TESTING ADJUSTING BALANCING MANUAL FOR TECHNICIANS

NEBB® National Environmental Balancing Bureau

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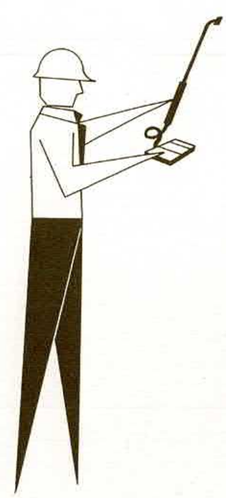

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NATIONAL ENVIRONMENTAL BALANCING BUREAU

TESTING, ADJUSTING, BALANCING MANUAL FOR TECHNICIANS

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This manual was developed using reliable engineering principles and research plus consultation with, and information obtained from, manufacturers, users, testing laboratories and others having specialized experience. The manual is subject to revision as further experience and investigation may show is necessary or desirable. System balancing which complies with this manual will not necessarily be acceptable if, when examined and tested, is found to have other features which impair the result contemplated by these procedures. Further, the National Environmental Balancing Bureau assumes no responsibility and accepts no liability for the application of the principles or techniques contained in this manual.

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FOREWORD

As heating, ventilating, and air conditioning systems, or environmental systems as they now are called, continue to become more sophisticated and complicated, testing, adjusting and balancing (TAB) technicians no longer can be just meter readers. Not only must they be able to competently measure and record the necessary data, but they must be able to understand fully how to perform their work. Furthermore, they must be knowledgeable of the fundamentals of how these environmental systems function, must determine quickly what is malfunctioning, and then must be able to correct the problem.

This manual was written to be not only a basic educational text on testing, adjusting and balancing work, but to be a comprehensive reference manual that will be carried to every TAB project along with the necessary tools and instruments. By using the table of contents and the index, answers to many common TAB problems may be found along with the basic methods and/or procedures needed to arrive at the solutions to these problems.

It is the desire of the NEBB Research and Development Committee that this *Testing, Adjusting and Balancing Manual for Technicians* will not only make the work of TAB technicians easier to do and understand, but that they will find their job more interesting and productive.

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INTRODUCTION

The first purpose of this manual is to provide a person who has a basic mechanical aptitude with the fundamental requirements to become a Testing, Adjusting and Balancing (TAB) technician who can successfully perform testing, adjusting and balancing of HVAC systems while under the direction of a qualified NEBB Supervisor.

The second purpose of this manual is to provide the experienced TAB technician with an easy to use reference containing all of the necessary procedures, tables, check lists, equations and equivalents to use in the field.

A TESTING, ADJUSTING AND BALANCING (TAB)

Testing, Adjusting and Balancing may be defined as the three major steps used on the job to achieve proper operation of HVAC (heating, ventilating, and air conditioning) systems. These systems also are called "environmental systems."

1. Testing

Testing may be described as the use of specialized instruments to measure temperatures, pressures, rotational speeds, electrical characteristics, velocities and air and water quantities for an evaluation of equipment and system performance.

2. Adjusting

Adjusting may be described as the final setting of balancing devices such as dampers and valves, in addition to automatic control devices such as thermostats and pressure controllers to achieve maximum specified system performance and efficiency during normal operation.

3. Balancing

Balancing is the methodical regulation of system fluid flows (air or water) through the use of acceptable procedures to achieve the desired or specified flow quantities (CFM or GPM).

B TAB TEST REPORTS

The testing, adjusting and balancing (TAB) test reports are considered essential to the HVAC systems designer and installers to better enable them to evaluate the results of their design, the performance of the equipment and installation techniques under actual operating conditions. It is well known that no HVAC system installation is "perfect" and therefore does not require TAB work. However, with good engineering practices, realistic equipment performance ratings and good workmanship in the system installation, adequate results can be obtained to satisfy a given set of design conditions within a reasonable set of limitations. The TAB work then "fine tunes" the system to meet the actual field conditions.

The testing, adjusting and balancing of an HVAC system also is the means used to determine and monitor system performance and may be utilized again and again well after the project is completed.

Testing and Balancing reports also can be used:

- to assist personnel responsible for the efficient operation of the HVAC systems,
- as a record of existing conditions,
- to compare periodic tests to original conditions for determining deterioration or reduced efficiency if any exist,
- for existing conditions when modifications or changes have been made in the HVAC system,
- in energy conservation programs as existing conditions for base energy level calculations,
- for procedures and reports that can be used to verify energy conservation results, and
- for comparison of design versus actual field performance.

C NEBB

The National Environmental Balancing Bureau (NEBB) was formed as a nonprofit organization by the Mechanical Contractors' Association of America (MCAA) and the Sheet Metal and Air Conditioning Contractors' National Association (SMACNA) to establish and direct a management oriented national program to upgrade and maintain uniform standards for the testing, adjusting and balancing of environmental systems and for the measuring of sound and vibration in environmental systems.

It now is the largest international testing, adjusting and balancing organization in the world.

1. Certification Programs

a. PURPOSE

The purpose of NEBB certification programs is to offer tangible proof of competent firms qualified in the proper methods, skills and procedures for:

(1) the testing, adjusting and balancing (TAB) of environmental systems.

(2) the measuring of sound and vibration (S&V) in environmental systems.

(3) testing of Clean Rooms.

b. OBJECTIVES

The purpose will be accomplished by meeting the following objectives:

- To establish industry standards, procedures and specifications for the testing, adjusting and balancing of environmental systems and measuring sound and vibration in environmental systems.
- To set minimum educational standards and other requirements for the qualification of supervisory personnel employed by firms who perform this work.
- To establish an educational program of instruction, the purpose of which is to train supervisory personnel and technicians in the proper methods and procedures in the testing, adjusting and balancing of environmental systems, and/or measuring sound and vibration in environmental systems; and to this end, to accredit NEBB schools established by local Chapters.
- To certify as qualified for performance or supervision of testing, adjusting and balancing of environmental systems, and/or measuring sound and vibration in environmental systems; those firms who meet the requirements for certification as established by NEBB, who comply with the objectives of NEBB, and who employ supervisory personnel who have met the qualifications as established by NEBB.
- To serve as a focal point for educational and technical materials pertinent to testing, adjusting and balancing and measuring sound and vibration of environmental or HVAC systems, and clean room testing.
- To promote the concept of total responsibility for the testing, adjusting and balancing of environmental systems, measuring sound and vibration in environmental systems, and the testing of clean rooms.

2. What NEBB Does

a. NEBB ACHIEVES ITS OBJECTIVES BY:

(1) Creating national standards, procedures and programs.

(2) Establishing Local or Regional Chapters which implement and promote NEBB programs through:

- participation in the certification process,
- courses of instruction for the training of supervisors, and
- a triparite (engineer, contractor, and owner or building official) review board which evaluates contractor compliance with NEBB's standards and procedures—in the event a dispute regarding such compliance arises between a certified firm and the consulting engineer or owner's representative.

(3) Establishing professional qualifications for TAB supervisors which include management responsibilities, reputable conduct, extensive experience, passing of appropriate written examinations and demonstration of certain practical working knowledge and proficiency in the use of instruments required for effective TAB.

(4) Certifying qualified firms. Certification involves strict conformance to the high standards and procedures established by NEBB, the employment of qualified supervisors, the possession of certain necessary instruments, a competent instrument maintenance program and use of NEBB Reporting Forms.

(5) Encouraging the advisory participation of design engineers, technical societies, governmental bodies, owners' organizations and manufacturers.

(6) Requiring review of contract plans and specifications to confirm that adequate provisions for testing and balancing have been included.

b. NEBB CERTIFICATION

NEBB certification is assurance that the NEBB certified firm has:

(1) A Qualified TAB Supervisor

A responsible management-level person supervising the job when the environmental system is tested, adjusted and balanced to specifications. The supervisor must meet stringent standards which include passing college level written examinations, practical examinations and experience requirements.

(2) Proper Instrumentation

A complete set of instruments required for the sophisticated techniques and procedures necessary to "fine-tune" modern day environmental systems.

(3) Maintained Proficiency for TAB Work

A contractor, once having met NEBB's rigid requirements for certification, must continue his technical competence in testing, adjusting and balancing. Maintaining NEBB certification requires that:

- the contractor's certification must be reviewed and renewed every two years,
- the contractor must maintain a NEBB-qualified supervisor on his staff in order to maintain certified status, and
- the contractor's TAB supervisor must renew his qualification every two years. Among other requirements, the supervisor must keep abreast of developments in TAB work by attending and successfully completing periodic TAB seminars.

c. NEBB ASSURES COMPETENCE, QUALITY AND CONFIDENCE

The NEBB program affords building owners, architects, engineers and other agents a reliable basis for specifying testing, adjusting and balancing work. It provides the industry with contractors highly competent in the TAB field and proper execution of TAB projects by insuring compliance with NEBB standards and procedures.

d. NEW DIMENSIONS TAB—SOUND AND VIBRATION

The professional excellence NEBB brought to Air and Hydronics TAB has expanded to additional fields—Sound and Vibration Measurement. NEBB has an extensive program of standards, instructional materials and educational programs for sound and vibration measurements and testing. This discipline adds new dimensions to NEBB capabilities to serve needs of the industry.

e. THE NEBB SPECIFICATION

The NEBB specification for testing, adjusting and balancing work can be incorporated in the contract documents by the specifying authority. NEBB TAB Supervisors are fully cognizant of the obligations contained in the NEBB Specification, have received training and have passed examinations to comply with the certification requirements, and annually attend additional TAB educational sessions to update their expertise.

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CHAPTER 1 HVAC FUNDAMENTALS

This Chapter contains the basics of "heat flow" and the "properties of air" which sound somewhat uncomplicated. However, when "thermodynamics" (heat flow) and "psychrometrics" (air-vapor relationships) are mentioned, many people take a glance at the material and "give-up."

It is not necessary that you thoroughly understand these subjects, but it is necessary that you can follow what is happening to heat transfer and air and hydronic systems as they pertain to HVAC systems.

A HEAT TRANSFER

A basic understanding of heat transfer principles is important to all TAB technicians. After all, it is not just the quantity of air and water flow that is important to a functional system, but that the proper rates of heat transfer are maintained.

1. Heat Intensity

The intensity of heat of substance is called *temperature* and is measured by a thermometer of other temperature indicating device. The Fahrenheit scale is used in the United States, while the Celsius (formerly called Centrigrade) scale is mostly used elsewhere.

The following equation can be used to convert temperatures from the Celsius scale to the Fahrenheit scale:

Equation 1-1

$$°F = 1.8°C + 32°$$

Equation 1-2 can be used to convert Fahrenheit scale temperatures to Celsius scale temperatures:

Equation 1-2

$$°C = \frac{(°F - 32°)}{1.8}$$

The temperature at which the continued removal of heat from a substance results in the substance having no molecular action is called *absolute zero,* which is minus 460°F on the Fahrenheit scale and minus 273°C on the Celsius scale. The thermodynamic absolute temperature (T) used in temperature/pressure/calculations can be obtained in degrees Rankine by using Equation 1-3, and in degrees Kelvin by using Equation 1-4. The relationship between the temperature scales is shown in Figures 1-1 and 1-2.

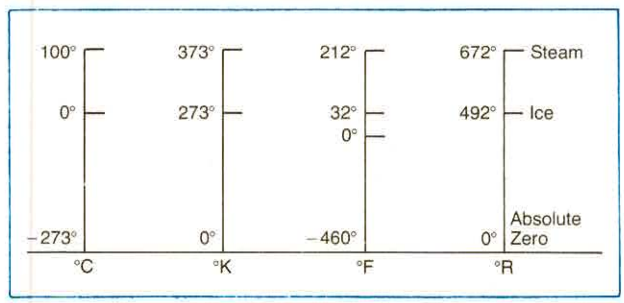


Figure 1-1 RELATIONSHIP BETWEEN TEMPERATURE SCALES

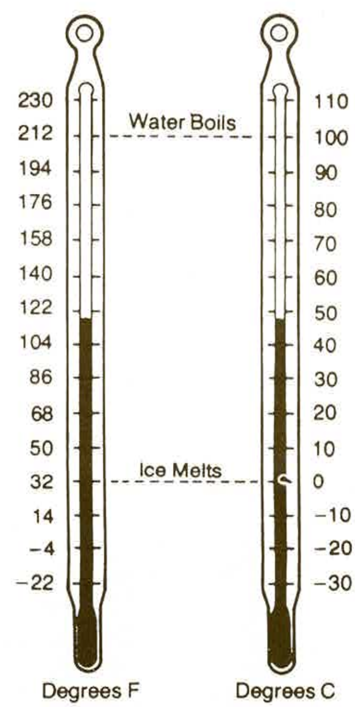


Figure 1-2 COMPARISON OF FAHRENHEIT AND CELSIUS THERMOMETERS

Equation 1-3

$°T = °F + 460° = °R$

Where:

$°T$ = Absolute temperature (Rankine Scale [°R])
$°F$ = Fahrenheit temperature

Equation 1-4

$°T = °C + 273° = °K$

Where:

$°T$ = Absolute temperature (Kelvin Scale [°K])
$°C$ = Celsius temperature

2. Heat Quantity

The quantity of heat is found by measuring the temperature and weight of a substance. In the United States the quantity or amount of heat in a substance is measured in *British Thermal Units* (Btu) which is defined as the amount of heat required to raise the temperature of one pound of water from 59°F to 60°F. In the metric system, the unit of measurement is called calorie which is the amount of heat required to raise the temperature of one gram of water from 4°C to 5°C.

It is easy to realize that it takes far more Btu's to heat a swimming pool from 94°F to 95°F than it does to heat a cup of coffee the same one degree. The Btu is seldom used without further definition either as flow rate with time or as a limited quantity of heat contained in some weight or volume of matter. By using the Btu per hour (Btuh), it is possible to determine the amount of heat flowing or heat transferred in a given amount of time. Using weight for mass, it is possible to establish the heat contained in a substance used as a heat source, such as Btu per pound of coal, or Btu per pound of air.

Example 1A

How many Btu's are required to increase the temperature of 8.33 pounds of water (1 gallon) from 80 degrees fahrenheit to 120 degrees fahrenheit?

Solution

8.33 pounds × (120°F − 80°F)
× (1 Btu/lb water) = 333.2 Btu

As seen in Example 1A, heat transfer in Btu's may be determined when the quantity and temperature difference of water is known. However, when applied to HVAC heat transfer equipment, time is also considered allowing a rate of heat transfer to be established. In other words, how many Btu's may be transferred in a given time period such as one minute or one hour. The most common heat transfer rates and their definitions are given below:

Btuh (BRITISH THERMAL UNITS PER HOUR)—Represents how many Btu's are transferred in a one hour period.

Mbh (THOUSAND BTU'S PER HOUR)—Represents how many thousands of Btu's are transferred in a one hour period.

NOTE: 1 mbh = 1000 Btuh

Ton (TON OF REFRIGERATION)—Represents a heat transfer rate of 12000 Btuh or 12 mbh. Heat transfer rates of air conditioning and refrigeration equipment are normally expressed in tons or tonnage.

3. Methods of Heat Transfer

In HVAC systems, as in natural processes, heat is transferred by three means:

(a) radiation
(b) convection
(c) conduction

a. RADIATION

Radiation is a form of energy transfer similar to that of light waves and radio waves, without heating the intervening space. Energy waves of the sun, for example, can be felt by a person until a heavy cloud layer passes in front of it. The change is felt immediately, but the air in the space in between was not heated directly by the sun's rays. Heat from an outdoor campfire or an infrared heater is another example of radiant heat which is readily felt even when the temperature is quite cold, but only on the side facing the fire or heat.

b. CONVECTION

Most of the heat transfer in the HVAC industry is by *convection*. Convection is the transfer of heat by movement of a fluid such as air or water over a substance. The heat flow can be either to or from the substance or object. For example, air flowing over a bank of hot pipes will become heated by transfer of heat from the pipes to the air. If, however, the pipes were cold and the air was warm, the air could be cooled by transferring heat to the colder pipes. Natural convection occurs when cool air surrounds a hot object, becomes heated, and then the heated air rises allowing the surrounding cooler air to move in to be heated by the object. In a similar manner, chilled air will fall, allowing the warmer surrounding air to come into contact with the cold object, basically reversing the above process.

When a fan is used to propel the air across a hot or cold surface, heat transfer generally increases with an increase in air velocity. *Forced convection* (air moved across the surface by a fan) therefore is a more efficient method of heat transfer which produces a greater volume of transferred heat. Major factors in the transfer of heat by convection are:

(a) temperature difference
(b) flow velocity
(c) type of fluid (or gas)
(d) conductivity of heat transfer material
(e) size and shape of the transfer surfaces
(f) condition of the transfer surfaces

c. CONDUCTION

Conduction is the flow of heat through a substance or the flow of heat from one body to another when the bodies are in direct physical contact with one another. The heating of the handle of a poker placed in a fire is a good example. It is common knowledge that the ability of various materials to conduct heat differs considerably. The best conductors of heat are metals such as aluminum and copper but glass also is a good conductor of heat. Poorer conductors such as wood, mineral wool, air, cork, etc. are called insulators.

d. CONDUCTIVITY (k)

The ability of a substance to transfer heat by conduction is called *thermal conductivity* (k). Conductivity is defined as the amount of heat in Btu per hour flowing through one inch of thickness of one square foot of a homogeneous materials when the difference in temperature between the faces is one degree Fahrenheit. Therefore, materials having the lowest conductivity numerical values are the best insulators.

e. CONDUCTANCE (C)

Thermal conductance (C) is a property of an object made of nonhomogeneous material such as hollow clay tile or concrete blocks where each succeeding inch of thickness is not identical with the preceding inch. Therefore, it is necessary to indicate the heat flow rate through the entire object. Conductance is defined as the heat flow rate in Btu per hour per one square foot of nonhomogeneous material of a certain specified thickness for a one degree difference in temperature between the two surfaces of the material. Care should be taken not to confuse conductivity and conductance.

f. RESISTANCE (R)

Almost everyone is now familiar with "R values" as all insulation and most homes being sold have these terms in the salespersons vocabulary. R values can be added together along with the "k values" and "C values". *Thermal resistance* (R) or resistivity is the reciprocal of the heat transmission coefficient (U). The overall resistance (R_t) is equal to the sum of the resistances and resistivities of the insulation and substances from which the wall, ceiling, floor, etc. is built. The coefficient of heat transfer "U" can be obtained by taking the reciprocal of the resistance as shown in the following equation:

Equation 1-5

$$U = \frac{1}{R_t} = \frac{1}{R_1 + R_2 + R_3 \ldots + R_n}$$

Where:

R_t = overall resistance total

U = coefficient of heat transfer

$R_1 + R_2 \ldots R_n$ = individual resistances

Equation 1-6

$$Q = A \times U \times \Delta t$$

Where:

Q = the rate of heat transfer of flow (Btuh)

A = the area of a surface (square feet)

U = coefficient of heat transfer

Δt = °F temperature difference between the temperatures on each side of the surface.

The lower that the value of "U" is in Equation 1-6 ($Q = A \times U \times \Delta t$), the lower the heat gain or loss is through a building surface. As "U" is the reciprocal of "R" (Equation 1-5), the value of "U" decreases as the value of "R" increases.

The importance of increasing insulation "R" values of walls and ceilings of a building is obvious as is reducing the infiltration loads.

Keeping the "Δt" low also is another method of reducing the heat flow. Winter heat flow in northern climates has a much greater "Δt" than summer heat flow. For example, a building gains heat in the summertime when the outside temperature is 95°F and the inside temperature is 78°F (Δt = 17°F) because the heat must flow from the outside to the inside (always warmer to the cooler). The building loses heat in the wintertime when the inside temperature is 72°F and the outside temperature is 0°F (Δt = 72°F).

It is not necessary for the TAB technician to become expert in building load calculations, as that is the function of the HVAC system designer. It is not necessary that one even "speak this language." However, be receptive to the various terms, concepts and issues that are important to the designer. Building heat gain calculations will vary from designer to designer for the same building and the same temperatures. It's not that the science is poorly developed or even poorly applied. The difference arises from the many complex variables that must be used to accurately determine the heat flow into and out of a building.

Example 1B

An exposed wall of 800 square feet in a building has a "U" factor or coefficient of heat transfer of 0.78. By insulating the wall, "U" becomes 0.08. In a 10 hour period with 65°F inside and 0°F outside, how much heat would be saved?

Solution

$Q = A \times U \times \Delta t$

$Q = 800 \times (0.78 - 0.08) \times (65° - 0°)$

$Q = 800 \times 0.7 \times 65$

$Q = 36{,}400$ Btuh difference

Heat saved = 10 hr. × 36,400 Btuh = 364,000 Btu

4. Types of Heat Transfer

Of the many types of heat found in thermodynamics, there are two basic types of heat transfer a TAB technician must be aware of: sensible heat and latent heat.

a. SENSIBLE HEAT

Sensible heat is any heat transfer that causes a change in temperature. Heating and cooling of air and water that may be measured with a thermometer is sensible heat. Heating or cooling coils that simply increase or decrease the air temperature are examples of sensible heat.

b. LATENT HEAT

Latent heat is any heat transfer that causes a change of state from a solid to a liquid, a liquid to a gas, or vice versa. Evaporation of water is an example of a latent heat transfer. Latent heat transfer at terminal coils may be defined as any process which humidifies or dehumidifies the air. Both processes result in a change of actual moisture content in the air.

c. TOTAL HEAT (ENTHALPY)

Total heat is the sum of the sensible heat and latent heat in an exchange process. In many cases, the addition or subtraction of latent and sensible heat at terminal coils appears simultaneously. Total heat also is called *enthalpy,* both of which can be defined as the quantity of heat energy contained in that substance.

At any given time, a substance has only one value of enthalpy and a related specific temperature value on the thermometer. If the enthalpy is increased, the temperature increases. Conversely, if the temperature is decreased, the enthalpy decreases. The ability to increase or decrease enthalpy and temperature together is the basis for heat transfer in environmental systems and only *differences* in enthalpy and temperature are normally of importance.

B FLUID MECHANICS

To fully understand the heat transfer process and the consequences to the TAB procedures, it becomes necessary to understand a little about the fluids involved and their individual properties and characteristics. For the purposes of TAB work, fluids may be considered to be any gas or liquid used to transfer heat. The relationship of the physical properties of a fluid is the subject of fluid mechanics.

1. Fluid Properties

The basic categories of fluid properties are: state, compressibility, viscosity, weight or density, volume as specific volume, violatility, vapor pressure, specific heat and heat content. These properties will be discussed only as they relate to environmental or HVAC systems.

a. STATE

The state of a fluid refers to its form, either liquid or gas. Liquids used in environmental systems are water, thermal fluids such as ethylene glycol solutions, and refrigerants in the liquid state. Gases are steam, evaporated refrigerants and the air-water vapor mixture found in the atmosphere. Some substances, including commonly used refrigerants, may exist in any of three states. A simple example is water, which may be solid (ice), liquid (water), or gas (steam or water vapor). Figure 1-3, Temperature/Heat Diagram, indicates the heat required to convert ice at minus 40°F (Point A) to steam near 280°F (Point F). The extra heat (144 Btu per pound) required from Point B to C is called *heat of fusion* and the extra heat (970 Btu per pound) required from Point D to E is called *heat of vaporization.*

b. COMPRESSIBILITY

Compressibility of a fluid, the ease which it may be reduced in volume by the application of pressure, depends upon the state of the fluid as well as the type of fluid itself. In TAB work, consider that water may not be compressed. Air is a compressible gas, but

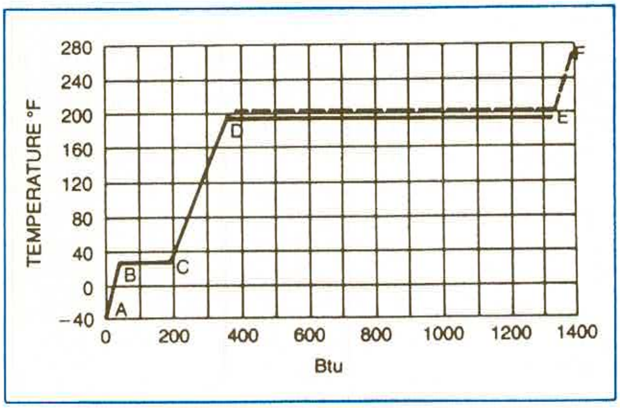


Figure 1-3 TEMPERATURE/HEAT DIAGRAM

that factor is usually not considered during normal testing and balancing procedures. See Chapter 2 of ENVIRONMENTAL SYSTEMS TECHNOLOGY for further information regarding fluid compressibility.

c. VISCOSITY

The Viscosity of a fluid refers to the liquid state and to the ease with which it flows, or the difficulty experienced in making it flow. The higher the viscosity of the fluid, the greater the pressure that is needed to cause it to flow in piping, thus requiring more energy.

d. WEIGHT

The weight of a substance is the amount of force it exerts under pull by the earth's gravitational field and that force is measured in pounds in the United States.

e. DENSITY

The density of a substance relates to the nearness and the number of particles or molecules of the substance in a given volume. It is recognized that it is easier to walk in still air than in still water, so it can be assumed that the density of air is much less than that of water. Density is referred to in terms of units of weight per unit of fixed volume, and when used in environmental systems, pounds per cubic feet is used.

f. STANDARD CONDITIONS

The standard conditions referred to in environmental system work for air are: *dry* air at 70°F, and at an atmospheric pressure of 29.92 inches mercury (in.Hg.). For water, standard conditions are 68°F at the same barometric pressure. At these standard conditions, the density of air is 0.075 pounds per cubic feet and the density of water is 62.4 pounds per cubic foot.

g. SPECIFIC VOLUME

Specific volume is the reciprocal of density and is used to determine the cubic feet of volume, if the pounds of weight are known. Both density and specific volume are affected by temperature and pressure. The specific volume of air under standard conditions is 13.33 cubic feet per pound and the specific volume of water at standard conditions is 0.016 cubic feet per pound.

h. VOLATILITY

Volatility, surface tension and capillary action of a fluid are incidental to environmental systems. *Volatility* is the rapidity with which liquids evaporate. Gasoline, for example, evaporates extremely rapidly and therefore is highly volatile.

i. VAPOR PRESSURE

Vapor pressure denotes the lowest absolute pressure that a given liquid at a given temperature will remain liquid before evaporating into its gaseous form or state.

j. SPECIFIC HEAT

Specific heat (C_p) is the amount of heat energy in Btu's required to raise the temperature of one pound of substance one degree Fahrenheit. The following are specific heat values at standard conditions:

water—$C_p = 1.00$ Btu/lb°F
air—$C_p = 0.24$ Btu/lb/°F

Using these values in simple equations, gallons per minute or cubic feet per minute may be determined in a system if the Btu per hour and the temperature difference are known. This relationship will be discussed later.

