BSR/ASHRAE Standard 111-2008R

Public Review Draft

Measurement, Testing, Adjusting and Balancing of Building Heating, Ventilation and Air-Conditioning Systems

First Public Review (July 2020)
(Complete Draft for Full Review)

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FOREWORD

Insert prior to publication.

1. PURPOSE

1.1 To provide uniform procedures for measurement, testing, adjusting, balancing, evaluating, and reporting the performance of building heating, ventilating, and air conditioning systems in the field.

2. SCOPE

2.1 This standard applies to: building heating, ventilating, and air conditioning (HVAC) systems of the air moving and hydronic types and their associated heat transfer, distribution, refrigeration, electrical power, and control subsystems.

2.2 This standard includes:

2.2.1 Methods for determining thermodynamic, hydraulic, hydronic, mechanical, and electrical conditions.

2.2.2 Methods for determining room air change rates, room pressurization, and cross contamination of spaces.

2.2.3 Procedures for measuring and adjusting outdoor ventilation rates to meet specified requirements.

2.2.4 Methods for validating collected data while considering system effects.

2.3 This standard establishes:

2.3.1 Minimum system configuration requirements to ensure that the system can be field tested and balanced.

2.3.2 Minimum instrumentation required for field measurements.

2.3.3 Procedures for field measurements in HVAC testing and balancing and equipment testing.

2.3.4 Formats for recording and reporting results.

2.3.5 The field data collected and reported under this standard is intended for use by building designers, operators and users, and by manufacturers and installers of HVAC systems.

3. DEFINITIONS & SYMBOLS

Note: Terminology is as defined below. All other definitions shall be as listed in the *ASHRAE TERMINOLOGY OF HEATING, VENTILATION, AIR CONDITIONING, & REFRIGERATION.*

airflow measuring stations: specialized devices that are permanently mounted in an airstream to provide a continuous output of the average velocities at the plane of measurement, for HVAC control or system monitoring. Averaging errors can be reduced with the use of multiple independent sensors across the plane of measurement.

The most popular technologies used by airflow measuring stations are: Thermal Dispersion and Velocity Pressure. The first may be either digital or analog electronic in design, while the later uses a pressure transducer to convert pressure differences sensed to a single non-linear analog signal representing average velocity pressure. Linearization and conversion of velocity pressure to velocity is accomplished in a separate device or calculated by the host controller.

Ak factor: the effective area of an air terminal, equal to the measured airflow rate divided by the velocity reading of an instrument used in a prescribed manner.

fan velocity pressure: the velocity pressure corresponding to the average velocity through the fan outlet. The kinetic energy per unit volume of air exiting the fan.

(PD): abbreviation for pressure difference and delta P.

pump suction lift: may also be referred to as Total Suction Lift, defined as the height from the water surface in a sump to the centerline of the propeller shaft or to the eye of the pump impeller. (Karassik, *Pump Handbook* 2nd ed., pp. 9.48)

reference instrument: an instrument that has been calibrated by an accredited testing laboratory, used to verify the accuracy of field instruments. This instrument is typically not used in the field.

resolution (instrument): a measure of the smallest incremental change which an instrument can display.

static discharge head: the static pressure of a fluid at the outlet of the pumping device, expressed in terms of the height of a column of the fluid, or of some manometric fluid which it would support.

static pressure rise: the algebraic difference between the static pressure at the fan outlet and the static pressure at the fan inlet.

static suction lift: the same as the static suction head but a negative value and measured at the inlet to the pumping device.

thermal dispersion: a means of measurement where thermal transfer is used to determine air velocity. The energy dispersed from a heated sensor in the airstream (typically a thermistor or RTD) is directly related to the air velocity passing the sensor element. Accurate velocity measurement by thermal means requires precise and high-resolution temperature measurement at each point where velocity is sampled.

Thermal methods are used by many technologies in the design of multi-point arrays using independent or inter-dependent sensing elements. These sensing elements are permanently fixed in a plane of the airstream to provide a cross-sectional average. Thermal methods are generally more sensitive to lower velocities than other measurement methods. Thermal measurement reliability is a function of sensor stability, sensor independence, calibration method and reference quality, sensor type, and the accuracy of the ambient temperature measurement points at the velocity, as required for accurate velocity determination at existing ambient conditions.

velocity pressure: the difference between two pressures (equalized Total and equalized Static) as used by various devices to allow the calculation of air velocity at the plane of measurement. The in-duct pressure devices include self-averaging arrays or interconnected Pitot static probes, fixed array of Pitot-static tubes, fan inlet Piezo rings, air measuring dampers used in combination with intake louvers, flow-cross or flow-ring sensors used in VAV boxes, etc. These devices must be mated with a pressure transducer to provide an electronic signal for use by the BAS or application-specific controller.

Due to the non-linear square root relationship between velocity pressure and velocity, a dedicated device or method of linearization must be employed to provide a useful linear duct velocity. This non-linear relationship accounts for the increased measurement uncertainties experienced at flows below 3 m/s (600 FPM) [*ASHRAE Fundamentals 2013*]. Velocity pressure arrays utilizing a single pressure transducer must avoid potential averaging errors due to duct disturbances by seeking better measurement conditions.

4. INSTRUMENTATION

4.1 Scope. This section covers the required instrumentation to obtain the measurements necessary for air or fluid system balancing, and other instruments which are useful or necessary in special situations. Included for each instrument will be a description, recommended uses, limitations, and accuracy requirements.

4.2 General. Follow the instruments operating instructions and procedures for the application of these instruments for field measurements.

4.3 Calibration. Instruments shall be verified and calibrated by laboratories accredited to ISO/IEC 17025 and ANSI Z540-1 over the intended operating range by comparison to a calibrated reference instrument. If the reading on the instrument to be verified is not within manufacturer published accuracy of the reading on the reference instrument, then the instrument being verified must be calibrated by an accredited testing laboratory before it can be used. The interval between verification shall not exceed 6 months.

4.4 Air Balancing Instruments. The minimum required instruments for air balancing are:

4.4.1 Pitot Static Tube

4.4.2 Digital Manometer

4.4.3 Tachometer

4.4.4 Combination Voltmeter and Ammeter

4.4.5 Three Phase Power Meter

4.4.6 Thermometry (DB, WB)

4.4.7 Hygrometer

4.4.8 Flow Capture Hood

4.4.9 Vane Anemometer

4.4.10 Thermal Anemometer

4.5 Pitot Static Tube Uses

4.5.1 Measurement of airstream "total pressure" by connecting the inner tube outlet connector to one side of a manometer or draft gauge.

4.5.2 Measurement of airstream "static pressure" by connecting the outer tube side outlet connector to one side of a manometer or draft gauge.

4.5.3 Measurement of airstream "velocity pressure" by connecting both the inner and outer tube connectors to opposite sides of a manometer or draft gauge.

4.5.4 This instrument, when used with a manometer or manometer, is a most reliable and rugged instrument. Its use as a direct measurement tool is preferred over many other methods for the field measurement of air velocity, system total air, outdoor air, return air quantities, fan static pressure, fan total pressure, and fan outlet velocity pressures where such measured quantities may be required within the range or capabilities of the instrument.

4.5.5 Limitations

4.5.5.1 Pitot static tubes shall not be used to measure velocities below 450 FPM, regardless of the electronic sensors used to identify differentials in pressure, due to the inherent high uncertainties in Pitot static measurement.

4.5.5.2 A reasonably large space is required adjacent to the duct penetrations for maneuvering the instrument.

4.5.5.3 Care must be taken to avoid pinching or puncturing the instrument tubing.

4.5.5.4 Because of the distance between the impact and static holes, the Pitot static tube cannot be used to measure flow through orifice-type openings.

4.5.5.5 The Pitot static tube is susceptible to plugging in air streams with heavy dust or moisture loadings.

4.5.5.6 Acceptance of the standard Pitot static tube rests in the accuracy on the correct determination of the static pressure. The total pressure is not affected by yaw or angularity up to approximately 8 degrees on either side of parallel flow. The static pressure, however, is extremely sensitive to direction of flow.

4.5.6 Accuracy of field measurement. Rigorous error analysis shows that flow rate determinations by the Pitot static tube and manometer combination method range from 5% to 10% error. Experience shows that qualified technicians can obtain measurements that range within 5% and 10% accuracy of actual flow under good field conditions. It has also been determined that suitable traverse conditions do not always exist, and measurements can then exceed a ±10% error rate.

4.6 Digital Manometers. Recommended uses: with Pitot tubes or static pressure probes.

4.6.1 Limitations

4.6.1.1 Not to be used to measure air velocities less than 600 FPM (3.0 m/s).

4.6.1.2 Battery life

4.6.1.3 Accuracy

4.6.1.4 Static: ±2% of reading, ±0.0001 in. H2O (± 0.025 Pa)

4.6.1.5 Differential: ±2% of reading absolute

4.7 Tachometers

4.7.1 Chronometric Tachometers

4.7.1.1 The shaft end must be accessible.

4.7.1.2 Slip between the shaft and instrument will reduce accuracy.

4.7.2 Electronic Tachometer (Stroboscope and Photoelectric):

4.7.2.1 Care must be taken to avoid reading multiples of the actual RPM when using the electronic tachometer. Readings must be started at the lower end of the scale.

4.7.3 Dual Function Tachometer

4.7.3.1 Battery operated

4.7.4 Accuracy of field measurements

4.7.4.1 Chronometric Tachometers, Electronic Tachometer, and Dual Function Tachometer: Within one-half of a scale division mark.

4.7.4.2 Electronic Tachometer (Stroboscope and Photoelectric): Within one-half of a scale division.

4.8 Combination Voltmeter and Voltmeter and Ammeter

4.8.1 Recommended Uses: Measurement of operating voltages and currents of electric motors and of electric resistance heating coils.

4.8.2 Limitations

4.8.2.1 Non-sinusoidal wave forms, such as those at the output of a variable frequency drive, can affect measurement accuracy.

4.8.2.2 The proper range shall be selected. When in doubt, begin with the highest range for both voltage and current scales. Accuracy of reading low currents can be improved by looping the conductor wire around the jaw once and dividing the current reading by 2.

4.8.2.3 Depending upon the conditions at the point of measurement and the size of the volt-ammeter, access for measurement may be restrictive. Caution is required, particularly when taking measurements under confined conditions.

4.8.2.4 To determine distortion of current readings by other fields, move the meter along the wire to verify that the reading remains constant.

Accuracy of field measurements: ±3% of full scale.

4.9 Three Phase Electrical Power Meter

4.9.1 Recommended Uses: Measurement of power consumption in watts (or kW) and power quality for electrical motors serving pumps, fans, and other equipment.

4.9.2 Limitations: Current probes are available in various ranges from milli-amps to thousands of amps. It is critical that the appropriate current probe is selected for the anticipated current of the load being measured to ensure accuracy of measurement. Manufacturer specifications must be referenced to determine appropriate ranges.

4.9.3 Most three-phase meters will measure voltage and amperage on all three phases simultaneously. Correct orientation and phasing of the probes must be verified by the field technician according to manufacturer instructions. Failure to verify phasing between voltage and amperage probes may result in incorrect power consumption and quality data.

4.9.4 Most three-phase power meters have the capability to log data over a period. If this function is utilized, the technician must verify the setup of the logging intervals to ensure appropriate storage space and timing of data recording.

4.9.5 Most three-phase power meters can be configured to measure single-phase power also. This typically involves a setup change in the meter. The technician should reference manufacturer instructions for proper configuration settings.

4.9.6 Accuracy of field measurements: ±2% of full scale.

4.10 Thermometry (DB, WB)

4.10.1 Dial Thermometers

4.10.1.1 Recommended uses: Suitable for checking both air and water temperature in ducts and pipe thermometer wells.

4.10.1.2 Limitations

a. Dial thermometers have a relatively long-time lag, so enough time must be allowed for the thermometer to reach equilibrium and the pointer to come to rest.

b. Vibration shall be avoided; avoid mounting on vibrating equipment or piping. Wall mounting preferred. Another alternative is to install pressure/temperature test ports that can be used with a portable stem probe and gauge (or thermometer) through an elastic, durable self-sealing material. The test port shall be capped when not in use for additional sealing security.

4.10.1.3 Accuracy of field measurements: Within one-half of a scale division mark.

4.10.2 Electronic Thermometry

4.10.2.1 Recommended uses: Suitable for all TAB temperature measurements, including air and other gases, liquids, and surfaces of pipes and other components with the appropriate probe. The manufacturer directions must be followed regarding proper use of probe and maximum allowable temperature for the probe and or thermometer. Equipment is available to measure from -380°F to 2250°F (-230°C to 1230°C). Common ranges used are +14°F to +248°F (-10°C to 120°C).

4.10.2.2 Limitations

a. Batteries must be recharged or changed when required.

b. In piping applications, it should be remembered that the surface temperature of the pipe is not equal to the fluid temperature and that a relative comparison is more reliable than an absolute reliance on readings at a single circuit or terminal unit.

c. Be sure measurement is taken at least as long as response time.

4.10.2.3 Accuracy of field measurements: when properly used, the instrument accuracy shall be attainable in the field.

4.11 Hygrometer

4.11.1 Recommended Use: Establishing relative humidity, wet bulb temperatures, and dew point temperatures.

4.11.2 Limitations

4.11.2.1 Batteries must be recharged or changed when required.

4.11.2.2 Be sure measurement is taken at least as long as response time.

4.11.3 Accuracy of field measurements

4.11.3.1 Relative humidity: ±3.0% RH

4.11.3.2 Temperatures: ±1.0°F (±0.5°C)

4.12 Digital Flow Capture Hood

4.12.1 Recommended Use: Taking direct readings on diffusers and grilles

4.12.2 Limitations

4.12.2.1 Batteries must be recharged or changed when required.

4.12.2.2 The hood skirt must completely cover the diffuser or grille.

4.12.2.3 Field correction factors must be verified when taking readings above 1000 CFM supply and 700 CFM return or exhaust.

4.12.2.4 Accuracies will change when diffusers and/or grilles are mounted in close proximity.

4.12.2.5 Accuracy of field measurements: ±3% of reading ±7 ft^3/min (±12 m^3/h) at flows > 50 ft^3/min (>85 m^3/h).

4.12.3 Vane Anemometer

4.12.3.1 Recommended Uses

a. Measurement of supply return and exhaust air quantities at air inlets and outlets.

b. Measurement of air quantities at the faces of return air dampers or openings, velocities across the filter or coil face areas, etc.

4.12.3.2 Limitations

a. Total inlet area of the instrument must be in the measured air stream.

b. Not suitable for measurement in ducts.

c. It is fragile and cannot be used in dusty, high temperature, or corrosive air.

d. The instrument must be calibrated in the field for correction factor by Pitot static tube traverse within the limitations of the system.

e. Since the instrument has a turbine wheel of very low inertia, caution is advised as to reliability of readings in non-uniform, turbulent, or stratified air streams. This is likely to occur downstream of dampers, face and bypass coils, or any device which causes turbulence in the airstream being measured.

f. Batteries must be recharged or changed when required.

4.12.3.3 Accuracy of field measurements: ±1.0% of reading ±4 ft./min (±0.02 m/s)

4.13 Thermal Anemometer - Single point

4.13.1 Recommended uses: Used to measure very low air velocities, such as room air currents and face velocities in hoods (10 to 600 FPM, 0.05 to 3.0 m/s).

4.13.2 Limitations

4.13.2.1 The probe that is used with this instrument is directional for velocity readings and must be used only in the direction of calibration.

4.13.2.2 Probes are subject to fouling by dust and corrosive air.

4.13.2.3 In general, these instruments should not be used in flammable or explosive atmosphere. However, there are special thermal anemometer probes available for use in these environments.

4.13.2.4 Cannot be used for duct traverses because it does not indicate reverse flows.

4.13.3 Accuracy of field measurements: ±3% of reading or ±3 Ft./min (±0.015 m/s), whichever is greater.

4.14 Other Measuring Instruments for Certain Situations, Air or Fluid Systems

4.14.1 Diaphragm Type Differential Pressure Gauge

4.14.1.1 Recommended uses

a. Use with Pitot static tube or with static probe.

b. Use with specially constructed induction unit primary air total pressure measuring tip for primary air distribution balancing on high pressure induction systems.

4.14.1.2 Limitations

a. Should not be used in preference to liquid or electronic manometer.

b. Readings should be made in mid-scale of range.

c. Should not be mounted on a vibrating surface.

d. Should be held in same position as when "zeroed."

e. Should be checked against a known pressure source with each use.

4.14.2 Barometer

4.14.2.1 Recommended uses: Primarily to measure atmospheric pressure that is required to correct actual airflow to standard conditions.

4.14.2.2 Actual pressure at local elevation must be used for air density calculations (See Appendix D, Example D.1.2).

4.15 Fluid Systems Measuring Instruments

4.15.1 Surface Temperature Probe

4.15.1.1 Recommended uses

a. In balancing water circuits thermally, whenever balancing by flow, measurements are not practical.

b. For evaluation of some types of boilers, furnaces, ovens, etc., where temperatures are over 100°F (40°C).

4.15.1.2 Limitations: In piping applications it should be remembered that the surface temperature of the conduit is not equal to the fluid temperature and that a relative comparison is more reliable than an absolute reliance on readings at a single circuit or terminal unit.

4.15.1.3 Accuracy of field measurements

a. Above 212°F (100°C) ±0.05% +0.17°F (+0.3°C) J, K, T, and E thermocouples.

b. Below 212°F (100°C) ±0.20% +0.17°F (+0.3°C)J, K and E types; T type ±0.50% +0.17°F (+0.3°C).

4.15.2 Calibrated Pressure Gauge

4.15.2.1 Recommended uses: Primarily for checking pump pressures, coil, chiller, and condenser pressure drops and pressure drops across orifice plates, Venturis, and other flow calibrated devices.

4.15.2.2 Limitations

a. Pressure ranges shall be such that the anticipated working pressure range is in the middle two-thirds of the scale range, and the gauge shall not be exposed to pressures greater than the maximum dial reading. Similarly, where there is exposure to vacuum, use compound gauge.

b. Reduce or eliminate pressure pulsations by installing a snubber or needle valve in waterline.

c. Eliminate vibration by avoiding mounting on vibrating equipment or piping. Wall mounting preferred. Another alternative is to install pressure/temperature test ports that can be used with a portable stem probe and gauge (or thermometer) through an elastic, durable self-sealing material. The test port shall be capped when not in use for additional sealing security.

4.15.2.3 Accuracy of field measurements: Within one-half of a scale division mark.

4.15.3 Differential Pressure Gauge

4.15.3.1 Recommended uses: This instrument, when furnished in differential pressure ranges below atmosphere, calibrated in inches of mercury (kPa), and above atmosphere, calibrated in inches of water (Pa), can be used with water hose flexible connectors for water distribution balancing.

4.15.3.2 Limitations: Some applications require use of a snubber or needle valve. A three-valve cluster for shutoff and bypass is necessary to prevent over-pressure damage when used as a portable test gauge.

4.15.3.3 Accuracy of field measurements: Within one-half of a scale division mark

4.15.4 Differential Pressure Manifold Gauge

4.15.4.1 Recommended uses

a. This instrument assembly is used to indicate the pressure at each point by alternating valve opening and closing.

b. By using the same gauge, it eliminates the error from using two separate, permanently mounted gauges which are subject to possible vibration damage and differences in calibration.

4.15.4.2 Accuracy of field measurements: Within one-half of a scale division mark

4.15.5 Fluid System Digital Electronic Differential Pressure Meters

4.15.5.1 Recommended uses: For measurement of fluid flow, temperature, and differential pressure; for computing the setting of compatible valves by proportional balancing procedures.

4.15.5.2 Limitations: Computing feature is limited to compatible valves

4.15.5.3 Accuracy of field measurements: Differential pressure within 1 ft WC or 2% of valve readout (whichever is greater). Flow same as differential pressure via computing feature.

4.15.6 Ultrasonic Flow Meters

4.15.6.1 Recommended uses: To measure flow in full pipes. Excellent when low or zero pressure drop is a requirement. Best fitted for larger pipes and most manufacturer specifications are based on flows of 1 fps or greater.

4.15.6.2 Limitations

a. Doppler Flowmeters: Liquid must contain particulate or gas bubbles.

b. Transit Time Flowmeters: Liquid must be acoustically transparent (implies low particulate content - e.g., typical lake or river water or cleaner).

c. Portable (Strap on) Flowmeters: Pipe parameters (pipe, diameter, wall thickness, and material of construction) must be known or determinable. Pipe must be acoustically transparent (concrete or lined pipe is not).

4.15.6.3 Accuracy of field measurements

a. Doppler Flowmeters:

    1) Strap-on transducers: Typ. 3% to 5% of reading

    2) Integral transducers: Typ. 2% to 3% of reading

b. Transit Time Flowmeters

    1) Strap-on transducers: Typ. 2% to 3% of reading

    2) Integral transducers: Typ. 1% to 2% of reading

    3) Integral transducers factory mounted to a calibrated flow tube: Typ. 0.5% to 1% of reading

4.15.7 Turbine Flow Meters

4.15.7.1 Recommended uses: Measure flow in pipe with clean fluid flow.

4.15.7.2 Limitations: Care must be exercised to maintain the turbine flow meter as wear may affect the wheel bearings. The bearings may drag if impurities lodge in them and debris can clog or break the wheel.

4.15.7.3 Accuracy of field measurements: 2% of reading

5. AIR SYSTEM MEASUREMENTS

5.1 Scope. This section sets forth techniques for

5.1.1 The field measurement of air temperature, air pressure, air velocity, and motor input power

5.1.2 Calculating airflow, pressure differentials, fan power, heat content, humidity, and air density.

5.2 General

5.2.1 This section will apply to both new and existing HVAC systems. Certain characteristics describing the system performance can be measured directly, while others must be calculated from the measured data. The methods for determining each are covered in this section.

5.2.2 The actual air performance determined from field measurement may differ from design conditions. These differences can often be explained by considering system effects that cause changes to design performance as a result of adverse or unexpected system conditions. Refer to Informative Appendix B, System Effects.

5.2.3 The accuracy that can be expected under field conditions for each of the performance characteristics is discussed in the following subsections. Certain system characteristics can be measured by several alternate methods. Many times, the system configuration will not allow the most accurate method to be used. Alternate methods are presented in this section with a discussion of the expected accuracy.

5.3 Temperatures

5.3.1 Air temperatures consist of the dry bulb temperature (DB) and the wet bulb temperature (WB). These are required to determine density, humidity (moisture content), and heat content of the air handled by the system.

5.3.1.1 Dry Bulb Measurements: The following shall be considered to ensure that temperature measurements are representative of the airstream being tested at the plane of interest:

a. When temperature stratification exists, measure temperatures at points locations established by the Log-Tchebycheff Rule (ISO/3966).

b. The sensor shall be shielded if thermal radiation or liquid contact could influence the reading.

c. Temperature measurements shall be made to ensure that a steady state value is being recorded or, if there are oscillations in the values, that representative average values can be determined.

5.3.1.2 Wet Bulb Measurements

a. Wet bulb temperatures are established using a sensing probe such as a thermocouple, RTD, or thermistor covered with a wet sock. The following must be followed to obtain accurate readings and taken in the same manner as procedures noted in 5.3.3.

b. Distilled water shall be used to wet the sock of the wet bulb sensing device.

c. The wetting media covering the sensor shall be clean and remain wet while the measurement is being made.

d. The air velocity across the sensor shall be between 300 and 2000 FPM (1.5 and 10 m/s).

5.4 Density

5.4.1 Methods for determining flow rate require that the air density be known. Density is also required when calculating or estimating pressure changes (losses or gains) across system components.

5.4.2 Data Required: The pressure and temperature of the airstream must be obtained at each location where it is desired to make a density determination. The absolute pressure is required and is determined by adding the measured static pressure value at the location to the barometric pressure as determined for the atmosphere to which the static pressure measurement is referred. The dry bulb temperature is always required. The wet bulb temperature is also required unless it is known that the air is saturated with water vapor or that the water vapor content of the air is insignificant. It should be noted that incorrect assumptions as to whether the air is dry or saturated can result in substantial errors in determining the air density.

5.4.3 Density Determination

5.4.3.1 The procedures of this subsection are applicable for dry air, air that is saturated with water vapor, and air that is partially saturated with water vapor. The density of the airstream may be determined by using the Psychrometric Density Chart, Figure D1, or the Psychrometric Density Table D3, found in Appendix D, or a calculation using the *ASHRAE Handbook of Fundamentals* Chapter 1 "Psychometrics". Each of the procedures requires that the following data at the location of interest be known.

a. Barometric Pressure

b. Static Pressure

c. Dry bulb and Wet Bulb temperatures.

The procedures for determining air density are illustrated in Appendix D, Examples D.1.1, D.1.2, and D.1.3.

5.4.3.2 Although the pressure and dry bulb temperature of the air stream must be obtained in each location at which a density value is required, the wet bulb temperature is required for only one location if the air stream does not gain or lose water vapor between locations. The density at the location, for which the wet bulb temperature is not obtained, can be calculated based on the density being directly proportional to absolute pressure and inversely proportional to absolute temperature.

