CHAPTER 37

TESTING, ADJUSTING, AND BALANCING

SYSTEMS that control the environment in a building change with time and use, and must be rebalanced accordingly. The designer must consider initial and supplementary testing and balancing requirements for commissioning. Complete and accurate operating and maintenance instructions that include intent of design and how to test, adjust, and balance the building systems are essential. Building operating personnel must be well-trained, or qualified operating service organizations must be employed to ensure optimum comfort, proper process operations, and economical operation.

This chapter does not suggest which groups or individuals should perform the functions of a complete testing, adjusting, and balancing procedure. However, the procedure must produce repeatable results that meet the design intent and the owner's requirements. Overall, one source must be responsible for testing, adjusting, and balancing all systems. As part of this responsibility, the testing organization should check all equipment under field conditions to ensure compliance.

Testing and balancing should be repeated as systems are renovated and changed. Testing boilers and other pressure vessels for compliance with safety codes is not the primary function of the testing and balancing firm; rather it is to verify and adjust operating conditions in relation to design conditions for flow, temperature, pressure drop, noise, and vibration. ASHRAE *Standard* 111 details procedures not covered in this chapter.

TERMINOLOGY

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, and (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 to meet design specifications, and recording and reporting the results.

The following definitions are 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.

GENERAL CRITERIA

Effective and efficient TAB requires a systematic, thoroughly planned procedure implemented by experienced and qualified staff. All activities, including organization, calibration of instruments, and execution of the work, should be scheduled. Air-side work must be coordinated with water-side and control work. Preparation includes planning and scheduling all procedures, collecting necessary data (including all change orders), reviewing data, studying the system to be worked on, preparing forms, and making preliminary field inspections.

Air leakage in a conduit (duct) system can significantly reduce performance, so conduits (ducts) must be designed, constructed, and installed to minimize and control leakage. During construction, all duct systems should be sealed and tested for air leakage. Water, steam, and pneumatic piping should be tested for leakage, which can harm people and equipment.

Design Considerations

TAB begins as design functions, with most of the devices required for adjustments being integral parts of the design and installation. To ensure that proper balance can be achieved, the engineer should show and specify a sufficient number of dampers, valves, flow measuring locations, and flow-balancing devices; these must be properly located in required straight lengths of pipe or duct for accurate measurement. Testing depends on system characteristics and layout. Interaction between individual terminals varies with pressures, flow requirements, and control devices.

The design engineer should specify balancing tolerances. Minimum flow tolerances are ±10% for individual terminals and branches in noncritical applications and ±5% for main air ducts. For critical water systems where differential pressures must be maintained, tolerances of ±5% are suggested. For critical air systems, recommendations are the following:

Positive zones:

Supply air	0 to +10%
Exhaust and return air	0 to −10%

The preparation of this chapter is assigned to TC 9.7, Testing and Balancing.

Negative zones:

Supply air	0 to −10%
Exhaust and return air	0 to +10%

Balancing Devices. Balancing devices should be used to provide maximum flow-limiting ability without causing excessive noise. Flow reduction should be uniform over the entire duct or pipe. Single-blade dampers or butterfly balancing valves are not good balancing valves because of the uneven flow pattern at high pressure drops. Pressure drop across equipment is not an accurate flow measurement but can be used to determine if the manufacturer design pressure is within specified limits. Liberal use of pressure taps at critical points is recommended.

AIR VOLUMETRIC MEASUREMENT METHODS

General

The pitot-tube traverse is the generally accepted method of measuring airflow in ducts; ways to measure airflow at individual terminals are described by manufacturers. The primary objective is to establish repeatable measurement procedures that correlate with the pitot-tube traverse.

Laboratory tests, data, and techniques prescribed by equipment and air terminal manufacturers must be reviewed and checked for accuracy, applicability, and repeatability of results. Conversion factors that correlate field data with laboratory results must be developed to predict the equipment's actual field performance.

Air Devices

Generally, loss coefficients given by air diffuser manufacturers should be checked for accuracy by field measurement and by comparing actual flow measured by pitot-tube traverse to actual measured velocity. Air diffuser manufacturers usually base their volumetric test measurements on a deflecting vane anemometer. The velocity is multiplied by an empirical effective area to obtain the air diffuser's delivery. Accurate results are obtained by measuring at the vena contracta with the probe of the deflecting vane anemometer.

Methods advocated for measuring airflow of troffer-type terminals are similar to those for air diffusers. A capture hood is frequently used to measure device airflows, primarily of diffusers and slots. Loss coefficients should be established for hood measurements with varying flow and deflection settings. If the air does not fill the measurement grid, the readings will require a correction factor (similar to the loss coefficient).

Rotating vane anemometers are commonly used to measure airflow from sidewall grilles. Effective areas (loss coefficients) should be established with the face dampers fully open and deflection set uniformly on all grilles. Correction factors are required when measuring airflow in open ducts [i.e., damper openings and fume hoods (Sauer and Howell 1990)].

All flow-measuring instruments should be field-verified by running pitot-tube traverses to establish correction and/or density factors.

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 *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.

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.

Mixture Plenums

Approach conditions are often so unfavorable that the air quantities comprising a mixture (e.g., outside air and return air) cannot be determined accurately by volumetric measurements. In such cases, the mixture's temperature indicates the balance (proportions) between the component airstreams. Temperatures must be measured carefully to account for stratification, and the difference between outside and return temperatures must be greater than 20°F. The temperature of the mixture can be calculated as follows:

$$Q_t t_m = Q_o t_o + Q_r t_r \tag{1}$$

where

Q_t = total measured air quantity, %
Q_o = outside air quantity, %
Q_r = return air quantity, %
t_m = temperature of outside and return mixture, °F
t_o = outside temperature, °F
t_r = return temperature, °F

Pressure Measurement

Air pressures measured include barometric, static, velocity, total, and differential. For field evaluation of air-handling performance, pressure should be measured per ASHRAE *Standard* 111 and analyzed together with manufacturers' fan curves and system effect as predicted by AMCA *Standard* 210. When measured in the field, pressure readings, air quantity, and power input often do not correlate with manufacturers' certified performance curves unless proper correction is made.

Pressure drops through equipment such as coils, dampers, or filters should not be used to measure airflow. Pressure is an acceptable means of establishing flow volumes only where it is required by, and performed in accordance with, the manufacturer certifying the equipment.

Stratification

Normal design minimizes conditions causing air turbulence, to produce the least friction, resistance, and consequent pressure loss. Under certain conditions, however, air turbulence is desirable and necessary. For example, two airstreams of different temperatures can stratify in smooth, uninterrupted flow conditions. In this situation, design should promote mixing. Return and outside airstreams at the inlet side of the air-handling unit tend to stratify where enlargement of the inlet plenum or casing size decreases air velocity. Without a deliberate effort to mix the two airstreams (e.g., in cold climates, placing the outside air entry at the top of the plenum and return air at the bottom of the plenum to allow natural mixing), stratification can be carried throughout the system (e.g., filter, coils, eliminators, fans, ducts). Stratification can freeze coils and rupture tubes, and can affect temperature control in plenums, spaces, or both.

Stratification can also be reduced by adding vanes to break up and mix the airstreams. No solution to stratification problems is guaranteed; each condition must be evaluated by field measurements and experimentation.

BALANCING PROCEDURES FOR AIR DISTRIBUTION

No one established procedure is applicable to all systems. The bibliography lists sources of additional information.

Instruments for Testing and Balancing

The minimum instruments necessary for air balance are

- Manometer calibrated in 0.005 in. of water divisions
- Combination inclined/vertical manometer (0 to 10 in. of water)
- Pitot tubes in various lengths, as required
- Tachometer (direct-contact, self-timing) or strobe light
- Clamp-on ammeter with voltage scales (RMS type)
- Rotating vane anemometer
- Flow hood
- Dial thermometers (2 in. diameter and 1°F graduations minimum) and glass stem thermometers (1°F graduations minimum)
- Sound level meter with octave band filter set, calibrator, and microphone
- Vibration analyzer capable of measuring displacement velocity and acceleration
- Water flowmeters (0 to 50 in. of water and 0 to 400 in. of water ranges)
- Compound gage
- Test gages (100 psi and 300 psi)
- Sling psychrometer
- Etched-stem thermometer (30 to 120°F in 0.1°F increments)
- Hygrometers
- Digital thermometers, relative humidity and dew-point instruments

Instruments must be calibrated periodically to verify their accuracy and repeatability before use in the field.

Preliminary Procedure for Air Balancing

Before balancing the system,

1. Obtain as-built design drawings and specifications, and become thoroughly acquainted with the design intent.
2. Obtain copies of approved shop drawings of all air-handling equipment, outlets (supply, return, and exhaust), and temperature control diagrams, including performance curves. Compare design requirements with shop drawing capacities.
3. Compare design to installed equipment and field installation.
4. Walk the system from the air-handling equipment to terminal units to determine variations of installation from design.
5. Check dampers (both volume and fire) for correct and locked position and temperature control for completeness of installation before starting fans.
6. Prepare report test sheets for both fans and outlets. Obtain manufacturer's outlet factors and recommended test procedure. A summation of required outlet volumes permits cross-checking with required fan volumes.
7. Determine the best locations in the main and branch ductwork for the most accurate duct traverses.
8. Place all outlet dampers in the full open position.
9. Prepare schematic diagrams of system as-built ductwork and piping layouts to facilitate reporting.
10. Check filters for cleanliness and proper installation (no air bypass). If specifications require, establish procedure to simulate dirty filters.
11. For variable-volume systems, develop a plan to simulate diversity.

Equipment and System Check

1. All fans (supply, return, and exhaust) must be operating before checking the following items:
   - Motor amperage and voltage to guard against overload.
   - Fan rotation.
   - Operability of static pressure limit switch.
   - Automatic dampers for proper position.
   - Air and water controls operating to deliver required temperatures.
   - Air leaks in the casing and in the areas the coils and filter frames must be stopped. Note points where piping enters the casing to ensure that escutcheons are right. Do not rely on pipe insulation to seal these openings (insulation may shrink). In prefabricated units, check that all panel-fastening holes are filled to prevent whistling.
2. Traverse the main supply ductwork whenever possible. All main branches should also be traversed where duct arrangement permits. Traverse points and method of traverse should be selected as follows:
   - Traverse each main or branch after the longest possible straight run for the duct involved.
   - For test hole spacing, refer to Chapter 14 of the *ASHRAE Handbook—Fundamentals*.
   - Traverse using a pitot tube and manometer where velocities are over 600 fpm. Below this velocity, use either a micromanometer and pitot tube or an electronic multimeter and a pitot tube.
   - Note temperature and barometric pressure and correct for standard air quantity if needed.
   - After establishing the total air being delivered, adjust fan speed to obtain design airflow, if necessary. Check power and speed to confirm motor power and/or critical fan speed are not exceeded.
   - Proportionally adjust branch dampers until each has the proper air volume.
   - With all dampers and registers in the system open and with supply, return, and exhaust blowers operating at or near design airflow, set the minimum outside and return air ratio. If duct traverse locations are not available, this can be done by measuring the mixture temperature in the return air, outside air louver, and the filter section. The mixture temperature may be approximated from Equation (1).

     The greater the temperature difference between hot and cold air, the easier it is to get accurate damper settings. Take the temperature at many points in a uniform traverse to be sure there is no stratification.

     After the minimum outside air damper has been set for the proper percentage of outside air, run another traverse of mixture temperatures and install baffling if variation from the average is more than 5%. Remember that stratified mixed-air temperatures vary greatly with outside temperature in cold weather, whereas return air temperature has only a minor effect.
3. Balance terminal outlets in each control zone in proportion to each other, as follows:
   - Once the preliminary fan quantity is set, proportion the terminal outlet balance from the outlets into the branches to the fan. Concentrate on proportioning the flow rather than the absolute quantity. As fan settings and branch dampers change, the outlet terminal quantities remain proportional. Branch dampers should be used for major adjusting and terminal dampers for trim or minor adjustment only. It may be necessary to install additional sub-branch dampers to decrease the use of terminal dampers that create objectionable noise.
   - Normally, several passes through the entire system are necessary to obtain proper outlet values.
   - The total tested outlet air quantity compared to duct traverse air quantities may indicate duct leakage.
   - With total design air established in the branches and at the outlets, (1) take new fan motor amperage readings, (2) find static pressure across the fan, (3) read and record static pressure across each component (intake, filters, coils, mixing dampers), and (4) take a final duct traverse.

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 air flow to force the air to the end of the system.

VARIABLE-VOLUME SYSTEMS

Many types of variable air volume (VAV) systems have been developed to conserve energy. They can be categorized as pressure-dependent or pressure-independent.

Pressure-dependent systems incorporate air terminal boxes with a thermostat signal controlling a damper actuator. The air volume to the space varies to maintain the space temperature; the air temperature supplied to the terminal boxes remains constant. The balance of this system constantly varies with loading changes; therefore, any balancing procedure will not produce repeatable data unless changes in system load are simulated by using the same configuration of thermostat settings each time the system is tested (i.e., the same terminal boxes are fixed in the minimum and maximum positions for the test).

Pressure-independent systems incorporate air terminal boxes with a thermostat signal used as a master control to open or close the damper actuator, and a velocity controller used as a sub-master control to maintain the maximum and minimum amounts of air to be supplied to the space. Air volume to the space varies to maintain the space temperature; air temperature supplied to the terminal remains constant. Care should be taken to verify the operating range of the damper actuator as it responds to the velocity controller to prevent dead bands or overlap of control in response to other system components (e.g., double-duct VAV, fan-powered boxes, retrofit systems). Care should also be taken to verify the action of the thermostat with regard to the damper position, as the velocity controller can change the control signal ratio or reverse the control signal.