2. Air-Heat Flow Equations

a. SENSIBLE HEAT

Sensible heat was defined as the heat associated with temperature differences as measured by a dry bulb thermometer. The sensible heat flow equations for air are:

Equation 1-7

For standard air conditions:

$$Q\ (\text{Sens.}) = 1.08 \times \text{cfm} \times \Delta t$$

Where:

Q = Heat flow (Btuh)
cfm = Airflow (cu. ft/min)
Δt = Temperature difference (°F)

Equation 1-8

For non-standard air conditions:

$Q\ (\text{Sens.}) = 60 \times C_p \times d \times cfm \times \Delta t$

Where:

C_p = Specific heat (Btu/lb °F)

d = Density (lb/cu ft)

(For standard air, $C_p = 0.24$ and $d = 0.075$.)

b. LATENT HEAT

Latent heat is the heat used to convert a liquid into a gas or vapor without a change in dry bulb temperature (such as water boiling at 212°F) or the heat released when vapor condenses into a liquid, again without a change in dry bulb temperature.

Equation 1-9

$Q\ (\text{Latent}) = 4840 \times cfm \times \Delta W$

Where:

Q = Heat flow (Btuh)

cfm = Airflow (cu. ft./min)

ΔW = Humidity Ratio (lb. H_2O/lb. dry air)

c. TOTAL HEAT (ENTHALPY)

Changes of the *enthalpy* or *total heat* content of air use the following equation:

Equation 1-10

$Q\ (\text{Total}) = 4.5 \times cfm \times \Delta h$

Where:

Q = Total heat flow (Btuh)

cfm = Airflow (cu. ft./min)

Δh = Enthalpy difference (Btu/lb. dry air)

3. Hydronic-Heat Flow Equation

The heat flow equation used for water systems is:

Equation 1-11

$Q = 500 \times gpm \times \Delta t$

Where:

Q = Heat Flow (Btuh)

gpm = Gallons per minute (water only)

Δt = Temperature difference (°F)

4. Fluid Statics

Fluid Statics as applied to TAB work, refers to a condition of a quantity of fluid at rest. It is the direct result of gravity and weight. Static pressure is used in both air and water testing to determine the potential for the movement of fluid within a system. Pressures in air systems are normally measured in units of inches of water (in.w.g.). A pressure unit of one inch of water is equivalent to the static pressure found at the base of a column of water one inch high. Pressures in water systems are normally measured in pounds per square inch (psi), but are converted to feet of water (ft.w.g.) for the purpose of evaluating pump and equipment performance. Figure 1-4 indicates the relationship of static heads and gauge pressures.

5. Fluid Dynamics

Fluid Dynamics is used to describe the condition of motion of a fluid within a system. The velocity of a fluid is based upon the cross-sectional area and the volume of a fluid passing through it. The importance of this property is that volume may be determined for air or water systems when the area and velocity are known. The following equation is the basis for determining all flow quantities during the TAB procedure:

Equation 1-12

$Q = (A) \times (V)$

Where:	Air	Water
Q = Volume of fluid flow	(CFM)	(GPM)
A = Cross-sectional area	(sq. ft.)	(sq. ft.)
V = Velocity	(fpm)	(fps)

6. Friction

Friction is the resistance found at the duct and piping walls. Resistance creates a static pressure loss in systems. The primary purpose of a fan or pump is to produce a design volume of fluid at a pressure equal to the frictional resistance of the system and the other dynamic pressure losses of the components. Later chapters will review pump and fan pressures in addition to system pressure losses.

7. Absolute Pressure

It was established earlier that air at standard conditions (70°F air at sea level with a barometric pressure of 29.92 in.Hg.) exerted a pressure of 14.696 psi. This is the pressure in a system when the pressure gauge reads zero. So the *absolute pressure* of a system is the gauge pressure in pounds per square inch added to the atmospheric pressure of 14.696 psi (use 14.7 psi in *environmental system work*) and the symbol is "psia."

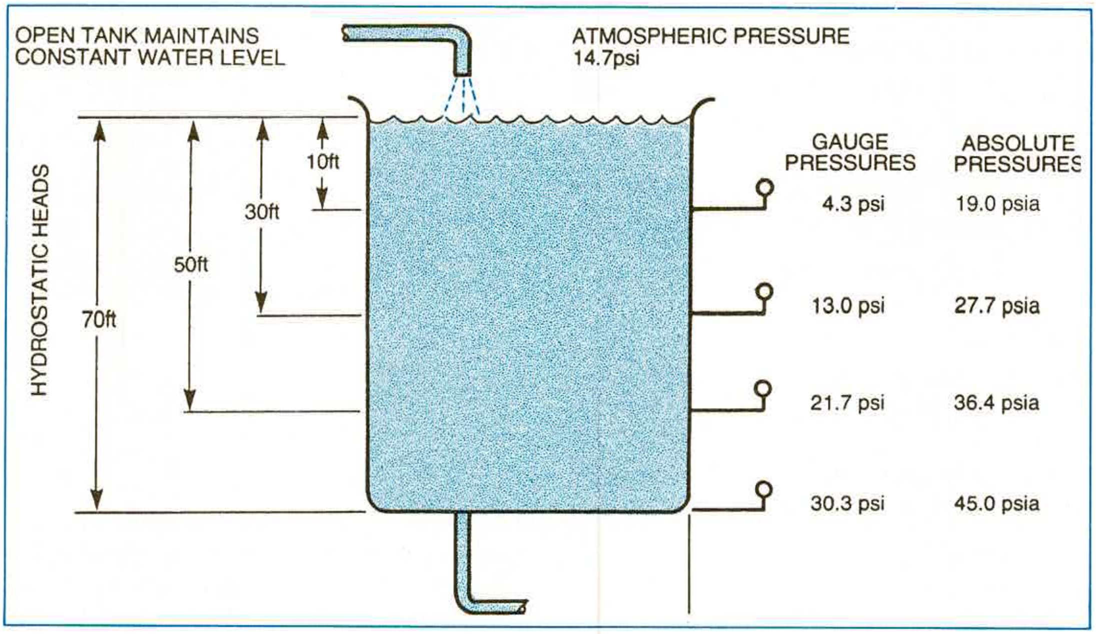


Figure 1-4 TANK STATIC HEAD

Equation 1-13

$P_a = P_g + 14.7$ psi

Where:

P_a = Absolute Pressure (psia)

P_g = Gauge Pressure (psi)

Example 1C

What is the absolute pressure of a system at the point where an accurate pressure gauge shows a reading of 10.0 psi?

Solution

Absolute Pressure $= P_g + 14.7 = 10.0 + 14.7$

$P_a = 24.7$ psia

C PRESSURES

The two main types of pressures used in TAB are *static* and *dynamic*. The static pressures are the easiest to understand. The pressure required to pump up a tire or to inflate a balloon is a static pressure. It is exerted equally in all directions which is what gives the distended balloon its spherical shape. The proof that it is exerted in all directions is simple. A hole anywhere in the balloon causes all the air to leak out.

This is not always true in dynamic situations. In certain locations within a piping system such as on the leaving side of an orifice, dynamic pressures can be *negative or zero.* Static heads also can be caused by heights of liquids or can be caused by external pressures. For example, the automatic cold water make-up valve in a boiler heating water system may be set between 15 and 30 pounds and it will operate to maintain that setting in the system. This then is the *base operating point of pressure.* All fluids below that level will be higher in pressure, all fluids above that datum line or height will be lower in pressure. *Static heads* are therefore important in hydronic systems, but they are so small for air systems, that they are ignored in all TAB work calculations.

In any system with a fan, pump or that has gravity flow, *dynamic heads* are developed. *Head* in this case only means resistance, friction, or pressure loss, which can be expressed in terms of *heights.* Dynamic pressures are caused by velocity and they act in the direction of flow. An example of this is putting one's hand into the airstream while a passenger in a car moving at 60 miles per hour. The pressure effects of the air can be felt upon your hand. As the velocity of the car decreases, so do the effects of the air pressures, until a stationary object is disturbed only by the wind currents in the air.

The *total pressures* within a system are the sum of its dynamic (velocity) and static pressures which are expressed in the following equation:

Equation 1-14

$TP = SP + V_p$

Where:

TP = Total pressure (in. w.g.)

SP = Static pressure (in. w.g.)

V_p = Velocity pressure (in. w.g.)

This equation is used extensively for air systems. Remember that a fluid flowing in a conduit at a certain velocity has a velocity pressure which is used to overcome the resistance of the walls of the conduit upon the medium.

However, in most hydronic system calculations for TAB work, the velocity pressure (V_p) is dropped from equations as the values are too small to affect the results.

D SYSTEM PRESSURES AND RESISTANCES

The pressure that a fan or pump overcomes is composed of three separate factors:

(1) Velocity head

(2) Friction losses

(3) Static head

Although *velocity head* is insignificant in pumped hydronic systems, when the fluid is air, velocity head is a significant factor. To illustrate the principles of friction losses and static head, assume that there is an ordinary garden hose stretched out on the ground, and that the bibcock has just been turned off. It is, therefore, still full of water but no water is flowing. To start water flow out of the end of the hose two conditions must take place. First, *pressure* must be applied to one end of the hose. In this case, by opening the bibcock or valve. The pressure within the house water system will force additional water into the hose, displacing the water already in the hose by forcing it out the open end. This could also be accomplished by disconnecting the hose from the bibcock and blowing air through it, again forcing the water out the open end. The second condition is that to move water through the pipe, the fluid must be replaced with another fluid. An equation has been developed for these conditions:

Equation 1-15

$$P = F/A$$

Where:

P = Pressure

F = Applied force

A = Area of the cross section

Equations 1-12 and 1-15 determine how much fluid moves through a pipe, hose, or duct in a given time period. Using these equations, notice what happens when "A", the cross-sectional area, becomes smaller. The pressure "P" is increased and the fluid flow rate "Q" gets smaller. Do not confuse "Q"—the fluid flow rate with "Q"—the heatflow rate used earlier. The same letter designations often are used for different terms.

To apply this change to the garden hose, when the hose is folded back on itself, the kink in the middle reduces the area that the fluid flows through. Naturally, the flow out of the end of the hose is reduced. It might also be observed that a little leak at the hose coupling to the bibcock now gets larger, an indication that the pressure has increased.

A height difference can also be applied to the garden hose application. If the nozzle is raised to the second floor, no pressure is required to cause the water to run out of the bottom end. But it's not possible to make water run out of the top end unless pressure is applied. In fact, pressure is required at the bottom end just to hold the water in the hose, and even more pressure is required to make it flow out the top end at the same time rate as when it was level. This is an illustration of *static head* as shown in Figure 1-5 for an opening piping system used with a cooling tower.

Remember that *fluid statics* refers to the pressures caused by height differences rather than by flow resistances. Again using the garden hose which was extended vertically, the bibcock is adjusted so that water barely trickles out of the very topmost nozzle of the hose. The pressure at the nozzle is at atmospheric pressure or 0 pounds per square inch gauge (psig). The pressure down at the bibcock is obviously much higher, because a 50 foot high column of water is pressuring down against it. The subsection, "Fluid dynamics" (the word dynamics means in motion) deals with the movement of fluids.

Example 1D

In Figure 1-5(A), if the static pressure "A" equals 10 feet and the suction head "B" equals 16 inches, calculate the static head in (a) feet, (b) psi, (c) in. w.g., (d) in. Hg.

Solution

(a) A − B = static head = 10′ − 1.33′
= 8.67 ft. of water.

Using equivalents from Table 12-11 in Chapter XII:

(b) 8.67 ft. × 0.433 = 3.75 psi

(c) 8.67 ft. × 12 = 104 in. w.g.

(d) 3.75 psi × 2.04 = 7.65 in. Hg.

Both static head and a smaller area result in increased pressures and decreased flow rates. These are examples of *resistances* which have been added to the system. Additional resistances or line restrictions in normal piping systems can be caused by coils, valves, and pipe fittings. Elbows, tees, and other types of restrictions can be expressed in *equiv-*

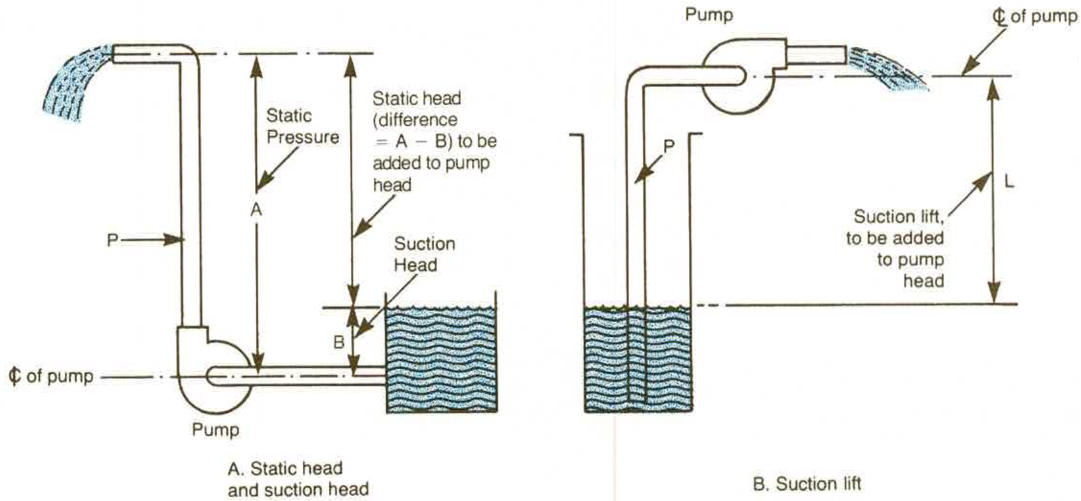


Figure 1-5 ILLUSTRATIONS OF STATIC HEAD, SUCTION HEAD, AND SUCTION LIFT

alent length of pipe in *feet.* This concept is very important in the sizing of hydronic systems.

Again using the garden hose for this illustration, 10 or 20 sections of 50 foot hose are added to the garden hose. When the bibcock is fully opened, it is found that the same amount of water does not come out of the long hose system as came out of the single garden hose. Length becomes part of the equation for flow, and the flow varies inversely according to length. Therefore, the longer the hose, the less the water flow because of the increased pressure losses due to increased resistance.

A kink in the hose might be worth 2 or 3 additional 50 foot hose sections. In other words, restrictions in the line may be represented by various lengths of straight pipe. In fact, all flow calculations used to be made in this manner for both air and hydronic systems. Design engineers still use charts and tables that indicate how many equivalent feet of pipe that one fitting is worth for hydronic systems. However, the new SMACNA and ASHRAE duct fitting pressure loss coefficient tables use the more efficient "inches of water guage" (in. w.g.) for the pressure losses in lieu of the old "equivalent feet" when designing air systems. But the primary pressures concerning pumps and hydronic systems are the static heads and the system pressure losses.

Equations that can be used to solve various hydronic flow problems, may be found in Chapter V of this manual or in Section IX of the NEBB "Procedural Standards for Testing, Adjusting, Balancing of Environmental Systems."

E SUMMATION

Testing, adjusting and balancing work is not all definitions and theories, but it does require some amount of comprehension of simple equations, basic fundamentals, the principles and the methods. Future chapters will expound and expand on the "tip of the iceberg" presented in this chapter. Enjoy the learning experience for which it was intended.

NEBB

CHAPTER 2 TAB INSTRUMENTATION AND USE

A INTRODUCTION

Many different types of testing and measuring instruments are used in TAB work. These include instruments used for measuring pressures, temperatures, fluid flows, electrical circuits and rotational speeds. These and others are used by all the TAB technicians who must know how to use and respect them. Many of these instruments are expensive; and most are delicate, so they must be treated with the utmost care. They must be protected from dirt as well as shock and jarring movements. Some can be damaged by exceeding their rated capacity; and they should always be kept in their cases when not in use.

Some instrument require periodic calibration to insure their accuracy. NEBB has established criteria for instrument calibration and the frequency. If the TAB technician suspects inaccurate readings with any instruments, they should be checked. Often, a quick comparison can be made with another instrument known to be accurate. Otherwise, it must be sent to a qualified calibration laboratory for certification. Very seldom do any two instruments read exactly the same, even when just returned from a repair and calibration facility. There are however, certain degrees of accuracy of tolerance that they must stay within. To avoid minute differences on a particular job, it is best, if possible, to use the same instruments throughout the entire balancing process.

Many new electronic instruments are being introduced which offer increased ease of use and speed of operation, as well as improved accuracy. They probably will eventually replace many of the conventional instruments now in use. However, they should be approved by the National NEBB Technical Committee before being used on NEBB Certified projects.

Proper use of all instruments is mandatory if accurate results are to be obtained. Review the instrument manufacturer's instructions for its proper use and application to insure accurate results. These instruments are a necessary and important part of the TAB profession, but they are of little value unless the person using them can interpret the readings and accurately record the data. Therefore a well qualified TAB technician must be able to properly measure and test the HVAC equipment and systems with the instruments, correctly interpret the information, record the data, and pass to the NEBB Supervisor neat and accurate test report forms.

B AIR MEASURING INSTRUMENTS

1. Manometers

A *manometer* is a TAB instrument used to read very low pressures, such as those found in HVAC air duct systems. Most manometers read in *inches of water* (in.w.g.) or *inches of mercury* (in. Hg.). These units may be defined as the amount of pressure required to raise the level of the fluid one inch in a tube. You soon will find that many measurements in TAB work will be in *inches of water* (in.w.g.), and most others will be in *feet of water* (ft. w.g.) or *pounds per square inch* (psi).

a. U-TUBE MANOMETERS

The basic instrument for measuring air system pressures and partial vacuums is the *U-tube manometer.* In its simple form, it is a practical instrument to use and is inherently 100 percent accurate.

The *U-tube manometer* consists of a U-shaped tube about half filled with liquid. With both ends open as in Figure 2-1 (A), the liquid is at the same height in each leg as atmospheric pressure is imposed on the top of the liquid in both legs.

When a positive pressure is applied to one leg, as in Figure 2-1 (B), the liquid is forced down in that leg and up in the other. The difference in height *h* indicates the pressure. Note that in this case, the total pressure imposed on the left leg is the sum of atmospheric pressure plus gauge pressure, while atmospheric pressure alone is imposed on the right leg. The difference *h* therefore represents gauge pressure.

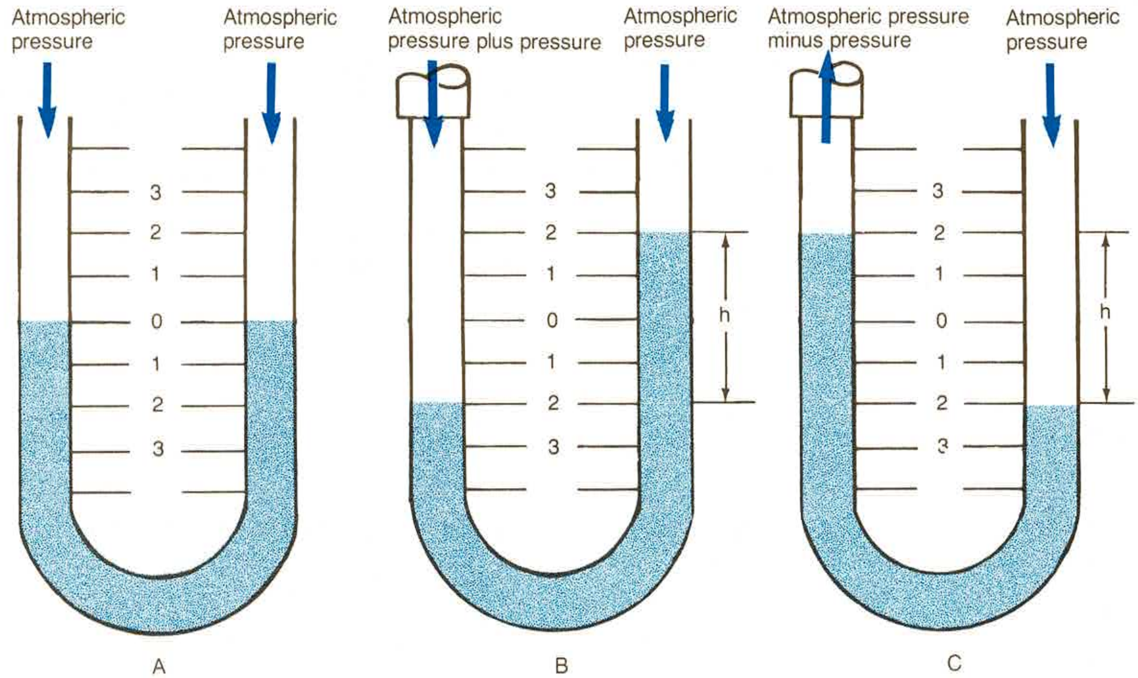


Figure 2-1 PRESSURE MEASUREMENTS WITH A U-TUBE MANOMETER

When a vacuum is applied to one leg as in Figure 2-1 (C), the liquid rises in that leg and falls in the other leg. The difference in height *h* indicates the amount of vacuum. Since, in Figure 2-1 (C), atmospheric pressure is pressing downward on the liquid in the right-hand leg, *h* represents the amount by which the measured pressure is less than atmospheric pressure.

In reading a U-tube manometer, note that a pressure or vacuum causes the liquid level in one leg to fall below the zero mark, while the liquid level in the other leg rises above the zero mark. The pressure *"h"* is the *sum* of the readings at these two levels. It would therefore appear that the reading could be said to be equal to twice the scale indication at either level. But this condition is rarely true in practice, as the liquid level is seldom precisely at the zero scale mark when the manometer is set up ready for use. Some manometers have movable scales which can be set to zero indication. However, for proper results, always take a reading at each of the two liquid levels and add them together.

As a primary measuring instrument, giving readings directly in inches of water, a manometer would be filled with plain water. Preferably, distilled water should be used to avoid mineral deposits within the manometer tube, as an accumulation of such deposits can interfere with the required liquid level meniscus and a clouded tube can make it hard to see the graduations clearly.

In many commercial instruments, the liquid used is oil rather than water. One advantage of oil over water is that oil is lighter than water for the same pressure difference. This expands the scale and makes for easier and more precise reading. For a given pressure, the difference in the height of the liquid levels in a manometer depends on the density of the liquid. Oil having a specific gravity of about 0.83, as compared to water, will rise a distance of about 1 divided by 0.83, or approximately 1.2 inches at a pressure of one inch of water.

Where a manometer is intended to be used with oil, it is essential that the oil has the same specific gravity for which the manometer scale was constructed, and this can be assured only by using oil obtained from the manufacturer of the manometer. Usually the oil is identified by a colored dye, commonly red. If there is any doubt as to whether the liquid for a specific manometer, having vertical legs or a vertical indicating tube, should be oil or water, it can be easily determined by measuring the scale—the distance covered by one inch on the scale should be exactly one inch for water, or about 1¼ inches for oil. Such a determination is not so easily made on an inclined manometer. Therefore, when necessary, observe the manufacturer's instructions either as marked on the instrument, or as in separate instructions.

Manometers may use mercury as well as water or oil. However, since mercury weighs 13.6 times as much as water, it will move only 0.07 of the distance that water would move at a given pressure. Mercury is therefore not suitable as the liquid for measuring the low pressures that occur in air systems, but it is used for measuring pressures in hydronic systems.

Because of a property of liquids called surface tension, the upper surface of the liquid column is curved rather than flat. The curved upper surface of the liquid column is called the *meniscus.* Water and oil have the ability to wet the surface of the manometer tubes and so form a meniscus shaped as in Figure 2-2 (A). Such a liquid column should be read by noting the scale division that is located at the bottom of the meniscus. On the other hand, mercury forms a meniscus curved upward as shown in Figure 2-2 (B), and should be read at the top of the curved surface.

In reading a manometer, care must be taken to avoid error due to parallax, which is the effect that occurs when the eye is not exactly perpendicular to the scale.

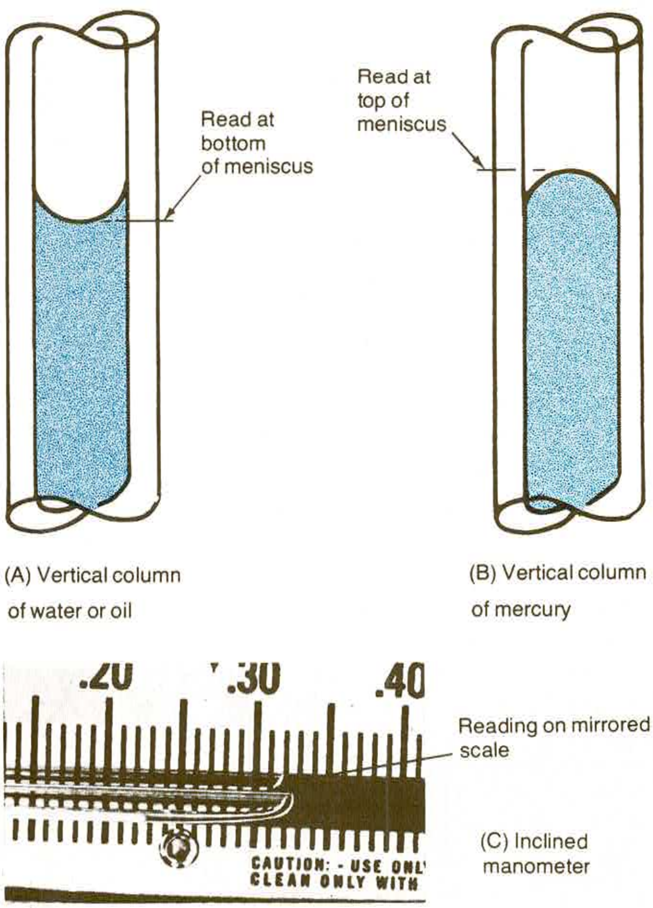


Figure 2-2 READING THE MENISCUS OF MANOMETERS

Some manometers have a mirrored scale to help eliminate the error. When the meniscus and its reflection are aligned, the line of sight will be perpendicular to the scale and the liquid column. Figure 2-2 (C) illustrates this effect; in this case, the reading is 0.32.

U-tube manometers normally are used for testing pressures above 1.0 in.w.g. They are available in many sizes up to 36 in.w.g., and a few are even larger. For higher pressures, mercury usually is used instead of water. Other types of manometers are used for readings under 1.0 in.w.g.

Table 2-1
TAB INSTRUMENTS REQUIRED FOR NEBB CERTIFICATION

Required for Air and/or Hydronic Certification*

Electronic Tachometer
12″ Mercury Thermometer, −40°F to + 120°F
12″ Mercury Thermometer, 0°F to + 220°F
Dial Thermometer, −40°F to + 120°F
Dial Thermometer, 0° to 220°F
Volt-Ammeter

Additional Requirements for Air Certification*

Inclined Manometer, 0″ to 1″
Combination Inclined and Vertical Manometer, 0″ to 5″
U-Tube Manometer, 18″
Pitot Tube, 18″
Pitot Tube, 36″
Deflecting Vane Anemometer (**Alnor Velometer preferably Model 6000BP) Range 100 to 3000 feet/min.