5.4.3.3 Example: The density at Location 1 ($dens_1$) can be established based on the test determination of barometric pressure (P_b), static pressure (P_{s1}), and dry bulb (t_{d1}) and wet bulb (t_{w1}) temperatures. The density at Location 2 ($dens_2$) can be calculated knowing the static pressure (P_{s2}) and temperature (t_{d2}) at Location 2.

The density at Location 2 is calculated as:

I-P:

$$dens_2 = (dens_1) \times [(P_{s2} + 13.6 \times P_b)/(P_{s1} + 13.6 \times P_b)] \times [(t_{d1} + 460)/(t_{d2} + 460)]$$

Where:

$dens_1$ and $dens_2$ are in lb/ft^3

P_{s1} and P_{s2} are in inches of water

P_b is in inches of mercury

t_{d1} and t_{d2} are in °F

SI:

$$dens_2 = (dens_1) \times [(P_{s2} + P_b)/(P_{s1} + P_b)] \times [(t_{d1} + 273)/(t_{d2} + 273)]$$

Where:

$dens_1$ and $dens_2$ are in kg/m^3

P_{s1} and P_{s2}, and P_b are in Pa

t_{d1} and t_{d2} are in °C

In the example, P_b is determined for the atmosphere to which the measurements of P_{s1} and P_{s2} are referred. Referring static pressure measurements to a common atmosphere is the usual practice. In the event that the static pressures cannot be referred to a common atmosphere, the absolute pressure for each location is calculated by using the static pressure measurement at the location and the barometric pressure for the atmosphere to which the static pressure measurement is referred. However, for purposes of accuracy, static pressure measurements that are used in the determination of fan static pressure must be referred to a common atmosphere.

5.4.4 Pressure

5.4.4.1 The pressures involved with air measurements are barometric pressure, static pressure, velocity pressure, total pressure, and differential pressure. One or more of these pressures is required to determine air density, airflow, resistance to airflow of system components, fan performance, and to make certain system component adjustments. In general, all measurements discussed are some variation of static pressures or velocity pressures in the system.

5.4.4.2 Experience must be used in selecting a pressure measurement location. Even the best location available in the field must be evaluated for system effects that influence the accuracy of the measurement. An example is the measurement of fan total pressure. Rarely, if ever, can the fan pressure be measured in strict accordance with AMCA-specified methods. Therefore, experienced engineers are to use this field measurement method as a guide only.

5.4.5 Barometric Pressure

5.4.5.1 A barometric pressure measurement is required in the field for a reference in determining air density.

5.4.5.2 Barometric pressure shall be measured directly using a barometer with an accuracy of 0.25% or better. Measurements at the beginning and end of a test period shall be averaged.

5.4.6 Static Pressure

5.4.6.1 All values of static pressure (design values and test values) must be referenced to the same value of atmospheric (barometric) pressure. Static pressure measurements can be either positive or negative. Positive values are those greater than atmospheric pressure. Negative values are those less than atmospheric pressure.

5.4.6.2 A straight run of duct upstream of the measurement location usually results in acceptable conditions at the location. Regions immediately downstream from elbows, obstructions and abrupt changes in airway area are generally unsuitable locations. Regions where unacceptable airflow irregularities are present shall be avoided.

5.4.6.3 Special considerations should be given in determining fan static pressure rise (See Section 7.5.7 for the definition of fan static pressure rise). It is recommended that the measurements be made at locations near the fan inlet and near the fan outlet, and that the duct between the measurement location and the fan inlet and outlet be straight and without change in a cross-sectional area. In this manner, the duct friction loss is usually insignificant, and considerations of velocity pressure conversions and calculations of pressure losses for duct fitting and other system components can be avoided.

a. In the event the fan is ducted on the outlet side, the static pressure measurement location downstream of the fan shall not be less than one equivalent diameter from the fan.

b. The static pressure measurement location upstream of the fan should not be less than 0.5 equivalent diameter from the fan inlet. In the event static pressure measurements must be obtained in an inlet box, the measurement location should be as indicated in Appendix D, Figure D7. In the case of double inlet fans, static pressure measurements must be made in both inlet boxes in order to determine the average static pressure on the inlet side of the fan.

5.4.7 Velocity Pressure: The velocity pressure (P_v) is a component of Velocity (V) and the pressure that is created due to the velocity and density of the fluid: i.e., it is a measure of the kinetic energy that exists in a moving airstream. Velocity Pressure is Total Pressure minus Static Pressure. Velocity pressure is measured as covered in Sections 5.6.3.2 and 5.6.3.3 using a differential pressure sensor or manometer.

5.4.8 Velocity: Velocity can be determined directly with sensors using non-pressure technologies, or it can be determined through calculation from Velocity Pressure. For standard conditions of temperature and barometric pressure (density), using I-P units, V = [use formula or refer to appropriate paragraph in ASHRAE 41.2].

5.4.9 Total Pressure: Total pressure is the sum of the static pressure and the velocity pressure at a given location. Total pressure is measured using a Pitot static tube or an impact tube properly connected to a manometer. The criteria for selecting an appropriate measurement location are the same as that for the measuring of velocity pressure. See Sections 5.6.3.2 and 5.6.3.3.

5.4.9.1 Fan Total Pressure (P_{tf}) is the algebraic difference between the fan outlet total pressure (P_{t2}) and the fan inlet total (P_{t1}) pressure. It is the measure of the total mechanical energy added to the air by the fan and is measured as illustrated in Figure 1.

$$P_{tf} = P_{t2} - P_{t1}$$

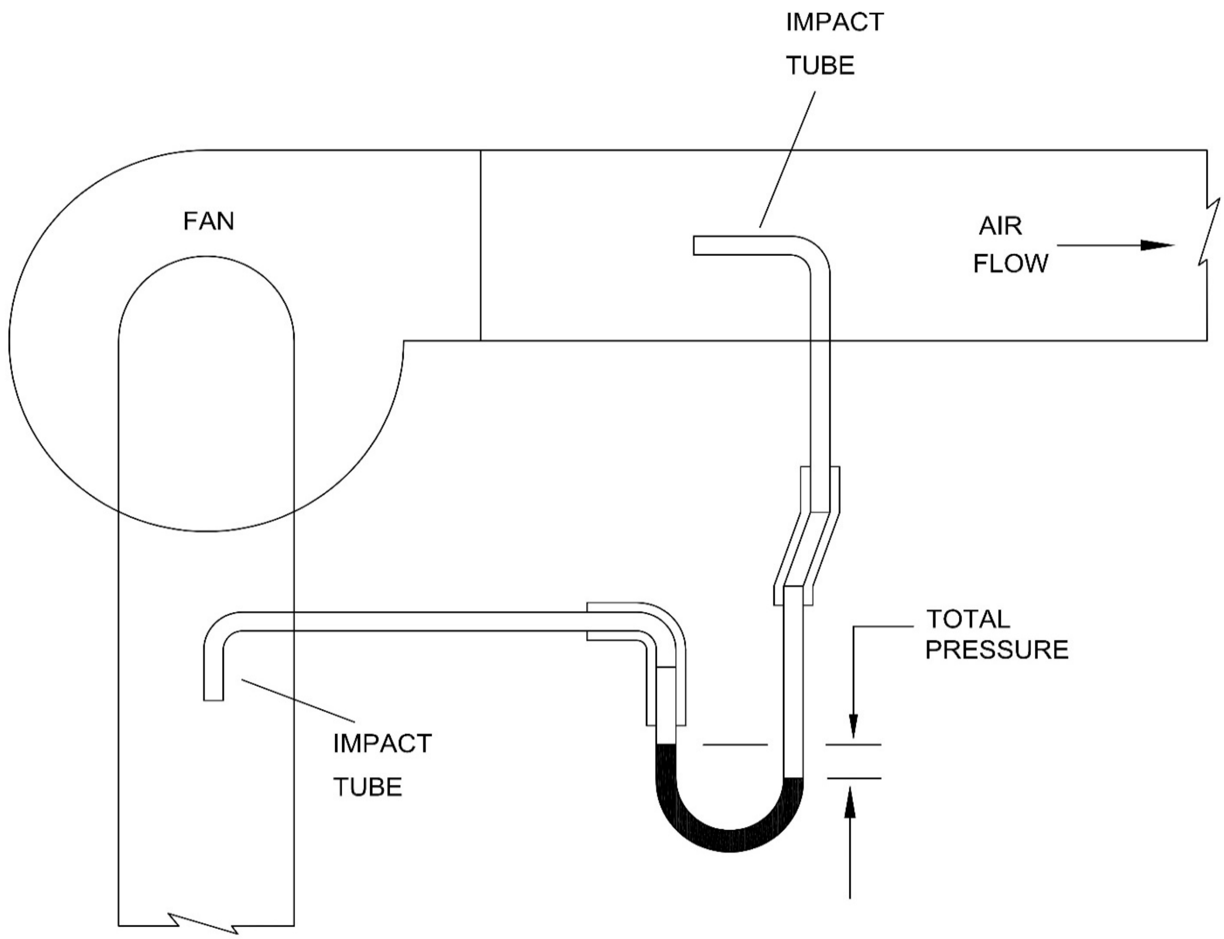
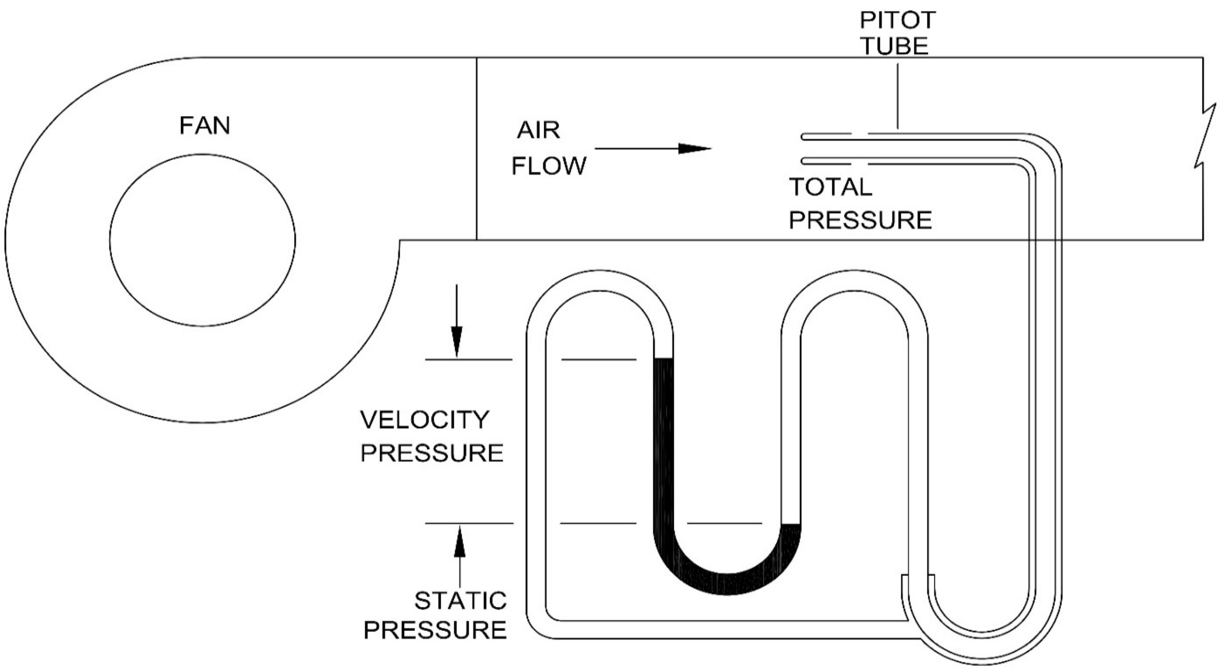


Figure 1 Fan Total Pressure (Ptf)

5.4.10 Fan Velocity Pressure (P_{vf}) is the velocity pressure corresponding to the average velocity through the fan outlet. It is the kinetic energy per unit volume of air exiting the fan and is measured as illustrated in Figure 2.



VELOCITY PRESSURE = TOTAL PRESSURE - STATIC PRESSURE

Figure 2 Fan Velocity Pressure (Pvf)

5.4.11 Differential Fan Static Pressure (P_{sf}) is the algebraic difference between the discharge fan static pressure (P_{sf1}) and the suction static pressure of the fan (P_{sf2}). Differential Pressures: differential pressure is the difference in static or total pressure across a device mounted in an air stream. The differential pressure is a measure of the resistance to airflow of a device. Fan static pressure is measured as illustrated in Figure 3.

P_{sf} = Fan Discharge Static Pressure (P_{sf1}) - Fan Suction Static Pressure (P_{sf2})

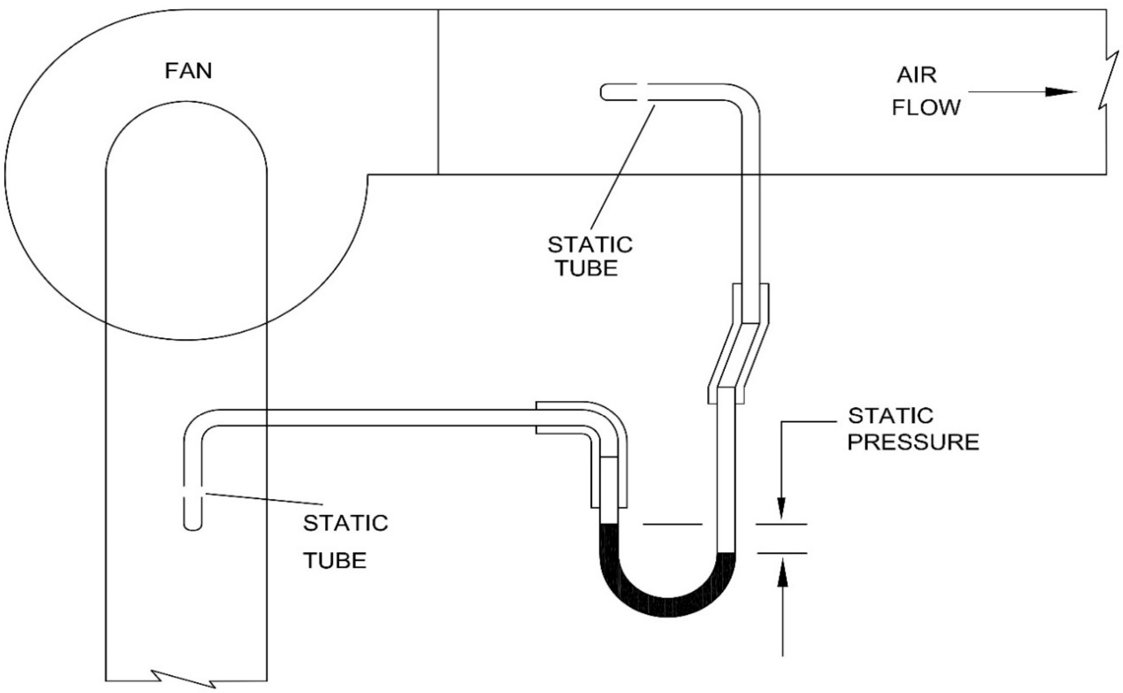


Figure 3 Fan Static Pressue (Psf)

5.4.11.1 Fan Static Pressure Rise is the algebraic difference between the fan inlet total pressure (P_{t1}) and the fan discharge static pressure (P_{sf1})

5.5 Measurements. Ideally, fan pressure measurements should be made near the fan inlet and outlet in a long straight duct of uniform cross section. In practice, this condition seldom exists, and the readings may be highly influenced by irregular airflow patterns. The interpretation of measurements under less than ideal conditions requires a complete understanding of fan system effect factors.

5.5.1 System Effects on Measurements: System effects result from conditions that cause pressure measurement errors. The poor selection of a measurement plane and excessive turbulence are the most common causes of field measurement errors.

5.5.2 Accuracy

5.5.3 Field test measurement accuracy for air pressures will range between 5% and 10% for ideal conditions. Much greater error can be expected when an available measurement location involves irregular airflow patterns.

5.5.3.1 Fan pressure measurement locations in the field usually have less than ideal conditions. When locations other than the fan inlet or outlet are used, pressure losses in ducts, fittings, and other devices must be accounted for. System effect factors for fan pressure losses must also be understood and accounted for.

5.6 Flow Rate

5.6.1 The recommendations in this section will apply to new and existing HVAC systems. It is assumed that an HVAC system will contain airways (ducts) suitable for flow measurement. Research and state of art allow the traversing of ducts as a field method for fan or system performance measurement and measuring coil velocities.

5.6.2 Flow rate is determined by using the area and the average velocity at a traverse plane. Location and definition of the area used with a velocity is explained in each subsection;

Where:

$Q = V \times A$

Q = Flow rate in CFM (m^3/s)

A = Cross sectional area at the traverse plane in ft^2 (m^2)

V = Average velocity in FPM (m/s)

Instrumentation is reviewed in each subsection and is detailed in Section 4.

5.6.3 Flow in Ducts

5.6.3.1 Instruments: The instruments recommended for use in measuring velocity pressure include a Pitot static tube and an inclined manometer or electronic instruments of comparable accuracy, or a permanent airflow stations with superior accuracy (these are described in Section 4).

5.6.3.2 The velocity pressure at a point in an air stream is numerically equal to the total pressure less the static pressure and is measured with the Pitot static tube connected to the inclined manometer as shown in Figure 2.

5.6.3.3 Location of Traverse Plane: The qualifications for a traverse plane that are considered suitable for manual velocity pressure measurements used in determining flow rate are as follows:

a. An ideal distribution would have 80% to 90% of the measurements greater than 10% of the maximum velocity. The worst distribution that could still be considered is when no less than 75% of the velocity pressure measurements are greater than 10% of the maximum velocity. (See Appendix D, Figure D8). However, when less than ideal distribution is encountered, traverse points must be maximized, and accuracy will be adversely affected. The traverse location is unacceptable if 0 or negative velocities are recorded.

b. The flow streams should be at right angles to the traverse plane. It is recommended that variations from this flow condition as a result of swirl or other mass turbulence be considered acceptable when the angle between the flow stream and the traverse plane is within 10° of this right angle.

c. The angle of the flow stream in any specific location is indicated by the orientation of the nose of the Pitot static tube that produces the maximum velocity pressure reading. However, when making a duct traverse, the nose of the Pitot static tube is held parallel to the side walls of the duct and pointing into the airflow.

d. The cross-sectional shape of the duct in which the traverse plane is located should not be irregular. Proper distribution of traverse points and accurate determination of the area of the traverse plane is difficult to achieve when the airway in which the traverse plane is located does not conform closely to a regular shape (round, oval, or rectangular).

e. Leakage must be considered when evaluating the accuracy of the traverse.

f. A location in a long, straight run of duct of uniform cross section will usually provide acceptable conditions for the traverse plane. In locating the traverse plane close to a fan, as is often done to minimize the effect of leakage, flow conditions upstream of the fan are usually more suitable. In some installations, more than one traverse plane may be required in order to account for the total flow. Also, more than one traverse plane location per system may be used to substantiate accuracy of the system performance.

g. In any installation in which a field test is anticipated, particularly when the requirement for a field test is an item in the specifications, provision should be included in the system for a suitable traverse plane location.

h. In any instance in which the fan is ducted on the outlet side and the traverse plane is to be located downstream from the fan, the traverse plane should be situated at enough distance downstream from the fan to allow the flow to diffuse to a more uniform velocity distribution and to allow the conversion of velocity pressure to static pressure. The information presented in Appendix D provides guidance for the location of the traverse plane in these cases. (See Figures D6 and D7). The location of the traverse plane on the inlet side of the fan should be more than 0.5 equivalent diameter from the fan inlet. In any case in which the traverse plane must be located within an inlet box, the plane should be located a minimum of 12 inches downstream from the leaving edges of the damper blades and more than 0.5 equivalent diameter upstream from the edge of the inlet cone.

i. Regions immediately downstream from elbows, obstructions, and abrupt changes in airway area are not suitable traverse plane locations. Regions where unacceptable levels of swirl are usually present should be avoided, such as the region downstream from an axial flow fan that is not equipped with straightening vanes.

5.6.4 The Traverse: to determine the velocity in the traverse plane, a straight average of individual point velocities will give satisfactory results when point velocities are determined by the following rule.

5.6.4.1 The Equal Area Rule is shown in Figure D9 for traverses in a rectangular duct.

5.6.4.2 The Log-Tchebycheff Rule (ISO/3966) is shown in Figure D9A. *Research Project 1245 (Reference 13.13) Determining the Effects of Duct Fittings on Volumetric Air Flow Measurements* (pp. 95 - 97) concluded the following: “The preponderance of evidence indicates that the Log-Tchebycheff (LT) traverse method, when used in rectangular ducts, will yield more accurate results than the Equal Area (EA) method by a consistent 3% to 4% of reading.” “In the absence of any flow reversals, in-duct traverse measurements with a thermal anemometer yield results that are comparable in accuracy to measurements with a Pitot-static probe. However, the Pitot-static probe has the ability to detect "apparent" negative velocity pressure values (i.e., negative pressure differences between the stagnation and the static pressure taps) which identify unacceptable traverse plane locations. Hence, the Pitot-static probe may be the preferred tool for in-duct traverse measurements.” “Current recommendations (*ASHRAE Fundamentals*, Chapter 37, Paragraph 6.3) state that "If negative velocity pressure readings are encountered, they are considered a measurement value of zero and calculated in the average velocity pressure." From the results of this research, this recommendation is completely wrong. The directional characteristics given in Figure 6.31 indicate that negative velocity pressure readings can result if the flow is misaligned with respect to the Pitot-static probe axis by ±55° to ±115° or ±150° to ±180°. It is now clear that negative velocity pressure readings indicate a totally unacceptable traverse location, and corresponding errors in flow value can subsequently exceed 50% of reading if they are used in a volumetric flow rate measurement. A different traverse location where no negative velocity pressure readings occur MUST be used. It is impossible to calculate flow values by including negative velocity pressure readings (even considering them to be zero values) without giving rise to unacceptably large errors in the flow results."

5.6.5 Circular Ducts

5.6.5.1 Appendix D, Figure D10A shows the measuring points for a circular duct traverse using the Log-Linear Rule and three symmetrically disposed diameters. Points on two perpendicular diameters may be used where access is limited.

5.6.6 Oval Ducts

5.6.6.1 Appendix D, Figure D10B shows the measuring points for a commonly accepted flat oval duct traverse.

5.6.7 Accuracy: Certification of airflow rates to specifications is the most difficult field measurement that the TAB engineer must perform. Most TAB procedures require measurements in the ducts as the most accurate documentation of system performance. These measurements are at one location; therefore, proper analysis of losses (duct leakage) must be considered. The duct traverse as outlined in this section is the only accepted method for field testing flow rate performance of fans. Error analysis shows that flow rate determinations by this method can range from 2% to 10% error. Experience shows that qualified technicians can obtain measurements that range between 5% and 10% accuracy of actual flow under good field conditions. The following rules will maintain the accuracy of the traverse:

5.6.7.1 No reading shall be less than zero.

5.6.7.2 75% of the velocity pressure readings shall be greater than 1/10 of the maximum velocity pressure reading.

5.6.7.3 The airstream should be at right angles to the traverse plane. The Pitot static tube position at the maximum velocity pressure for any location should be within 5o of perpendicular to the traverse plane. The cross-sectional area and shape of the duct shall be uniform in the vicinity of the traverse plane. The traverse location shall not be in a transition. The traverse plane should be in a straight run of duct, sufficiently upstream and downstream of elbows, transitions, fans, etc. When the design engineer deems it necessary to field measure performance to a greater accuracy, installation of permanently fixed airflow stations must be considered.

5.6.8 Airflow Rate: After Dampers and After Conditions that Cause Irregular Flow

5.6.9 TAB personnel are often faced with having to measure airflow rates when no suitable ductwork exists for traversing (i.e., coil banks of walk in fan rooms, outside air inlets with louvers or dampers, ceiling return plenums with frame dampers, etc.). The instrument usually chosen for these measurements is the rotating vane anemometer.

5.6.10 Airflow Rate at Coils: Investigations by Sauer and Howell have led to the development of a procedure for measuring airflow rates at the downstream faces of cooling and heating coils using a rotating vane anemometer (See Reference 13.11). A calculated correction factor, unique to each coil installation, is applied to the indicated air velocity to correct for the Venturi-like effect of the air passing through the coil. Research shows that this correction factor is a function of the measured air velocity (MV), number of coil rows (ROWS), number of fins per inch (FPI), tube spacing (SP), and the tube outside diameter (OD). The research has developed correction factors to be applied for three different size anemometer heads: the 4-inch (100 mm) diameter, the 2.75-inch (69 mm) diameter, and the 1-inch (25 mm) diameter. Accuracy should be within ±7% of actual airflow using this procedure which includes ±1% to 2% instrument error. This procedure compares favorably with Pitot static tube measurement accuracy and is applicable within a range of face velocities from 200 to 1,500 FPM. A Pitot static tube traverse is not practical below 600 FPM velocity. Specific procedures for calculating the conversion factor and the standard and actual airflow volumes are contained in Appendix D4.