In a pressure-dependent system, setting minimum airflows to the space (other than at no flow) is not suggested unless the terminal box has a normally closed damper and the manufacturer of the damper actuator provides adjustable mechanical stops. The pressure-independent system requires verifying that the velocity controller is operating properly; it can be adversely affected by inlet duct configurations (Griggs et al. 1990). The primary difference between the two systems is that the pressure-dependent system supplies a different amount of air to the space as pressure upstream of the terminal box changes. If the thermostats are not calibrated properly to meet the space load, zones may overcool or overheat. When zones overcool and receive greater amounts of supply air than required, they decrease the amount of air that can be supplied to overheated zones. The pressure-independent system is not affected by improper thermostat calibration in the same way as a pressure-dependent system, because minimum and maximum airflow limits may be set for each zone.

Static Control

Static control saves energy and prevents overpressurizing the duct system. The following procedures and equipment are some of the means used to control static pressure.

No Fan Volumetric Control. Sometimes referred to as "riding the fan curve," this type of control should be limited to systems with minimum airflows of 50% of peak design and forward-curved fans with flat pressure curves. Pressure and noise are potential problems, and this control is not energy-efficient.

System Bypass Control. As system pressure increases due to terminal boxes closing, a relief damper bypasses air back to the fan inlet. The economy of varied fan output is lost, and the relief damper is usually a major source of duct leakage and noise. The relief damper should be modulated to maintain a minimum duct static pressure.

Discharge Damper. Losses and noise should be considered.

Vortex Damper. Losses from inlet air conditions are a problem, and the vortex damper does not completely close. The minimum expected airflow should be evaluated.

Variable Inlet Cones. System loss can be a problem because the cone does not typically close completely. The minimum expected airflow should be evaluated.

Varying Fan Speed Mechanically. Slippage, loss of belts, cost of belt replacement, and the initial cost of components are concerns.

Variable Pitch-in-Motion Fans. Maintenance and preventing the fan from running in the stall condition must be evaluated.

Varying Fan Speed Electrically. Varying the voltage or frequency to the fan motor is usually the most efficient method. Some motor drives may cause electrical noise and affect other devices.

In controlling VAV fan systems, location of the static pressure sensors is critical and should be field-verified to give the most representative point of operation. After the terminal boxes have been proportioned, static pressure control can be verified by observing static pressure changes at the fan discharge and the static pressure sensor as load is simulated from maximum to minimum airflow (i.e., set all terminal boxes to balanced airflow conditions and determine whether any changes in static pressure occur by placing one terminal box at a time to minimum airflow, until all terminals are placed at the minimal airflow setting). The maximum to minimum air volume changes should be within the fan curve performance (speed or total pressure).

Diversity

Diversity may be used on a VAV system, assuming that the total airflow is lower by design and that all terminal boxes will never fully open at the same time. Duct leakage should be avoided. All ductwork upstream of the terminal box should be considered medium-pressure, whether in a low- or medium-pressure system.

A procedure to test the total air on the system should be established by setting terminal boxes to the zero or minimum position nearest the fan. During peak load conditions, care should be taken to verify that adequate pressure is available upstream of all terminal boxes to achieve design airflow to the spaces.

Outside Air Requirements

Maintaining a space under a slight positive or neutral pressure to atmosphere is difficult with all variable-volume systems. In most systems, the exhaust requirement for the space is constant; hence, outside air used to equal the exhaust air and meet minimum outside air requirements for building codes must also remain constant. Because of the location of the outside air intake and pressure changes, this does not usually happen. Outside air should enter the fan at a point of constant pressure (i.e., supply fan volume can be controlled by proportional static pressure control, which can control the return air fan volume). Makeup air fans can also be used for outside air control.

Return Air Fans

If return air fans are required in series with a supply fan, control and sizing of the fans is most important. Serious over- and underpressurization can occur, especially during economizer cycles.

Types of VAV Systems

Single-Duct VAV. This system uses a pressure-dependent or -independent terminal and usually has reheat at a minimal setting on the terminal unit, or a separate heating system.

Bypass. This system uses a pressure-dependent damper, which, on demand for heating, closes the damper to the space and opens to the return air plenum. Bypass sometimes uses a constant bypass airflow or a reduced amount of airflow bypassed to the return plenum in relation to the amount supplied to the space. No economic value can be obtained by varying fan speed with this system. A control problem can exist if any return air sensing is done to control a warm-up or cool-down cycle.

VAV Using Single-Duct VAV and Fan-Powered, Pressure-Dependent Terminals. This system has a primary source of air from the fan to the terminal and a secondary powered fan source that pulls air from the return air plenum before the additional heat source. This system increases maintenance of terminal filters, motors, and capacitors. In some fan-powered boxes, backdraft dampers allow duct leakage when the system calls for the damper to be fully closed. Typical applications include geographic areas where the ratio of heating hours to cooling hours is low.

Double-Duct VAV. This type of terminal uses two single-duct variable terminals. It is controlled by velocity controllers that operate in sequence so that both hot and cold ducts can be opened or closed. Some controls have a downstream flow sensor in the terminal unit. The total-airflow sensor is in the inlet and controlled by the thermostat. As this inlet damper closes, the downstream controller opens the other damper to maintain set airflow. Low pressure in the decks controlled by the thermostat may cause unwanted mixing of air, which results in excess energy use or discomfort in the space.

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 control is used, obtain the fan manufacturer's data on 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 and (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.

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.

Report Information

To be of value to the consulting engineer and owner's maintenance department, the air-handling report should consist of at least the following items:

1. *Design*
   - Air quantity to be delivered
   - Fan static pressure
   - Motor power installed or required
   - Percent of outside air under minimum conditions
   - Fan speed
   - Input power required to obtain this air quantity at design static pressure
2. *Installation*
   - Equipment manufacturer (indicate model and serial numbers)
   - Size of unit installed
   - Arrangement of air-handling unit
   - Nameplate power and voltage, phase, cycles, and full-load amperes of installed motor
3. *Field tests*
   - Fan speed
   - Power readings (voltage, amperes of all phases at motor terminals)
   - Total pressure differential across unit components
   - Fan suction and fan discharge static pressure (equals fan total pressure)
   - Plot of actual readings on manufacturer's fan performance curve to show the installed fan operating point
   - Measured airflow rate

   It is important to establish the initial static pressures accurately for the air treatment equipment and the duct system so that the variation in air quantity due to filter loading can be calculated. It enables the designer to ensure that the total air quantity will never be less than the minimum requirements. Because the design air quantity for peak loading of the filters has already been calculated, it also serves as a check of dirt loading in coils.
4. *Terminal Outlets*
   - Outlet by room designation and position
   - Manufacture and type
   - Size (using manufacturer's designation to ensure proper factor)
   - Manufacturer's outlet factor (where no factors are available, or field tests indicate listed factors are incorrect, a factor must be determined in the field by traverse of a duct leading to a single outlet)
   - Design air quantity and velocity required to obtain it
   - Test velocities and resulting air quantity
   - Adjustment pattern for every air terminal
5. *Additional Information (if applicable)*
   - Air-handling units
     - Belt number and size
     - Drive and driven sheave size
     - Belt position on adjusted drive sheaves (bottom, middle, and top)
     - Motor speed under full load
     - Motor heater size
     - Filter type and static pressure at initial use and full load; time to replace
     - Variations of velocity at various points across face of coil
     - Existence of vortex or discharge dampers, or both
   - Distribution system
     - Unusual duct arrangements
     - Branch duct static readings in double-duct and induction system
     - Ceiling pressure readings where plenum ceiling distribution is used; tightness of ceiling
     - Relationship of building to outside pressure under both minimum and maximum outside air
     - Induction unit manufacturer and size (including required air quantity and plenum pressures for each unit) and test plenum pressure and resulting primary air delivery from manufacturer's listed curves
   - All equipment nameplates visible and easily readable

Many independent firms have developed detailed procedures suitable to their own operations and the area in which they function. These procedures are often available for information and evaluation on request.

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. Air-side 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 to less than design reduces heat transfer directly and linearly with respect to flow. In contrast, the heat transfer rate for the water side does not vary linearly with the water flow rate through a heat exchanger (coil), because of the characteristics of heat exchangers. As a result, water-side 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°F Δt) 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 air-side coefficient; a change in internal or water-side 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.

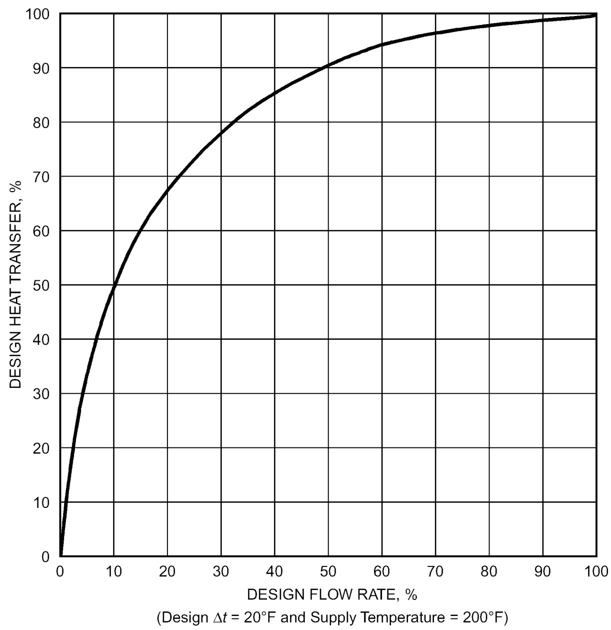

Fig. 1 Effects of Flow Variation on Heat Transfer from a Hydronic Terminal

Tests of hydronic coil performance show that when flow is throttled to the coil, the water-side 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 water-side 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 terminals 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 (β = 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 contracta 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.

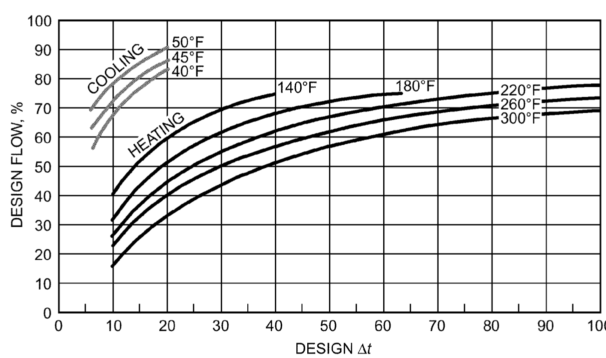

Fig. 2 Percent of Design Flow Versus Design Δt to Maintain 90% Terminal Heat Transfer for Various Supply Water Temperatures

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 Δt. In coils with larger water-side design Δt, heat transfer can increase.

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 a 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.

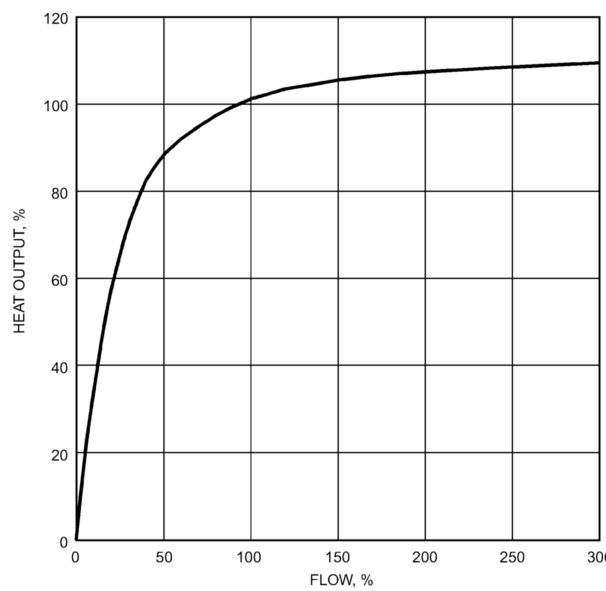

Fig. 3 Typical Heating Coil Heat Transfer Versus Water Flow

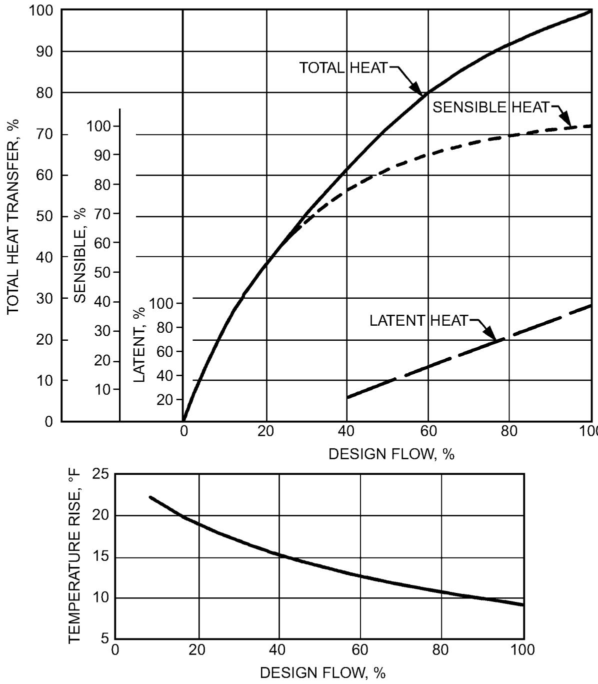

Fig. 4 Chilled Water Terminal Heat Transfer Versus Flow

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

Water-side 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 water side 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 water-side 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:

- Flowmeters (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 C_v, a flow versus handle position and pressure drop table, or a slide rule flow calculator
- Dynamic balancing valves or flow-limiting valves (for prebalanced systems only); field adjustment of these devices is not normally required or possible (Chapter 46, Design and Application of Controls)
- Pumps with factory-certified pump curves
- Components used as flowmeters (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 ΔP 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 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 C_v for a balancing valve or to size a control valve.