**Magnehelic or Hayes Draft Gauge 0 to ½″
**Magnehelic or Hayes Draft Gauge 0 to 1″
**Magnehelic or Hayes Draft Gauge o to 5″
Smoke Candles
Smoke Generator, Aspirating Type
Sling Psychrometer

Additional Requirements for Hydronic Certification*

Contact Pyrometer, Thermocouple Type
Calibrated Test Gauge, 0 to 30 psi
Calibrated Test Gauge, 0 to 60 psi
Calibrated Test Gauge, 0 to 200 psi
Calibrated Test Gauge, Compound, −30″ to 30 psi
Calibrated Test Gauge, Compound, −30″ to 60 psi
U-Tube Manometer, 36″ or Well Type Manometer, 18″

**Instrument specification and calibration requirements contained in Section II, NEBB Procedural Standards for Testing, Adjusting and Balancing of Environmental Systems, current edition.*

***Trade name used for descriptive purposes only, instruments of equal capacity are acceptable.*

The U-tube manometer does not require calibration. When kept clean and used with the proper fluid, it is inherently accurate. It is one of the NEBB required instruments for use by NEBB Certified TAB Contractors (see Table 2-1).

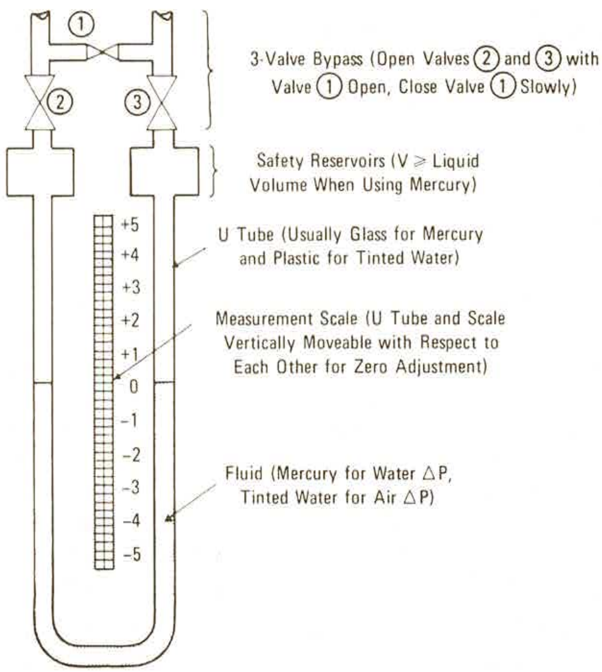


Figure 2-3 DIAGRAM OF A U-TUBE MANOMETER

b. INCLINED MANOMETERS

The inclined manometer (Figure 2-4) is a variation of a well-type manometer, in that the indicating leg or tube is placed in a sloping position rather than a vertical one. The purpose of this is to improve accuracy in reading the scale—for the same pressure, the distance along the inclined scale is considerably greater than on the vertical. Therefore, scale divisions on an inclined scale can be of greater length than on a vertical scale. Depending on the angle of incline, for a pressure of one inch w.g., the inclined scale might be 6 or 8 inches long. This makes it feasible to mark an inclined scale in smaller graduations than a vertical scale, and so increases the precision with which the inclined scale can be read, usually in a range of 0.1 to 1.0 in.w.g.

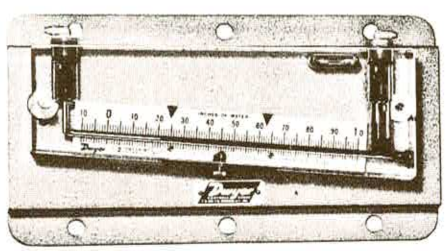


Figure 2-4 INCLINED MANOMETER

The actual length of scale graduations depends on the angle of the incline. It is therefore essential that the inclined scale be set at the exact angle for which it was designed. For this purpose, each inclined manometer is equipped with a level, and portable inclined manometers have a leveling screw. Therefore, before taking a reading on an inclined manometer, the instrument must be leveled. Also, it is important to avoid moving the manometer out of position when in use—it could easily be thrown out of level if moved about on a sloping or uneven surface. It is a good idea, after taking a reading, to check the instrument to see that it is still level.

c. INCLINED-VERTICAL MANOMETERS

The inclined-vertical manometer, pictured in Figure 2-5, is a well-type manometer that combines an inclined manometer with a vertical well-type instrument. The inclined portion affords precision in measuring low pressures, such as one in.w.g. and lower. The vertical portion gives the manometer the ability to measure higher pressures, such as up to 8 or 10 in.w.g. in a compact instrument.

The scale of some instruments have two sets of graduations—inches of water and velocity in feet per minute. The combination inclined-vertical manometer is particularly suitable for use with a Pitot tube to measure duct velocities, and so for added convenience, the scale can be read indirectly in velocity.

Since a portion of this manometer is actually an inclined manometer, it is essential that the inclined-

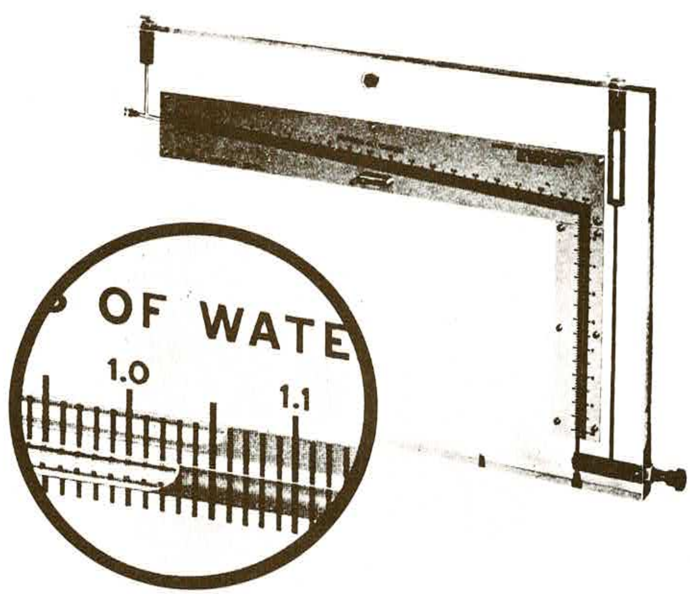


Figure 2-5 INCLINED-VERTICAL MANOMETER

vertical manometer be carefully leveled before taking readings. Inclined/vertical manometers have an essential built-in level device so that the readings on the inclined portion of the instrument can be accurate. A built in zero adjustment in the form of a threaded piston in the bottom well is provided to zero the fluid. As these instruments use a colored oil instead of water, the proper fluid must be used to obtain correct readings.

When this instrument is used with a pitot tube, it can measure the various pressure readings of HVAC duct systems and for pressure drops across components. When kept clean and used properly it is inherently accurate and requires no calibration.

When using the inclined portion of the instrument, the TAB technician should be cautious about "paralax" or the ability to properly "eyeball" the fluid level in relation to the scale. Some instruments have a mirror located behind the scale to aid the user. The manufacturer's instructions should be followed. This also is a NEBB required instrument for Certified NEBB TAB Contractors.

d. MICROMANOMETERS AND HOOK GAUGES

Micromanometers (Figure 2-6) and hook gauges are manometers made to give very precise readings, accuracy within plus or minor 0.001 inch being possible. These are delicate instruments, in general being more adapted to precise industrial and laboratory testing, rather than field testing. However, they have some application in commercial testing and balancing

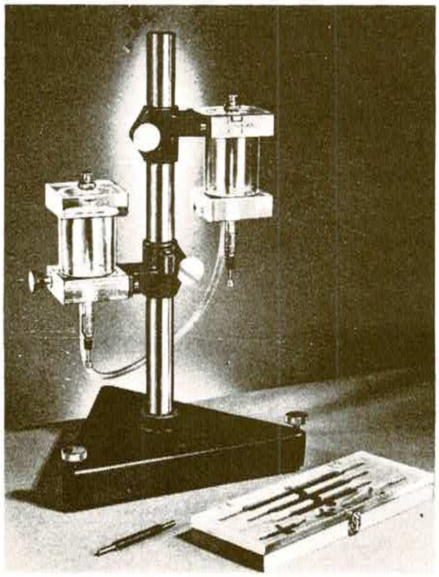


Figure 2-6 HOOK GAUGE

where it may be necessary to measure very low pressures as at exhaust hoods and air distributing ceilings. They are also useful for measuring air velocities below 600 feet per minute, where the corresponding velocity pressures are very low (approximately 0.025 in.w.g. and less). Using water as the liquid, these are primary measuring devices and can be used to calibrate other instruments.

Each of the two vials of this U-tube type manometer contains a pointer (often called a *hook*). The measuring principle is based on the fact that water has surface tension, and as the point of the hook is adjusted by the micrometer upward toward the surface of the water, surface tension causes the point to form a small dimple on the water before the pointer breaks through the surface. Formation of the dimple indicates contact of the point with the water surface, a reading at this setting being taken on the micrometer.

To use the micrometer, the vials are set at precisely the same level by using a gauge rod of precise length (or using a longer gauge rod, the vials can be set at a precise distance one above another for a greater pressure range such as up to 4 in.w.g.). The position of each hook is adjusted until it dimples the water surface, and its micrometer is then set to zero. When the pressure to be measured is imposed on one of the vials, the hooks are again adjusted to dimple the water surfaces. The pressure, as determined by the difference in height of the two water surfaces, is determined by reading the two micrometers and adding their readings together.

Although very precise instruments, there are disadvantages in using them. Both instruments must be very carefully leveled immediately before each reading, and they should be checked after being read to be sure they have not moved. Also, they are difficult to use if mounted on a surface that vibrates, or if there are pulsations in the pressure to be measured.

e. ELECTRONIC METERS

Electronic Meters are relatively new at this time. They contain no fluid as they are battery powered. Most have a digital display, and some offer readings in temperature or pressure plus velocity. Many specify that they are as accurate as a micromanometer and are able to read very low pressures. They will operate in any position and are light and small. As they all are battery powered, some can be re-charged. At this time they are still quite expensive but they offer great potential advantages of being fast and easy to use by TAB technicians. NEBB has set up an instrument accuracy test program with Tennessee Technological University to verify the manufacturers proclaimed accuracy.

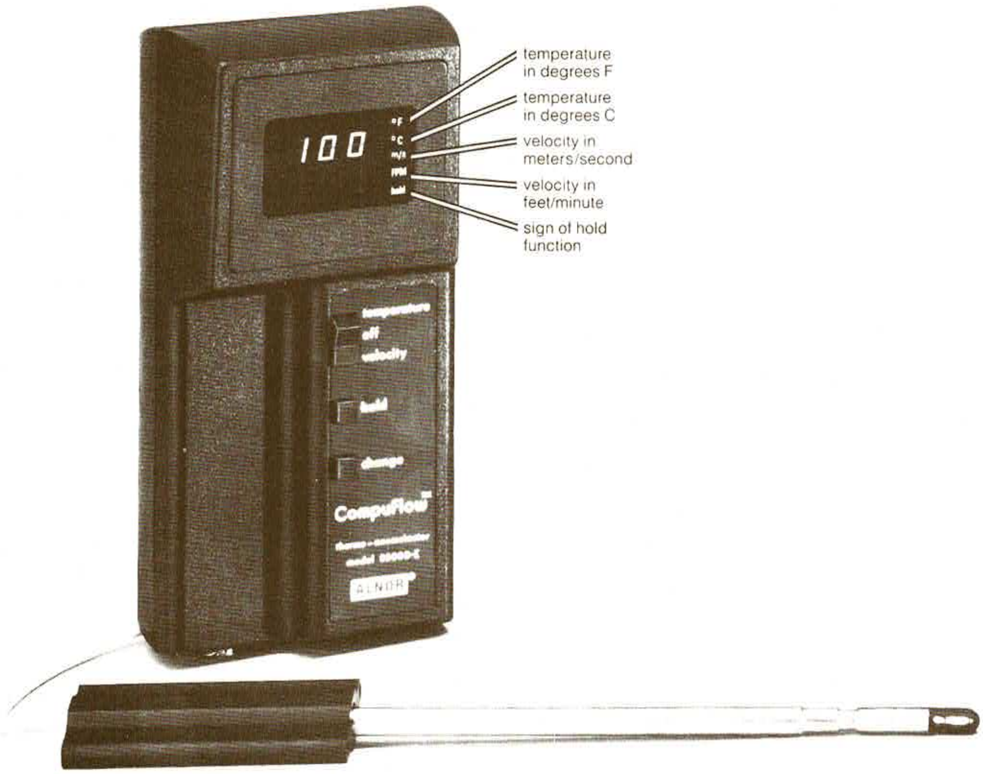


Figure 2-7 ELECTRONIC AIRFLOW INSTRUMENT

f. DRY-TYPE PRESSURE GAUGES

As noted above, manometers have certain disadvantages: they contain liquid that may be spilled, or which can be blown out by pressure beyond the range of the instrument; inclined manometers must be leveled. Dry type gauges overcome these objections.

Figure 2-8 shows a Magnehelic Gauge which is one type of dry type gauge. This gauge contains a diaphragm which readily moves with changes in the pressure imposed on it. However, its movement is restricted by the range spring which is calibrated to bend a definite amount when the diaphragm is subjected to a give pressure. Since it is a delicate type of instrument, having jewelled bearings, it should be handled carefully. Portable instruments for field use are available with carrying cases, and accessories normally used with the gauge can be stored in the case.

Advantages of dry-type gauges include:

(a) Absolutely level mounting is not necessary, although if the position of the gauge is changed, resetting of the zero adjustment may be required for proper gauge indication.

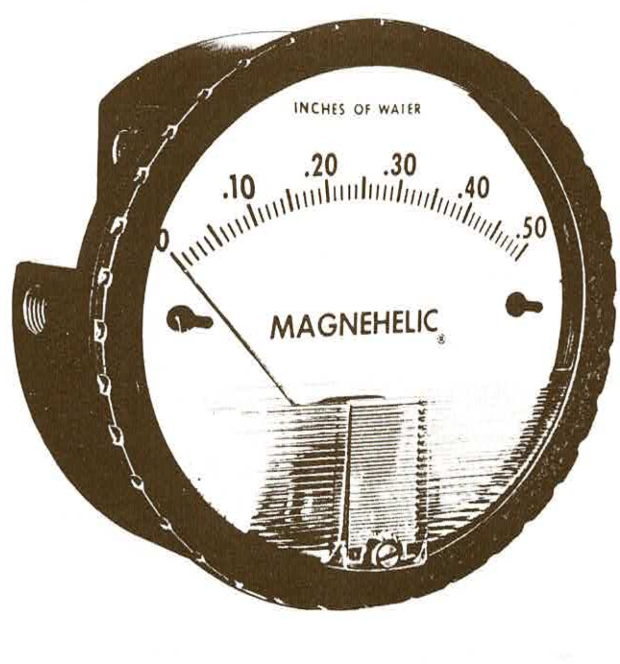


Figure 2-8 MAGNEHELIC GAUGE

(b) When all pressure taps are open to atmosphere, the gauge should read zero. If it does not, it should be so set by turning an appropriate screw on the face of the gauge.

(c) The gauge will measure differential pressure when two sensing elements are connected to it. If only one sensing element is connected to the gauge, the instrument will read gauge pressure, but in this case, the other pressure tap on the gauge must be open to atmosphere.

(d) The gauges are available in a variety of pressure ranges. Gauges with ranges of 0 to 0.5 in., 0 to 1 in., and 0 to 5 in. water pressure should give good coverage for general use.

(e) Calibration of the gauge should be checked periodically against a comparison gauge, using a micromanometer, a hook gauge, or an inclined manometer of known accuracy. To make the check, connect the gauge to the test gauge with tubing leading from a tee; connect the third leg of the tee to a pressure, comparing the gauge readings. A sufficiently skilled instrument mechanic can make any necessary calibration adjustments according to the manufacturer's instructions.

Magnehelic gauges normally are used in the vertical position; but the higher ranges (1.0 in.w.g. and up) can be used horizontally when they have been zeroed in that position. They are small, lightweight, inexpensive and relatively easy to use. They are accurate enough for most TAB work as long as they are not abused. They must be treated gently. Gauges with the three ranges mentioned above are NEBB required instruments.

g. PRESSURE SENSING DEVICES

In order for any of the gauges described above to be of use, they must be properly connected to the source of pressure that is to be measured. Normally, 3⁄16 inch rubber or plastic tubing is used to connect the instrument to a sensing element, the design of which depends upon the type of pressure to be measured—essentially whether it is desired to measure static pressure or total pressure (velocity pressure not being read directly but as the difference between total pressure and static pressure). This is illustrated in Figures 2-9 and 2-11.

In measuring static pressure, every effort must be made to eliminate the effect of airflow in the duct. This can best be done by arranging that holes through which static pressure is sensed, are at a 90-degree angle to actual airflow. Be sure that the holes are sharp and free of burrs, as they could tend to act as tiny air scoops, set up turbulence or direct a flow of air either toward or away from the sensing hole.

Figure 2-10 shows some types of elements used for sensing static pressure. Several of these devices are intended for permanent installation, such as related to a permanently mounted draft gauge to indicate

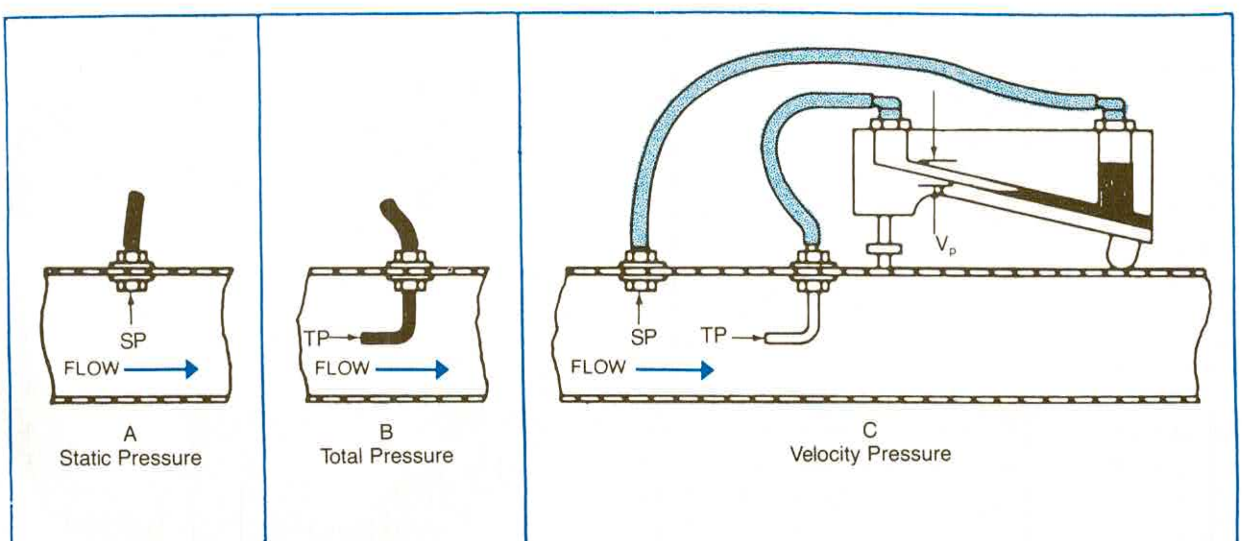


Figure 2-9 AIRFLOW PRESSURE MEASUREMENTS

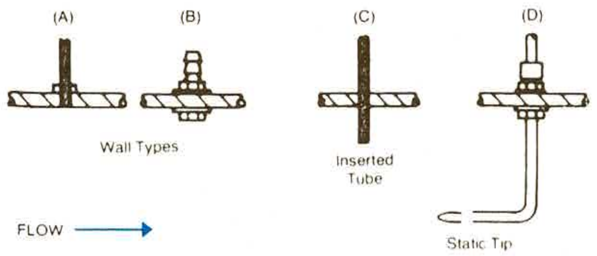


Figure 2-10 STATIC PRESSURE SENSING DEVICES

pressure drop across a filter, but they can serve to illustrate the principles involved.

Figure 2-10 (A) shows a simple through-wall tap. The hole through the duct wall should be sharp and free of burrs on the inner face of the duct, and the axis of the hole must be perpendicular to the direction of flow. A tubing connection must be provided, soldered or otherwise fastened to the duct. Figure 2-10 (B) shows a somewhat similar tap, which is a manufactured item and can be readily installed in a hole drilled in the duct. The elements shown in Figures 2-10 (A) and (B) are suitable static pressure sensing devices where airflow is relatively smooth and without turbulence, typically where air velocities are below 1,500 feet per minute and flow is not disturbed as by turns or other duct fittings. If turbulence exists, impingement, aspiration or unequal distribution of moving air at the sensing opening can significantly reduce the accuracy of readings.

Figure 2-10 (C) shows a simple tube inserted through the duct wall. It has limitations similar to those of the elements shown in Figures 2-10 (A) and 2-10 (B). However, it could be even more critical as to the end of the tube being perfectly perpendicular to the direction of airflow. This device is typical of improper practice sometimes used in field testing. There is often the tendency to take the end of the rubber tube leading from the manometer and push it through a hole in the duct, or to connect the tube to a piece of metal tubing, which is then inserted in a hole in the duct or worked through the flexible connections at a fan. This is not a recommended practice.

Figure 2-10 (D) shows a static pressure tip that is a desirable sensing device, especially for sensing pressure drop across equipment such as air filters, or heating or cooling coils. The probability of air turbulence at such locations requires that the pressure sensing openings be located away from the duct walls to minimize impingement and aspiration. The end portion of the tip contains a number of small holes drilled radially and at a 90-degree angle to the axis of the tube. When the tip is installed so the part containing the holes is parallel to the direction of airflow, the axis of the holes will be perpendicular to the airflow and so able to properly sense static pressure.

2. Pitot Tubes and Their Use

a. THE PITOT TUBE

Figure 2-9 (C) shows an arrangement for measuring velocity pressure using separate elements for measuring static pressure and total pressure. The static pressure sensor should follow the principles discussed above for devices of this type. The element which senses total pressure may be referred to as an *impact tube*. It is an open tube, faced directly into the airstream so as to receive the effects of both

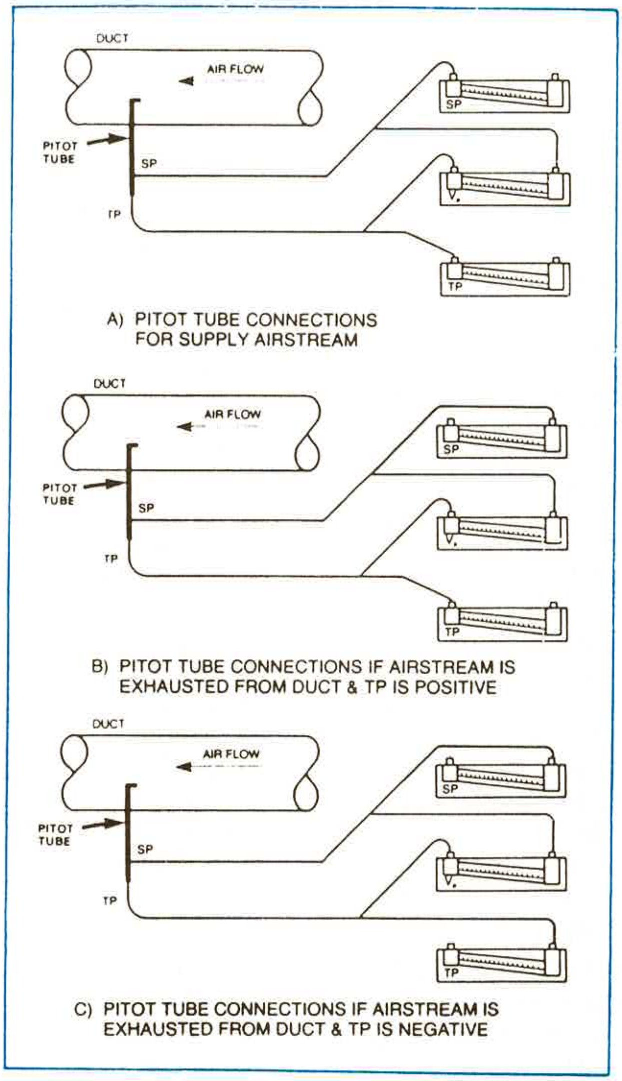
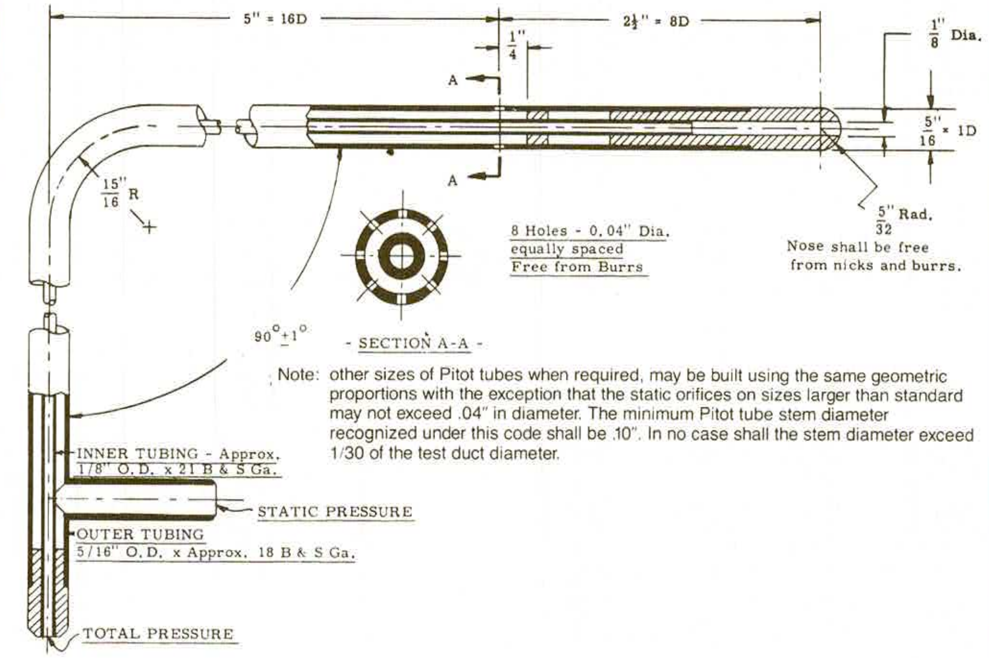


Figure 2-11 PITOT TUBE CONNECTIONS



Note: other sizes of Pitot tubes when required, may be built using the same geometric proportions with the exception that the static orifices on sizes larger than standard may not exceed .04" in diameter. The minimum Pitot tube stem diameter recognized under this code shall be .10". In no case shall the stem diameter exceed 1/30 of the test duct diameter.

Figure 2-12 PITOT TUBE

static pressure and velocity. Essentially, the Pitot tube combines an impact tube with an efficient static pressure sensor of the type shown in Figure 2-10 (D).

As indicated in Figure 2-12, a Pitot tube contains an impact tube on which total pressure is imposed, fastened concentrically inside a second tube of slightly larger diameter which senses static pressure through radially located sensing holes near the tip. The air space between the inner and outer tubes serves to transfer pressure from the sensing holes to the static pressure connection at the opposite end of the Pitot tube and then, through connecting tubing, to the low or negative pressure side of a manometer. The impact tube runs to the total pressure connection of the Pitot tube, and when connected to the high pressure side of the manometer, velocity pressure is indicated directly on the instrument; see Figure 2-11 or 2-9 (C). The Pitot tube is an important measuring device and for accuracy, it must be carefully made. They are available in various lengths, normally from 8 inches to 60 inches.