5.6.11 Flow Rate Approximation by Temperature Ratio: Some components of system airflow are virtually impossible to measure with an anemometer or Pitot tube. For example, outside air measurements are affected by lack of ductwork and unpredictable turbulence (i.e., after louvers). Outside airflow rate can be determined if the total supply volume is known from a duct traverse. The method involves temperature measurements of the outside air, the return air, and the supply air (mixed air) using the following formula:

$$\% \, OA = (T_{RA} - T_{MA}) / (T_{RA} - T_{OA})$$

and

$$\% \, OA = (T_{OA} - T_{MA}) / (T_{OA} - T_{RA})$$

Where:

T_{RA} = Temperature Return Air

T_{MA} = Temperature Mixed Air

T_{OA} = Temperature Outside Air

This equation can be further generalized and solved such that any two volumetric components of a three-component mixed air stream can be determined if the air streams differ in temperature and the volume of one air stream can be measured.

5.6.11.1 Accuracy: Accuracy of the temperature ratio method depends on the turbulence present to create a non-stratified airstream resulting in a more accurate temperature determination. Accuracy also depends on the relative temperature differences between the streams to be mixed. Too low a difference will magnify a temperature error and too large a difference will introduce error due to the density difference (i.e., temperature difference of 25°F to 50°F / 14°C to 28°C is acceptable). Overall, the accuracy of the flow rate being determined is dependent on the accuracy of the duct traverse and the temperature averaging measurements of the three airstreams. Under good field conditions, the flow rate determination of the unknown airstreams should be within 10% of the actual condition, but narrow temperature differentials between the airstreams, temperature sensor error, and temperature sampling errors can produce significant errors. This method should only be used to produce estimates of airflow rates for diagnostic or preliminary purposes.

5.7 Flow Rate at Intake and Discharge Openings and Grilles: Discharge opening and supply grille measurements are limited to the use of the rotating vane anemometer, since research indicates it is reliable, accurate, and repeatable for field use if properly applied, which involves the use of appropriate correction factors. More research is needed to determine the limitations of instantaneous type devices, such as swinging vane anemometers and hot wire devices. The rotating vane anemometer is now available as an electronic instantaneous reading device of high accuracy. Intake openings that are provided with thermal dispersion sensor arrays and installed per manufacturer guidelines can also provide very accurate and reliable measurements. Having been used in the field for over 20 years, a body of data is available supporting both their usage and their abilities to perform measurements where other instruments are incapable.

5.7.1 Procedures: To obtain the air velocity from the readings of an air velocity meter, a correction factor must be applied in addition to the corrections for instrument calibration.

5.7.1.1 Flow at Air Diffusion Devices

a. Effective area shall be determined by measuring the actual flow rate to the diffuser by duct traverse and dividing this flow rate by the average velocity measurement from the outlet.

b. Capture hoods are also a common measurement method at diffusers. They can give repeatable readings for proportioning airflow. The same cautions described above disallow use for certification of airflow. Its potential application in the field is very diverse, and each application can alter the diffuser performance with a system effect. For actual flows, the hood must be field calibrated against a duct traverse for each typical situation.

5.8 Flow Rate at Volume Pressure Assemblies

5.8.1 These devices have many functions such as constant volume boxes, VAV boxes, constant pressure boxes, blending or mixing boxes, etc. The device is typically located ahead of supply air diffusers between the diffuser and the branch ducts of the system.

5.8.2 Manufacturers typically label these devices with tables that give various pressure, velocity, and flow rate specifications. The device may also be factory calibrated for maximum and minimum airflow rates. The specification table applies only to the device and does not include effects of downstream ductwork.

5.8.3 The TAB technician must be cautioned about using the device specification to determine field flow rates. If ductwork entrance and exit conditions do not approximate the same situation as the manufacturer test set up, the specification flow rate can be seriously altered.

5.9 Field certification of the flow rate must be established by acceptable duct traverses.

5.9.1 System Fan/Flow Rate Equations

5.9.1.1 Fan Equations (I-P Units)

$CFM_2/CFM_1 = RPM_2/RPM_1$

$P_2/P_1 = (RPM_2/RPM_1)^2 \times (d_2/d_1)$

$BHP_2/BHP_1 = (RPM_2/RPM_1)^3 \times (d_2/d_1)$

RPM (fan)/Fan motor pulley = Pitch diameter motor pulley/Pitch diameter fan pulley

Where:

CFM = cubic feet per minute

RPM = revolutions per minute

P = static or total pressure (in. wg)

BHP = brake horsepower

d = density (lb/ft^3)

5.9.1.2 Fan Equations (SI Units)

$(L/s_2)/(L/s_1) = (m^3/s_2)/(m^3/s_1) = (rad/s_2)/(rad/s_1)$

$P_2/P_1 = [(rev/s_2)/(rev/s_1)]2\ (d_2/d_1)$

$kW_2/kW_1 = [(rev/s_2)/(rev/s_1)]^3\ (d_2/d_1)$

Where:

L/s = liters per second

m^3/s = cubic meters per second

rad = radians per second

P = static or total pressure (Pa)

rev/s = revolutions per second

kW = kilowatts

5.9.1.3 Flow Rate Equations (I-P Units)

Q = V x A

V = velocity (FPM)

$V = 1096 (P_v/d)^{1/2}$

P_v = velocity pressure (in. wg)

d = density (lb/ft^3), (for Standard Air, d = 0.075 lb/ft^3)

A = area (ft^2)

5.9.1.4 Flow Rate Equations (SI Units)

Q = V x A

V = velocity (m/s)

$V= 1.414 (P_v/d)^{1/2}$

P_v = velocity pressure (Pa)

d = density (kg/m^3), (for Standard Air, d = 1.2 kg/m^3)

A = area (m^2)

5.10 Heat Content

5.10.1 General

5.10.2 This subsection presents methods for determining the heat content of an airstream.

5.10.3 The performance of system components that add heat to or extract heat from the airstream can be determined by calculating the difference in the heat content of the airstream entering and leaving the component.

5.10.4 The heat content of an airstream consists of sensible heat and latent heat. The sum of these two equals the total heat content of the airstream.

5.10.5 Instruments

5.10.5.1 Thermometers for measuring dry bulb and wet bulb temperatures.

5.10.5.2 Psychrometric Charts

5.10.6 Data Required

5.10.6.1 The heat content of the airstream, enthalpy, is usually expressed as Btu/lb of dry air. Heat content cannot be measured directly but can be determined from the dry bulb and wet bulb temperatures of the airstream using a psychrometric chart.

5.10.7 Test Method

5.10.7.1 Obtain the dry bulb and wet bulb temperatures of the airstream at the point where it is desired to determine the heat content. Find the intersection of the constant dry bulb and wet bulb lines for the temperatures measured on the psychrometric chart. Determine the value of the enthalpy line that passes through this intersection. The value of this enthalpy line is a measure of the heat content of the airstream in Btu/lb of dry air at the desired point.

5.10.7.2 It should be noted that a psychrometric chart is applicable for a given barometric pressure. If the barometric pressure at the test site is appreciably different than that of the psychrometric chart being used, an appropriate correction should be made to the enthalpy value obtained from the chart. The barometric correction can be determined for any psychrometric chart drawn for 29.92 inches HG barometric pressure using Figure 4, Barometric Enthalpy Corrections.

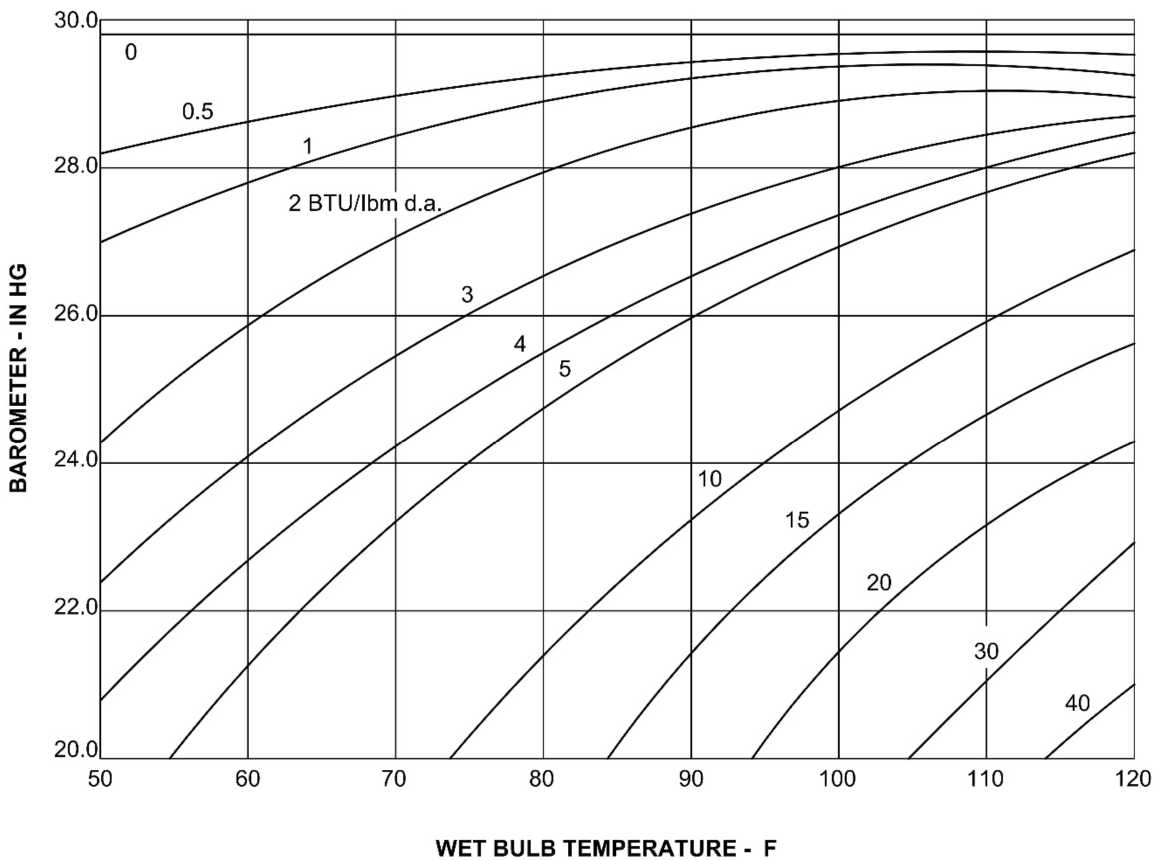


FIGURE 4 Barometric Enthalpy Corrections

5.11 Humidity

5.11.1 General: The state of an air water vapor mixture is completely defined by specifying the pressure, temperature, and humidity. Humidity refers to the amount of water vapor present in an air water vapor mixture. The two principal measures of humidity are relative humidity (RH) and humidity ratio (W).

5.11.2 Definitions

5.11.2.1 Relative Humidity: RH is the ratio of the vapor pressure existing (p_w) compared to the vapor pressure at saturation (p_s) for the same dry bulb temperature (and is usually expressed in "%").

$RH = p_w/p_s$

5.11.2.2 Humidity Ratio: W is the actual weight of water vapor existing (W_v) per unit mass of dry air (W_{da}). Most psychrometric charts express this as pounds of water vapor per pound of dry air or grains of water vapor per pound of dry air.

$W = (W_v/W_{da} = (0.622)\ [(p_s) \times (RH)] / [p_b - (p_s) \times (RH)]$

Where:

p_b = barometric pressure (in. Hg)

p_s = vapor pressure at saturation (in. Hg)

5.11.3 Instruments

5.11.3.1 Battery powered hygrometer

5.11.3.2 Powered dew point indicator

5.11.3.3 Powered electrical and/or electronic psychrometer

5.11.4 Humidity Determinations

5.11.4.1 The hygrometer is the only device that produces a direct reading of RH. The other devices produce dry bulb or wet bulb temperature values, or both, that are then used to obtain values of humidity.

5.11.4.2 To determine the humidity when using a dew point indicator, find the dew point indicator value on the saturation line of the psychrometric chart. The humidity ratio line that intersects the saturation line at this point yields the value of W. To determine RH, the airstream dry bulb temperature (at the location where RH is desired) must be measured. Using the psychrometric chart, the intersection of the dry bulb line and the W line, as determined previously, will yield the RH.

5.11.4.3 The intersection of the wet bulb temperature line and the value of the test site barometer will yield a W correction value to be added to the value of W obtained from the psychrometric chart drawn for 29.92 inches HG. Corrected values of RH can then be determined using the equation below:

$W = 0.622\ [(p_s) \times (RH)] / [p_b\ (p_s) \times (RH)]$

Rearranging terms,

$RH = WP_b/[(0.622 + W) \times (p_s)]$

Where:

W = corrected value of humidity ratio (determined as noted above)

p_b = barometric pressure at test site

p_s = vapor pressure for water (at saturation) at the dry bulb temperature

5.12 Fan Power Determination

5.12.1 General

5.12.2 Fan power is defined as either:

5.12.2.1 Power input to the fan shaft, or

5.12.2.2 The total of the power input to the fan shaft and the power transmission loss

5.12.2.3 When comparing fan power data from field tests to the fan manufacturer rated performance characteristics, the values of fan power determined from field tests should be obtained on the same basis as that used by the manufacturer in determining the fan power rating. The basis for the fan power should accompany the rated fan power information. If it does not, it should be obtained from the fan manufacturer. Rated fan power for belt driven fans may or may not include belt drive losses. In most instances where a power transmission loss occurs, the loss must be determined and subtracted from the motor power output to obtain power input to the fan shaft.

5.12.3 Instruments

5.12.3.1 The higher levels of accuracy, measurements of current, voltage, watts, and power factor can be obtained by using a quality industrial type power analyzer. Such instruments are available with accuracies of 1% full scale for volts, amps, and power factor and 2% full scale for watts. In many cases, accuracy level requirements will permit the use of a clamp on type volt amp meter with accuracies of 3% full scale.

5.12.4 Test Methods: Since accuracy requirements for field test determinations of fan power input vary considerably, a number of test methods are recommended. These methods are intended to provide economical and practical alternatives for dealing with the various levels of accuracy requirements.

5.12.5 Phase Current Method

5.12.5.1 This is a method for estimating the power output of three-phase motors, based on the relationship of motor current and motor power output. The method, described in Appendix D, Example D3 and Figure D11, requires measurements of the phase currents and voltages supplied to the motor while driving the fan. Depending on the operating load point of the motor, it may also involve the measurements of the no load phase currents.

5.12.5.2 This method is convenient and sufficiently accurate for most field tests where the motor operates near full load; the closer the actual phase current is to the motor nameplate value of full load amps, the greater the accuracy. Since fan motors are normally selected for operation at or near the full load point, this method provides a reasonably accurate estimate of the power output of the fan motor. Fan power input is determined by using the motor power output and, where applicable, the power transmission loss. For motors that are not operating near full load, this method can lead to gross inaccuracies. (See Appendix D.3 for further explanation).

5.12.5.3 Typical Motor Performance Data

a. Typical motor performance data may be used in the determination of fan power input. These data, which are referred to as typical in that the data and the actual performance of the motor are expected to correspond closely, can usually be obtained from the motor manufacturer. The data provided can be in a variety of forms but are sufficient to determine motor power output based on electrical input measurements. It is important that the power supplied to the motor during the field test be consistent with that used as the basis for the motor performance data. The phase voltage should be stable and balanced, and the average should be within 2% of the voltage indicated in the performance data.

b. Depending on the form of the typical motor performance data, motor power output is determined by one of the following methods.

5.12.5.4 The motor power output, HP, is the value in the typical motor performance data that corresponds to the field test measurement of watts input to the motor.

5.12.5.5 Using the field test measurement of watts input and the corresponding typical motor performance data values of torque output and speed, the motor power output is calculated as:

BHP = (T x RPM)/63025

Where:

T = torque output (in in-lb)

RPM = motor speed

5.12.5.6 Using the field test measurement of watts input and the corresponding typical motor performance data value of motor efficiency, the motor power output is calculated as:

BHP = (watts input x motor efficiency)/746

5.12.5.7 Using the field test measurements of amps, input, and volts, the typical motor performance data values of power factor (pf) and motor efficiency corresponding to the measured amps input, the motor power output is calculated as:

For Single Phase Motors: BHP = (amps x volts x pf x motor efficiency)/746

For Three Phase Motors: BHP = (1.73 x amps x volts x pf x motor efficiency)/746

In both equations, amps and volts are the field test measurement values and, in the case of three-phase motors, are the averages of the measured phase values.

5.12.5.8 The fan power input is determined by using the motor power output and, where applicable, the power transmission loss.

5.12.5.9 Specific manufacturer motor performance data for the motor being tested may be used to estimate actual operating horsepower at load. Most motor manufacturers publish performance curves for each style and size motor they manufacture. These curves show the relationship between actual operating horsepower, motor speed (RPM), motor efficiency at load, and motor power factor at load.

From field measurements of motor electrical input kilowatts, voltage, and amperage, the following relationship can be used to determine motor power factor:

kW = (1.73205 x V x A x pf)/1000

Rearranging terms,

pf = (kW x 1000)/(1.73205 x V x A)

Where:

kW = three-phase kilowatts

V = three-phase voltage

A = average amperage across all three phases

Note: for single-phase motors, the factor 1.73205 becomes 1.0.

Using the calculated power factor and/or the measured motor RPM, determine the estimated motor efficiency and the percent of full load horsepower from the manufacturer motor performance curves.

5.12.6 Power Transmission Losses

5.12.6.1 Several types of power transmission equipment are used in driving fans. Those in which power transmission losses should be considered in the determination of fan power input include belt drives, gear boxes, fluid drives, and electromechanical couplings.

5.12.6.2 Information as to whether the fan power input ratings include power transmission losses is included in the published fan performance ratings or is otherwise available from the fan manufacturer. It is important that this be established and that the fan power input be determined accordingly in order to provide a valid comparison of field test results to the fan performance ratings. In most cases, fan power input ratings do not include power transmission losses.

5.12.6.3 In view of the lack of published information available for use in calculating belt drive losses, a graph is included in Appendix D, Figure D12 for this purpose. As indicated in the graph, belt drive loss, expressed as a percentage of motor power output, decreases with increasing motor power output and increases with increasing speed. This graph is based on experience and a limited amount of test data, and some departure from this graph may be expected. However, it serves as a reasonable guide in evaluating belt drive losses, including variable speed drives.

5.12.6.4 For other types of power transmission equipment, it is suggested that the fan manufacturer be consulted to establish whether transmission losses are included in the fan ratings, and if so, the magnitudes of the losses allowed in the ratings. Otherwise, it will be necessary to consult the manufacturer of the power transmission equipment for the information regarding transmission losses.

6. Hydronic Systems

6.1 Design Requirements: Hydronic System Test, Adjust, and Balance specification shall include by the system designer the following as minimum;

6.1.1 Terminal Heat Transfer Devices Design: Flow rate, entering water temperature, differential temperature, pressure drop, fluid working conditions.

6.1.2 Flow Tolerance: The required flow of fluid to provide the minimum acceptable heat transfer from the terminal device. Required flow tolerance shall allow flow deviation of no greater than ±10% fluid terminal flow. Minimum acceptable heat transfer shall be no less than 97.5% heat emission at simulated design flow operating conditions. Required flow tolerance shall be calculated from heat exchanger manufacturer data and characteristic noting the minimum percentage of design flow required to achieve 97.5% heat transfer at design operating conditions.

6.1.2.1 The designer shall also specify part load operating requirements and required flow percentages for required heat transfer.

6.1.2.2 By default, the hydronic system design shall be expected to operate at part load providing no less than 97.5% of the part load required heat transfer unless specified differently. It shall be the responsibility of the designer to specify pipe sizes and pressure losses and demonstrate that the piping network provides the required pressure drop relationships between distribution piping and terminal piping to reduce inter branch flow hydraulic interference due to control valve operation.

6.1.3 Hydronic System Schematic Drawing: Schematic piping diagram shall show pipe flow connections starting at pump and progressing to each heat transfer terminal through main distribution piping, sub-distribution piping, and branch distribution piping. Each illustrated pipe shall specify pipe sizes, designated design flow, friction loss rate in piping, pipe length, design pressure loss and flow data at branch distribution points and major branch components such as control valves, terminals, flow measurement devices, and flow adjustment devices.

6.1.4 Pressure Measurement and Flow Measurement Points: All heat transfer terminal branches shall be equipped with a fluid flow measurement device causing a fixed differential pressure difference as follows:

6.1.4.1 Branches serving one terminal greater than 10 GPM or greater design flow rate.

6.1.4.2 Branches serving more than one but less than ten terminal devices with a combined design flow of 15 GPM or less.

6.1.4.3 The fluid flow measurement device shall cause a differential pressure signal no less than 2 psig in differential pressure measurement at the terminal design flow rate which may be related to calculate the measured flow rate. The device shall be permanently labeled with design flow rate and expected signal pressure drop in psig, manufacturer name, and model number. Exception: Non-Intrusive sensing devices permanently fixed to piping utilizing sonic or other high frequency measurement methods.

6.1.4.4 All distribution pipe transition points to and from sub-distribution circuits shall be provided with pressure measurement points utilizing a self-sealing hypodermic access point for pressure or temperature measurement.

6.1.4.5 Measurement points shall also be provided for Entering and leaving points of pumps and pump trim devices such as suction diffusers, discharge valves, or piping strainers.

6.1.4.6 Control system sensor measuring locations

6.1.4.7 Heat Exchangers, Boilers, Chillers, System Pressure Control vessels, and other system source and load devices.

6.1.5 Specification of Testing and Balancing Methods and devices

6.1.5.1 The designer shall specify required hydronic system component and system tests. General specification by the designer of the application of *ASHRAE Standard 111*, without specific reference or requirements, shall invoke application of minimum standard requirements as herein specified.

6.1.5.2 Unless specified by default, the designer shall specify the method of system adjustment per one of the following methods.

6.1.5.3 Proportional Balance: all connected terminal loads are manually adjusted to provide terminal path head losses to be equal with respect to the pump and the flow required for terminal heat transfer flow tolerance. Proportional flow balance shall ensure that all terminals receive the same percentage of flow as capable of being produced by the pump and limiting the total pump output to no more than 10% greater than the block connected flow load.

a. Terminals shall be equipped with a flow adjustment valve, which may be separate from the flow measurement device.

b. Hydronic systems with multiple distribution risers, serving multiple terminal units per riser from a common distribution pipe, shall provide adjustment valves for the riser, floor, and terminal.

6.1.5.4 Reverse-Return: System piping shall be considered a proportional balance method so long as the designer can demonstrate that connected terminals stay within the required flow tolerance, operate in similar manner under similar operating conditions, and are part of the same temperature control zone. Systems with multiple distribution risers, serving multiple terminal units per riser from a common distribution pipe, shall provide adjustment valves for the riser and each floor. Terminal valves are recommended.

6.1.5.5 Pipe Pressure Drop: Adjustment in Direct Return system piping shall be considered a proportional balance method so long as the designer can demonstrate the fluid flow interaction and there is a terminal path head loss adjustment device to allow adjustability to the required flow tolerance if necessary.

6.1.5.6 Proportionally balanced systems shall be measured while in operation at the provided flow measurement device. Adjustment shall be made through the provided adjustment device using either field measurement and calculation, numerical calculation from design drawings, or a combination of both. Pre-calculation and adjustment of required valve settings shall be an acceptable method of proportional balance but shall require field measurement and verification of flow in operation.

6.1.5.7 Maximum Flow Limiting: All connected terminal loads are adjusted manually or automatically to provide no more than terminal design flow and the flow tolerance with a maximum deviation of ±10%.

a. Provide Automatic Flow Limiting valve and cartridge set for the terminal design flow rate. Pressure control mechanism shall control over the entire differential head range of the provided pump.

6.1.5.8 Pressure Independent Control: Each terminal control valve or branch is provided with a differential pressure control regulator (either system or externally powered) to limit terminal maximum flow rate and stabilize temperature control flow response. Differential pressure regulators may be provided as integral or separate devices to the terminal temperature control valve and shall be adjustable to the required differential pressure.

6.1.5.9 Alternate methods of balance shall be acceptable through calculation and demonstration of field flows, providing required flow for comfort heat transfer flow tolerance at start-up and operating conditions and during simulated partial loading with respect to flow. All terminals shall be measurable during flow simulation. All terminal temperature controllers shall be under normal temperature control. All flow and temperature control deviations shall be reported.

6.2 Design Guidance / System Effect

6.2.1 Design flow shall be measured and tested at conditions that simulate design and part load flow conditions. This requires that all service and control valves be open, all strainers cleaned, all pumps operational for design conditions, and control valves set to the specific lift required for the desired part load condition. It is noted that flow conditions may be simulated, but temperature responses associated with actual flows and loads may not produce proper results.

6.2.2 Improper design and specification may result in throttled or closed control valves significantly impacting the flow of fluid through the system due to inter-branch flow hydraulic interference. This may have a compound effect on system flow related to the magnitude of flow in each branch, riser pipe size and terminal geographical location to the pump, and to the type of pump control employed, either constant speed or variable speed under differential pressure feedback control.