The flow coefficient C_v 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

$$C_v = Q\sqrt{s_f/\Delta p} \tag{2}$$

where

C_v = flow coefficient at 1 psi drop
Q = design flow for terminal or valve, gpm
Δp = pressure drop, psi
Δh = pressure drop, ft of water
s_f = specific gravity of fluid

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

$$C_v = 1.5Q\sqrt{s_f/\Delta h} \tag{3}$$

HYDRONIC BALANCING METHODS

Various techniques are used to balance hydronic systems. Balance by temperature difference and water balance by proportional method are 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 overload protection information
3. Control valve C_v 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. Flowmeter 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 revent. Retest and revent 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 flowmeter 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.

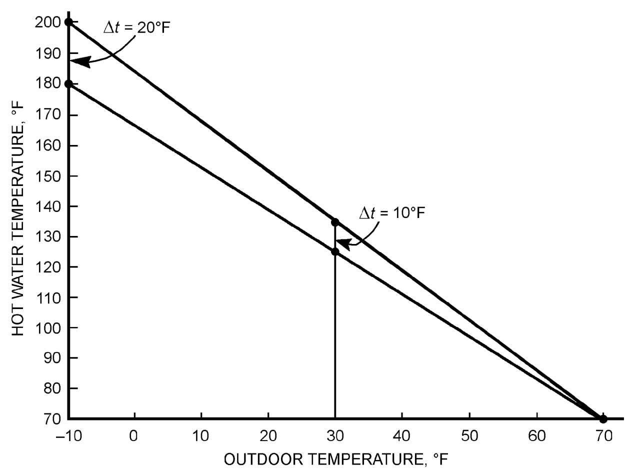

Fig. 5 Water Temperature Versus Outside Temperature Showing Approximate Temperature Difference

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 water-side 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:

$$\text{Flow quotient} = \frac{\text{Actual flow rate}}{\text{Design flow rate}} \tag{4}$$

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 water side 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 to change. 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 2 fps, which may alter the heat transfer characteristics of the system.

Other Balancing Techniques

Flow Balancing by Rated Differential Procedure. This procedure depends on deriving a performance curve for the test coil, comparing water temperature difference Δt_w to entering water temperature t_{ew} minus entering air temperature t_{ea}. One point of the desired curve can be determined from manufacturer's ratings, which are published as $(t_{ew} - t_{ea})$. A second point is established by observing that the heat transfer from air to water is zero when $(t_{ew} - t_{ea})$ is zero (consequently, $\Delta t_w = 0$). With these two points, an approximate

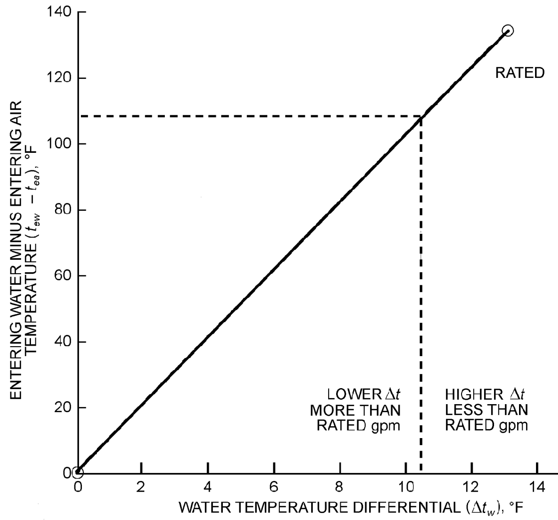

Fig. 6 Coil Performance Curve

performance curve can be drawn (see Figure 6). Then, for any other ($t_{ew} - t_{ea}$), this curve is used to determine the appropriate Δt_w. The basic curve applies to catalog ratings for lower dry-bulb temperatures, providing a 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. The curve may also be used for cooling coils for sensible transfer (dry coil).

Flow Balancing by Total Heat Transfer. This procedure determines water flow by running an energy balance around the coil. From field measurements of airflow, wet- and dry-bulb temperatures up- and downstream of the coil, and the difference Δt_w between entering and leaving water temperatures, water flow can be determined by the following equations:

$$Q_w = q/500\,\Delta t_w \tag{5}$$

$$q_{cooling} = 4.5\,Q_a(h_1 - h_2) \tag{6}$$

$$q_{heating} = 1.08\,Q_a(t_1 - t_2) \tag{7}$$

where

Q_w = water flow rate, gpm
q = load, Btu/h
$q_{cooling}$ = cooling load, Btu/h
$q_{heating}$ = heating load, Btu/h
Q_a = airflow rate, cfm
h = enthalpy, Btu/lb
t = temperature, °F

Example 1. Find the water flow for a cooling system with the following characteristics:

Test data

t_{ewb} = entering wet-bulb temperature = 68.5°F
t_{lwb} = leaving wet-bulb temperature = 53.5°F
Q_a = airflow rate = 22,000 cfm
t_{lw} = leaving water temperature = 59.0°F
t_{ew} = entering water temperature = 47.5°F

From psychrometric chart

h_1 = 32.84 Btu/lb
h_2 = 22.32 Btu/lb

Solution: From Equations (5) and (6),

$$Q_w = \frac{4.5 \times 22{,}000(32.84 - 22.32)}{500(59.0 - 47.5)} = 181 \text{ gpm}$$

The desired water flow is achieved by successive manual adjustments and recalculations. Note that these temperatures can be greatly influenced by the heat of compression, stratification, bypassing, and duct leakage.

General Balance Procedures

All the variations of balancing hydronic systems cannot be listed; however, the general method should balance the system and minimize operating cost. Excess pump pressure (operating power) can be eliminated by trimming the pump impeller. Allowing excess pressure to be absorbed by throttle valves adds a lifelong operating-cost penalty to the operation.

The following is a general procedure based on setting the balance valves on the site:

1. Develop a flow diagram if one is not included in the design drawings. Illustrate all balance instrumentation, and include any additional instrument requirements.
2. Compare pumps, primary heat exchangers, and specified terminal units, and determine whether a design diversity factor can be achieved.
3. Examine the control diagram and determine the control adjustments needed to obtain design flow conditions.

Balance Procedure—Primary and Secondary Circuits

1. Inspect the system completely to ensure that (1) it has been flushed out, it is clean, and all air is removed; (2) all manual valves are open or in operating position; (3) all automatic valves are in their proper positions and operative; and (4) the expansion tank is properly charged.
2. Place controls in position for design flow.
3. Examine the flow diagram and piping for obvious short circuits; check flow and adjust the balance valve.
4. Take pump suction, discharge, and differential pressure readings at both full and no flow. For larger pumps, a no-flow condition may not be safe. In any event, valves should be closed slowly.
5. Read pump motor amperage and voltage, and determine approximate power.
6. Establish a pump curve, and determine approximate flow rate.
7. If a total flow station exists, determine the flow and compare with pump curve flow.
8. If possible, set the total flow about 10% high using the total flow station first and the pump differential pressure second; then maintain pumped flow at a constant value as balance proceeds by adjusting the pump throttle valve.
9. Any branch main flow stations should be tested and set, starting by setting the shortest runs low as balancing proceeds to the longer branch runs.
10. With primary and secondary pumping circuits, a reasonable balance must be obtained in the primary loop before the secondary loop can be considered. The secondary pumps must be running and terminal units must be open to flow when the primary loop is being balanced, unless the secondary loop is decoupled.

FLUID FLOW MEASUREMENT

Flow Measurement Based on Manufacturer's Data

Any component (terminal, control valve, or chiller) that has an accurate, factory-certified flow/pressure drop relationship can be used as a flow-indicating device. The flow and pressure drop may be used to establish an equivalent flow coefficient as shown in Equation (3). According to the Bernoulli equation, pressure drop varies as the square of the velocity or flow rate, assuming density is constant:

$$Q_1^2/Q_2^2 = \Delta h_1/\Delta h_2 \tag{8}$$

For example, a chiller has a certified pressure drop of 25 ft of water at 100 gpm. The calculated flow with a field-measured pressure drop of 30 ft is

$$Q_2 = 100\sqrt{30/25} = 109.5 \text{ gpm}$$

Flow calculated in this manner is only an estimate. The accuracy of components used as flow indicators depends on (1) the accuracy of cataloged information concerning flow/pressure drop relationships and (2) the accuracy of the pressure differential readings. As a rule, the component should be factory-certified flow tested if it is to be used as a flow indicator.

Pressure Differential Readout by Gage

Gages are used to read differential pressures. Gages are usually used for high differential pressures and manometers for lower differentials. Accurate gage readout is diminished when two gages are used, especially when the gages are permanently mounted and, as such, subject to malfunction.

A single high-quality gage should be used for differential readout (Figure 7). This gage should be alternately valved to the high- and low-pressure side to establish the differential. A single gage needs no static height correction, and errors caused by gage calibration are eliminated.

Differential pressure can also be read from differential gages, thus eliminating the need to subtract outlet from inlet pressures to establish differential pressure. Differential pressure gages are usually dual gages mechanically linked to read differential pressure. The differential pressure gage readout can be stated in terms of psi or in feet of head of 60°F water.

Conversion of Differential Pressure to Head

Pressure gage readings can be restated to fluid head, which is a function of fluid density. The common hydronic system conversion factor is related to water density at about 60°F; 1 psi equals 2.31 ft. Pressure gages can be calibrated to feet of water head using this conversion. Because the calibration only applies to water at 60°F, the readout may require correction when the gage is applied to water at a significantly higher temperature.

Pressure gage conversion and correction factors for various fluid specific gravities (in relation to water at 60°F) are shown in Table 2. The differential gage readout should only be defined in terms of the head of the fluid actually causing the flow pressure differential. When this is done, the resultant fluid head can be applied to the C_v to determine actual flow through any flow device, provided the manufacturer has correctly stated the flow to fluid head relationship.

For example, a manufacturer may test a boiler or control valve with 100°F water. If the test differential pressure is converted to head at 100°F, a C_v independent of test temperature and density may be calculated. Differential pressures from another test made in the field at 250°F may be converted to head at 250°F. The C_v calculated with this head is also independent of temperature. The manufacturer's data can then be directly correlated with the field test to establish flow rate at 250°F.

A density correction must be made to the gage reading when differential heads are used to estimate pump flows as in Figure 8. This is because of the shape of the pump curve. An incorrect head difference entry into the curve due to an uncorrected gage reading can cause a major error in the estimated pumped flow. In this case, the gage reading for a pumped liquid that has a specific gravity of 0.9 (2.57 ft liquid/psi) was not corrected; the gage conversion is assumed to be 2.31 ft liquid/psi. A 50% error in flow estimation is shown.

Differential Head Readout with Manometers

Manometers are used for differential pressure readout, especially when very low differentials, great precision, or both, are required. But manometers must be handled with care; they should not be used for field testing because fluid could blow out into the water and rapidly deteriorate the components. A proposed manometer arrangement is shown in Figure 9.

Figure 9 and the following instructions provide accurate manometer readings with minimum risk of blowout.

1. Make sure that both legs of the manometer are filled with water.
2. Open the purge bypass valve.
3. Open valved connections to high and low pressure.

Table 2 Differential Pressure Conversion to Head

Fluid Specific Gravity	Corresponding Water Temperature, °F	Foot Fluid Head Equal to 1 psi[a]	Correction Factor When Gage is Stated to Feet of Water (60°F)[b]
1.5		1.54	
1.4		1.65	
1.3		1.78	
1.2		1.93	
1.1		2.10	
1.0	60	2.31	1.00
0.98	150	2.36	1.02
0.96	200	2.41	1.04
0.94	250	2.46	1.065
0.92	300	2.51	1.09
0.90	340	2.57	1.11
0.80		2.89	
0.70		3.30	
0.60		3.85	
0.50		4.63	

[a]Differential psi readout is multiplied by this number to obtain feet fluid head when gage is calibrated in psi.
[b]Differential feet water head readout is multiplied by this number to obtain feet fluid head when gage calibration is stated to feet head of 60°F water.

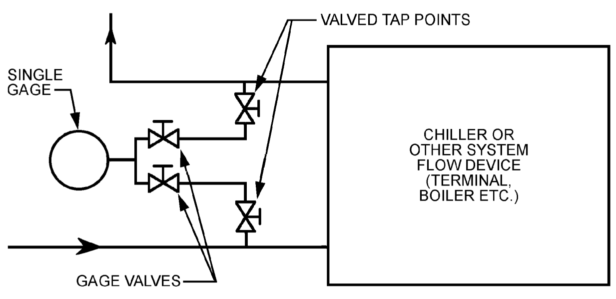

Fig. 7 Single Gage for Reading Differential Pressure

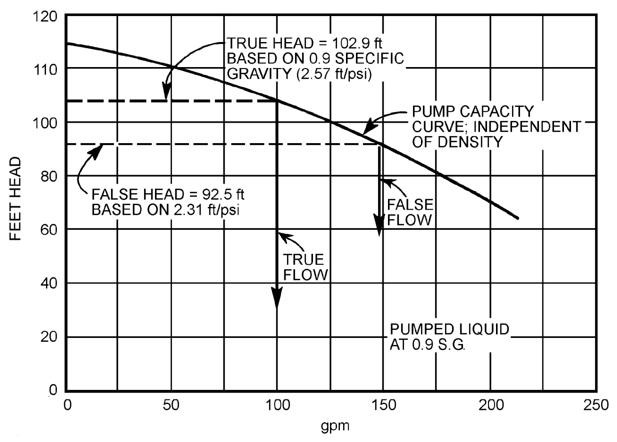

Fig. 8 Fluid Density Correction Chart for Pump Curves

4. Open the bypass vent valve slowly and purge air here.
5. Open manometer block vents and purge air at each point.
6. Close the needle valves. The columns should zero in if the manometer is free of air. If not, vent again.
7. Open the needle valves and begin throttling the purge bypass valve slowly, watching the fluid columns. If the manometer has an adequate available fluid column, the valve can be closed and the differential reading taken. However, if the fluid column reaches the top of the manometer before the valve is completely closed, insufficient manometer height is indicated and further throttling will blow fluid into the blowout collector. A longer manometer or the single gage readout method should then be used.