Smaller "pocket-size" Pitot tubes can be used in ducts smaller than 8-inch diameter. To insure accurate sensing of total pressure, any size Pitot tube tip must be pointed directly into, or parallel with, the airstream. The Pitot tube tip is parallel with the static pressure outlet tube, and so the latter can be used as a pointer to align the tip properly. When the Pitot tube is correctly aligned, the full effect of the air velocity will be obtained, and the pressure indication will be maximum.

Pitot tubes have smooth, well-rounded tips to minimize turbulence. They should be treated with care, so that they will not become bent or mashed; the noses should be protected so they will not become dented or otherwise roughened, and the small static pressure sensing holes should be kept open and clean.

As the Pitot tube can be used to measure any one of three basic pressures (total pressure, static pressure and velocity pressure) when used with the proper hose hook ups between the Pitot tube and the manometer (as shown in the Figure 2-11), it is accurate and reliable and is the preferred method of measuring air velocities and pressures in the field. The accuracy of the readings of the Pitot tube in a duct are dependent on the uniformity of the airflow in a cross section of the duct. Pitot tube traverses should be taken in a length of straight duct, preferably 6 to 10 duct diameters downstream of any elbows, branches, transitions or other obstructions to uniform airflow and at least several duct diameters upstream of another turn, branch, etc.

b. USE OF THE PITOT TUBE

The primary use of the Pitot tube by the TAB technician will be measuring velocities in ducts to determine the duct airflow (cfm). The procedures for this are listed below:

(1) Measure the size of the duct. This means the free inside dimensions of the duct where the air is passing through. If the duct has a fibrous glass lining, the dimensions inside the insulation are what you want to use. From these dimensions, determine the cross-sectional area by multiplying the height in inches times the width in inches divided by 144. This will give you the duct cross-sectional area (A) in square feet (sq. ft.).

(2) To perform a Pitot tube traverse of a duct, the readings must be taken in the duct at equal intervals. A rectangular duct, such as the 48″ × 36″ duct in Figure 2-13, should be divided into equal areas and 48 readings taken. A minimum of 16 readings should be taken in small ducts and the readings should be no more than 6 inches apart for maximum accuracy. The NEBB "Rectangular Duct Traverse Reports" should be prepared in advance for each duct traverse to be made. Readings will be taken at the center of each area.

Equation 2-1

$$\text{Area (sq. ft.)} = \frac{\text{Height (in.)} \times \text{Width (in.)}}{144}$$

or

$$A = \frac{H \times W}{144}$$

Example 2A

If an unlined duct is 16 inches wide and 18 inches high, what is the area in square feet?

Solution

$$A = \frac{H \times W}{144} = \frac{18 \times 16}{144} = 2.0 \text{ sq. ft.}$$

When measuring internally insulated ducts it is wise to stop occasionally and blow out any glass fibers that collect in the Pitot tube to maintain as accurate a set of readings as possible. Passing the tube through the insulation does dislodge small amounts of fiberglass and will eventually clog the tube. Remove the plastic tube connections to the manometer and blow into the bottom of the inside tube to discharge any fiberglass fibers from the head end of the tube.

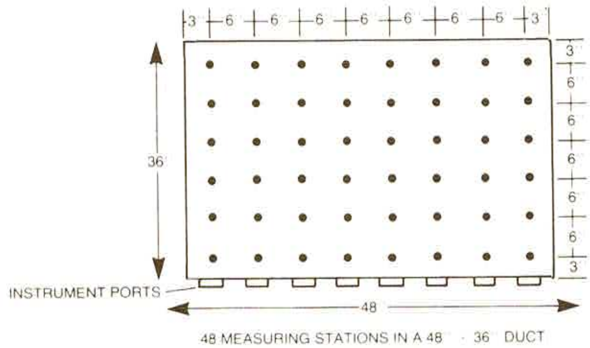


Figure 2-13 RECTANGULAR DUCT TRAVERSE

In round ducts, readings should be taken at the centers of equal concentric areas. Preferably twenty readings should be taken, ten along each of two diameters. The divisions between each reading are to be of equal area. This means that the dimensions themselves will not be the same (See Figure 2-14). Less readings may be taken with smaller ducts. A convenient chart (Table 2-2) can be used for round ducts up to 36 inches in diameter, which gives the dimensions in inches from the pipe center for the test locations. For larger ducts, Table 2-3 has the necessary constants that are multiplied by the diameter of the duct being traversed. Once these dimensions have been determined, you can layout the NEBB "Round Duct Traverse Reports" for each set of traverses being made.

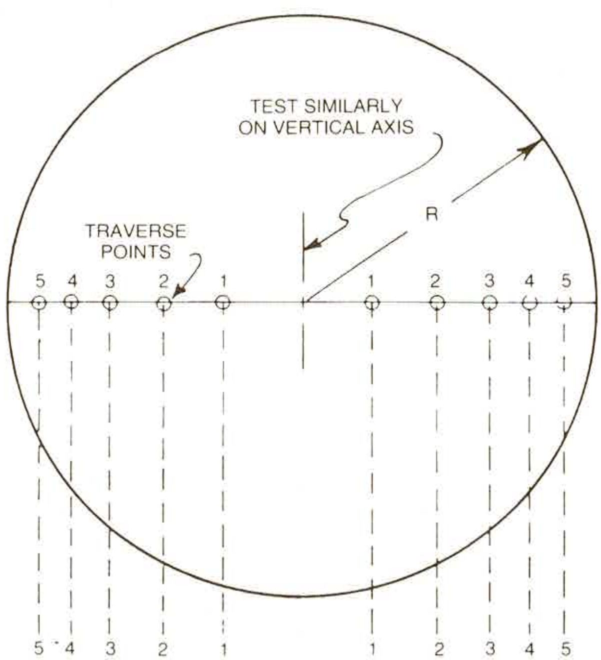


Figure 2-14 ROUND DUCT TRAVERSE

Table 2-2 ROUND DUCT TRAVERSE

Pipe diam.	Readings in one diam.	Distances of Pitot Tube Tip From Pipe Center				
		Point 1	Point 2	Point 3	Point 4	Point 5
3 in.	6	.612″	1.061″	1.369″		
4 in.	6	.812″	1.414″	1.826″		
5 in.	6	1.021″	1.768″	2.282″		
6 in.	6	1.225″	2.121″	2.738″		
7 in.	6	1.429″	2.475″	3.195″		
8 in.	6	1.633″	2.828″	3.651″		
9 in.	6	1.837″	3.182″	4.108″		
10 in.	8	1.768″	3.062″	3.950″	4.677″	
12 in.	8	2.122″	3.674″	4.740″	5.612″	
14 in.	10	2.214″	3.834″	4.950″	5.857″	6.641″
16 in.	10	2.530″	4.382″	5.657″	6.693″	7.589″
18 in.	10	2.846″	4.929″	6.364″	7.530″	8.538″
20 in.	10	3.162″	5.477″	7.077″	8.367″	9.487″
22 in.	10	3.479″	6.025″	7.778″	9.203″	10.435″
24 in.	10	3.795″	6.573″	8.485″	10.040″	11.384″
26 in.	10	4.111″	7.120″	9.192″	10.877″	12.333″
28 in.	10	4.427″	7.668″	9.900″	11.713″	13.282″
30 in.	10	4.743″	8.216″	10.607″	12.550″	14.230″
32 in.	10	5.060″	8.764″	11.314″	13.387″	15.179″
34 in.	10	5.376″	9.311″	12.021″	14.233″	16.128″
36 in.	10	5.692″	9.859″	12.728″	15.060″	17.176″

For distances of traverse points from pipe diameters other than those given in Table No. 2-2, use constants in Table 2-3.

Example 2B

A ten point traverse is made in both a horizontal and vertical plan across a duct.

A 20″ diameter duct requires 20 readings (see Table 2-2).

Solution

Reading points in Figure 2-15 can be calculated from Table 2-3 as follows:

#1 = 20 × .1581 = 3.16″ from the center
#2 = 20 × .2738 = 5.48″ from the center
#3 = 20 × .3535 = 7.08″ from the center
#4 = 20 × .4183 = 8.37″ from the center
#5 = 20 × .4743 = 9.49″ from the center

Note that these dimensions check with those in Table 2-2, and that the readings are numbered from the center of the duct. Reading point #1 is 3⅛ inches from the center of the duct. All of the #5 readings are ½ inch from the outside of the duct or 9½ inches from the center. These points will divide the round duct into equal areas and assure an accurate velocity pressure profile at four different points in each quadrant of the duct.

(3) Once the Pitot tube traverse hole dimensions have been determined, the TAB technician can mark off the Pitot tube. Common practice is to use tape. Electrical plastic tape will work

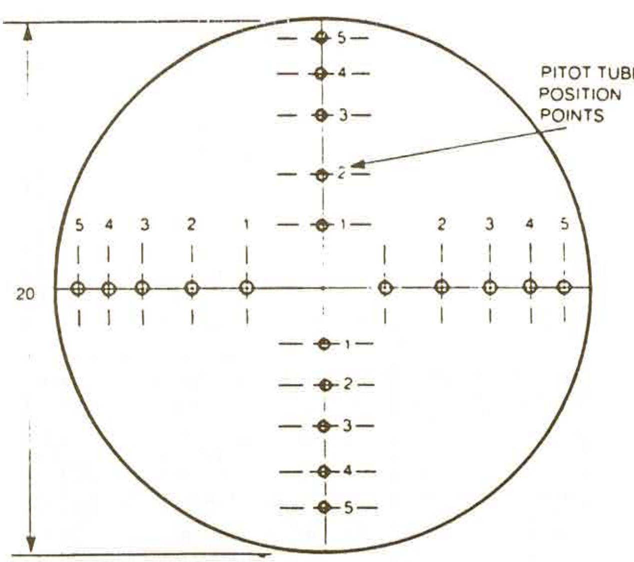


Figure 2-15 EXAMPLE TRAVERSE

Table 2-3 ROUND DUCT TRAVERSE CONSTANTS

Readings in one Diameter	Constants To Be Multiplied By Pipe Diameter For Distances of Pitot Tube Tip From Pipe Center				
	Point 1	Point 2	Point 3	Point 4	Point 5
6	.2041	.3535	.4564		
8	.1768	.3062	.3953	.4677	
10	.1581	.2738	.3535	.4183	.4743

well, and can usually be re-used several times.

(4) Using the dimensions determined above, mark off the locations on the duct where you will need holes for the Pitot tube. The holes should be located in a section of straight duct from 6 to 10 duct diameters long. They should be nearer to the downstream end of a straight section of duct. Quite often, this much straight duct is not available. Therefore, you will have to use the best or longest straight section available, possibly with a sacrifice in accuracy. **Never traverse in an elbow, offset or transition.** Allow room to swing the Pitot tube while inserting and removing it from the duct. Many times you may need to use Pitot tubes of different lengths in the same hole due to a lack of room beside the duct to move the tube in and out. Do not locate holes in the bottom of ducts that carry moisture laden air; instead, put these holes in the side of the duct where possible.

(5) Drill holes in the duct slightly larger than the Pitot tube diameter, usually 3⁄8″ diameter. Where the duct is insulated make provision for neat repair of the insulation.

(6) The holes will have to be capped when the testing is complete. Snap in plastic and rubber caps are available. Some specifications will call for metal test ports with a screw on cap. These are available in various lengths for use with insulated ducts.

(7) Set up the manometer in a convenient location. After connecting the tubing from the manometer to the Pitot tube, be sure to level and zero the manometer. Use a good grade of tubing that is thick enough so that it will not kink, but thin enough to be flexible. Clear plastic tubing that has a 3⁄16 inch inside diameter (usually 5⁄16 inch outside diameter) works well.

(8) Proceed to take velocity pressure readings at each location and record the readings on the NEBB Test Report Form sheets in the appropriate places. It is important that the Pitot tube be held straight and directly into the air stream. Notice that the static pressure port faces the same directions as the Pitot tube and can be used as a guide. Be sure to hold the Pitot tube at each position long enough to let the manometer fluid settle into position. If the fluid will not stay still while the Pitot tube is at any one location, air turbulence is indicated. Record the average reading, midway between the points. Turbulence should be noted on the test report form because turbulent readings will not be accurate. It is always advisable to take a static pressure reading at the Pitot tube duct opening. Some specifications require it, and it can be very useful for future adjustments or troubleshooting.

(9) Once the readings are recorded they will have to be converted to duct velocities before they can be totalled and averaged. When adding the readings, the values must be velocities in feet per minute (fpm). **Velocity pressures cannot be totalled and averaged.** Unless every reading in the traverse was identical, a very unlikely condition, adding and averaging the velocity pressures will result in erroneous results. Since many inclined manometers have scales that also read velocities in feet per minute (fpm), you can record these figures instead of the velocity pressures. These figures can be added and averaged; otherwise it will be necessary to convert the velocity pressures to fpm.

(10) Pitot tubes of 18 inch and 36 inch sizes are NEBB required instruments for NEBB Certified TAB Contractors.

c. VELOCITIES/VELOCITY PRESSURES

To review velocity pressure readings are taken at equal intervals over a cross section of the duct. Good practice dictates *not less than* 16 readings in any duct and in larger ducts readings should be taken on not less than 6″ centers. The velocity pressures are then changed to velocity values, added together, and divided by the total number of readings to get the average velocity. *Do not average the velocity pressure readings.*

It is not unusual to make a negative pressure reading in ducts with considerable turbulence. The negative readings are added in at zero value but are counted in the number of readings to obtain the average velocity. Assume a duct with 16 positive pressure readings and 4 negative pressure readings. The 16 positive pressure readings would be added together and averaged by all 20 readings (16 positive plus 4 zero readings).

To convert velocity pressure to velocity or velocity to velocity pressure, Table 2-4 or Table 12-4 (in the back) can be used. With an electronic calculator in hand, it often is faster to calculate the values using Equations 2-2 and 2-3 for standard air (near sea level conditions).

Equation 2-2

$$V = 4005\sqrt{V_p}$$

Equation 2-3

$$V_p = \left(\frac{V}{4005}\right)^2$$

Equation 2-4

$$V = 1096\sqrt{\frac{V_p}{d}}$$

Where:

V = Velocity (fpm)

V_p = Velocity Pressure (in.w.g.)

d = Density (lb./cu.ft.)

Note: Standard air density = 0.075 lb./cu.ft.

Equation 2-4 is used in areas of HVAC work where the air density is a factor, such as high temperature air found in heat recovery situations or locations at higher altitudes such as Denver at 5000 feet.

Table 2-4 VELOCITY PRESSURES VS. VELOCITIES

Velocity Pressure, In. w.g.	Velocity, Fpm	Velocity Pressure, In. w.g.	Velocity, Fpm	Velocity Pressure, In. w.g.	Velocity, Fpm	Velocity Pressure, In. w.g.	Velocity, Fpm	Velocity Pressure, In. w.g.	Velocity, Fpm
0.01	400	0.29	2150	0.58	3050	1.28	4530	2.40	6200
.02	565	.30	2190	.60	3100	1.32	4600	2.44	6260
.03	695	.31	2230	.62	3150	1.36	4670	2.48	6310
.04	800	.32	2260	.64	3200	1.40	4730	2.52	6360
.05	895	.33	2300	.66	3250	1.44	4800	2.56	6410
0.06	980	0.34	2330	0.68	3300	1.48	4870	2.60	6460
.07	1060	.35	2370	.70	3350	1.52	4930	2.64	6510
.08	1130	.36	2400	.72	3390	1.56	5000	2.68	6560
.09	1200	.37	2440	.74	3440	1.60	5060	2.72	6610
.10	1270	.38	2470	.76	3490	1.64	5120	2.76	6650
0.11	1330	0.39	2500	0.78	3530	1.68	5190	2.80	6700
.12	1390	.40	2530	.80	3580	1.72	5250	2.84	6750
.13	1440	.41	2560	.82	3620	1.76	5310	2.88	6800
.14	1500	.42	2590	.84	3670	1.80	5370	2.92	6840
.15	1550	.43	2620	.86	3710	1.84	5430	2.96	6890
0.16	1600	0.44	2650	0.88	3750	1.88	5490	3.00	6940
.17	1650	.45	2680	.90	3790	1.92	5550	3.04	6980
.18	1700	.46	2710	.92	3840	1.96	5600	3.08	7030
.19	1740	.47	2740	.94	3880	2.00	5660	3.12	7070
.20	1790	.48	2770	.96	3920	2.04	5710	3.16	7120
0.21	1830	0.49	2800	0.98	3960	2.08	5770	3.20	7160
.22	1880	.50	2830	1.00	4000	2.12	5830	3.24	7210
.23	1920	.51	2860	1.04	4080	2.16	5880	3.28	7250
.24	1960	.52	2880	1.08	4160	2.20	5940	3.32	7300
.25	2000	.53	2910	1.12	4230	2.24	5990	3.36	7340
0.26	2040	0.54	2940	1.16	4310	2.28	6040	3.40	7380
.27	2080	.55	2970	1.20	4380	2.32	6100	3.44	7430
.28	2120	.56	2990	1.24	4460	2.36	6150	3.48	7470

Example 2C

The velocity pressure of a 48″ × 24″ duct is 0.16 in.w.g. Find the velocity.

Solution

Using Equation 2-2:

$$V = 4005\sqrt{V_p}$$

$$V = 4005\sqrt{0.16\text{ in.w.g.}} = 1602\text{ fpm}$$

Example 2D

If the system in Example 2C was at an altitude where the air density was 0.07 lb./cu.ft., what would be the system air velocity?

Solution

$$V = 1096\sqrt{\frac{V_p}{d}}$$

$$V = 1096\sqrt{\frac{0.16\text{ in.w.g.}}{0.07\text{ lb./cu.ft.}}} = 1657.0\text{ fpm}$$

This example shows that the same velocity pressure reading from a system containing air of a lesser density will result in a higher velocity.

Density corrections can be made from Table 2-5 in this chapter or from Tables 12-3 and 13-1 in Chapters XII and XIII. TAB work up to 2000 feet altitude and between 30°F and 120°F normally is not corrected for density.

Table 2-5 CORRECTION FACTORS FOR AIR DENSITY (To be Applied to Velocities from Table 2-4)

Density	Factor	Density	Factor	Density	Factor
0.010	2.7378	0.040	1.3689	0.070	1.0348
0.011	2.6104	0.041	1.3521	0.071	1.0275
0.012	2.4993	0.042	1.3359	0.072	1.0203
0.013	2.4012	0.043	1.3203	0.073	1.0133
0.014	2.3139	0.044	1.3052	0.074	1.0064
0.015	2.2354	0.045	1.2906	0.075	0.9997
0.016	2.1644	0.046	1.2765	0.076	0.9931
0.017	2.0998	0.047	1.2629	0.077	0.9866
0.018	2.0407	0.048	1.2496	0.078	0.9803
0.019	1.9862	0.049	1.2368	0.079	0.9741
0.020	1.9359	0.050	1.2244	0.080	0.9680
0.021	1.8893	0.051	1.2123	0.081	0.9620
0.022	1.8458	0.052	1.2006	0.082	0.9561
0.023	1.8053	0.053	1.1892	0.083	0.9503
0.024	1.7673	0.054	1.1782	0.084	0.9446
0.025	1.7316	0.055	1.1674	0.085	0.9391
0.026	1.6979	0.056	1.1569	0.086	0.9336
0.027	1.6662	0.057	1.1467	0.087	0.9282
0.028	1.6362	0.058	1.1368	0.088	0.9229
0.029	1.6077	0.059	1.1271	0.089	0.9177
0.030	1.5807	0.060	1.1177	0.090	0.9126
0.031	1.5550	0.061	1.1085	0.091	0.9076
0.032	1.5305	0.062	1.0992	0.092	0.9026
0.033	1.5071	0.063	1.0908	0.093	0.8978
0.034	1.4848	0.064	1.0822	0.094	0.8930
0.035	1.4634	0.065	1.0739	0.095	0.8883
0.036	1.4430	0.066	1.0657	0.096	0.8836
0.037	1.4233	0.067	1.0577	0.097	0.8791
0.038	1.4045	0.068	1.0499	0.098	0.8746
0.039	1.3864	0.069	1.0423	0.099	0.8701
0.040	1.3689	0.070	1.0348	0.100	0.8658

The average velocity obtained then must be inserted in Equation 2-5 to obtain the system airflow.

Equation 2-5

$$Q = A \times V$$

Where:

Q = Airflow (cfm)

A = Area of duct cross-section (sq. ft.) [see Equation 2-1]

V = Velocity average (fpm)

Example 2E

Find the airflow of the 48″ × 24″ duct in Example 2C.

Solution

Using Equation 2-5:

$$Q = A \times V = \frac{48 \times 24}{144} \times 1602 = 12{,}816\text{ cfm}$$

Example 2F

The following velocity pressures were obtained from a 10 inch diameter duct. Calculate the duct airflow. Velocity Pressures from duct traverse: 0.46, 0.50, 0.52, 0.50, 0.49, 0.45, 0.51, 0.51, 0.55, and 0.60.

Solution

Obtain the following velocities from Table 2-4:

V_p (in.w.g.)	fpm
0.46	2716
0.50	2832
0.52	2888
0.50	2832
0.49	2804
0.45	2687
0.51	2860
0.51	2860
0.55	2970
0.60	3102
Total - 10 Items in column	28,551

$$V_{ave} = \frac{28{,}551}{10} = 2855\text{ fpm}$$

$$Q = A \times V = \frac{\pi R_2}{144} \times V$$

$$Q = \frac{\pi(10)^2}{144} \times 2855$$

$$Q = 6229\text{ cfm}$$

3. Anemometers

Anemometers are used to read air velocities directly. They are used to measure airflow at diffusers, registers, hoods, filter banks, coils or in open space. Some types of anemometers must be timed in their use so a stop watch will be necessary. Most measurements are taken to ultimately determine the airflow. Since anemometers read in terms of velocity, it will be necessary to convert to cfm using Equation 2-5 ($Q = A \times V$).

Most diffusers and grille manufacturers have developed *free area factors* ("Ak or K") for use with the various instruments. Each outlet manufacturer's "K" factors will be different: and also, for different instruments the "K" factor also will be different.

When planning to take airflow measurements at an outlet, first obtain the outlet manufacturers data. Certain instruments will be recommended along with the correct "K" factor for each type and size of outlet. The procedure for measuring velocities usually includes a particular way to position the anemometer which must be followed if accurate results are to be obtained.

a. ROTATING VANE ANEMOMETER

The rotating vane anemometer (Figure 2-16) consists of a propeller in a housing connected to a dial that is calibrated in feet. It must be timed when used, preferably for one minute with a stop watch which will result in a reading of feet per minute (fpm). The most common sizes are 3 inch and 4 inch diameter models. Due to the construction characteristics of each type and model, a correction curve or chart is furnished and *must be used.* Most units correct upwards at lower velocities and correct downwards at higher velocities. These instruments are not usually satisfactory below 200 fpm, although some of the newer ones claim accuracy as low as 30 fpm. Many TAB technicians make correction charts from the curves which are easier and quicker to use in the field.

There is some controversy over whether to move the rotating vane anemometer across the face of a grille or to take several fixed readings and add them, similar to a traverse. The difference doesn't appear to be large but if there is a large difference in velocity between one end of a grille and the other, the later method should be used. The instrument mainly is used for measuring velocities of registers, grilles, and hoods; and for approximate or comparative velocity readings only of filters, coils and damper openings.

When a 14" × 6" supply register is to be measured to determine the airflow in cfm with a rotating vane anemometer, the following procedure is used:

(1) Measure and identify the register. Obtain the manufacturers "Ak" or "K" factor and recommended test instrument and reading procedure.

Equation 2-6

$$Q = V_c \times K$$

Where:

Q = airflow (cfm)

V_c = Velocity corrected (fpm)

K = Mfr's. factor

(2) When the manufacturer recommends a rotating vane anemometer, hold the anemometer against the register with the dial away from the airflow. Move the anemometer back and forth slowly covering all of the area of the register. Then criss-cross the register in an "X" pattern, timing it so you just finish at the end of one full minute. Stop the anemometer and the watch at precisely one minute and read the dial for the velocity in fpm. Correct the dial reading from the curve and record the corrected reading.

(3) Using Equation 2-6 ($Q = V_c \times K$), determine the cfm.

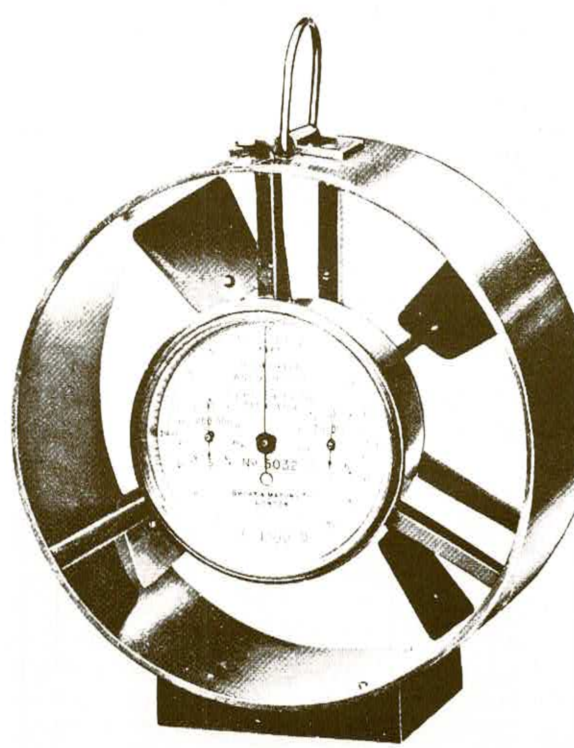


Figure 2-16 ROTATING VANE ANEMOMETER

Example 2G

The Manufacturer has published an "Ak" factor of 0.43 for the grille being measured. The corrected velocity reading for the anemometer was 987 fpm. Find the airflow.

Solution

$Q = V_c \times K = 987 \times 0.43 = 424$ cfm

Readings of 30 seconds may be taken on smaller registers by multiplying the answer by two. On very large grilles and registers, you may need to divide it into sections and read and calculate the cfm for each section individually and add them together for the total cfm. Using this instrument will require some practice to get familiar with the patterns of movement in relation to timing.

One of the main advantages of rotating vane anemometers is that they read and average a much larger area than deflecting vane or hot wire anemometers which take only spot readings. This larger area average is desirable on registers and grilles where the dampers have been throttled. The rotating vane anemometer does require calibration every 6 months.

b. DEFLECTING VANE ANEMOMETER

The deflecting vane anemometer (Figure 2-17) is recommended by more outlet manufacturers than any other. "K" factors are available from almost all manufacturers for most diffusers, registers, and grilles. The instrument reads directly in terms of velocity, and it is available with several attachments and in different ranges. The most popular model reads in two ranges (0-1250 fpm and 0-2500 fpm) with a handy range switch on the probe (see Figure 2-18). A separate "Low-flow" probe attaches to the instrument directly and reads 0-300 fpm. The main disadvantage of the "Low-Flow" probe is that it can't be used with a hose and must be left on the instrument.

The probe for diffusers will be used most often, and it is always used with two hoses. The diffuser manufacturers provide a recommended procedure for positioning the tip on a diffuser, what ring to read, how many readings to take, and the "K" factor to use (see Figure 2-19).

