.0	2.13	0.38	0.31	0.27	0.24	Embossed Vane Runner
4	4.5	3.25	0.26	0.21	0.18	0.16	Embossed Vane Runner

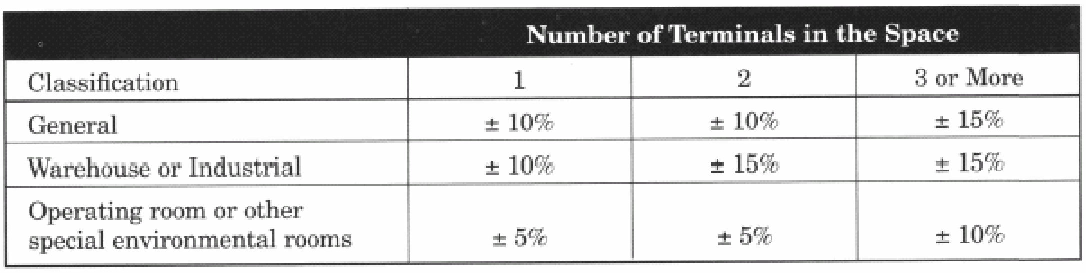
*Numbers are for reference only.

Fig. 6.3 Loss Coefficients of Elbows with Turning Vanes.

Air Balancing Tolerances

All air quantities shall be measured according to National Standards. All air

balancing tolerances shall be superseded by the specification if different. Total air quantity to each space shall be within ± 10% of design. Terminals shall be adjusted to design quantities in accordance with Figure 6.5.



	Number of Terminals in the Space		
Classification	1	2	3 or More
General	± 10%	± 10%	± 15%
Warehouse or Industrial	± 10%	± 15%	± 15%
Operating room or other special environmental rooms	± 5%	± 5%	± 10%

Figure 6.5 Percent Tolerance Between Air Terminals Within a Space

Dual-Duct Systems

Most constant-volume dual-duct systems are designed to handle part of the total system's supply through the cold duct and smaller air quantities through the hot duct. Balancing should be accomplished as follows:

1. When adjusting multizone or dual-duct constant-volume systems. establish the ratio of the design volume through the cooling coil to total fan volume to achieve the desired diversity factor. Keep the proportion of cold to total air constant during the balance. However, check each zone or branch with this component on full cooling. If the design calls for full flow through the cooling coil, the entire system should be set to full flow through the cooling side while making tests. Perform the same procedure for the hot-air side.

2. Check the leaving air temperature at the nearest terminal to verify that hot and cold damper inlet leakage is not greater than the established maximum allowable leakage.

3. Check apparatus and main trunks, as outlined in the section on Equipment and System Check.

4. Determine whether static pressure at the end of the system (the longest duct run) is at or above the minimum required for mixing box operation. Proceed to the extreme end of the system and check the static pressure with an inclined manometer. Pressure should exceed the minimum static pressure recommended by the mixing box manufacturer.

5. Proportionately balance diffusers or grilles on the low-pressure side of the box, as described for low-pressure systems in the previous section.

6. Change control settings to full heating, and ensure that the controls and dual-duct boxes function properly. Spot-check the airflow at several diffusers. Check for stratification.

7. If the engineer has included a diversity factor in selecting the main apparatus, it will not be possible to get full flow from all boxes simultaneously, as outlined in item 3 under Equipment and System Check. Mixing boxes closest to the fan should be set to the opposite hot or cold deck to the more-critical-season airflow to force the air to the end of the system.

Balancing the VAV System

The general procedure for balancing a VAV system is.

1. Determine the required maximum air volume to be delivered by the supply and return air fans. Load diversity usually means that the volume will be somewhat less than the outlet total.

2. Obtain fan curves on these units, and request information on surge characteristics from the fan manufacturer.

3. If inlet vortex damper is used, obtain the fan manufacturer's data on damper control (deaeration) of the fan when used with the damper. If speed control is used, find the maximum and minimum speed that can be used on the project.

4. Obtain from the manufacturer the minimum and maximum operating pressures for terminal or variable-volume boxes to be used on the project.

5. Construct a theoretical system curve, including an approximate surge area. The system curve starts at the boxes' minimum inlet static pressure, plus system loss at minimum flow, and terminates at the design maximum flow. The operating range using an inlet vane damper is between the surge line intersection with the system curve and the maximum design flow. When variable speed control is used, the operating range is between.

1. The minimum speed that can produce the necessary minimum box static pressure at minimum flow still in the fan's stable range

2. The maximum speed necessary to obtain maximum design flow

6. Position the terminal boxes to the proportion of maximum fan air volume to total installed terminal maximum volume.

7. Set the fan to operate at approximate design speed (increase about 5% for a fully open inlet vane damper).

8. Check a representative number of terminal boxes. If static pressure varies widely, or if airflow at several boxes is below flow, check every box.

9. Run a total air traverse with a pitot tube.

10. Increase speed if static pressure and/or volume are low. If volume is correct but static is high, reduce speed. If static is high or correct but volume is low, check for system effect at the fan. If there is no system effect, go over all terminals and adjust them to the proper volume.

11. Run Steps (7) through (10) with the return or exhaust fan set at design flow as measured by a pitot-tube traverse and with the system set on minimum outside air.

12. Proportion the outlets, and verify design volume with the VAV box on maximum flow.

Verify minimum flow setting.

13. Set terminals to minimum, and adjust the inlet vane or speed controller until minimum static pressure and airflow are obtained.

14. Temperature control personnel, balancing personnel, and the design engineer should agree on the final placement of the sensor for the static pressure controller. This sensor must be placed in a representative location in the supply duct to sense average maximum and minimum static pressures in the system.

15. Check return air fan speed or its inlet vane damper, which tracks or adjusts to the supply fan airflow, to ensure proper outside air volume.

16. Operate the system on 100% outside air (weather permitting), and check supply and return fans for proper power and static pressure.

Fan Tracking

As supply system airflows and pressures change, consideration must be given as to how the return air, outside air and relief air track.

Volumetric Tracking

Volumetric tracking is the preferred method of measurement. Tracking is accomplished by measuring the airflows of two fan systems. This maintains a difference for constant minimum outside air volume and building pressurization. Fan selection is critical to prevent surges or unstable operation of the controls. Surge can occur as the fan approaches a point where there is insufficient air entering the fan wheel to completely fill the space between the blades. Surge will be indicated by a fluctuation in CFM and static pressure, and will create a noisy operation.

Sizing of the calibrated airflow station is also critical in the air handling unit operation. Unstable operation of controls occurs when the velocity drops below 600 fpm (3 *m/s)* where the air measuring station is out of range. Most transducers require at least 0.02"wc of pressure to operate properly. Consideration must also be made on the high end to not exceed the transducer's range.

Plenum Pressurization Tracking

Plenum pressurization is the control of fan speed and minimum outside air

volumes by constant plenum pressure. The control of supply fan speed or vane control is accomplished by maintaining a constant duct pressure at various volumes at a point designated for control, usually 2/3 distance down the supply duct. The minimum outside air volume is maintained with a constant mixed air plenum pressure controlling the return air damper. The purge (exhaust) damper is controlled by building pressure. The return fan speed is controlled by a constant return fan discharge plenum pressure. The

maximum out-side air damper for economizer is controlled by mixed air temperature.

Prior to setting, the plenum pressure tracking, the supply, return and exhaust system must be balanced. The supply duct static pressure set point is measured and recorded (Point A) (Figure 10.1). With the system set for maximum airflow, return air damper closed, economizer and relief air dampers 100% open, set the return/relief airflow, adjust, measure and record. Read and record the discharge air static pressure of the return/ relief fan (Point B), which will be the static pressure controller set point. Set the building pressure controller to maintain and control the relief air dampers. Read and record the set point. Release the economizer dampers and set to minimum outside condition. Adjust the mixed air pressure sensor to obtain the minimum outside air. Read and record the mixed air pressure sensor set point (Point C).

Set the supply system to minimum airflow and verify the mixed air pressure sensor is controlling by measuring and recording system static profiles with the system at maximum, a midpoint and minimum airflow.

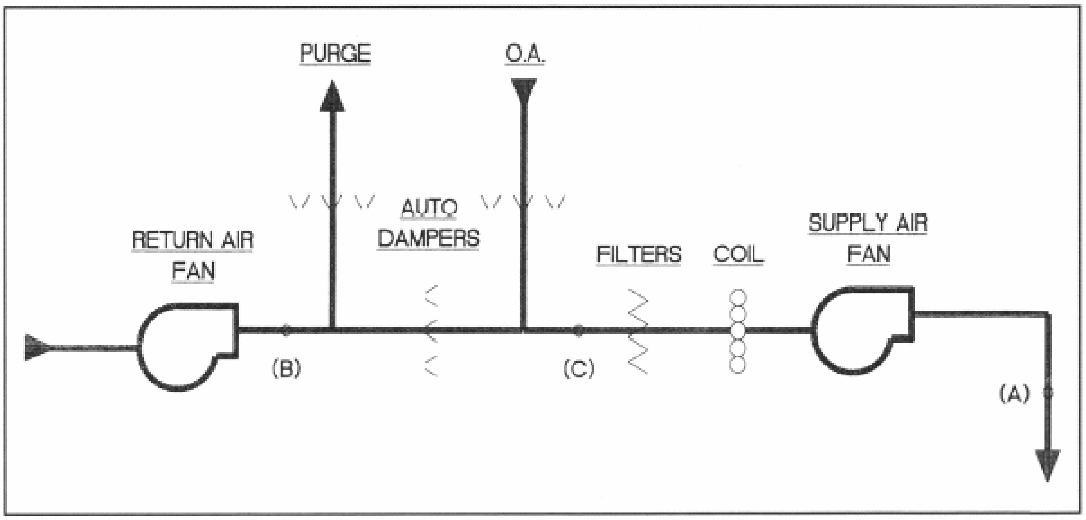


Fig. 10.1 Plenum Pressure Tracking Points

Induction Systems

Most induction systems use high-velocity air distribution. Balancing should be accomplished as follows:

1. For apparatus and main trunk capacities, perform general VAV balancing procedures.

2. Determine primary airflow at each terminal unit by reading the unit plenum pressure with a manometer and locating the point on the charts (or curves) of air quantity versus

static pressure supplied by the unit manufacturer.

3. Normally, about three complete passes around the entire system are required for proper adjustment. Make a final pass without adjustments to record the end result.

4. To provide the quietest possible operation, adjust the fan to run at the slowest speed that provides sufficient nozzle pressure to all units with minimum throttling of all unit and riser dampers.

5. After balancing each induction system with minimum outside air, reposition to allow maximum outside air and check power and static pressure readings.

Duct System Pressure

The duct system designer calculates the static pressure losses of the straight section of ductwork using engineering tables and charts. To these losses, the losses of the entire duct fitting are calculated and added along with the pressure loss data of all manufactured items such as filters, coils, dampers and diffusers or grills. The duct system fan(s) are selected from the total static pressure losses of the longest run(s) of the supply air and return air ducts connected to the fan(s).

The three duct pressure that technicians measure in the field are total pressure (TP), static pressure (SP), and velocity pressure (Vp). The three pressures are related by the following equation:

Total pressure in w.g. (TP) = Static pressure in w.g. (SP) + Velocity pressure in w.g. (Vp)

Static pressure

Static pressure (SP) is exerted equally in all directions at any point or cross section of the duct. It also is a measure of the potential energy to produce and maintain airflow against duct resistance. Static pressure may be positive or negative to the atmosphere, but it can be measured indirectly by subtracting the velocity pressure from the total pressure (SP = TP – Vp).

Velocity pressure

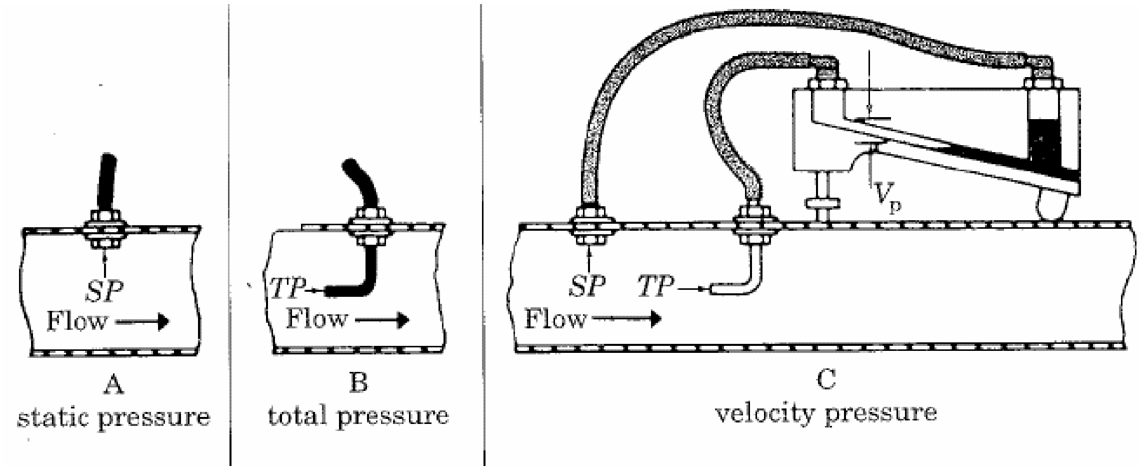
Velocity pressure (Vp) is exerted only in the direction of airflow and is a measure of kinetic energy resulting from the airflow. Velocity pressure cannot be measured directly by a pitot tube and pressure gauge or manometer, but it can be measured indirectly by subtracting the static pressure from the total pressure (Vp = TP – SP).

Vp = (Velocity in fpm / 4005)^2

Total pressure

Total pressure (TP) of a duct is measured by the impact of the moving air stream on the end of a Pitot tube directly facing and perpendicular to the airflow. Total pressure

determines how much energy is in the airflow at any point in the system. Total pressure always decline in value from the fan to any terminal device (diffuser or grille), but it can be measured indirectly by adding the static pressure to the Velocity pressure (TP = SP + Vp).



Example 4: A 48 X 16 in. duct handles 10,700 cfm at 2.3 in. w.g. static pressure. Find the total pressure and velocity pressure of the duct.

Velocity fpm = cfm / area in ft^2

Velocity pressure = (velocity fpm / 4005)^2

Total pressure = static pressure + velocity pressure

10,700 cfm / (48 X 16 / 144) = 2006 fpm

(2006 fpm / 4005)^2 = 0.25 in. w.g. velocity pressure

2.3 in. w.g. static pressure + 0.25 in. w.g. velocity pressure = 2.55 in. w.g. total pressure

Calculation of CFM from Heat flow

The temperature of air decreases progressively as the air passes through an evaporator coil. The drop in air temperature is greatest across the first row of the coil and diminishes as the air passes across each succeeding row. The fact that the temperature difference between the air and the refrigerant is greatest across the first row, and becomes less and less as the temperature of the air is reduced in passing across each succeeding row. The temperature difference is least across the last row of the coil.

External factors affect coil performance. Principal among these are the circulation, velocity, and distribution of air in the refrigerated space and over the coil. These factors

are closely related and in many cases is dependent one on the other.

Heat from the product or condition space is carried to the evaporator by air circulation. When air circulation is inadequate, heat is not carried from the product or condition space to the evaporator at a rate sufficient to allow the evaporator to perform at peak efficiency or capacity. It is important also that the circulation of air is evenly distributed in all parts of the refrigerated space and over the coil. Poor distribution of the air circulating can result in uneven temperatures and dead spots in the air condition space. Uneven distribution of air over the coil surface causes some parts of the surface to function less efficiently than other and lowers evaporator capacity and efficiency.

When air velocity is low, the air passing over the coil stays in contact with the coil surface longer. More heat is removed and is cooled with a greater range. Thus, the temperature difference increases, the refrigerant temperature decreases, resulting in a loss of capacity and efficiency because the rate of heat transfer is lowered. As air velocity increases, a greater quantity of air is brought into contact with the evaporator coil. Consequently, the temperature difference decreases, the refrigerant temperature increases, resulting in a gain of capacity and efficiency because the rate of heat transfer is increases.

Air volume change across the coil will increase or decrease the refrigerant temperature that increases or decreases the efficiency and capacity of a system. Increasing the heat by increasing the airflow or increasing the evaporating coil size by 10% will be decrease of water removed from the air by 2% to 4%, but there will be a 4% to 8% capacity increase. This will result in a change of the load in the system from a 72% to 74% sensible heat increase and from a 28% to 26% latent heat decrease. An increase of the refrigerant temperature of 1ºF will increase compressor efficiency by 1% to 2%.

The reason for the increase in heat and the increase in refrigeration temperature is due to the increasing size and/or airflow of the overall coil. Over size an evaporating coil or increase airflow, decreases the water removing capacity of a coil.

Decrease heat by a decrease of airflow or an evaporating coil size by 10% will result in a 2% to 4% increase of water removed from the air, will cause a 4% to 8% capacity decrease. The capacity decrease results in a 72% to 70% decrease of sensible heat change to the load in the system, thereby causing a 28% to 30% latent heat increases. A decrease of the refrigerant temperature of 1ºF will decrease compressor efficiency by 1% to 2%.

The decrease compression efficiency is caused by the decrease in heat, and the refrigeration temperature that is due to the decreasing size and/or airflow of the overall coil. Under sizing an evaporating coil or a decrease of refrigerant temperature, increases

the water removing capacity of a coil.

Lowering the refrigeration below the airs dew point in an existing evaporator coil to remove more water from the air or to increase water-removing capacity. This can be done by lowing the airflow or downsizing the coil but not increasing the coil size. The coolest refrigerant temperature is 34 degrees (no ice on the coil).

It is possible to check the airflow of a Fan coil by knowing the air entering and the leaving air-dry bulb and well bulb temperature.

Calculation of heat

Calculation of heat movement from the air to the refrigerant is done by the loss of heat in the air. Air that enters the evaporator has heat. This heat is called the enthalpy of the air. Enthalpy of air is defined as the sum of the internal energy of the air. Air heat is called Enthalpy because it is a combination of sensible and latent heat. Sensible heat raises or lowers the temperature of the air. Latent heat adds or removes the water from the air. The formula for this heat movement is.

Air Density X 60 X CFM X Δ Enthalpy = BTUH

The first part of this equation is the weight of the air in pounds of air per hour (air density X 60 X Cfm), the next part (Δ enthalpy) is the total heat removed per pound of air in enthalpy.

Pounds of Air per hour = Air Density X 60 X CFM

Δ Enthalpy = Enthalpy of the air into an evaporator (h1) - Enthalpy of the air of an evaporator (h2)

The formula for CFM = BTUH / (Air density X 60 X Δ Enthalpy)

Btuh is the total heat that has been removed from the air. In a water system the total heat add to the water in a fan coil or a chiller is equal to

Btuh = GPM X 60 X 8.337 X Δ Temperature water

Air Density

Air density is defined as the mass per unit volume of air. As the temperature of a given mass of air increases, its volume increases [i.e. thermal expansion] and its density decreases. As the temperature of a given mass of air decreases, its volume decreases and its density increases.

100ºF dry bulb and 30% RH a 1lb of air is 14.4F³ a density of 0.0694

62ºF dry bulb and 30% RH a 1lb of air is 13.2F³ a density of 0.0757

As the relative humidity increases at a given temperature the given mass of air increases and it's, volume increases and density decreases. As the relative humidity decreases at a given temperature, the given mass of air and volume decreases and its density increases.

76ºF dry bulb and 100% RH a 1lb of air is 13.9F³ a density of 0.0719

76°F dry bulb and 0% RH a 1lb of air is 13.5F³ a density of 0.0740

The Foot³ for one pound of air must be located first before air density can be calculated using the formula. Air density is equal to one pound of air divided by the foot³ per pound of air (Air Density = 1lb / ft³). One pound of air changes its Foot³ with the dry and wet bulb (See Table 8A, 8B). The formula of total heat uses the air density of the air leaving the evaporator. To using, Table 8 A&B for ft³ and 9 A to C for air density.

To use table 8:

(1) Measure the dry bulb temperature with a digital thermometer of the air leaving the evaporator.

(2) Measure the wet bulb (water vaporization temperature) temperature with a digital sling psychrometer of the air leaving the evaporator.

(3) Find the measured indoor entering dry bulb temperature on the left side of the table and the wet bulb temperature on top line.

(4) Where the lines cross is the Foot³. See table 8A & 8B

(5) Divide the Foot³ by 1 pound for air density.

See Table 9 and 9B & 9C for air density

62ºF dry bulb and 46º wet bulb 1lb of air is 13.21F³ with a density of 0.0756

Foot³ of one pound of air

Table 8A

Dry Bulb on left, Wet Bulb on top

	70	68	66	64	62	60	58	56	54	52	50	48	46	44
70	13.69	13.65	13.62	13.59	13.56	13.54	13.51	13.49	13.46	13.44	13.42	13.39	13.37	13.35
69		13.63	13.60	13.57	13.54	13.52	13.49	13.46	13.44	13.42	13.39	13.37	13.35	13.33
68		13.61	13.58	13.55	13.52	13.49	13.67	13.44	13.42	13.39	13.37	13.35	13.33	13.31
67			13.56	13.53	13.50	13.47	13.45	13.42	13.40	13.78	13.35	13.33	13.31	13.29
66			13.54	13.51	13.48	13.45	13.43	13.40	13.38	13.36	13.33	13.31	13.29	13.27
65				13.49	13.46	13.43	13.41	13.38	13.36	13.33	13.31	13.29	13.27	13.25
64				13.47	13.44	13.41	13.39	13.36	13.34	13.31	13.29	13.27	13.25	13.23
63					13.42	13.39	13.37	13.34	13.32	13.29	13.27	13.25	13.23	13.21
62					13.40	13.37	13.35	13.32	13.30	13.27	13.25	13.23	13.21	13.19
61						13.35	13.32	13.30	13.28	13.25	13.23	13.21	13.19	13.17
60						13.33	13.30	13.28	13.25	13.23	13.21	13.19	13.17	13.15

Foot^3 of one pound of air at 29.92”

Table 8B

Dry Bulb on left, Wet Bulb on top

	60	58	56	54	52	50	48	46	44	42	40	38	36	34
65	13.43	13.40	13.38	13.35	13.33	13.31	13.29	13.27	13.25	13.23				
64	13.41	13.38	13.36	13.33	13.31	13.29	13.27	13.25	13.23	13.21				
63	13.39	13.36	13.34	13.31	13.29	13.27	13.25	13.23	13.21	13.19	13.17			
62	13.37	13.34	13.32	13.29	13.27	13.25	13.22	13.21	13.19	13.17	13.15			
61	13.35	13.32	13.29	13.27	13.25	13.23	13.21	13.19	13.17	13.15	13.13			
60	13.33	13.30	13.27	13.25	13.23	13.21	13.19	13.17	13.15	13.13	13.11			
59		13.28	13.25	13.23	13.21	13.19	13.17	13.15	13.13	13.11	13.09	13.08		
58		13.26	13.23	13.21	13.19	13.17	13.15	13.13	13.11	13.09	13.07	13.05		
57			13.21	13.19	13.17	13.15	13.13	13.11	13.09	13.07	13.05	13.03		
56			13.19	13.17	13.15	13.13	13.11	13.8	13.07	13.05	13.03	13.01		
55				13.15	13.13	13.11	13.08	13.06	13.04	13.03	13.01	12.99	12.97	
54				13.13	13.11	13.08	13.06	13.04	13.03	13.01	12.98	12.97	12.95	
53					13.08	13.06	13.04	13.03	13.00	12.98	12.97	12.95	12.93	
52					13.06	13.04	13.03	13.00	12.98	12.96	12.95	12.93	12.91	12.89
51						13.02	13.00	12.98	12.96	12.95	12.93	12.91	12.89	12.87
50						13.00	12.98	12.96	12.94	12.92	12.91	12.88	12.87	12.85
49							12.96	12.94	12.92	12.90	12.88	12.87	12.85	12.83
48							12.94	12.92	12.90	12.88	12.87	12.85	12.83	12.81
47								12.90	12.88	12.86	12.84	12.83	12.81	12.79
46								12.88	12.86	12.84	12.82	12.80	12.79	12.77
45									12.84	12.82	12.80	12.78	12.77	12.75
44									12.82	12.80	12.78	12.76	12.75	12.73
43										12.78	12.76	12.74	12.73	12.71
42										12.76	12.74	12.72	12.71	12.69
41											12.72	12.70	12.69	12.67
40											12.70	12.68	12.66	12.65
39												12.66	12.64	12.63
38												12.64	12.62	12.61
37													12.60	12.59
	60	58	56	54	52	50	48	46	44	42	40	38	36	34

Foot^3 of one pound of air at 29.92”

Table 8A & 8B

Air density change with altitudes table 9 is the correction for altitude

Table 9Air Density at Altitudes

Foot^3 per 1lb	Altitude 0 Feet	Altitude 1,000 Feet	Altitude 3,000 Feet	Altitude 5,000 Feet	Altitude 7,000 Feet	Altitude 9,000 Feet
12.0	0.08333	0.08033	0.07467	0.06933	0.06433	0.05958
12.1	0.08264	0.07967	0.07405	0.06876	0.06380	0.05909
12.2	0.08197	0.07902	0.07344	0.06820	0.06328	0.05861
12.3	0.08130	0.07837	0.07285	0.06764	0.06276	0.05813
12.4	0.08065	0.07774	0.07226	0.06710	0.06226	0.05766
12.5	0.08000	0.07712	0.07168	0.06656	0.06176	0.05720
12.6	0.07937	0.07651	0.07111	0.06603	0.06127	0.05675
12.7	0.07874	0.07591	0.07055	0.06551	0.06079	0.05630
12.8	0.07813	0.07531	0.07000	0.06500	0.06031	0.05586
12.9	0.07752	0.07473	0.06946	0.06450	0.05984	0.05543
13.0	0.07682	0.07415	0.06892	0.06400	0.05938	0.05500
13.1	0.07634	0.07359	0.06840	0.06351	0.05893	0.05458
13.2	0.07576	0.07303	0.06788	0.06303	0.05848	0.05417
13.3	0.07519	0.07248	0.06737	0.06256	0.05805	0.05376
13.4	0.07463	0.07194	0.06687	0.06209	0.05761	0.05336
13.5	0.07407	0.07141	0.06637	0.06163	0.05719	0.05296
13.6	0.07353	0.07088	0.06588	0.06118	0.05676	0.05257
13.7	0.07299	0.07036	0.06540	0.06073	0.05635	0.05219
13.8	0.07246	0.06986	0.06493	0.06029	0.05594	0.05181
13.9	0.07194	0.06935	0.06446	0.05986	0.05554	0.05144
14.0	0.07143	0.06886	0.06400	0.05943	0.05514	0.05107
14.1	0.07092	0.06837	0.06355	0.05901	0.05475	0.05071
14.2	0.07042	0.06789	0.06310	0.05859	0.05437	0.05035
14.3	0.06993	0.06741	0.06266	0.05818	0.05399	0.05000
14.4	0.06944	0.06694	0.06222	0.05778	0.05361	0.04965
14.5	0.06897	0.06648	0.06179	0.05738	0.05324	0.04931
14.6	0.06849	0.06603	0.06137	0.05699	0.05288	0.04897
14.7	0.06803	0.06558	0.06095	0.05660	0.05252	0.04864
14.8	0.06757	0.06514	0.06054	0.05622	0.05216	0.04831
14.9	0.06711	0.06470	0.06013	0.05584	0.05181	0.04799
15.0	0.06667	0.06427	0.05973	0.05547	0.05147	0.04767
Foot^3 per 1lb	Altitude 0 Feet	Altitude 1,000 Feet	Altitude 3,000 Feet	Altitude 5,000 Feet	Altitude 7,000 Feet	Altitude 9,000 Feet