An error is often introduced when converting inches of gage fluid to feet of the test fluid. The conversion factor changes with test fluid temperature, density, or both. Conversion factors shown in Table 2 are to a water base, and the counterbalancing water height H (Figure 9) is at room temperature.

Orifice Plates, Venturi, and Flow Indicators

Manufacturers provide flow information for several devices used in hydronic system balance. In general, the devices can be classified as (1) orifice flowmeters, (2) venturi flowmeters, (3) velocity impact meters, (4) pitot-tube flowmeters, (5) bypass spring impact flowmeters, (6) calibrated balance valves, (7) turbine flowmeters, and (8) ultrasonic flowmeters.

The **orifice flowmeter** is widely used and is extremely accurate. The meter is calibrated and shows differential pressure versus flow. Accuracy generally increases as the pressure differential across the meter increases. The differential pressure readout instrument may be a manometer, differential gage, or single gage (Figure 7).

The **venturi flowmeter** has lower pressure loss than the orifice plate meter because a carefully formed flow path increases velocity head recovery. The venturi flowmeter is placed in a main flow line where it can be read continuously.

Velocity impact meters have precise construction and calibration. The meters are generally made of specially contoured glass or plastic, which permits observation of a flow float. As flow increases, the flow float rises in the calibrated tube to indicate flow rate. Velocity impact meters generally have high accuracy.

A special version of the velocity impact meter is applied to hydronic systems. This version operates on the velocity head difference between the pipe side wall and the pipe center, which causes fluid to flow through a small flowmeter. Accuracy depends on the location of the impact tube and on a velocity profile that corresponds to theory and the laboratory test calibration base. Generally, the accuracy of this **bypass flow impact** or differential velocity head flowmeter is less than a flow-through meter, which can operate without creating a pressure loss in the hydronic system.

The **pitot-tube flowmeter** is also used for pipe flow measurement. Manometers are generally used to measure velocity head differences because these differences are low.

The **bypass spring impact flowmeter** uses a defined piping pressure drop to cause a correlated bypass side branch flow. The side branch flow pushes against a spring that increases in length with increased side branch flow. Each individual flowmeter is calibrated to relate extended spring length position to main flow. The bypass spring impact flowmeter has, as its principal merit, a direct readout. However, dirt on the spring reduces accuracy. The bypass is opened only when a reading is made. Flow readings can be taken at any time.

The **calibrated balance valve** is an adjustable orifice flowmeter. Balance valves can be calibrated so that a flow/pressure drop relationship can be obtained for each incremental setting of the valve. A ball, rotating plug, or butterfly valve may have its setting expressed in percent open or degree open; a globe valve, in percent open or number of turns. The calibrated balance valve must be manufactured with precision and care to ensure that each valve of a particular size has the same calibration characteristics.

The **turbine flowmeter** is a mechanical device. The velocity of the liquid spins a wheel in the meter, which generates a 4 to 20 mA output that may be calibrated in units of flow. The meter must be well maintained, as wear or water impurities on the bearing may slow the wheel, and debris may clog or break the wheel.

The **ultrasonic flowmeter** senses sound signals, which are calibrated in units of flow. The ultrasonic metering station may be installed as part of the piping, or it may be a strap-on meter. In either case, the meter has no moving parts to maintain, nor does it intrude into the pipe and cause a pressure drop. Two distinct types of ultrasonic meter are available: (1) the transit time meter for HVAC or clear water systems and (2) the Doppler meter for systems handling sewage or large amounts of particulate matter.

If any of the above meters are to be useful, the minimum distance of straight pipe upstream and downstream, as recommended by the meter manufacturer and flow measurement handbooks, must be adhered to. Figure 10 presents minimum installation suggestions.

Using a Pump as an Indicator

Although the pump is not a meter, it can be used as an indicator of flow together with the other system components. Differential pressure readings across a pump can be correlated with the pump curve to establish the pump flow rate. Accuracy depends on (1) accuracy of readout, (2) pump curve shape, (3) actual conformance of the pump to its published curve, (4) pump operation without cavitation, (5) air-free operation, and (6) velocity head correction.

When a differential pressure reading must be taken, a single gage with manifold provides the greatest accuracy (Figure 11). The pump suction to discharge differential can be used to establish pump differential pressure and, consequently, pump flow rate. The single gage and manifold may also be used to check for strainer clogging by measuring the pressure differential across the strainer.

If the pump curve is based on fluid head, pressure differential, as obtained from the gage reading, needs to be converted to head, which is pressure divided by the fluid weight per cubic foot. The pump differential head is then used to determine pump flow rate

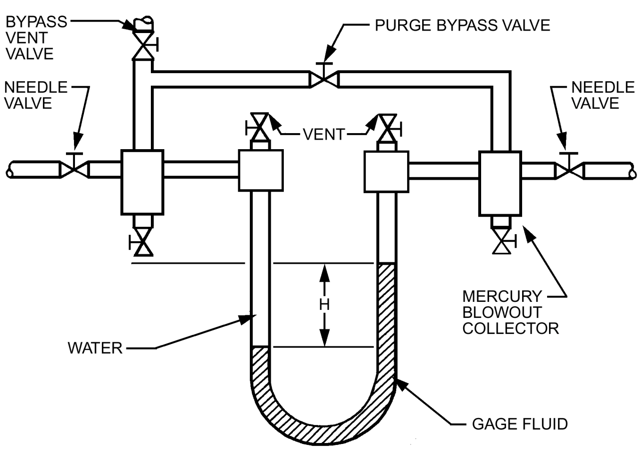

Fig. 9 Fluid Manometer Arrangement for Accurate Reading and Blowout Protection

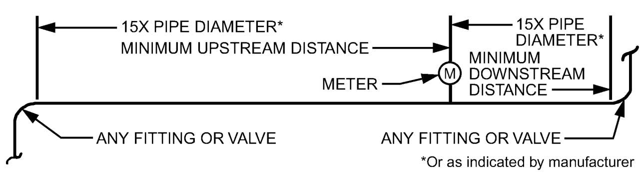

Fig. 10 Minimum Installation Dimensions for Flowmeter

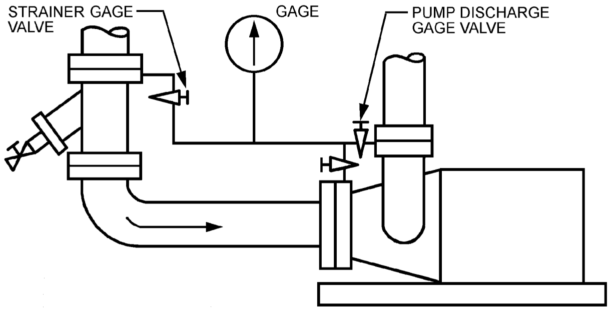

Fig. 11 Single Gage for Differential Readout Across Pump and Strainer

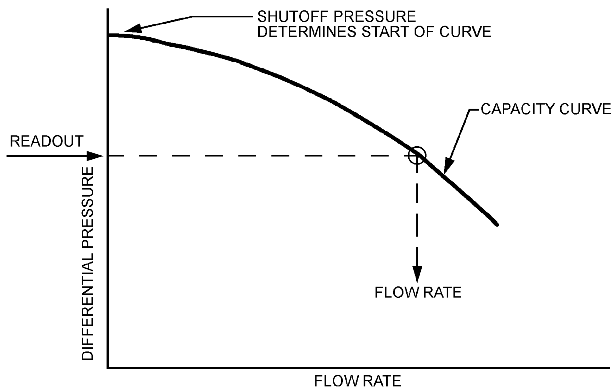

Fig. 12 Differential Pressure Used to Determine Pump Flow

(Figure 12). As long as the differential head used to enter the pump curve is expressed as head of the fluid being pumped, the pump curve shown by the manufacturer should be used as described. The pump curve may state that it was defined by test with 85°F water. This is unimportant, since the same curve applies from 60 to 250°F water, or to any fluid within a broad viscosity range.

Generally, pump-derived flow information, as established by the performance curve, is questionable unless the following precautions are observed:

1. The installed pump should be factory calibrated by a test to establish the actual flow-pressure relationship for that particular pump. Production pumps can vary from the cataloged curve because of minor changes in impeller diameter, interior casting tolerances, and machine fits.
2. When a calibration curve is not available for a centrifugal pump being tested, the discharge valve can be closed briefly to establish the no-flow shutoff pressure, which can be compared to the published curve. If the shutoff pressure differs from that published, draw a new curve parallel to the published curve. While not exact, the new curve will usually fit the actual pumping circumstance more accurately. Clearance between the impeller and casing minimize the danger of damage to the pump during a no-flow test, but manufacturer verification is necessary.
3. Differential head should be determined as accurately as possible, especially for pumps with flat flow curves.
4. The pump should be operating air-free and without cavitation. A cavitating pump will not operate to its curve, and differential readings will provide false results.
5. Ensure that the pump is operating above the minimum net positive suction head.
6. Power readings can be used (1) as a check for the operating point when the pump curve is flat or (2) as a reference check when there is suspicion that the pump is cavitating or providing false readings because of air.

Table 3 Instruments for Monitoring a Water System

Point of Information	Manifold Gage	Single Gage	Thermometer	Test Well	Pressure Tap
Pump—Suction, discharge	x				
Strainer—In, out					x
Cooler—In, out		x	x		
Condensers—In, out		x	x		
Concentrator—In, out		x	x		
Absorber—In, out		x	x		
Tower cell—In, out				x	x
Heat exchanger—In, out	x		x		
Coil—In, out				x	x
Coil bank—In, out		x	x		
Booster coil—In, out					x
Cool panel—In, out					x
Heat panel—In, out				x	x
Unit heater—In, out					x
Induction—In, out					x
Fan coil—In, out					x
Water boiler—In, out			x		
Three-way valve—All ports					x
Zone return main			x		
Bridge—In, out			x		
Water makeup		x			
Expansion tank		x			
Strainer pump					x
Strainer main	x				
Zone three-way—All ports				x	x

7. The flow determined by the pump curve should be compared to the flow measured at the flowmeters, flow measured by pressure drops through circuits, and flow measured by pressure drops through exchangers.
8. The pump flow derived from the pressure differential at the suction and discharge connections is only an indicator of the actual flow—it cannot be used to verify the test and balance measurements. If the pump flow is to be used for balancing verification it needs to be determined using the Hydraulic Institute procedure or by measuring the flow through a properly installed metering station 15 to 20 straight pipe diameters downstream from the pump discharge.

The power draw should be measured in watts. Ampere readings cannot be trusted because of voltage and power factor problems. If motor efficiency is known, the wattage drawn can be related to pump brake power (as described on the pump curve) and the operating point determined.

Central Plant Chilled Water Systems

For existing installations, establishing accurate thermal load profiles is of prime importance because it establishes proper primary chilled water supply temperature and flow. In new installations, actual load profiles can be compared with design load profiles to obtain valid operating data.

To perform proper testing and balancing, all interconnecting points between the primary and secondary systems must be designed with sufficient temperature, pressure, and flow connections so that adequate data may be indicated and/or recorded.

Water Flow Instruments

As indicated previously, proper location and use of instruments is vital to accurate balancing. Instruments for testing temperature and pressure at various locations are listed in Table 3. Flow-indicating devices should be placed in water systems as follows:

- At each major heating coil bank (10 gpm or more)
- At each major cooling coil bank (10 gpm or more)
- At each bridge in primary-secondary systems
- At each main pumping station
- At each water chiller evaporator

- At each water chiller condenser
- At each water boiler outlet
- At each floor takeoff to booster reheat coils, fan coil units, induction units, ceiling panels, and radiation (Do not exceed 25 terminals off of any one zone meter probe.)
- At each vertical riser to fan coil units, induction units, and radiation
- At the point of tie-in to existing systems

STEAM DISTRIBUTION

Procedures for Steam Balancing Variable Flow Systems

Steam distribution cannot be balanced by adjustable flow-regulating devices. Instead, fixed restrictions built into the piping in accordance with carefully designed pipe and orifice sizes are used to regulate flow.

It is important to have a balanced distribution of steam to all portions of the steam piping at all loads. This is best accomplished by properly designing the steam distribution piping, which includes carefully considering steam pressure, steam quantities required by each branch circuit, pressure drops, steam velocities, and pipe sizes. Just as other flow systems are balanced, steam distribution systems are balanced by ensuring that the pressure drops are equalized at design flow rates for all portions of the piping. Only marginal balancing can be done by pipe sizing. Therefore, additional steps must be taken to achieve a balanced performance.

Steam flow balance can be improved by using spring-type packless supply valves equipped with precalibrated orifices. The valves should have a tight shutoff between 25 in. of Hg and 60 psig. These valves have a nonrising stem, are available with a lockshield, and have a replaceable disk. Orifice flanges can also be used to regulate and measure steam flow at appropriate locations throughout the system. The orifice sizes are determined by the pressure drop required for a given flow rate at a given location. A schedule should be prepared showing (1) orifice sizes, (2) valve or pipe sizes, (3) required flow rates, and (4) corresponding pressure differentials for each flow rate. It may be useful to calculate pressure differentials for several flow rates for each orifice size. Such a schedule should be maintained for future reference.