Example 2H

A diffuser manufacturer states that four readings are to be taken equally spaced around the outer ring of diffuser. The "K" factor for an 8 inch diameter neck diffuser with the cones up that provides a vertical air pattern is 0.20. The readings are: 920 fpm, 1160 fpm, 1200 fpm, and 1045 fpm. Find the diffuser airflow.

Solution

Divide the total of the readings by the number of readings taken to obtain the average velocity reading. Then multiply the average velocity in fpm times the "K" factor to obtain the cfm.

Reading	Velocity
1	920 fpm
2	1160 fpm
3	1200 fpm
4	1045 fpm

Total – 4325 fpm / 4 Readings = 1081 fpm

$Q = V_c \times K$

$Q = 1081 \text{ fpm} \times 0.20 = 216$ cfm

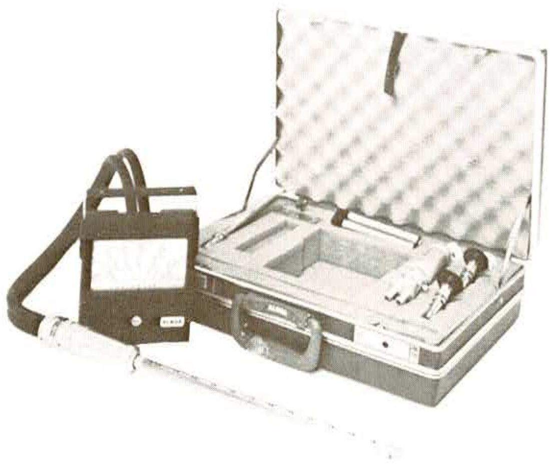
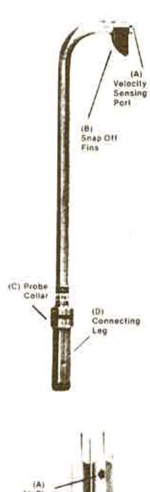


Figure 2-17 VELOMETER® SET



A) DIFFUSER PROBE

The Diffuser Probe is designed to measure the velocity at diffusers, registers and grilles. The *volume* of air being supplied or exhausted can be determined by multiplying the measured velocity times an air flow factor.

The velocity sensing port **(A)** senses the velocity at the diffuser, register, or grille.

The snap-off fins **(B)** allow you to accurately position the probe vertically, horizontally or radially.

The probe collar **(C)** acts as a stop when connecting the probe to the Range Selector, and the O-ring acts as a seal.

The connecting leg **(D)** is mounted into the Range Selector.

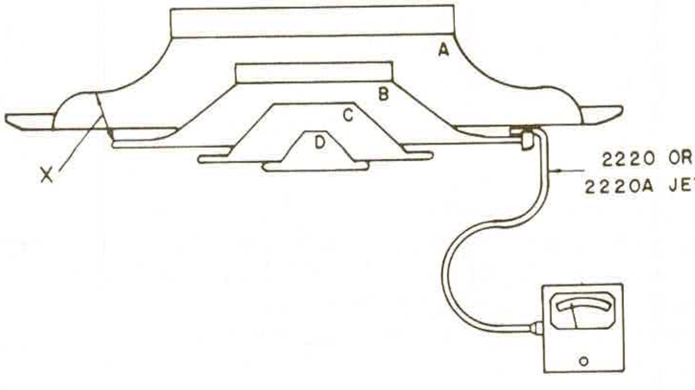
B) LO-FLOW PROBE

The Lo-Flow Probe is designed for measuring velocities below 300 feet per minute in open spaces; it attaches directly to the Meter without hoses.

An arrow **(A)** on the probe serves as a reminder of the direction you must orient the probe and the Meter when taking measurements.

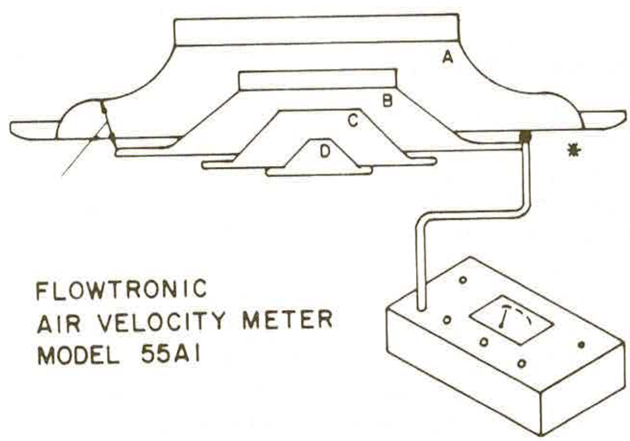
Figure 2-18 VELOMETER® ATTACHMENTS

NECK DIAM. IN.	CONES X	UP FACTOR	CONES X	UP FACTOR
6	5/8	0.12	1 5/16	0.15
8	13/16	0.20	1 3/4	0.27
10	1	0.32	2 3/16	0.42
12	1 3/16	0.45	2 5/8	0.59
14	1 3/8	0.61	3 1/16	0.80
16	1 5/8	0.79	3 1/2	1.05
18	1 13/16	1.00	3 15/16	1.30
20	2	1.25	4 3/8	1.63
24	2 3/8	1.80	5 1/4	2.35
30	3	2.80	6 5/8	3.60
36	3	3.18	6 5/8	4.15



ALNOR VELOMETER

NECK DIAM. IN	CONES X	UP FACTOR	CONES X	UP FACTOR
6	5/8	0.14	1 5/16	0.18
8	13/16	0.25	1 3/4	0.32
10	1	0.39	2 3/16	0.50
12	1 3/16	0.56	2 5/8	0.71
14	1 3/8	0.76	3 1/16	0.98
16	1 5/8	1.00	3 1/2	1.30
18	1 13/16	1.25	3 15/16	1.60
20	2	1.55	4 3/8	2.00
24	2 3/8	2.25	5 1/4	2.90
30	3	3.55	6 5/8	4.50
36	3	4.10	6 5/8	5.10



* ROTATE PROBE FOR MAXIMUM VELOCITY

Figure 2-19 DIFFUSER FLOW FACTORS

The deflecting vane anemometer also is available with static pressure probes and a Pitot tube. The static pressure probe must be used with specific published procedures and holes of a given size. The Pitot tube reads directly in fpm only. The instrument must *not* be used in contaminated air, or for measuring extremely hot or cold air.

The deflecting vane anemometer must be checked for calibration every 6 months, and either it or a hot wire anemometer is a NEBB required instrument for NEBB Certified TAB Contractors.

c. HOT WIRE ANEMOMETER

The hot wire anemometer (Figure 2-20) uses the principle that the resistance in a wire will increase when heated. The probe has a fine wire as thin as a hair heated by the battery. The main advantage of the hot wire anemometer is its ability to accurately read

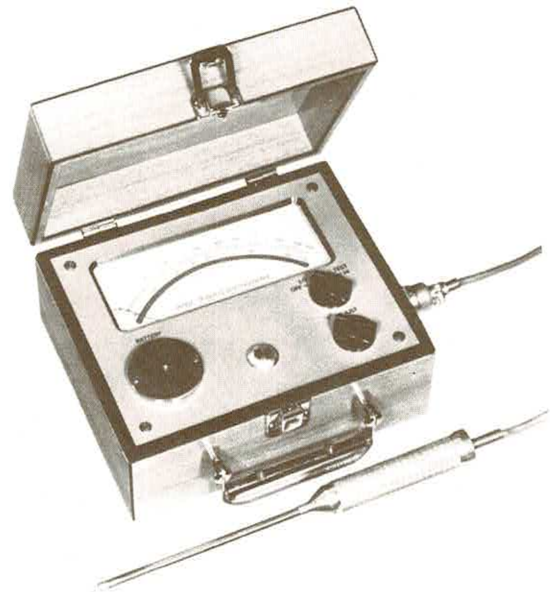


Figure 2-20 HOT-WIRE ANEMOMETER

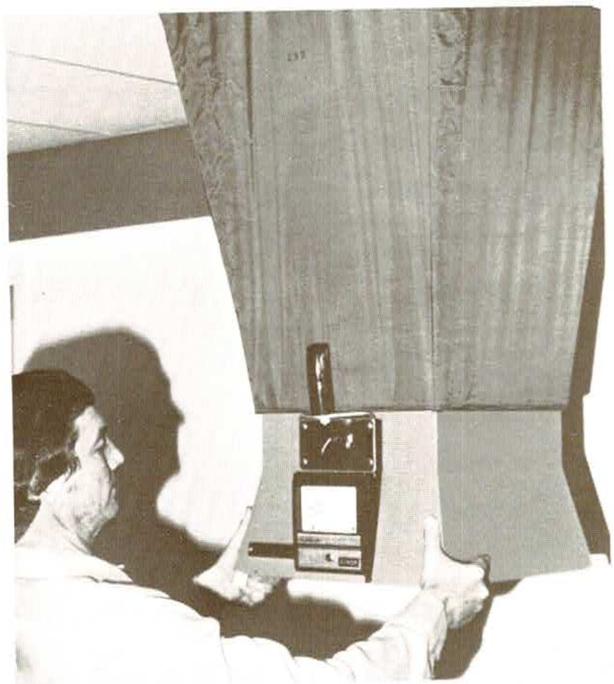


Figure 2-21 FLOW MEASURING HOOD

extremely low velocities such as room air currents. It also is very well suited to read air velocities around the face of hoods. However, its use is not limited to just low velocities, as it will do just about everything that deflecting vane anemometers will do. Disadvantages are the probe on these instruments is extremely directional and must be held at right angles to the airflow; and it is very delicate.

There are many newer makes and models of these instruments that have appeared on the market recently. It appears that this type of unit will be used much more in the future, as many will read temperatures as well as velocities. With attachments, they also can read static pressure. One newer model is available with a hot wire Pitot probe, a digital dial and will automatically total up readings when taking Pitot tube traverses. The accuracies of some of the newer instruments have not yet been verified. Always check with your NEBB Supervisor to determine the status of new instruments that are furnished.

Caution must be used to prevent dust and corrosive gases from damaging the hot wire tip. Batteries must be kept fresh or charged for proper operation. Either an approved hot wire anemometer or a deflecting vane anemometer is a required NEBB instrument for NEBB Certified Contractors.

4. Flow Measuring Hoods

The flow hood rapidly is becoming the most popular instrument in the TAB industry for measuring the airflows of all types of registers, grilles, diffusers and troffers. They read directly in "cfm" and eliminate the need for calculations, and for obtaining "K" factors and other exact data from the manufacturer. Because they are easy to use, many building owners and consulting engineers now have them and they are using them to verify the TAB contractor's reading.

The two main advantages of the flow hood are speed of operation and accuracy. The hood is held up to the ceiling around the diffuser for several seconds, and then you can read and record the airflow directly in cfm with no further calculation. Many diffusers, particularly those with perforated faces, are very difficult to read accurately with any other instrument. Some diffuser manufacturers now have discontinued testing for and publishing "K" factors. Instead they are recommending using a flow hood.

The flow hood consists of a measuring section that has an air measuring device somewhat similar to those installed in ducts for monitoring cfm for control purposes. Inside is a series of Pitot tubes arranged in a pattern similar to a traverse. Pressures from these sensing elements are directed into a manometer or a deflecting vane anemometer that is calibrated directly in cfm. From the measuring section, a variety of sizes of collector skirts are available. A gasketed frame is provided on the larger end so all you have to do is hold it up tight against the surface around the diffuser and read the dial.

Table 2-6 AIR MEASURING INSTRUMENTS

	RECOMMENDED USES	CALIBRATION REQUIRED	ACCURACY OF FIELD MEASUREMENT
U-TUBE MANOMETER	Air and gas (with water or oil instrument): Measuring pressure drops above 1" w.g. across filters, coils, eliminators, fans, grilles and duct sections. Measuring low manifold gas pressures.	None (Zero adjustment required for each set-up).	
MANOMETER VERTICAL/ INCLINED	Use with Pitot tube or static probe to determine static pressure, total pressure and velocity pressure in ductwork.	Same as U-Tube Manometer.	
MICRO-MANOMETER (HOOK GAUGE)	For air velocities below 600 ft./min. or low air pressure or vacuum readings. Used to calibrate other instruments. Difficult to use in the field.	Same as above. (All units must be accurately leveled before and after each reading.)	
PITOT TUBE	Measurement of airstream "total pressure", Measurement of airstream "static pressure", and Measurement of airstream "velocity pressure".	None required. However, the instrument must be maintained in clean condition.	Accuracy for field use is ±5% for the combination of the Pitot tube and the indicating instruments.
PRESSURE GAUGE (MAGNEHELIC)	Use with Pitot tube or static probe to determine static pressure, total pressure and velocity in ductwork.	None. Check against inclined draft gauge frequently.	Readable to 0.05 inches.
ROTATING VANE ANEMOMETER	Measurement of supply, return and exhaust air quantities at registers and grilles. Measurement of air quantities at the faces of maximum return air dampers or openings, total air across the filter or coil face areas, etc.	By an approved test agency every 6 months depending on usage. Check against recently calibrated instrument on each TAB project.	Average ±10%.
FLORITE ANEMOMETER	For measuring relatively uniform airflow at grilles and other air terminals.	By an approved test agency every 6 months depending on use. Frequent comparison with an instrument of known accuracy is recommended.	Within 10% of the range of the instrument.
PITOT TYPE ANEMOMETER	This instrument may be used for measurements of air velocity through both supply and return air terminals using the proper jet and the proper air terminal k factor (effective area) for the air flow calibration. The instruments may also be used for measuring some lower velocities where the instrument case itself is placed in the airstream.	The instrument should be checked by an approved (test) agency every 6 months or less depending on usage. Check against recently calibrated instrument on each project.	Accuracy is within ±10% when the instrument is in calibration and is used in accordance with the manufacturer's recommendations.
HOT WIRE ANEMOMETER	Used to measure very low air velocities such as room air currents and airflow in hoods and troffers. It is used for measurements at grilles and diffusers, although much less frequently than other velocity measuring instruments.	By the manufacturer or factors approved agency every 6 months. Check against recently calibrated instrument on each project. When in use frequently check zero or the calibration point setting.	Accuracy is ±10%.
FLOW MEASURING HOOD	To measure air distribution devices directly in cfm. When balancing a large number of ceiling diffusers or balancing troffer diffusers.	The flow measuring instrument used with the hood should be calibrated by the manufacturer or factory approved agency every 6 months	If the hood is properly shaped and positioned at the air terminal, accuracy of field measurements will be within the limitations of the flow reading instrument.

Most flow hoods are available with from two to four ranges. A selector switch is provided to change ranges and switch from supply airflow to exhaust airflow. For very low cfm, some are equipped with a perforated plate or blank off panels that are inserted in the measuring section to reduce the area and increase the velocity of the air. There is also a model that uses a solid state digital electronic manometer. It will read from 25 to 2500 cfm with no range switch required. It will also read supply or exhaust automatically. A minus sign appears on the display when a return or exhaust is being read.

With the correct attachments, some flow hoods have manometers that can be removed and used with a Pitot tube. When reading higher airflows, the hood will create some static pressure in the system while being used. This will reduce the cfm coming from the outlet being tested. A curve is furnished so that a correction can be made if needed. Due to the weight and from having to hold it up tight, continuous use may cause fatigue. Inaccurate readings will result if the hood isn't held up in a tight position. The hoods, even when packed for transport, are large and bulky. Electronic models will need to have their batteries charged or changed frequently. Calibration should be verified every 6 months.

5. Smoke Devices

Smoke devices (Figure 2-22) are useful for the detection of leaks and to study air currents.

a. SMOKE CANDLES

Smoke candles are available in 30 second to 10 minute durations. Various colors are available. They will produce a lot of smoke when needed in a continuous stream. They are useful for finding leaks in ducts and from other airtight spaces.

b. SMOKE STICKS

Smoke sticks are glass tubes filled with titanium tetrachloride. Breaking off the tip on one end will release the smoke which emits in a continuous stream about twice as heavy as a cigarette, for about ten minutes. They are useful for observing air currents in rooms, hoods, etc., and for identifying positive and negative pressures.

c. ASPIRATING SMOKE GENERATORS

Smoke generators produce intermittent puffs of smoke as the operator squeezes the squeeze bulb. Cartridges are inserted in the gun and when the bulb is squeezed, smoke emits. They are useful for the same purposes as the smoke sticks.

CAUTION: Although not toxic, some smokes will irritate people so avoid breathing where possible. Before using, all occupants should be advised. When large smoke candles are used, smoke detection equipment should be de-activated and the proper authorities notified. Most smokes leave a slight residue when used.

C HYDRONIC MEASURING

1. Pressure Gauges

a. PRESSURE GAUGES (BOURDON TYPE)

Bourdon tube pressure gauges (Figure 2-23) are used primarily for measuring static pressures at

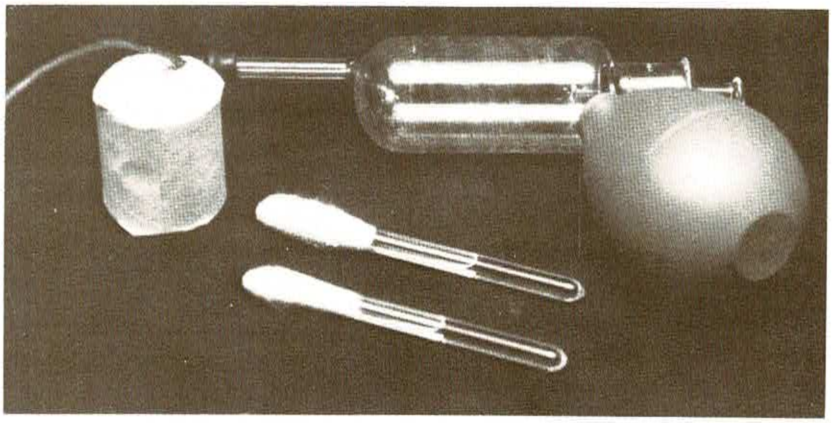


Figure 2-22 SMOKE DEVICES

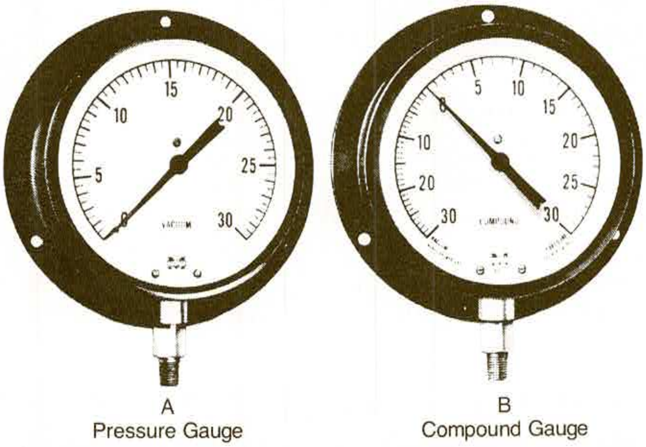


Figure 2-23 CALIBRATED PRESSURE GAUGE

pumps, primary heat exchange equipment, and terminal units. Test gauges are usually 3½″ to 6″ in diameter with bottom or back connections. Test gauges are available with various ranges of pressure or vacuum, or both which are called compound gauges.

Some precautions in the use and care of bourdon tube gauges are:

(1) Pressure gauges should be selected so the pressure to be measured falls in the upper half of the scale range.

(2) The gauge should not be exposed to pressures greater than the maximum dial reading. Similarly, a compound gauge should be used when vacuum conditions could occur.

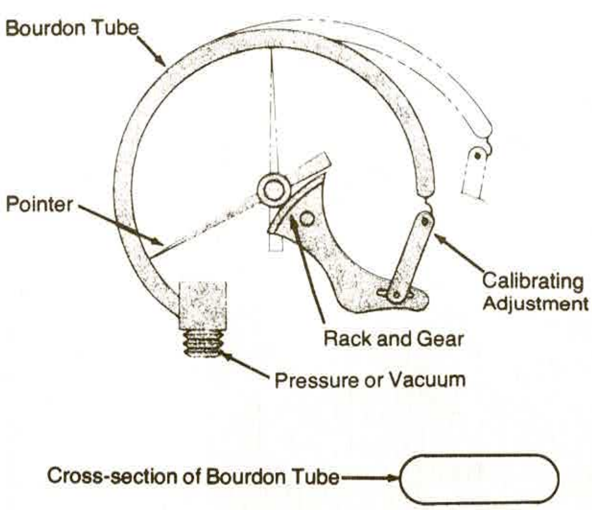


Figure 2-24 BOURDON TUBE GAUGE MECHANISM

(3) Test gauges should not be allowed to vibrate with pulsations in system pressures. Use of a gauge snubber, restrictor, or a needle valve at the gauge will extend the gauge life in addition to allowing greater accuracy of pressure readings.

(4) Always apply and remove pressure from a gauge slowly to prevent damage of internal linkage.

(5) Use of a test manifold (see Figure 2-25 and 2-26) whenever possible. A manifold allows the measurement of two pressures with the gauge at one location. This not only decreases wear on the gauge, but increases accuracy of differential pressure readings, as both pressures are read at the same static elevation.

(6) Bourdon tube gauges are sensitive to shock and abuse while not in use. Do not subject a test gauge to freezing conditions or constant vibration in a vehicle. Wrap the gauge with shock absorbing material when transporting.

b. DIFFERENTIAL PRESSURE GAUGES

A differential pressure gauge (Figure 2-27) is an instrument that reads directly in the difference between two pressures. Unlike a bourdon tube gauge that al-

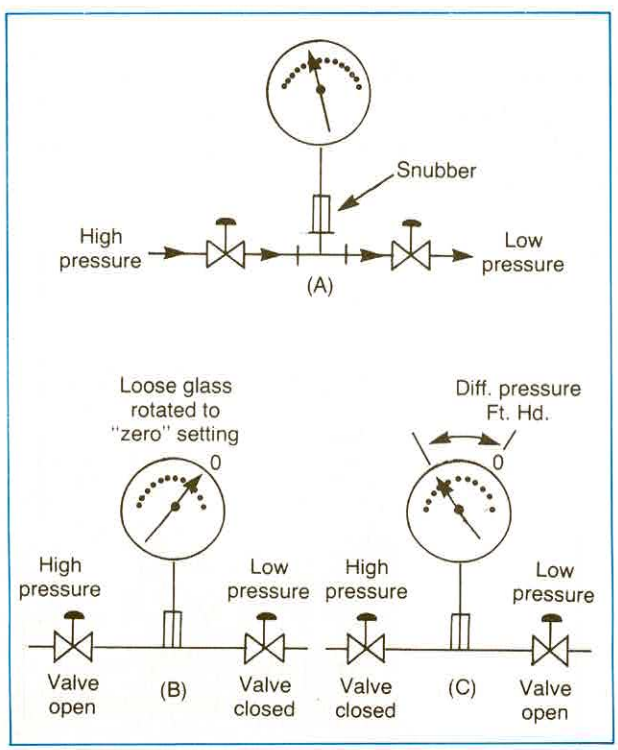


Figure 2-25 SINGLE GAUGE FOR MEASURING DIFFERENTIAL PRESSURES

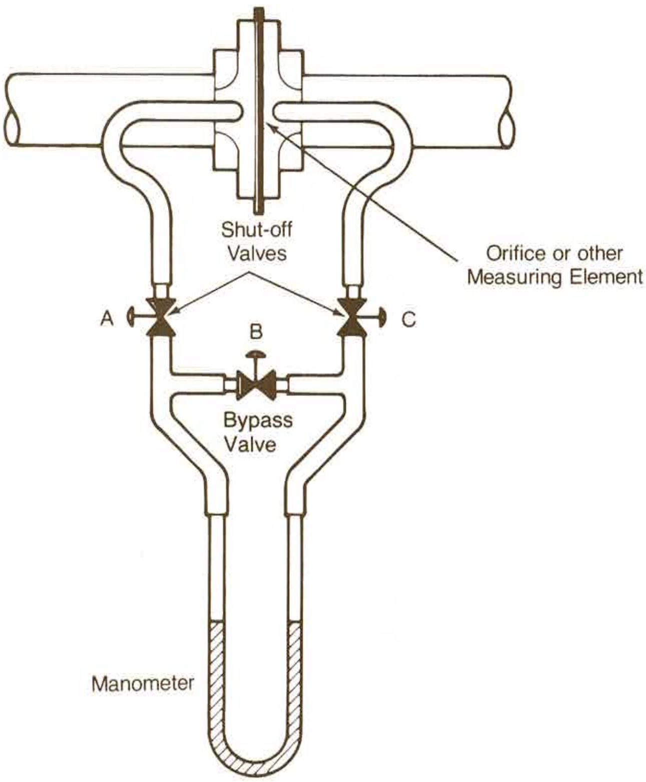


Figure 2-26 THREE-VALVE CLUSTER MANIFOLD

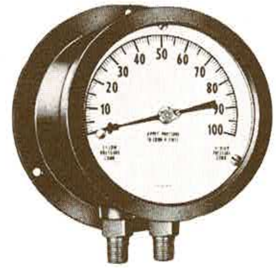


Figure 2-27 DIFFERENTIAL PRESSURE GAUGE

lows the physical reading of two static pressures and manual calculation of differential pressure, a differential pressure gauge compares two pressures in a bellows or mechanical assembly and outputs the pressure differential on the dial.

Differential gauges are used with most flow meters and combination balance valve/flow meters because of their greater accuracy and available speed when measuring small amounts of differential pressure. Figure 2-28 shows a typical differential pressure gauge in use with an annular flow indicator. Some precautions in the use and care of differential pressure gauges are:

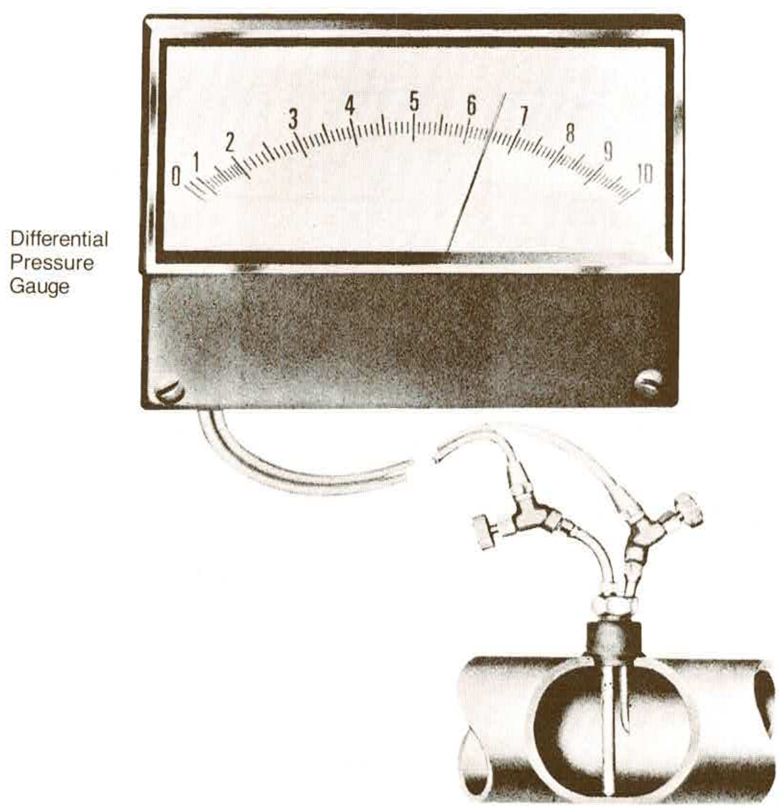


Figure 2-28 ANNULAR FLOW INDICATOR

(1) Do not use on components with differential pressures greater than the scale of the gauge. Many differential pressure gauges are provided with equalizer valves which should remain open when connecting and disconnecting the meter. Closing the equalizing valve will produce the differential reading. If by closing the valve, the differential is approaching the maximum dial reading, reopen the equalizing valve and use a differential gauge with a larger scale.