Table 9

Altitude correction for Cu ft per 1 pound of Air: Cu ft per 1 pound of Air / Altitude Ratio

Altitude correction for Air density is: Air density X Altitude Ratio

Table 9 B

Air Density at 29.92”

Dry Bulb on left, Wet Bulb on top

	70	68	66	64	62	60	58	56	54	52	50	48	46	44
70	0.0731	0.0732	0.0734	0.0736	0.0737	0.0738	0.0739	0.0741	0.0743	0.0744	0.0745	0.0747	0.0748	0.0749
69		0.0733	0.0735	0.0736	0.0738	0.0739	0.0741	0.0743	0.0744	0.0745	0.0747	0.0748	0.0749	0.0750
68		0.0735	0.0736	0.0737	0.0739	0.0741	0.0743	0.0744	0.0745	0.0747	0.0748	0.0749	0.0750	0.0751
67			0.0737	0.0738	0.0740	0.0742	0.0744	0.0745	0.0746	0.0748	0.0749	0.0750	0.0751	0.0752
66			0.0738	0.0739	0.0742	0.0743	0.0745	0.0746	0.0747	0.0749	0.0750	0.0751	0.0752	0.0754
65				0.0741	0.0743	0.0744	0.0746	0.0747	0.0749	0.0750	0.0751	0.0752	0.0753	0.0754
64				0.0743	0.0744	0.0745	0.0747	0.0748	0.0750	0.0751	0.0752	0.0753	0.0754	0.0755
63					0.0745	0.0746	0.0748	0.0749	0.0751	0.0752	0.0753	0.0754	0.0755	0.0757
62					0.0746	0.0747	0.0749	0.0751	0.0752	0.0753	0.0754	0.0755	0.0757	0.0758
61						0.0748	0.0751	0.0752	0.0753	0.0754	0.0755	0.0757	0.0758	0.0759
60						0.0750	0.0752	0.0753	0.0754	0.0755	0.0757	0.0758	0.0759	0.0760

Table 9 C Air Density at 29.29”

Dry Bulb on left, Wet Bulb on top

	60	58	56	54	52	50	48	46	44	42	40	38	36	34
65	0.0744	0.0746	0.0747	0.0749	0.0750	0.0751	0.0752	0.0753	0.0754	0.0755				
64	0.0745	0.0747	0.0748	0.0750	0.0751	0.0752	0.0753	0.0754	0.0755	0.0757				
63	0.0746	0.0748	0.0749	0.0751	0.0752	0.0753	0.0754	0.0755	0.0757	0.0758	0.0759			
62	0.0747	0.0749	0.0751	0.0752	0.0753	0.0754	0.0755	0.0757	0.0758	0.0759	0.0760			
61	0.0748	0.0751	0.0752	0.0753	0.0754	0.0755	0.0757	0.0758	0.0759	0.0760	0.0762			
60	0.0750	0.0752	0.0753	0.0754	0.0755	0.0757	0.0758	0.0759	0.0760	0.0762	0.0763			
59		0.0753	0.0754	0.0755	0.0757	0.0758	0.0759	0.0760	0.0762	0.0763	0.0764	0.0765		
58		0.0754	0.0755	0.0757	0.0758	0.0759	0.0760	0.0762	0.0763	0.0764	0.0765	0.0767		
57			0.0757	0.0758	0.0759	0.0760	0.0762	0.0763	0.0764	0.0765	0.0767	0.0768		
56			0.0758	0.0759	0.0760	0.0762	0.0763	0.0764	0.0765	0.0767	0.0768	0.0769		
55				0.0760	0.0762	0.0763	0.0764	0.0765	0.0767	0.0768	0.0769	0.0770	0.0771	
54				0.0762	0.0763	0.0764	0.0765	0.0767	0.0768	0.0769	0.0770	0.0771	0.0772	
53					0.0764	0.0765	0.0767	0.0768	0.0769	0.0770	0.0771	0.0772	0.0774	
52					0.0765	0.0767	0.0768	0.0769	0.0770	0.0771	0.0772	0.0774	0.0775	0.0776
51						0.0768	0.0769	0.0770	0.0771	0.0772	0.0774	0.0775	0.0776	0.0777
50						0.0769	0.0770	0.0771	0.0772	0.0774	0.0775	0.0776	0.0777	0.0779
49							0.0771	0.0772	0.0774	0.0775	0.0776	0.0777	0.0779	0.0780
48							0.0772	0.0774	0.0775	0.0776	0.0777	0.0779	0.0780	0.0781
47								0.0775	0.0776	0.0778	0.0779	0.0780	0.0781	0.0782
46								0.0776	0.0778	0.0779	0.0780	0.0781	0.0782	0.0783
45									0.0779	0.0780	0.0781	0.0782	0.0783	0.0784
44									0.0780	0.0781	0.0782	0.0784	0.0785	0.0786
43										0.0782	0.0784	0.0785	0.0786	0.0787
42										0.0784	0.0785	0.0786	0.0787	0.0788
41											0.0786	0.0787	0.0789	0.0790
40											0.0787	0.0789	0.0790	0.0791
39												0.0790	0.0791	0.0792
38												0.0791	0.0792	0.0793
37													0.0793	0.0794

Enthalpy

(H1-H2)

Enthalpy of air is defined as the sum of the internal energy of the air. Air heat is called Enthalpy because it is a combination of sensible and latent heat. Sensible

heat raises or lowers the temperature of the air. Latent heat adds or removes the water from the air.

The air that enters an evaporator has heat and that heat can be measured by the wet bulb temperature. Wet bulb is a good indication of the total heat of the air the enthalpy.

- **Air at 67ºF dry bulb and 67ºF wet bulb has an enthalpy of 31.62.**
- **Air at 70ºF dry bulb and 67ºF wet bulb has an enthalpy of 31.59.**
- **Air at 80ºF dry bulb and 67ºF wet bulb has an enthalpy of 31.51.**
- **Air at 90ºF dry bulb and 67ºF wet bulb has an enthalpy of 31.43.**

When calculating the total heat that an evaporator has removed from the air the wet bulb is taken from the air going in and out of the evaporator (Δ enthalpy (h1-h2)).

Example 5: using table 11

- **Air going in to an evaporator coil at 80ºF dry bulb and 67ºF wet bulb has an enthalpy of 31.62.**
- **Air coming out of an evaporator coil at 62ºF dry bulb and 57ºF wet bulb has an enthalpy of 24.48.The Δ enthalpy (h1-h2) is 31.62-24.48= 7.14 Δ enthalpy.**

Table 11

Enthalpy at saturation, Btu per pound of dry air at 29.92”

Wet-Bulb (F)	.0	.1	.2	.3	.4	.5	.6	.7	.8	.9	Wet-Bulb (F)
30	10.92	10.96	11.00	11.04	11.08	11.12	11.16	11.20	11.24	11.29	30
31	11.33	11.37	11.41	11.45	11.49	11.53	11.57	11.61	11.65	11.70	31
32	11.76	11.80	11.84	11.88	11.92	11.96	12.00	12.04	12.08	12.12	32
33	12.17	12.21	12.25	12.29	12.33	12.38	12.42	12.46	12.50	12.54	33
34	12.59	12.63	12.67	12.71	12.75	12.80	12.84	12.88	12.92	12.97	34
35	13.01	13.05	13.09	13.14	13.18	13.22	13.27	13.31	13.35	13.40	35
36	13.44	13.48	13.53	13.57	13.61	13.66	13.70	13.74	13.79	13.83	36
37	13.87	13.92	13.96	14.01	14.05	14.10	14.14	14.19	14.23	14.28	37
38	14.32	14.36	14.41	14.45	14.50	14.54	14.59	14.64	14.68	14.73	38
39	14.77	14.82	14.86	14.91	14.96	15.00	15.05	15.09	15.14	15.18	39
40	15.23	15.28	15.32	15.37	15.42	15.47	15.51	15.56	15.61	15.65	40
41	15.70	15.75	15.80	15.84	15.89	15.94	15.99	16.04	16.08	16.13	41
42	16.17	16.22	16.27	16.32	16.32	16.42	16.46	16.51	16.56	16.61	42
43	16.66	16.71	16.76	16.80	16.80	16.90	16.95	17.00	17.05	17.10	43
44	17.15	17.20	17.25	17.30	17.30	17.40	17.45	17.50	17.55	17.60	44
45	17.65	17.70	17.75	17.80	17.85	17.91	17.96	18.01	18.06	18.11	45
46	18.16	18.21	18.26	18.32	18.37	18.42	18.47	18.52	18.58	18.63	46
47	18.68	18.73	18.79	18.84	18.89	18.95	19.00	19.05	19.10	19.16	47
48	19.21	19.26	19.32	19.37	19.43	19.48	19.53	19.59	19.64	19.70	48
49	19.75	19.81	19.86	19.92	19.97	20.03	20.08	20.14	20.19	20.25	49
50	20.30	20.36	20.41	20.47	20.53	20.58	20.64	20.69	20.75	20.81	50
51	20.86	20.92	20.98	21.03	21.09	21.15	21.21	21.26	21.32	21.38	51
52	21.44	21.49	21.55	21.61	21.67	21.73	21.78	21.84	21.90	21.96	52
53	22.02	22.08	22.14	22.20	22.26	22.32	22.38	22.44	22.50	22.56	53
54	22.62	22.68	22.74	22.80	22.86	22.92	22.98	23.04	23.10	23.16	54
55	23.22	23.28	23.34	23.41	23.47	23.53	23.59	23.65	23.72	23.78	55
56	23.84	23.90	23.97	24.03	24.10	24.16	24.22	24.29	24.35	24.42	56
57	24.48	24.54	24.61	24.67	24.74	24.80	24.86	24.93	24.99	25.06	57
58	25.12	25.19	25.25	25.32	25.38	25.45	25.52	25.58	25.65	25.71	58
59	25.78	25.85	25.92	25.98	26.05	26.12	26.19	26.26	26.32	26.39	59
60	26.46	26.53	26.60	26.67	26.74	26.81	26.87	26.94	27.01	27.08	60
61	27.15	27.22	27.29	27.36	27.43	27.50	27.57	27.64	27.70	27.78	61
62	27.85	27.92	27.99	28.07	28.14	28.21	28.28	28.35	28.43	28.50	62
63	28.57	28.64	28.72	28.79	28.87	28.94	29.01	29.09	29.16	29.24	63
64	29.31	29.39	29.46	29.54	29.61	29.62	29.76	29.83	29.91	29.98	64
65	30.06	30.14	30.21	30.29	30.37	30.45	30.52	30.60	30.68	30.75	65
66	30.83	30.91	30.99	31.07	31.15	31.23	31.30	31.38	31.46	31.54	66
67	31.62	31.70	31.78	31.86	31.94	32.02	32.10	32.18	32.26	32.34	67
68	32.42	32.50	32.59	32.67	32.75	32.84	32.92	33.00	33.08	33.17	68
69	33.25	33.33	33.42	33.50	33.59	33.67	33.75	33.84	33.92	34.01	69
70	34.09	34.18	34.26	34.35	34.43	34.52	34.61	34.69	34.78	34.86	70
71	34.95	35.04	35.13	35.21	35.30	35.39	35.48	35.57	35.65	35.74	71
72	35.83	35.92	36.01	36.10	36.19	36.28	36.38	36.47	36.56	36.65	72
73	36.74	36.83	36.92	37.02	37.11	37.20	37.29	37.38	37.48	37.57	73
74	37.66	37.76	37.85	37.95	38.04	38.14	38.23	38.33	38.42	38.51	74
75	38.61	38.71	38.80	38.90	39.00	39.09	39.19	39.28	39.38	39.47	75
76	39.57	39.67	39.77	39.87	39.97	40.07	40.17	40.27	40.37	40.47	76

To use Table 11:

1. Using a digital sling psychrometer or a sling psychrometer and; familiarize yourself with its operation.

2. **Wet the bulb with distilled water for a sling psychrometer; obtain the wet bulb and dry bulb reading of the air going into the fan coil. Leave the psychrometer in the air until the wet bulb stops moving (this may take 5 minutes).**

Wet bulb temperature_____________°F

Dry bulb temperature______________ºF

3. Using a digital sling psychrometer or the sling psychrometer obtain the wet bulb and dry bulb reading of the air coming out of the fan coil. Leave the psychrometer in the air until the wet bulb stops moving (this may take 5minutes).

Wet bulb temperature _____________°F

Dry bulb temperature______________ºF

4. Using the wet bulb temperature find the enthalpy of the air that is going in and out of the evaporator (table 11) or with the wet and dry bulb temperature use (Table 11B, C or D) to find the enthalpy.

Enthalpy air in ____________ EP

Enthalpy air out ____________ EP

Difference in EP _____________ΔEP

Table 11B, 11C and 11D are Enthalpy of air at different dry bulb temperature

Table 11A

Enthalpy at saturation, Btu per pound of dry air at 29.92”

Dry Bulb on left, Wet Bulb on top

	100	98	96	95	94	93	92	91	90	89	88	87	86	85	84
100	71.76	68.23	64.87	63.26	61.69	60.16	58.67	57.22	55.80	54.42	53.08	51.76	50.49	49.24	48.03
98		68.26	64.90	63.29	61.72	60.19	58.70	57.25	55.83	54.45	53.11	51.79	50.52	49.27	48.05
96			64.94	63.32	61.75	60.22	58.73	57.28	55.86	54.48	53.13	51.82	50.54	49.29	48.08
94					61.78	60.25	58.76	57.31	55.89	54.51	53.16	51.85	50.57	49.32	48.11
92							58.79	57.38	55.92	54.54	53.19	51.88	50.59	49.35	48.13
90									55.94	54.56	53.22	51.90	50.62	49.37	48.15
88											53.24	51.93	50.65	49.40	48.18
86													50.67	49.42	48.20
84															48.22

Enthalpy at saturation, Btu per pound of dry air at 29.92”

Dry Bulb on left, Wet Bulb on top

	83	82	81	80	79	78	77	76	75	74	73	72	71	70
94	46.91	45.76	44.63	43.53	42.45	41.41	40.38	39.38	38.41	37.46	36.52	35.62	34.73	33.87
92	46.94	45.79	44.65	43.55	42.48	41.43	40.41	39.41	38.43	37.48	36.54	35.64	34.75	33.88
90	46.97	45.81	44.68	43.58	42.50	41.45	40.43	39.43	38.45	37.50	36.57	35.66	34.77	33.90
88	46.99	45.83	44.69	43.60	42.52	41.47	40.45	39.46	38.47	37.52	36.59	35.68	34.79	33.92
86	47.02	45.86	44.72	43.62	42.55	41.50	40.47	39.48	38.49	37.54	36.61	35.70	34.81	33.94

Enthalpy at saturation, Btu per pound of dry air at 29.92”

Dry Bulb on left, Wet Bulb on top

Table 11 B

Enthalpy at saturation, Btu per pound of dry air at 29.92”

Dry Bulb on left, Wet Bulb on top

	84	83	82	81	80	79	78	77	76	75	74	73	72	71	70
86	48.20	47.02	45.86	44.73	43.62	42.55	41.50	40.47	39.47	38.49	37.54	36.61	35.70	34.81	33.94
84	48.23	47.04	45.88	44.75	43.65	42.57	41.52	40.49	39.49	38.51	37.56	36.63	35.72	34.83	33.96
82			45.90	44.77	43.67	42.59	41.54	40.51	39.51	38.53	37.58	36.65	35.74	34.85	33.98
80					43.69	42.62	41.56	40.54	39.53	38.55	37.60	36.67	35.76	34.87	34.00
78							41.58	40.56	39.55	38.57	37.62	36.69	35.77	34.88	34.02
76									39.57	38.59	37.64	36.71	35.79	34.90	34.03
74											37.66	36.72	35.81	34.92	34.05
72													35.83	34.94	34.07
70															34.09

Table 11 C

Enthalpy at saturation, Btu per pound of dry air at 29.29”

Dry Bulb on left, Wet Bulb on top

	69	68	67	66	65	64	63	62	61	60	59	58	57	56
86	33.10	32.27	31.46	30.67	29.90	29.14	28.40	27.68	26.98	26.29	25.61	24.95	24.31	23.67
84	33.11	32.29	31.48	30.69	29.91	29.16	28.42	27.70	26.99	26.30	25.62	24.96	24.32	23.68
82	33.13	32.30	31.49	30.70	29.93	29.17	28.43	27.71	27.00	26.31	25.64	24.98	24.33	23.70
80	33.15	32.32	31.51	30.72	29.94	29.19	28.45	27.73	27.02	26.33	25.65	24.99	24.34	23.71
78	33.17	32.34	31.53	30.73	29.96	29.20	28.46	27.74	27.03	26.34	25.66	25.00	24.35	23.72
76	33.18	32.35	31.54	30.75	29.98	29.22	28.48	27.75	27.05	26.35	25.68	25.01	24.37	23.73
74	33.20	32.37	31.56	30.77	29.99	29.23	28.49	27.77	27.06	26.37	25.69	25.03	24.38	23.74
72	33.22	32.39	31.58	30.78	30.01	29.25	28.51	27.78	27.07	26.38	25.70	25.04	24.39	23.75
70	33.24	32.41	31.59	30.80	30.02	29.26	28.52	27.80	27.09	26.39	25.71	25.05	24.40	23.76
68		32.42	31.61	30.81	30.04	29.28	28.54	27.81	27.10	26.41	25.73	25.06	24.41	23.78
66				30.83	30.06	29.30	28.55	27.82	27.11	26.42	25.74	25.07	24.42	23.79
64						29.31	28.57	27.84	27.13	26.43	25.75	25.09	24.44	23.80
62								27.85	27.14	26.45	25.76	25.10	24.45	23.81
60										26.46	25.78	25.11	24.46	23.82

Table 11 D

Enthalpy at saturation, Btu per pound of dry air at 29.92”

Dry Bulb on left, Wet Bulb on top

	58	57	56	55	54	53	52	51	50	49	48	47	46	45
68	25.06	24.41	23.78	23.15	22.54	21.94	21.36	20.78	20.21	19.66	19.12	18.61	18.09	17.58
66	25.07	24.42	23.79	23.16	22.55	21.95	21.37	20.79	20.22	19.67	19.13	18.62	18.10	17.59
64	25.09	24.44	23.80	23.17	22.57	21.96	21.38	20.80	20.23	19.68	19.14	18.63	18.11	17.59
62	25.10	24.45	23.81	23.18	22.58	21.97	21.39	20.81	20.24	19.69	19.15	18.64	18.12	17.60
60	25.11	24.46	23.82	23.19	22.59	21.98	21.40	20.82	20.25	19.70	19.16	18.64	18.13	17.60
58	25.12	24.47	23.83	23.20	22.60	21.99	21.41	20.83	20.26	19.71	19.17	18.65	18.13	17.61
56			23.84	23.21	22.61	22.00	21.42	20.84	20.27	19.72	19.18	18.65	18.14	17.61
54					22.62	22.01	21.43	20.85	20.28	19.73	19.18	18.66	18.14	17.62
52							21.44	20.86	20.29	19.74	19.19	18.66	18.15	17.62
50									20.30	19.75	19.21	18.67	18.15	17.63
48											19.21	18.67	18.16	17.63
46													18.16	17.64

To calculate the total heat removed from air that has gone through an evaporator

Use a digital sling psychrometer or a sling psychrometer and; familiarize yourself with its operation.

(1) Wet the bulb with distilled water for a sling psychrometer; obtain the wet bulb and dry bulb reading of the air going into the fan coil. Leave the psychrometer in the air until the wet bulb stops moving (this may take 5 minutes or more).

Wet bulb temperature_____________°F

Dry bulb temperature _____________°F

(2) Using a digital sling psychrometer or the sling psychrometer obtain the wet bulb and dry bulb reading of the air coming out of the fan coil. Leave the psychrometer in the air until the wet bulb stops moving (this may take 5 minutes).

Wet bulb temperature____________°F

Dry bulb temperature ____________°F

(3) On Table 8 A&B find, the Foot^3 of the leaving air of the evaporate.

(4) Divide the Foot^3 by one pound of air; this is the air density of the air. Air density____________________ or use Table 9B or 9C for air density

(5) Find the CFM of the system

CFM______________________

(6) Using the Wet bulb find the enthalpy of the air that is going in and out of the evaporator Tables 11

Enthalpy air in ____________ EP

Enthalpy air out ____________ EP

Difference in EP ____________ΔEP

(7) Run the formula; Air Density X 60 X CFM X ΔEP = BTUH

Example 6: Find the total heat that of a evaporator fan coil is absorbing from air with 4050 cfm and entering air of 76° dry bulb and 64°wet bulb, leaving air of 58°dry bulb and 54°wet bulb.

Solution

76° dry bulb and 66°wet bulb has a enthalpy of 30.75

58°dry bulb and 54°wet bulb has a enthalpy of 22.60 and a air density of 0.0757

Air density X 60 X cfm X Δ enthalpy = Btuh

0.0757 X 60 X 4050 X 8.15 = 149,920 Btuh

Example 7: Find the airflow and velocity fpm of a fan coil with a duct of 48 X 36 and a 75 gpm with an entering water of 45° and a leaving water 56°. On the airside, the entering air temperature is 78° dry bulb and 65° wet bulb with a leaving air temperature of 60° dry bulb and 56° wet bulb.

Solution

Btuh = gpm X 60 X 8.337 X Δ Temperature water

Cfm = Btuh / (air density X 60 X Δ enthalpy (h1-h2)

Velocity fpm = airflow cfm / area in ft²

Air density of leaving air 60º dry bulb 56º wet bulb = 0.0753

Enthalpy of entering air of 78º dry bulb 65º wet bulb = 29.96

Enthalpy of leaving air of 60º dry bulb 56º dry bulb = 23.82

Area in in² to ft² of a 48” X 36” duct: 48” X 36”= 1728 in² / 144 = 12 ft²

75 gpm X 60 X 8.337 X 11ºΔ Temperature = 412,681 Btuh

412,681btuh / (air density 0.0753 X 60 X [(h1) 29.96 – (h2) 23.82] = 14,876 cfm

14,876 cfm / 12 ft² = 1239 fpm

Kitchen Ventilation

Air Balancing

Balancing is best performed when the manufacturers of all the equipment can provide a certified reference method of measuring the airflows, rather than depending on generic measurements of duct flows or other forms of measurement in the field, which can be in error by 20% or more. The equipment manufacturer should be able to develop a reference method of measuring airflow in a portion of the equipment that is dynamically stable in the laboratory as well as in the field. This method should relate directly to airflow by graph or formula.

The general steps for air balancing in restaurants are as follows:

1. Exhaust hoods should be set to their proper flow rates, with supply and exhaust fans on.

2. Next, supply airflow rate, whether part of combined HVAC units or separate replacement air units, should be set to design values through the coils and the design supply flows from each outlet, with approximately correct settings on the outside airflow rate. Then, correct outside and return airflow rates should be set proportionately for each unit, as applicable. These settings should be made with exhaust on, to ensure adequate relief for the outside air. Where outside air and return air flows of a particular unit are expected to modulate, there should ideally be similar static losses through both airflow paths to preclude large changes in total supply air from the unit. Such changes, if large enough, could affect the efficiency of heat exchange and could change airflows within and between zones, thereby upsetting air distribution and balance.

3. Next, outside air should be set with all fans (exhaust and supply) operating. Pressure difference between inside and outside should be checked to see that (1) non kitchen zones of the building are at a positive pressure compared to outside and (2) kitchen zone pressure is negative compared to the surrounding zones and negative or neutral compared to outside. For applications with modulating exhaust, every step of exhaust

and replacement should be shut off, one-step at a time. Each combination of operation should be rechecked to ensure that design pressures and flows are maintained in each zone and between zones. This requires that the replacement airflow rate compensate automatically with each increment of exhaust. It may require some adjustments in controls or in damper linkage settings to get the correct proportional response.

4. When the preceding steps are complete, the system is properly integrated and balanced. At this time, all fan speeds and damper settings (at all modes of operation) should be permanently marked on the equipment and in the test and balance report. Air balance records of exhaust, supply, return, fresh air, and individual register airflows must also be completed. These records should be kept by the food service facility for future reference.

5. For new facilities, after two or three days in operation (no longer than a week and usually before the facility opens), all belts in the system should be checked and readjusted because new belts wear in quickly and could begin slipping.

6. Once the facility is operational, the performance of the ventilation system should be checked to verify that the design is adequate for actual operation, particularly at maximum cooking and at outside environmental extremes. Any necessary changes should be made, and all the records should be updated to show the changes.

Rechecking the air balance should not be necessary more than once every 2 years unless basic changes are made in facility operation. If there are any changes, such as adding a new type of cooking equipment or deleting exhaust connections, the system should be modified accordingly.

Multiple-Hood Systems

Kitchens exhaust systems serving more than a single hood present several design challenges not encountered with single-hood systems. One of the main challenges of multiple-hood exhaust systems is air balancing. Because balancing dampers are not permitted in the exhaust ducts, the system must be balanced by design. Zoning may be desirable for a balanced design and to improve energy conservation. Hood accessories are now available to allow balancing at individual hoods. Additionally, most filters come in varying sizes to allow pressure loss equalization at varying airflows. Some hoods and grease filters have adjustable baffles that allow airflow to be adjusted at the hood. These may be helpful for relatively fine balancing, but the system must provide most of the balancing. System zoning is preferred, because incorrect installation of a multi branch system can lead to complex problems. Adjustable filters should not be used when they can be interchanged between hoods or within the same hood because an interchange could disrupt the previously achieved balance. Balancing can also be accomplished by changing the number and/or size of filters.

For correct flow through a branch duct in a multiple-hood system, the static pressure loss of the branch must match the static pressure loss of the common duct upstream from the point of connection. Any exhaust points subsequently added or removed must be designed to comply with the minimum velocities required by code and to maintain the

balance of the remaining system. In cases such as master kitchen-exhaust systems, which are sometimes used in shopping center food courts, no single group is responsible for the entire design. The base building designer typically lays out ductwork to (or through) each tenant space and each tenant selects a hood and lays out connecting ductwork. Often the base building designer has incomplete information on tenant exhaust requirements. Therefore, one engineer must be responsible for defining criteria for each tenant's design and for evaluating proposed tenant work to ensure that tenant designs match the system's capacity. The engineer should also evaluate any proposed changes to the system, such as changing tenancy. Rudimentary computer modeling of the exhaust system may be helpful (Elovitz 1992). Given the unpredictability and volatility of tenant requirements, it may not be possible to balance the entire system perfectly. However, without adequate supervision, the probability that at least part of the system will be badly out of balance is very high.