After the appropriate regulating orifices are installed in the proper locations, the system should be tested for tightness by sealing all openings in the system and applying a vacuum of 20 in. of Hg, held for 2 hours. Next, the system should be readied for warm-up and pressurizing with steam following the procedures outlined in Section VI of the ASME *Boiler and Pressure Vessel Code*. After the initial warm-up and system pressurization, evaluate system steam flow, and compare it to system requirements. The orifice schedule calculated earlier will now be of value should any of the orifices need to be changed.

Steam Flow Measuring Devices

Many devices are available for measuring flow in steam piping: (1) steam meters, (2) condensate meters, (3) orifice plates, (4) venturi fittings, (5) steam recorders, and (6) manometers for reading differential pressures across orifice plates and venturi fittings. Some of these devices are permanently affixed to the piping system to facilitate taking instantaneous readings that may be necessary for proper operation and control. A surface pyrometer used in conjunction with a pressure gage is a convenient way to determine steam saturation temperature and the degree of superheat at various locations in the system. Such information can be used to evaluate performance characteristics.

COOLING TOWERS

Field-testing cooling towers is demanding and difficult. ASME *Standard* PTC 23 and CTI *Standard Specification* ATC-105 establish procedures for these tests. Certain general guidelines for testing cooling towers are as follows:

Conditions at Time of Test

- Water flow within 15% of design
- Heat load within 30% of design and stabilized
- Entering wet bulb within 12°F of design

Using these limitations and field readings that are as accurate as possible, a projection to design conditions produces an accuracy of ±5% of tower performance.

Conditions for Performing Test

- Water-circulating systems serving tower should be thoroughly cleaned of all dirt and foreign matter. Samples of water should be clear and indicate clear passage of water through pumps, piping, screens, and strainers.
- Fans serving cooling tower should operate in proper rotation. Foreign obstructions should be removed. Permanent obstruction should be noted.
- Interior filling of cooling tower should be clean and free of foreign materials such as scale, algae, or tar.
- Water level in tower basin should be maintained at proper level. Visually check basin sump during full flow to verify that centrifugal action of the water is not causing entrainment of air, which could cause pump cavitation.
- Water-circulating pumps should be tested with full flow through tower. If flow exceeds design, it should be valved down until design is reached. The flow that is finally set should be maintained throughout the test period. All valves, except necessary balancing valves, should be fully open.
- If makeup and blowdown have facilities to determine flow, set them to design flow at full flow through the tower. If flow cannot be determined, shut off both.

Instruments

Testing and balancing agencies provide instruments to perform the required tests. Mechanical contractors provide and install all components such as orifice plates, venturis, balancing valves, thermometer wells, gage cocks, and corporation cocks. Designers specify measuring point locations.

Instruments used should be recently calibrated as follows:

Temperature

- Thermometer with divisions of 0.2°F in a proper well for water should be used.
- Thermometer with solar shield and 0.2°F divisions or thermocouple with 0.2°F readings having mechanical aspiration for wet-bulb readings should be used.
- Sling psychrometer may be used for rough checks.
- Thermometer with 0.2°F divisions should be used for dry-bulb readings.

Water Flow

- Orifice, venturi, and balancing valve pressure drops can be read using a manometer or recently calibrated differential pressure gage.
- Where corporation cocks are installed, a pitot tube and manometer traverse can be made by trained technicians.

Test Method

1. Conduct water flow tests to determine volume of water in tower, and of makeup and blowdown water.
2. Conduct water temperature tests, if possible, in suitable wells as close to tower as possible. Temperature readings at pumps or condensing element are not acceptable in tower evaluation. If there are no wells, surface pyrometer readings are acceptable.
3. Take makeup water volume and temperature readings at point of entry into tower.
4. Take blowdown volume and temperature readings at point of discharge from tower.
5. Take inlet and outlet dry- and wet-bulb temperature readings using prescribed instruments.

- Use wet-bulb entering and leaving temperatures to determine tower actual performance against design.
- Use wet- and dry-bulb entering and leaving temperatures to determine evaporation involved.

6. If tower has a ducted inlet or outlet where a reasonable duct traverse can be made, use this air volume as a cross-check of tower performance.
7. Take wet- and dry-bulb temperature readings 3 to 5 ft from the tower on all inlet sides, halfway between the base and the top of inlet louvers at no more than 5 ft spacing horizontally the readings. Note any unusual inlet conditions.
8. Note wind velocity and direction at time of test.
9. Take test readings continually with a minimum time lapse between readings.
10. If the first test indicates a tower deficiency, perform two additional tests to verify the original readings.

TEMPERATURE CONTROL VERIFICATION

The test and balance technician should work closely with the temperature control installer to ensure that the project is completed correctly. The balancing technician needs to verify proper operation of the control and communicate findings back to the agency responsible for ensuring that the controls have been installed correctly. This is usually the HVAC system designer, although others may be involved. Generally, the balancing technician does not adjust, relocate, or calibrate the controls. However, this is not always the case, and differences do occur with VAV terminal unit controllers. The balancing technician should be familiar with the specifications and design intent of the project so that all responsibilities are understood.

During the design and specification phase of the project, the designer should specify verification procedures for the controls and responsibilities for the contractor who installs the temperature controls. It is important that the designer specify (1) the degree of coordination between the installer of the control and the balancing technician and (2) the testing responsibilities of each.

Verification of control operation starts with the balancing technician reviewing the submitted documents and shop drawings of the control system. In some cases the controls technician should instruct the balancing technician in the operation of certain control elements, such as digital terminal unit controllers. This is followed by schedule coordination between the control and balancing technicians. In addition, the balancing and controls technicians need to work together when reviewing the operation of some sections of the HVAC system, particularly with VAV systems and the setting of the flow measurement parameters in digital terminal unit controllers.

Major mechanical systems should be verified after testing, adjusting, and balancing is completed. The control system should be operated in stages to prove it can match system capacity to varying load conditions. Mechanical subsystem controllers should be verified when balancing data are collected, considering that the entire system may not be completely functional at the time of verification. Testing and verification should account for seasonal variations; tests should be performed under varying outside loads to ensure operational performance. Retesting a random sample of terminal units may be desirable to verify the control technician's work.

Suggested Procedures

The following verification procedures may be used with either pneumatic or electrical controls:

1. Obtain design drawings and documentation, and become well acquainted with the design intent and specified responsibilities.
2. Obtain copies of approved control shop drawings.
3. Compare design to installed field equipment.
4. Obtain recommended operating and test procedures from manufacturers.
5. Verify with the control contractor that all controllers are calibrated and commissioned.
6. Check location of transmitters and controllers. Note adverse conditions that would affect control, and suggest relocation as necessary.
7. Note settings on controllers. Note discrepancies between set point for controller and actual measured variable.
8. Verify operation of all limiting controllers, positioners, and relays (e.g., high- and low-temperature thermostats, high- and low-differential pressure switches, etc.).
9. Activate controlled devices, checking for free travel and proper operation of stroke for both dampers and valves. Verify normally open (NO) or normally closed (NC) operation.
10. Verify sequence of operation of controlled devices. Note line pressures and controlled device positions. Correlate to air or water flow measurements. Note speed of response to step change.
11. Confirm interaction of electrically operated switch transducers.
12. Confirm interaction of interlock and lockout systems.
13. Coordinate balancing and control technicians' schedules to avoid duplication of work and testing errors.

Pneumatic System Modifications

1. Verify main control supply air pressure and observe compressor and dryer operation.
2. For hybrid systems using electronic transducers for pneumatic actuation, modify procedures accordingly.

Electronic Systems Modifications

1. Monitor voltages of power supply and controller output. Determine whether the system operates on a grounded or nongrounded power supply, and check condition. Although electronic controls now have more robust electronic circuits, improper grounding can cause functional variation in controller and actuator performance from system to system.
2. Note operation of electric actuators using spring return. Generally, actuators should be under control and use springs only upon power failure to return to a fail-safe position.

Direct Digital Controllers

Direct digital control (DDC) offers nontraditional challenges to the balancing technician. Many control devices, such as sensors and actuators, are the same as those in electronic and pneumatic systems. Currently DDC is dominated by two types of controllers: fully programmable or application-specific. Fully programmable controllers offer a group of functions linked together in an applications program to control a system such as an air-handling unit. Application-specific controllers are functionally defined with the programming necessary to carry out the functions required for a system, but not all adjustments and settings are defined. Both types of controllers and their functions have some variations. One of the functions is adaptive control, which includes control algorithms that automatically adjust settings of various controller functions.

The balancing technician must understand controller functions so that they do not interfere with the test and balance functions. Literacy in computer programming is not necessary, although it does help. When testing the DDC,

1. Obtain controller application program. Discuss application of the designer's sequence with the control programmer.
2. Coordinate testing and adjustment of controlled systems with mechanical systems testing. Avoid duplication of efforts between technicians.
3. Coordinate storage (e.g., saving to central DDC database and controller memory) of all required system adjustments with control technician.

In cases where the balancing agency is required to test discrete points in the control system,

1. Establish criteria for test with the designer.

2. Use reference standards that test the end device through the entire controller chain (e.g., device, wiring, controller, communications, and operator monitoring device). An example would be using a dry block temperature calibrator (a testing device that allows a temperature to be set, monitored, and maintained in a small chamber) to test a space temperature sensor. The sensor is installed with extra wire so that it may be removed from the wall and placed in the calibrator chamber. After the system is thermally stabilized, the temperature is read at the controller and the central monitor, if installed.
3. Report findings of reference and all points of reading.

FIELD SURVEY FOR ENERGY AUDIT

An energy audit is an organized survey of a specific building to identify and measure all energy uses, determine probable sources of energy losses, and list energy conservation opportunities. This is usually performed as a team effort under the direction of a qualified energy engineer. The field data can be gathered by firms employing technicians trained in testing, adjusting, and balancing.

Instruments

To determine a building's energy use characteristics, existing conditions must be accurately measured with proper instruments. Accurate measurements point out opportunities to reduce waste and provide a record of the actual conditions in the building before energy conservation measures were taken. They provide a compilation of installed equipment data and a record of equipment performance before changes. Judgments will be made based on the information gathered during the field survey; that which is not accurately measured cannot be properly evaluated.

Generally, instruments used for testing, adjusting, and balancing are sufficient for energy conservation surveying. Possible additional instruments include a power factor meter, light meter, combustion testing equipment, refrigeration gages, and equipment for recording temperatures, fluid flow rates, and energy use over time. Only high-quality instruments should be used.

Observation of system operation and any information the technician can obtain from the operating personnel pertaining to the operation should be included in the report.

Data Recording

Organized record keeping is extremely important. A camera is also helpful. Photographs of building components and mechanical and electrical equipment can be reviewed later when the data are analyzed.

Data sheets for energy conservation field surveys contain different and, in some cases, more comprehensive information than those used for testing, adjusting, and balancing. Generally, the energy engineer determines the degree of fieldwork to be performed; data sheets should be compatible with the instructions received.

Building Systems

The most effective way to reduce building energy waste is to identify, define, and tabulate the energy load by building system. For this purpose, load is defined as the quantity of energy used in a building, or by one of its subsystems, for a given period. By following this procedure, the most effective energy conservation opportunities can be achieved more quickly because high priorities can be assigned to systems that consume the most energy.

A building can be divided into nonenergized and energized systems. Nonenergized systems do not require outside energy sources such as electricity and fuel. Energized systems (e.g., mechanical and electrical systems) require outside energy. Energized and nonenergized systems can be divided into subsystems defined by function. Nonenergized subsystems are (1) building site, envelope, and interior; (2) building use; and (3) building operation.

Building Site, Envelope, and Interior. These subsystems should be surveyed to determine how they can be modified to reduce the building load that the mechanical and electrical systems must meet without adversely affecting the building's appearance. It is important to compare actual conditions with conditions assumed by the designer, so that mechanical and electrical systems can be adjusted to balance their capacities to satisfy actual needs.

Building Use. These loads can be classified as people occupancy or operation loads. People occupancy loads are related to schedule, density, and mixing of occupancy types (e.g., process and office). People operation loads are varied and include (1) operation of manual window shading devices; (2) setting of room thermostats; and (3) conservation-related habits such as turning off lights, closing doors and windows, turning off energized equipment when not in use, and not wasting domestic hot or chilled water.

Building Operation. This subsystem consists of the operation and maintenance of all the building subsystems. The load on the building operation subsystem is affected by factors such as (1) the time at which janitorial services are performed, (2) janitorial crew size and time required to clean, (3) amount of lighting used to perform janitorial functions, (4) quality of equipment maintenance program, (5) system operational practices, and (6) equipment efficiencies.

Building Energized Systems

The energized subsystems of the building are generally plumbing, heating, ventilating, cooling, space conditioning, control, electrical, and food service. Although these systems are interrelated and often use common components, logical organization of data requires evaluating the energy use of each subsystem as independently as possible. In this way, proper energy conservation measures for each subsystem can be developed.

Process Loads

In addition to building subsystem loads, the process load in most buildings must be evaluated by the energy field auditor. Most tasks not only require energy for performance, but also affect the energy consumption of other building subsystems. For example, if a process releases large amounts of heat to the space, the process consumes energy and also imposes a large load on the cooling system.

Guidelines for Developing a Field Study Form

A brief checklist follows that outlines requirements for a field study form needed to conduct an energy audit.