(2) Purge all air from gauge before reading. Air contained within gauge and bellows will result in erroneous readings.

(3) Differential pressure gauges are extremely sensitive to shock, vibration, and temperature. Take all necessary precautions recommended by the meter manufacturer to insure gauge safety.

2. Manometers

U-tube manometers (Figure 2-29) are primarily used for differential pressure measurement applications similar to a differential pressure gauge. Fundamental principles of manometers used in air system testing was discussed in the first section of this chapter and applies to manometers used in water testing. Several distinctions of manometers used in hydronic testing are:

(1) Water manometers normally contain mercury rather than oil.

(2) Water manometers must withstand greater operating pressures than air manometers, and must use manifolds for prevention of loss of mercury to the piping system.

(3) Air must be purged from instrument for correct reading to be obtained.

Figure 2-26 shows a typical mercury manometer with three-valve cluster manifold. Section II of the NEBB "Procedural Standards for Testing, Adjusting and Balancing of Environmental Systems" contains the calibration requirements of pressure gauges and the types and sizes required by NEBB Certified Contractors.

3. Flow Meters

Some flow meters (Figure 2-28) work by measuring the maximum velocity or "V_{max}". They do this by having a differential pressure gauge connected to two tubes inserted down the centerline of the pipe which record the difference in pressures. The gauge reading, when referred to appropriate calibration data, will indicate the fluid flow in "gpm". The calibration data is used to convert the pressure head differential to the flow velocity by using different multiplier constants, depending on the inside pipe diameter.

The velocity profiles become less bullet-shaped (flatter) as the inside pipe diameter increases. This allows

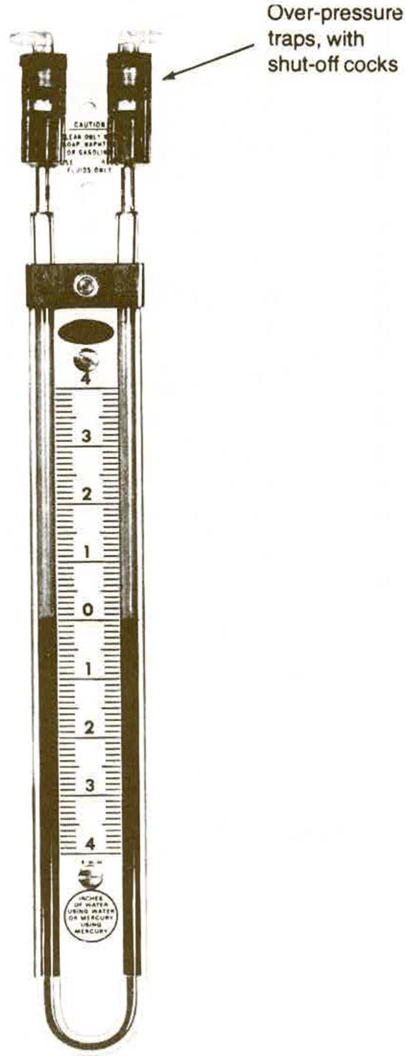


Figure 2-29 MANOMETER EQUIPPED WITH OVER-PRESSURE TRAPS

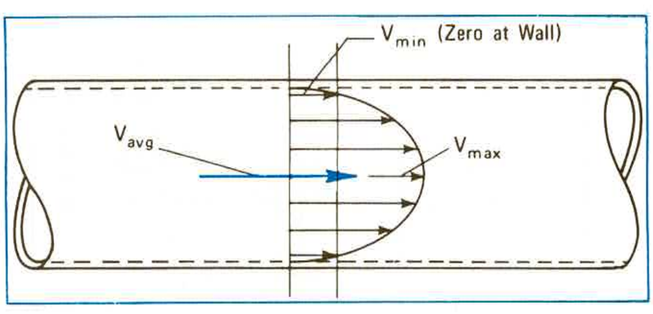


Figure 2-30 VELOCITY PROFILE

the ratio of "V_{max}" to "V_{ave}" to approach unity. This ratio can be called a "coefficient", which is adjusted by knowing the inside pipe diameter. The pipe area (A) also can be calculated if the inside pipe diameter is known. "$Q = V \times A \times C$", "V" becomes "V_{max}" and "C" becomes the coefficient (a constant).

Some flow meters position the sensing element directly in the center of the pipe. In order to do that, one must know the diameter of the pipe as the given variable. "V_{max}" is measured and in some cases the flow meters are read directly in gallons per minute or any other required measure of flow rate. The coefficient "C" must have been determined experimently by inducing various flow rates and recording the various head measurements for each diameter pipe. Manufacturers of such flow meters often can send test data illustrating "C" and flow characteristics on the same chart.

Another type of flow meter seems similar, but works differently. It also has a probe with an opening at the center of the pipe, which is measuring "V_{max}". The same principals are applied except that this device extracts water from the center of the tube and diverts it to a vertical column in which a ball of fixed diameter rests in a vertical conical-shaped chamber. The cross sectional area increases with the height. This means that the ball (of fixed weight) will rise to increasingly greater heights as the water flow increases. As the ball rises, the area around it increases, but the ball diameter stays constant. Therefore, the total area (the donut-shaped area formed by the wall minus the ball) also increases, and the velocity of the water decreases. When the velocity is constant, the ball will remain in one position. If the velocity becomes higher, the ball would rise higher, and if lower, the ball would sink lower.

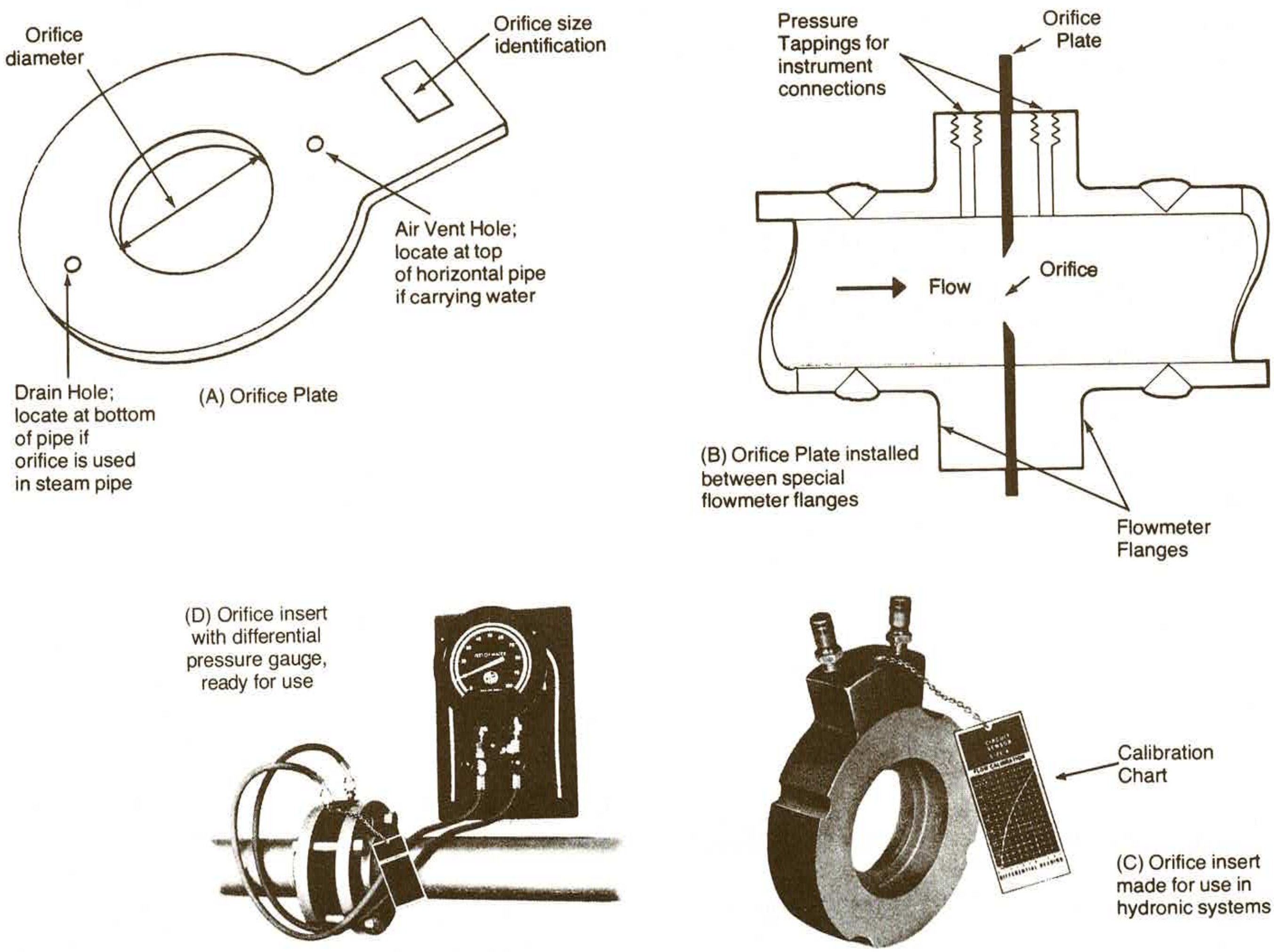


Figure 2-31 THE ORIFICE AS A MEASURING DEVICE FOR FLOW IN PIPING

When given other flow rates, the ball will ascend to one specific level, the level at which the area produces a velocity sufficient to hold the ball at that height. These tubes therefore can be calibrated to a specific flow rate. When the lab technicians of the manufacturer do the calibration, they take into account the different "C factors" for the different diameter pipes.

Another type of measuring device is an orifice meter (Figure 2-31). Orifice plates are essentially fixed circular openings through which the fluid flows. It is a restriction in the line. One would expect that it would take more pressure upstream to force the fluid through the restricted opening. The faster the fluid is flowing, the more upstream pressure is required. In this way, the pressure differential (that is, the upstream pressure minus the downstream pressure) is related to the velocity of the fluid.

The *venturi* (Figure 2-32) operates on the general principles and the same theories as the orifice but the shape of the venturi is somewhat different. A venturi meter has an improved accuracy over the orifice plate type by reading the pressures upstream and at the vena-contracta. The correction factor "C" still must be determined experimentally and is numerically close to that of the orifice.

4. Calibrated Balancing Valves

Another useful device (Figure 2-33) is the calibrated balancing valve. These valves perform dual duty as flow measuring devices and as balancing valves. They are similar to ordinary balancing valves, but the manufacturer has provided pressure taps into the inlet and outlet; and has calibrated the device by setting up known flow quantities while measuring the resistance which results from the different valve positions. These positions usually are graduated on the valve body (as a dial) and the handle as a pointer to indicate the reading. The manufacturer then publishes a chart or graph which illustrates the percentage amount that the valve is open (the dial settings), the pressure drop and the resulting flow.

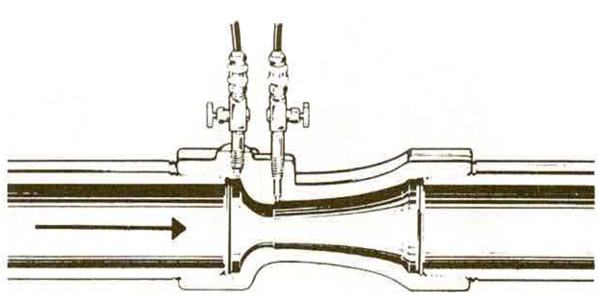


Figure 2-32 CROSS-SECTION OF A VENTURI

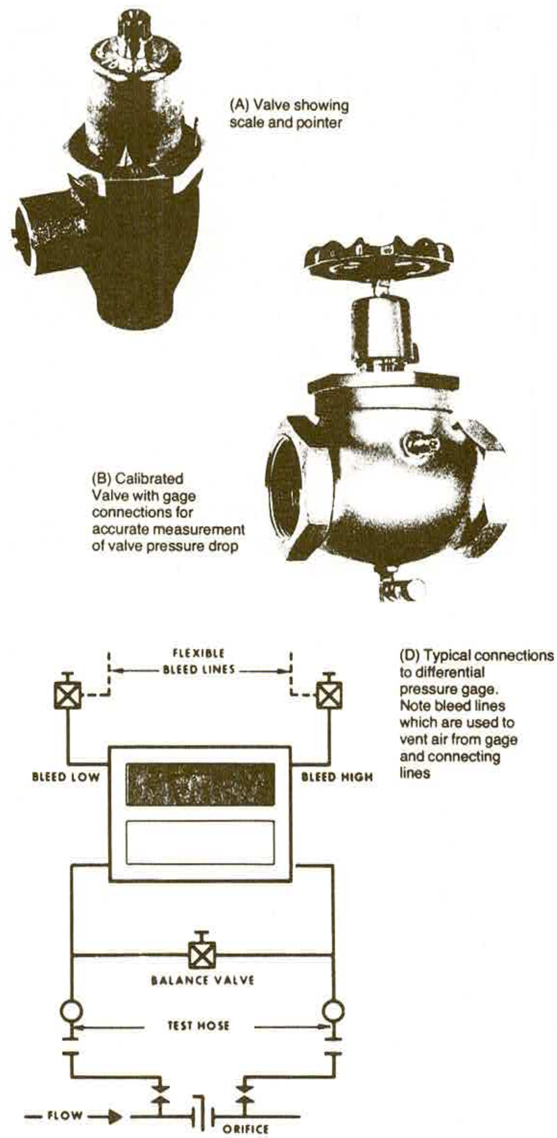


Figure 2-33 CALIBRATED BALANCING VALVE

An application in which these valves are indispensable would be a school building with central station chilling and a four pipe system connected to classroom unit ventilators. The system designer would specify that connections be furnished so that flow data for heating and cooling can be taken at each of the unit ventilators. Since stop valves are also required, it would be convenient for the installer to purchase these dual function valves. When connections are not provided, a great many hours could be spent drilling and tapping holes to connect pressure gauges. The same applies to thermometer wells and all other taps or connections needed for the balancing work.

Table 2-7 HYDRONIC MEASURING INSTRUMENTS

	Recommended Uses	Calibration Required	Accuracy of Field Measurement
U-TUBE MANOMETER	Measuring fluid pressure drops through coils, chillers, condensers and other heat exchangers, also across orifices and venturis	None (Zero adjustment required for each set-up)	
PRESSURE GAUGE	The same use as the U-Tube Manometer but for higher pressures.	By an approved test agency every 24 months depending on usage.	1/2 of 1% or 1/2 of scale division.
DIFFERENTIAL PRESSURE GAUGE	Same as pressure gauge.	Same as pressure gauge.	1/2 of 1% or 1/2 of scale division.
FLOW MEASURING DEVICES	Used for accuracy of measurement in fluid system when installed properly.	As required by the manufacturer.	Depends on instrument used.

Table 2-8 TEMPERATURE MEASURING INSTRUMENTS

	Recommended Uses	Calibration Required	Accuracy of Field Measurement
GLASS TUBE THERMOMETERS	Used to measure temperature of air or fluids.	None	1/2 of 1% or 1/2 of scale division.
DIAL THERMOMETERS	Used to measure temperature of air or fluids.	Check against mercury thermometer.	1/2 of 1% or 1/2 of scale division.
PYROMETERS	Used to measure surface temperature devices such as pipe or duct.	Every 12 months.	1/2 of 1% or 1/2 of scale of division.
PSYCHROMETERS	Used to measure both wet bulb and dry bulb temperatures to determine wet bulb depression and relative humidity.	None	1/2 of 1% or 1/2 of scale division.

D TEMPERATURE MEASURING INSTRUMENTS

1. Thermometers—Mercury Type

Mercury-filled glass thermometers (Figure 2-34) have a useful temperature range of from minus 40°F to 950°F. They are available in a variety of standard temperature ranges, scale graduations and lengths.

The complete stem immersion calibrated thermometer, as the name implies, must be used with the stem completely immersed in the airstream 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. Thermometers calibrated for partial stem immersion are more commonly used in conjunction with thermometer test wells designed to receive them or by inserting them through small holes drilled in the ducts. No emergent stem correction is required for the partial stem immersion type.

When the temperatures of the surrounding surfaces are substantially different from the measured airstream, there is considerable radiation effect upon the thermometer reading if left unshielded or otherwise unprotected from these radiation effects. Proper

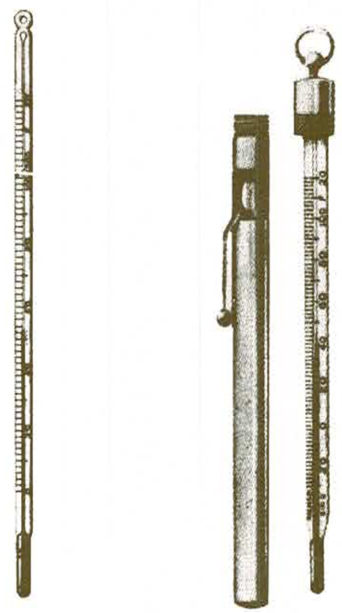


Figure 2-34 GLASS TUBE THERMOMETERS

shielding or aspiration of the thermometer bulb and stem can minimize these radiation effects.

2. Dial Thermometers

Dial thermometers (Figures 2-35 and 2-36) approved by NEBB are constructed with various size dial heads, from 1¾″ to 5″, with stainless steel encapsulated temperature sensing elements. Hermetically sealed, they are rust, dust and leak proof and are actuated by sensitive bi-metallic helix coils. Some can be field calibrated. Sensing elements range in length from 2½″ to 24″ and are available in many temperature ranges with and without thermometer wells. A disadvantage is that the sensing time lag is relatively long. They also must be field checked and calibrated against a glass stem mercury type thermometer on each TAB Project.

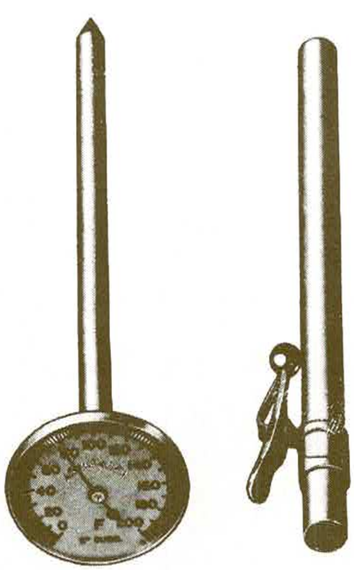


Figure 2-35 DIAL THERMOMETER

3. Electric/Electronic Thermometers

Electric or electronic thermometers have instrument cases with either an analog or digital readout, batteries, various switches, knobs, etc. to adjust the meter, and temperature sensing elements which usually are remote from the case and connected by means of wires or cables. Some also have multiple sensors that can be interchanged to meet job requirements and conditions.

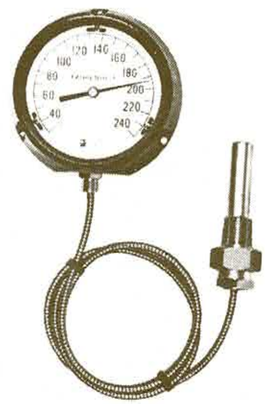


Figure 2-36 FLEXIBLE CAPILLARY TYPE DIAL THERMOMETER

4. Pyrometers

Pyrometers (Figure 2-37) normally used in measurements or surface temperatures in heating and air conditioning applications, use a thermocouple as a sensing device and a milli-voltmeter (or potentiometer) with a scale calibrated for reading temperatures directly. A variety of types, shapes and scale ranges are available; and often can be used as air thermometers also.

It should be remembered that the surface temperature of a conduit or duct is not equal to the airstream temperature and that a relative comparison is more reliable than an absolute reliance on readings at a single circuit or terminal unit.

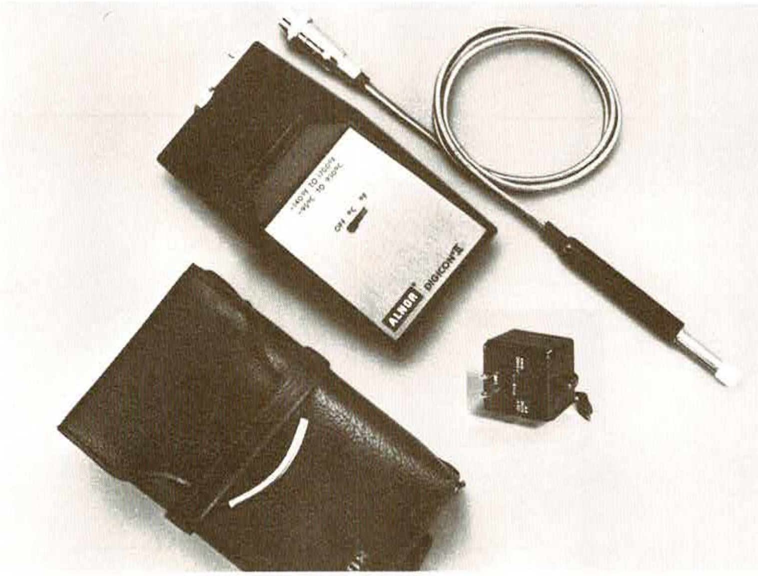


Figure 2-37 ELECTRONIC THERMOMETER/ PYROMETER SET

5. Psychrometers

The sling psychrometer (Figure 2-38) consists of two mercury filled thermometers, one of which has a cloth wick or sock around its bulb. The two thermometers are mounted side by side on a frame fitted with a steady motion through the surrounding air. The whirling motion is periodically stopped to take readings of the wet and dry bulb thermometers (in that order) until such time as consecutive readings become steady. Due to evaporation, the wet bulb thermometer will indicate a lower temperature than the dry bulb thermometer, and the difference is known as the wet bulb depression.

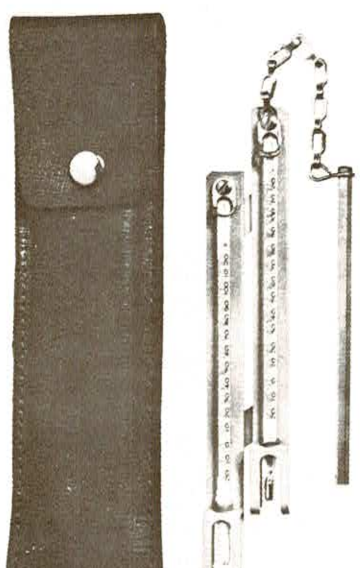


Figure 2-38 SLING PSYCHROMETER

The sling psychrometer can be used in determining the psychrometric properties of the conditioned spaces, return air, outdoor air, mixed air and conditioned supply air. The readings taken from the sling psychrometer can be spotted on a standard psychrometric chart from which all other psychrometric properties of the measured air can be determined.

Accurate wet bulb readings require an air velocity of between 1000 to 1500 fpm across the wick, or a correction must be made; therefore, an instrument with an 18″ radius should be whirled at a rate of two revolutions per second. Significant errors will result if the wick becomes dirty or dry.

E ELECTRICAL MEASURING INSTRUMENTS

1. Safety

Live electric circuits are always dangerous . . . and sometimes fatal! Electric shock can result in serious burns or death. Use extreme caution around live electric circuits and panels. Do **not** put your hands into an electric panel to pry wires out of the way, and do **not** try to force the jaws of a meter into position. As a TAB technician, you will need to measure voltage and amperage in TAB work, and possibly electrical resistance if troubleshooting.

When taking amperage readings do not attach the instrument and then start the motor. Position the instrument and read it after the motor is running at full speed. The inrush current required to start a motor is from three to five times higher than the load rated full nameplate current. Therefore, starting the motor with the instrument attached can damage the instrument.

Readings may be taken at the motor leads or from the load terminals of the starter. To determine the amperages of single phase motors, place the clamp about one wire. When involved with three phase current, take readings on each of three wires and average the results.

To measure voltage with portable test instruments, set the meter to the most suitable range, connect the test lead probes firmly against the terminals or other surfaces of the line under test. Then read the meter, making certain to read the correct scale if the meter has more than one scale. When reading single phase voltage the leads should be applied to the two load terminals. The resulting single reading is the voltage of the current being applied to the motor.

2. Multimeter (Volt/ohmmeter)

a. ANALOG TYPE

The analog multimeter (volt/ohmmeter) (Figure 2-39) is used for measuring voltages as well as resistance (ohms). Some types also will read amperage, but with a limited range which is not high enough for most motors. Two test leads are provided for taking the readings. There is one switch that is used to select the function, such as Volts AC, Volts DC, or ohms. Another switch often selects the range such as 0-150 volts, 0-500 volts, etc.

To use the meter for measuring voltage, hook up the leads to the correct receptacles for the function being tested. Note that the red lead is used in the positive (+) or hot wire opening of the receptacle. The black lead is usually plugged into the negative (−) or common wire opening or terminal. Set the meter for AC or DC voltage. Always select a range higher than the voltage expected. If there is any question, always start out on the highest range and work down if necessary. Firmly touch the test leads to the points being tested and read the voltage.

On single-phase circuits, read across the two load terminals. This single reading is the only voltage that can be measured at the circuit. On three-wire, three-phase circuits, it will be necessary first to read across poles 1 and 2, then poles 2 and 3, and finally poles 3 and 1. The three readings should be very close and the readings should be averaged. If there is much difference between the three readings, notify the proper people so that corrections can be made before possible damage results to the connected equipment.

For resistance measurements, the test leads must be connected to the correct receptacles or terminals on the meter. Set the function switch on "ohms" and set the range switch to the proper range. You will then have to set the zero adjustment to get the dial to read zero with the leads in tight contact with each other. Most meters also have an ohms adjustment that is also adjusted for full scale meter deflection with the leads separated. If you can't obtain full scale deflection, the battery in the meter is probably weak and will need to be replaced.

Once set up, proceed to touch the leads across the component being tested and read the resistance. *Never try to read resistance in a live circuit or the meter will be changed.* It may be necessary to disconnect one end of the tested component to eliminate stray current from causing an erroneous reading.

Amperage readings with this type of meter are more involved and limited to low currents. The circuit must be broken into and the meter connected in series with the circuit.

b. DIGITAL TYPE

Use and operation of a digital multimeter (Figure 5-40) is almost identical to an analog type. The digital types are required for the precise low voltage readings required for testing and adjusting solid state controls. Many of the newer electronic and computer controlled V.A.V. boxes require a digital multimeter to set up and adjust. The TAB technician will find that these meters will be needed more often as these newer electronic controls become more popular.