For greatest success with multiple-hood exhaust systems, minimize pressure losses in the ducts by keeping velocities low, minimizing sharp transitions, and using hoods with relatively high-pressure drops. When pressure loss in the ducts is low compared to the loss through the hood, changes in pressure loss in the ductwork because of field conditions or changes in design airflow will have a smaller effect on total pressure loss and thus on actual airflow.

Minimum code-required air velocity must be maintained in all parts of the exhaust ductwork at all times. If fewer or smaller hoods are installed than the design anticipated, resulting in low velocity in portions of the ductwork, the velocity must be brought up to the minimum. One way is to introduce replacement air, preferably untempered, directly into the exhaust duct where it is required (see Figure 13). The bypass duct should connect to the top or sides (at least 2 in. from the bottom) of the exhaust duct to prevent backflow of water or grease through the bypass duct when fans are off. This arrangement is shown in NFPA *Standard* 96 and should be discussed with the authority having jurisdiction.

A fire damper should be provided in the bypass duct, located close to the exhaust duct. Bypass duct construction should be the same as the exhaust duct construction, including enclosure and clearance requirements, for at least several feet beyond the fire damper. Means to adjust the bypass airflow must be provided upstream of the fire damper. All dampers must be in the clean bypass air duct so they are not exposed to grease-laden exhaust air. The difference in pressure between replacement and exhaust air

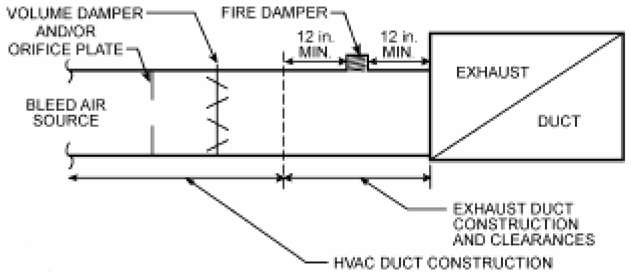


Fig. 13 Method of Introducing Replacement Air Directly into Exhaust Duct

duct may be great; the balancing device must be able to make a fine airflow adjustment against this pressure difference. It is best to provide two balancing devices in series, such as an orifice plate or blast gate for coarse adjustment followed by an opposed-blade damper for fine adjustment.

Directly measuring air velocities in the exhaust ductwork to assess exhaust system performance may be desirable. Velocity (pitot-tube) traverses may be performed in kitchen exhaust systems, but holes drilled for the pitot tube must be liquid tight to maintain the fire-safe integrity of the ductwork per NFPA *Standard* 96. Holes should never be drilled in the bottom of a duct, where they may collect grease. Velocity traverses should not be performed when cooking is in progress because grease collects on the instrumentation.

PRINCIPLES AND PROCEDURES FOR BALANCING HYDRONIC SYSTEMS

Both air- and water-side balance techniques must be performed with sufficient accuracy to ensure that the system operates economically, with minimum energy, and with proper distribution. Airside balance requires precise flow measuring because air, which is usually the prime heating or cooling transport medium, is more difficult to measure in the field. Reducing airflow below design airflow reduces heat transfer directly and linearly with respect to waterside flow. In contrast, the heat transfer rate for the waterside does not vary linearly with the water flow rate through a heat exchanger (coil), because of the characteristics of heat exchangers. As a result, waterside heat transfer is less sensitive to changes in flow, and the required accuracy of flow is lower when using traditional design criteria. The relatively high pressures associated with hydronic systems allow for easier measurement of pressure, although application of flow and head relationships should be thoroughly understood.

Heat Transfer at Reduced Flow Rate

The typical heating-only hydronic terminal (200°F, 20°FΔ*t*) gradually reduces heat output as flow is reduced (Figure 1). Decreasing water flow to 20% of design reduces heat transfer to 65% of that at full design flow. The control valve must reduce water flow to 10% to reduce heat output to 50%. This relative insensitivity to changing flow rates is because the governing coefficient for heat transfer is the airside coefficient; a change in internal or waterside coefficient with flow rate does not materially affect the overall heat transfer coefficient. This means (1) heat transfer for water-to-air terminals is established by the mean air-to-water temperature difference, (2) heat transfer is measurably changed, and (3) a change in mean water temperature requires a greater change in water flow rate.

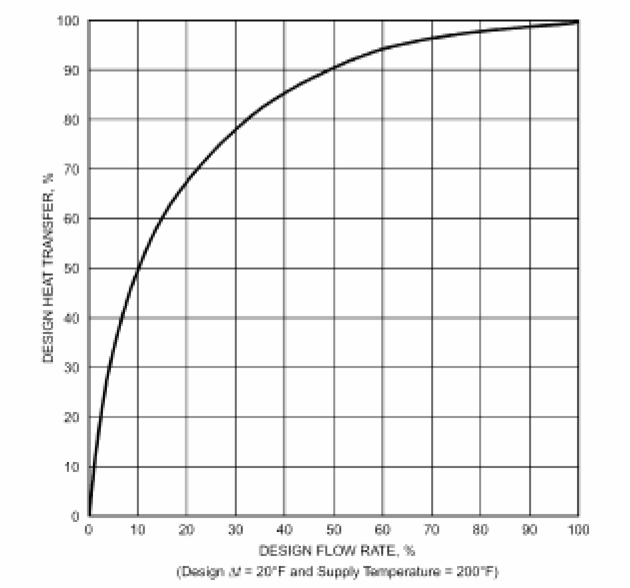


Fig. 1 Effects of Flow Variation on Heat Transfer

from a Hydronic Terminal

Tests of hydronic coil performance show that when flow is throttled to the coil, the waterside differential temperature of the coil increases with respect to design selection. This applies to both constant volume and variable-volume air-handling units. In constantly circulated coils that control temperature by changing coil entering water temperature, decreasing source flow to the circuit decreases the waterside differential temperature.

A secondary concern applies to heating terminals. Unlike chilled water, hot water can be supplied at a wide range of temperatures. Inadequate terminal heating capacity caused by insufficient flow can sometimes be overcome by raising supply water temperature.

Design below the 250°F limit (ASME low-pressure boiler code) must be considered.

Figure 2 shows the flow variation when 90% terminal capacity is acceptable. Note that heating tolerance decreases with temperature and flow rates and that chilled-water terminal are much less tolerant of flow variation than hot-water terminals.

Dual-temperature heating/cooling hydronic systems are sometimes first started during the heating season. Adequate heating ability in the terminals may suggest that the system is balanced. Figure 2 shows that 40% of design flow through the terminal provides 90% of design heating with 140°F supply water and a 10°F temperature drop. Increased supply water temperature establishes the same heat transfer at terminal flow rates of less than 40% design.

Sometimes, dual-temperature water systems have decreased flow during the cooling season because of chiller pressure drop; this could cause a flow reduction of 25%. For example, during the cooling season, a terminal that heated satisfactorily would only receive 30% of the design flow rate.

Although the example of reduced flow rate at Δt = 20°F only affects heat transfer by 10%, this reduced heat transfer rate may have the following negative effects:

- **Object of the system is to deliver (or remove) heat where required. When flow is reduced from design rate, system must supply heating or cooling for a longer period to maintain room temperature.**
- **As load reaches design conditions, the reduced flow rate is unable to maintain room design conditions.**
- **Control valves with average range ability (30:1) and reasonable authority (B = 0.5) may act as on-off controllers instead of throttling flows to the terminal. The resultant change in riser friction loss may cause overflow or underflow in other system terminals. Attempting to throttle may cause wear on the valve plug or seat because of higher velocities at the vena contract of the valve. In extreme situations, cavitations may occur.**

Terminals with lower water temperature drops have greater tolerance for unbalanced conditions. However, larger water flows are necessary, requiring larger pipes, pumps, and pumping cost.

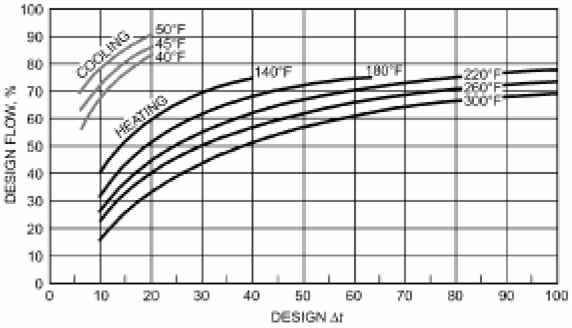


Fig. 2 Percent of Design Flow Versus Design Δ*t* to Maintain 90% Terminal Heat Transfer for Various Supply Water Temperatures

System balance becomes more important in terminals with a large temperature difference. Less water flow is required, which reduces the size of pipes, valves, and pumps, as well as pumping costs. A more linear emission curve gives better system control. If flow varies by more than 5% at design flow conditions, heat transfer can fall off rapidly, ultimately causing poorer control of the wet-bulb temperature and potentially decreasing system air quality.

Heat Transfer at Excessive Flow

Increasing the flow rate above design in an effort to increase heat transfer requires careful consideration. Figure 3 shows that increasing the flow to 200% of design only increases heat transfer by 6% but increases resistance or pressure drop four times and power by the cube of the original power (pump laws) for a lower design Δ*t*. In coils with larger waterside design Δ*t*, heat transfer can increase.

Hydronic Pressure Measuring Instruments

Manometer U-tube

The U-tube manometer has primarily been used for measuring pressure drops across terminal heat exchangers and flow meters. There is no calibration required and other instruments can be verified against the manometer.

Digital Manometer for Water Measurement

There are several manufacturers of digital manometers and each has specific functions it can perform. The standard requirements of the digital manometer is it can read in English units or metric, has a broad range of pressures, can display in pounds per square inch (psi), foot head (R), inches of water column (in. wc), and kilo Pascal's (kPa). Each manometer should have auto zeroing capabilities.

The manufacturer's pressure rating and calibration instructions should be followed. When used to measure pressures, the calibration shall be verified by using a U-tube manometer or a calibrated pressure gauge. The calibration verification can also be accomplished by using a dead weight kit.

The accuracy of the gauge reading on differential pressure shall be ±2% of reading ±1 digit.

Differential Pressure Gauges

The differential pressure gauge is used to measure pressure differentials across terminal heat exchangers and flow meters. The manufacturer's pressure ratings and calibration instructions shall be followed. The instrument calibration verification shall be performed by comparing the instrument readings with a U-tube manometer and/or a calibrated gauge.

The accuracy of the instrument shall be ±1/2 % of full-scale differential pressure in ranges 0-30" wc to 0-200 psi (75 kpa to 1379 kpa). in addition, ±3/4% of full-scale differential pressure for ranges above 200 psi (1379 kpa).

Test Gauges

The most commonly used gauge for measuring pressure in a hydronic system is the Bourdon tube gauge. The accuracy of the gauge shall be 0.5% of full scale. The gauge should be selected so the indicated pressure is between the middle and upper half of the scale. When measuring hydronic differential pressures across any device with a Bourdon tube gauge, the same gauge shall be used for all readings. Pulsations shall be dampened to stop fluctuations in the readings. The gauge must be located at the same elevation for all readings to eliminate the effect of the weight of the fluid in the system at different heights. The calibration shall be verified by checking against a sheltered gauge prior to use on every project and dead weight tested on *six-month* intervals.

Non-Intrusive Meters

These meters strap on to the pipe and use non-intrusive transducers to measure acoustical signals and determine the flow. The non-intrusive meter may be a fixed meter or a portable meter.

The transducers must be attached directly to a clean pipe without insulation. The following must be known regarding the pipe:

- Pipe internal diameter
- Pipe external diameter
- Pipe material
- W Pipe classification

There are two types of non-intrusive meters: the Transit Time meter and the Doppler meter.

Transit Time

The Transit Time meter is designed for clean liquid applications and is used in HVAC hydronic flow measurement. The pipe must be free of air. The manufacturer's installation information must be followed to maintain the meter's accuracy of ±2% of full scale.

Doppler

The Doppler meter is not suited for HVAC system hydronic measurement because it requires systems handling particulate matter to be accurate.

Hydronic Flow Measuring

Orifice Plate

A calibrated orifice plate is a plate permanently installed in the pipe that as been calibrated by measuring the pressure drop against a known flow. The accuracy is dependent on the pipe location and the instrument used to measure pressure differential (Figure 11.1). The orifice shall be located according to the manufacturer's recommendations.

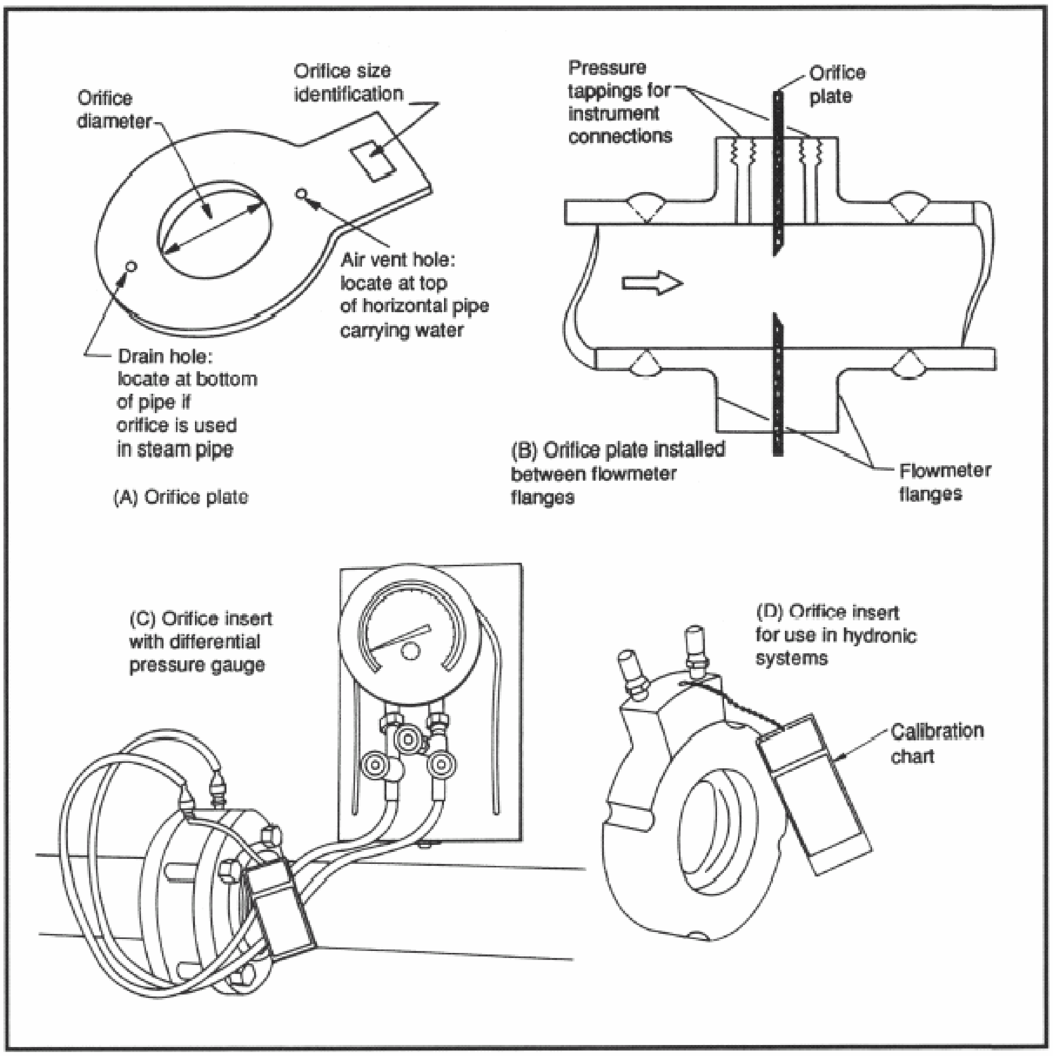


Fig. 11.1 Orifices used as flow-measuring devices in piping

Venturi

The venturi is a device permanently installed in the pipe that has been calibrated by measuring the pressure drop against a known flow. Venturis must be located according to the manufacturer's recommendations to maintain accuracy of 2% (Figure 11.2).

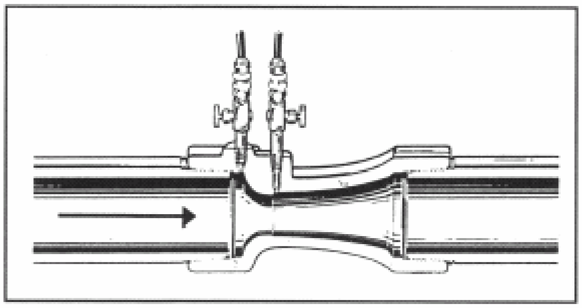


Fig. 11.2 A Venturi in a Piping System

Hydronic Pitot Tube Traverse

The Pitot rod is a pair of tubes in a casing. One tube measures a static head pressure, and the other tube which faces in the direction of flow, measures the total head pressure. The Pitot rod is a manually inserted device used to take a series of velocity measurements, which are averaged and used to determine flow. (Figures 11.3 and 11.4). A manometer is attached to measure flow.

GPM = FPM X Gallons per foot

FPM = 1040 X √ VP

Example 7A

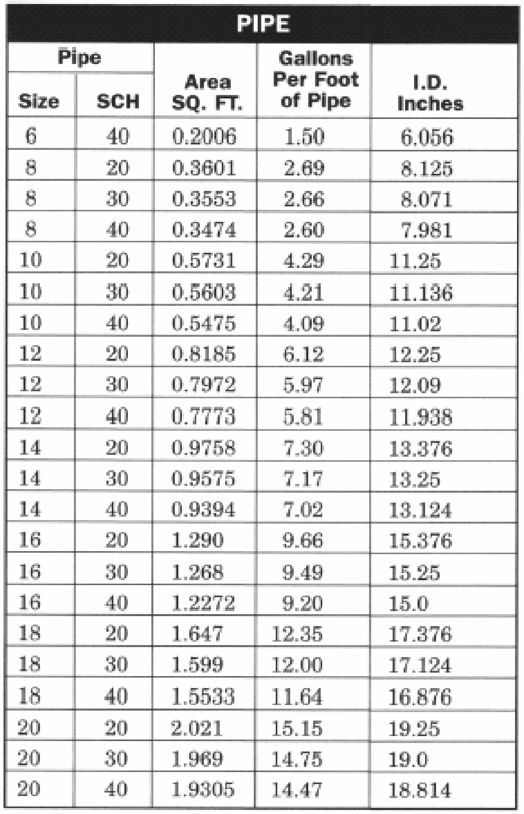
Find the Gpm of a system that has a Cfm of 10,000 with a entering air of 80°F Dry bulb, 67°F wet bulb and a leaving air of 62°F dry bulb , 57.5°F wet bulb. The water entering is 45°F and leaving is 55°F.

80°F dry with a 67°F wet is a 31.51 enthalpy and 62°F dry 57.5°F wet is a enthalpy of 24.77 with a air density of 0.0757

Solution

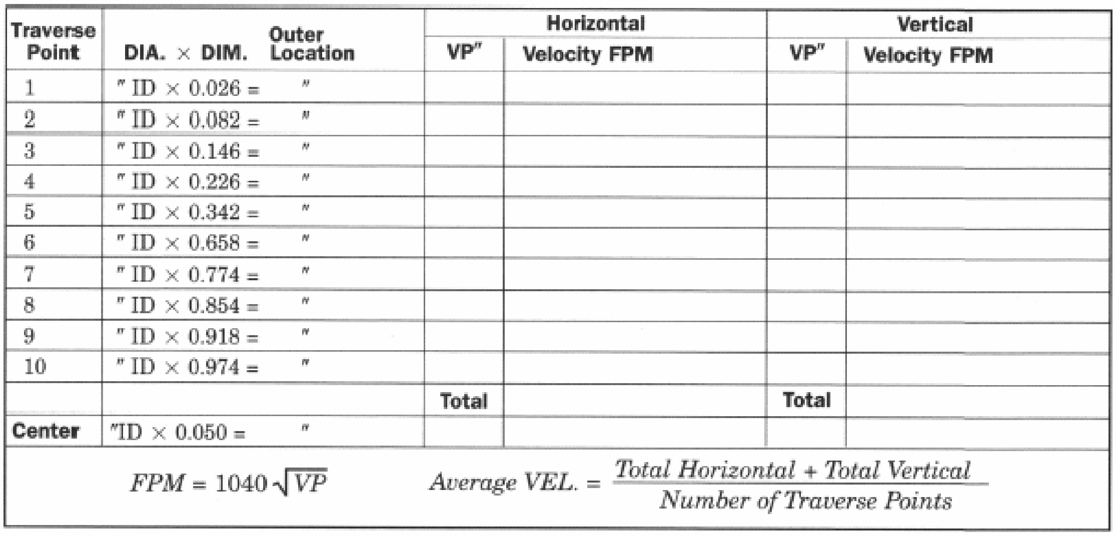
Gpm = Air Density X CFM X Δ Enthalpy / 8.337 X Δ temperature of the water

0.0757 X 10,000 X 6.74 / 8.337 X 10 = 5102.18 / 83.37 = 61.2 GPM



PIPE				
Pipe		Area SQ. FT.	Gallons Per Foot of Pipe	I.D. Inches
Size	SCH			
6	40	0.2006	1.50	6.056
8	20	0.3601	2.69	8.125
8	30	0.3553	2.66	8.071
8	40	0.3474	2.60	7.981
10	20	0.5731	4.29	11.25
10	30	0.5603	4.21	11.136
10	40	0.5475	4.09	11.02
12	20	0.8185	6.12	12.25
12	30	0.7972	5.97	12.09
12	40	0.7773	5.81	11.938
14	20	0.9758	7.30	13.376
14	30	0.9575	7.17	13.25
14	40	0.9394	7.02	13.124
16	20	1.290	9.66	15.376
16	30	1.268	9.49	15.25
16	40	1.2272	9.20	15.0
18	20	1.647	12.35	17.376
18	30	1.599	12.00	17.124
18	40	1.5533	11.64	16.876
20	20	2.021	15.15	19.25
20	30	1.969	14.75	19.0
20	40	1.9305	14.47	18.814

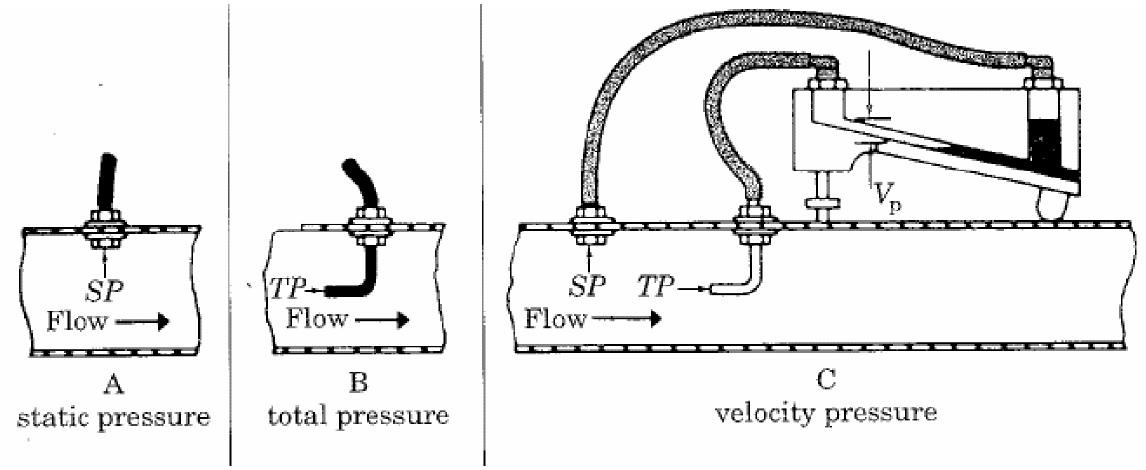
Fig. 11.3 Pipe Data



Traverse Point	DIA. × DIM.	Outer Location	Horizontal		Vertical	
			VP"	Velocity FPM	VP"	Velocity FPM
1	" ID × 0.026 =	"				
2	" ID × 0.082 =	"				
3	" ID × 0.146 =	"				
4	" ID × 0.226 =	"				
5	" ID × 0.342 =	"				
6	" ID × 0.658 =	"				
7	" ID × 0.774 =	"				
8	" ID × 0.854 =	"				
9	" ID × 0.918 =	"				
10	" ID × 0.974 =	"				
			Total		Total	
Center	"ID × 0.050 =	"				

$FPM = 1040\sqrt{VP}$ $\quad Average\ VEL. = \dfrac{Total\ Horizontal + Total\ Vertical}{Number\ of\ Traverse\ Points}$

Fig. 11.4 Calculating Flow Quantities from Hydronic Pitot Tube Traverses



Pressure Total pressure Velocity pressure

Example 8: Find the GPM of a 6" pipe SCH 40 with a pressure of 42.45 and a total pressure of 42.47. Use fig. 11.3 for Gallons per foot

Solution

- **Total pressure – pressure = velocity pressure**
- **1040 X √ velocity pressure = FPM**
- **FPM X Gallons per foot = GPM**
- **42.47 total pressure – 42.45 pressure = 0.02 velocity pressure**
- **1040 X √ 0.02 = 147.08 FPM**
- **147.08 FPM X 1.5 Gallons per foot = 220.62 GPM**

Example 9: Find the ton of a chiller of example 8 with an entering water of 55.8 °F and a leaving water of 45.2 °F.

Solution 1

- **Tons = GPM X 60 X 8.337 X Δ Temperature of the water / 12,000 btuh per ton**
- **220.62 gallons X 60 minutes X 8.337 X 10.6 ΔT = 1,169,800.4 btuh**
- **1,169,800.4 btuh / 12,000 btuh per ton = 97.48 tons**

Solution 2

- **Tons = GPM X Δ Temperature of the water X specific heat X specific gravity / 24**
- **220.62 gallons X 10.6 ΔT X 1 SH X 1 SG = 2338.57**
- **2338.57 / 24 = 97.44 tons**

Calibrated Balancing Valves

Calibrated balancing valves are designed as a flow meter and a balancing valve. The valve has pressure taps installed on the inlet and outlet. The manufacturer has calibrated the valve by measuring the resistance at various valve positions against known flow quantities. The valve has a graduated scale or dial to indicate the degree open. The calibration data shows the flow rate in gallons per minute (gpm) *(Us)* versus the measured pressure drop. Pressure drops are measured with an appropriate differential gauge. The flow is determined from the manufacturer's chart. The location of the valve must follow the manufacturer's recommendations.

Spring-loaded Constant Flow Devices

Spring-loaded constant flow devices are for constant flow systems. The valve is a spring-loaded piston that maintains *25%* flow as long as the pressure differential is within the manufacturer's range. Differential pressures must be taken to assure pressure drop is within the recommended range. These valves are sensitive to dirt and debris, and should have a strainer installed upstream.

Heat Exchanger Pressure Drop Taps

Manufacturers of heat exchangers, e.g., coils, evaporators, and condensers, have flow curves for their exchangers that provide flow in gpm vs. pressure drop in feet of water. To obtain optimal pressure drops, pressure taps must be located at the inlet and outlet connection of the exchanger prior to any turns or valves. Dirt, debris and scaling will provide an inaccurate flow measurement.