Inspection and Observation of All Systems. Record physical and mechanical condition of the following:

- Fan blades, fan scroll, drives, belt tightness, and alignment
- Filters, coils, and housing tightness
- Ductwork (equipment room and space, where possible)
- Strainers
- Insulation ducts and piping
- Makeup water treatment and cooling tower

Interview of Physical Plant Supervisor. Record answers to the following survey questions:

- Is the system operating as designed? If not, what changes have been made to ensure its performance?
- Have there been modifications or additions to the system?
- If the system has had a problem, list problems by frequency of occurrence.
- Are any systems cycled? If so, which systems and when, and would building load permit cycling systems?

Recording System Information. Record the following system/equipment identification:

- Type of system: single-zone, multizone, double-duct, low- or high-velocity, reheat, variable-volume, or other

- System arrangement: fixed minimum outside air, no relief, gravity or power relief, economizer gravity relief, exhaust return, or other
- Air-handling equipment: fans (supply, return, and exhaust) manufacturer, model, size, type, and class; dampers (vortex, scroll, or discharge); motors manufacturer, power requirement, full load amperes, voltage, phase, and service factor
- Chilled- and hot-water coils: area, tubes on face, fin spacing, and number of rows (coil data necessary when shop drawings are not available)
- Terminals: high-pressure mixing box manufacturer, model, and type (reheat, constant-volume, variable-volume, induction); grilles, registers, and diffusers manufacturer, model, style, and loss coefficient to convert field-measured velocity to flow rate
- Main heating and cooling pumps, over 5 hp: manufacturer, pump service and identification, model, size, impeller diameter, speed, flow rate, head at full flow, and head at no flow; motor data (power, speed, voltage, amperes, and service factor)
- Refrigeration equipment: chiller manufacturer, type, model, serial number, nominal tons, brake horsepower, total heat rejection, motor (horsepower, amperes, volts), chiller pressure drop, entering and leaving chilled water temperatures, condenser pressure drop, condenser entering and leaving water temperatures, running amperes and volts, no-load running amperes and volts
- Cooling tower: manufacturer, size, type, nominal tons, range, flow rate, and entering wet-bulb temperature
- Heating equipment: boiler (small through medium) manufacturer, fuel, energy input (rated), and heat output (rated)

Recording Test Data. Record the following test data:

- Systems in normal mode of operation (if possible): fan motor running amperes and volts and power factor (over 5 hp); fan speed, total air (pitot-tube traverse where possible), and static pressure (discharge static minus inlet total); static profile drawing (static pressure across filters, heating coil, cooling coil, and dampers); static pressure at ends of runs of the system (identifying locations)
- Cooling coils: entering and leaving dry- and wet-bulb temperatures, entering and leaving water temperatures, coil pressure drop (where pressure taps permit and manufacturer's ratings can be obtained), flow rate of coil (when other than fan), outside wet and dry bulb, time of day, and conditions (sunny or cloudy)
- Heating coils: entering and leaving dry-bulb temperatures, entering and leaving water temperatures, coil pressure drop (where pressure taps permit and manufacturer's ratings can be obtained), and flow rate through coil (when other than fan)
- Pumps: no-flow head, full-flow discharge pressure, full-flow suction pressure, full-flow differential pressure, motor running amperes and volts, and power factor (over 5 hp)
- Chiller (under cooling load conditions): chiller pressure drop, entering and leaving chilled water temperatures, condenser pressure drop, entering and leaving condenser water temperatures, running amperes and volts, no-load running amperes and volts, chilled water on and off, and condenser water on and off
- Cooling tower: water flow rate in tower, entering and leaving water temperatures, entering and leaving wet bulb, fan motor [amperes, volts, power factor (over 5 hp), and ambient wet bulb]
- Boiler (full fire): input energy (if possible), percent CO_2, stack temperature, efficiency, and complete Orsat test on large boilers
- Boiler controls: description of operation
- Temperature controls: operating and set point temperatures for mixed air controller, leaving air controller, hot deck controller, cold deck controller, outside reset, interlock controls, and damper controls; description of complete control system and any malfunctions
- Outside air intake versus exhaust air: total airflow measured by pitot-tube traverses of both outside air intake and exhaust air systems, where possible. Determine whether an imbalance in the exhaust system causes infiltration. Observe exterior walls to determine whether outside air can infiltrate return air (record outside air, return air, and return air plenum dry- and wet-bulb temperatures). The greater the differential between outside and return air, the more evident the problem will be.

REPORTS

The report is a record of the HVAC system balancing and must be accurate in all respects. If the data are that incomplete, it must be listed as a deficiency for correction by the installing contractor.

Reports should comply with ASHRAE *Standard* 111, be complete, and include the location of test drawings. An instrument list including serial numbers and current and future calibration dates should also be provided.

TESTING FOR SOUND AND VIBRATION

Testing for sound and vibration ensures that equipment is operating satisfactorily and that no objectionable noise and vibration are transmitted to the building structure and occupied space. Although sound and vibration are specialized fields that require expertise not normally developed by the HVAC engineer, the procedures to test HVAC are relatively simple and can be performed with a minimum of equipment by following the steps outlined in this section. Although this section provides useful information for resolving common noise and vibration problems, Chapter 47 should be consulted for information on problem solving or the design of HVAC.

Testing for Sound

Present technology does not test whether equipment is operating with desired sound levels; field tests can only determine sound pressure levels, and equipment ratings are almost always in terms of sound power levels. Until new techniques are developed, the testing engineer can only determine (1) whether sound pressure levels are in desired limits and (2) which equipment, systems, or components are the source of excessive or disturbing transmission.

Sound-Measuring Instruments. Although an experienced listener can often determine whether systems are operating in an acceptably quiet manner, sound-measuring instruments are necessary to determine whether system noise levels are in compliance with specified criteria, and if not, to obtain and report detailed information to evaluate the cause of noncompliance. Instruments normally used in field testing are as follows.

The **precision sound level meter** is used to measure sound pressure level. The most basic sound level meters measure overall sound pressure level and have up to three weighted scales that provide limited filtering capability. The instrument is useful in assessing outside noise levels in certain situations and can provide limited information on the low-frequency content of overall noise levels, but it provides insufficient information for problem diagnosis and solution. Its usefulness in evaluating indoor HVAC sound sources is thus limited.

Proper evaluation of HVAC sound sources requires a sound level meter capable of filtering overall sound levels into frequency increments of one octave or less.

Sound analyzers provide detailed information about sound pressure levels at various frequencies through filtering networks. The most popular sound analyzers are the octave band and center frequency, which break the sound into the eight octave bands of audible sound. Instruments are also available for 0.33, 0.1, and narrower spectrum analysis; however, these are primarily for laboratory and research applications. Sound analyzers (octave midband or center frequency) are required where specifications are based on noise criteria (NC) and room criteria (RC) curves or similar frequency criteria and for problem jobs where a knowledge of frequency is necessary to determine proper corrective action.

Personal computers are a versatile sound-measuring tool. Software used on portable computers has all the functional capabilities described previously, plus many that previously required a fully equipped acoustical laboratory. This type of sound-measuring system is many times faster and much more versatile than conventional sound level meters. With suitable accessories, it can also be used to evaluate vibration levels. Accuracy and calibration to applicable standards are of concern for software.

Regardless of which sound-measuring system is used, it should be calibrated before each use. Some systems have built-in calibration, while others use external calibrators. Much information is available on the proper application and use of sound-measuring instruments.

Air noise, caused by air flowing at a velocity of over 1000 fpm or by winds over 12 mph, can cause substantial error in sound measurements because of wind effect on the microphone. For outside measurements or in drafty places, either a wind screen for the microphone or a special microphone is required. When in doubt, use a wind screen on standard microphones.

Sound Level Criteria. Without specified values, the testing engineer must determine whether sound levels are within acceptable limits (Chapter 47). Note that complete absence of noise is seldom a design criterion, except for certain critical locations such as sound and recording studios. In most locations, a certain amount of noise is desirable to mask other noises and provide speech privacy; it also provides an acoustically pleasing environment, since few people can function effectively in extreme quiet. Table 1 in Chapter 7 of the *ASHRAE Handbook—Fundamentals* lists typical sound pressure levels. In determining allowable HVAC equipment noise, it is as inappropriate to demand 30 dB for a factory where the normal noise level is 75 dB as it is to specify 60 dB for a private office where the normal noise level might be 35 dB.

Most field sound-measuring instruments and techniques yield an accuracy of ±3 dB, the smallest difference in sound pressure level that the average person can discern. A reasonable tolerance for sound criteria is 5 dB; if 35 dBA is considered the maximum allowable noise, the design engineer should specify 30 dBA.

The measured sound level of any location is a combination of all sound sources present, including sound generated by HVAC equipment as well as sound from other sources such as plumbing systems and fixtures, elevators, light ballasts, and outside noises. In testing for sound, all sources from other than HVAC equipment are considered background or ambient noise.

Background sound measurements generally have to be made (1) when the specification requires that the sound levels from HVAC equipment only, as opposed to the sound level in a space, not exceed a certain specified level; (2) when the sound level in the space exceeds a desirable level, in which case the noise contributed by the HVAC system must be determined; and (3) in residential locations where little significant background noise is generated during the evening hours and where generally low allowable noise levels are specified or desired. Because background noise from outside sources such as vehicular traffic can fluctuate widely, sound measurements for residential locations are best made in the normally quiet evening hours. Procedures for residential sound measurements can be found in ASTM *Standard* E1574, Measurement of Sound in Residential Spaces.

Sound Testing. Ideally, a building should be completed and ready for occupancy before sound level tests are taken. All spaces in which readings will be taken should be furnished with whatever drapes, carpeting, and furniture are typical as these affect the room absorption, which can affect the sound levels and the subjective quality of the sound. In actual practice, because most tests must be conducted before the space is completely finished and furnished for final occupancy, the testing engineer must make some allowances. Because furnishings increase the absorption coefficient and reduce by about 4 dB the sound pressure level that can be expected between most live and dead spaces, the following guidelines should suffice for measurements made in unfurnished spaces. If the sound pressure level is 5 dB or more over the specified or desired criterion, it can be assumed that the criterion will not be met, even with the increased absorption provided by furnishings. If the sound pressure level is 0 to 4 dB greater than the specified or desired criterion, recheck when the room is furnished to determine compliance.

Follow this general procedure:

1. Obtain a complete set of accurate, as-built drawings and specifications, including duct and piping details. Review specifications to determine sound and vibration criteria and any special instructions for testing.
2. Visually check for noncompliance with plans and specifications, obvious errors, and poor workmanship. Turn system on for aural check. Listen for noise and vibration, especially duct leaks and loose fittings.
3. Adjust and balance equipment, as described in other sections, so that final acoustical tests are made with the HVAC as it will be operating. It is desirable to perform acoustical tests for both summer and winter operation, but where this is not practical, make tests for the summer operating mode, as it usually has the potential for higher sound levels. Tests must be made for all mechanical equipment and systems, including standby.
4. Check calibration of instruments.
5. Measure sound levels in all areas as required, combining measurements as indicated in Step (3) if equipment or systems must be operated separately. Before final measurements are made in any particular area, survey the area using an A-weighted scale reading (dBA) to determine the location of the highest sound pressure level. Indicate this location on a testing form, and use it for test measurements. Restrict the preliminary survey to determine location of test measurements to areas that can be occupied by standing or sitting personnel. For example, measurements would not be made directly in front of a diffuser located in the ceiling, but would be made as close to the diffuser as standing or sitting personnel might be situated. In the absence of specified sound criteria, the testing engineer should measure sound pressure levels in all occupied spaces to determine compliance with criteria indicated in Chapter 47 and to locate any sources of excessive or disturbing noise. With octave band sound level measurements, overall NC and RC values can be determined. Measurements of 63 to 8000 Hz should be considered the minimum when sound levels in the 16 and 31.5 Hz octave band are desired, from completeness and evaluation of the possibility of noise-induced vibration as evaluated by RC and NCB methods.
6. Determine whether background noise measurements must be made.
   - If specification requires determining sound level from HVAC equipment only, background noise readings must be taken with HVAC equipment turned off.
   - If specification requires compliance with a specific noise level or criterion (e.g., sound levels in office areas shall not exceed 35 dBA), ambient noise measurements must be made only if the noise level in any area exceeds the specified value.
   - For residential locations and areas requiring very low noise, such as sound recording studios and locations used during the normally quieter evening hours, it is usually desirable to take sound measurements in the evening and/or take ambient noise measurements.
7. For outside noise measurements to determine noise radiated by outside or roof-mounted equipment such as cooling towers and condensing units, the section on Sound Control for Outdoor Equipment in Chapter 47, which presents proper procedure and necessary calculations, should be consulted.

Noise Transmission Problems. Regardless of precautions taken by the specifying engineer and installing contractors, situations can occur where the sound level exceeds specified or desired levels, and there will be occasional complaints of noise in completed installations. A thorough understanding of Chapter 47 and the section on Testing for Vibration in this chapter is desirable before attempting to resolve any noise and vibration transmission problems. The following is intended as an overall guide rather than a detailed problem-solving procedure.

All noise transmission problems can be evaluated in terms of the source-path-receiver concept. Objectionable transmission can be resolved by (1) reducing noise at the source by replacing defective equipment, repairing improper operation, proper balancing and adjusting, and replacing with quieter equipment; (2) attenuating paths of transmission with silencers, vibration isolators, and wall treatment to increase transmission loss; and (3) reducing or masking objectionable noise at the receiver by increasing room absorption or introducing a nonobjectionable masking sound. The following discussion includes ways to identify actual noise sources using simple instruments or no instruments and possible corrections.