6.2.3 Pipe system sizes should be selected with respect to their overall friction loss, location, and effect on the flow required for comfort/heat transfer conditioning. In general, the majority of terminal path head loss should be in the terminal branch piping, heat transfer device, control and adjustment device, not in the distribution piping transporting water to the terminal. The terminal path with the largest calculated friction loss shall be used to select the pump head required for the system. The branch pressure drop should never be less than combined pressure drops of the supply and return riser pressure loss. Branch pressure drop ratio to riser pressure drop ratio should be selected to maintain the required flow tolerance of the system.

6.2.4 Diversity Pump Selection and System Sizing Criteria: application by designers of “diversity factors” in which pumps and their maximum operable flow are capable of should be very carefully considered prior to actual field application.

6.2.4.1 In general, the application of a diversity factor often leads to the pump size being reduced, but not the corresponding chillers or pipe sizing. As a result, should improper specification of control sequences of operation or equipment be applied, it may be impossible to adjust control valves to an over-ridden part load point of operation that allows for fluid to flow to all terminals.

6.2.4.2 If diversity is applied by the designer, they must ensure that proportional balancing is employed, as well as proportional flow control devices. The designer shall specify how full flow conditions (where all valves are controlled to open in response to temperature control) shall be treated by the control system to ensure that all terminals receive the required flow for comfort control. The designer shall undertake detailed calculations to show what a simulated full block load flow condition requires for balancing device adjustment so that the balancing technician has a reference point against which to measure and report.

6.2.4.3 It is recommended that piping networks employed in single building entities with diverse system designs sub-divide the piping distribution system into common temperature control zones such as “North,” “East,” “South,” and “West.” It is recommended that these systems employ modulating proportional control, not two position control.

6.2.5 Variable Speed Variable Flow Pump Control Systems and Application. Pumping systems employ VSVF pump controls. Designer applied systems may provide only one point of measurement for control of the pump speed in relation to a zone serving as a surrogate representing the loading condition of the entire piping network, or several others.

6.2.5.1 General Sensor Application Requirements: Differential pressure shall be the controlled variable used to control pump speed. The designer shall calculate the required pump head for all branch flow paths with respect to the pump and apply the path with the greatest head loss for the selection of the pump head and system flow rate.

6.2.5.2 Preferentially, the piping network shall be arranged in a direct return piping system configuration. The supply pipe shall provide a point of constant entering water temperature. The return pipe shall provide a point of varying temperature from the connected terminal load. Branches shall be designated as the connected piping, terminal, and control devices located between the supply and return piping.

6.2.5.3 The designer shall arrange the piping network such that branches represent common heat transfer loading requirements within flow tolerance requirements and temperature-controlled zone characteristics.

6.2.5.4 The design pressure drop of the branches should be correlated to be within the heat transfer flow tolerance.

6.2.5.5 Branches or nested branches with design differential pressures or flow rates exceeding the required flow tolerance of the differential pressure control branch shall be analyzed by the designer for flow interference effect, and preferentially alternative pump or piping schemes shall be employed to reduce or eliminate their flow interference.

6.2.5.6 The differential pressure sensor measurement nodes shall be located entering the common piping from the distribution main or sub-main (riser) supply piping representing the controlled branch and the leaving piping entering the common piping of the return distribution main or sub-main.

6.2.5.7 The sensed zone shall provide typical control operation for the connected zones of the distribution piping. The sensed zone should follow the same time schedule constraints as the other connected branches. The sensed zone should cover similar loading characteristics as the other connected branches. (Example: it might be that an infrequently used auditorium or meeting room represents point of design head loss; however, when unused, the temperature control valve is closed to reduce energy consumption while other connected zones are still under operation. A normally operating zone should be employed for typical pump control with enhanced sequences of operation to cover periods when the auditorium comes into use).

6.2.5.8 Where system size or designer preference requires application of more than one differential pressure point of sensing and control, the designer shall specify their preferred arrangement of the loop control sequence of operation.

6.2.5.9 Multi-Sensor control loops preferably shall use discriminatory selection of the sensor with the greatest control deviation from setpoint for control of the pump variable speed control algorithm.

6.2.5.10 Alternatively, averaging of multiple sensors may be considered by the designer under the guidance of analyzing the system flow control points of operation with respect to flow.

6.2.5.11 Piping and pumping design and selection may lead the designer to apply a variable speed pump control strategy that employs no external differential pressure sensor requirement, relying on algorithms that translate the relationship of flow, head, and pump speed vs. pump motor energy use to determine pump control output and speed. The designer shall still designate the highest head loss branches for the proper adjustment and testing of the system.

6.2.6 In the circuit with the greatest hydraulic loss, balancing devices shall be open at design flow conditions. Balancing devices requiring a minimum differential pressure for the design flow at open condition shall have such that overflow through an open control valve allows the device to immediately begin limiting the flow.

6.2.7 In the circuit with the greatest hydraulic loss, entering pressure shall provide differential head such that the heat transfer terminal coil is flooded, air is vented, and a static pressure 4 psig greater than atmospheric pressure exists.

6.2.8 Terminals: Terminal performance is affected by

6.2.8.1 Entering water temperature, hydronic design differential temperature drops (coil mass and surface area), entering air conditions, air side differential temperature, air film coefficient, and air volume.

6.2.8.2 Properly sized control valve: Control valves should be sized with complimentary authority to the operating coil characteristic under full and part load to allow for linear proportional control. Poor valve authority will increase operating condition flow lowering operating differential water temperature, sequencing of distribution and source equipment when there is more than one, and possibly may cause the control valve to hunt at certain points of controller operation or system loading.

6.2.8.3 Stable Operation Differential Pressure: Typical proportional control algorithms rely on a design construct of the controller range of output being linked to the linear control of heat emission from the controlled terminal. This relationship implies that terminal entering water temperature should be kept stable and that available differential pressure be kept constant.

6.2.8.4 Control Valve Authority: In a simple pumped hydronic system, control valve authority provides an index number presented as a percentage calculated by dividing the pressure drop of the control valve at design flow conditions divided by the total pressure drop of the hydraulic path that it serves, which in a proportionally balanced system equals the pump head. This relationship may be modified as well to account for controlled points of differential pressure, either through variable speed pump control, application of differential pressure regulators, or decoupled pumping systems. For example, in a pump utilizing variable speed control maintaining a fixed differential pressure across the most hydraulically significant branch, the control valve authority is calculated by dividing the pressure drop of the control valve at design flow by the controlled pressure drop of the branch.

6.2.8.5 Control Valve Turndown: Control Valve Hysteresis

6.2.9 Boilers: boiler performance will be affected by:

6.2.9.1 Examine and determine if boiler includes factory mounted circulation pump, the operating characteristics and control sequence of operations, and inclusion in the attached pump and piping system network IT services.

6.2.9.2 Examine boiler pressure safety and relief valve sizing for appropriate capacity and pressure setting. Ensure that boiler operation does not cause spurious relief valve operation or leakage. Fluid leakage from relief valve may cause suspended solids in operating fluid to precipitate and coat relief valve operating components causing failure of valve to operate as intended and potentially cause exceptional damage to the boiler and surrounding components.

6.2.9.3 Examine flow switch size and settings and ensure proper integration into boiler firing controls. Instantaneous hot water boilers require immediate shut down of gas burner when liquid flow decreases below boiler minimum flow requirements where the mismatch of gas energy input is greater than waterside heat transfer forming steam or overheating and distortion of heat transfer tubes.

6.2.9.4 Boiler heat transfer output will be reduced by water precipitate scale which may be due to improper air management, chemical treatment, fire side soot deposits, inefficient combustion, or improper combustion cycle airflow.

6.2.9.5 Unsteady water lines in gauge glasses (possibly causing intermittent safety shutdown of the burner by the water level control) can be caused by priming due to grease or dirt in the boiler, erratic return of condensate (excessive boiler pressure or low pump head), or the bottom of water column being connected to a waterway carrying water at high velocity.

6.2.10 Chillers: Selected Chillers shall be provided with manufacturer data regarding the minimum required operating water velocity to the evaporator bundle, design water flow rate, minimum water flow rate, quantity of heat exchanger tubes in evaporator bundle per pass, tube nominal dimensions, chiller manufacturer recommended points of flow, and temperature measurement.

6.2.10.1 Chiller performance is affected by the following:

a. A reduction in water flow rate through either the condenser or evaporator sections can cause short cycling and possible mechanical damage.

b. Load temperature shock of the chiller can occur due to "line size" control valves (which should be sized for the controlled flow rate) which allow large flows of liquid at seasonal changeover.

c. Condenser water flow rates less than design or at temperatures above design can cause high head and automatic safety shutdown of the compressor.

d. Sequencing of multiple chillers can be affected by the manner, order, and location of the automatic temperature control valve.

6.2.10.2 Chillers shall be provided with design pressure loss versus flow and partial flow pressure losses in operating characteristic curve.

6.2.11 Cooling Towers and Air-Cooled Condensers: Cooling tower capacities are adversely affected by:

6.2.11.1 A centrifugal or propeller blower operating backwards or in reverse causing low airflow and the fan motor to draw near full load amps.

6.2.11.2 Sump outlet clogged with debris.

6.2.11.3 Pipeline strainer dirty or with a strainer basket having mesh that is too fine.

6.2.11.4 Low net positive suction head (NPSH) on condenser water pump causing cavitation and reduced flow.

6.2.11.5 Insufficient make-up water.

6.2.11.6 Obstructed air inlet.

6.2.11.7 Diverting valve piped to pump suction pipe instead of above sump, resulting in sump overflow and air suction at pump on startup.

6.2.11.8 Outlet vortexing

6.2.11.9 Fluid short circuiting

6.2.11.10 Improper pipe sizing (too small) of sump transfer piping in conjoined cooling towers causing variable sump water heights, air entrainment in fluid, and sump overflow on automatic fill.

6.2.12 Air-cooled condenser capacities are adversely affected by

6.2.12.1 Low airflow from fans rotating in reverse.

6.2.12.2 Dirty coils causing compressor high head pressure and automatic shutdown.

6.2.12.3 Dampers or damper controls inoperative.

6.2.12.4 Short circuiting

6.2.13 Pumps

6.2.13.1 Pump flow performance characteristic curve: Field performance shall be confirmed through field testing. Determine if pump suction nozzle is same size as pump discharge nozzle. Pump nozzles may differ in size, with the discharge nozzle being one or more pipe sizes smaller than the suction. In such cases, field measurements shall correct the operating characteristic curve for velocity head and translation of field flow readings.

6.2.13.2 The flow indicated by a centrifugal pump will be different from design and published pump curves when:

a. Pump rotation is backwards.

b. Inlet piping conditions create high pressure losses or re-circulation within the nozzle entrance.

c. Net positive suction head available (NPSHA) is less than the manufacturer stated requirements.

d. System resistance to flow is different than that used to select the pump. (For pumps with a flat curve, a small change in head results in a large change in flow rate and motor load).

e. The pump impeller size is the wrong diameter.

f. When the impeller is installed backwards on a double suction pump.

g. Outlet piping conditions create turbulence and high-pressure losses.

h. Fluid viscosity is greatly different from water.

i. Head Conversion: Flow implied from pump curves utilizing pressure gauges requires conversion from psig to Ft. Head. Conversion factors shall account for the operating density of the pumped fluid.

j. Measured operating motor speed differs from published pump characteristic curves. Characteristic curves shall be re-calibrated to the field operating condition.

6.2.13.3 Measure and confirm requirements for pump suction and discharge piping.

a. Avoid and or note locating of strainers in suction piping. Application of pump suction diffusers may include a “startup” strainer being included to prevent large pipe debris from entering the pump impeller until the system has been thoroughly flushed. Remove startup strainer from pump accessory after flushing and prior to pump being placed into service and final flow testing and adjustment.

b. Avoid pump piping applications which would cause suction lift of water entering the pump. Avoid piping which introduces multiple pipes joining at the pump suction within 50 pipe diameters. Avoid piping and fitting orientations which would add swirl of the fluid to the pump suction.

c. Note pump discharge and suction velocities. Avoid pump selections with high velocities greater than ten feet per second (fps) in HVAC applications.

d. In applications of multiple pumps on a single headered manifold, note pump manifold dimensions, allowing for proper separation between pumps to avoid excess losses and potential NPSH impacts. Utilize recommendations of Hydraulic Institute Rotodynamic Pumps Guideline for NPSH Margin (ANSI/HI 9.6.1-2017).

e. Avoid and eliminate pump discharge valve throttling losses. Variable speed pump applications shall not require discharge throttling for adjustment and shall utilize limiting of pump speed to adjust pump discharge head. Non-variable speed pump applications which require the addition of discharge head through pump discharge valve throttling during system balancing shall eliminate throttling effects through adjustment of the pump impeller size for pumps ¼ HP or greater in motor size, opening the discharge valve after pump adjustment.

6.2.13.4 Pump motor overloading can occur at startup or during operation when

a. The system is not properly balanced and has excessive flow operating beyond the acceptable horsepower margin of the pump motor, speed drive or combined combination.

b. The system resistance allows the pump to operate at an unrated or unsatisfactory condition of flow performance and head considered to be off the pump flow characteristic curve, or at an operating point of low flow as to be considered within a zone of incipient cavitation for too long of an operation period.

c. Incorrect motor installed on pump.

d. Wrong impeller is installed on the pump.

e. Operating damage has occurred to the pump coupler, bearings, or shaft alignment.

6.2.14 Air Management and System Pressure Management

6.2.14.1 Examine and determine the type of Air Management and System Pressure Management system being applied.

a. Air Management systems collect air at one location and transfer it to a plain (steel) vessel acting as a compression tank for fluid expansion and contraction. This tank shall require specialized fittings for air to be directed into the vessel as a compression cushion and fluid to expand and contract into the vessel based on fluid temperature. This fitting shall provide a labyrinth pathway preventing gravity circulation and heat transfer of the controlled fluid between the tank and the system piping. Local jurisdiction building codes may require additional devices such as drain valves and sight glasses to indicate fluid level within the vessel. Sight glasses require forms of elastomeric gasketing to hold and seal the glass tube, which may “dry” out over time, allowing the seal to be broken and air to freely enter and leave the vessel. Safety consideration should enter the site selection to allow for isolation valves at the entering and leaving (top and bottom) of the connection points to the tank to prevent unsealing. Similarly, properly operating compression vessels with a warm fluid layer and saturated gas layer of collected air can cause rust on the interior tank walls, particularly at the top of the tank. Corrosion due to rust will cause pin hole leaks over time, again causing air to freely enter and leave the vessel causing a loss of expansion control and system pressure management. Localized manual air vents shall be provided to allow air discharge while filling or maintaining the system as required to eliminate entrapped air from piping collection points and prevent water flow blockage.

b. Air Elimination systems may collect air at many locations and automatically discharge it from the system utilizing and automatic air vent. A specialized expansion vessel containing an elastomeric diaphragm or bladder shall be integrated within the vessel to separate interface from the system fluid and the gaseous fill of the device which fluid will act to compress with changes in fluid temperature expansion and contraction. These devices have manufacturer rated fluid acceptance volumes for proper operation and shall be checked against the fluid constraints of piping system volume to ensure proper sizing. These devices may be filled with air or nitrogen. These devices shall always be filled with gas to a pressure equal to that required by the system design to assure internal fluid system pressures greater than atmospheric and incorporating a safety margin as determined by the system designer. Fluid pressures falling below atmospheric will open automatic air vents allowing air to re-enter the piping system. The device must be charged with gas in a dry state not connected to the influences of the filled fluid system to ensure proper pressure setting. Local jurisdiction building codes may require specific piping configurations to attach the device to the system. Connected to the system, the device gas pressure may not be checked unless isolated from the fluid system, vented to atmosphere on the fluid side of the diaphragm. Piping considerations for such testing and potential maintenance should be considered by the system designer in consultation with local jurisdiction building codes. Verify that the pressure regulator of the make-up water supply is not set too low so that automatic air vents at the highest elevations of the system do not expel air or induce air into the system.

6.2.14.2 The pipe network design shall be examined for co-mingling of more than one integral piping system and separated by heat exchanger(s). Separate system fluid fills and pressure management systems shall be provided on each side of the heat exchanger. All fluid circulation systems of any operating temperature shall be equipped with an air and pressure management system.

6.2.14.3 System piping and sources (chillers and boilers) shall be protected by a properly selected safety pressure relief valve selected to the pressure and temperature requirements of the local jurisdiction building codes, ASME Pressure Vessel Code, and the National Fuel Gas Code.

6.2.14.4 Preferentially, the pipe which connects the expansion tank to the liquid circulation main should be connected on the suction side of and close to the circulating pump. The point of tank interface to the pipe is a point of no pressure change; however, fluid friction losses due to pipe flow reduce entering pressure to the pump from the set system pressure of the tank. Expansion tank pressure remains constant whether the pump is on or off. Ignoring minor pressure losses due to friction head loss in the piping from the interface to the pump suction, all developed pump energy in the form of head will act to increase the leaving fluid pressure from the pump. Keeping operating pressures greater than atmospheric helps to eliminate

a. Cavitation in the pump and erosion of the impeller or volute.

b. Suction of air into the system at the pump shaft seals and at automatic air vents when applied.

c. Reduced or no heat transfer at heat exchangers due to air binding of entrained air at high points or fittings of the piping systems or apparatus.

6.2.14.5 Air separators shall be located at the point of warmest temperature or water lowest pressure point in the system.

6.2.14.6 Pumps shall pump into the load e.g., through the points of highest resistance such as chillers and terminals.

6.2.14.7 Confirm proper operation of the pressure relief valve to prevent damage to the system or harm to occupants and operators. Improper operation may occur when

a. Basic compression tanks are flooded or waterlogged due to insufficient air present in the tank and water expansion due to increased operating temperature.

b. Compression or expansion tanks are sized too small.

c. The liquid flow switch on an instantaneous boiler is not set properly to close the main gas valve when a reduced liquid flow occurs. This causes some fluid to turn to steam, resolution, and increased pressure.

d. Relief valves are set wrong.

6.2.15 Reduced liquid flow rates can occur when

6.2.15.1 Strainers are clogged with debris or the strainer mesh is too fine.

6.2.15.2 Generated pump head is inadequate to overcome system resistance.

6.2.15.3 Systems are improperly vented and flushed allowing for air binding in fittings and pipe transitions.

6.2.15.4 Fluid re-circulation occurs in the eye of the impeller.

6.2.15.5 A system shutoff valve has been partially closed.

6.2.16 Terminal or Coil heat transfer effectiveness will be affected by

6.2.16.1 Improper coil piping and orientation to the airstream and supply and return piping. Coil circuiting should also be examined.

6.2.16.2 Air stream particulate binding to the fin surface of the coil as either dirt, carried biological material such as pollen or seeds, suspended salt in coastal climates, recirculated exhaust gases or lint. Airflow is reduced modifying the operating heat transfer. Such conditions can occur rapidly, and in as little time as three months depending on location.

6.2.16.3 Air binding in the coil. This may occur when supply pipe connection is made above the return pipe connection or was not properly vented.

6.2.17 Known Operating Anomalies:

6.2.17.1 Cold Leg Startup: Can occur in a hot water heating system when a distribution main in a ceiling space supplies terminal on the floor level below as well as the floor level above. With balance valves in normal positions, the pressure difference between supply and return mains is not sufficient to lift the weight of the cold-water leg. (Temporary closure of the balance valves on most of the upper floor level terminals will initiate flow through the terminals on the lower floor level).

6.2.17.2 Three-way flow control valves are improperly selected and balanced causing flows greater than terminal design flow at part load conditions. Flow is short circuited from other terminals in the system. See Hydronic System appendixes on Control Valve Authority. Balancing valves shall be installed in the bypass piping of a three-way valve, and considerations given to the method of balance at part load.

6.2.17.3 Pump shutoff head versus the control valve rating should be checked to ensure the control valve has the correct pressure rating and that the operator will close against the supply pressure.

6.2.17.4 Minimum required differential pressure for proper temperature control valve and controller operation in variable speed pumping scenarios:

a. Recognition and examination shall be given to the relationship known as “Pump Control Area” and the effects of non-sequential control valve positioning on other branches flow, balancing and differential pressure control.

b. Recognition and examination shall be given to theoretical energy saving strategies that would reset the pump operating minimum differential pressure setpoint lower than that required to provide design flow and heat transfer within correlation of the temperature controller operating adjustments (For Example; P, I, D). In addition, control valves that require a minimum operating differential pressure such as Pressure Independent Control (PIC) Valves shall be examined so that enough differential pressure is available to provide for their proper operation and not cause hunting and water hammer from the differential pressure control element of the PIC style valve.

c. All control valves, including PIC valves shall be laboratory performance tested by the manufacturer to International Society of Automation Standard 75 series requirements to establish unitary flow and pressure losses, and conformance to characteristic specifications. In addition, valves with internal or external pressure regulators such as PIC valves shall have the operating flow and head characteristic verified and published individually for each incremental flow rate range of valve operation. For example, in a PIC valve rated for selectable operating flow ranges from 1 to 15 GPM over a gross operating range of 2 to 32 psig, a valve selected by the designer to operate from 0 to 2 GPM shall be provided with a flow characteristic curve demonstrating that when set for 100 % open, a flow of 2 GPM is attained at 2 psig differential pressure across the PIC valve body and maintained at 2 GPM to the rated limit of operating differential pressure at 32 psig. Similarly, incremental positions of less than 2 GPM at 10% increments from 0% stroke (closed) to 100% stroke (open) generate specified flows within the manufacturer specified control range of 2 psig to 32 psig differential across the valve body. Valve performance shall be tested both opening and closing demonstrating valve flow hysteresis.

d. Control valves shall be operated under pump differential pressure control, and not under pump differential temperature control.

6.2.17.5 Differential pressure sensors applied to variable speed pump operation shall be located across distant branch zones recognized and tested as the hydraulically most significant zone, e.g., the zone for which the pump head was selected to operate. Sensors may not be located across the supply and return piping within the mechanical spaces providing housing of the pumps. Similarly, pump control algorithms not relying on direct differential pressure sensing shall indicate how they ensure maintenance of the minimum required differential pressure at the hydraulically most significant zone during part load operating conditions, or they shall not be applied.

7. HYDRONIC MEASUREMENTS

7.1 Scope

7.1.1 For purposes of this section, "hydronic systems" includes piping systems that carry water, oil, antifreeze solutions, steam, and steam condensate.

7.1.2 This section sets forth standard techniques for field measurement of temperatures, pressures, and related electrical data of hydronic systems.

7.1.3 This section sets forth standard techniques for calculation of fluid flow rates, velocity pressure, heat content, and pump performance in hydronic systems.

7.1.4 The minimum requirements of Section 4, Instrumentation, must be met or exceeded to comply with the criteria for measurement.

7.2 General

7.2.1 The recommendations in this section will apply to both new and existing HVAC systems. Certain characteristics describing the system performance are measured directly; others are calculated from measured data. The methods for determining each are covered in this section.

7.2.2 Actual hydronic performance determined from field measurement may differ from design conditions. These differences can often be explained by reviewing for system effects that cause changes as a result of adverse or unexpected conditions.

7.2.3 Certain system characteristics can be measured by several alternate methods. Many times the hydronic system configuration and installation will not allow the most accurate method to be used.

7.2.4 The accuracy that can be expected under field conditions for each of the performance characteristics is also discussed.

7.2.5 Balanced System: A balanced hydronic system shall provide flow in a simulated design flow condition within the specified flow tolerance to each heat exchanger. The required flow tolerance shall be calculated from the normalized coil heat transfer characteristic with respect to flow. 97.5% coil sensible heat transfer shall be used to determine the required flow tolerance. The maximum flow deviation at 97.5% heat transfer shall be no greater than ±10% of the design flow rate.

7.2.5.1 Coil heat transfer characteristic is a function of construction geometry, related in part to design differential temperature and entering water temperature.

7.2.5.2 At simulated coil partial load conditions, there will be deviation of flow from that required to maintain the part load condition. Several hydraulic effects interact to create these deviations, which may be overcome in large part through system design selections of pipe friction loss, organization of like system loads, balancing methods, and maintenance of key operating characteristics such as constant coil entering water temperature and constant branch operating differential pressure to allow proper operation of the temperature controller and control valve.

7.2.6 Simulated flow conditions are with all automatically actuated flow control valves for the purposes of temperature or other control in an open state.

7.2.7 Systems sized with use of a diversity factor shall provide the proportional (percentage) flow with respect to the specified system diversity to each heat exchanger with the automatically actuated flow control valve at the simulated design flow state.

7.3 Temperatures

7.3.1 General

7.3.2 Temperatures of fluids, such as water, oil, antifreeze solutions, heat transfer fluids, etc., will be used to determine the heat content using either the Fahrenheit (F) I-P scale or the Celsius (C) SI scale. For nominal commercial HVAC work, the normal operating range is from 40°F (5°C) to 225°F (107°C), 350°F (176°C) maximum for steam.