Generalized Chilled Water Terminal—Heat Transfer Versus Flow

Heat transfer for a typical chilled-water coil in an air duct versus water flow rate is shown in Figure 4. The curves are based on ARI rating points: 45°F inlet water at a 10°F rise with entering air at 80°F db and 67°F wb. The basic curve applies to catalog ratings for lower dry-bulb temperatures providing consistent entering air moisture content (e.g., 75°F db, 65°F wb). Changes in inlet water temperature, temperature rise, air velocity, and dry- and wet-bulb temperatures cause terminal performance to deviate from the curves. Figure 4 is only a general representation and does not apply to all chilled-water terminals. Comparing Figure 4 with Figure 1 indicates the similarity of the nonlinear heat transfer and flow for both the heating and cooling terminals.

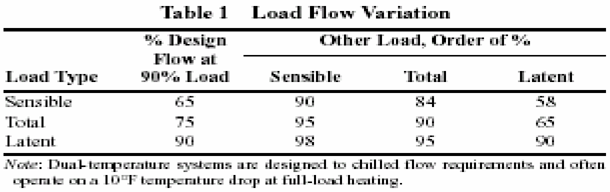


Table 1 Load Flow Variation

Load Type	% Design Flow at 90% Load	Other Load, Order of % Sensible	Total	Latent
Sensible	65	90	84	58
Total	75	95	90	65
Latent	90	98	95	90

Note: Dual-temperature systems are designed to chilled flow requirements and often operate on a 10°F temperature drop at full-load heating.

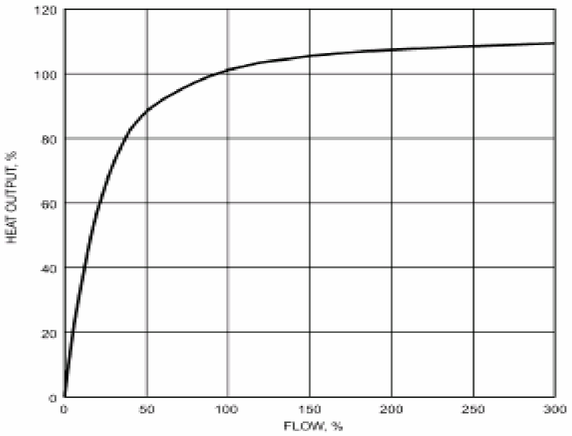


Fig. 3 Typical Heating Coil Heat Transfer Versus Water Flow

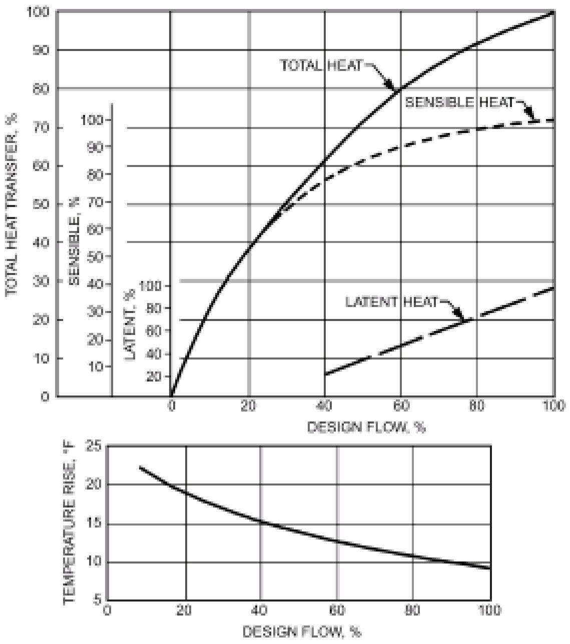


Fig. 4 Chilled Water Terminal Heat Transfer Versus Flow

Table 1 shows that if the coil is selected for the load and flow is reduced to 90% of load, three flow variations can satisfy the reduced load at various sensible and latent combinations.

Flow Tolerance and Balance Procedure

The design procedure rests on a design flow rate and an allowable flow tolerance. The designer must define both the terminal's flow rates and feasible flow tolerance, remembering that the cost of balancing rises with tightened flow tolerance. Any overflow increases pumping cost, and any flow decrease reduces the maximum heating or cooling at design conditions.

WATER-SIDE BALANCING

Waterside balancing adjustments should be made with a thorough understanding of piping friction loss calculations and measured system pressure losses. It is good practice to show expected losses of pipes, fittings, and terminals and expected pressures in operation on schematic system drawings.

The waterside should tested by direct flow measurement. This method is accurate because it deals with system flow as a function of differential pressures, and avoids compounding errors introduced by temperature difference procedures. Measuring flow at each terminal enables proportional balancing and, ultimately, matching pump head and flow to actual system requirements by trimming the pump impeller or reducing pump motor power. Often, reducing pump-operating cost will pay for the cost of waterside balancing.

Equipment

Proper equipment selection and preplanning are needed to successfully balance hydronic systems. Circumstances sometimes dictate that flow, temperature, and pressure be measured. The designer should specify the water flow balancing devices for installation during construction and testing during hydronic system balancing. The devices may consist of all or some of the following:

- Flow meters (ultrasonic stations, turbines, venturi, orifice plate, multiported pitot tubes, and flow indicators)
- Manometers, ultrasonic digital meters, and differential pressure gages (analog or digital)
- Portable digital meter to measure flow and pressure drop
- Portable pyrometers to measure temperature differentials when test wells are not provided
- Test pressure taps, pressure gages, thermometers, and wells.
- Balancing valve with a factory-rated flow coefficient *Cv*, a flow versus handle position and pressure drop table, or a slide rule flow calculator
- Dynamic balancing valves or flow-limiting valves (for pre-balanced systems only); field

adjustment of these devices is not normally required or possible

- **Pumps with factory-certified pump curves**

- **Components used as flow meters (terminal coils, chillers, heat exchangers, or control valves if using manufacturer's factory certified flow versus pressure drop curves); not recommended as a replacement for metering stations**

Record Keeping

Balancing requires accurate record keeping while making field measurements. Dated and signed field test reports help the designer or customer in work approval, and the owner has a valuable reference when documenting future changes.

Sizing Balancing Valves

A balancing valve is placed in the system to adjust water flow to a terminal, branch, zone, riser, or main. It should be located on the leaving side of the hydronic branch. General branch layout is from takeoff to entering service valve, then to the coil, control valve, and balancing/service valve. Pressure is thereby left on the coil, helping keep dissolved air in solution and preventing false balance problems resulting from air bind.

A common valve sizing method is to select for line size; however, balancing valves should be selected to pass design flows when near or at their fully open position with 12 in. of water minimum pressure drop. Larger *P* is recommended for accurate pressure readings. Many balancing valves and measuring meters give an accuracy of ±5% of range down to a pressure drop of 12 in. of water with the balancing valve wide open. Too large of a balancing valve pressure drop will affect the performance and flow characteristic of the control valve. Too small a pressure drop will affect its flow measurement accuracy as it is closed to balance the system. Equation (2) may be used to determine the flow coefficient *Cv* for a balancing valve or to size a control valve.

The flow coefficient *CV* is defined as the number of gallons of water per minute that flows through a wide-open valve with a pressure drop of 1 psi at 60°F. This is shown as $\{CV = Q \times \sqrt{(SF / \Delta P)}\} \quad (2)$

$Q = CV / \sqrt{(SF / \Delta P)}$

Where

CV **= flow coefficient at 1 psi drop**

Q **= design flow for terminal or valve, gpm**

ΔP **= pressure drop, psi**

ΔH **= pressure drop, ft of water**

SF = specific gravity of fluid

If pressure drop is determined in feet of water, Equation (2) can

be shown as {CV = 1.5 X Q √ (SF / ΔH)} (3)

HYDRONIC BALANCING METHODS

Various techniques are used to balance hydronic systems. Balance by temperature difference and water balance by proportional method is the most common.

Preparation

Minimally, preparation before balancing should include collecting the following:

1. Pump submittal data; pump curves, motor data, etc

2. Starter sizes and overloads protection information

3. Control valve *Cv* ratings and temperature control diagrams

4. Chiller, boiler, and heat exchanger information; flow head loss

5. Terminal unit information; flow head data

6. Pressure relief and reducing valve setting

7. Flow meter calibration curves

8. Other pertinent data such as readout conversion charts

System Preparation for Static System

1. Examine piping system: Identify main pipes, risers, branches and terminals on as-built drawings. Check that flows for all balancing devices are indicated on drawings before beginning work. Check that design flows for each riser equal the sum of the design flows through the terminals.

2. Examine reducing valve

3. Examine pressure relief valves

4. Examine expansion tank

5. For pumps, confirm

Location and size

Vented volute

Alignment

Grouting

Motor and lubrication

Nameplate data

Pump rotational direction

6. For strainers, confirm

Location and size

Mesh size and cleanliness

7. Confirm location and size of terminal units

8. Control valves:

Confirm location and size

Confirm port locations and flow direction

Set all valves open to coil

Confirm actuator has required force to close valve under loaded conditions

9. Ensure calibration of all measuring instruments, and that all calibration

Data are known for balancing devices

Pump Start-Up

1. Set pump throttle valve to nearly closed position.

2. Start pump and confirm rotational direction; rewire if rotation is incorrect.

3. Open throttle valve slowly until differential head readout applied to the pump curve indicates that flow approximates design.

4. Slowly close pump throttle valve to shutoff. Read pump differential head from gages.

If shutoff head corresponds with the published curve, the previously prepared velocity head correction curve can be used as a pump flow calibration curve.

A significant difference between observed and published shutoff head can be caused by an unvented volute, a partially plugged impeller, or by an impeller size different from that specified.

Confirmation of System Venting

1. Confirm tank location and size.

2. Shut off pump; record shutoff gauge pressure at tank junction.

3. Start pump and record operating pressure at tank junction.

4. Compare operating to shutoff pressures at tank junction. If there is no pressure change, the system is air-free.

5. Eliminate free air.

No air separation: Shut off pump and re-vent. Retest and re-vent until tank junction pressure is stable.

Air separation: Operate system until free air has been separated out, indicated by stable tank junction pressure.

Balancing

For single-, multiple-, and parallel pump systems, after pump start-up and confirmation of system venting,

1. Adjust pump throttle until pump head differential corresponds to design.

2. Record pump motor voltage and amperage, and pump strainer head, at design flow.

3. Balance equipment room piping circuit so that pumped flow remains constant over alternative flow paths.

4. Record chiller and boiler circuits (for multiple-pump systems, requires a flow meter installed between header piping). For multiple-pump systems only,

5. Check for variable flow in source circuits when control valves are operated.

6. Confirm

Pump suction pressure remains above cavitations range for all operating conditions.

Pump flow rates remain constant.

Source working pressures are unaffected. For parallel-pump systems, follow Steps (1) to (4), then shut off pumps alternately and

5. Record head differential and flow rate through operating pump, and operating pump motor voltage and current.

6. Confirm that operational point is satisfactory (no overload, cavitation potential, etc.).

Balance by Temperature Difference

This common balancing procedure is based on measuring the water temperature difference between supply and return at the terminal. The designer selects the cooling and/or heating terminal for a calculated design load at full-load conditions. At less than full load, which is true for most operating hours, the temperature drop is proportionately less. Figure 5 demonstrates this relationship for a heating system at a design ΔT of 20°F for outside design of 10°F and room design of 70°F.

For every outside temperature other than design, the balancing technician should construct a similar chart and read off the ΔT for balancing. For example, at 50% load, or 30°F outside air, the ΔT required is 10°F, or 50% of the design drop.

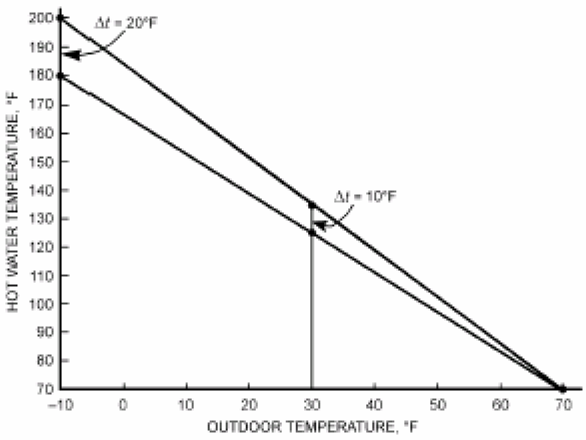


Fig. 5 Water Temperature Versus Outside Temperature Showing Approximate Temperature Difference

This method is a rough approximation and should not be used where great accuracy is required. It is not accurate enough for cooling or heating systems with a large temperature drop.

Water Balance by Proportional Method

Preset Method. A thorough understanding of the pressure drops in the system riser piping, branches, coils, control valves and balancing valves is needed. Generally, several pipe and valve sizes are available for designing systems with high or low-pressure drops. A flow-limiting or trim device will be required. Knowing system pressure losses in design allows the designer to select a balancing device to absorb excess system pressures in the branch, and to shift pressure drop (which might be absorbed by a balancing device nearly close to achieve balance) to the pipes, coils, and valves so the balancing device merely trims these components' performance at design flow. It may also indicate where high-head-loss circuits can exist for either relocation in the piping network, or hydraulic isolation through hybrid piping techniques. The installed balancing device should never be closed more than 40 to 50%; below this point flow reading accuracy falls to ±20 to 30%. Knowing a starting point for setting the valve (preset) allows the designer to iterate system-piping design. This may not always be practical in large systems, but minimizing head and flow saves energy over the life of the facility and allows for proper temperature control. In this method,

1. Analyze the piping network for the largest hydraulic loss based on design flow and pipe friction loss. The pump should be selected to provide the total of all terminal flows, and the head required to move water through the hydraulically greatest circuit. Balance devices in this circuit should be sized only for the loss required for flow measurement

accuracy. Trimming is not required.

2. Analyze differences in pressure drop in the pumping circuit for each terminal without using a balancing device. The difference between each circuit and the pump head (which represents the drop in the farthest circuit) is the required drop for the balancing device.

3. Select a balancing device that will achieve this drop with minimum valve throttling. If greater than two pipe sizes smaller, shift design drop into control valve or coil (or both), equalizing pressure drop across the devices.

4. Monitor system elevations and pressure drops to ensure air management, minimizing pocket collections and false pressure references that could lead to phantom balancing problems. 5. Use proportional balancing methods as outlined for field-testing and adjustment.

Proportional Balancing

Proportional waterside balancing may use design data, but relies most on as-built conditions and measurements and adapts well to design diversity factors. This method works well with multiple-riser systems. When several terminals are connected to the same circuit, any variation of differential pressure at the circuit inlet affects flows in all other units in the same proportion. Circuits are proportionally balanced to each other by a flow quotient:

Flow quotient = Actual flow rate / Design flow rate (4)

To balance a branch system proportionally,

1. Fully open the balancing and control valves in that circuit.

2. Adjust the main balancing valve for total pump flow of 100 to 110% of design flow.

3. Calculate each riser valve's quotient based on actual measurements. Record these values on the test form, and note the circuit with the lowest flow quotient.

***Note*: When all balancing devices are open, flow will be higher in some circuits than others. In some, flow may be so low that it cannot be accurately measured. The situation is complicated because an initial pressure drop in series with the pump is necessary to limit total flow to 100 to 110% of design; this decreases the available differential pressure for the distribution system. After all other risers are balanced; restart analysis of risers with unmeasurable flow at Step (2).**

4. Identify the riser with the highest flow ratio. Begin balancing with this riser, then continue to the next highest flow ratio, and so on. When selecting the branch with the highest flow ratio,

Measure flow in all branches of the selected riser.

In branches with flow higher than 150% of design, close the balancing valves to reduce flow to about 110% of design.

Readjust total pump flow using the main valve.

Start balancing in branches with a flow ratio greater than or equal to 1. Start with the branch with the highest flow ratio.

The reference circuit has the lowest quotient and the greatest pressure loss. Adjust all other balancing valves in that branch until they have the same quotient as the reference circuit (at least one valve in the branch should be fully open).

When a second valve is adjusted, the flow quotient in the reference valve will also change; continued adjustment is required to make their flow quotients equal. Once they are equal, they will remain equal or in proportional balance to each other while other valves in the branch are adjusted or until there is a change in pressure or flow.

When all balancing valves are adjusted to their branches' respective flow quotients, total system water flow is adjusted to the design by setting the balancing valve at the pump discharge to a flow quotient of 1.

Pressure drop across the balancing valve at pump discharge is produced by the pump that is not required to provide design flow. This excess pressure can be removed by trimming the pump impeller or reducing pump speed. The pump discharge-balancing valve must then be reopened fully to provide the design flow.

As in variable-speed pumping, diversity and flow changes are well accommodated by a system that has been proportionately balanced. Because the balancing valves have been balanced to each other at a particular flow (design), any changes in flow are proportionately distributed.

Balancing the waterside in a system that uses diversity must be done at full flow. Because the components are selected based on heat transfer at full flow, they must be balanced to this point. To accomplish full-flow proportional balance, shut off part of the system while balancing the remaining sections. When a section has been balanced, shut it off and open the section that was open originally to complete full balance of the system. When balancing, care should be taken if the building is occupied or if load is

nearly full.

Variable-Speed Pumping. To achieve hydronic balance, full flow through the system is required during balancing, after which the system can be placed on automatic control and the pump speed allowed changing. After the full-flow condition is balanced and the system differential pressure set point is established, to control the variable-speed pumps, observe the flow on the circuit with the greatest resistance as the other circuits are closed one at a time. The flow in the observed circuit should remain equal to, or more than, the previously set flow. Water flow may become laminar at less than 2fps, which may alter the heat transfer characteristics of the system.

Pump Hydronic Volume Measurement

Gauge Locations

To measure the pumping head, connect a test gauge-to-gauge taps on the pump (Figure 11.5). Some pumps are manufactured with gauge taps at the inlet and outlet flanges or pump body. If these are not present, taps should be installed:

1. In the suction and discharge pipe, as close to the pump as possible.
2. With no fittings between the pump and the taps
3. With gauge cocks
4. So that gauges can be installed at the same height with respect to pump centerline. If this is not possible, use a manifold-equipped test gauge or correct for the height difference.

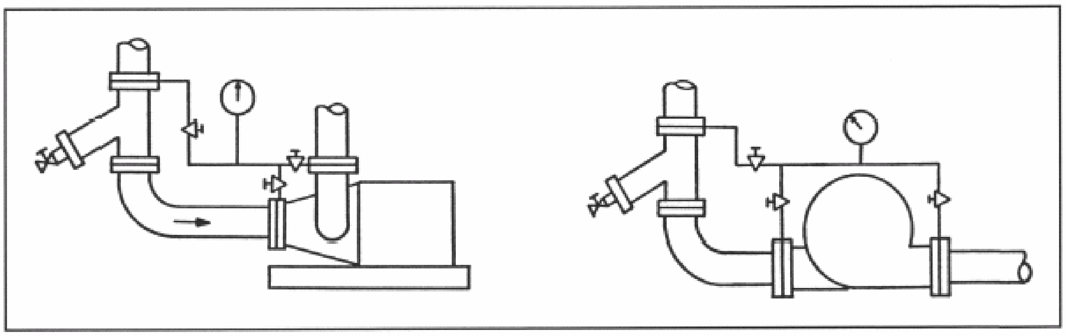


Fig. 11.5 End suction pump and Double suction pump

Verification of Impeller Size

In order to determine the amount of flow in the system using the pump, the impeller size must be verified.

1. Obtain the correct pump curve.

2. Turn the pump off

3. Close the discharge valve. Verify that the suction valve is fully open. The suction valve should always be fully open when the pump is in normal operation.

4. Turn the pump on.

5. Using a calibrated test gauge, measure the pressure on the suction and discharge sides of the pump (differential head pressure)

6. Calculate the rise across the pump by subtracting the suction pressure from the discharge pressure. If the suction is in a vacuum, convert the reading (inches Hg) to psi and add the result to the discharge pressure [inches Hg / 2.05 = PSI].

7. Multiply the pump psi rise by 2.31 feet to obtain rise in feet of water.

8. Plot the rise on the vertical axis of the pump curve at zero flow. This indicates the starting point of the flow curve. The impeller size is determined by comparing the adjacent listed sizes on the pump curve

Determine Pump Flow

1. Verify all valves in the system are fully open

2. **Open the pump discharge valve to its full-open position**

3. **Determine the flow head by measuring the discharge pressure and the suction pressure of the pump using the same calibrated test gauge**.

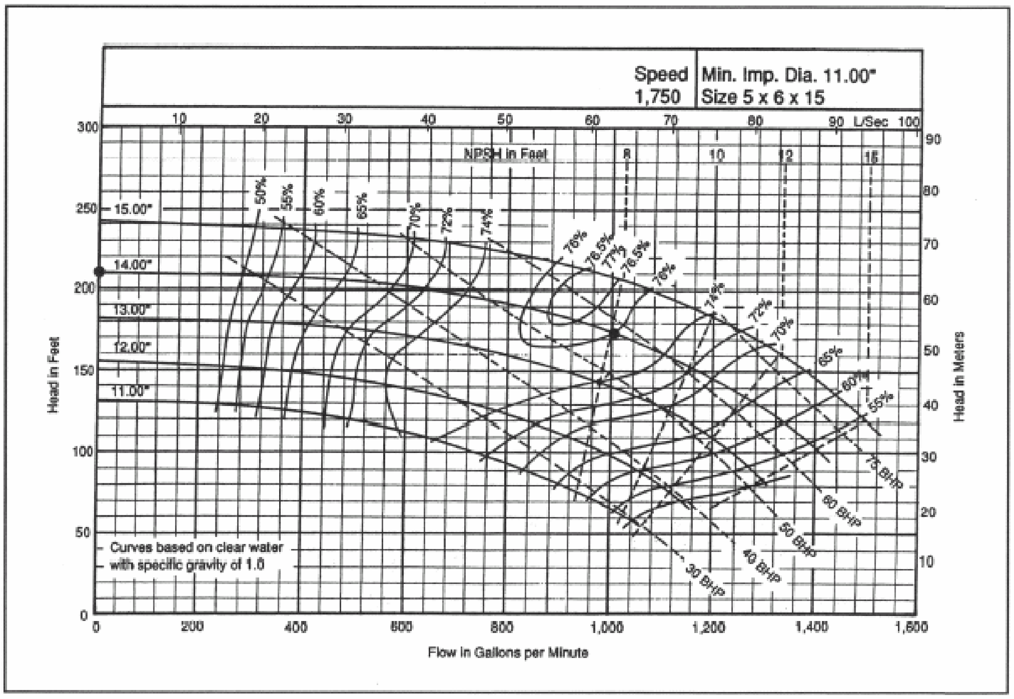


Fig. 11.6 Standard pump curve operating point with a 210' shut-off head operating at 175' and 1000 gpm. Motor should be 75 hp to be non-overloading.

4. Calculate the psi (pa) rise across the pump by subtracting the suction pressure from the discharge pressure.

5. Multiply the pump psi rise by 2.31 feet/psi (110 pa) to obtain rise in feet of water.

6. Plot the rise on the pump curve using the determined shut-off head curve and establish flow (Figure 11.6). Include pump curve in report.

7. Measure the motor amperage and voltage. Compare the amperage to the motor nameplate. Note: The pump should be non-overloading through the entire operating range, if not, note what condition will cause overloading.

8. Record the motor nameplate data, the actual rpm, amperage, voltage, and Watts.

9. Record the pump nameplate showing the pump size, impeller size, gpm (Vs), and head. Record the actual operating head and actual shut-off head, discharge and suction pressure.

Centrifugal Pump Performance

Suction Conditions - Cavitations

The suction conditions of a centrifugal pump are among the most important factors affecting its operation. After water enters the pump inlet, pressure between the pump inlet and the eye of the impeller is reduced. This region within the pump has the lowest pressure of the entire system. It is critical that this pressure not be lower than the vapor pressure of the liquid being pumped. Because centrifugal pumps are designed to pump liquids and not vapor, when a liquid vaporizes in the pump, one or more of the following symptoms may occur: snapping and crackling noises at the inlet, severe vibration, a drop in head and brake horsepower, a reduction in flow, or complete stoppage of flow. This condition is called "cavitation" and usually results in pitting and erosion of the vane tips or impeller inlet, thus reducing pump life.

Net Positive Suction Head (NPSH)

To eliminate cavitations problems it is necessary to maintain a minimum suction pressure at the pump inlet to overcome the internal losses in the pump. This minimum pressure, called net positive suction head (NPSH), combines all the factors limiting the suction side of the pump: internal losses, elevation of the suction supply, friction losses, vapor pressure, and

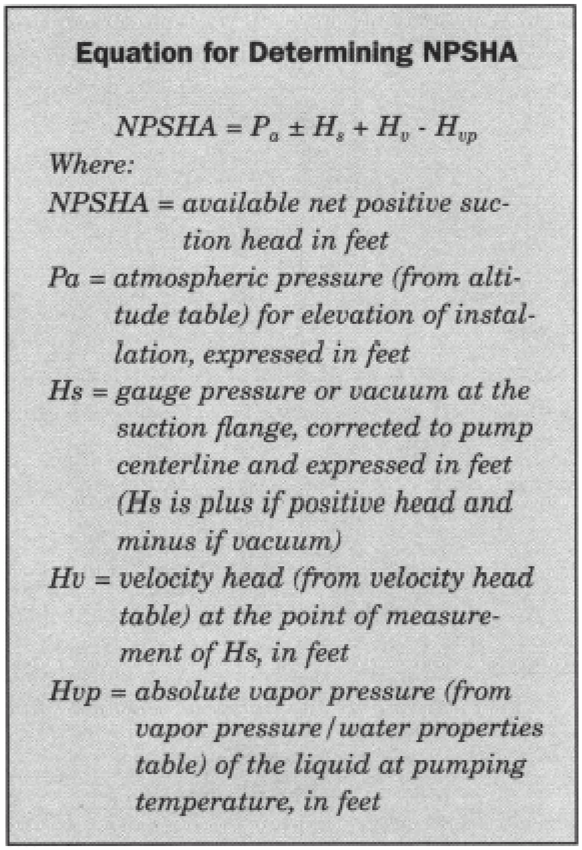
altitude of the installation. NPSH is further broken down into required net positive suction head and available net positive suction head.

Net positive suction head required (NPSHR) refers to the internal pump losses and is a characteristic of the pump design. It is the actual absolute pressure required by the pump to overcome the pump's internal losses and allow the pump to operate satisfactorily. A pump curve (Figure 11.6) will give the full range of NPSHR values for each impeller size and capacity.

Net positive suction head available (NPSHA) is a characteristic of the system in which the pump operates. It depends on such conditions as elevation of the suction supply in relation to the pump, the friction loss in the suction pipe, the altitude of the installation, or the pressure on the suction supply, and vapor pressure.

Both available and required NPSH vary with capacity for a given pump and suction system. The NPSHA is decreased as the capacity is increased due to the increased friction losses in the suction piping. The NPSHR is a function of the velocity and friction in the pump inlet, and therefore increases as the capacity increases. If it is necessary to determine the NPSHA while testing a system, use Equation 11.1. For an example, see

Appendix 11.1.