When troubleshooting in the field, the engineer should listen to the offending sound. The best instruments are no substitute for careful listening, as the human ear has the remarkable ability to identify certain familiar sounds such as bearing squeak or duct leaks and is able to discern small changes in frequency or sound character that might not be apparent from meter readings only. The ear is also a good direction and range finder; noise generally gets louder as one approaches the source, and direction can often be determined by turning the head. Hands can also identify noise sources. Air jets from duct leaks can often be felt, and the sound of rattling or vibrating panels or parts often changes or stops when these parts are touched.

In trying to locate noise sources and transmission paths, the engineer should consider the location of the affected area. In areas remote from equipment rooms containing significant noise producers but adjacent to shafts, noise is usually the result of structure-borne transmission through pipe and duct supports and anchors. In areas adjoining, above, or below equipment rooms, noise is usually caused by openings (acoustical leaks) in the separating floor or wall or by improper, ineffective, or maladjusted vibration isolation systems.

Unless the noise source or path of transmission is quite obvious, the best way to identify it is by eliminating all sources systematically as follows:

1. Turn off all equipment to make sure that the objectionable noise is caused by the HVAC. If the noise stops, the HVAC components (compressors, fans, and pumps) must be operated separately to determine which are contributing to the objectionable noise. Where one source of disturbing noise predominates, the test can be performed starting with all equipment in operation and turning off components or systems until the disturbing noise is eliminated. Tests can also be performed starting with all equipment turned off and operating various component equipment singularly, which permits evaluation of noise from each individual component.

   Any equipment can be termed a predominant noise source if, when the equipment is shut off, the sound level drops 3 dBA or if, when measurements are taken with equipment operating individually, the sound level is within 3 dBA of the overall objectionable measurement.

   When a sound level meter is not used, it is best to start with all equipment operating and shut off components one at a time because the ear can reliably detect differences and changes in noise but not absolute levels.
2. When some part of the HVAC system is established as the source of objectionable noise, try to further isolate the source. By walking around the room, determine whether the noise is coming through air outlets or returns, hung ceiling, or floors or walls.
3. If the noise is coming through the hung ceiling, check that ducts and pipes are isolated properly and not touching the hung ceiling supports or electrical fixtures, which would provide large noise radiating surfaces. If ducts and pipes are the source of noise and are isolated properly, possible remedies to reduce the noise include changing flow conditions, installing silencers, and/or wrapping the duct or pipe with an acoustical barrier or lagging such as a lead blanket or other materials suitable for the location (see Chapter 47).
4. If noise is coming through the walls, ceiling, or floor, check for any openings to adjoining shafts or equipment rooms, and check vibration isolation systems to ensure that there is no structure-borne transmission from nearby equipment rooms or shafts.
5. Noise traced to air outlets or returns usually requires careful evaluation by an engineer or acoustical consultant to determine the source and proper corrective action (see Chapter 47). In general, air outlets can be selected to meet any acoustical design goal by keeping the velocity sufficiently low. For any given outlet, the sound level increases about 2 dB for each 10% increase in airflow velocity over the vanes, and doubling the velocity increases the sound level by about 12 to 15 dB. Approach conditions caused by improperly located control dampers or improperly sized diffuser necks can increase sound levels by 10 to 20 dB. Using variable-frequency drive (VFD) speed controllers on air-handling units can help evaluate air velocity concerns. Dampers used to limit airflow often influence overall sound levels.

   A simple, effective instrument that aids in locating noise sources is a microphone mounted on a pole. It can be used to localize noises in hard-to-reach places, such as hung ceilings and behind heavy furniture.
6. If noise is traced to an air outlet, measure the A-weighted sound level close to it but with no air blowing against the microphone. Then, remove the inner assembly or core of the air outlet and repeat the reading with the meter and the observer in exactly the same position as before. If the second reading is more than 3 dB below the first, a significant amount of noise is caused by airflow over the vanes of the diffuser or grille. In this case, check whether the system is balanced properly. As little as 10% too much air will increase the sound generated by an air outlet by 2.5 dB. As a last resort, a larger air outlet could be substituted to obtain lower air velocities and hence less turbulence for the same air quality. Before this is considered, however, the air approach to the outlet should be checked.

   Noise far exceeding the normal rating of a diffuser or grille is generated when a throttled damper is installed close to it. Air jets impinge on the vanes or cones of the outlet and produce **edge tones** similar to the hiss heard when blowing against the edge of a ruler. The material of the vanes has no effect on this noise, although loose vanes may cause additional noise from vibration.

   When balancing air outlets with integral volume dampers, consider the static pressure drop across the damper, as well as the air quantity. Separate volume dampers should be installed sufficiently upstream from the outlet that there is no jet impingement. Plenum inlets should be brought in from the side, so that jets do not impinge on the outlet vanes.
7. If air outlets are eliminated as sources of excessive noise, inspect the fan room. If possible, change fan speed by about 10%. If resonance is involved, this small change can make a significant difference.
8. Sometimes fans are poorly matched to the system. If a belt-driven fan delivers air at a higher static pressure than is needed to move the design air quantity through the system, reduce fan speed by changing sheaves. If the fan does not deliver enough air, consider increasing fan speed only after checking the duct system for unavoidable losses. Turbulence in the air approach to the fan inlet increases fan sound generation and decreases its air capacity. Other parts that may cause excessive turbulence are dampers, duct bends, and sudden enlargements or contractions of the duct. When investigating fan noise, seek assistance from the fan supplier or manufacturer.
9. If additional acoustical treatment is to be installed in the ductwork, obtain a frequency analysis. This involves the use of an octave-band analyzer and should generally be left to a trained engineer or acoustic analysis consultant.

Testing for Vibration

Vibration testing is necessary to ensure that (1) equipment is operating within satisfactory vibration levels and (2) objectionable vibration and noise are not transmitted to the building structure. Although these two factors are interrelated, they are not necessarily interdependent. A different solution is required for each, and it is essential to test both the isolation and vibration levels of equipment.

General Procedure.

1. Make a visual check of all equipment for obvious errors that must be corrected immediately.
2. Make sure all isolation is free-floating and not short-circuited by obstruction between equipment or equipment base and building structure.
3. Turn on the system for an aural check of any obviously rough operation. Checking bearings with vibration measurement instrumentation is especially important because bearings can become defective in transit and/or if equipment was not properly stored, installed, or maintained. Defective bearings should be replaced immediately to avoid damage to the shaft and other components.
4. Adjust and balance equipment and systems so that final vibration tests are made on equipment as it will actually be operating.
5. Test equipment vibration.

Instruments. Although instruments are not required to test vibration isolation systems, they are essential to test equipment vibration properly.

Sound-level meters and **computer-driven sound-measuring systems** are the most useful instruments for measuring and evaluating vibration. Usually, they are fitted with accelerometers or vibration pickups for a full range of vibration measurement and analysis. Other instruments used for testing vibration in the field are described as follows.

Reed vibrometers are relatively inexpensive and are often used for testing vibration, but their relative inaccuracy limits their usefulness.

Vibrometers are moderately priced and measure vibration amplitude by means of a light beam projected on a graduated scale.

Vibrographs are moderately priced mechanical instruments that measure both amplitude and frequency. They provide a chart recording amplitude, frequency, and actual wave form of vibration. They can be used for simple, accurate determination of the natural frequency of shafts, components, and systems by a **bump test**.

Reed vibrometers, vibrometers, and vibrographs have largely been supplanted by electronic meters that are more accurate and have become much more affordable.

Vibration meters are moderately priced, relatively simple-to-use modern electronic instruments that measure the vibration amplitute. They provide a single broadband (summation of all frequencies) number identifying the magnitude of the vibration level. Both analog and digital readouts are common.

Vibration analyzers are relatively expensive electronic instruments that measure amplitude and frequency, usually incorporating a variable filter.

Strobe lights are often used with many of the other instruments for analyzing and balancing rotating equipment.

Stethoscopes are available as inexpensive mechanic's type (basically, a standard stethoscope with a probe attachment), relatively inexpensive models incorporating a tunable filter, and moderately priced powered types that electronically amplify sound and provide some type of meter and/or chart recording.

The choice of instruments depends on the test. Vibrometers and vibration meters can be used to measure vibration amplitude as an acceptance check. Because they cannot measure frequency, they cannot be used for analysis and primarily function as a go/no-go instrument. The best acceptance criteria consider both amplitude and frequency. Anyone seriously concerned with vibration testing should use an instrument that can determine frequency as well as amplitude, such as a vibrograph or vibration analyzer.

Vibration measurement instruments (both meters and analyzers) made specifically for measuring machinery vibration typically use **moving coil velocity transducers**, which are sizable and rugged. These are typically limited to a lower frequency of 500 cycles per minute (cpm) [8.33 Hz] with normal calibration. If measuring very-low-speed machinery such as large fans, cooling towers, or compressors operating below this limit, use an adjustment factor provided by the instrument manufacturer or use an instrument with a lower low-frequency limit, which will typically use a smaller accelerometer as the vibration pickup transducer.

Testing Vibration Isolation.

1. Ensure that equipment is **free-floating** by applying an unbalanced load, which should cause the equipment to move freely and easily. On floor-mounted equipment, check that there are no obstructions between the base or foundation and the building structure that would cause transmission while still permitting equipment to rock relatively free because of the application of an unbalanced force (Figure 13). On suspended equipment, check that hanger rods are not touching the hanger. Rigid connections such as pipes and ducts can prohibit mounts from functioning properly and from providing a transmission path. Note that the fact that the equipment is free floating does not mean that the isolators are functioning properly. For example, a 500 rpm fan on isolators with a natural frequency of 500 cpm (8.33 Hz) could be free-floating but would actually be in resonance, resulting in transmission to the building and excessive movement.
2. Determine whether isolators are adjusted properly and providing desired isolation efficiency. All isolators supporting a piece of equipment should have approximately the same deflection (i.e., they should be compressed the same under the equipment). If not, they have been improperly adjusted, installed, or selected; this should be corrected immediately. Note that isolation efficiency cannot be checked by comparing vibration amplitude on equipment to amplitude on the structure (Figure 14).

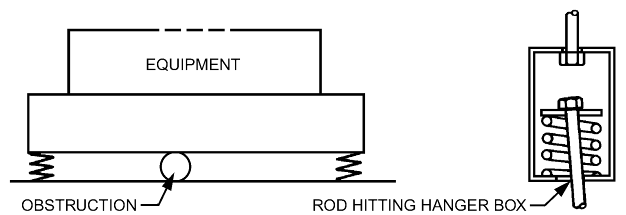

Fig. 13 Obstructed Isolation Systems

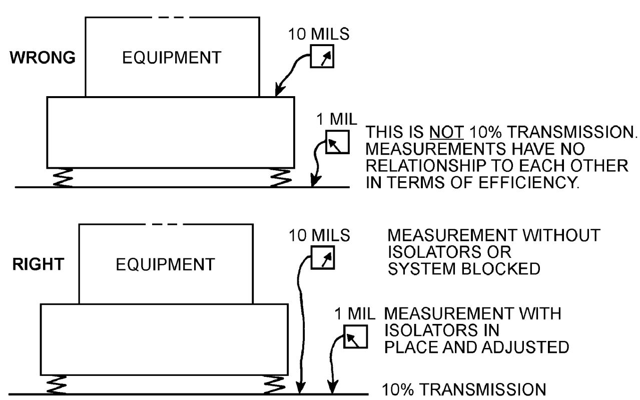

Fig. 14 Testing Isolation Efficiency

The only accurate check of isolation efficiencies is to compare vibration measurements of equipment operating with isolators to measurements of equipment operating without isolators. Because this is usually impractical, it is better to check whether the isolator's deflection is as specified and whether the specified or desired isolation efficiency is being provided. Figure 15 shows natural frequency of isolators as a function of deflection and indicates the theoretical isolation efficiencies for various frequencies at which the equipment operates.

Although it is easy to determine the deflection of spring mounts by measuring the difference between the free heights with a ruler (information as shown on submittal drawings or available from a manufacturer), such measurements are difficult with most pad or rubber mounts. Further, most pad and rubber mounts do not lend themselves to accurate determination of natural frequency as a function of deflection. For such mounts, the most practical approach is to check that there is no excessive vibration of the base and no noticeable or objectionable vibration transmission to the building structure.

If isolators are in the 90% efficiency range, and there is transmission to the building structure, either the equipment is operating roughly or there is a flanking path of transmission, such as connecting piping or obstruction, under the base.

Testing Equipment Vibration. Testing equipment vibration is necessary as an acceptance check to determine whether equipment is functioning properly and to ensure that objectionable vibration and noise are not transmitted. Although a person familiar with equipment can determine when it is operating roughly, instruments are usually required to determine accurately whether vibration levels are satisfactory.

Vibration Tolerances. Vibration tolerance criteria are listed in Table 40 of Chapter 47. These criteria are based on equipment installed on vibration isolators and can be met by any reasonably smoothly running equipment. Note that values in Chapter 47 are based on the root-mean-square (RMS) values; other sources often use peak-to-peak or peak values, especially for displacements. For sinusoid responses, it is simple to obtain peak from RMS values, but in application the relationship may not be so straightforward. The main advantage of RMS is that the same instrumentation can be used for both sound and vibration measurements by simply changing the transducer. Also, there is only one recognized reference level for the decibel used for sound-pressure levels, but for vibration levels there are several recognized ones but no single standard. A common mistake in interpreting vibration data is misunderstanding the reference level and whether the vibration data are RMS, peak-to-peak, or peak values. Use great care in publishing and interpreting vibration data and converting to and from linear absolute values and levels in decibels.