7.3.3 The quantity or amount of heat in a fluid is measured in British Thermal Units (Btu) or in kilojoules (kJ). Heat flow is measured in Btu per hour (Btuh) or watts (W).

7.3.4 Steam temperatures vary above 212°F (100°C) under pressure to below 212°F (100°C) when in a partial vacuum.

7.3.5 Instruments

7.3.6 Temperature measurements of fluid shall be made using the following instruments.

7.3.6.1 Dial thermometer with bimetallic helix coil.

7.3.6.2 Thermocouple with millivolt meter or potentiometer read out device.

7.3.6.3 Electric resistance thermometer utilizing Thermistor or RTD sensing element and integrated digital display.

7.3.7 Fluid Immersion

7.3.7.1 Wells: Test Wells for thermometer insertion installed at the desired locations permit accurate readings without removal or loss of the system fluid. Heat transfer fluid or mastic shall be used to ensure thermal contact between the thermometer and the test well.

7.3.7.2 Measurement Port: Self-sealing test and measurement ports are an alternate to thermometer wells to provide accurate readings with less measurement response lag and with essentially no removal or loss of system fluid. Such devices also allow for measurement of localized fluid pressure. Sensing devices are "injected" into the fluid stream using a hypodermic style probe. Typically, an electronic resistance temperature probe must be applied due to the diametric size constraints of the measurement port. Ports should be applied knowing the criteria of the hypodermic needle diameter prior to port selection and installation. The needle shall contact the fluid stream when injected into the system.

7.3.8 Thermometer Radiation Effects: When the temperatures of the surrounding surfaces are substantially different from the measured fluid, there is considerable radiation effect upon the thermometer reading if left unprotected. Proper shielding or aspiration of the thermometer bulb and stem can minimize these radiation effects.

7.3.9 Emergent Stem Correction: The complete stem-immersion type of calibrated mercury thermometer must be used with the stem completely immersed in the fluid in which the temperature is to be measured. If complete immersion of the thermometer stem is not possible or practical, then a correction must be made for the amount of emergent liquid column. The correction equation is:

Stem Correction = kn (t_b - t_s)

Where:

k (Hg), ˚C = 1.00016

k (Hg), ˚F = 1.00009

n = number of degrees of emergent liquid column

t_b = temperature of bath

t_s = temperature of stem

7.3.10 Thermometers calibrated for partial stem immersion are more commonly used. They are used in conjunction with thermometer test wells designed to receive them. No emergent stem correction is required for the partial stem immersion type.

7.3.11 Surface Measurements

7.3.12 The surface of the pipe or conduit where the measurements are to be made with a thermocouple device must be clean and free of scale, rust, insulation, etc.

7.3.13 The surface temperature of the conduit is not equal to the fluid temperature so that a relative comparison is more reliable than an absolute reliance on readings at a single point. The thermocouple and pipe must be installed at the point of measurement.

7.3.14 Surface temperature measurements are the least accurate and are to be used only when other measurement methods are not possible.

7.3.15 Accuracy: Under reasonable conditions and with properly calibrated equipment, the accuracy of field measurements should be within one half of a scale division mark.

7.4 Fluid Properties

7.4.1 Specific Gravity: The ratio of the mass of a given volume of a substance to the mass of an equal volume of water usually at an equal 40°F (4°C).

7.4.2 Density: The density of 68°F (20°C) water at atmospheric pressure of 29.9 inches Hg (101 kPa) is 62.3 lb/ft^3 (998.2 kg/m^3) standard conditions. Specific gravity under the same conditions is 1.0.

7.4.3 Specific Volume: The reciprocal of specific density. Water specific volume is 0.016 ft^3/lb at (0.001 m^3/kg) standard conditions.

7.4.4 Viscosity

7.4.5 The property of a fluid to resist flow. The viscosity of some fluids can change with a change in temperature. Several measurement methods are used for viscosity.

7.4.6 As viscosity increases, pump efficiency and capacity are reduced while the pump horsepower and the system friction loss are increased.

7.4.7 Vapor Pressure

7.4.8 The vapor pressure of a fluid can limit the suction lift of a hydronic pump. Vapor pressure denotes the lowest absolute pressure witnessed with a given liquid at a given temperature. If the pressure in a pump system is not equal to or greater than the vapor pressure of the liquid, the liquid will flash into a gas. It is for this same reason that pressure must be available on the suction side of the pump when handling hot water or volatile liquids, such as gasoline. Without sufficient pressure, the liquid will flash into a vapor and becomes unpumpable.

7.4.9 Many process applications use pressurized vessels on the suction side to overcome vapor pressure of some liquids. The amount of pressure needed depends on the liquid and liquid temperature. The higher the temperature, the higher the vapor pressure. Vapor pressure is measured in pounds per square inch (kPa) absolute.

7.4.10 Thermal Transfer Fluids

7.4.11 Water and steam are the commonly used heat transfer fluids with a usable range of 32°F to 350°F (0°C - 177°C). Below 32°F (0°C), air and refrigerants such as halogenated hydrocarbons, ammonia, brines, and/or solutions of glycol and water are used in the HVAC industry.

7.4.12 Above 350°F (177°C), oils or organic compounds that have vapor pressures lower than those of water must be used. Toxicity, corrosiveness, and flammability often limit the use of some of these in HVAC environments.

7.5 Pressure

7.5.1 Instruments: Pressure and/or vacuum measurements of fluids shall be made using the following instruments:

7.5.1.1 Dial-type pressure gauge

7.5.1.2 Dial-type differential pressure gauge

7.5.1.3 Electronic sensor with read out device

7.5.1.4 Fluid system digital hydronic flow meters

7.5.2 Gauge/Absolute Pressure

7.5.2.1 The pressure of most hydronic systems is measured in terms of pounds per square inch (psi [kPa]) or feet of water (ft wg [Pa]). This indicated pressure is known as the gauge pressure (psig), and the measuring device should indicate a zero reading when not connected.

7.5.2.2 For HVAC work, atmospheric pressure can be assumed to be 14.7 psi (101.3 kPa) at sea level even though barometric conditions constantly change. Absolute pressure (psia) equals the gauge pressure plus the 14.7 psi (101.3 kPa) of atmospheric pressure.

7.5.9 Definitions

7.5.10 Static Head: The pressure due to the weight of the fluid above the point of measurement. In a closed system, the pump capacity is not affected as the static head is equal on both sides of the pump.

7.5.11 Suction Head: The height of fluid surface above the centerline of the pump on the suction side. This value usually is subtracted from the static head of the pump discharge piping.

7.5.12 Differential Pressure: The pressure difference existing between two measured pressures. If possible, the same gauge should be used to take both readings.

7.5.13 Gauge Connections

7.5.14 Valve connections should be provided at the desired locations to permit accurate readings of pressures or vacuums. Pressures should be applied slowly to the gauge and released slowly by gradually turning the cock or valve handle.

7.5.15 Reduce or eliminate pressure pulsations by installing a needle valve or a pulsation dampener (or snubber) between the gauge and the system. Steam gauges should have a "pig tail" installed at the gauge inlet.

7.6 Flow Rates

7.6.1 General

7.6.2 Heat transferred in a system is directly dependent on the fluid flow rate. Flow rate can be determined with reasonable accuracy by indirect methods; however, it is more expedient to take flow rate readings directly from the desired locations.

7.6.3 In hydronic work, velocity usually is in terms of feet per second (fps), [meters per second (m/s)]. Volume flow rates are in terms of gallons per minute (GPM) [liters per second (L/s) or cubic meters per hour (m^3/hr)].

7.6.4 Instrument Measurement of Fluid Flow

7.6.5 At least one permanent flow measuring devices indicating total system flow should be installed in the proper locations when the system is being installed.

7.6.6 System flow measurement should be affected by application of In-line Turbine meters, In-line Electromagnetic meters, Insertion Electromagnetic meters, or Ultrasonic meters and minimally self-indicating at the point of measurement.

7.6.7 System Component Measurement of Fluid Flow

7.6.8 System components specified with laboratory measured flow coefficients (C_V) can be used as a flow-verification device. Care should be exercised, however, in interpreting results from devices not specifically designed for flow measurement.

7.6.9 Manufacturer statements of flow and pressure drop for heat exchangers shall be established by laboratory flow test for maximum accuracy. Heat exchangers, and in particular coils are not recommended for establishing flow based on pressure drop unless manufacturer supplied test data is provided to indicate coil water side pressure drop relationship to flow rate. Unfortunately, many components are rated only to a calculated pressure drop which may or may not conform to reality.

7.6.9.1 Pumps are imprecise flow meters at best. The pump may be used as a general indicator of flow through differential head readings across the pump, correlated with the pump flow characteristic curve to establish the pump flow rate. Establishing flow accuracy correlation is dependent upon:

7.6.9.2 Actual conformance of pump to the published curve. Preferentially, the pump is tested at the factory and provided with a certified pump characteristic curve conforming to Hydraulic Institute testing standards. The certified curve should indicate the factory flow test results and also any corrections for actual pump operating speed, velocity head correction, and fluid density.

7.6.9.3 Pump curve shape at the point of interpretation, and the accuracy of the differential pressure measurement device. Wherever possible, differential head readings should be also correlated with operating motor power readings.

7.6.9.4 Pump field operation without cavitation and free of entrained air in the working fluid.

a. Readings from proper gauge connection location as utilized in factory testing.

b. Pump and System Flow Rate Equations.

   1) Pump Equations (I-P Units)

$GPM_2/GPM_1 = RPM_2/RPM_1$

$GPM_2/GPM_1 = D_2/D_1$

$H_2/H_1 = (RPM_2/RPM_1)^2$

$H_2/H_1 = (D_2/D_1)2$

$BHP_2/BHP_1 = (RPM_2/RPM_1)^3$

$BHP_2/BHP_1 = (D_2/D_1)^3$

Where:

GPM = gallons per minute

RPM = revolutions per minute

D = impeller diameter (in.)

H = head (ft wg)

BHP = brake horsepower

(2) Pump Equations (SI Units)

$(L/s_2)/(L/s_1) = (m^3/s_2)/(m^3/s_1) = (rad/s_2)/(rad/s_1)$

$m^3/s_2)/(m^3/s_1) = D_2/D_1$

$H_2/H_1 = [(rad/s_2)/(rad/s_1)]^2$

$H_2/H_1 = (D_2/D_1)^2$

$BP_2/BP_1 = [(rad/s_2)/(rad/s_1)]^3$

$BP_2/BP_1 = (D_2/D_1)^3$

Where:

L/s = liters per second

m^3/s = cubic meters per second

rad/s = radians per second

D = impeller diameter

H = head (kPa) or (Pa)

BP = brake horsepower

7.6.10 Hydronic Flow Rate Equations (I-P Units)

$Q = 500 \times GPM \times t$

$P_2/P_1 = (GPM_2/GPM_1)^2$

$P = (GPM/C_v)^2$

Where:

GPM = gallons per minute

Q = heat flow (Btuh)

t = temperature difference (°F)

P = pressure difference (psi)

C_v = valve constant (dimensionless)

7.6.11 Hydronic Horsepower Equations (SI Units)

$WHP = (GPM \times H \times SG)/3960$

$BHP = (GPM \times H \times SG)/(3960 \times E_p(\text{decimal}) = WHP/E_p$

$E_p = (WHP \times 100)/BHP$ (in percent)

Where:

WHP = water horsepower

GPM = gallons per minute

BHP = brake horsepower

H = head (ft wg)

SG = specific gravity (use 1.0 for water)

E_p = efficiency of pump

$NPSH_A = P_a \pm P_s + (V^2/2g) - P_{vp}$

$h = f\,(L/D) \times (V^2/2g)$

Where:

$NPSH_A$ = net positive suction head available

P_a = atmospheric pressure (use 34 ft wg)

P_s = pressure at pump centerline (ft wg)

$V^2/2g$ = velocity head at point P_s (ft wg)

P_{vp} = absolute vapor pressure (ft wg)

h = head loss (ft)

f = friction factor (dimensionless)

L = length of pipe (ft)

D = internal diameter (ft)

V = velocity (ft/sec)

g = gravity acceleration (32.2 ft/sec^2)

7.6.12 Hydronic Flow Rate Equations (SI Units)

$Q = 4190 \times m^3/s \times t$ or $Q = 4.19 \times L/s \times t$

$P_2/P_1 = [(m^3/s_2)/(m^3/s_1)]^2 = [(L/s_2)/(L/s_1)]^2$

$P = [(^{m3}/s)/C_v]^2 = [(L/s)/C_v]^2$

Where:

Q = heat flow (kilowatts)

t = temperature difference (°C)

m^3/s = (used for larger volumes) = cubic meters per second

L/s = liters per second

P = pressure difference (Pa or kPa)

C_v = valve constant (dimensionless)

$WP(kW) = 9.81 \times m^3/s \times H(m) \times SG$

$WP(W) = [\sim L/s \times H(Pa) \times SG]/1002$

$BP(W) = WP/E_p$ or $[L/s \times H(Pa) \times SG]/1002 \times E_p$ (decimal)

$E_p = (WP \times 100)/BP$ (in percent)

Where:

WP = water power (kW or W)

m^3/s = cubic meters per second

L/s = liters per second

BP = brake horsepower (W)

SG = specific gravity (use 1.0 for water)

E_p = efficiency of pump

H = head, Pa (m)

$NPSH_A = P_a \pm P_s + (V^2/2g) - P_{vp}$

$h = f(L/D) \times (V^2/2g)$

Where:

$NPSH_A$ = net positive suction head available

Pa = atmospheric pressure (Pa), (standard atmospheric pressure = 101.325 Pa)

$V^2/2g$ = velocity head at point P_{sl} (m)

P_{vp} = absolute vapor pressure

P_{sl} = pressure at pump center line (Pa)

g = gravity acceleration (9.807 m/s^2)

h = head loss (m)

f = friction factor (dimensionless)

L = length of pipe (m)

D = internal diameter (m)

V = velocity (m/s)

7.7 Pump Tests

7.7.1 Field tests of an installed pump should include the following:

7.7.2 Verification of Pump Impeller Size: Using the pump manufacturer certified performance characteristic curve:

7.7.2.1 Turn pump on, either utilizing a variable speed drive bypass power supply if provided, or by setting variable speed control to manual mode and adjusting for 100% speed.

7.7.2.2 Measure actual operating pump speed. Traditional measurement shall be by Stroboscope or Tachometer. Correct pump curve to actual measured rotational speed. Note: rapid technological evolution also may offer several other robust and accurate methods for rotational speed measurement utilizing unconventional methods of data collection and display, such as mobile device applications and self-powered sensors collecting energy from the applied measurement.

7.7.2.3 With pump operating, slowly close pump discharge valve. This is shut-off head. Read pressure differential across pump. Convert psi gauge differential pressure to feet of water (kPa); be sure to correct for any difference in gauge heights. Note measured differential head. Re-open pump discharge valve.

7.7.2.4 Examining manufacturer pump curve, compare measured differential head and mark location on ordinate axis (y-axis) of pump curve indicating “Head.” Indicate supplied impeller size from pump manufacturer submittal data and pump manufacturer identification and serial number tag affixed to the pump. If “no flow” discharge head point of pump curve does not correlate closely to measurements, verify deviation with manufacturer and or correct actual pump operation characteristic. In general, centrifugal pumps may have a minor decrease in head from maximum head indicated on pump curve similar in concept to a fan curve “hump,” but of significantly less magnitude. By manufacturer standard measurement consensus agreement, the point of maximum head is extended to the ordinate as a flat line due to the insignificance of the general deviation.

7.7.2.5 Verify with manufacturer if this procedure is applicable to this equipment. If the discharge shut-off head method is rejected by the manufacturer, the impeller must be physically measured with a ruler and/or caliper.

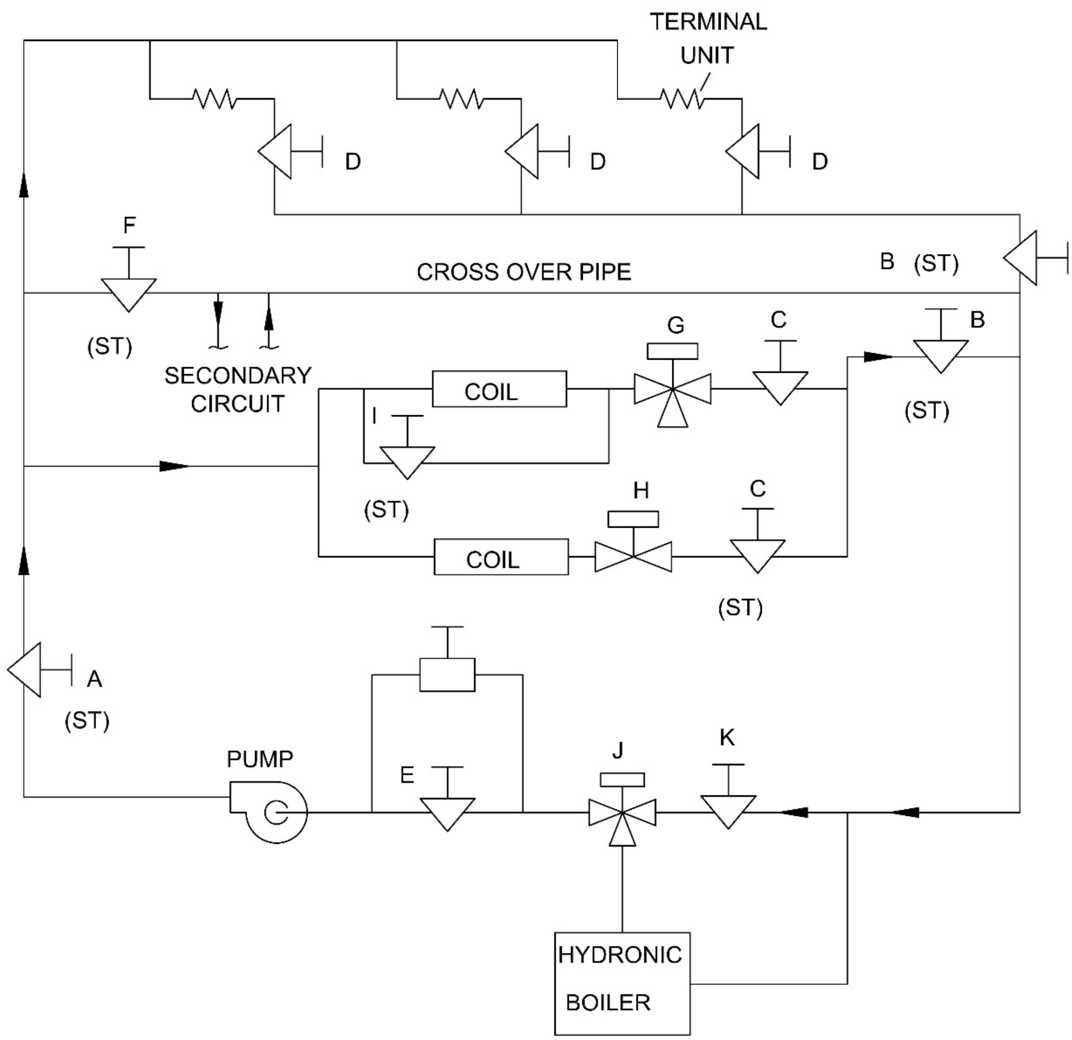
7.7.2.6 Anytime shut-off head is taken, the valves must be closed slowly to prevent water hammer, which could cause serious damage to the piping system. The discharge valve shall be closed for only the time required to measure the discharge and suction pressures.

7.7.3 Verification of Pump Capacity by Manufacturer Characteristic Curve

7.7.3.1 Turn pump on to 100% speed, opening all control valves to 100% open for full system flow effect.

7.7.3.2 Measure the operating pump differential head. Using a corrected pump flow characteristic curve, interpret the intersection of the differential head and the curve to establish the operating flow rate. If the pressure gauges accuracy band does not provide a discrete enough window of flow with respect to the curve, electrically establish actual motor operating horsepower to provide two points of curve intersection.

7.7.3.3 Prior to reading, establish that gauge readings are taken at the same points utilized by the manufacturer to establish the flow characteristic curve in the factory and laboratory. Ensure that pressure gauges are properly connected and flooded (no trapped air) to the suction and discharge pipes.



TYPICAL BALANCING VALVE LOCATIONS (A TO H)

A. PUMP DISCHARGE
B. SUB-RETURN MAIN
C. COILS
D. TERMINAL UNIT OUTLETS
E. CHEMICAL FEEDER BY-PASS
F. CROSS OVER PIPES
G. MIXING VALVE AT COIL
H. TWO WAY VALVE AT COIL
I. BYPASS VALVE AROUND COIL
J. MIXING VALVE FOR BOILER
K. BYPASS VALVE AROUND BOILER

FIGURE 5 Typical Hydroncic Balancing Stations (ST)

7.7.3.4 Measure motor voltage and amperes and establish motor operating horsepower.

7.7.4 Pump Pressure Gauges

7.7.4.1 Each installed pump shall have gauges installed for verification of pump operation during system commissioning and facility operation.

7.7.4.2 It is preferential to utilize one common pressure gauge with signal piping and valve selectivity to measure all points in the area of the pump.

7.7.4.3 One gauge will create the same reading error in all readings reducing overall reading errors due to multiple gauges.

7.7.4.4 Application of a static pressure gauge allows both pump and differential pressure readings to be established by calculation and is preferable to utilizing a differential pressure gauge and a separate static pressure gauge.

7.7.4.5 Ensure that permanently installed pressure gauge(s) and spot reading ports are properly connected to yield correct readings.

7.7.4.6 If there are no pump gauge taps, then consult manufacturer installation guidelines for the proper gauge connection locations to suction and discharge pipes.

7.7.4.7 Such connections should be placed as close to the pump as possible and there must be no fittings between the pump and the gauge connections.

7.7.5 There should always be a gauge valve directly ahead of each gauge. This will permit removal of the gauge if service is required and may also be used to dampen system pressure pulsations to obtain a steadier, more accurate gauge reading.

7.7.5.1 When applying multiple gauges, they should be mounted so they are at the same level.

7.7.5.2 Gauges mounted on different levels must have readings corrected for the head difference between the gauge readings. It is, therefore, recommended that after taking a set of readings, the gauges be interchanged and read again for comparison with the first set of readings unless one gauge is a compound type.

7.7.6 A preferred method is to mount a pressure gauge on piping which connects the factory pressure test holes, usually located in the suction and discharge flanges. A shut off valve is located on each side of the gauge tee fitting. A portable hose assembly valved with an added air bleed valve at the gauge is a useful portable test assembly.

7.8 Miscellaneous Pump Test Procedures

7.8.1 Check System Resistance

7.8.2 With all valves open throughout the system, read pressure differential across pump and so mark on head capacity curve.

7.8.3 Pressure readings taken close to the pump discharge may be affected by turbulence from the pump. Flows calculated from pressure differential readings at the pump suction and discharge must be considered as approximate and not used to verify hydronic balance or flow meter readings.

7.8.4 Record Actual Conditions: Read motor nameplate voltage and amperes. Measure motor voltage and amperage. Actual amperage should not exceed motor nameplate amperes for the pump to be a non-overloading pump. The pump head capacity curve should always be under the rated horsepower curve. (See Figure E2).

8. AIR TESTING, ADJUSTING, AND BALANCING

8.1 Scope: This section sets forth requirements for:

8.1.1 System Preparation and Obtaining Data

8.1.2 Obtaining all approved performance data

8.1.3 Verifying installation and conditions of all equipment and systems

8.1.4 System Testing and Adjusting, including procedures to test and adjust equipment and test the performance of systems.

8.1.5 System balancing, including balancing procedures for various types of systems and equipment.

8.2 General

8.2.1 The requirements set forth in this section shall apply to both new and existing HVAC supply, return, and exhaust systems. The requirements of Section 4, Instrumentation, and Section 5, Air Measurement, shall apply as a minimum to system testing, adjusting, and balancing.

8.3 System Preparation

8.3.1 Prior to the air system testing, adjusting, and balancing, obtain and verify the following:

8.3.1.1 Obtain updated construction drawings, specifications, approved shop drawings and submittals, addenda, bulletins, and change orders related to air systems.

8.3.1.2 Prepare field data forms to record testing and balancing process.

8.3.1.3 Obtain system leakage rate data where duct leak testing is specified.

8.3.1.4 Verify that fans are installed, rotating correctly with proper RPM, controlled to supply the required airflow rate, and that all installation, start-up, lubrication, and safety requirements have been met.

8.3.1.5 Check for clean filters properly mounted and sealed.

8.3.1.6 Fire, smoke, automatic, and volume control dampers are operable, accessible, and are in an open or normal position.

8.3.1.7 Controls are installed, operable, and calibrated.

8.3.1.8 Air Terminal Devices are installed, operable, and accessible.

8.3.1.9 Air Outlet and Inlet Devices are installed and accessible.

8.3.1.10 Access doors are installed and secured.

8.3.2 Perform the following in accordance with design documents before beginning air system testing, adjusting, and balancing:

8.3.2.1 Verify that all dampers are in an open position and all air terminal devices or automatic air volume control devices are in an acceptable mode.