Equation for Determining NPSHA

$$NPSHA = P_a \pm H_s + H_v - H_{vp}$$

Where:

NPSHA = available net positive suction head in feet

Pa = atmospheric pressure (from altitude table) for elevation of installation, expressed in feet

Hs = gauge pressure or vacuum at the suction flange, corrected to pump centerline and expressed in feet (Hs is plus if positive head and minus if vacuum)

Hv = velocity head (from velocity head table) at the point of measurement of Hs, in feet

Hvp = absolute vapor pressure (from vapor pressure / water properties table) of the liquid at pumping temperature, in feet

Equation 11.1

Automatic Valves

Automatic control valves (ACV) are used to control flow rate or to mix or divert water streams. They are classified as two-way or three-way construction, and as modulating or “two position.” To determine flow value use Equation 11.2.

Equation for Valve Flow

$$GPM = C_v\sqrt{P_d}$$

Where:
GPM = *Gallons per Minute*
C_v = *Valve co-efficient*
P_d = *Pressure Drop in PSI*

Equation 11.2

When valves are selected at low authority (low-pressure drop), the ability to control the flow near shutoff is reduced. The balancing valve is used to set the flow at peak flow for the control valve.

Equation for Valve Authority

$$\text{Valve Authority} = \frac{\text{Pressure Drop Across the Control Valve}}{\text{Pressure Drop Across the System}}$$

Equation 11.3

Two-way Valves

Two-way valves control the flow rate by closing off when heat transfer is

not required at the terminal. They may be either single-seated or double seated. Single-seated valves (Figure 11.22) are the most common.

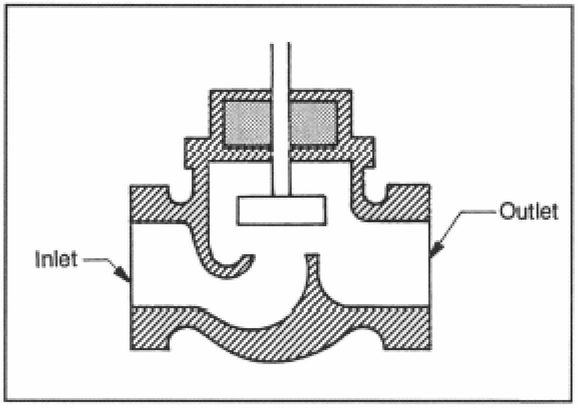


Fig. 11.22 Single-Seated, Two-Way Valve

All automatic control valves must be installed with the direction of flow opposing the closing action of the valve. In other words, the direction of flow must be such that the flow and pressure tend to hold the valve open. If the valve is installed the opposite way, it will produce chattering. Double-seated or balanced valves (Figure 11.23) are generally used when high differential pressures are encountered and tight shutoff is not required. The flow direction through these valves tends to close one port while opening the other port. This design creates a balanced thrust condition that lets the valve close smoothly, without water hammer, regardless of the differential pressure.

Three-way Valves

Three-way valves are generally classified as mixing or diverting. This refers to the internal construction of the valve, not the application. The internal differences are necessary so that the valve seats against flow. Mixing valves have two inlets and one outlet (Figure 11.24). Diverting valves have one inlet and two outlets (Figure 11.25). Either valve may perform a temperature-control action (mixing application) or a flow-control action (bypassing application) depending on its location in the system (Figure 11.26). Diverting valves, however, should not be substituted for mixing valves and vice-versa. Using either design for the wrong application would tend to cause chatter. Another type of three-way valve is used in the supply line to coils in a three-pipe system (Figure 11.27). This valve has two inlets and one outlet. One inlet is supplied with heating water and the other is supplied with chilled water. Depending on the thermostatic controls in the

occupied space, the valve opens to allow either heating water or chilled water into the coil.

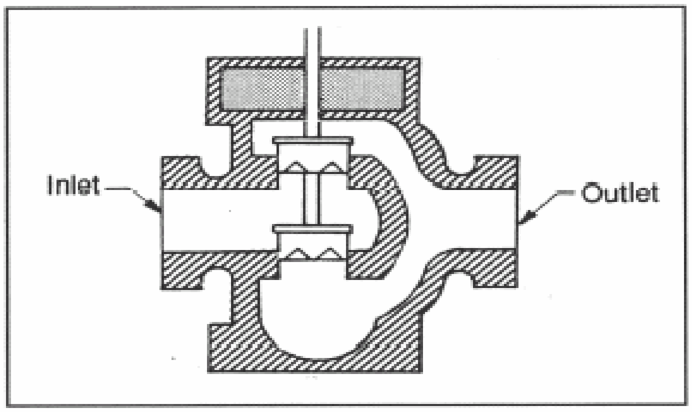


Fig. 11.23 Double-Seated, Two-Way Valve

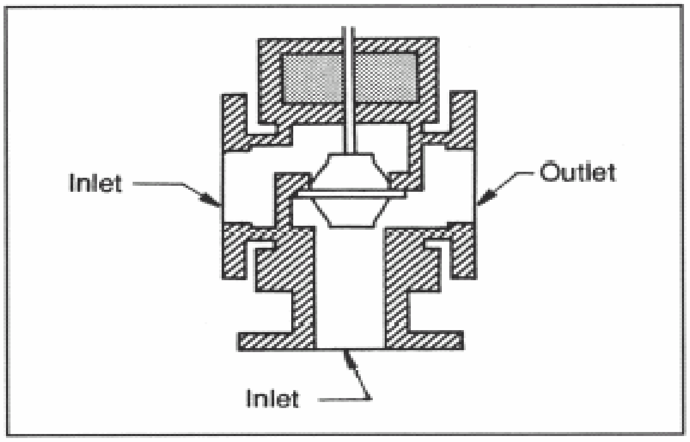


Fig. 11.24 Three-Way Mixing Valve

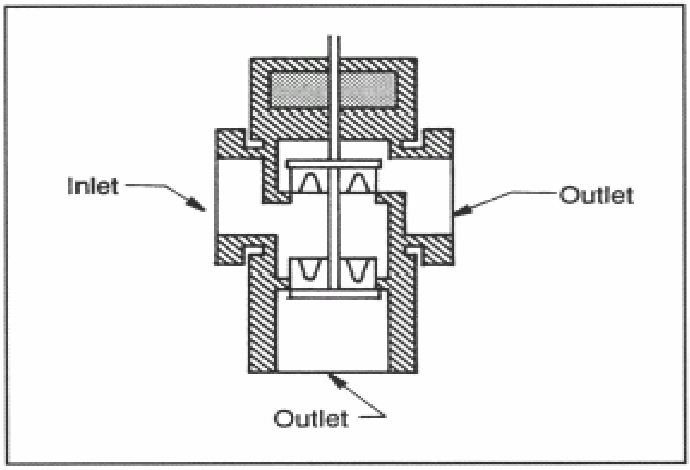


Fig. 11.25 Three-Way Diverting Valve

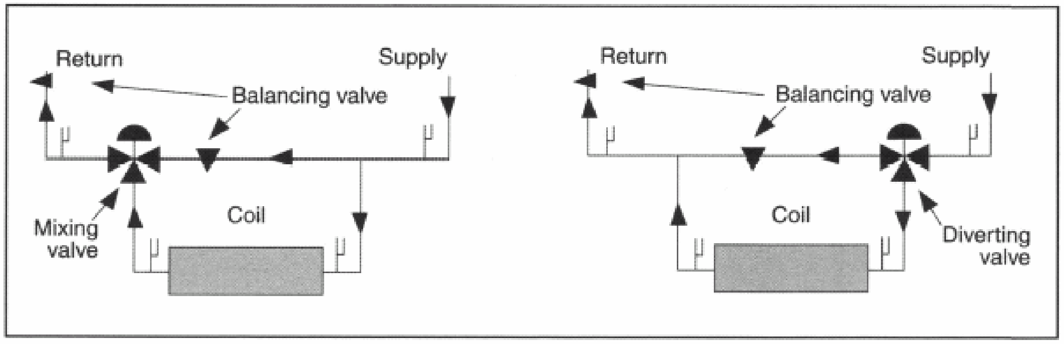


Fig. 11.26 Three-Way Valve Installations Controling Flow Rate

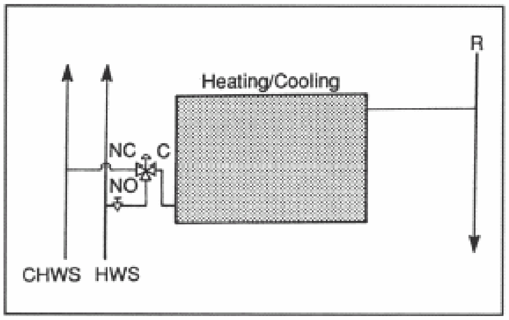


Fig. 11.27 Three-Pipe System with Three-Way Supply Valves

11.5.6 Heat Transfer Devices

11.5.6.1 Heat Exchangers

The term "heat exchanger" describes any heat transfer device as well as a particular category of devices. Heat exchangers are made in various types and sizes, and are designed for a number of fluid combinations. The four basic types of heat exchangers are: shell and tube, U-tube, helical, and plate (Figures 11.28 & 11.29). Some of the combinations of fluids used in heat exchangers are: steam to water (converter), water to steam (generator), refrigerant to water (condenser), water to refrigerant (chiller), and water to water (heat exchanger).

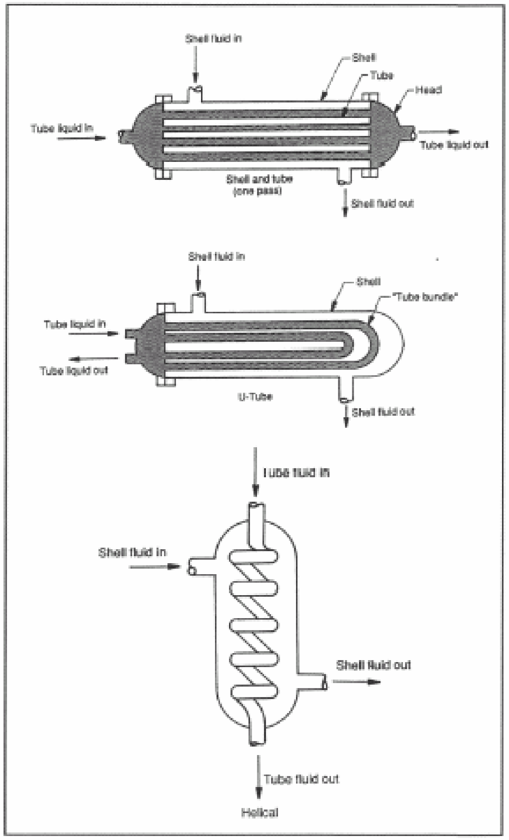


Fig. 11.28 Basic Types of Heat Exchangers

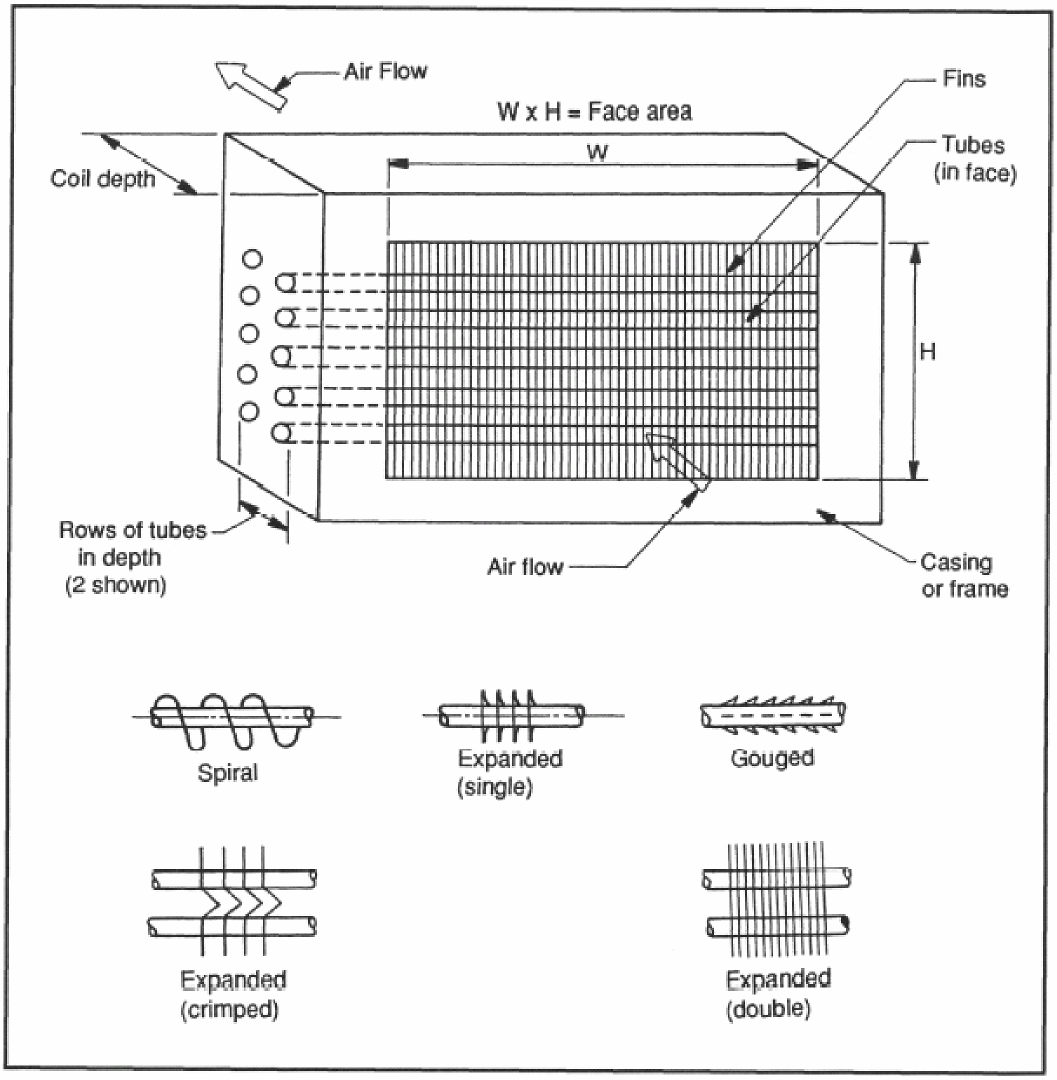


Fig. 11.30 Typical Water Coil

Procedures

Constant Volume Water Balancing Procedure

The entire water system must be operational: all inspections performed as described in Chapter 6, final strainers installed and clean, all controls operational with all valves fully open. Set the system for balancing in the following manner:

1. **Determine the total coil flow and determine how many pumps will operate**
2. **Record the pump (model and serial number, the motor name-plate, the design gpm flow, and the design head**
3. **Verify the impeller size on each pump**

4. Read and record the actual pump head and flow, actual motor amperage and voltage, starter data and motor overload protection devices size and ratings.
5. Set each pump to provide approximately 10% more water flow than designed.
6. Record all flow meter nameplate data. Read all flow meter flows and pressure drops throughout the entire system, (*i.e.* chillers, converters, coils etc.), and establish flow using: Compare heat exchanger flow and flow meter flow with pump flow.
7. Starting with the coil having the greatest flow, adjust the coils to the design flow. With full flow through the coil, measure and set the pressure differential at A-B by adjusting the balancing valve 1. Read the differential pressure at A-C. Set the three-way valve to bypass the coil. Adjust balancing Valve 2 in the bypass to match the pressure differential through the valve and coil (A-C) (Figure 1.45).

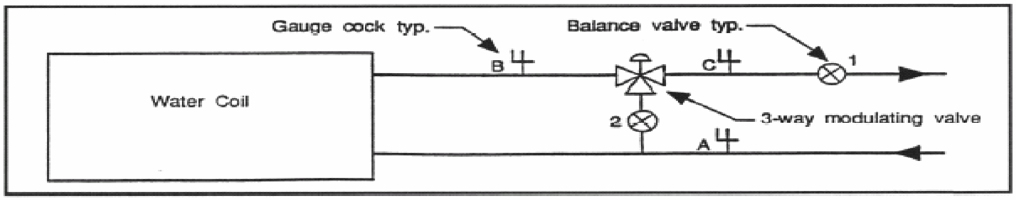


Fig. 11.45 Constant-Flow System with 3-Way Valve

8. Continue to proportion the water to each coil until all coils are balanced within 10% of their design flow with at least one balancing valve in the system remaining 100% open.

9. Re-verify and record the final pump gpm and head, amperages and voltage of each pump, discharge valve position, each heat exchange gpm and pressure drop, each coil gpm and pressure drop, and each flow meter flow. If the discharge valve has to be throttled to obtain design flow, note the valve position and, if possible, the differential across the valve. It should be recommended that the impeller be trimmed to meet final system requirements with the discharge valve full open.

10. Verify the systems control sequence, observing if the correct controller controls the correct valve.

11. Verify the coil or heat exchanger's controller calibration and record the entering and leaving air and water temperatures with the coil air and water set to design (cooling coil db/wb, heating coil db).

12. Record the coil design and actual flows on a coil summary sheet and verify total coil flow is within 10% of total pump flow, total primary heat exchanger flow and total flow meter flow.

13. Show the actual gpm *(Vs)* and head on the pump curve and establish and record the BHP (W).

Variable Volume Water Balancing Procedure

The entire water system must be operational: all inspections performed, final strainers installed and clean, all controls operational with all valves fully open. Set the system for balancing in the following manner:

1. Determine the total coil flow and determine:

2. Diversity-is there enough pumping capacity to provide 100% design water flow to each coil? If there is not enough pumping capacity to provide 100% design water flow to each coil, close the valves to the coils closest to the pump until the pump can provide 100% flow to the open coils. If there are three-way valves or a bypass valve in the system, observe that it is closed prior to balancing.

3. Record the pump (s) model and serial number, the motor nameplate, the design gpm flow, and the design head. Record the variable frequency drive (VFD) data if installed.

4. If a VFD is installed, set the VFD bypass for 100%. Verify the impeller size on each pump.

5. Read and record the actual pump head and flow with actual motor amperage and voltage. Record starter data and motor over-load protection devices size and rating.

6. Set each pump to provide approximately 10% more water flow than designed.

7. Record all flow meter nameplate data. Read all flow meter flows and pressure drops throughout the entire system, *(ie.* chillers, converters, coils, etc.), and establish flow using Equation 11.4.

8. Starting with the coil having the greatest flow, adjust the coil to the design flow.

9. Continue to proportion the water to each coil until all coils are balanced within ± 10% of design flow with the balancing valve of at least one coil remaining 100% open. If the system has diversity, close an equal amount of valves nearest to the pump and open the initially closed valves. Continue to proportion the water to each coil (starting with the coil closest to the pump) until all the coils are balanced within ± 10% of the design flow.

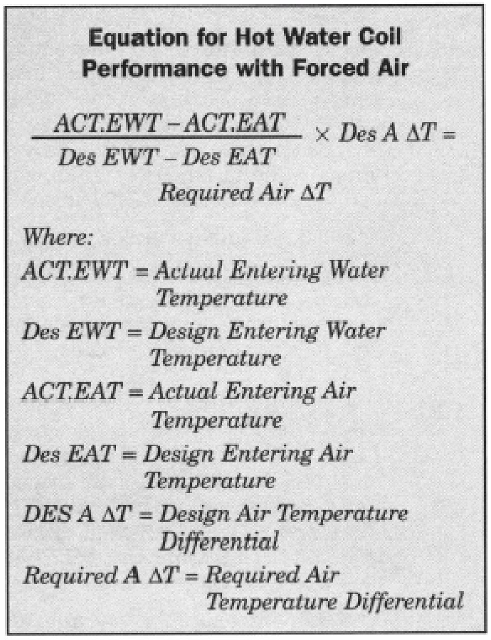
10. Re-verify and record the final pump gpm and head, amperage and voltage of each pump (discharge valve position if the system does not have a variable frequency drive). When the system has a VFD and the VFD is at 100% with the discharge valve throttled to obtain design flow, it is recommended that the impeller be trimmed or the VFD limited with the discharge valve fully open.

11. Record the system's pressure differential at the pressure transmitter with the system at full flow. Adjust the coil closest to the pump closed and verify the pressure transmitter setting. Continue with the next coil until enough coils are closed to equal the minimum speed of the pump or the minimum flow through the primary heat exchangers. Verify that the closed valves closest to the pump remain closed and are not lifting off their seats. Note that most bypasses in two-way valve systems act as a relief valve and the pump will ride its pressure curve until bypass opens. Record the minimum flow across the heat exchanger.

12. Verify the coil or heat exchangers controller calibration, and record the entering and leaving water temperatures.

13. Record the coils design and actual flow (coils open for diversity) on a coil summary sheet and verify total coil flow equals within 10% of total pump flow, total primary heat exchanger flow, and total flow meter flow.

14. Show the actual gpm and head on the pump curve and establish and record the brake horsepower. (Note: with VFD, discharge valve and VFD must be at l00%.)



Equation for Hot Water Coil Performance with Forced Air

$$\frac{ACT.EWT - ACT.EAT}{Des\ EWT - Des\ EAT} \times Des\ A\ \Delta T = Required\ Air\ \Delta T$$

Where:

ACT.EWT = Actual Entering Water Temperature

Des EWT = Design Entering Water Temperature

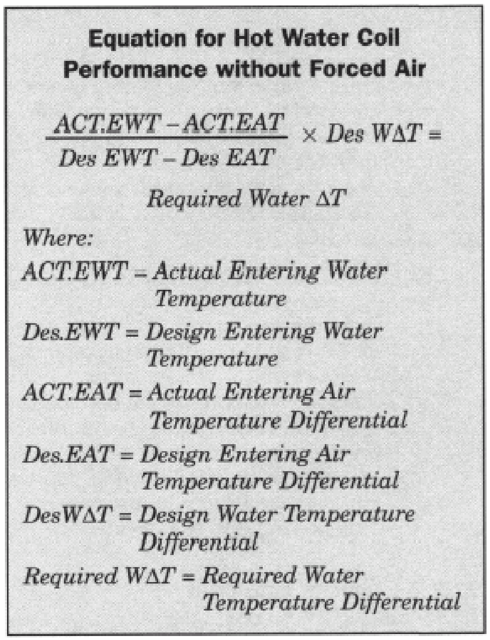
ACT.EAT = Actual Entering Air Temperature

Des EAT = Design Entering Air Temperature

DES A ΔT = Design Air Temperature Differential

Required A ΔT = Required Air Temperature Differential

Equation 11.5



Equation for Hot Water Coil Performance without Forced Air

$$\frac{ACT.EWT - ACT.EAT}{Des\ EWT - Des\ EAT} \times Des\ W\Delta T = Required\ Water\ \Delta T$$

Where:

ACT.EWT = Actual Entering Water Temperature

Des.EWT = Design Entering Water Temperature

ACT.EAT = Actual Entering Air Temperature Differential

Des.EAT = Design Entering Air Temperature Differential

DesWΔT = Design Water Temperature Differential

Required WΔT = Required Water Temperature Differential

Equation 11.6

Expansion or Compression Tanks

Water expands, when heated, in direct proportion to its temperature. In hydronic systems, an allowance must be made for this expansion, otherwise a pipe or piece of equipment could burst.

Open systems have expansion tanks to safely accommodate the increased volume of heated water. These tanks, open to the atmosphere, are installed about 3 feet above the highest point in the system. As the water volume increases, the water level rises in the tank (Figure 11.10).

A better solution is to use a closed expansion or compression tank containing a gas, such as air. When water expands into such closed tanks, it compresses the gas (Figure 11.11).

Once the system is filled with water (up to about two-thirds of the tank's diameter) and the water is heated or cooled to operating temperature, the expansion tank will accommodate the fluctuations in water volume and will control the pressure change in the system.

The point at which the expansion tank connects to the system is called "the point of no pressure change." To avoid pumping problems, expansion

tanks are installed on the suction side of the pump.

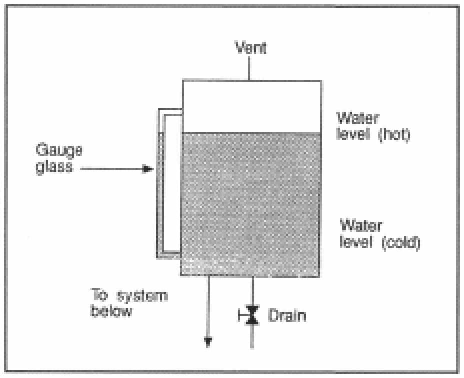


Fig. 11.10 Open Expansion Tank

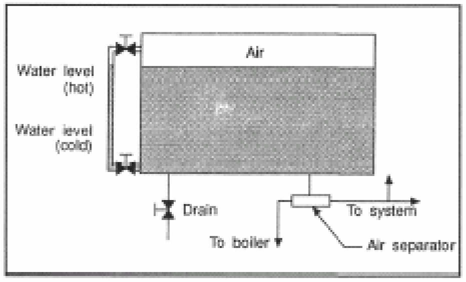


Fig. 11.11 Closed Expansion Tank

Pressure-Relief Valves

Pressure-relief valves (Figure 1 1.12) are safety devices for protecting against system damage and personal injury. The pressure-relief valve opens on a pre-set value so that the system pressure cannot exceed this amount.

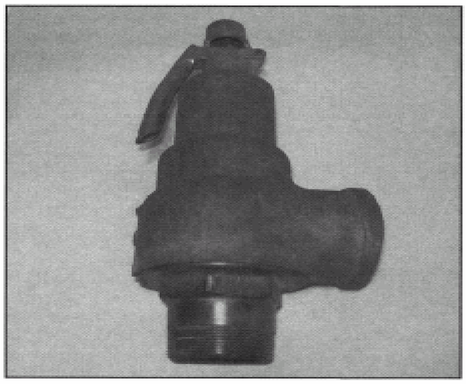


Fig. 11.12 Pressure-Relief Valve

Air Control

Air is always present in hydronic heating and cooling systems. It gets into the system when it is initially filled through make-up water and when the system is drained for repairs and maintenance. Air is released from the water upon start-up, particularly when the water is heated. Much of this air will find its way to the expansion tank through an air separator. However, some of the air will find its way to other high points in the system.

This entrapped air must be vented or it will block flow through the terminals. In fact, air in a system can restrict flow to the point of causing total shut-off. Air can also result in noise in the system, pump cavitation, and seal failure. Air control devices such as air vents and air separators are designed to remove air from the system.

Air Vents

Air vents should be located at each point in the piping at which the water

changes its direction to a downward flow, and at all high points in the system. Air vents can be either automatic or manually operated. A manual air vent is simply a valve that is opened to allow the entrained air to escape.

Because automatic air vents occasionally malfunction, manual valves should be provided at each vent to permit servicing without shutting down and draining the system.

Air Separators

A boiler dip-tube air separator is a tube in the top or top side of the boiler. When water is heated, air is released and collects at a high point in the boiler. The dip tube allows this collected air to rise into the expansion tank. The air-separating tank is an in-line, low-velocity, dip-tube air separator.