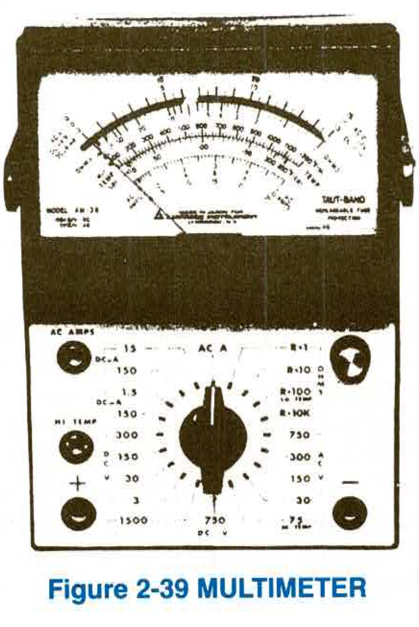


Figure 2-39 MULTIMETER

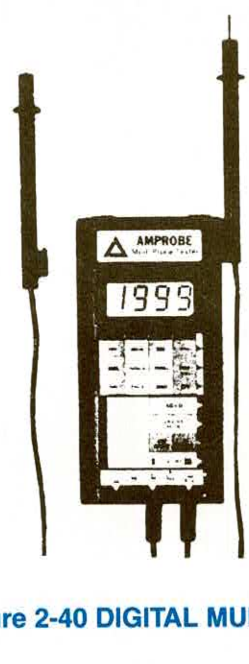


Figure 2-40 DIGITAL MULTIMETER

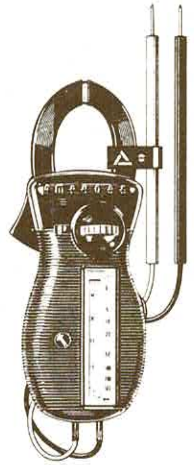


Figure 2-41 CLAMP-ON VOLT-AMMETER

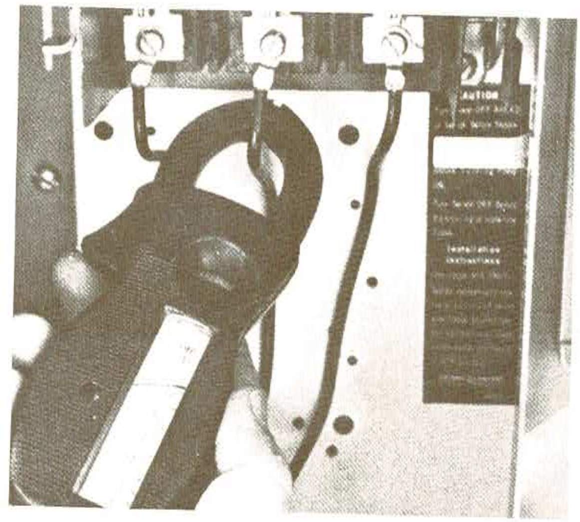


Figure 2-43 MEASURING AMPERAGE

3. Volt-Ammeter

The volt-ammeter (Figures 2-41 and 2-42) is currently the most popular volt-ammeter in use by TAB technicians. The voltmeter is only AC and operates similar to a multimeter. The ammeter operates by *induction,* similar to the operation of a transformer. By opening the jaws on the meter and putting them around the wire and closing them (see Figure 2-43), the current going through the wire will induce a current to the meter and provide a reading. Always select a range higher than what is expected to be in the wire and work down to lower ranges if necessary. These meters can be damaged easily by high readings above scale ranges.

Never hang a meter on while a motor is starting. The inrush current of a starting motor will be so great that it will usually "peg" the meter and damage it. Wait until the motor is up to full speed. Amperage readings are taken on one leg for single-phase circuits. For three-phase circuits, a reading is necessary on each of the three legs. NEBB requires that all three readings to be entered on the NEBB Test Report Forms.

There are many makes and models of volt-ammeters available. Some have digital dials and some also include a single range ohmmeter. The volt-ammeter is a NEBB required instrument and it must be checked for calibration every 6 months.

Figure 2-42 DIGITAL VOLT-AMMETER

F ROTATION MEASURING INSTRUMENTS

Tachometers are one type of instruments used to measure the speed at which a shaft or wheel is turning, usually in terms of revolutions per minute (rpm). The TAB technician will need to measure the rpm of fans, motors, pumps, etc., as a matter of routine. Rotating equipment also can cause severe injury, so be extremely cautious around rotating equipment, especially when the guards have been removed. Don't wear loose clothing that might get wrapped around a shaft or pulley causing serious injury.

1. Direct Contact Tachometers

A direct contact tachometer is one in which the shaft of the tachometer makes contact with the rotating shaft to be tested. The tachometer must be held exactly in line with the driving shaft. Rubber tips are provided that slip on to the tachometer shaft. In years past, many manufacturers of fans and motors would countersink the ends of their shafts so a pointed rubber tip could be inserted easily for testing. Since most manufacturers have abandoned this practice, flat rubber tips are being provided with most instruments so that a reading can be obtained from the end of a flat shaft. The user must be sure the tip is centered on the shaft. Wobbly readings will be inaccurate. Use firm pressure so there is no slippage.

a. CHRONOMETRIC TACHOMETERS

The chronometric tachometer (Figure 2-44) is a revolution counter with a built in stop watch which eliminates human error for timing the reading. The TAB technician just aligns the tach with the rotating shaft, makes contact and presses the start button. The counter then counts for a predetermined time span that is controlled automatically by the built-in stop watch. The counter stops automatically and you can then remove the instrument and read the dial which is calibrated directly in rpm. This is a very accurate instrument when used in a countersunk hole. Calibration should be checked every 12 months.

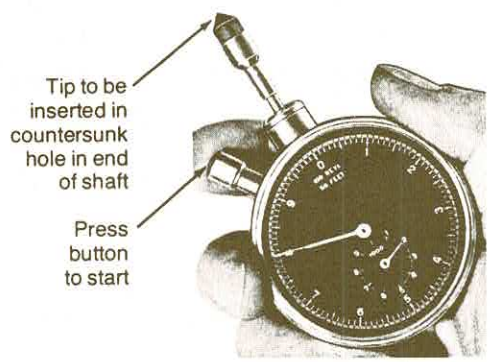
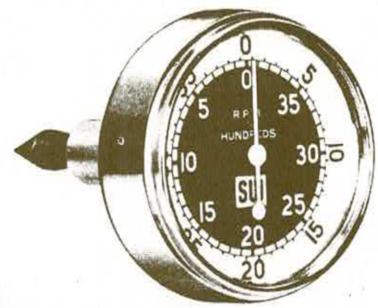
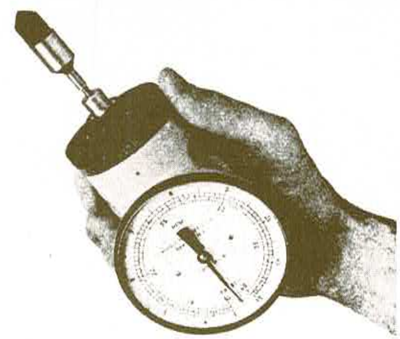


Figure 2-44 CHRONOMETRIC TACHOMETER



(A) Tachometer with single speed range



(B) Tachometer with multiple speed ranges

Figure 2-45 CENTRIFUGAL TACHOMETERS

b. CENTRIFUGAL TACHOMETERS

A centrifugal tachometer (Figure 2-45) uses a mechanical centrifugal mechanism similar to an automobile speedometer. It is easy to use; just hold it against the rotating shaft and read instantly the results. It reads rotation in either direction. They are inexpensive but the dial divisions do not usually provide for precise rpm readings. Usually you can get within 20 or 25 rpm's or so, but no closer. Calibration should be checked every 12 months.

c. ELECTRONIC TACHOMETERS

Electronic tachometers are somewhat similar in looks and use to centrifugal tachometers. The digital readings are obtained through electronic means rather than mechanically. Most of the newer ones have a holding provision that will keep the reading in memory. They are very accurate and easy to use. Batteries must be maintained. Calibration should be checked every 12 months.

2. Non-Contact Rotation Measuring Instruments

Non-contact electronic instruments, as the title implies, actually do not contact the rotating parts. They measure the rpm by flashing a light at the moving parts and by either electronically controlling the flash rate or counting the reflections, determine the rpm. They are very accurate and inherently safer to use since no contact need be made. They are used extensively on inline fans, pumps and other locations where access or the removal of belt guards would make the use of a direct contact tachometer difficult. A non-contact electronic instrument is a NEBB required instrument and calibration should be checked every 24 months.

a. PHOTO TACHOMETER

The photo tachometer beams a small light at the rotating part. Most models will require a single mark on

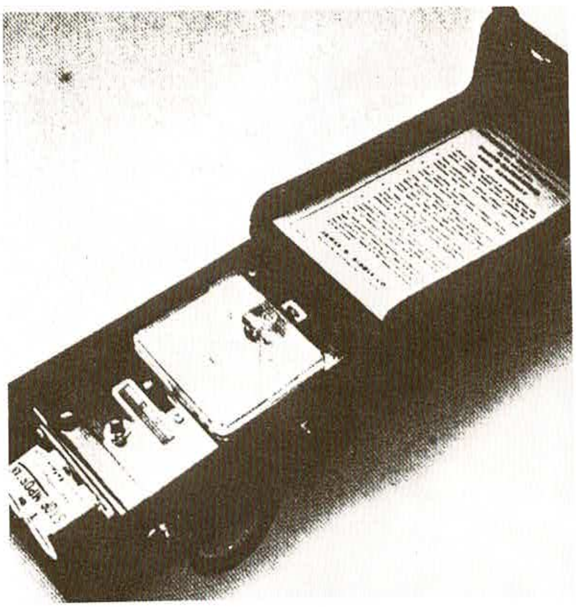


Figure 2-46 PHOTO TACHOMETER

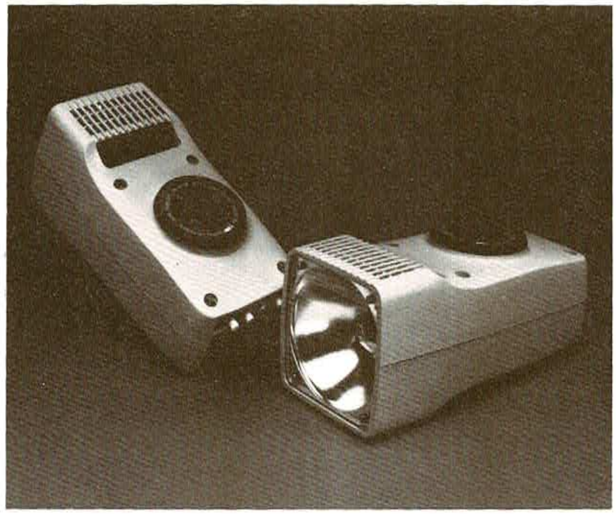


Figure 2-47 STROBOSCOPE

the rotating part such as a piece of reflecting tape or a chalk mark. Every rotation of the rotating part will allow the light emitting from the photo tachometer to be reflected back to the instrument where it is sensed, counted electronically, and then the rpm is displayed on the dial.

Most of the newer ones use digital readout and have a memory for storing the results. They are small, lightweight, battery powered, and are very accurate. They are very good for measuring extremely high rpm, but they can be difficult to use in high light areas, such as on roof equipment on a sunny day. Some models are equipped with an attachment for direct contact use.

Many new photo tachometers have been appearing on the market recently. If a purchase is expected, be sure that the instrument meets the standards set by the National NEBB Technical Committee. Recent experience has shown that some units have had a short battery life.

b. STROBOSCOPE

The stroboscope (Figure 2-47) has an electronically controlled flashing light. The frequency of the flash,

Table 2-9 ROTATION MEASURING INSTRUMENTS

	Recommended Uses	Calibration Required	Accuracy of Field Measurement
REVOLUTION COUNTER	To read RPM of rotating shaft with separate timing device.	None	±5% when used properly.
CENTRIFUGAL TACHOMETER	Same as above but direct reading requiring no timing device.	At least every 12 months.	1/4 of 1% of the dial range.
CHRONOMETRIC TACHOMETER	Same as above but combines a revolution counter with synchronized stop watch.	At least every 12 months.	± 2% of full scale.
STROBOSCOPE	To read RPM of any moving part by use of an electronically flashing light.	Calibrate against known value (Fluorescent light)	± 2% of full scale.
PHOTO TACHOMETER	Same as Stroboscope.	Same as Stroboscope.	±2% of full scale.

which is of very short duration, is adjustable and controlled by a calibrated dial on the instrument. When the frequency of the flashing light is exactly the same as the rpm of the rotating part that the flashing light is beamed at, the rotating part will appear to stand still. To obtain this, the TAB technician will point the light at the rotating object and then adjust the frequency of the flash until the object stands still. The dial reading is the frequency of the flash, which also is the rpm to be recorded.

These instruments will require some practice if erroneous readings are to be avoided. Multiples of the rpm of the rotating part, called harmonics, will also make the rotating part appear stationary. Speeds of 1/2, 2, 3 or other multiples of the actual rpm will give this effect. By starting at a lower rpm reading and adjusting upward, carefully notice how high you can go before multiple readings are observed. On a shaft with a keyway, only one keyway should be seen. If two or more appear, the stroboscope is flashing too fast. With some practice and by following the instructions, the TAB technician will understand and be able to correctly operate this instrument.

The stroboscope is extremely accurate when used properly. Built in calibration aids are provided and should be checked before each measurement. Most stroboscopes need 115 volt power and will require use of an extension cord. Although some newer ones are light and are battery operated, most are heavy and bulky. Some models require a ten minute warmup period and they are difficult to use in bright light.

NEBB

CHAPTER 3 ELECTRICAL EQUIPMENT AND SYSTEMS

A INTRODUCTION

1. Safety First

The first priority for a TAB technician working with electricity must be **caution.** The voltages and currents involved in HVAC work can cause shock, severe burns and death. When performing TAB work, should you accidentally discover a live circuit, do not touch any conductor or wet surfaces that could ground you and cause electrocution. It is a good practice to wear rubber soled shoes and not to use aluminum ladders while working around electric circuits.

When it becomes necessary for the TAB technician to work on wiring, such as to reverse motor rotations or to change heater coils, be sure that the circuits are de-energized and locked out. Tag them, noting that work is being done and not to turn them back on. Carry a padlock (and keys) to physically lock panels where possible. Before touching any electric circuit or component, verify that the electricity is off by use of a voltmeter. Test across the circuit and between all legs of the circuit and ground. All readings should be "zero". If you get any voltage reading when the circuit is "off", don't touch it. Have an electrician check out the circuit to find the reason for the voltage reading.

When working in panels, be cautious of other live circuits. The circuit you are working on may be de-energized, but quite often there are other circuits in the same panel that still can be energized and potentially dangerous.

2. Understanding Electricity

The TAB technician *must* have a basic knowledge of electricity. HVAC systems are powered predominently by electric motors, and many temperature control systems are also electric. The TAB technician must know how to test electric motors and insure that they are properly protected. This chapter will provide the basics needed by the TAB field technician. The subject of electricity is interesting and diverse, and most TAB technicians probably will want to obtain a deeper understanding than presented here. A good starting reference is the NEBB Publication, "Environmental Systems Technology".

B ELECTRICAL THEORY

It is important to memorize the definitions of *volt, amp,* and *ohm*. Notice the similarities to water and air systems which have pressure, flow and resistances. In electricity, it is a flow of electrons rather than water or air. *Direct current* (DC) flows very much as hydronic systems would (in one end of a circuit and out the other end). *Alternating current* (AC), when viewed as a hydronic system, would reverse its flow in the conduit 120 times per second as it flows in the same direction 60 times per second (60 Hertz or cycles per second, 50 times per second (50 Hz) in some countries). This produces pressure pulses that are sent down the conduit.

Electrical systems are used to transmit power as hydronic systems are used to transmit heat. Heat is transported by the mass of the flowing fluid; therefore, the flow must be like DC current: to flow in only one direction so as to continually exchange the fluid mass (the heat). AC current is different, being similar in concept to a hydraulic system in an auto. Pressure (voltage) is applied at one end, and although there is not a continuous flow to the other end of the system, power is still transported by the fluid (a hydraulically pulsed system) to a piston. As there is little friction loss with this "pulse" system compared to one with continuous flow, alternating current must be used when transmitting electrical power at high voltages. Direct current is still in use in the "downtown" areas of some large cities, and in some special applications (elevators, etc.), but the TAB technician seldom will encounter this type of electrical current.

Electrically, a similar "piston mechanism" is needed. This mechanism is found in the form of magnetic fields. A magnetic field is the "magic" that transforms power into motion within motors. It is also the magnetic field which allows a transformer to change the

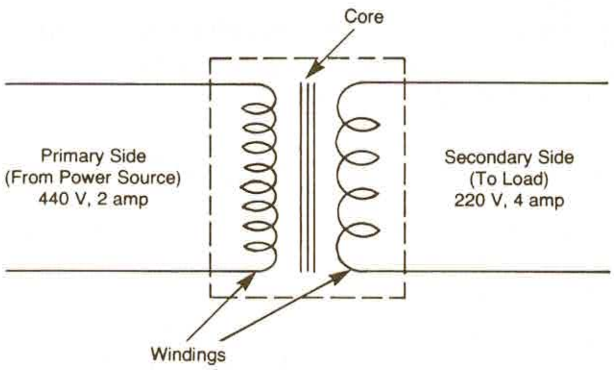


Figure 3-1 BASIC TRANSFORMER

voltage on the incoming (primary) side to a different voltage on the outgoing (secondary) side (see Figure 3-1).

Most of us are familiar with the older series Christmas tree light circuits, where if one bulb burnt out, the whole string of lights went out. These circuits were similar to the one shown in Figure 3-2 "Series Circuit." However, the Christmas lights used for outside decorating burned independently because they were wired in parallel, that is, each bulb received 120 volts as shown in Figure 3-3. Knowing the circuit voltage (E) and the wattage (W) or resistance (R) of the bulbs, the current flow or amperage (I) can be calculated from Equation 3-1 and 3-2. Equation 3-1 also is known as "Ohm's Law."

Equation 3-1
(Ohm's Law)

$$E = I \times R \text{ or } I = \frac{E}{R}$$

Equation 3-2 (a)

$$P = E \times I \text{ (Single-Phase)}$$

Equation 3-2 (b)

$$P = E \times I \times 1.73 \text{ (Three-Phase)}$$

Where:

E = Volts

I = Current Flow (Amps)

R = Resistance (Ohms)

P = Power (Watts)

1000 Watts is equal to a kilowatt, which is a standard unit of power measurement used by utility companies.

Example 3A

If each bulb in Figure 3-3 is rated at 60 watts, find the total current flow of the circuit in amps.

Solution

Using Equation 3-2 (a)

$$P = E \times I;\ I = \frac{P}{E} = \frac{3 \times 60 \text{ Watts}}{120 \text{ Volts}} = 1.5 \text{ Amps}$$

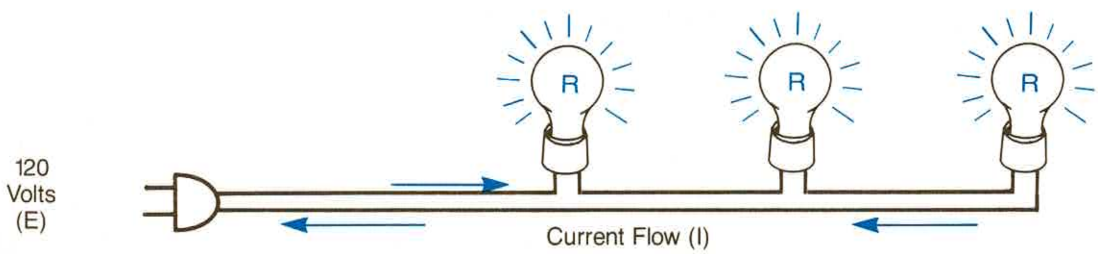


Figure 3-2 SERIES CIRCUIT

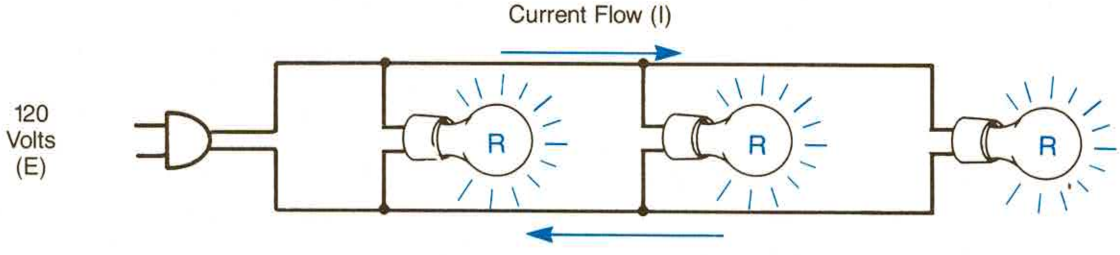


Figure 3-3 PARALLEL CIRCUIT

Ohm's Law may be illustrated with an example of a 24-inch diameter water main flowing from a dammed water reservoir. If the water level is 230 feet above the main (230 volts), and a gate valve, located 100 feet from the end of the pipe is barely opened (high resistance), the water flow (amps) will be very small; small enough in fact that a man could hold his hand in the water flow coming out of the end of the pipe.

I = E/R, so "I" becomes higher as "R" decreases in value. As the gate valve is opened (resistance decreases), the flow continues to build up until the man would be swept off of his feet. If a smaller 2-inch diameter pipe was used, the pressure (230 ft. w.g.) would be the same; but the flow with the valve wide open, would never equal that of the 24-inch diameter main, or even come close to it.

The same concept applies to electrical wiring. However, unlike the water pipes, the wire will make an attempt to handle the flow; and in extreme instances, will become red hot and glow (as in electric resistance heaters). This is the reason fuses and circuit breakers are used to protect the wiring system and equipment from being overloaded.

C ELECTRIC WIRING

1. Wire

Wire used in electric circuits can be made from many different materials. When electricity (electrons) flows easily in the material, it is called a *good conductor.* If it is difficult to pass an electric current through a material, it is known as an *insulator.*

Some good conductors of electricity are copper, silver and aluminum. Most wire is made from copper because it is a good conductor and can be purchased at a reasonable price. Silver is a better conductor of electricity than copper, but higher cost and poorer availability usually prevent it from being used as wire.

In some cases, the path for the electrons is provided as printed wiring on specially made "printed circuit boards". Many radios and controls have circuits of this type. Many or all of the parts are connected with thin strips of copper or silver. This is "printed wiring". While it is used extensively in certain applications, many of the electric circuits in equipment with printed wiring are still connected with conventional wires.

Copper wire is used the most, while aluminum wire is used for high voltage lines that carry electricity long distances from power plants to users. The wiring used in HVAC work and in the service connections to equipment can be a single solid wire or a single wire made up of many strands of smaller wires. Stranded wiring normally is more flexible.

2. Wire Covering/Insulation

The solid or stranted wires can be coated with varnish, left bare or covered with an insulating material and/or a braided covering.

Coated wire usually is referred to as "magnet wire". It can be found in electromagnets, coils for transformers and in the antennas of portable radios. The varnish coating must be removed when this type of wire is soldered.

Insulated wire is used for lamp cords, house wiring, control wiring and telephone cables. By definition, a cable is an assembly of two or more wires inside a common covering. Cables used in control wiring and telephone circuits have color-coded insulation on the individual wires for each in connecting or tracing.

Wire with braided covering called "shielded cable" is used extensively in control and communications work. Applications include lead-in cables that connect antennas to TV sets, leads for electrical instruments and transmission line cables. This type of cable is used because the braided covering reduces magnetic interference problems such as in radio and television reception. It is much more expensive than other wire, but it is used because of its special "shielding" properties. A "coaxial cable" uses two conductors, one over the other. The outer braided conductor shields magnetic fields from the inner conductor.

3. Wire Sizes

Table 3-1 lists various sizes of wire. Note that the size is given by a number, ranging from 18 to 2000. This numbering system is known as the American Wire Gauge (AWG). It is used to give specific information about the various sizes of wire.

In the chart, from size 18 to size 0, the larger the wire number, the smaller the diameter of the wire; then the opposite is true. It is important that the proper size of wire is selected for the job.

Figure 3-4 shows the relationship between pipe/water flow and wire/current flow. If the power wires connected to equipment are too small, the reduced current flow will not allow the job to be done without equipment damage or failure.

Round wire is measured in "circular mils." A mil is one thousandth of an inch (.001 in.). A circular mil is the cross-sectional area of a wire that has a diameter of one mil.

Table 3-1 PROPERTIES OF CONDUCTORS [Reprinted with permission from The National Electrical Code Handbook (1981)]

Size AWG, MCM	Area Cir. Mils	Concentric Lay Stranded Conductors: No. Wires	Concentric Lay Stranded Conductors: Diam. Each Wire Inches	Bare Conductors: Diam. Inches	Bare Conductors: *Area Sq. Inches	DC Resistance Ohms/M Ft. At 25°C, 77°F. Copper: Bare Cond.	DC Resistance Copper: Tin'd. Cond.	DC Resistance Aluminum
18	1620	Solid	.0403	.0403	.0013	6.51	6.79	10.7
16	2580	Solid	.0508	.0508	.0020	4.10	4.26	6.72
14	4110	Solid	.0641	.0641	.0032	2.57	2.68	4.22
12	6530	Solid	.0808	.0808	.0051	1.62	1.68	2.66
10	10380	Solid	.1019	.1019	.0081	1.018	1.06	1.67
8	16510	Solid	.1285	.1285	.0130	.6404	.659	1.05
8	16510	7	.0486	.1458	.0167	.653	.679	1.07
6	26240	7	.0612	.184	.027	.410	.427	.674
4	41740	7	.0772	.232	.042	.259	.269	.424
3	52620	7	.0867	.260	.053	.205	.213	.336
2	66360	7	.0974	.292	.067	.162	.169	.266
1	83690	19	.0664	.332	.087	.129	.134	.211
0	105600	19	.0745	.372	.109	.102	.106	.168
00	133100	19	.0837	.418	.137	.0811	.0843	.133
000	167800	19	.0940	.470	.173	.0642	.0668	.105
0000	211600	19	.1055	.528	.219	.0509	.0525	.0836
250	250000	37	.0822	.575	.260	.0431	.0449	.0708
300	300000	37	.0900	.630	.312	.0360	.0374	.0590
350	350000	37	.0973	.681	.364	.0308	.0320	.0505
400	400000	37	.1040	.728	.416	.0270	.0278	.0442
500	500000	37	.1162	.813	.519	.0216	.0222	.0354
600	600000	61	.0992	.893	.626	.0180	.0187	.0295
700	700000	61	.1071	.964	.730	.0154	.0159	.0253
750	750000	61	.1109	.998	.782	.0144	.0148	.0236
800	800000	61	.1145	1.030	.833	.0135	.0139	.0221
900	900000	61	.1215	1.090	.933	.0120	.0123	.0197
1000	1000000	61	.1280	1.150	1.039	.0108	.0111	.0177
1250	1250000	91	.1172	1.289	1.305	.00863	.00888	.0142
1500	1500000	91	.1284	1.410	1.561	.00719	.00740	.0118
1750	1750000	127	.1174	1.526	1.829	.00616	.00634	.0101
2000	2000000	127	.1255	1.630	2.087	.00539	.00555	.00885

* Area given is that of a circle having a diameter equal to the overall diameter of a stranded conductor.