8.3.2.2 Verify that all air inlet or outlet deflectors are in the position indicated by the manufacturer when using Ak factors to determine airflow rate and obtain correction factors for all velocity measuring instruments.

8.3.2.3 Verify that all automatic controls in the system are set in the testing mode and all computer programs have been properly loaded (where applicable) and parameters set.

8.4 Air System Testing and Adjusting

8.4.1 Perform the following tests and adjustments before beginning the air system balancing:

8.4.1.1 Record nameplate data on fan, motor, and air handling cabinet. Also, record sizes of sheaves, belts, and shafts.

8.4.1.2 Test and record the fan RPM to confirm rated speed.

8.4.1.3 Measure and record motor operating amperes and voltages.

8.4.1.4 Set system in the minimum outdoor air mode and then perform a Pitot static tube velocity traverse of main ducts and adjust fan speeds for total design supply and return airflow rates. Total design flow must include estimated duct leakage plus 5% of system total to allow for balancing effects. Minimum outdoor air quantities must be maintained during all system modes established by velocity traverse or other methods.

8.4.1.5 For special systems in Section 8.6 that use VAV or CV pressure independent air terminal devices, set system static pressure and proceed to test and balance all of the air terminal devices and their downstream air inlet or outlet devices, being sure the air terminal device inlet pressure is in the correct range. Air terminal device adjustments must be done per manufacturer literature. The following steps occur after all air terminal devices and related air inlet or outlet devices are balanced:

8.4.1.6 Measure and record the static pressure resistance of the duct system and the static pressure drop across coils, filters, etc., in the cabinet or out in the duct system.

8.4.1.7 Measure and record the pressures at fan suction and discharge per the pressure rating required, either static or total.

8.4.1.8 After the system is balanced, test the system in the maximum outdoor air mode. If motor overloads or airflow rates are excessive, adjust manual dampers to obtain the same conditions as recorded with minimum outside air.

8.4.1.9 Measure and record outdoor, return, and supply air temperatures with the system set at minimum outdoor air mode at design airflow or diversity and cooling or heating medium set for design flow. Verify coil capacities by the following formulas:

a. Sensible Heat

Btuh = CFM x 1.08 x T

Where:

Btuh = Btu per hour, sensible heat

CFM = cubic feet per minute, volume of airflow

1.08 = constant, 60 min/hr x 0.075 lb/ft^3 x 0.24 Btu/lb/°F

T = dry bulb temperature difference of the air entering and leaving the coil. In applications where airflow to the conditioned space needs to be calculated, the T is the difference between the supply air dry bulb temperature and the room temperature dry bulb.

b. Total Heat Airside

$Btuh_t$ = CFM x 4.5 x h_t

Where:

$Btuh_t$ = Btu per hour, total heat

CFM = cubic feet per minute, volume of airflow

4.5 = conversion factor, 60 min/hour x 0.075 lb/ft^3

h_t = change in total heat content of the supply air (enthalpy), Btu/lb (from wet bulb temperatures and psychrometric chart or table of properties of mixtures of air and saturated water vapor).

c. Total Heat Waterside

Btuh = GPM x 500 x T_w

Where:

Btuh = Btu per hour, water

GPM = water volume in gallons per minute

500 = conversion factor, 60 min/hour x 8.33 lbs/gallon x 1 Btu/lb/°F

T_w = temperature difference between the entering and leaving water

8.5 Air System Balancing: Balance the air system by the procedure outlined below.

8.5.1 Traverse Procedure

8.5.1.1 After the air system has been prepared according to Section 8.3 and Section 8.4, balance by the procedures set forth in the following subsections:

8.5.1.2 Note: When system characteristics prevent design flow rates, balance the system components to equal percentages of design unless otherwise instructed by the design engineer.

8.5.1.3 Balancing Sub-main Air Ducts

a. Perform a velocity traverse of each sub-main duct to determine flow rate through each.

b. Adjust the main volume control dampers to provide the required flow through each sub-main air duct.

8.5.1.4 Balancing Branch Air Ducts

a. Beginning at the sub-main duct closest to the fan, or with the highest percentage of required flow, perform a velocity traverse of each branch on that sub-main duct run.

8.5.1.5 Proceeding from the branch with the highest percentage of required flow, adjust the branch volume control dampers to provide the required flow through each branch duct.

8.5.1.6 Proceed to the sub-main duct with the next highest percentage of required flow, and traverse and adjust each branch per the previous steps.

a. Continue until all branches are balanced.

8.5.1.7 Balancing Air Terminal Device Flow Rates

a. Starting at the air terminal device with the highest percentage of design flow and ending with the air terminal device having the lowest percentage of design flow, adjust the air terminal device volume control to provide an airflow rate within 10% of design. Note: If balanced properly without excess pressure, then at least one air terminal device on each branch should have the volume control damper fully open. Branch dampers may require readjustment.

b. Continue until all air terminal devices are balanced to within 10% of design.

8.5.1.8 Final Adjusting and Balancing

a. Measure and record the final airflow rates at each air terminal device. If it is necessary to adjust the airflow rate through an air terminal device by 5% or less in order to achieve the final setting within 10% of design, then it is not necessary to adjust nearby air terminal devices which have been final measured. Otherwise, nearby air terminal devices should be re-measured and adjusted accordingly, if required.

b. Secure, mark, seal, and record the final setting positions of all volume control dampers installed in sub-main or branch ducts.

c. Measure and record the final airflow rates at velocity traverses in main, sub-main, and branch ducts. Do not adjust related volume control dampers.

d. Measure and record the data required in Section 8.4.

e. Reset all controls for normal operations.

8.6 Airside Systems. In addition to the applicable procedures set forth in Sections 8.3, 8.4, and 8.5, the following airside systems require additional balancing procedures.

8.6.1 Single duct pressure dependent system

8.6.2 Multi-zone system

8.6.3 Single duct, fan powered, pressure dependent system

8.6.4 Single duct, pressure independent system

8.6.5 Dual duct, pressure independent system, constant volume at variable volume

8.6.6 Laboratory testing

8.6.7 Note: For systems using fan volume controls, balance at less than 100% volume setting to allow for future pressure loss of wet coils, damper movement, or dirty filter, or simulate pressure losses with volume controls at 100%

8.7 Single Duct, Pressure Dependent Systems – All equipment in the system should be in operation before the test and balance technician begins the procedure. All controls should be installed and operational.

8.7.1 Procedures

8.7.1.1 Measure the airflow quantity of the supply, return, and outside air by traverse, unless it is impossible to do so.

8.7.1.2 When the air quantity cannot be obtained by traverse, the sum of the outlet or inlet quantities as the total CFM (L/s) of the fan can be used. If a traverse is not performed, then note the reason why on the test and balance report.

8.7.1.3 Proportionally balance the air distribution systems. Verify that at least one outlet or inlet damper is fully open on every branch duct and at least one branch duct balancing damper is fully open.

8.7.1.4 Adjust the fan speed to obtain 100% to 110% of design airflow.

8.7.1.5 Record the final measurements of the air distribution.

8.7.1.6 Set the system in the normal mode and record the following final conditions:

8.7.1.6.1 Supply, return, and outside air quantity

8.7.1.6.2 Motor voltage, current, kW, and actual motor speed

8.7.1.6.3 Fan speed

8.7.1.6.4 Static pressure profile

8.7.1.6.5 Coil capacity test results (including outside and return air temperatures)

8.7.1.6.6 If the system is equipped with an economizer, then repeat the data in Step 8.7.1.6 above in the economizer mode. (Coil capacity tests need not be repeated if the coils are not pertinent to the economizer mode). Confirm that the economizer mode is entered at the designated and appropriate enthalpy condition for all units. Also verify that the economizer mode is de-energized at the appropriate conditions.

8.7.1.7 For units with economizer control, confirm that the outdoor air damper returns to the minimum position at appropriate temperature (enthalpy) level. Confirm that the damper positions are in the same minimum position whether opening on startup or closing from maximum position on economizer lockout.

8.7.1.8 For units with demand control ventilation, confirm the override of the outdoor air damper control upon demand for ventilation.

8.7.2 Report

8.7.2.1 Include all test data on the appropriate data sheet forms.

8.7.2.2 List any uncorrected deficiencies that affect the test results on the deficiency form and include it with the report.

8.7.2.3 Include the report contents procedure requirements and all device test reports required.

8.7.3 Multi-Zone Systems

8.7.3.1 Operation: All equipment in the system should be in operation before the test and balance technician begins the procedure. All controls should be installed and operational.

8.7.3.2 Procedures

a. Measure the total airflow quantity for each zone, return, and outside air by traverse, unless it is impossible to do so. If a traverse is not performed, then note the reason why on the test and balance report.

b. When the quantity cannot be obtained by traverse, use the sum of the outlet quantities as the total CFM (L/s) of the zones.

c. Proportionally balance the outlets in each zone.

d. Proportionally balance each zone.

e. Proportionally balance the return air system.

f. Set the fan speed to obtain 100% to 110% of design airflow.

g. Take final measured data with the zone dampers in full cool and repeat in full heat or bypass mode.

h. Test the zone temperature control-mixing dampers for proper shut-off of both hot and cold decks and report the percentage of leakage.

i. Verify that all zone-mixing dampers are controlled by the proper space thermostat or sensor.

j. Confirm that the appropriate reset of hot and cold duct temperature setpoints based on zone demand has occurred.

k. For units with economizer control, confirm that the outdoor air damper returns to the minimum position at the appropriate temperature (enthalpy) level. Confirm that damper positions are at the same minimum position whether they are opening on startup or closing from maximum position on economizer lockout.

l. For units with demand control ventilation, confirm that the override of the outdoor air damper control has occurred on demand for ventilation.

m. At the completion of the balancing, set the system to automatic. If the system is equipped with an economizer, data shall be recorded in the economizer mode. (Coil capacity tests unrelated to the economizer need not be repeated). Record the following final conditions:

1. Supply, return, and outside air quantity
2. Hot and cold duct temperatures
3. Motor, voltage, current, kW, actual motor speed
4. Fan speed
5. Static pressure profile (including a static pressure in each zone)
6. Coil capacity test

8.7.4 Report

8.7.4.1 Include all test data on the appropriate data forms

a. List any uncorrected deficiencies that affect the test results on the deficiency form and include this information with the report.

b. Include the report contents procedure requirements and all device test reports required.

8.7.5 Single Duct, Fan-Powered Pressure Dependent Systems

8.7.5.1 All equipment in the system should be in operation before the test and balance technician begins the procedure. All controls should be installed and operational.

8.7.5.2 Procedures

a. Set the terminal box according to the appropriate procedure given in the component section for VAV box testing.

b. After all VAV terminals are adjusted, set the space thermostats in either a full heating or cooling position as required to satisfy the design diversity factor of the system, if applicable. Select the thermostats to be set for heating or cooling in order to simulate as nearly as practical the manner in which the system will respond to the cooling load shift of the building. Record the box numbers/designations used to achieve the diversity and record data for the boxes in full cooling and minimum cooling.

c. With proper diversity established, adjust the supply fan capacity to provide the design total CFM (L/s) with the automatic volume control device fully open to provide adequate, but not excessive, static pressure in the branch duct to the VAV terminal, which is the most difficult to supply. Record the individual boxes that are most difficult to supply.

d. Proportionally balance the return air system.

e. Determine the final total airflow quantity of supply, return, and outside air by a traverse unless it is impractical to do so. If a Pitot tube traverse is not performed, then note the reason why on the test and balance report.

f. Where the quantity cannot be determined by a traverse, use the sum of all low-pressure terminals as the total CFM (L/s) of the fan.

g. Measure and record the static pressure at the sensor for the automatic volume control device under maximum and minimum conditions. This is the static pressure control setpoint, unless otherwise noted. Note the control system readout.

h. With the system set for diversity, record the following final conditions:

   1. Supply, return and outside air quantity

   2. Motor, voltage, current, kW, and actual motor speed

   3. Fan speed

   4. Static pressure profile, including the pressure at the static sensor

   5. Coil capacity tests

i. If the system is equipped with an economizer, repeat the data in Step 8.7.5.2 above in the economizer mode. (Coil capacity tests need not be repeated if the coils are not pertinent to the economizer mode).

8.7.5.3 Report

a. Include all test data on the appropriate data sheet forms.

b. List any uncorrected deficiencies that affect the test results on the deficiency form and include this information with the report.

c. Include the report contents procedure requirements and all device test reports required.

8.7.6 Dual Duct, Pressure Independent Systems

8.7.6.1 This type of system uses control schemes that supply a varying quantity of heated or cooled air to the space. The hot duct and cold duct each have their own volume controller.

a. Operate all associated fans.

b. Set the air terminal devices being tested on full cooling or for diversity.

c. Take static pressures at all systemic components.

d. If all air terminal devices are constant volume, then set thermostats to obtain all the airflow through the cold ducts. Traverse the main ducts if more than 10% of the rated fan airflow is measured in the hot duct. During the balancing process, find and have hot valve leakage or crossed box supplies corrected.

e. If the air terminal devices have a variable volume feature, then adjust to full flow via thermostats so the sum total airflow rate of the air terminal devices equals the fan design flow rate during the balancing procedures.

f. Test the inlet static pressure at several of the most difficult to supply air terminal devices and make system adjustments for adequate pressure at the air terminal device inlet (CV or VAV) to provide the required flow rate through the air terminal device and downstream ductwork.

g. With the air terminal device (CV or VAV) set for 100% cold air delivery and with the hot duct temperature at least 20°F (10°C) warmer than the cold duct, test the air terminal device for hot valve leakage. Measure the temperature of the cold inlet duct air and the supply air temperature at two air outlet devices.

h. If the duct splits at the discharge, measure the temperature at an air outlet device on each branch. If the average supply air temperature at the air outlet device is higher than the cold inlet duct temperature by more than 5% of the difference between cold duct and hot duct temperatures, then request the installer to correct the deficiency. Also test for, report, and have corrected any air mixing deficiencies which result in 3°F (2°C) or more difference between air outlet device supply temperatures supplied by an air terminal device.

i. Proportionally balance all air outlet devices. Consider each air terminal device and associated downstream low-pressure ductwork as a separate, independent system. Verify the action of the thermostat (direct acting or reverse acting) and the volume damper position (normally closed or normally open). Verify the range of the damper motor as it responds to the velocity controller. Consult the air terminal device manufacturer data for the required pressure drop range across the air terminal device. The total required inlet static pressure is the manufacturer drop plus the downstream resistance. Record the static pressure drop across the air terminal device and the inlet static pressure. These readings should be within the required range. Verify that the air terminal device will operate at maximum flow when the inlet static pressure to the air terminal device is within the proper operating range by reading out the downstream air outlet devices.

j. Proportionally balance all air outlet devices.

k. Test the VAV air terminal device for both maximum and minimum flow as applicable. Consult manufacturer recommendations on the proper procedure for setting velocity controllers if required. Include both quantities on the report.

l. Measure and record the final total airflow rate that velocity traverses in the hot and cold ducts with system set for maximum cold duct airflow.

m. Measure and record the data required in Section 8.4.1.9 (a) (b) and (c), plus the duct static pressure sensed by the static pressure probe for automatic control of supply duct pressure existing when fan is at design flow rate.

n. Reset all controls for normal operation.

o. At the completion of balancing:

    1. The inlet manual damper to at least one VAV air terminal device on each branch duct will be fully open.

    2. At least one damper in each branch duct will be fully open.

    3. Reset the system to normal operating conditions.

8.7.6.2 Report

a. Include all test data on the appropriate data sheet forms.

b. List any uncorrected deficiencies that affect the test results on the deficiency form and include this information with the report.

c. Include the report contents procedure requirements and all device test reports required.

8.7.7 Laboratory Testing and Balancing

8.7.7.1 General

8.7.7.2 Note: a, b and c below are written for laboratories that are negative. The exhaust and supply airflow percentages will switch when the laboratories are designed to be positive.

a. For each fume hood, verify by traverse that the airflow is between 100% and 110% of design (The design airflow is the volume of exhaust that produces the required face velocity at sash opening, i.e., FV * Area).

b. For each laboratory balance, the supply airflow should be between 90% and 100% of design. Avoid any direct velocity from the ceiling diffuser toward the fume hoods. Verify airflow measurements by establishing correction factors from traverses.

c. Balance the general exhaust system airflow to between 100% and 110% of design. When flow hoods are used to measure general exhaust airflow rates, care should be taken when reading multiple exhaust grilles that the flow hood does not add restriction, forcing the air to another exhaust grille.

d. After the correct airflow for the hood has been established and all exhaust and supply air systems have been balanced, verify that the face velocities do not fall below the design face velocity as directed by the safety officer. Face velocities shall be taken at equal areas as described in ANSI/ASHRAE 110.

e. Make a sketch of the tested hood indicating each face velocity, the sash opening (height, width, and area), the position of the internal baffles, the traversed CFM, the laboratory room number, and exhaust system number. After the face velocities have been determined to be within the established limits, observe smoke flows into the hood to determine that no reverse flows are present.

f. A sticker indicating the inspection test result shall be placed on the side of the hood, at the maximum sash height measured, indicating the following:

   1. Height of sash (in inches)
   2. Average velocity: FPM
   3. Highest velocity: FPM
   4. Lowest velocity: FPM
   5. Person performing the test:
   6. Date of test:

g. It shall be noted in the sketch that all face readings are for reference only. The flow is established by traverse. At the present time, there is no way to take the average velocity times the face area to determine total flow. Each velocity-measuring instrument will require different correction factors, and these corrections are often different for different sized hoods of the same type.

h. If the hood does not pass this requirement, a caution tag shall be placed on the sash. The caution tag shall be a fluorescent orange tag stating: THIS HOOD DOES NOT MEET SPECIFIED FLOW REQUIREMENTS   DATE:

8.7.7.3 Tracking the Laboratory Control: Track the laboratory in the following manner by establishing airflows for each air terminal device:

a. All hood sashes open, minimum cooling.

b. All hood sashes closed, minimum cooling.

c. All hood sashes closed, maximum cooling.

d. Note velocity at the door during the set up of items (a) through (c) in this list.

e. Identify the point at which the hood face velocity falls below the target velocity. (Any time a minimum CFM is set, the hood will track linearly until it reaches the minimum airflow point; then the face velocity will increase).

f. Indicate flows on a drawing of the laboratory at maximum and minimum conditions and velocities recorded at the door.

g. Track the entire exhaust system from maximum flow to minimum flow by observing the static pressure entering the most remote hood exhaust air terminal device and the exhaust fan static pressure controller maintains setpoint.

h. Track the entire supply system from maximum flow to minimum flow, observing the static pressure entering the most remote supply air terminal device and the supply air static pressure controller maintains setpoint.

8.8 Verification of Control Operation

8.8.1 The TAB technician is responsible for verifying that the control system is operating as specified, and for reporting any installation problems discovered (1) setting controls to a proper fixed mode to prevent changes during balancing, and (2) verifying proper operation. Actual adjusting, moving, or recalibrating controls is normally the responsibility of the control contractor. However, TAB technicians should work closely with the control contractor to ensure system operation within design limitations, identify and correct any problems, and ensure the safety of the system and components. The performance of the automatic controls for the HVAC system is inspected and tested in each seasonal mode. In addition, the performance of all life safety devices and their interface with the HVAC systems shall be verified and reported.

a. Verify that controllers, including limiting controllers such as firestats and freezestats, are calibrated and in control.

b. Verify that controller setpoints meet design intent.

c. Confirm that the sequences of operation for any control mode are in compliance with the approved drawings.

d. Check that the control terminations are according to the approved drawings.

e. Verify the settings, operation, and adjustment of all end switches, mercury switches, solenoid valves, contractors, etc.

f. Check the operation of lockout or interlock systems.

g. Check the operation of all valve and damper actuators.

h. Determine that all controlled devices are properly connected.

i. Verify the operation of pilot positioners.

j. Confirm that all controlled devices are operated by the intended controller and note any overlap of controlled devices.

k. Prove that all controlled devices are in the position indicated by the controller (open, closed, or modulating).

l. Determine the integrity of all controlled devices in terms of tightness of close-off and full-open position. This includes dampers in multi-zone units, mixing boxes, and VAV air terminal devices.

m. Ensure that all controlled devices have free travel.

n. Verify that all controlled devices are properly installed in the distribution system in relation to direction of flow and location.

o. Confirm the proper operation of all controlled devices as applicable to normally open or normally closed.

p. Test the fail-safe modes of all controlled devices.

q. Examine the span of controls from a normally open position to a normally closed position, observing any dead bands, excessive pressures, and leading or lagging of simultaneously or sequentially controlled devices.

r. Check the location and installation of all sensors to determine if they will sense only the intended temperature(s), humidity, or pressure(s). Also check for potential erratic operation due to outside influences such as sunlight, drafts, outside walls, etc.

8.8.1.1 For pneumatic systems:

a. Check main supply air for proper pressures.

b. Observe the operation of the compressor and dryer.

c. With the system in normal operation, test each control loop at both ends of the control range to verify that all control loops and their individual field points are responding correctly.

d. Check the calibration of all field sensors.

e. Verify the calibration and response time of all transducers.

8.8.1.2 For direct digital systems:

a. Verify each point is named properly and where applicable setpoints meet the project requirements and hold the calibration tolerances.

b. With the system in normal operation, test each control loop at both ends of the control range to prove that all control loops and their individual field points are responding correctly.

c. Check the calibration of all field sensors.

d. Verify the calibration and response time of all transducers.

e. Determine if the system has lightning protection and battery backup.

f. Confirm the application and accuracy of the software algorithms for each control loop.

8.9 Thermal Performance Verification

8.9.1 After performing all previous procedures prescribed by Sections 8.3 through 8.8 and Section 9.3 of this standard, the system shall be set to simulate design conditions. Measure and record a complete set of DBT and WBT for air entering and leaving coils and heat exchangers and air in conditioned rooms or spaces. If conditions cannot be simulated and this affects verification, it shall be documented in testing and balancing report.

8.10 Outside Air Ventilation Verification

8.10.1 After completion of the balancing procedures in Sections 8.3 through 8.8, the system outside air rate shall be verified. This is necessary to assure that the design minimum outdoor air is being supplied to the occupied spaces. Obtain the minimum outside air rate and the appropriate balance conditions from the design documents. Determine the total system actual flow rate by traverse or other approved method and the return air rate by the same method. If adequate space is not available to perform a proper traverse, utilize the temperature ratio method if the outside temperature is at least 20° above or below the return air temperature. Adjust the outside air rate to equal the required flow rate by balancing the return air system to allow sufficient outside air to enter the system. This setting should be locked in and marked as the minimum outside air setting.

After setting the outside air rate, recheck the total system flow to assure that it has not changed.

9. HYDRONIC TESTING AND BALANCING

9.1 Scope: This section sets forth standard procedures for testing and balancing hydronic systems which include: water, thermal transfer fluids, steam, and condensate.

9.2 General Requirements: The techniques set forth in this section shall apply to both new and existing systems. Unless otherwise noted, each subsection listed under Section 9 shall apply to all hydronic systems. Any deviation from the procedure set forth, due to unusual circumstances, shall be documented and included as part of the final balancing report.

9.3 Sequence of Procedures

9.3.1 Contract Documents: Obtain a set of contract documents with all applicable addenda including specifications, complete set of approved equipment and control submittals, and manufacturer catalogs.

9.3.2 Data Sheets: Prepare field balancing data sheets or report forms with all pertinent design data and number in sequence starting at pump to end of system. Check sum of branch circuit flows against approved pump flow rate. If variation exceeds 5%, obtain approval to correct.

9.3.3 Schematics: Collect fluid flow schematic of designer and specified piping data and verify with as-built conditions. If schematic is unprepared, create fluid flow schematic of system to be balanced and show number on sketch that will correspond to number on appropriate balancing sheet in report.

9.3.4 Inspection: Field check system to ensure it can be balanced and has the proper balancing stations, including flow measuring device, temperature well, pressure taps, and balancing devices.

9.3.5 Preparation: The installed system shall be prepared for balancing. All of the following must provide satisfactory results before balancing procedures commence.

9.3.5.1 Utilizing fluid flow schematic and spreadsheet, calculate friction head loss of each terminal flow path with respect to the pump. Establish estimated head loss required for each circuit to cause all paths to have a similar common head loss.

9.3.5.2 Pre-check the following components operation and status as follows:

a. Pump rotation.

b. Inspect operating motor load of the pump. If motor is overloaded, throttle the main flow balancing device so motor nameplate rating is not exceeded, determine if impeller requires trimming to be a non-overloading pump.

c. Open manual and automatic valves for maximum system flow.

d. Operate the system for, at minimum, one day prior to balancing procedures.

e. The system shall be completely flushed and air-vented.

f. Inspect system fill valve to ensure proper setting for pressure at top of the system and allow required pressure for venting.

g. Inspect air vents for positive pressure. A forceful liquid flow should exit from all vents when manually operated.

h. Inspect for proper operating pressure and that the expansion tank is operable and properly setup.

i. Inspect strainers and see that they are clean and have the correct mesh for system fluid.

j. Inspect all check or combination check valves for proper installation relative to desired flow and to ensure that combination check valves are not in manual open position

k. Prepare balance plan implementation drawings, numbering and labeling valves with corresponding designed water-flows which correspond to report labels.

l. Set pump for constant pressure during the balancing procedure.

m. Set the pressure regulating valve at the specified pressure.