The centrifugal air separator works on the action of centrifugal force and

low-velocity separation. Water circulation through the air separator creates a vortex or whirlpool in the center of the tank where entrained air collects and rises into the expansion tank.

These devices are used at the point of lowest pressure and highest temperature in the system, which suggests for hot water systems, it is located between the heat exchanger and the pump inlet, and for chilled water systems it is located between the system and pump inlets (the pump will pump through the heat exchanger).

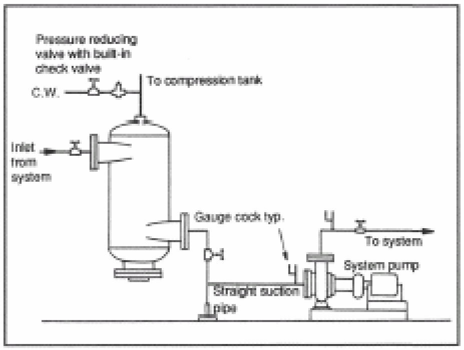


Fig. 11.13 Air Separator

Strainers

The purpose of strainers (Figure 11.14) is to catch sediment or other foreign material in the water. A strainer contains a mesh screen formed into a sleeve that fits inside the strainer body. This sleeve must be removed and cleaned. A strainer with a dirty sleeve or a sleeve with a screen that is too fine means that there will be excessive pressure drop across the strainer and lower water flow. Strainers placed in the suction side of the pump must be properly sized and kept clean to avoid cavitation. Strainers should have pressure taps installed at the inlet and outlet.

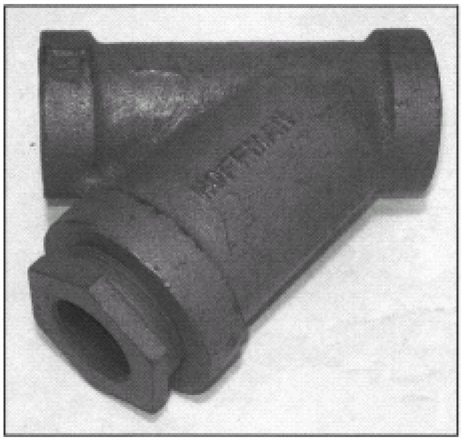


Fig. 11.14 Strainers

Strainers should only be used where necessary to protect the components in the system. For example, in the case of cooling towers, strainers in the tower basin may provide adequate protection. Individual fine mesh strainers may need to be installed before automatic control valves or spray nozzles, which operate with small clearances and require protection from materials that might pass through the pump strainer.

Valves

Valves are used in hydronic systems to:

- **control water flow**
- **isolate part, or all of the system for servicing**

- **allow draining of all or part of the system**
- **restrict operation to specific subsystems**
- **allow quick filling of the system by bypassing the pressure-reducing valve**

Manual Valves

Manual valves fall into two major categories: manual service valves and check valves. Manual service valves regulate flow rate. These include gate, globe, plug, ball, and butterfly valves. Check valves limit the direction of flow. Another manual valve, the combination valve, both regulates flow and limits direction of flow.

Gate Valves

Gate valves (Figure 11.15) regulate flow only in that they are either fully open or fully closed. They should not be used for throttling purposes because they produce a high-velocity water stream when partly closed, resulting in erosion of the valve seat. This is called wiredrawing, and will result in leakage when the valve is fully closed. Gate valves should be used for tight shutoff, and for servicing equipment. Gate valves allow a straight-through flow pattern, resulting in low-pressure drop.

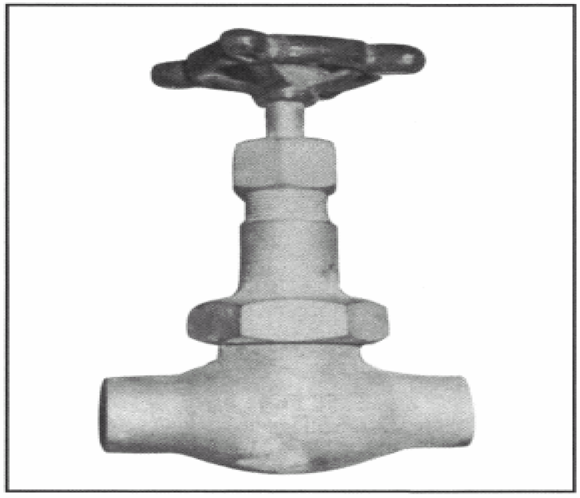


Fig. 11.15 Gate Valve

Globe Valves

Globe valves (Figure 11.16) can be used in partially open positions and therefore can be used for throttling fluid flow. However, globe valves have high-pressure drops, even when fully open. Therefore, globe valves can unnecessarily increase the required pump head, and should not be used for balancing. They can, however, be used in water make-up lines. Note: If globe valves are ever tampered with, or used to shut off a system, it is difficult to reset the system.

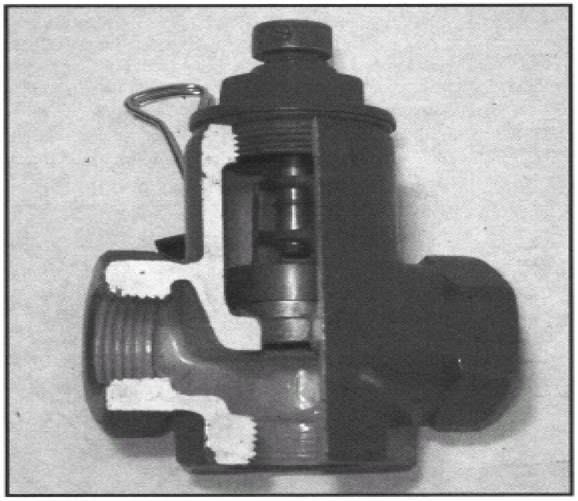


Fig. 11.16 Globe Valve

Plug Valves

Plug valves (Figure 11.17) are used for balancing water flow. Some plug valves have adjustable memory stops that are set during the final balance so that if the valve is closed, it can later be reopened to the original setting. Plug valves have a low-pressure drop and therefore add little to the pumping head. They can also be used for tight shutoff. Plug valves have good throttling characteristics (Figure 11.18). Throttling characteristics refer to the relationship of the valve disc to the percentage of flow.

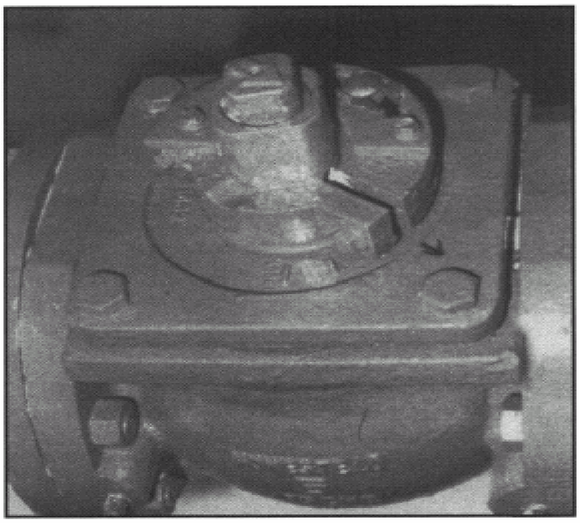


Fig. 11.17 Plug Valve

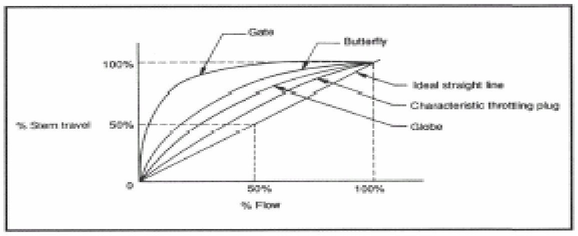


Fig. 11.18 Valve Throttling Characteristics

Valves have linear throttling characteristics when the disc-open percentage is the same as the flow percentage. For example, a valve would be said to have linear or straight-line throttling characteristic if closing it 50% reduced

flow by 50%. For adjusting purposes, the closer the valves are to linear flow the better.

Ball Valves

Ball valves (Figure 11.19) are similar to plug valves, and are often used for water balancing. They have low-pressure drops and good flow characteristics. Set the memory stop, if available.

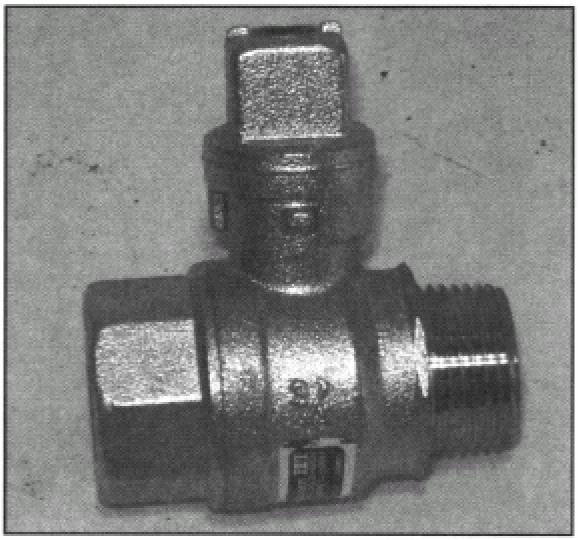


Figure 11.19 Ball Valve

Butterfly Valves

Butterfly valves (Figure 11.20) consist of a heavy ring enclosing a disc that

rotates on an axis and, in principle, is similar to a round single-blade damper. Butterfly valves have low-pressure drops and are used as throttling valves, but do not have good throttling characteristics.

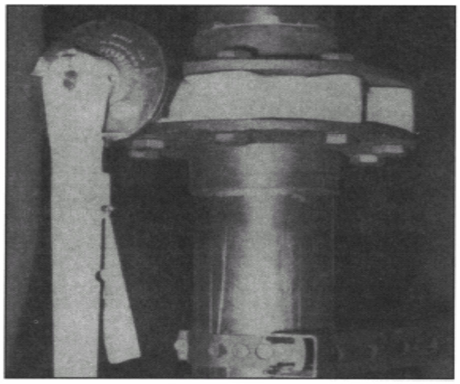


Fig. 11.20 Butterfly Valve

Check Valves

Check valves allow water to flow in one direction and stops its flow in the other direction. There are two types of check valves: swing and spring loaded. Check valves should be used at the discharge of the pump. With swing check valves, when the system is off the weight of the gate and gravity closes the valve. The valve is opened when the system is turned on,

in addition, pressure from the fluid flowing in the proper direction causes the gate to swing open. Spring-loaded check valves have springs that keep the valve closed. Water pressure from the proper direction against the spring opens the valve.

Combination Valves

Combination valves (Figure 11.21) are sometimes referred to as a multipurpose or triple-duty valves. They are a combination of a check valve, a calibrated balancing valve, and a shutoff valve. They are made in a straight or angle pattern.

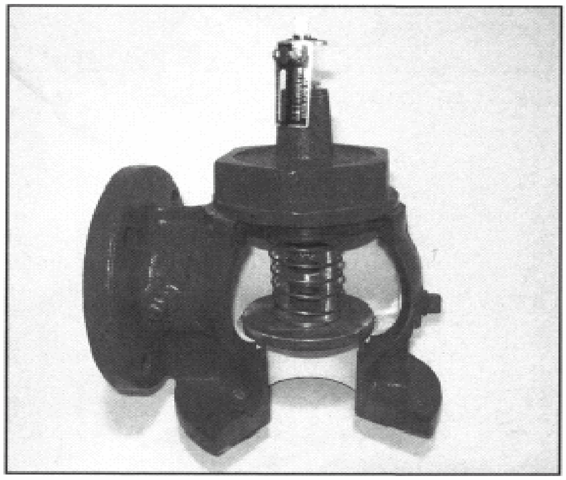


Fig. 11.21 Combination Valve

Combination valves act as a check valve, preventing back flow when the

pump is off, and can be closed for tight shutoff, or can be partially closed for balancing. These valves have pressure taps for connecting flow gauges and for reading pressure drop across the valve. A calibration chart is provided with the valve for conversion of pressure drop to gpm. The valves also have memory stops.

Series loop

The series loop is a simple and inexpensive piping arrangement generally limited to residential and small commercial building applications. In series loops (Figure 11.32), supply water is pumped through each terminal in series and then returned to the boiler. Since all the water is circulated in succession through each terminal, there are several disadvantages to the series loop arrangement. First, making repairs at any one terminal requires shutting the whole system down. Second, it is not possible to provide capacity controls on individual terminals, because valuing down one terminal would reduce flow

to all the terminals down the line. Instead, space heating must be controlled by dampening the airflow.

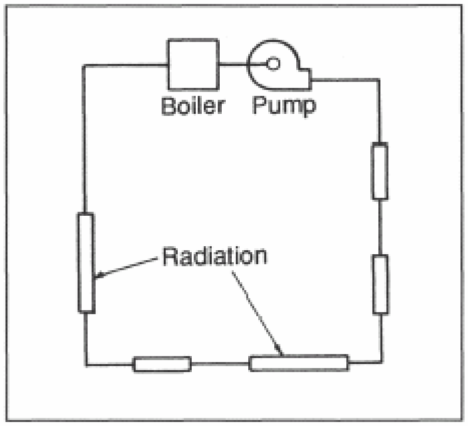


Fig. 11.32 Series Loop System–One Circuit

These disadvantages can be partially remedied by designing the piping with two or more circuits, and installing balancing valves in each circuit (Figure 11.33). This type of arrangement is called a split series loop.

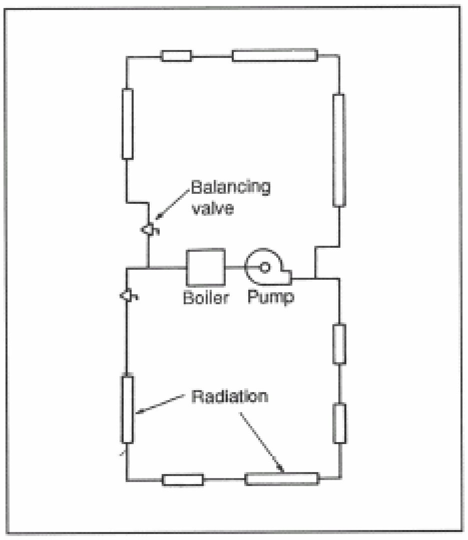


Fig. 11.33 A Spilt Series Loop System–Two Circuits

The series loop circuit length and pipe size are also very important in the design because they directly influence the water flow rate, temperature, and pressure drop. For instance, as the heating supply water flows through the terminals, its temperature drops continuously as it gives up heat in each terminal. If there are too many terminals in series, the water temperature in the last terminal may be too cool to be effective.

One Pipe Main

One-pipe mains are used in houses and small commercial and industrial heating applications where individual space control is needed. This arrangement uses a single-loop main but differs from series loop arrangements in that each terminal is connected by a supply and return branch pipe to the main (Figure 11.34). Because the terminal has a

higher-pressure drop than the main, the water circulating in the main will tend to flow through the straight run of the tee fittings, starving the terminal.

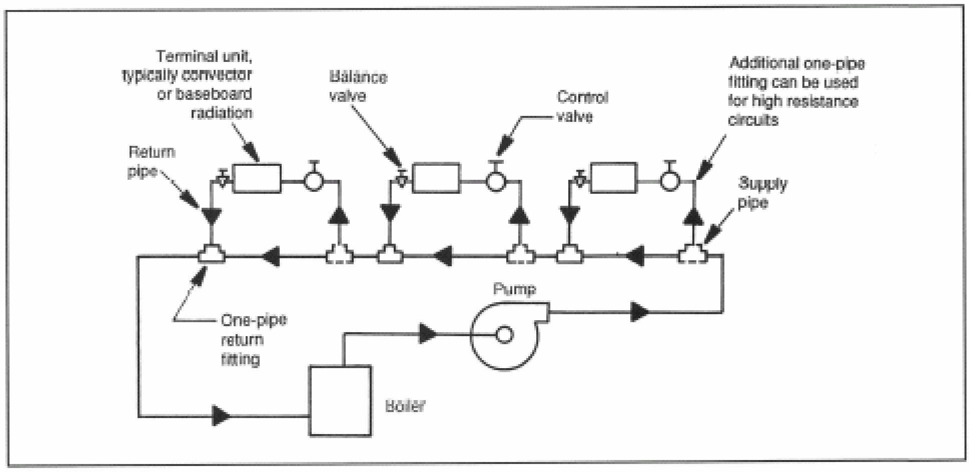


Fig. 11.34 A One-Pipe Main System

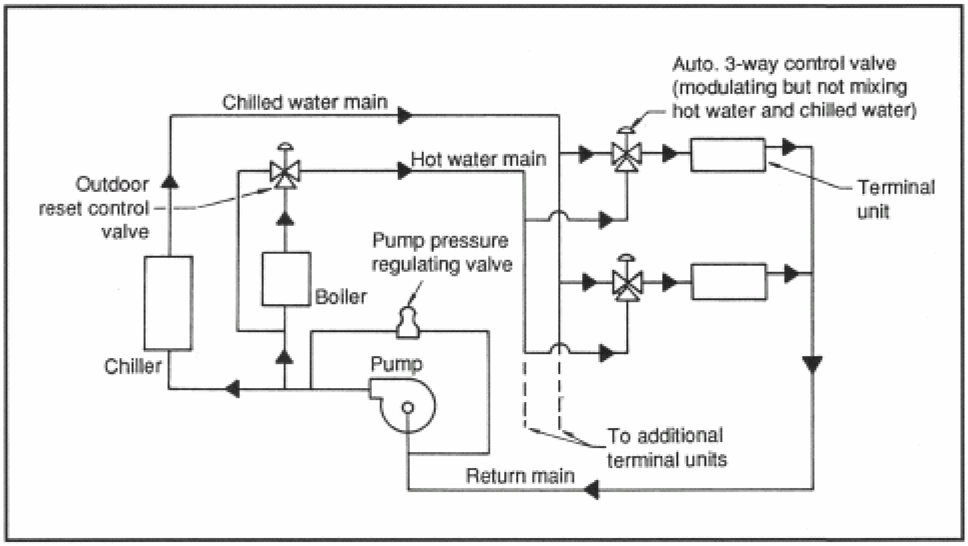


Fig. 11.38 Elementary Three-Pipe Hydronic System

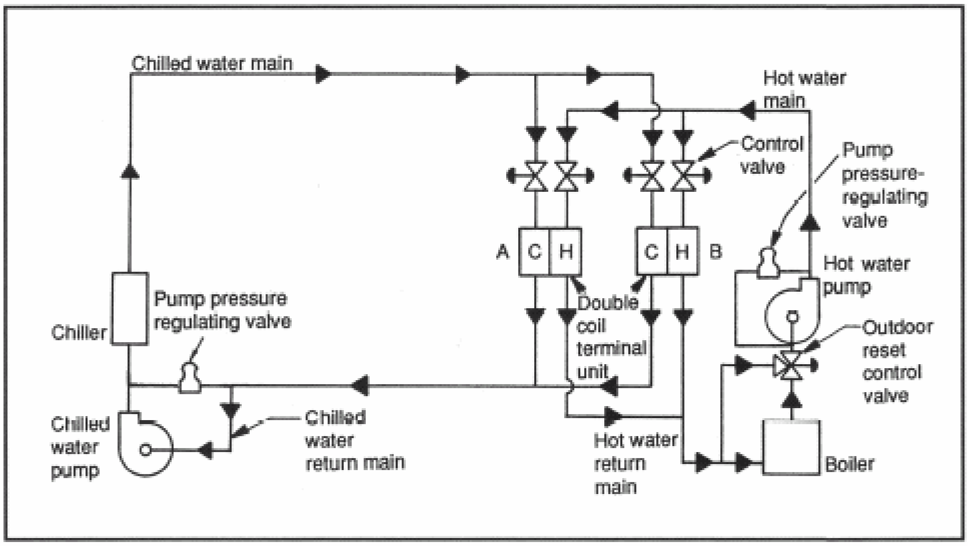


Fig. 11.39 Elementary Four-Pipe Hydronic System

Pump Circuits

Primary-Secondary

Primary-secondary pump circuits reduce pumping horsepower requirements while increasing system control quality. Primary-secondary

circuits are designed so that when two circuits are interconnected, flow in one circuit will not cause flow in the other if the pressure drop is eliminated in the pipe common to both circuits.

Figure 11.40 shows a primary-secondary pump circuit. The common pipe (A-H), which may vary in length, has a negligible pressure drop and ensures the isolation of the secondary circuit from the primary circuit. Therefore, in order to overcome the pressure loss in the secondary circuit (A-B-C-D-E-F-G-H) and provide design flow, a secondary pump (Figure 11.41) must be installed.

The primary pump and the secondary pump have no effect on each other. The function of the primary pump is simply to circulate water around the primary circuit. The secondary pump then supplies the terminals. The secondary flow may be (1) less than, (2) equal to, or (3) greater than the primary flow (Figures 11.42, 11.43, and 11.44).

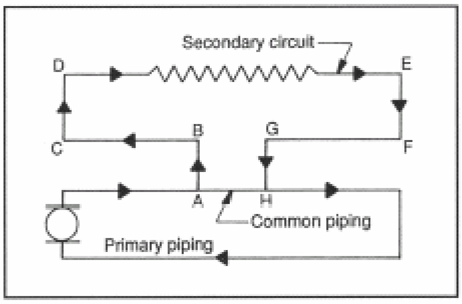


Fig. 11.40 Primary-Secondary Circuit

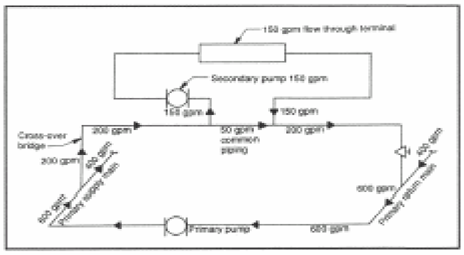


Fig. 11.42 Hydronic System with Secondary Flow Less Than Primary Flow

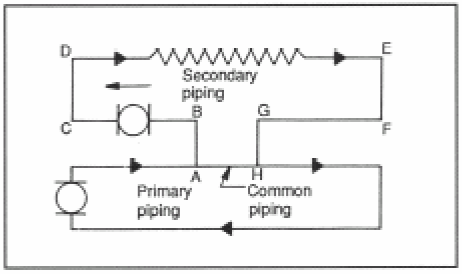


Fig. 11.41 Secondary Circuit Pump Added

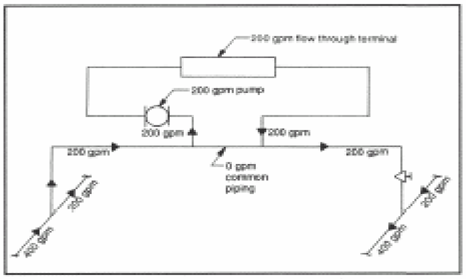


Fig. 11.43 Hydronic System with Secondary Flow Equal To Primary Flow

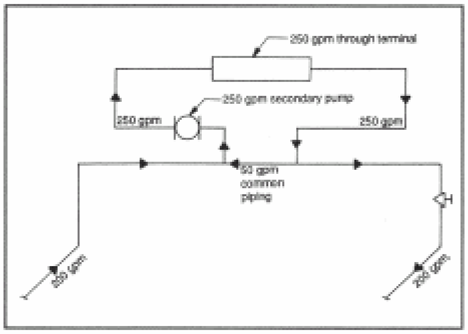


Fig. 11.44 Hydronic System with Secondary Flow Greater Than Primary Flow

Basic Testing

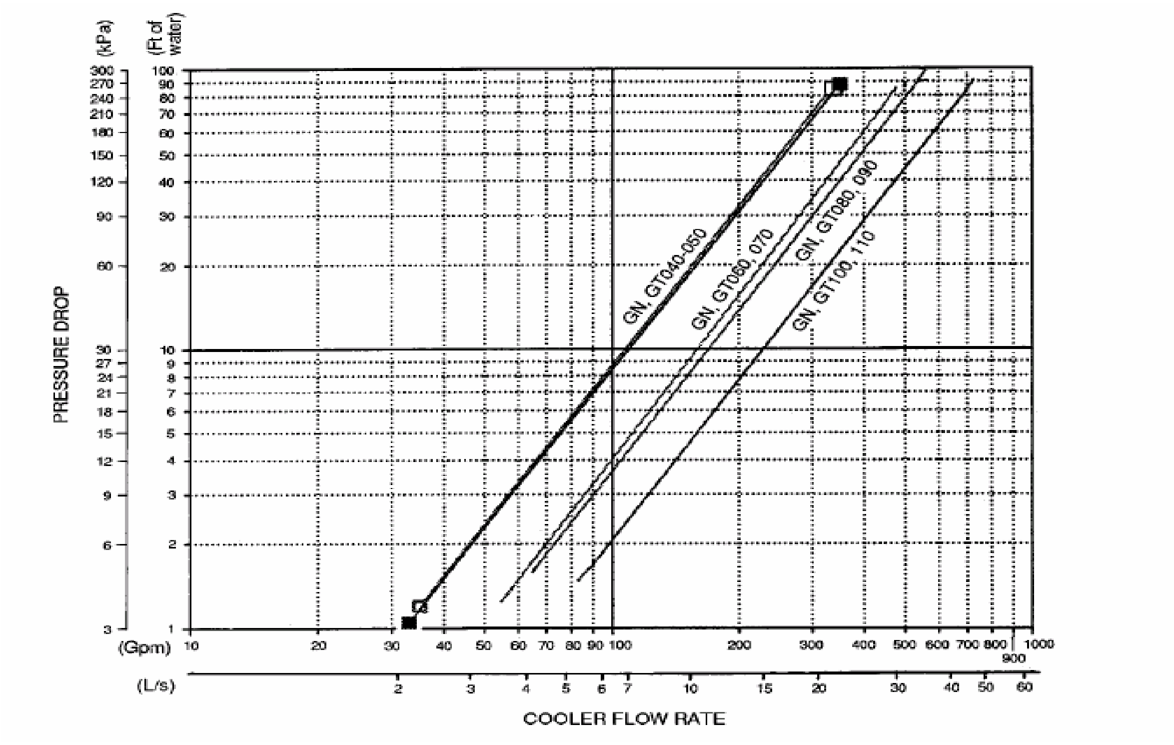
Water Cooled Chiller

The water-cooled chiller consists of one or more of the following: compressors, liquid coolers and water-cooled condensers. The basic test for a water-cooled chiller is shown on the NTT "Chillers" report form. The basic procedure is to measure and/or report the

following:

- **Chiller manufacturer nameplate data**
- **Rated capacity in tons**
- **Model and serial number**

- **Evaporator:**
  - **the design and actual entering water temperature**
  - **the design and actual leaving water temperature**
  - **the design and actual pressure drop in feet (kpa)**
  - **the design and actual GPM *(Vs)***



Example 10: Find the GPM of a GN, 070-chiller, entering water pressure of 42.45 psig and a leaving pressure of 36.82 psig.