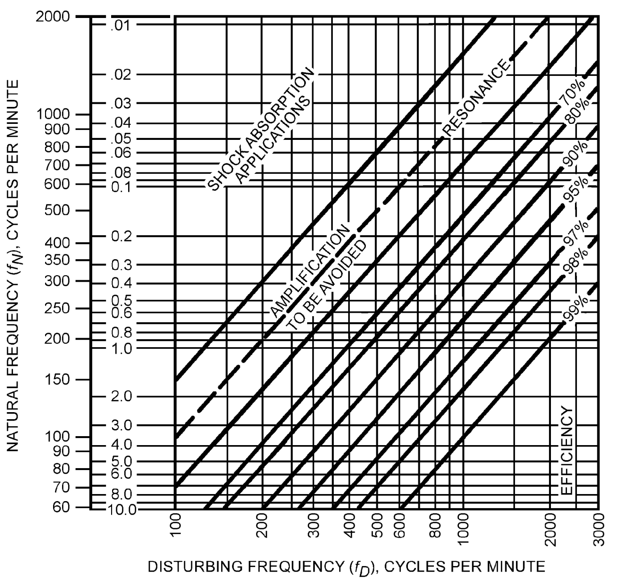

Fig. 15 Isolator Natural Frequencies and Efficiencies

Procedure for Testing Equipment Vibration.

1. Determine operating speeds of equipment from nameplates, drawings, or a speed-measuring device such as a tachometer or strobe, and indicate them on the test form. For any equipment where the driving speed (motor) is different from the driven speed (fan wheel, rotor, impeller) because of belt drive or gear reducers, indicate both driving and driven speeds.
2. Determine acceptance criteria from specifications, and indicate them on the test form. If specifications do not provide criteria, use those shown in Chapter 47.
3. Ensure that the vibration isolation system is functioning properly (see the section on Testing Vibration Isolation).
4. Operate equipment and make visual and aural checks for any apparent rough operation. Any defective bearings, misalignment, or obvious rough operation should be corrected before proceeding further. If not corrected, equipment should be considered unacceptable.
5. Measure and record vibration at bearings of driving and driven components in horizontal, vertical, and, if possible, axial directions. At least one axial measurement should be made for each rotating component (fan motor, pump motor).
6. Evaluate measurements.

Evaluating Vibration Measurements.

Amplitude Measurement. When specification for acceptable equipment vibration is based on amplitude measurements only, and measurements are made with an instrument that measures only amplitude (e.g., a vibration meter or vibrometer),

- No measurement should exceed specified values or values shown in Tables 40 or 41 of Chapter 47, taking into consideration reduced values for equipment installed on inertia blocks
- No measurement should exceed values shown in Tables 40 or 41 of Chapter 47 for driving and driven speeds, taking into consideration reduced values for equipment installed on inertia blocks. For example, with a belt-driven fan operating at 800 rpm and having an 1800 rpm driving motor, amplitude measurements at fan bearings must be in accordance with values shown for 800 cpm (13.3 Hz), and measurements at motor bearings must be in accordance with values shown for 1800 cpm (30 Hz). If measurements at motor bearings exceed specified values, take measurements of the motor only with belts removed to determine whether there is feedback vibration from the fan.
- No axial vibration measurement should exceed maximum radial (vertical or horizontal) vibration at the same location.

Amplitude and Frequency Measurement. When specification for acceptable equipment vibration is based on both amplitude and frequency measurements, and measurements are made with instruments that measure both amplitude and frequency (e.g., a vibrograph or vibration analyzer),

- Amplitude measurements at driving and driven speeds should not exceed specified values or values shown in Tables 40 or 41 of Chapter 47, taking into consideration reduced values for equipment installed on inertia blocks. Measurements that exceed acceptable amounts may be evaluated as explained in the section on Vibration Analysis.
- Axial vibration measurements should not exceed maximum radial (vertical or horizontal) vibration at the same location.
- The presence of any vibration at frequencies other than driving or driven speeds is generally reason to rate operation unacceptable; such vibration should be analyzed as explained in the section on Vibration Analysis.

Vibration Analysis. The following guide covers most vibration problems that may be encountered.

Axial Vibration Exceeds Radial Vibration. When the amplitude of axial vibration (parallel with shaft) at any bearing exceeds radial vibration (perpendicular to shaft, vertical or horizontal), it usually indicates misalignment, most common on direct-driven equipment because flexible couplings accommodate parallel and angular misalignment of shafts. Such misalignment can generate forces that cause axial vibration. Axial vibration can cause premature bearing failure, so misalignment should be checked carefully and corrected promptly. Other possible causes of large-amplitude axial vibration are resonance, defective bearings, insufficient rigidity of bearing supports or equipment, and loose hold-down bolts.

Vibration Amplitude Exceeds Allowable Tolerance at Rotational Speed. The allowable vibration limits established by Table 37 of Chapter 47 are based on vibration caused by rotor imbalance, which results in vibration at rotational frequency. Although vibration caused by imbalance must be at the frequency at which the part is rotating, a vibration at rotational frequency does not have to be caused by imbalance. An unbalanced rotating part develops centrifugal force, which causes it to vibrate at rotational frequency. Vibration at rotational frequency can also result from other conditions such as a bent shaft, an eccentric sheave, misalignment, and resonance. If vibration amplitude exceeds allowable tolerance at rotational frequency, the following steps should be taken before performing field balancing of rotating parts:

1. Check vibration amplitude as equipment goes up to operating speed and as it coasts to a stop. Any significant peaks at or near operating speed, as shown in Figure 16, indicate probable resonance (i.e., some part having a natural frequency close to the operating speed, resulting in greatly amplified levels of vibration)

   A bent shaft or eccentricity usually causes imbalance that results in significantly higher vibration amplitude at lower speeds, as shown in Figure 17, whereas vibration caused by imbalance generally increases as speed increases.

   If a bent shaft or eccentricity is suspected, check the dial indicator. A bent shaft or eccentricity between bearings as shown in Figure 18A can usually be compensated for by field balancing, although some axial vibration might remain. Field balancing cannot correct vibration caused by a bent shaft on direct-connected equipment, on belt-driven equipment where the shaft is bent at the location of sheave, or if the sheave is eccentric (Figure 18B). This is because the center-to-center distance of the sheaves will fluctuate, each revolution resulting in vibration.
2. For belt- or gear-driven equipment where vibration is at motor driving frequency rather than driven speed, it is best to disconnect the drive to perform tests. If the vibration amplitude of the motor operating by itself does not exceed specified or allowable values, excessive vibration (when the drive is connected) is probably a function of bent shaft, misalignment, eccentricity, resonance, or loose holddown bolts.
3. Vibration caused by imbalance can be corrected in the field by firms specializing in this service or by testing personnel if they have appropriate equipment and experience.

Table 4 Common Causes of Vibration Other than Unbalance at Rotation Frequency

Frequency	Source
0.5 × rpm	Vibration at approximately 0.5 rpm can result from improperly loaded sleeve bearings. This vibration will usually disappear suddenly as equipment coasts down from operating speed.
2 × rpm	Equipment is not tightly secured or bolted down.
2 × rpm	Misalignment of couplings or shafts usually results in vibration at twice rotational frequency and generally a relatively high axial vibration.
Many × rpm	Defective antifriction (ball, roller) bearings usually result in low-amplitude, high-frequency, erratic vibration. Because defective bearings usually produce noise rather than any significantly measurable vibration, it is best to check all bearings with a listening device.

Vibration at Other than Rotational Frequency. Vibration at frequencies other than driving and driven speeds is generally considered unacceptable. Table 4 shows some common conditions that can cause vibration at other than rotational frequency.

Resonance. If resonance is suspected, determine which part of the system is in resonance.

Isolation Mounts. The natural frequency of the most commonly used spring mounts is a function of spring deflection, as shown in Figure 19, and it is relatively easy to calculate by determining the difference between the free and operating height of the mount, as explained in the section on Testing Vibration Isolation. This technique cannot be applied to rubber, pad, or fiberglass mounts, which have a natural frequency in the 300 to 3000 cpm (5 to 50 Hz) range. Natural frequency for such mounts is determined by a bump test. Any resonance with isolators should be immediately corrected because it results in excessive movement of equipment and more transmission to the building structure than if equipment were attached solidly to the building (installed without isolators).

Components. Resonance can occur with any shaft, structural base, casing, and connected piping. The easiest way to determine natural

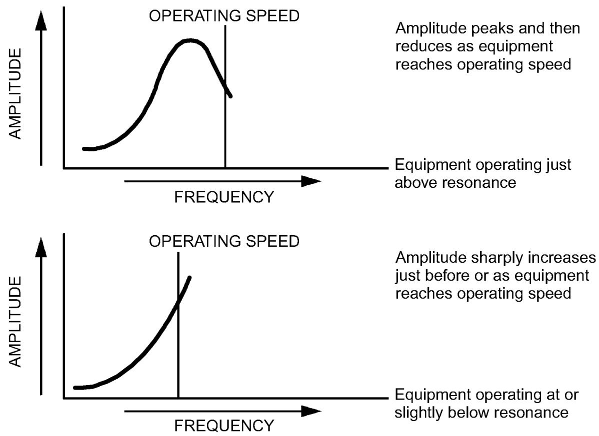

Fig. 16 Vibration from Resonant Condition

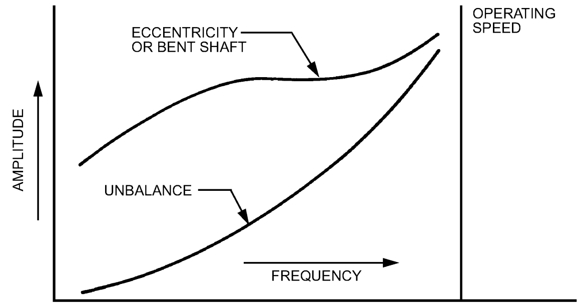

Fig. 17 Vibration Caused by Eccentricity

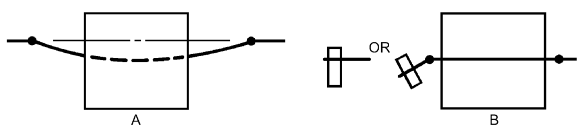

Fig. 18 Bent Shafts

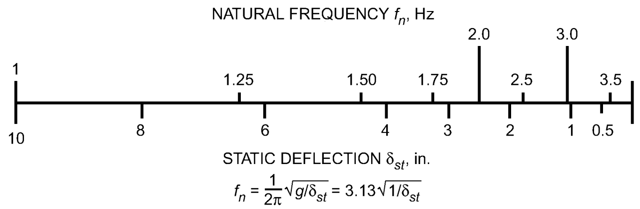

Fig. 19 Natural Frequency of Vibration Isolators

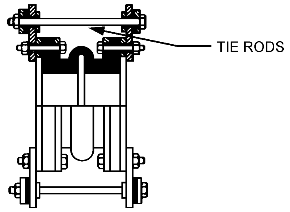

Fig. 20 Typical Tie Rod Assembly

frequency is to perform a bump test with a vibration spectrum analyzer. This test consists of bumping the part and measuring; the part will vibrate at its natural frequency, which will show up as a response peak on the analyzer. However, most of these instruments are restricted to frequencies above 500 cpm (8.3 Hz). They therefore cannot be used to determine natural frequencies of most isolation systems, which usually have natural frequencies lower than 500 cpm (8.3 Hz).

Checking for Vibration Transmission. The source of vibration transmission can be checked by determining frequency with a vibration analyzer and tracing back to equipment operating at this speed. However, the easiest and usually best method (even if test equipment is being used) is to shut off components one at a time until the source of transmission is located. Most transmission problems cause disturbing noise; listening is the most practical approach to determine a noise source because the ear is usually better than instruments at distinguishing small differences and changes in character and amount of noise. Where disturbing transmission consists solely of vibration, an instrument will probably be helpful, unless vibration is significantly above the sensory level of perception. Vibration below sensory perception is generally not objectionable.

If equipment is located near the affected area, check isolation mounts and equipment vibration. If vibration is not being transmitted through the base, or if the area is remote from equipment, the probable cause is transmission through connected piping and/or ducts. Ducts can usually be isolated by isolation hangers. However, transmission through connected piping is very common and presents many problems that should be understood before attempting to correct them (see the following section).

Vibration and Noise Transmission in Piping. Vibration and noise in connected piping can be generated by either equipment (e.g., pump or compressor) or flow (velocity). Mechanical vibration from equipment can be transmitted through the walls of pipes or by a water column. Flexible pipe connectors, which provide system flexibility to permit isolators to function properly and protect equipment from stress caused by misalignment and thermal expansion, can be useful in attentuating mechanical vibration transmitted through a pipe wall. However, they rarely suppress flow vibration and noise and only slightly attenuate mechanical vibration as transmitted through a water column.

Tie rods are often used with flexible rubber hose and rubber expansion joints (Figure 20). Although they accommodate thermal movements, they hinder vibration and noise isolation. This is because pressure in the system causes the hose or joint to expand until resilient washers under tie rods are virtually rigid. To isolate noise adequately with a flexible rubber connector, tie rods and anchor piping should not be used. However, this technique generally cannot be used with pumps on spring mounts, which would still permit the hose to elongate. Flexible metal hose can be used with spring-isolated pumps because wire braid serves as tie rods; metal hose controls vibration but not noise.

Problems of transmission through connected piping are best resolved by changes in the system to reduce noise (improve flow characteristics, reduce impeller size) or by completely isolating piping from the building structure. Note, however, that it is almost impossible to isolate piping completely from the structure, as required resiliency is inconsistent with rigidity requirements of pipe anchors and guides. Chapter 47 contains information on flexible pipe connectors and resilient pipe supports, anchors, and guides, which should help resolve any piping noise transmission problems.

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