9.3.6 Test and Balance Procedure

9.3.6.1 Calculated or Pre-Set Methodology: The basis of Proportional Balancing methods is that all heat transfer fluid flow paths have the same fluid head loss with respect to the pump serving their system. When a flow in a circuit is changed, the flow in the circuit units will change in the same relationship or proportion. The “Pre-Set” method provides an excellent starting point for any balance methodology. PSM calculates each path’s design head loss without the addition of any balancing valve losses. Based on the calculated system path head losses, the most significant (greatest) path head loss is determined and the required incremental head loss for each path to make all paths equal is determined. Manually set balance valves may be designated to their respective terminal paths and adjusted to the required valve position either prior to or during installation to deliver the design flow rate.

a. Read and Set Balance: Without calculating required settings, and starting at the terminals closest to the pump, flow devices are read and then adjusted within flow tolerance to the terminal design flow rate, sequentially going to each further valve in the system. After valves are adjusted, the valves are re-read, adjusting only those outside of the specified flow tolerance range. This method is not recommended.

9.3.6.2 Proportional Balance: The basis of Proportional Balancing methods is that all heat transfer fluid flow paths have the same fluid head loss with respect to the pump serving their system. If the pump operation provides greater head and flow than required, all circuits receive a similar percentage of overflow. If the pump operation provides for less flow and head than required, or if the pump was selected in a manner to operate in such a manner such as with diverse pump system selections, all circuits receive a similar percentage of underflow.

a. It is advised that during preparations, the technician shall have calculated the circuit head losses and potential valve adjustments even while not implementing those adjustments.

b. Technicians shall detail their analysis and compensate for the style of manually adjusted balance device employed. Systems implementing fixed measuring points on each circuit may utilize flow read from the device to make adjustments and proportional settings. Designer specification of other flow reading styles may employ computer applications to calculate what valve adjustments are employed and specific procedures.

c. Project specifications referencing ANSI/ASHRAE 111 (this document) for general system commissioning and performance requirements shall utilize pre-determined calculations for estimating of balancing device settings (Pre-Set) coupled with direct field reading and adjustment procedures (Proportional).

9.3.6.3 Automatic Maximum Flow Limiting: “AFL” balancing implements one major aspect of system balance; individual path flow rates do not exceed design flow. AFL balancing does not provide proportional responses and operates according to the installed system hydraulics. AFL devices closest to a pump will close off greater than design flow in response to the provided differential pressure to the branch but will not operate proportionally to other circuits. Circuits with less than design flow rates shall operate at a deficit until path control valves for circuits closer to the pump act to close in response to temperature controllers. Depending on the system loading, these circuits could remain starved for system flow for substantial periods of system operation time.

a. It is advised that during preparations, the technician shall have identified each AFL valve and the designated flow cartridge setting. System calculation shall verify that total installed flow rates are within the flow design tolerance.

b. Technicians shall inspect the installed AFL valve cartridge and note indicated flow rates from fixed measuring points on each circuit.

9.3.6.4 Pressure Independent Control Valves: PIC valves may implement most major aspects of proportional balancing through the combination of their integrated differential pressure regulator on the temperature control valve, and through the integration of programmed logic through a programmable control system providing the control signal to the PIC valve. Flow shall be read from the provided fixed measuring points on each circuit.

a. The designer shall specify the controller sequence of operation under balanced system operation and for system commissioning. These sequences shall determine interaction such that coil flow tolerance is achieved, allow for technician override during the testing phase, and account for system operation modes when required system flow as determined by temperature controller output is non-attainable from installed pumping system components.

b. Technicians shall inspect the installed PIC valve and note indicated valve flow range and maximum valve flow setting. Technicians shall read flow from fixed measuring points on each circuit for the maximum actuator flow control signal, and an incremental flow signal if implemented meant to limit maximum controlled flow of the valve in normal operation.

9.3.6.5 Other Balancing Methodologies: Several different methodologies of fluid circuit control are available to the system designer, each requiring application guidance to the test and balance technician. Pressure Independent Control can be provided at individual circuits, or across several circuits through application of a Differential Pressure Regulator; Circulating Pumps may be employed as flow control devices in either a binary (full flow, no flow) operating state or as a proportioning device through application of a variable speed drive.

a. The designer shall specify the methodology of operation of the device, the fixed flow measuring points, and any required accessories to prevent gravity circulation as required. All specified operations and devices shall allow for establishment of the terminal required heat transfer at 97.5% for comfort or other required conditions.

b. Technicians shall read flow from fixed measuring points on each circuit. Technicians shall implement other readings such as temperature to determine iof conditions such as gravity circulation can exist during operation.

9.3.7 Primary Constant Flow Systems

9.3.7.1 Operation: All equipment in the system should be in operation before the technician begins the procedure. All controls should be installed, calibrated, and fully operational.

9.3.7.2 Procedures

a. Perform the shut-off head-pressure test on each pump to determine the impeller size and pump operating curve.

b. Make the initial setting of pump flow at approximately 110% of design flow, with all system coil valves in the wide-open position. If a pressure bypass valve is used to maintain constant flow in a two-way valve system, verify that the pressure bypass valve is closed.

9.3.7.3 Proportioning

a. If the balance valves are manual, proportionally balance the flow through each coil or element in the system. Make at least two passes to obtain a proportional balance. More passes may be required to obtain a proportional balance, but a minimum of two passes should be performed.

b. If the system uses automatic flow control valves, verify that the correct valve has been installed in each application and measure the pressure at the valve to assure that the system pressure does not exceed the valve maximum or minimum capacities. The valve pressure capacity should be designed to exceed maximum pump pressure capacity. It is also advisable to check the water system cleanness and strainer application to protect these valves.

c. If three-way valves are present at each coil or element, cycle the controls to modulate the valve to bypass after setting the coil flow, and set the flow through the bypass to match the coil flow. Modulate the control valve from the coil to bypass position and monitor the flow to assure that flow does not increase. If the flow does increase, adjust the bypass balancing valve until the modulated flow equals the coil flow. Re-measure the full bypass flow, record all flows, and report the excess modulated flow as a deficiency.

d. Make a final adjustment at the pump, if necessary, to obtain 100 to 110% of design flow. If this flow is not attainable, make a note of the reason for the deviation.

e. Perform a final pass through each coil or element, recording the final flows and pressure drops.

f. Record all final pump operating heads and flows.

g. Measure and record the amperage and voltage on all pumps before and after the system balancing.

h. Verify that all control valves are the three-way type, or that a pressure bypass is installed in the proper location to maintain constant or minimum flow. If a pressure bypass valve is installed, measure and record the system operating pressure at the control point with all valves in the system at full flow. Modulate the coil valves to simulate minimum flow to the coils and check the total system flow and operation of the bypass valve. Record the control pressure operating point.

i. Upon completion of the testing and balancing, record the data below on appropriate report forms.

- Design water flow, GPM (L/s)

- Design pressure criteria

- All nameplate data

- Amperage and voltage measurements

- kW (where applicable)

- Design pump and motor RPM

- Actual motor RPM (if accessible)

- Actual pump RPM if different from the motor RPM

- Pump shutoff head pressure

- Pump operating head pressure

- Pump suction pressure (both operating and shut-off)

- Pump discharge pressure (both operating and shut-off)

- Position of the balancing valve at the final balance

- Actual water flow GPM (L/s)

- Actual pressure measurements at the flow elements, coils, etc.

- Actual bypass pressure and control setting at the final balance (if installed)

9.3.7.4 Report

a. Include all test data on approved Water Balance Data Sheets as described in the report contents procedure.

b. In the conclusion, list both the conditions at the time of the test and the results.

c. List any uncorrected deficiencies that affect the test results on the deficiency report form.

d. Include the required data from the report contents procedure in Section 6 and all device test reports required.

9.3.8 Constant Flow Primary/Variable Flow Secondary Systems

9.3.8.1 Operation: All equipment in the system should be in operation before the technician begins this procedure. All controls should be installed, calibrated, and fully operational.

9.3.8.2 Procedures

a. Perform a shut-off head pressure test on each pump to determine the impeller size and pump operating curve.

b. With all system valves in the wide-open position, set the primary and secondary pump operating head at approximately 110% of design flow.

c. Balance the water flow in the primary loop to 100 to 110%. Balance the flow between the primary elements (chillers, boilers, etc.), if there is more than one element per pump.

d. With all control and balance valves open, balance the secondary pumps and flow to 110% of design flow. If the secondary loop system includes variable speed drives controlled by a differential pressure sensor, set the initial pressure necessary to control the pumps. A good starting point for this setting is 5 psi (34.5 kPa) above the design pressure drop on the highest pressure drop coil(s). This pressure may need to be adjusted higher during the testing operation to provide full flow to the coil that is most difficult to serve.

9.3.8.3 Proportioning

a. If the balance valves are manual and if there is no diversity in the secondary system, proportionally balance the system, making at least two passes to obtain a proportional balance. More passes may be required to obtain a proportional balance, but a minimum of two passes should be performed. If there is diversity in the secondary system, select enough coils to match the secondary pump design GPM (L/s) while closing the remaining valves. Proportionally balance the flow through the selected valves as indicated above. Once the system has been proportionally balanced, select enough open valves to match the diversity and close them. Balance the flow at the newly opened valves. Record the coil numbers and locations necessary to produce the diversity readings.

b. If the system uses automatic flow control valves, verify that the correct valve has been installed in each application and measure the pressure at the valve to assure that the system pressure does not exceed the valve maximum or minimum capacities. The valve pressure capacity should be designed to exceed maximum pump pressure capacity. It is also advisable to check the water system cleanness and the strainer application to protect these valves.

c. Make final adjustments at the pumps to obtain 100% to 110% flow. If this flow is not obtainable, make note of the reason for the deviation.

d. Perform a final pass through the primary and secondary components, recording final flow and pressure drops with the system set for diversity as described above, if applicable.

e. Record all final pump operating heads and flows.

f. With the system set at maximum flow, assure that there is proper flow direction at the primary/secondary bridge. Verify temperatures in the primary and secondary loops to assure that there is no mixing of secondary return at full cooling.

g. Record the actual pressure at the control point for the variable-volume secondary system. Compare this to the actual control device readout or setpoint.

h. Set the pump control of the secondary system at minimum flow.

i. Measure and record the power on all pumps before and after the system balancing.

j. Upon completion of the test and balance procedure, record the data below on appropriate report forms.

- Design water flow, GPM (L/s)
- Design pressure criteria
- All nameplate data
- Power measurements
- kW (where applicable)
- Design pump and motor RPM
- Actual motor RPM (if accessible)
- Actual pump RPM if different from the motor RPM
- Pump shut-off head pressure
- Pump operating head pressure
- Pump suction pressure (both operating and shut-off)
- Pump discharge pressure (both operating and shut-off)
- Position of the balancing valve at the final balance
- Actual water flow GPM (L/s)
- Actual pressure measurements at the flow elements, coils, etc.
- Actual bypass pressure and control setting at the final balance (if installed)

9.3.8.4 Report

a. Include all test data on approved Water Balance Distribution Equipment Data Sheets as required by the report contents procedure.

b. In the conclusion, list the conditions at the time of the test and the results.

c. List any uncorrected deficiencies that affect the test results on the deficiency report form.

9.3.8.5 Flow at the Pump

a. Determine flow at the pump with the system control valves open. The pump must be operating at 60 Hz. Follow pump tests and pump test procedures in Appendix E, respectively. Prior to balancing the system coils, verify flows across all primary heat exchangers (i.e., chiller evaporators) and primary flow stations. Adjust the pump flow to within ±10% of design.

9.3.8.6 Proportion Flows at the Coils

b. Determine the flow at each coil using the catalogued flow pressure drop relationship or flow meters at each coil. After all the coil flows have been determined, throttle the balancing valve on the coils having greater than design flow. Measure the flows on each coil and repeat throttling the balancing valves having greater than design flow until all coils are operating within the specified design limits. This will allow all control valves to operate in their specified range.

9.3.9 Final Balance

9.3.9.1 With all coils proportioned within the specified design limits, total all coil design flows. Determine the final flow at the pump, across all primary heat exchangers and primary flow meters. Record all final data. Note if the discharge valve was throttled or the VFD was adjusted below 60 Hz. The design team should be consulted if the discharge valve was throttled or the VFD was set below 60 Hz to determine if the impeller should be trimmed or left in the balanced condition until future expansions are complete. With the system set for design flow, determine the system differential at the measuring station. This differential will be recorded and given to the control contractor for their system control point if a VFD is installed. Set system coil control valves back to their design setpoints. Release any overridden control devices and allow the system to reach equilibrium before measuring temperatures.

9.3.10 Final Temperatures

9.3.10.1 With the system at equilibrium, measure the coil temperatures by setting the coil to the design water flow and airflow. Measure the entering and leaving water temperatures and the entering and leaving coil temperatures (DB and WB for cooling coils and DB only for heating coils). Record the flows and temperatures and calculate and record the water MBH capacity. Set the balance valve back to the final balance position and release any overridden points.

9.3.11 Determine flow at pump.

9.3.11.1 If flow is not within 5% of design, consult with client to arrange a change of impeller, approval to throttle, or change of design flow rate.

9.3.11.2 If deficiency cannot be corrected, proceed to balance proportionally.

9.3.12 Pump Impeller Size

9.3.12.1 To determine the pump head capacity curve for centrifugal pumps, close off the discharge valve on the pump and measure pressure at the pump inlet and discharge. With this information, the pump head capacity curve can be established starting at no flow.

9.3.12.2 Verify with manufacturer if this procedure is applicable to this equipment. (Note: Never use this technique on positive displacement pumps or damage may result).

9.4 Variable Speed Pumping Systems

9.4.1 Balance variable speed pumping systems (i.e., systems with automatic two position valves) by setting system to maximum flow through heat exchange terminals and proceed in accordance with Section 9.3.

9.4.2 If diversity in flow design exists, it will be necessary to close the automatic two-way valves on parts of the system closest to the pump and proportion the water to the remaining terminals at 100% design flow (the terminals closed will be the diversity coils, and their total flow will be equal to the total design flow of all connected terminals minus the pump flow design). When all the terminals are proportioned, open the automatic control valves on the diversity coils. To proportion the diversity coils, close off the automatic control valves on the next group of terminals to equal the design diversity flow and proportion the diversity coils. Verify the flow at all terminals with open control valves. (Note all terminals will be balanced to 100% flow so the automatic valves (CV) will be at design flow). When the system is proportioned, the terminal automatic control valves will be put into control and setpoints verified.

9.4.3 The variable frequency drive (VFD) speed controller will be set with the hydronic system set for diversity. The pressure differential controller will be set to decrease the pump speed as the pipe pressure increases. This setting must be at 60 Hz at the VFD if the system has no future capacity. Test the pump with the VFD in bypass mode (the VFD bypassed and the pump motor operating at across the line full voltage); if the pressure increases, the pump horsepower must not exceed nameplate and the system pressure must be below the seating pressure of the valves. If the nameplate is exceeded, the operating head is off the published curve or valve seats are being lifted, the impeller must be trimmed.

9.5 Primary-Secondary-Tertiary Pumping Systems

9.5.1 The primary system has pumps for the primary heat exchangers and the secondary system has pumps for the building terminal units. The secondary pumps will pull water from the primary supply header. The control and balancing of the secondary will follow Section 9. The balancing of the primary pumps will be as described in Appendix E. The control will depend on the system structure, i.e., having a decoupled loop, and staging the pumps to meet the secondary requirements or the system will state the heat exchangers and the pumps by load.

9.6 Verification of Control Operation

9.6.1 Sensing Devices

9.6.1.1 Pressure, temperature, and flow sensing devices should be verified through their normal operating range. Observe operation of controlled device. Device should travel full-open to full-closed. Malfunctions are to be reported for correction. Retest after corrections.

9.6.2 Control Valves

9.6.2.1 For variable flow systems with the system set for minimum flow, verify that the VFD and/or the pressure differential control reduce(s) the pressure so all automatic two-way control valves closest to the pump will not lift off the seat. This is accomplished by verifying there is zero flow through the coil by verifying the coil differential pressure is zero.

9.6.3 Pressure Differential Controller Valve

9.6.3.1 Before balancing, inspect to assure bypass control valves are closed. After balancing, set the system to maximum flow. Obtain the pressure differential required to operate the system at peak flow. Verify sequence of operation with installer or manufacturer of controls by closing off one coil at a time and verifying that system differential is maintained and flow (at the most restrictive coil) is maintained.

9.6.4 Other Controls

9.6.4.1 Simulate operation in presence of or with approval of control contractor, in accordance with design requirements and manufacturer recommendations. Report any malfunctions for correction. Use the same procedure as in Section 8.8.

10. EQUIPMENT FIELD TESTING

10.1 Scope

10.1.1 The tests in this section are limited to capacity tests. Rating performance tests and part loading performance tests are not within the scope of this standard.

10.1.2 All tests should be taken at design conditions for greatest accuracy but may be performed within ±10°F (5°C) of design conditions and prorated using manufacturer performance test data. It is imperative that all parties agree on test procedures and how testing will be accomplished.

10.2 Refrigeration

10.2.1 Electrically Driven Chillers

10.2.2 Chilled and condenser water systems shall have a hydronic balancing station in the main flow circuit piping to each unit. Temperature wells and pressure gauges shall be installed in the outlet and inlet piping close to the machine. Also, temperature wells should be installed close to the cooling tower. After balancing the chilled water distribution, proceed as follows:

10.2.3 Chilled Water Flow

a. Measure and record flow through the evaporator.

b. Measure and record temperatures in and out of the chiller evaporator in 0.1°F (0.05°C) increments.

c. Measure pressure in and out of the evaporator.

d. Calculate capacity as follows:

Water flow x density x specific heat x mass x T/rate = Capacity

or

GPM x 500 x T(°F) = Btuh

(L/s x kg/m^3 x kJ/ (kg x K) x kg x T(°C)/3.515 kW) = W

Rate is 12,000 Btuh (3.517 kW/h) per ton of capacity

10.2.4 Condenser Water Measurements

a. Measure and record water flow through the condenser.

b. Measure and record the temperatures entering and leaving condenser.

c. Measure pressure in and out of the condenser.

d. Calculate capacity as follows:

Flow x density x specific heat x mass x T/rate = Capacity

Rate – 15,000 Btuh (4.396 kW/h) = Capacity

(L/s x kg/m^3 x kJ/kg x K x kg x T(°C)/4.396 kW/h) = Capacity

This rate is for electric compressor type chillers. The evaporator tons of capacity and the condenser tons of capacity should match.

10.2.5 Absorption Chillers

10.2.5.1 Chilled water measurement at the evaporator is the same as for the electric compressor type above.

10.2.5.2 Condenser water measurements are as follows:

a. Measure and record flow through the condenser.

b. Measure and record the temperatures in and out of condenser in 0.1°F (0.05°C) increments.

c. Measure and record pressure drop across condenser.

d. Calculate capacity as follows:

Flow x density x specific heat x mass x T/rate = Capacity

Rate = 28,000 Btuh (8.21 kW/h) is the heat rejection rate for this unit

(L/s x kg/m^3 x kJ/kg x K x kg x T(°C)/8.21 kW/h) = Capacity

Note: Heat rejection rates can vary and always need to be verified with manufacturer design database.

10.2.6 Efficiency Calculation

10.2.6.1 Output capacity (tons) /Input kW = kW/ton compare to manufacturer rating data.

10.2.7 Air-cooled Condensers

10.2.7.1 Measure and record airflow through the condenser coils.

10.2.7.2 Measure and record dry bulb temperature in and out of coils.

10.2.7.3 Calculate capacity as follows:

Airflow CFM (L/s) x density x specific heat x time x T/Rate = Capacity

Rate = 15,000 Btuh (4.396 kW/h) per ton of capacity

Density at (0.075) ft^3/lb x Specific Heat at (0.24 Btu/lb) x time (60 min/hour) = 1.08 at standard conditions. Always correct to standard conditions for capacity calculations.

I-P Units

Btuh = 10,000 CFM x 1.08 x 45°F = 486,000 Btuh

486,000 Btuh/15,000 Btuh/ton = 32.4 tons capacity

SI Units

Heating value (hv) = 4719 L/s x 1.20 x 25°C = 141,570 W/(4.396 x 1000) = 32.2 tons. (kW x 1000 = Watts)

10.3 Power Measurements

10.3.1 Measurements

10.3.1.1 Measure and record all electrical input to the device or component being tested in the same time frame other tests are being performed to assure all tests are measuring the same load.

10.3.1.2 Measure amperage on each phase and record. A continuous amperage measurement with a data logger will indicate a change in load or conditions that affect the capacity being tested.

10.3.1.3 Measure and record kW input for use in calculations.

10.3.1.4 Measure volts on each phase and record.

10.3.1.5 Measure motor speed to verify they are operating at design speed, where possible.

10.3.1.6 Power for controls, not used to produce capacity, shall not be measured unless a part of manufacturer design database.

10.3.2 Power Calculations

10.3.2.1 Measured kW x 3412 = Btuh. Btuh = Energy “input.”

10.3.2.2 Measured evaporator Btuh/12,000 Btu per ton = ton hour of refrigeration effect = Energy output.

10.3.2.3 Measured Condenser Btuh/15,000 Btu per ton hour heat rejection (Electric compressors).

10.3.2.4 Measure Absorption Chiller Energy Input (total including electric motor kW/h) associated with chiller rating. Convert to Btuh for input.

10.3.2.5 Measure Evaporator Btuh as in Section 10.2.3 as output.

10.3.2.6 Final result is kW/ton or Btu/ton, a rating characteristic of the chiller.

Output/input = found by the Field test. Compare to published rating to determine operating efficiency.

Note: The evaporator tons capacity should match the condenser tons if the chiller package is functioning properly.

10.4 Cooling Towers for Water Cooled Condensers

10.4.1 Water Cooled Condenser

10.4.2 Water flow from condenser to cooling tower should be the same flow. There may be basins and bypasses in the path that may change this pattern. Flow measurements shall be made at both locations. For accurate flow measurements, good quality flow meters are required to be installed in the supply piping to each cooling tower and to each condenser. The condenser supply is from the cooling tower. This is a requirement even if there is more than one chiller or more than one cooling tower per system.

10.4.3 Measurements and Verification

10.4.4 Measure water flow in and out of the condenser as stated above.

10.4.5 Measure water flow in the cooling tower and balance each cooling tower for design flow.

10.4.5.1 Measure water flow of make-up water to each tower basin. Set overflow to zero during test so the evaporation rate can be determined. During test, isolate make-up water.

10.4.5.2 Verify control valves are operating properly.

10.4.5.3 Verify the entire condenser water system is operating correctly by allowing the system to go to full cooling. Not all chillers need to operate with adequate flow meters, but the controls must allow full flow through the condenser and cooling tower. Take final flow measurements and record for the final report.

10.4.5.4 Measure and record the temperatures on and off the cooling towers.

10.4.5.5 Measure and record outside wet bulb temperature.

10.4.5.6 If the cooling tower has nozzles and pressure gauges installed, take pressure measurements on each line to the nozzles at full flow and record.

10.4.5.7 Measure power usage, including nameplate data, as stated under power above. Include motor amperage, voltage, RPM, safety factor, overload protection, manufacturer, and rating. List nameplate HP and actual BHP.

10.4.5.8 List cooling tower airflow from BHP and manufacturer data.

10.4.5.9 Calculations and Verification

a. Calculate heat rejection of cooling tower as follows:

Flow x T x factor/rate = heat rejection tons

Examples:

1000 GPM x 20 T(°F) x 500/15,000 = 667 heat rejection tons

63L/s x 11.1 T(°C) x 4.2/4.396 kW = 668 heat rejection tons

b. Verify and record data that each cooling tower is operating within manufacturer design. Verify the following:

   - Water flow

   - Airflow

   - Power

   - Make-up water

   - Overflow water

   - Water treatment; obtain data from water treatment contractor.

c. If stand-by cooling towers and equipment are in the system, start up the stand-by units and shut down the tested units and repeat test.

10.5 Centrifugal and Rotary Screw Chillers

10.5.1 Chillers cannot normally be tested at full capacity in field installations due to lack of control of loads and atmospheric conditions. Testing of chillers in field conditions shall not commence until after the manufacturer has completed the required field start up procedure. Field testing shall include measurement, balance, and recording of the following:

10.5.2 Evaporator Section

10.5.2.1 Temperatures of water entering evaporator, temperature of water leaving evaporator section with 0.5°F (0.01°C) of specified value, chilled water flow rate within

5% of design (preferably from a calibrated flow meter), electrical power input to compressor (volts and amps for all phases along with calculated brake horsepower), chilled water pressure drop (inlet to outlet).