Solution

- **Δ Pressure drop X 2.31 = foot head**
- **Use chart for GN 070 chiller for GPM**
- **42.45 pressure – 36.82 pressure = 5.63 Δ Pressure drop**
- **5.63 Δ Pressure drop X 2.31 = 13.00 foot head**
- **using chart for GN 070 chiller with a 13.0 foot head gave a 180 gpm**

Example 11: **Find the tons of the chiller in example 10 with an entering water temperature of 56.8 °F and a leaving water temperature of 46.5 °F of a GN 070 chiller.**

Solution 1

- **Tons = GPM X 60 X 8.337 X Δ Temperature of the water / 12,000 btuh per ton**
- **180 gallons X 60 minutes X 8.337 X 10.3 ΔT = 927,407.8 btuh**
- **927,407.8 btuh / 12,000 btuh per ton = 77.28 tons**

Solution 2

- **Tons = GPM X Δ Temperature of the water X specific heat X specific gravity / 24**
- **180 gallons X 10.3 ΔT X 1 SH X 1 SG = 1854**
- **1854 / 24 = 77.25 tons**

Condenser

- **the design and actual entering water temperature**
- **the design and actual leaving water temperature**
- **the design and actual pressure drop in feet (kpa)**
- **the design and actual GPM *(Vs)***

- **Electrical (electrical measurements will only be taken on equipment rated under 600 volts)**
- **the design and actual voltage**
- **the design and actual amperages**

COOLING CAPACITIES — 60 Hz, ENGLISH

LCWT (F)	UNIT SIZE 30GXN,R	CONDENSER ENTERING AIR TEMPERATURE (F)														
		85			95			105			115			125		
		Cap.	Input kW	Cooler Flow Rate (Gpm)	Cap.	Input kW	Cooler Flow Rate (Gpm)	Cap.	Input kW	Cooler Flow Rate (Gpm)	Cap.	Input kW	Cooler Flow Rate (Gpm)	Cap.	Input kW	Cooler Flow Rate (Gpm)
40	080	75.8	84.1	181.3	70.0	88.7	167.6	63.7	93.8	152.4	57.5	100.6	137.7	52.0	108.3	124.4
	090	83.2	95.1	199.2	76.5	100.2	183.2	69.8	106.4	167.0	63.2	114.3	151.4	57.3	122.9	137.1
	106	97.1	108.6	232.4	89.7	114.2	214.7	81.8	120.8	195.9	74.2	128.8	177.7	66.9	138.0	160.3
	114	105.5	119.6	252.5	97.4	126.1	233.3	88.9	133.4	212.7	80.5	143.2	192.6	72.5	153.8	173.5
	125	114.5	130.3	274.0	108.3	141.7	259.2	102.9	154.1	246.3	97.2	170.6	232.7	70.3	153.0	168.4
	135	125.3	141.2	299.9	118.2	153.8	282.9	112.3	167.0	268.9	106.2	185.2	254.1	93.2	205.3	223.0
	150	137.0	155.4	327.9	130.4	168.9	312.1	123.9	183.7	296.7	117.5	202.0	281.4	110.7	224.0	265.1
	160	149.5	170.8	357.8	141.5	185.4	338.7	135.0	201.3	323.1	127.9	222.7	306.2	115.2	247.5	275.8
	174	163.1	186.2	390.4	156.6	201.2	374.8	148.6	219.3	355.7	141.0	239.7	337.5	88.4	195.9	211.7
	204	188.7	217.2	451.8	178.6	236.2	427.6	170.0	255.9	407.0	161.1	282.9	385.6	148.2	312.7	354.8
	225	205.8	237.2	492.7	195.3	256.9	467.5	186.2	279.9	445.7	176.2	308.6	421.7	111.5	224.5	267.0
	249	229.7	266.2	549.8	218.9	288.4	524.0	208.4	314.0	499.0	197.5	345.4	472.8	142.6	308.9	341.3
	264	241.8	282.7	578.9	229.7	305.8	550.0	219.3	333.0	524.9	207.7	366.4	497.3	146.0	333.9	349.5
	281	259.5	291.9	621.3	248.5	317.3	594.9	235.7	345.1	564.2	223.9	379.0	536.0	211.2	419.6	505.6
	301	281.2	317.8	673.1	267.1	345.4	639.4	254.3	374.5	608.8	241.3	413.3	577.7	227.4	458.3	544.5
	325	304.3	352.9	728.5	288.5	382.9	690.7	275.4	416.8	659.2	261.2	460.5	625.4	229.2	512.3	548.8
	350	327.9	395.3	784.9	313.0	427.2	749.4	298.5	468.1	714.5	282.9	515.1	677.2	162.6	345.0	389.3
	370	350.1	396.2	838.6	334.4	429.1	801.0	317.8	467.7	761.4	301.3	514.6	721.9	223.3	449.4	535.0
	390	372.7	427.0	892.8	355.0	463.0	850.2	337.8	504.2	809.3	320.2	553.3	767.0	238.8	539.4	572.1
	415	399.5	457.6	957.0	380.4	495.6	911.4	362.3	539.4	868.2	343.8	592.7	823.5	260.2	581.1	623.2
	450	422.6	479.7	1012.5	402.0	519.5	963.0	382.9	565.4	917.4	362.7	622.8	869.0	230.9	452.6	553.2
42	080	78.8	85.7	188.7	73.0	90.4	174.8	66.5	95.4	159.3	59.9	101.9	143.6	54.2	109.8	129.8
	090	86.4	97.2	206.9	79.7	102.3	191.0	72.7	108.2	174.2	65.8	115.9	157.6	59.6	124.6	142.8
	106	101.0	110.8	241.9	93.5	116.4	223.9	85.6	122.8	205.1	77.6	130.5	185.9	69.9	139.8	167.4
	114	109.8	121.8	263.1	101.6	128.5	243.5	92.9	135.8	222.5	84.1	145.1	201.4	75.8	155.7	181.6
	125	118.4	131.9	283.6	112.0	143.7	268.4	106.3	156.0	254.6	100.5	172.1	240.7	72.1	153.9	172.8
	135	129.3	143.1	309.9	122.5	155.9	293.6	116.2	168.9	278.4	109.9	186.6	263.3	93.2	205.4	223.2
	150	141.3	157.4	338.6	135.0	171.1	323.5	128.1	185.7	307.0	121.6	203.6	291.3	112.3	225.1	269.1
	160	154.5	172.9	370.2	146.4	187.9	350.6	139.4	203.8	334.0	132.3	224.7	317.0	115.3	247.3	276.2
	174	168.5	188.5	403.6	161.8	204.0	387.5	153.8	221.5	368.5	145.9	242.3	349.6	91.9	197.5	220.1
	204	194.7	220.2	466.4	185.2	239.4	443.6	175.8	259.7	421.1	166.6	285.7	399.1	150.1	313.2	359.6
	225	213.0	240.5	510.4	202.0	260.6	483.9	192.4	283.0	461.0	182.2	311.4	436.6	115.6	226.5	276.9
	249	237.5	269.6	569.1	226.3	292.2	542.2	215.5	317.9	516.2	204.2	348.9	489.3	147.6	310.9	353.5
	264	250.4	286.2	599.9	237.3	309.9	568.4	226.7	336.5	543.1	214.9	370.1	514.7	149.1	335.1	357.2
	281	267.9	295.5	641.8	257.8	320.7	617.6	243.8	349.3	584.0	231.7	382.1	555.2	218.7	423.0	523.9
	301	290.1	322.0	695.0	276.9	349.6	663.4	262.9	379.6	629.8	249.6	417.1	597.9	235.2	461.0	563.5
	325	315.1	357.1	754.8	298.3	387.9	714.7	284.6	421.1	681.8	270.1	464.8	647.0	229.3	512.2	549.4
	350	339.5	400.5	813.2	323.3	433.2	774.6	308.5	472.8	739.0	292.3	520.1	700.3	168.5	347.3	403.5
	370	362.0	401.1	867.8	345.7	434.9	828.5	328.5	473.4	787.3	311.4	518.9	746.5	229.2	451.9	549.3
	390	385.4	432.2	923.8	367.1	468.6	880.1	349.3	509.6	837.1	330.9	558.9	793.3	242.0	540.9	580.2
	415	412.9	463.6	989.7	393.3	501.8	942.8	374.5	545.3	897.8	355.3	598.1	851.3	263.5	582.6	631.8
	450	437.0	486.1	1047.6	415.5	526.9	996.1	395.5	572.4	948.1	374.8	628.6	898.5	239.3	456.6	573.8

44	**080**	81.8	87.5	196.1	75.8	92.3	181.8	69.4	97.3	166.3	62.5	103.5	149.9	56.4	111.1	135.3
	090	89.7	99.3	214.9	83.0	104.4	199.0	75.8	110.2	181.7	68.6	117.6	164.5	62.0	126.2	148.7
	106	105.0	113.1	251.6	97.5	118.7	233.8	89.4	124.8	214.2	81.1	132.7	194.5	73.1	141.6	175.2
	114	114.3	124.3	274.0	106.0	131.0	254.1	97.1	138.1	232.7	87.8	147.1	210.5	79.2	157.6	189.9
	125	122.2	133.9	293.0	116.0	145.8	278.1	109.8	158.0	263.2	103.8	173.6	248.9	73.4	155.0	176.0
	135	133.4	144.9	319.7	127.0	157.9	304.5	120.1	171.3	288.0	113.6	188.1	272.2	93.2	205.3	223.5
	150	145.8	159.4	349.6	139.9	173.0	335.4	132.4	188.0	317.3	125.6	205.8	301.2	112.9	225.6	270.6
	160	159.4	175.6	382.1	151.6	190.4	363.5	144.0	206.3	345.2	136.6	227.0	327.5	115.3	247.4	276.3
	174	174.1	190.9	417.3	167.0	206.7	400.3	159.3	224.1	381.9	150.9	245.2	361.7	95.5	199.4	228.9
	204	200.8	223.4	481.5	192.0	242.5	460.2	181.7	263.3	435.6	172.3	288.1	413.0	150.1	313.5	359.9
	225	220.0	244.2	527.4	209.0	264.4	501.0	198.8	286.8	476.6	188.3	315.1	451.4	119.6	228.3	286.8
	249	245.6	272.9	588.8	234.1	296.3	561.1	222.6	321.4	533.7	211.1	352.5	506.0	152.7	313.7	366.0
	264	259.0	290.1	621.0	245.2	314.4	587.9	234.1	340.2	561.2	222.0	374.3	532.2	151.3	337.4	362.7
	281	276.6	298.8	663.0	267.2	324.2	640.6	252.1	353.7	604.3	239.7	385.3	574.7	226.2	426.8	542.3
	301	299.3	326.2	717.4	286.9	353.7	687.9	271.7	384.6	651.2	258.1	420.5	618.7	239.2	462.8	573.5
	325	325.2	362.0	779.6	309.0	393.6	740.6	294.0	426.5	704.8	278.9	469.3	668.7	229.4	512.3	549.9
	350	351.3	405.7	842.1	333.8	439.3	800.1	318.6	477.6	763.8	301.9	525.7	723.8	174.3	350.2	417.8
	370	373.5	406.4	895.9	357.3	440.8	856.8	339.5	478.9	814.1	321.7	524.0	771.7	235.2	454.5	564.1
	390	398.2	437.6	955.2	379.3	474.5	909.7	360.7	515.5	865.3	341.9	564.6	820.1	244.7	542.9	585.0
	415	426.6	469.3	1023.3	406.2	509.1	974.3	386.5	551.8	927.2	366.8	604.8	879.8	266.9	584.2	640.0
	450	451.0	492.8	1081.7	429.7	534.5	1030.5	408.4	579.7	979.7	387.2	634.9	928.6	247.8	461.3	594.3
45	**080**	83.3	88.5	199.9	77.3	93.3	185.4	70.8	98.3	169.7	63.9	104.2	153.3	57.6	111.8	138.1
	090	91.3	100.3	219.0	84.7	105.5	203.1	77.3	111.3	185.4	70.0	118.5	167.9	62.6	126.5	150.0
	106	107.2	114.2	257.0	99.6	119.8	238.9	91.5	125.9	219.3	82.8	133.7	198.5	74.7	142.5	179.1
	114	116.6	125.6	279.7	108.2	132.2	259.5	99.3	139.1	238.1	89.7	148.2	215.2	80.9	158.5	194.1
	125	124.1	134.7	297.6	118.0	146.8	283.0	111.6	159.1	267.7	105.6	174.4	253.2	74.1	155.4	177.8
	135	135.4	145.9	324.7	129.3	158.8	310.1	122.1	172.6	292.9	115.5	188.9	277.1	93.2	205.5	223.6
	150	148.0	160.4	355.0	142.3	174.0	341.2	134.6	189.1	322.7	127.7	206.7	306.3	114.7	226.2	275.1
	160	161.8	176.8	388.0	154.3	191.7	369.9	146.3	207.8	350.9	138.9	227.8	333.0	115.3	247.3	276.6
	174	176.9	192.0	424.2	169.8	208.2	407.2	162.1	225.3	388.8	153.5	246.4	368.1	97.3	200.4	233.4
	204	203.9	224.9	489.0	195.3	244.1	468.4	184.7	265.1	443.0	175.1	289.7	419.8	150.2	313.3	360.1
	225	223.6	245.5	536.1	212.6	266.1	509.8	202.1	288.5	484.6	191.4	316.6	458.9	121.7	229.3	291.9
	249	249.6	274.7	598.5	238.2	298.3	571.1	226.3	323.4	542.8	214.6	354.2	514.5	155.3	315.6	372.3
	264	263.3	292.1	631.3	249.6	316.7	598.4	237.9	342.5	570.4	225.6	376.3	541.0	153.0	338.6	366.8
	281	281.0	300.6	673.7	271.8	326.1	651.7	256.5	355.9	615.1	243.8	387.0	584.6	230.1	428.4	551.7
	301	303.9	328.2	728.8	292.1	355.7	700.4	276.1	387.1	662.2	262.3	422.4	629.1	239.3	462.8	573.8
	325	330.2	364.9	791.8	314.4	396.1	753.9	298.8	429.4	716.5	283.5	471.4	679.8	229.4	512.3	550.1
	350	357.3	408.6	856.7	339.0	442.4	813.0	323.8	479.6	776.4	306.8	528.6	735.7	177.3	352.0	425.2
	370	379.3	409.4	910.0	363.2	443.7	871.5	344.9	482.2	827.5	326.9	527.1	784.2	238.0	455.9	571.1
	390	404.7	440.3	970.9	385.8	478.2	925.5	366.5	518.5	879.4	347.5	567.3	833.6	245.8	543.8	587.5
	415	433.5	472.5	1040.0	413.0	512.5	990.9	392.7	555.2	942.2	372.7	607.6	894.2	268.6	585.5	644.7
	450	458.1	496.4	1098.9	436.9	538.1	1048.1	414.7	583.9	995.1	392.2	637.9	940.9	252.0	463.8	604.8

Air Cooled Chiller

The air-cooled chiller consists of one or more of the following: compressors, liquid coolers and air-cooled condensers that include condenser fans. The basic procedure is to measure and/or report the following:

- **Chiller manufacturer nameplate data**
- **Rated capacity in tons**
- **Model and serial number**

Evaporator:

- **the design and actual entering water**
- **the design and actual leaving water temperature**
- **the design and actual pressure drop in feet (kpa)**
- **the design and actual GPM (l/s)**

Condenser

- **the design and actual temperature**

Electrical

- **the design and actual voltage for the compressor(s)**
- **the design and actual amperage for the compressor(s)**
- **the design and actual voltage for the condenser fan(s)**
- **the design and actual amperage for the condenser fan(s)**

Air Cooled Chiller With Remote Condenser(s)

The air-cooled chiller with remote condenser(s) consists of compressors and liquid coolers or evaporators that connect to one or more air-cooled condensers.

The basic test for the air-cooled chiller with remote condenser is the same as Air Cooled Chillers with the condenser data being not applicable.

Air Cooled Condenser

The basic test for the air-cooled condenser is shown on the Associated Air Balance Council's "Air Cooled Condensing Unit" report form. The basic procedure is to measure and/or report the following:

- **Condenser manufacturer nameplate data**
- **Rated capacity in tons**
- **Model and serial number**
- **The design and actual entering air temperature**
- **The design and actual voltage for the condenser fan(s)**
- **The design and actual amperage for the condenser fan(s)**

Field Performance Testing Of Chillers

The performance test for a water-cooled chiller, air-cooled chiller, and an air-cooled chiller with a remote condenser must be specified. This test is not considered a standard test of the Balance Council, and is over and above basic chiller testing. All parties involved in the performance test must agree to the procedure, date of test, and how the various measurements will be taken.

The following tasks are the responsibility of contractors and/or manufacturers involved and will be verified by the TAB Agency before the performance test is scheduled or performed:

- **Temperature taps or thermometer wells shall have been installed for measuring the temperature of water entering and leaving the evaporator and condenser. All persons involved in the test must agree on the method of temperature measuring**
- **The water circulating system shall have been thoroughly cleaned of all foreign matter. Samples of the water shall be clear and indicate clear passage of water through the pumps, piping, and evaporators**
- **The water circulating pump shall have been tested for correct operation, the strainers cleaned, and placed in normal operation**
- **The liquid chilling package shall have been leak tested, dehydrated, evacuated, and charged with the operating amount of refrigerant**
- **Water flow measuring devices shall be installed and functioning**

General Procedures

The performance test shall be conducted by the NTT Agency and the manufacturer in the presence of the purchaser, engineer, and final owner, should they desire to be present. There should be sufficient notice of the test date so that all concerned parties can arrange to be present. Additionally, if the manufacturer desires, he shall be given sufficient time before the test to inspect and prepare the system being tested.

All test instrumentation shall be agreed upon by the purchaser, engineer, manufacturer, and NTT Agency. All thermometers used in the test shall be tested before and after the test. Any thermometer whose corrected reading (actual reading plus calibration) differs from the average by more than k 0.5ºF (0.IºC) shall not be used.

The operating conditions of the test shall conform to the following:

- **Temperatures must be the average bulk stream**

- **Each reading of liquid temperatures shall not vary by more than 0.3°F (0.1ºC)**
- **The liquid cooling temperature range shall not vary more than 0.5°F (0.IºC)**

Range = Entering Liquid O F (ºC) - Leaving Liquid O F (ºC)

- **Individual dry bulb temperature readings shall not vary by more than 2°F (1°C) from the average value**
- **The arithmetic average of all required dry bulb air temperature readings shall not vary from the specified values by more than 1ºF (0.5°C)**
- **The chilled liquid flow rate shall not deviate more than ± 5% from the specified volume.**

Performance Test of Water Cooled

Liquid Chilling Systems

After steady-state condition has been established, the following can be recorded by taking three sets of data in five-minute intervals (test readings for each data set shall be taken as simultaneously as possible). All the required readings shall be taken at five-minute intervals and the test continued until at least three consecutive sets of readings are within the operating conditions.

The following data shall be recorded:

- **Temperature of liquid entering cooler, ºF (ºC)**
- **Temperature of liquid leaving cooler, ºF (ºC)**
- **Chilled liquid flow rate, gpm (Vs)**
- **Description of liquid sufficient to obtain necessary physical properties**
- **Power input to compressor, expressed as:**
- **Power input to motor, kW, Note: Manufacturer must furnish kW meter for all voltages over 600.**
- **Electrical power input to controls and auxiliary equipment, kW**
- **Condenser water pressure drop (inlets to outlets), ft. of water (kpa) (pressure drop X 2.31 = feet of water) (Note: Steady state will be determined by pressure drop).**
- **Dry bulb temperature of air entering condenser, ºF (ºC)**
- **Condenser fan motor power consumption, kW**
- **Fan rotative speed, rpm**
- **Barometric pressure, inches of HG**

The following general information shall be recorded:

- **Nameplate data including make, model, size, and refrigerant, sufficient to completely identify the liquid chilling package**
- **Electrical frequency in Hz**
- **Compressor driver or input rotative speed (rpm) for open type compressors**
- **Temperature of ambient air at cooler, ºF (ºC)**
- **Actual voltage, and current in amperes, for each phase, of all electric motor drives**
- **Date, time, and location of test**
- **Names of test supervisor and witnessing personnel**
- **Specified refrigerant**
- **Operating charge of refrigerant, lb (kg)**

Cooling Tower Testing

The NTT basic test for a cooling tower measures water flow, temperature both entering and leaving the tower, and electrical components. The performance test encompasses testing recommended by NTT or the Cooling Tower Institute (CTI). Establish the tower airflow by determining flow in pounds per hour, while CTI determines the tower airflow according

to the fans' brake horsepower.

The NTT performance test is not dependent upon the tower manufacturer's curves. A graph generated by the NTT performance test illustrates the cooling tower operation through a wide range of conditions.

Cooling tower performance test must be specified as per the NTT Cooling Tower performance procedure or the CT1 test. Performance testing is over and above the basic testing of towers.

Basic Test

The standard test for cooling towers is shown on the Associated Air Balance Council's "Cooling Tower Operation Test Report." The procedure is to measure and record:

- **Design and actual flow of the water**

- **Design and actual entering water temperature**
- **Design and actual leaving water temperature**
- **Design and actual entering wet bulb**
- **Design and actual ambient dry bulb**

Along with the data above, record the tower's nameplate and/or submittal data to indicate:

- **Tower designation**
- **Manufacturer**
- **Model number and serial number**
- **Rated capacity in tons**

Record the motor nameplate and heater element with actual amps, volts, calculated brake horsepower. Record the design and actual rpms as well.

Performance Test Parameters

1. The cooling tower performance test must be specified and is not a standard test of the TAB Agency. All parties involved in the cooling tower test must agree to the procedure, date of test, and how the various measurements will be taken. The test performed will be as prescribed by NTT or CTI.

2. While the following tasks are primarily the responsibility of other trades, they should be checked by the testing agency as well before scheduling or conducting the cooling tower performance test:

- **Temperature taps or thermometer wells shall have been installed for measuring the temperature of hot water entering the tower, cold water leaving the tower, and make-up water entering the tower. All persons involved in the test must agree on the method of temperature measuring.**
- **Provisions shall have been made to measure the re-circulating, make-up, and blow down water flow rate with the proper runs of straight pipe upstream and downstream of the measurement device. All persons involved in the test must agree on the method of water flow measurement.**
- **A 110-volt AC power source shall be available at the tower site for mechanically aspirated psychrometer.**
- **The water circulating system shall have been thoroughly cleaned of all foreign matter. Samples of water shall be clear.**
- **The cooling tower fans shall be rotating correctly and operating within ±10% of the design horsepower loading. All obstructions shall have been removed from the path**

of airflow. Permanent obstructions shall be noted by the NTT TAB Agency.

- The cooling tower wet-deck surface (fill) shall be clean and free of foreign materials such as scale, algae, or tar.

- The water in the upper and lower basins of the tower shall be set at proper operation levels as recommended by the tower manufacturer.

- Water circulating pumps shall have been tested for correct operation and placed in normal operating conditions.

- If performing the CT1 test, the cooling tower manufacturer shall have submitted a full set of performance curves for the cooling tower. Additionally, the manufacturer shall provide a listing of the design point conditions of:

- design liquid-to-gas ratio;
- hot water temperature;
- cold water temperature;
- inlet wet bulb temperature;
- inlet dry bulb temperature;
- fan motor horsepower (motor output power unless otherwise specified);
- fan motor efficiency;
- design barometric pressure;
- and density of air entering the fan(s) at design conditions.

General Procedures

The performance test is conducted by the NTT TAB Agency in the presence of authorized representatives of the manufacturer, the purchaser, and the owner, should they desire to be present. The NTT Agency shall give sufficient advance notice of the test date so that all concerned parties can arrange to be present. Additionally, the manufacturer shall be given sufficient time before the test to inspect and prepare the tower, if desired.

The NTT TAB Agency uses a team of at least one TAB engineer and one technician to perform this test. Representatives of the manufacturer and purchaser are allowed to observe, but shall not participate in, reading and recording of test data.

All test instrumentation shall be furnished, or inspected and approved, by the NTT TAB Agency. All thermometers used in the test are checked before and after the test by immersing them in a common bath, the temperature of which is between 70°F (32°C) and 90°F (63°C). Any thermometer whose corrected reading (actual reading plus calibration)

differs from the average by more than +0.15"F (0.l"C) shall not be used.

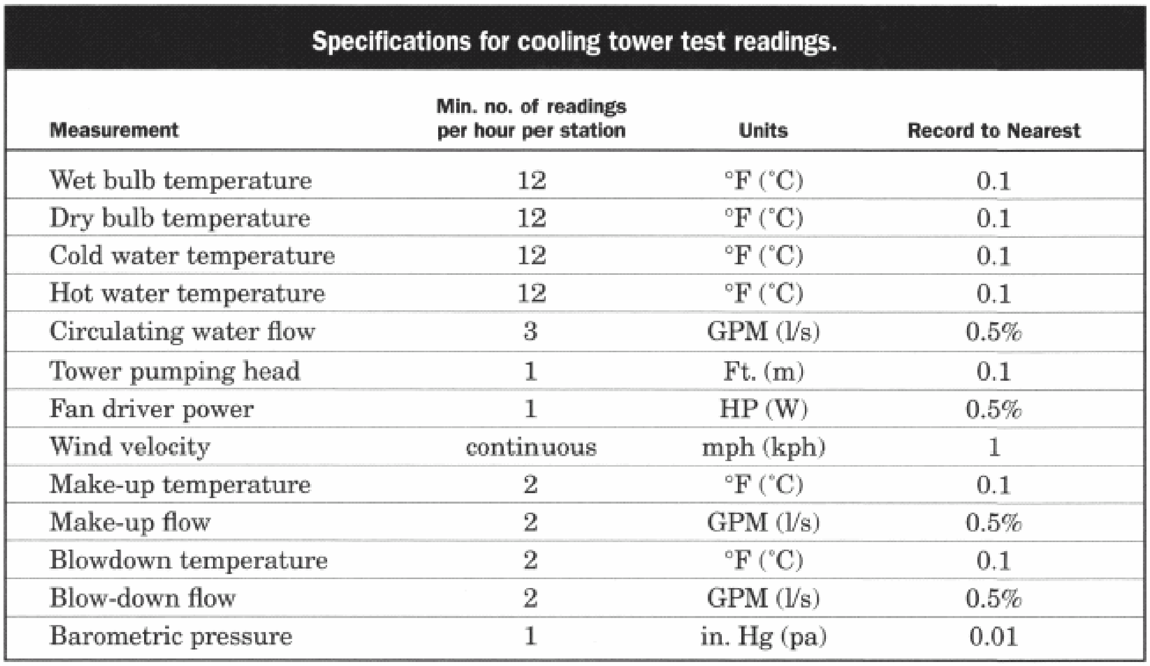
The operating conditions during the test must conform to the following:

- **Entering wet-bulb temperature shall be within ±15°F (1°C) of design.**
- **Range shall be within ±20% of design.**
- **Power input to fan (s) shall be within ±10% of design.**
- **Average wind velocity shall be 10 mph or less, and shall not exceed 15 mph for longer than one minute.**
- **Entering dry bulb, if applicable, shall not exceed ±20% of design.**
- **Circulating water flow shall be within ±10% of design.**
- **Barometric pressure shall be within ±1 in. Hg. (3.4 kpa) of design.**

For a valid test, the variations in the measured conditions over the one-hour test period must conform to the following:

- **Circulating water flow shall not vary by more than 2%.**
- **Range shall not vary by more than *5%.***
- **Total load shall not vary by more than *5%.***
- **Entering wet bulb temperature shall not exceed ±2ºF (1°C). Note: Air temperature readings may fluctuate momentarily, but variations in the averages during the test period should not exceed the indicated value.**
- **Btuh of a condenser are 15,000 btuh per ton.**
- **With a 10º ΔT of the water the GPM is 3 gpm per ton, change the gpm changes the ΔT**

The duration of the test shall be for not less than one hour, starting after the system has reached steady state conditions. During the test, readings shall be taken and recorded at regular intervals. The frequency of test readings, the units of measure, and the number of significant figures shall be as shown in Table 13.1.