10.5.3 Condenser - Water Cooled

10.5.3.1 Temperatures of water entering and leaving condenser section, condenser water flow rate within 5% of design (preferably from a calibrated flow meter), condenser water pressure drop (inlet to outlet).

10.5.4 Condenser – Air-cooled

10.5.4.1 Dry bulb temperature of air entering condenser and condenser fan motor power consumption (volts and amps for all phases of all motors along with a calculated brake horsepower).

10.5.4.2 Auxiliary Data

a. Nameplate data, including make, model, size, refrigerant, compressor driver RPM for open drive type compressors, ambient temperature at test site, and motor nameplate data.

11. REPORTING PROCEDURES AND FORMS

11.1 Scope

11.1.1 This section sets forth an outline for the reporting procedures and forms which make up the final report of operating conditions.

11.2 Reporting

11.2.1 Procedures: The supervising personnel should use a logical approach in preparing forms and recording data. This section will list form titles and entries commonly used and enable the forms to be designed to suit each particular job. All entries will not be required in every situation. Many excellent forms have been developed by various associations but are available for use by their members only.

11.2.1.1 Report Accuracy: Report and record accuracy shall continuously monitor and inspect for transcription accuracy;

11.2.1.2 Reports shall be the final permanent and legal record of system operating conditions after the last adjustments have been made.

11.2.1.3 Reports shall confirm that prescribed procedures have been followed.

11.2.1.4 Reports shall serve as a reference for maintenance personnel of the owner

11.2.1.5 Reports shall provide the designer with the operational system check and verification that system design assumptions were correct and could serve as an aid in diagnosing problems.

11.2.1.6 Reports shall include identification of project, system/unit, location, date, technician, page number, and remarks.

11.2.1.7 General Items: The report should contain the following, as applicable:

a. Title page

b. Name and address of TAB firm

c. Project name

d. Location

e. Architect

f. Engineer

g. Contractor

h. Report date

i. Signature of TAB firm person who approved report

j. Summary comments

k. Design versus Final performance

l. Notable characteristics of system

m. Description of systems operation sequence

n. Summary of outdoor and exhaust flows to indicate amount of building pressurization

o. Nomenclature sheet

p. Codes for boxes, reheat coils, terminals, etc., with data on manufacturer, type, size, fittings, etc.

q. Notes which explain in detail why certain final data in the body of the report varies from design values

r. Test conditions (To be stated on the fan or pump performance form)

s. Setting of outdoor, return, and exhaust dampers

t. Condition of Filters

u. Cooling coil – wet or dry

v. Face and bypass damper setting at coil

w. Fan drive setting (indicate setting percentage of maximum pitch diameter)

x. Setpoints of variable flow controller

y. Setting of supply air static pressure controller

z. Other systems operating which affect performances

aa. Form Titles and Entries

aa1. System Diagram: This form is to be used for schematic layout of air distribution systems and hydronic systems. A single line system diagram is highly recommended to ensure systematic and efficient procedures. Quantities of outside air, return air, relief air, sizes and airflow rates for main ducts, sizes and airflow rates for all air terminal devices, all dampers, and other regulating devices shall be shown. All air terminals should be numbered before filling out the Air Terminal Device Report. While diagrams are suggested, the use of this form is not mandatory.

aa2. Air Apparatus Test Report: The performance of air handling apparatus with coils is to be reported on this form. Motor voltage and amperage for three-phase motors should be reported for all three legs (T1, T2, T3). If the design engineer did not specify a design quantity for any item in the test data section, place an X in the space for the design quantity and record the actual quantity. However, if the equipment manufacturer furnished ratings, enter them in the design columns.

aa3. If motor ratings differ from design, provide an explanation at the bottom of the page. If there are split coils, record data for each airstream.

11.2.1.8 Unit Data

a. Make/Type

b. Model Number/Size

c. Serial Number

d. Arrangement/Class

e. Discharge

f. Sheave Make

g. Sheave Size/Bore

h. Number of Belts/Make/Size

i. Number of Filters/Type/Size

j. Make/Frame

k. HP/RPM

l. Volts/Phase/Hertz

m. Full Load Amps/S.F.

n. Sheave Make

o. Cooling Coil Differential Pressure

p. Heating Coil Differential Pressure

q. Outside Airflow Rate

r. Return Airflow Rate

s. Outside Air Damper Position

t. Return Air Damper Position

u. Vortex Damper Position or VFD HZ

v. Motor Data

w. Sheave Size/Bore

x. Sheave Center to Center & Adjustment

11.2.1.9 Test Data (List Design & Actual for each)

a. Total Airflow Rate

b. Total System Static Pressure

c. Fan RPM

d. Motor Volts, T1 T2, T2 T3, T3 T1

e. Motor Amps, A1, A2, A3

f. Discharge Static Pressure

g. Filter Differential Static Pressure

h. Preheat Coil Differential Static Pressure

i. Apparatus Coil Test Report: This form is to be used for recording performance of chilled water, hot water, steam, or DX coils, and for "run around" heat recovery systems.

j. Coil Data

k. System Number

l. Location

m. Coil Type

n. Number of Rows/Fins

o. Make/Model

11.2.1.10 Test Data (List Design & Actual for each)

a. Airflow Rate

b. Air Velocity

c. Air Pressure Drop

d. Entering/Leaving Air DB/WB

e. Water Flow Rate

f. Water Pressure Differential

g. Entering/Leaving Water Temperature

h. Exp. Valve/Refrigerant

i. Refrigerant Suction Pressure

j. Refrigerant Suction Temperature

k. Inlet Steam Pressure

11.2.2 Gas/Oil-Fired Heat Apparatus Test Report

11.2.2.1 Data for gas or oil-fired devices, such as unit heaters, duct furnaces, etc., will be recorded on this form. This report is not intended to be used in lieu of a factory startup equipment report but could be used as a supplement. All available design data should be reported. The "HP/RPM, FLA/S.F. (Service Factor), Drive Data" information could apply to the burner motor, burner fan motor, unit air fan motor, etc., depending on the application or equipment; therefore, designate the motor of the recorded data.

a. Unit Data

b. System Number

c. Location

d. Make/Type

e. Model Number/Size

f. Serial Number

g. Type Fuel/Input

h. Output/Btuh

i. Ignition Type

j. Burner Control

k. Volts/Phase/Hertz

l. HP/RPM

m. Full Load Amps/S.F.

n. Sheave Data

11.2.2.2 Test Data (List Design & Actual for each)

a. Airflow Rate

b. Entering/Leaving Air Temperatures

c. Entering/Leaving Air Pressure

d. Low Fire Input

e. High Fire Input

f. Manifold Pressure/CFH

g. High Limit Setting

h. Operating Setpoint

i. Voltage, T1-T2, T2-T3, T3-T1

j. Amps, A1, A2, A3

k. Heating Value of the Fuel

11.2.3 Electric Coil/Duct Heater Test Report

11.2.3.1 This form is to be used for electric furnaces or for electric coils installed in built up units or in ducts. "Min. Air Vel" is the manufacturer recommended minimum airflow velocity.

a. Unit Data

b. System/Location

c. Coil Number

d. kW

e. Stages

f. Volts/Phase/Hertz

g. Amps

h. Airflow Rate

i. Face Area

j. Minimum Air Velocity

11.2.3.2 Test Data (List Design & Actual for each)

a. kW

b. Air Velocity

c. Airflow Rate

d. Entering Air Temperature

e. Leaving Air Temperature

f. Voltage, T1-T2, T2-T3, T3-T1

g. Amps, A1, A2, A3

11.2.4 Fan Test Report

11.2.4.1 This form is to be used with supply, return, or exhaust fans.

11.2.4.2 Fan Data

a. System Number

b. Location

c. Make/Type

d. Model Number/Size

e. Serial Number

f. Arrangement/Class

g. Sheave Make

h. Sheave Size/Bore

i. Motor Data

j. Make/Frame

k. HP/RPM (w/Rad/s)

l. Volts/Phase/Hertz

m. Full Load Amps/S.F.

n. Sheave Make

o. Sheave Size/Bore

p. Number of Belts/Make/Size

q. Sheave Center Line Distance & Adjustment

11.2.4.3 Test Data (List Design & Actual for each)

a. Airflow Rate

b. Total System Static Pressure

c. Fan RPM

d. Discharge Static Pressure

e. Suction Static Pressure

f. Voltage, T1 T2, T2 T3, T3 T1

g. Amps, Al, A2, A3

11.3 Duct Traverse Report

11.3.1 Rectangular Duct Report: This form is to be used as a worksheet for recording the results of a traverse in a rectangular duct. Make a grid representing the duct cross section with a box for each test point. It is recommended that the velocity pressures be recorded in one half of each box provided and converted to velocities in the other half of box at a later time. The velocities shall be averaged. Do not average the velocity pressures.

11.3.1.1 Data Reported

a. System/Unit Number

b. Location/Zone

c. Traverse Air Temperature

d. Duct Static Pressure

e. Duct Size

f. Duct Area

g. Design Velocity

h. Design Flow Rate

i. Actual Average Velocity

j. Actual Flow Rate

k. Barometric Pressure

11.3.2 Round Duct Traverse Report: Record the results of a traverse in a round duct on this work sheet type form. Make a circle representing the duct cross section. Make columns with a number for each test point and for velocity pressures or velocities taken at points across two diameters at a right angle to each other.

11.3.2.1 Data Reported

a. System/Unit Number

b. Location/Zone

c. Traverse Air Temperature

d. Duct Static Pressure

e. Duct Size

f. Design Velocity

g. Design Flow Rate

h. Actual Average Velocity

i. Actual Flow Rate

j. Barometric Pressure

11.3.3 Oval Duct Traverse Report: Record the results of a Pitot tube traverse in a duct on this worksheet type form. Make columns with a number for each test point, the dimension along the major and minor axis, and velocity pressures or velocities taken at points across the two axes of the duct or show on a traverse sheet the horizontal reading.

11.3.3.1 Data Reported

a. System/Unit Number

b. Location/Zone

c. Traverse Air Temperature

d. Duct Static Pressure

e. Duct Size

f. Duct Area

g. Design Velocity

h. Design Flow Rate

i. Actual Average Velocity

j. Actual Flow Rate

k. Barometric Pressure

11.3.4 Air Terminal Device Report

11.3.4.1 This form can be used as both a worksheet and a final report form to record all readings on this test report form. However, it is not necessary to record preliminary velocity readings.

11.3.4.2 If the final adjusted flow rate of any air terminal device varies by more than ±10% from the design flow rate, a note should be placed in the remarks column indicating the amount of variance. The "remarks" section at the bottom of the sheet should be used to provide known or potential reasons for such deviation. All correction factors should be shown in the remarks column for all velocity measurement instruments.

11.3.4.2.1 Data Reported

a. System/Unit Number

b. Location/Zone

c. Test Apparatus

d. Area Served

e. Air Terminal Device Number (From System Diagram)

f. Air Terminal Device Type/Model

g. Air Terminal Device

h. Air Terminal Device Size

i. Design Flow Rate

j. Design Velocity

k. Preliminary Velocity (as needed)

l. Make Preliminary Flow Rate (as needed)

m. Final Velocity

n. Final Volume

o. Ak/Effective Area

11.3.5 System Coil Report

11.3.5.1 This form is used as a worksheet to report on reheat coils or on the water coil of terminal units. Any of the three alternate methods for determining water flow rate or heat transfer rate indicated on the test report form may be used.

11.3.5.2 Equipment Data

a. System/Unit Number

b. Location/Zone

c. Room Number/Riser Number

d. Coil Make

e. Model/Size

f. Design Flow Rate

g. Design Water Supply Temperature

h. Flow Meter Type/Size

11.3.5.3 Test Data

a. Flow Meter Reading (if available): Flow Rate

11.3.5.4 Item 1: Design Pressure Drop

a. Entering Water Pressure

b. Leaving Water Pressure

c. Actual Pressure Drop

11.3.5.5 Item 2: Design Water Temperature Drop

a. Entering/Leaving Water Temperature

b. Actual Water Temperature Drop

11.3.5.6 Item 3: Design Air Pressure Drop

a. Entering/Leaving Air Static Pressure

b. Actual Air Static Pressure Drop

11.3.5.7 Item 4: Design Air Temperature Drop [Cooling Coil (Db, Wb) Heating Coil (Db)]

a. Entering/Leaving Air Temperature

b. Actual Air Temperature Drop

11.3.6 Packaged Chiller Test Report

11.3.6.1 This form may be used to record the control settings and the entering and leaving conditions at the chiller. It does not attempt to indicate the performance or efficiency of the machine except as may be determined by the design engineer from the data contained therein.

11.3.6.2 This form, or the manufacturer form, should be substantially completed and verified by the manufacturer representatives and/or the equipment owner or installing contractor before the HVAC distribution systems are balanced. Temperature and pressure differential readings of the chiller unit evaporator and condenser should be recorded during the TAB procedures. Describe the flow measuring device when used.

11.3.6.3 Data Reported (list design and actual quantities, where appropriate)

11.3.6.3.1 Unit Data

a. Make/Type

b. Model Number/Size

c. Serial Number

d. Capacity Refrigerant

e. Refrigerant

f. Starter

g. Heater Size

11.3.6.3.2 Condenser Data

a. Condenser Pressure/Temperature

b. Entering/Leaving Water Pressure

c. Water Pressure Drop

d. Entering/Leaving Water Temperature

e. Water Temperature Drop

11.3.6.3.3 Evaporator Data

a. Evaporator Pressure/Temperature

b. Entering/Leaving Water Pressure

c. Water Pressure Drop

d. Entering/Leaving Water Temperature

e. Water Temperature Drop

f. Water Flow Rate

11.3.6.3.4 Compressor Data

a. Make/Model

b. Serial Number

c. Suction Pressure/Temperature

d. Discharge Pressure/Temperature

e. Oil Pressure/Temperature

f. Voltage, T1-T2, T2-T3, T3-T1

g. Amps, A1, A2, A3

h. kW Input

i. Crankcase Heater Amps

j. Chilled Water Control Settings

k. Cond. Water Control Setting

l. Low Pressure Cutout Setting

m. High Pressure Cutout Setting

11.3.6.3.5 Refrigeration Data

a. Oil Level Checked

b. Oil Failure Sw. Diff.

c. Refrigeration Level Checked

d. Relief Valve Setting

e. Unloader Setpoints

f. % Cylinders Unloaded

g. Purge Operation Checked

h. Bearing Temperature

i. Vane Position

j. Demand Limit

k. Low Temperature Cutout Setting

11.3.7 Package Rooftop/Heat Pump A/C Unit Test Report

11.3.7.1 Test Data from package units of all types is to be recorded on this form. If the unit has components other than the evaporator fan, DX coil, compressor and condenser fan(s), use the appropriate test report form for: water or steam coils, direct fired heaters, electric coils, or return air fans.

11.3.7.2 Data Reported (list design and actual quantities, where appropriate)

11.3.7.2.1 Unit Data

a. Make/Model Number

b. Type/Size

c. Serial Number

d. Filter Type/Size

e. Fan Sheave Make

f. Fans Sheave Diameter/Bore

g. No. Belts/Make/Size

h. Type of Heating Section* (*use other appropriate form)

11.3.7.2.2 Motor Data

a. Make/Frame

b. HP/RPM (w/Rad/s)

c. Volts/Phase/Hertz

d. Full Load Amps/S.F.

e. Sheave Make

f. Sheave Diameter/Bore

g. Sheave Centerline Distance and Adjustment

11.3.7.2.3 Evaporator Test Data

a. Total Airflow Rate

b. Total Static Pressure

c. Discharge Static Pressure

d. Suction Static Pressure

e. Outside Airflow Rate

f. Outside Air DB/WB

g. Return Airflow Rate

h. Return Airflow DB/WB

i. Entering Air DB/WB

j. Leaving Air DB/WB

k. Fan RPM

l. Voltage, T1-T2, T2-T3, T3-T1

m. Amps, A1, A2, A3

11.3.7.2.4 Condenser Test Data

a. Refrigerant/Weight

b. Compressor Manufacturer/Number

c. Compressor Model/Serial Number

d. Low Ambient Control

e. Suction Pressure/Temperature

f. Condenser Pressure/Temperature

g. Crankcase Motor Amps

h. Compressor Volts, T1-T2, T2-T3, T3-T1

i. Compressor Amps, A1, A2, A3

j. Low Pressure/High Pressure Cutout Setting

k. Number of Fans/Fan RPM

l. Condenser Fan HP/Airflow Rate

m. Condenser Fan Volts/Amps/Phase

11.3.8 Compressor and/or Condenser Test Report

11.3.8.1 This form may be used to record the control settings and the entering and leaving conditions at the unit. Since the balancing firm is not necessarily responsible for startup or the proper operation of the machine, this form does not attempt to indicate the performance or efficiency of the machine except as may be determined by the design engineer from the data contained therein.

11.3.8.2 This form, or the manufacturer form, should be substantially completed and verified by the manufacturer representatives and/or the equipment owner or installing contractor before the HVAC distribution systems are balanced. Temperature and pressure differential readings of the unit should be recorded during the TAB procedures.

11.3.8.3 This form may also be used to record data for the refrigerant side of unitary systems, "bare" compressors, separate air-cooled condensers, or separate water-cooled condensers.

11.3.8.4 Data Reported (list design and actual quantities, where appropriate)

11.3.8.4.1 Unit Data

a. Unit Make

b. Unit Model/Serial Number

c. Compressor Make

d. Compressor Model/Serial Number

e. Refrigerant Weight

f. Low Ambient Control

11.3.8.4.2 Test Data

a. Duct Inlet/Outlet Static Pressure

b. Entering/Leaving Air DB

c. Cond. Water Temperature In/Out

d. Cond. Water Pressure In/Out

e. Control Setting

f. Unloader Setpoints

g. Low Pressure/High Pressure Cutout Setting

h. Suction Pressure/Temperature

i. Cond. Pressure Temperature

j. Oil Pressure/Temperature

k. Voltage, T1-T2, T2-T3, T3-T1

l. Amps, A1, A2, A3

m. kW Input

n. Crankcase Heater Amps

o. Number of Fans/Fan RPM/Airflow Rate

p. Fan Motor Make/Frame/HP

q. Fan Motor Volts/Amps

11.3.9 Cooling Tower or Condenser Test Report

11.3.9.1 This form should be substantially completed and verified before the system is balanced. The "pump data" section is to be used for the re-circulating pump in evaporative condensers, not the system used with cooling towers (use Pump Test Report).

11.3.9.2 Data Reported (list design and actual quantities, where appropriate)

11.3.9.2.1 Unit Data

a. Make/Type

b. Model Number/Size

c. Serial Number

d. Nominal Capacity

e. Refrigerant

f. Water Treatment

11.3.9.2.2 Pump Data

a. Make/Model

b. Pump Serial Number

c. Motor Make/Frame

d. Motor HP/RPM (w/Rad/s)

e. Volts/Phase/Hertz

f. Water Flow Rate

11.3.9.2.3 Fan Data

a. Number of Fan Motors

b. Motor Make/Frame

c. Motor HP/RPM

d. Volts/Phase/Hertz

e. Motor Sheave Diameter/Bore

f. Fan Sheave Diameter/Bore

g. Sheave Centerline Distance

h. Number of Belts/Make/Size

11.3.9.2.4 Air Data

a. Duct Airflow Rate

b. Duct Inlet Static Pressure

c. Duct Outlet Static Pressure

d. Average Entering/Leaving WB

e. Ambient WB

11.3.9.2.5 Water Data

a. Entering/Leaving Water Pressure

b. Water Pressure Drop

c. Entering/Leaving Water Temperature

d. Water Temperature Drop

e. Water Flow Rate

f. Bleed Water Flow Rate

11.3.10 Heat Exchanger/Converter Test Report

11.3.10.1 This form is designed to record final conditions for steam or hot water heat exchangers.

11.3.10.2 Data Reported (list design and actual quantities, where appropriate)

11.3.10.2.1 Unit Data

a. Location

b. Service

c. Make/Type

d. Model Number/Size

e. Serial Number

f. Rating

11.3.10.2.2 Steam Test Data

a. Pressure

b. Flow Rate

11.3.10.2.3 Primary Water Test Data

a. Entering/Leaving Temperature

b. Temperature Drop

c. Entering/Leaving Pressure

d. Pressure Drop

e. Water Flow Rate

11.3.10.2.4 Secondary Water Test Data

a. Entering/Leaving Temperatures

b. Temperature Differential

c. Entering/Leaving Pressure

d. Pressure Differential

e. Water Flow Rate

f. Control Setpoint

g. Circuiting Type

11.3.11 Pump Test Report

11.3.11.1 This report form may be used as a work sheet. The final data on each pump is also recorded on this form. The actual impeller diameter entry is that indicated by plotting the head curve based on a no flow head test or by actual field measurement where possible.

11.3.11.2 Net Positive Suction Head (NPSH) is important for pumps in open circuits and for pumps handling fluids at elevated temperatures. (NPSH is defined in Section E.1.6.)

11.3.11.3 Data Reported

11.3.11.3.1 Design Data

a. Service/Location

b. Make

c. Model Number

d. Serial Number

e. Water Flow Rate/Head

f. Required NPSH

g. Pump RPM

h. Impeller Diameter

i. Motor Make/Frame

j. Motor HP/RPM (w/Rad/s)

k. Volts/Phase/Hertz

l. Full Load Amps/S.F.

m. Seal Type

11.3.11.3.2 Actual Test Data

a. Number of Flow Heads

b. Actual Impeller Diameter

c. Full-open Head

d. Full-open Flow Rate

e. Final Discharge Pressure

f. Final Suction Pressure

g. Final Head

h. Final Flow Rate

i. Voltage, T1-T2, T2-T3, T3-T1

j. Amps, A1, A2, A3

11.3.12 Boiler Test Report

11.3.12.1 This form may be used as a check sheet to record the control settings and the entering and leaving conditions at the boiler. Since the balancing firm is not necessarily responsible for startup or the proper operation of the machine, this form does not at-tempt to indicate the performance or efficiency of the boiler except as may be determined by the design engineer from the data contained therein.

11.3.12.2 This form, or the manufacturer form, should be substantially completed and verified by the manufacturer representatives and/or the installing contractor before the HVAC distribution systems are balanced. Temperature and/or pressure readings of the boiler should be entered during the TAB procedures.

11.3.12.3 A flue gas analysis normally is not in the scope of TAB procedures, but data could be added in the "remarks" section if available and required by the engineer/owner.

11.3.12.4 Data Reported (list design and actual quantities, where appropriate)

11.3.12.4.1 Unit Data

a. Location/Service

b. Make/Type

c. Model Number/Size

d. Serial Number

e. Fuel/Input

f. Number of Passes

g. Ignition Type

h. Burner Control

i. Volts/Phase/Hertz

11.3.12.4.2 Test Data

a. Operating Pressure/Temperature

b. Entering/Leaving Temperature

c. Number of Safety Valves/Size

d. Safety Valve Settings

e. High Limit Setting

f. Operating Control Setting

g. High Fire Setpoint

h. Low Fire Setpoint

i. Voltage, T1-T2, T2-T3, T3-T1

j. Amps, A1, A2, A3

k. Draft Fan Volts/Amps

l. Manifold Pressure

m. Safety Controls Check

11.3.13 Instrument Calibration Report

11.3.13.1 This form is to be used for recording the application and date of the most recent calibration test or calibration for each instrument used in the testing, adjusting, and balancing work covered by the report.

11.3.13.2 Data Reported

a. Instrument/Make

b. Serial Number

c. Application

d. Dates of use

e. Date(s) of Calibration

11.3.14 Component Failure Report

11.3.14.1 This form is intended to provide enough information to determine cause of failure and provide feedback to the manufacturer, designer, or installer. This form should be used as soon as a problem has occurred, and inclusion in the final report would be at the judgment of the balancer. It should be noted on the report, if appropriate, that the analysis and recommendations are not to be considered final or made by an expert on the subject.

11.3.14.2 Data Reported

a. Project

b. System

c. Component

d. Manufacturer

e. Serial Number

f. Model Number

g. Date

h. Architect/Engineer

i. Contractor

j. Submittal Data

k. Description and Problem

l. Field Test Results

m. Probable Cause

n. Recommendations

12. COMMISSIONING FOR TEST AND BALANCE

12.1 Commissioning is the process of verifying and documenting that the HVAC systems meet the requirements of the owner and the intentions of the design team. Commissioning should be implemented during the design phase and be carried through the occupancy phase and possibly further until the facility is 100% occupied. It is imperative that the design team and owner agree to the amount of testing, who will sign off on the tests, and the amount of time required.

12.2 Before final commissioning can take place, an approved test and balance report must be submitted to the commissioning authority. Depending on the experience of the TAB agency, the balance report can be used for functional performance testing provided a control point verification is included.

12.3 If required by the specification, the TAB agency shall submit to the commissioning authority a TAB plan. This plan should describe the systems to be commissioned, the format for reporting, an approved instrument list with calibration dates, resumes of personnel who will be doing and are responsible for the test and balance, dates various pieces of equipment will be tested, how to test special equipment if designed and/or installed, and a time period for correctional work to be accomplished and the TAB personnel to return to complete the project.

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