Specifications for cooling tower test readings.

Measurement	Min. no. of readings per hour per station	Units	Record to Nearest
Wet bulb temperature	12	°F (°C)	0.1
Dry bulb temperature	12	°F (°C)	0.1
Cold water temperature	12	°F (°C)	0.1
Hot water temperature	12	°F (°C)	0.1
Circulating water flow	3	GPM (l/s)	0.5%
Tower pumping head	1	Ft. (m)	0.1
Fan driver power	1	HP (W)	0.5%
Wind velocity	continuous	mph (kph)	1
Make-up temperature	2	°F (°C)	0.1
Make-up flow	2	GPM (l/s)	0.5%
Blowdown temperature	2	°F (°C)	0.1
Blow-down flow	2	GPM (l/s)	0.5%
Barometric pressure	1	in. Hg (pa)	0.01

Table 13.1 Specifications for cooling tower test readings.

Record test data on the NTT test data summary sheet (Appendix 13.1). Upon completion of the test, the sheet is signed by the NTT TAB engineer and any representatives of the manufacturer, purchaser, or owner who are present. Give one copy of each authenticated test form to each person signing the test form.

Instrumentation

Temperature measurements shall be made with calibrated mercury-in-glass or resistance thermometers. These thermometers must be accurate to ±O.1"F with indicators or recorders graduated in increments of not more than 0.2ºF (0.IºC).

All air temperature measurements shall be made with mechanically aspirated psychrometer using the type of thermometers described above. The temperature-sensitive element shall be shielded from direct sunlight or other significant sources of radiant heat.

When measuring wet bulb temperature, cover the temperature-sensitive element with a wick continuously wetted with distilled water of a temperature approximately equal to the wet bulb temperature being measured. The wick shall fit snugly over, and extend at least 3/4 inches (19 mm) past, the temperature-sensitive element. The air velocity over the element shall be at least 950 (4.8 m/s) and no greater than 1050 (5.3 m/s) feet per minute.

Any of the following calibrated devices may be used to measure water flow rate:

- **Venturi or flow nozzle**

- **Orifice plate**
- **Turbine meter (traverse)**
- **Pitot tube (traverse)**
- **Other mutually agreed upon flow measuring devices**

For these devices, ensure that the proper upstream and downstream flow conditions exist.

Report of Results

The report of test results must include:

- **A description of the cooling tower, including the name of the manufacturer, the model number, serial number, orientation, and principal dimensions.**
- **A sketch of the installation showing the location of instrumentation for measuring water flow, water temperatures, air temperatures, etc. Note any buildings, walls, obstructions, or other equipment in the immediate vicinity of the tower being tested. Give particular attention to any equipment or facilities discharging heat or vapor near the tower.**
- **A copy of all test data sheets, plus a single summary sheet tabulating all test data averages used in calculating results and comparing them to the associated design values for the same parameter (Appendix 13.1).**
- **A copy of all calculations made in determining the result.**
- **A brief summary stating the test result and mentioning any areas in which the test deviated from the test parameters.**
- **The distribution of the report shood be limited to the parties directly involved in the test.**

Evaluating the Test

Since a cooling tower test can seldom, if ever, be conducted at both design heat load and design entering wet bulb temperature, the test procedure and method for calculating results must address several issues, including:

- **Establishing the maximum deviations between the design conditions and actual test conditions that can be allowed while still yielding a valid test result.**
- **Specifying the instrumentation and procedures that must be used in order to obtain an accurate measurement of the actual cooling tower thermal performance.**
- **Establishing a consistent procedure for comparing the measured performance to the required performance.**

The degree to which the test conditions may deviate from design and still yield a valid test, as set forth in the test parameters, has been accepted practice in the cooling tower

industry for many years. They reasonably meet the need to obtain test data close to the tower design conditions, while not imposing such stringent limits on the test conditions as to severely limit the opportunities for obtaining a test.

Similarly, the instrumentation and procedures set forth in the test parameters are the culmination of many years experience in testing cooling towers of all sizes and configurations. They have been shown to give consistent and repeatable results.

Performance Test

The objective of this test is to determine a performance factor (P.F.) for the tower that is constant at all heat load conditions and all entering (W) wet bulb conditions that may occur. The performance factor (P.F.) will be used to determine the performance of the tower at the design conditions.

Following is an outline of the analysis that establishes this test procedure, according to applicable cooling tower factors and how they operate:

- **Water flow GPM and airflow across the tower are constant.**

- **At constant heat load, the approach will decrease as the entering WB increases. Approach is defined as the difference between the temperature of the water leaving the tower and the wet bulb temperature of the air entering the tower.**

- **Evaporation takes place in a tower but has no effect on the cooling of the water. It is simply a conversion of air sensible heat to latent heat.**

- **For standard cooling towers, the heat rejected per ton of refrigeration is 15,000 BTU/Hr. (3 GPM with a 10ºΔT)**

- **At any point of test, the water, air, and fill arrangement combine to allow the air to remove a given percentage of the theoretical heat available to it. This percentage, called the Performance Factor (P.F.), will remain constant at any entering WB temperature and heat load as long as the GPM (Vs), CFM (Vs), and fill do not change.**

Test Procedure

During the test procedure, the following data is recorded with the same intervals and no greater deviations than outlined in 13.3.3:

- **Water temperature entering the tower.**

- **Water temperature leaving the tower.**

- **WB temperature of the air entering the tower (use table 11 to convert to enthalpy H2).**

- **WB temperature of the air leaving the tower (use table 11 to convert to enthalpy H3).**

- **Water flow rate GPM.**

BTUH = GPM X 60 X 8.337 X ΔT of the water

Tons = GPM X ΔT of the water / 30

GPM = BTUH / 60 X 8.337 X ΔT of the water

GPM = (Tons X 30) / ΔT of the water

GPM = (Air Density X CFM X Δ Enthalpy h1-h2) / (8.337 X Water ΔT) .

CFM = (GPM X 8.337 X Water ΔT) / (Air Density X Δ Enthalpy h1-h2)

Example 12: Find the tons of a cooling tower with an entering water temperature of 90º and a leaving water temperature of 80º and a 145 gpm.

Solution 1

- **GPM x 60 X 8.337 X ΔT of the water = BTUH**
- **BTUH / 15,000 = Tons**
- **145 gpm X 60 X 8.337 X 10º ΔT = 725,319 btuh**
- **725,319 btuh / 15,000 = 48.35 tons**

Solution 2

- **GPM X ΔT of the water / 30 = tons**
- **(145 gpm X 10º ΔT) / 30 = 48.33 tons**

Example 13: Find the GPM of a cooling tower with an entering water temperature of 90º and a leaving water temperature of 80º and a entering air temperature of 95º dry bulb and 67º wet bulb, leaving air temperature of 92º dry bulb and 84º wet bulb and a 10,167 CFM.

Solution 1

- **GPM = (Air Density X CFM X Δ Enthalpy h1-h2) / (8.337 X Water ΔT)**
- **Air density of 92º dry bulb and 84º wet bulb is 0.071**
- **Enthalpy of 92º dry bulb and 84º wet bulb is 48.13**
- **Enthalpy of 95º dry bulb and 67º wet bulb is 31.38**
- **Water ΔT is 10º**
- **(0.071 X 10,167 X 16.75) / (8.337 X 10) = 12,091.10 / 83.37 = 145.02 gpm**

Analysis of the Test Data

The enthalpy (total heat see table 11) of the air at the leaving WB temperature minus the enthalpy of the air at the entering WB temperature determines the quantity of heat in BTU/LB. that the air picks up while passing through the tower. If this heat gain (enthalpy difference), is divided by the theoretical enthalpy difference that would result if the air left the tower at a wet bulb temperature equal to the temperature of the water entering the tower, the result is the Performance Factor. This theoretical enthalpy difference occurs at zero approach and represents a tower performance of 100%:

Measure Enthalpy Difference / Theoretical Enthalpy Difference = Performance Factor (PF)

If the water flow GPM and the airflow through the tower remain constant,

then the P.F. will remain the same as the heat load and WB temperature change.

With this as a basis, a cooling tower can be tested at a reasonable operating condition and from the data of this test calculate the performance of the tower at the specified full load conditions. It is possible to prepare a performance curve for the test tower at a given P.F. It has been found that the most useful curve is to plot approach vs. heat rejection at specified entering WB temperature, test water flow GPM and test P.F. From this curve, the engineer can determine if the condenser water system will satisfy the requirements of the operation of the tower at a wide range of operating conditions. See Example A in Appendix 13.2.

Cooling Tower Terms

EWT = Entering water temperature ºF (ºC)

LWT = Leaving water temperature ºF (ºC)

EWB =Air entering tower wet bulb ºF (ºC)

LWB =Air leaving tower wet bulbº F (ºC)

GPM = Gallons of water per minute over

d = Air density in lbs. per cubic ft. leaving (see tables 9)

R = Range across tower = EWT - LWT

A =Approach to wet bulb = LWT - EWB

h = Enthalpy of air at a wet bulb temperature of EWT (see table 11)

h1 = Enthalpy of air at LWB (see table 11)

h2 = Enthalpy of air at EWB (see table 11)

P.F. = Tower performance capacity factor. This is a percentage of the heat picked up by the air in passing through the tower to the amount it would have picked up if it had left the tower saturated at the EWT tower at test tower = (h2 – h3)/ (h-h3).

THR = Total heat rejected by tower in BTU per minute = GPM x* 8.337 *x (EWT - LWT)

Lam = lbs. of air per minute through tower = THR / (h.2 - h3)

CFM = Tower CFM = Lam / d (of leaving air)

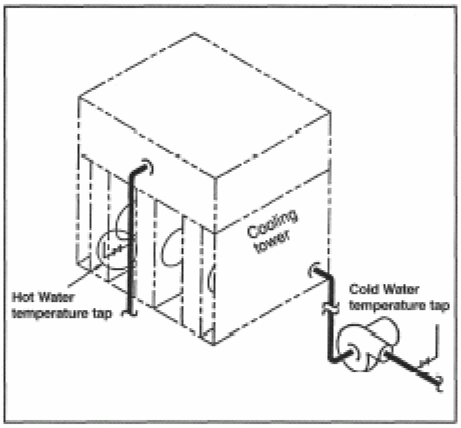


Fig. 13.1 Location of Water Temperature Taps

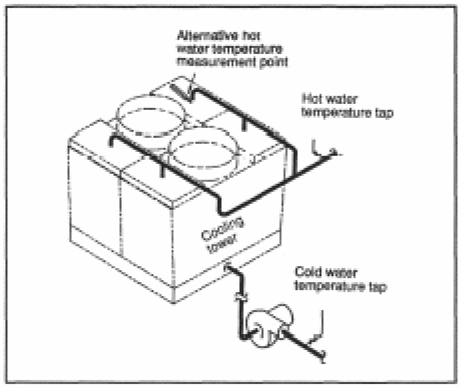


Fig. 13.2 Location of Water Temperature Taps

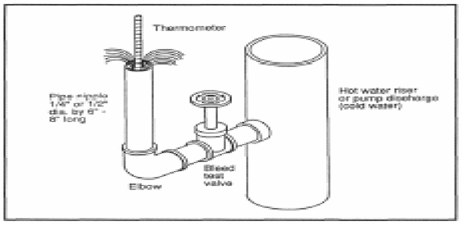


Fig. 13.3 Temperature Measured Through a Bleed Test Valve

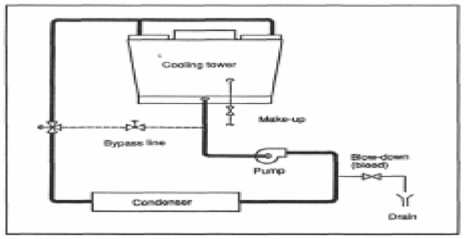


Fig. 13.4 Typical Cooling Tower System with Bypass

Water Temperature

Water temperatures are to be measured using calibrated mercury-in-glass or resistance thermometers, accurate to 0.1ºF (0.1º"C) with indicators or recorders graduated in increments no greater than 0.2º"F (0.2º"C). The temperature-sensitive element must be carefully located so that the temperature being sensed is representative of the true average temperature of the bulk fluid.

The hot water temperature entering the tower should be measured in the riser (s) leading to the tower inlet (s), as close as practical to the tower, or at the discharge of the inlet riser(s) into the flume or distribution box (es) on the tower (Figures 13.1 & 13.2).

The cold-water temperatures should be measured on the discharge side of the system circulating pump (Figures 13.1 *8z* 13.2). If the temperature of the cold water leaving the pump is being measured through a bleed test valve (as shown in Figure 13.3), the average temperature must be corrected for the heat added by the pump and the valve. The test valve must be sufficiently opened to provide adequate quantities to establish accurate temperature measurements.

Cold-water temperatures should not be taken in the tower basin or by bleeding water from the basin or outlet pipe since temperature gradients of several degrees may exist in the basin and the suction piping extending from the tower to the pump. Only after the water is well-mixed by the pump can one be assured the measured temperature is representative of the true cold water temperature leaving the tower.

Many cooling tower systems have an automatic or manual bypass arrangement (Figure 13.4) used to control temperatures in the cooling loop by diverting hot water from the cooling tower riser into the cold water line leaving the tower or into the tower basin. Where such a bypass line exists, it is imperative it be shut

The cold-water temperatures should be measured on the discharge side of the system circulating pump (Figures 13.1 *8z* 13.2). If the temperature of the cold water leaving the pump is being measured through a bleed test valve (as shown in Figure 13.3), the average temperature must be corrected for the heat added by the pump and the valve. The test valve must be sufficiently opened to provide adequate quantities to establish accurate temperature measurements. Cold-water temperatures should not be taken in the tower basin or by bleeding water from the basin or outlet pipe since temperature gradients of several degrees may exist in the basin and the suction piping extending from the tower to the pump. Only after the water is well-mixed by the pump can one be assured the measured temperature is representative of the true cold water temperature leaving the tower.

Many cooling tower systems have an automatic or manual bypass arrangement (Figure 13.4) used to control temperatures in the cooling loop by diverting hot water from the cooling tower riser into the cold water line leaving the tower or into the tower basin. Where such a bypass line exists, it is imperative it be shut with a liquid of high thermal conductivity (oil or water).

Inlet Air Temperature Measurements

The wet bulb temperature of the air entering the cooling tower is a required measurement on all tests. Similarly, the dry bulb temperature of the entering air stream is required on forced draft (blow through) towers to determine the density of the air entering the fan(s).

Wet bulb temperature must be measured with mechanically aspirated psychrometer. Mount the psychrometer on tripods or other support devices so they remain stationary for the duration of the test. A sufficient number of instruments, located within four feet of the cooling tower air inlet(s), shall be employed to ensure that the test average is an accurate representation of the true average inlet wet bulb temperature. See Figures 13.5 and 13.6 for typical situations.

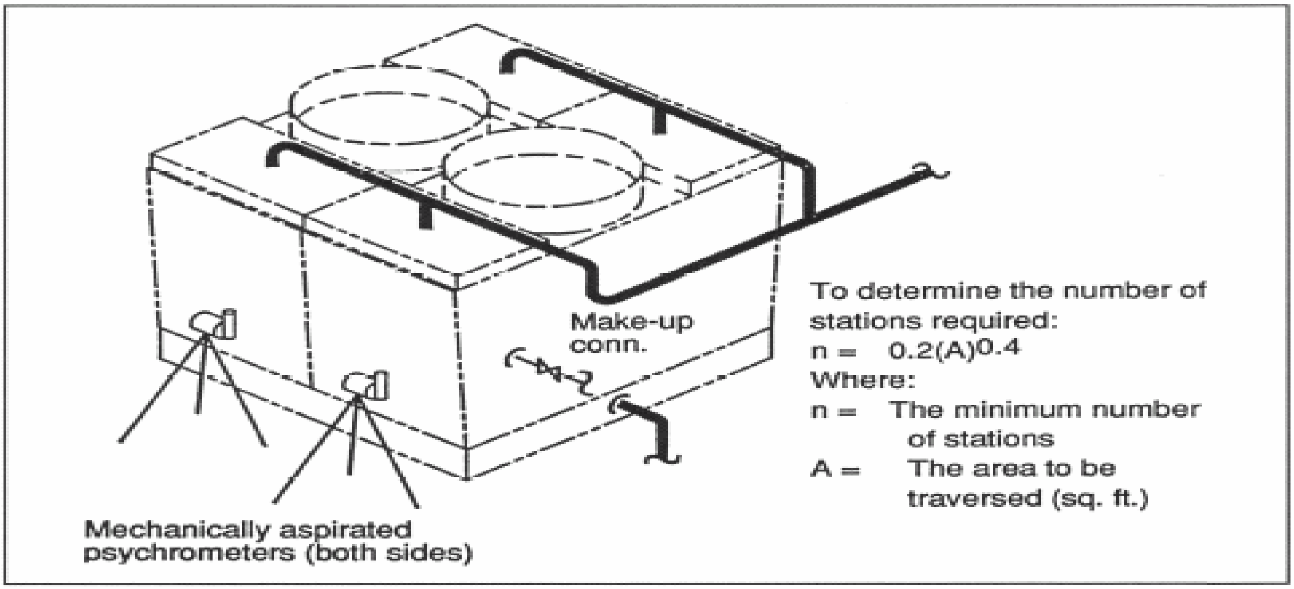


Fig. 13.5 Location of Psychrometers on Typical Crossflow Tower

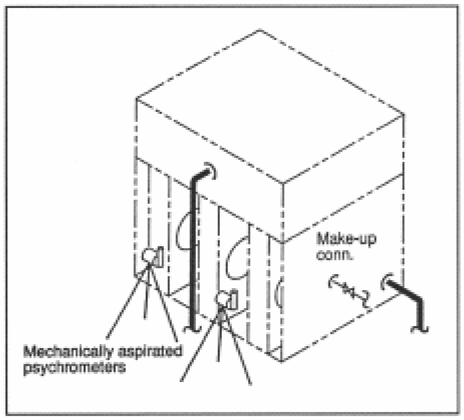


Fig. 13.6 Location of Psychrometers on Typical Forced-draft Tower

When dry bulb temperature measurement of the inlet air is made on forced draft (blow through) type towers for the purpose of density determination, a measurement accuracy of 2°F to 5°F (1ºC-2ºC) for the average inlet dry bulb temperature is usually satisfactory. In most cases this can be achieved by means of a single psychrometer situated at a representative location along an air inlet face.

Water Flow Rate Measurement

As specified in the test code, water flow rates may be measured by any of several procedures and devices. Since the circulating water, flow rate is one of, if not the single, most important test parameter; take considerable care to ensure it is measured accurately.

Install the flow measurement station in a straight section of pipe with the proper distances between it and any upstream or down stream fittings. Specific recommendations for various flow measurement devices can be obtained from the manufacturer. In pipes smaller than 8 inches (200 mm), the preferred measurement device is a venturi, flow nozzle, or orifice plate. Alternatively, a calibrated turbine meter may be used. For pipes 9 inches (225 mm) and larger, any of the aforementioned devices

can be used. If these become too expensive or unwieldy, measure the flow by Pitot tube traverse.

Taps for Pitot tube traverses can be installed in either vertical or horizontal pipe runs. As illustrated in Figure 13.7, the taps must be perpendicular to the pipe and at 90" from one another. Adequate clearance [roughly the pipe diameter, plus three feet (1 m)] must be available at each tap to install and position the Pitot tube.

If make-up enters or blow down leaves the tower or the system at a point where it will affect any of the measured test values, they must either be shut off for the duration of the test or the flow rates measured. Usually, these measurements lend themselves to a volumetric or bulk measurement procedure. In many instances, a water meter will have been installed in the make-up line and can be used for this purpose. In virtually all instances, the blow down can be valved off during the test period.

Fan Motor Power

Most cooling tower fans are driven by electric motors of 480V or less. The preferred method for measuring power is to use a calibrated, clamp-on wattmeter to measure the KW input to the motor(s) and convert this to

Horse power output using the motor manufacturer's efficiency values. Brake horsepower can be calculated.

Wind Velocity

When a cooling tower test is run under conditions of little or no wind, the parties to the test often may agree to forego measurement of wind velocity. Strict compliance with the test code, however, requires that wind velocity be measured with a meteorological anemometer and wind vane. For towers with an overall height of less than 20 feet (6 m), the point of measurement is 5 feet (1.5 m) above the basin curb at a distance 50 to 100 feet (15-30 m) upwind of the tower.

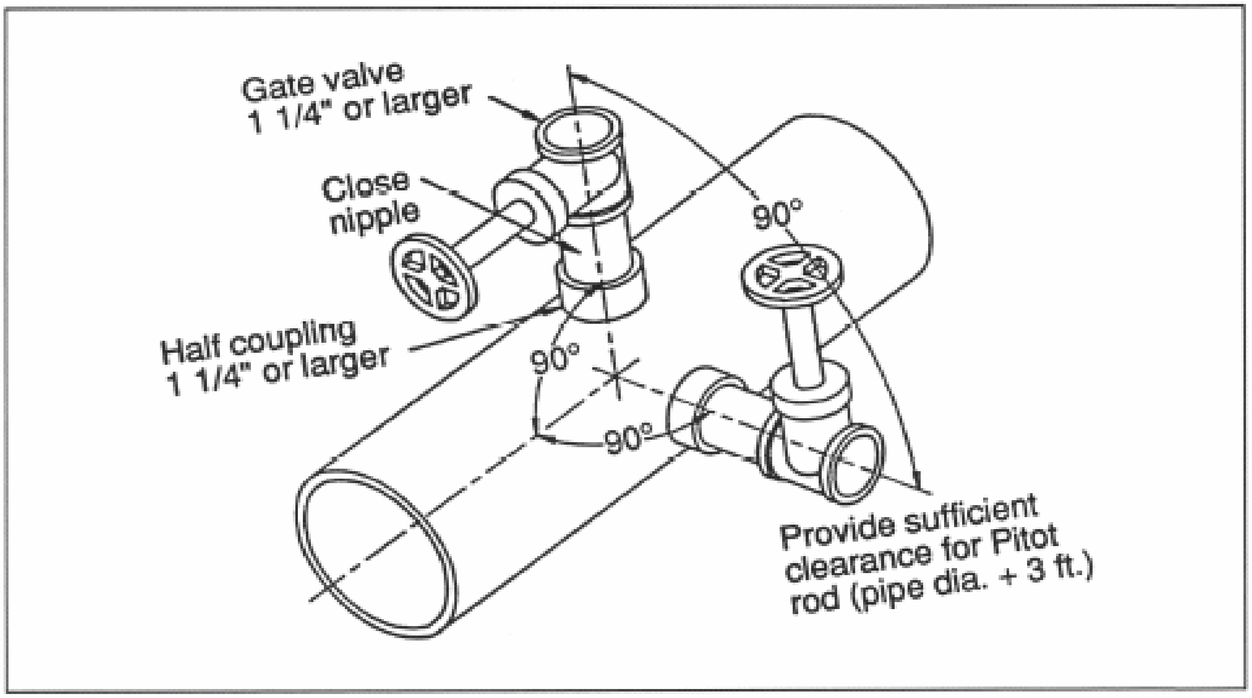
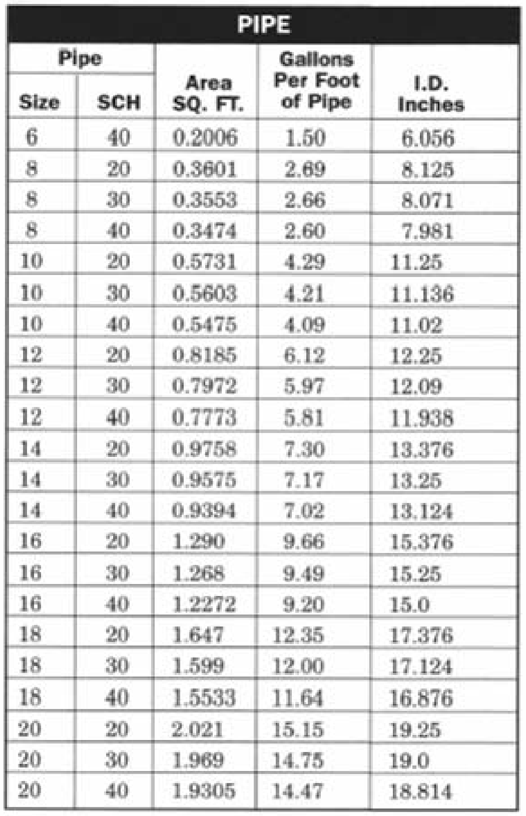


Fig. 13.7 Detail of Taps for Pitot-Tube Traverse

GPM = FPM X Gallons per foot

FPM = 1040 X √ VP



PIPE				
Pipe		Area SQ. FT.	Gallons Per Foot of Pipe	I.D. Inches
Size	SCH			
6	40	0.2006	1.50	6.056
8	20	0.3601	2.69	8.125
8	30	0.3553	2.66	8.071
8	40	0.3474	2.60	7.981
10	20	0.5731	4.29	11.25
10	30	0.5603	4.21	11.136
10	40	0.5475	4.09	11.02
12	20	0.8185	6.12	12.25
12	30	0.7972	5.97	12.09
12	40	0.7773	5.81	11.938
14	20	0.9758	7.30	13.376
14	30	0.9575	7.17	13.25
14	40	0.9394	7.02	13.124
16	20	1.290	9.66	15.376
16	30	1.268	9.49	15.25
16	40	1.2272	9.20	15.0
18	20	1.647	12.35	17.376
18	30	1.599	12.00	17.124
18	40	1.5533	11.64	16.876
20	20	2.021	15.15	19.25
20	30	1.969	14.75	19.0
20	40	1.9305	14.47	18.814

Fig. 11.3 Pipe Data

Example 14: Fine the tons of a condenser that has a 8" SCH 40 with a pressure of 23.347psig and a total pressure of 23.352 psig and a entering water of 85º and leaving water of 93º.

Solution

- **Total pressure – Pressure = VP**
- **FPM = 1040 X √ VP**
- **GPM = FPM X Gallons per foot**
- **Ton = GPM X 60 X 8.337 X Δ Temperature of the water**
- **23.352 psig – 23.347 = 0.005 VP**
- **1040 X √ 0.005 = 73.54 FPM**
- **73.54 X 2.6 gallons per foot = 191.20 GPM**
- **191.20 X 60 X 8.337 X 8 = 765,136.5 btuh**
- **765,136.5 / 15,000 = 51.01 tons**