The values given in the table are those given in Handbook 100 of the National Bureau of Standards except that those shown in the 8th column are those given in Specification B33 of the American Society for Testing and Materials, and those shown in the 9th column are those given in Standard No. S-19-81 of the Insulated Power Cable Engineers Association and Standard No. WC3-1969 of the National Electrical Manufacturers Association.

The resistance values given in the last three columns are applicable only to direct current. When conductors larger than No. 4/0 are used with alternating current, the multiplying factors in Table 9 compensate for skin effect.

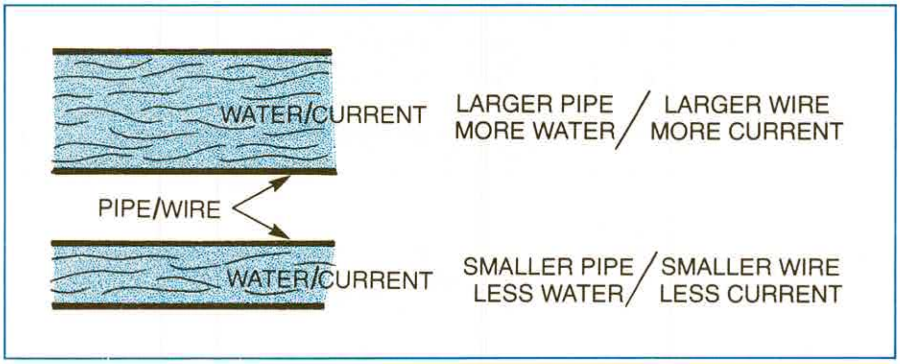


Figure 3-4 WIRE CAPACITY

Circular mils are easier to visualize through an example. Five squared is equal to 25. Therefore, a wire 5 mils in diameter would have a cross-sectional area that would contain 25 circular mils. Another wire has a diameter of 8 mils. Its area then would be 64 circular mils.

There is a good reason for converting the diameter of wire into circular mils. Working with the cross-sectional area of the wire makes it possible to select the right size of wire to safely carry the prescribed number of amps in the circuit.

4. Insulators

Insulation is a material used to cover wire. Wires are insulated to keep them from touching each other or contacting other conductors. If they did touch, it would cause a bright spark called an "arc". Anytime there is an arc, there is danger that a fire will start, a fuse will blow, or a circuit breaker will trip. Therefore, for protection and safety, wire is manufactured with an insulating cover of rubber or plastic molded on it.

Insulation comes in many colors, combinations of colors and striped patterns. Insulation also prevents any person touching the wire from getting a shock.

The best insulators are rubber, shellac, glass, mica, plastic and paper. Electrical tape and plastic shields are also forms of insulators. When a large number of wires are taped together, or encased in protective tubing, the assembly is called a "wiring harness."

Insulators for high voltage or other transmission service lines are best known from the many years of observance of "telephone poles" or the high metal towers which crisscross the country.

D ELECTRIC POWER

1. Electric Service

The standard electrical service in the United States is alternating current (AC) at 60 hertz (Hz), formerly known as cycles per second (cps). The two most common types of service normally used with HVAC systems are single-phase and three-phase circuits.

Single-phase, three-wire circuits are used for most residential services as well as for light commercial services. For heavier commercial, industrial or where ever larger motors are used, three-phase, three or four wire service is provided.

2. Single-Phase Circuits

A *measured voltage* may not be exactly one of the values of voltages indicated in Figure 3-5. Voltages can vary, and in normal situations, a variation of ± 10% will not adversely affect equipment operation. The basic 115 volt two-wire circuit shown in part "A" of Figure 3-5 is very common. There is a potential or "pressure" of 115 volts between the hot wire and the neutral or ground. The normal household circuit, such as a lamp, is representative of such a circuit. The 115 volt potential in the hot wire will exist between the hot wire and the neutral, or between the hot wire and any other ground, such as a pipe or a person which might contact the wire.

The *neutral or ground wire* is another matter. The neutral normally has no voltage potential. Theoretically, if the neutral contacts a pipe or a person, nothing will happen. The neutral is connected to the generator. The term "theoretically" is used, because in actual field conditions, stray currents can find their

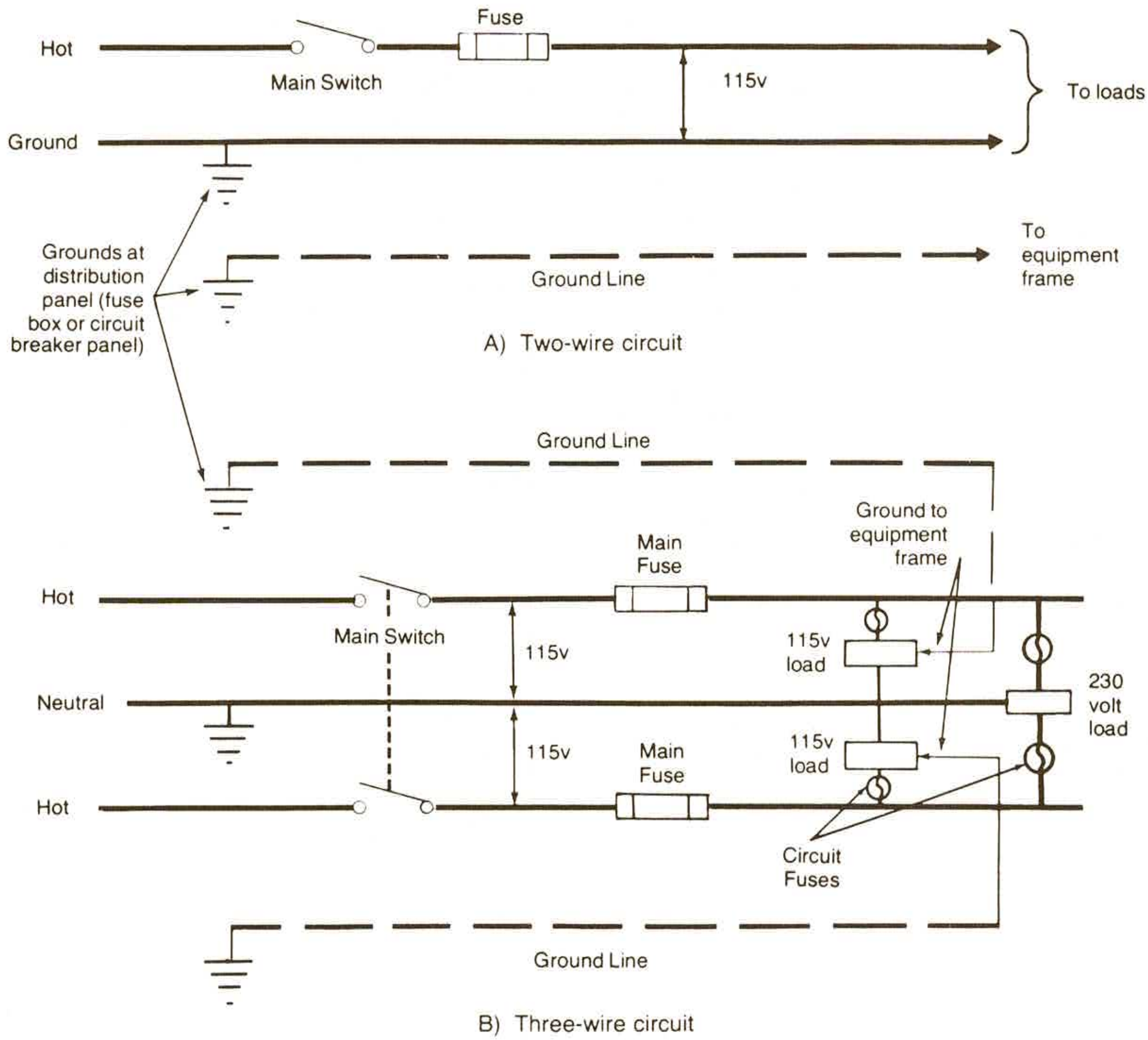


Figure 3-5 SINGLE-PHASE AC SERVICE

way into the neutral and then it can become dangerous. A neutral should be treated with the same respect as a known hot wire.

Part "B" of Figure 3-5 shows another common single phase circuit. It is also a "household" circuit which serves items requiring greater power such as ranges, clothes driers and air conditioners. This circuit represents the type of three-wire service normally entering a residence. Two of the three wires are "hot" wires and one is "neutral". The voltage potential between either of the hot wires and the neutral is the same 115 volts previously discussed. There are actually three circuits, two separate 115 volt circuits (from each of the hot wires to the neutral) and a 230 volt circuit (between the two hot wires). The neutral in a 230 volt connection serves as a ground only for safety, and is not connected as part of the power circuit. It is connected to the frame of the machine, to carry off any stray currents or short circuits resulting from failures. **Ground or neutral wires are never switched or fused.** The main advantage of the 230 volt, two hot wire circuit is that it allows each of the hot wires to carry half of the current flow. Therefore, twice the current will be handled by the same wire sizes.

3. Three-Phase Circuits

The three-phase (3ø) concept is somewhat more difficult to understand. In the case of the single-phase, three-wire circuit, two different electrical pulses are being sent down two different hot wires. After one starts, the second starts 1/120th of a second later. These pulses continue indefinitely at the same frequency and having the same "phase relationship" between the 2 wires. This can be thought of as + 115 volts and − 115 volts between the hot wires and the neutral wire.

Applying the same analysis to three-phase circuits, the 3ø generator sends a pulse down an additional (third) wire. There are now three pulses going down three wires. After the first pulse starts in the first wire, the second wire pulse starts 1/180th of a second later, and the third wire pulse starts 1/180th of a second after that. Each pulse will be 1/60th of a second long and each of the three wires will be out of phase with each other by 1/3rd of a pulse.

In large buildings, the use of many large motors running under light loads can cause these pulses to get out of phase. This causes what is known as a *low power factor.* Utility companies penalize the user for this low-power factor by increasing the charge per kilowatt of electrical power used.

Three-phase circuits are shown in Figures 3-6, 3-7 and 3-8. Three-phase circuits will have three hot wires with an equal voltage, usually 208, 230 or 460 volts between each "leg". A fourth neutral wire may be included to provide single-phase 110, 120 or 277 volt circuits for lighting and small power applications.

When voltage readings are taken with a volt meter there is no apparent way to tell the difference between 220 volt single phase circuits and 220 volt three-phase circuits. When measurements are taken, it is found that voltages do vary somewhat; that three-phase circuits are usually 220 volt, and that single-phase circuits are usually 230 volt. However, *phasing cannot be determined from measured voltage readings.*

The voltage readings of any 3 "hot legs" of three-phase equipment should essentially be the same. Variations exceeding ± 2 percent could damage equipment and should be reported (after the equipment is turned off) so that corrections can be made.

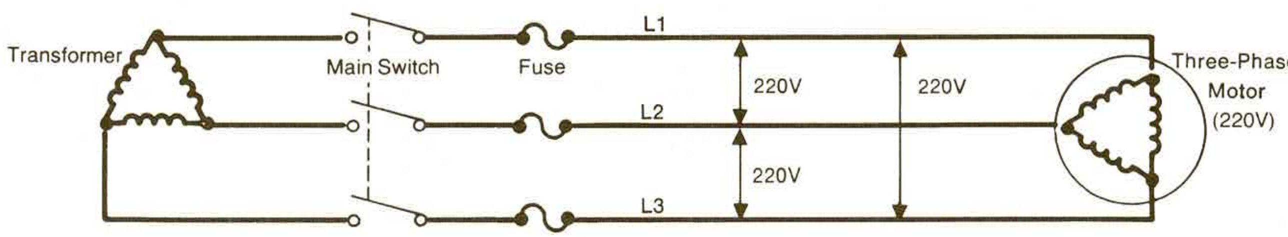


Figure 3-6 220-VOLT THREE-WIRE DELTA THREE-PHASE CIRCUIT

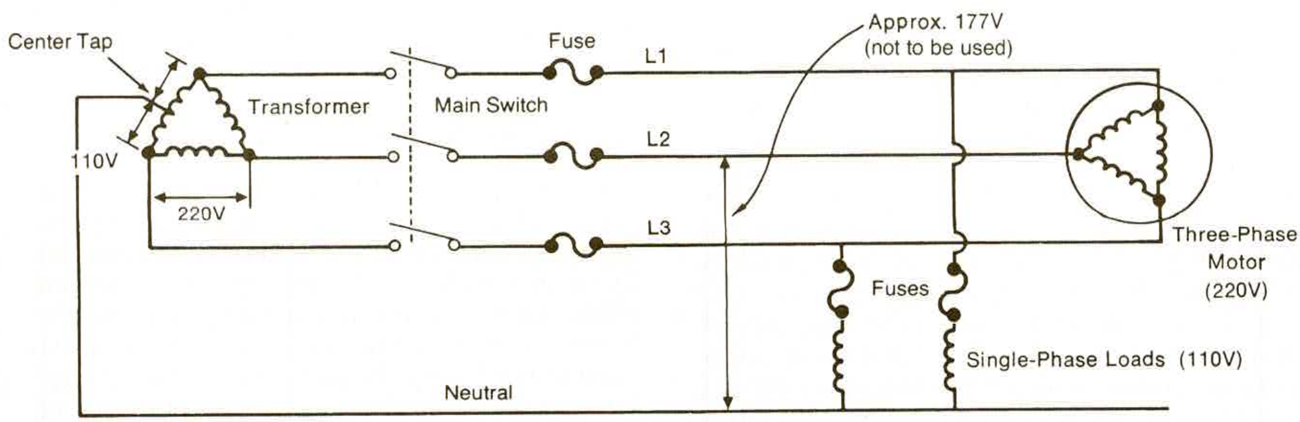


Figure 3-7 220-VOLT THREE-WIRE DELTA THREE-PHASE CIRCUIT WITH 110-VOLT SINGLE-PHASE SUPPLY

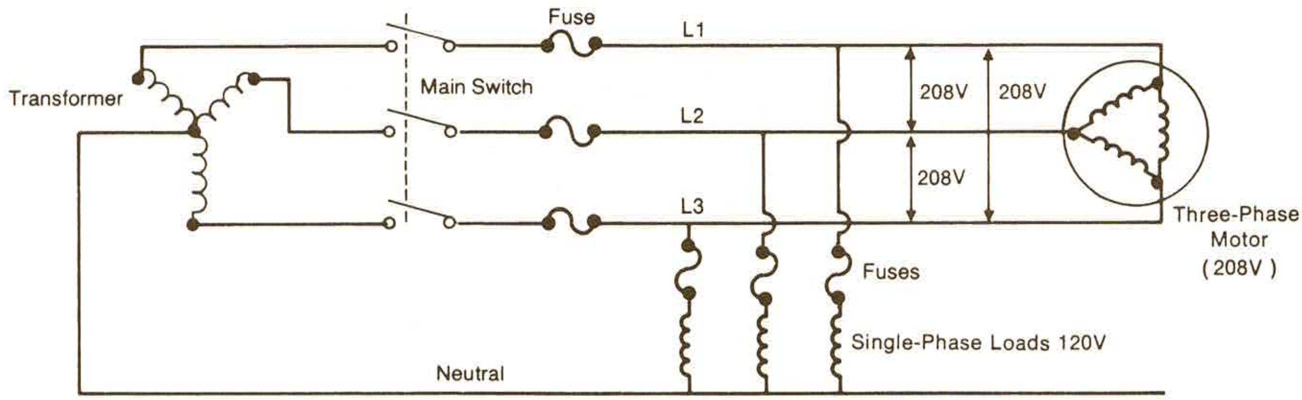


Figure 3-8 120/208-VOLT FOUR-WIRE WYE CIRCUIT

E MOTORS

1. Introduction

Motors used on HVAC equipment are designed for alternating current, except in rare cases. Most small motors will use single-phase current, while the larger motors will use three-phase current (some rural areas have only single-phase current). There are many different motor speeds, but approximately 1800 rpm and 3600 rpm are the most common. The actual speed of the motor will vary with the load imposed. "Split-phase", "capacitor start", "synchronous", "induction", "shaded-pole"—all are part of the many different types of motors that the TAB technician will need to know. The characteristics of each is important for troubleshooting, as the wrong type of motor is often used.

Motors rotating in the wrong direction is a common occurrence when a new system is started. The normal TAB procedures deal with this situation, as correct motor rotation is *vital* to the performance of the unit. The direction of the motor usually is changed in three-phase motors by switching any two of the three phase power-wires. In single-phase motors, the change of direction is accomplished by switching two of the internal motor leads that connect to the motor line terminal lugs.

A word of caution: Certain fans and most pumps will develop measurable pressures and some fluid flow when the rotation is incorrect. Rotation arrows can be found on many types of equipment. Correct rotation is obvious on some units. Flow and amperage readings also can be used to determine whether something is amiss. Whenever a piece of equipment does not perform as specified and the current flow is much lower than design, rotation is one item to be checked.

Except for some small motors, an attached name plate will supply the basic information which the TAB technician needs: full load amps, rpm, horsepower or watts capacity, voltages, line phase, and cycles. Many motor nameplates contain "starting load" amps which are quite large compared to the "full load" amps. Although this figure is not as important and is not recorded, it must be used by the system designer for electrical circuits, circuit breaker panels, and motor starting equipment. Information on special motors might have to be obtained from specification sheets or from the HVAC unit nameplate. Since voltage and amperage measurements are seldom the same as the nameplate values, the actual motor horsepower being produced can be estimated. The "no load" amps reading is difficult to obtain from close-coupled equipment where the load cannot easily be detached from the motor.

Figure 3-9 indicates the general ways in which the various motor factors are interrelated. The point at which the speed and the amperes cross, corresponds to 55% of the maximum amps and over 60% of the maximum horsepower. At this point, the speed is between 97% and 98% of the maximum synchronous speed (which is not a great change) and the efficiency curve stays fairly flat clost to 90%. The power factor also stays between 80% and 90%.

Equation 3-3

Single-Phase Circuits

$$Bhp = \frac{I \times E \times P.F. \times Eff.}{746}$$

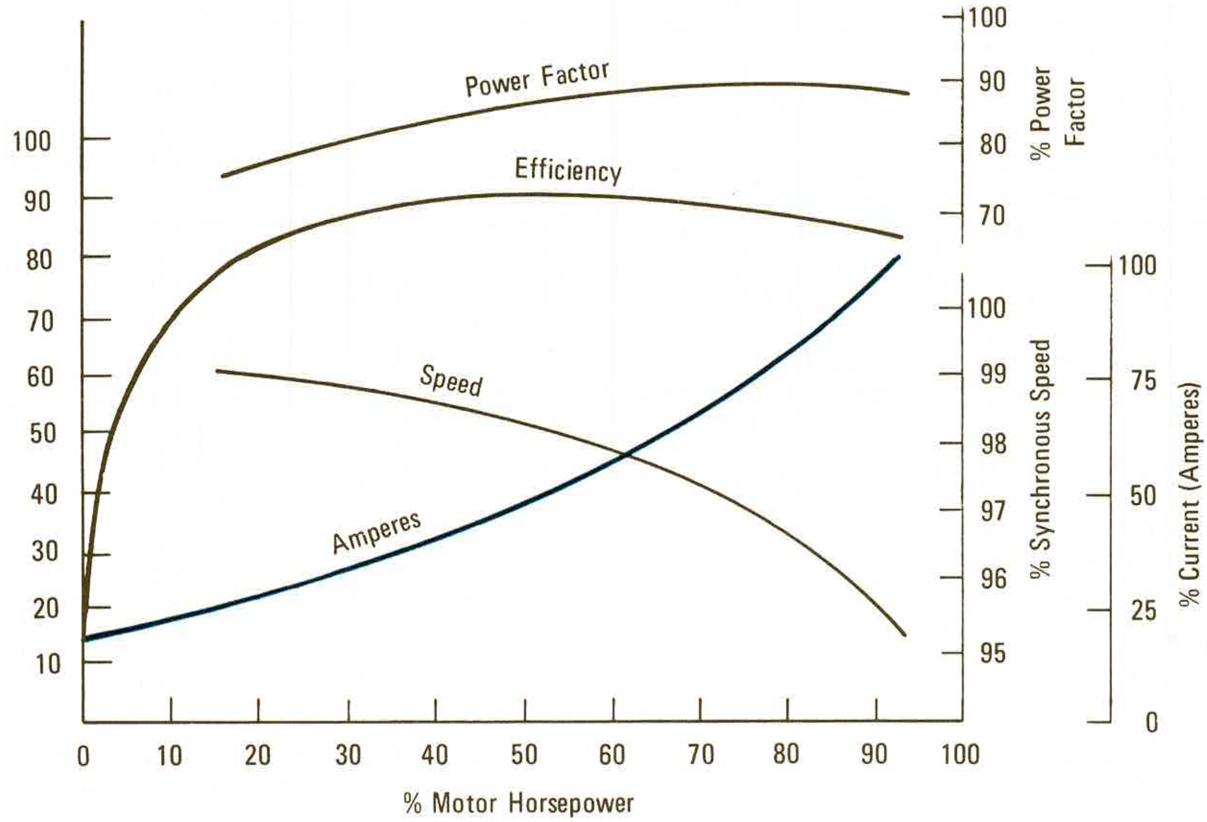


Figure 3-9 TYPICAL PERFORMANCE OF STANDARD SQUIRREL CAGE INDUCTION MOTOR

Equation 3-4

Three-Phase Circuits

$$\text{Bhp} = \frac{\text{I} \times \text{E} \times \text{P.F.} \times \text{Eff.} \times 1.73}{746}$$

Where:

Bhp = Brake horsepower

I = Amps

E = Volts

P.F. = Power factor

Eff. = Efficiency

In Equations 3-3 and 3-4, the power factor and efficiency values must be used to obtain the actual motor brake horsepower. As these values usually are difficult to obtain, a reasonable estimate can be used. Referring to Figure 3-9, the normal range of both curves is between 80% and 90%. Therefore, 80% might be used for one value and 90% for the other value to obtain a brake horsepower estimate.

Brake horsepower is calculated to verify that the proper size motor has been installed, i.e., that the installed motor is not overloaded and is operating within its service factor. It also is used to determine that the pump or fan is operating with the required efficiencies. The system designer usually has specified the total amount of power or energy that may be consumed to perform a specific function.

Example 3B

Find the brake horsepower of a three-phase fan motor rated at 10 HP which is drawing 22 amps, 20 amps, and 21 amps (on each leg) at 208 volts. (No power factors or efficiencies are available.)

Solution

Using Equation 3-4:

$$\text{Bhp} = \frac{\text{I} \times \text{E} \times \text{P.F.} \times \text{Eff.} \times 1.73}{746}$$

$$\text{Bhp} = \frac{21(\text{ave.}) \times 208 \times 0.8 \times 0.9 \times 1.73}{746}$$

$$\text{Bhp} = 7.29$$

2. Recording Motor Data

The TAB technician will be working with electric motors on most fans and pumps. Electric motors are classified according to size. Anything less than 1 HP is considered fractional HP and will usually be of single phase design. All conventional HVAC systems use motors which operate on alternating current (AC) at 60 Hertz in North America. Motors are available in various rpm, voltage, amperage, frame size, etc. The first place a TAB technician should start at a piece of HVAC equipment is the motor nameplate to record the data. The recorded data will include the following:

a. MAKE

The manufacturer producing this motor.

b. FRAME

The "National Electric Manufacturers Association: (NEMA) have standards for dimensions of motors including the shaft size, mounting dimensions, etc. This is important when replacing a motor. You must obtain a motor with the same electrical characteristics as well as the same frame number. Otherwise mounting alterations and/or drive changes may be necessary. This data is also important if the TAB technician needs to install a larger or smaller motor on a unit, so the best replacement frame available can be identified. The frame chart in Table 3-2 gives the dimensions for most motors used in HVAC work.

c. HORSEPOWER (HP)

This is the maximum horsepower or the amount of work that this motor is designed to produce. Although this motor could be overloaded and would produce more horsepower, damage and/or a shortened life can be expected. The TAB technician should never overload a motor. If he finds a motor running overloaded, he should report it immediately and possibly turn it off, depending on the amount of overload and the service.

d. REVOLUTIONS PER MINUTE

Revolutions per minute motors are designed to operate at different rpm according to their construction. The rpm on the nameplate is the rpm the motor will turn when loaded with the nameplate horsepower. With less load or more load, the rpm will vary but only a small amount. Motors are theoretically designed for a certain rpm based on the magnetic poles in the motor. However, there is always some slip within the motor. A typical motor would be designed for 1800 rpm but would be rated at 1725 rpm. The actual rpm may be more or less than 1725 but always less than 1800. Some motors are designed for multiple rpm by switching the winding connections so that two to four different speeds are available. These motors are quite common on small direct drive units.

TAB technicians will need to familiarize themselves with the wiring diagrams on these motors so that the correct speed/airflow (rpm/cfm) relationship can be obtained. Wiring diagrams are usually provided on the motor or the unit. If not, the submittal data should have instructions for this. Some installation have automatic switching provided for different modes of operation. Typical nameplate rpm's are 3450, 1775, 1160, 1050 and multiple combinations of these. Three and four speed motors with a 1050 rpm maximum are frequently used in small drive units.

e. VOLTS

The voltage rating on the nameplate is the voltage which the motor is designed to operate at. Variations of ± 10 percent should not adversely affect the operation. Many motors are designed to operate at either of two different voltages. A typical single phase motor may be designed for 115/230 volt operation. A three phase motor may be designed for 230/460 volt operation. A wiring diagram will usually be provided in the wiring junction box or its inside cover. Sometimes its part of the nameplate. It will designate how to connect the wires for either high or low voltage operation. The high and low voltage designation used on most diagrams refers to the higher and lower voltage ratings on the nameplate. The wiring connections should be verified if any of the following are observed:

- Incorrect RPM
- Abnormal sounding motor, usually a loud but different humming sound.
- Very uneven or extremely high amperage readings.
- A motor that hums loudly but won't rotate.
- Burning smell or smoke from the motor.

f. PHASE

Motors used for HVAC work will be either single phase or three phase.

g. HERTZ

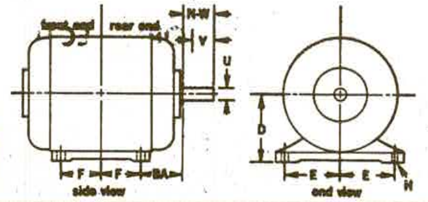
All motors will be for 60 hertz service. NOTE: Variable frequency drives are being used quite often to vary the speed of motors to match system requirements. Although the speed of the motor is varied by changing the frequency of the electricity coming into the motor, the motor itself will still be rated for 60 hertz.

h. FULL LOAD AMPS

This is the amperage that a motor will use when operating at the nameplate voltage and producing the

Table 3-2 NEMA MOTOR FRAME DIMENSION STANDARDS

Standardized motor dimensions as established by the National Electrical Manufacturers Association (NEMA) are tabulated below and apply to all base-mounted motors listed herein which carry a NEMA frame designation.



NEMA FRAME	D(*)	2E	2F	BA	H	N-W	U	V(§) Min.	Key Wide	Key Thick	Key Long	NEMA FRAME
42	2⅝	3½	1 11/16	2 1/16	9/32 slot	1⅛	⅜	—	—	21/64 flat	—	42
48	3	4¼	2¾	2½	11/32 slot	1½	½	—	—	29/64 flat	—	48
56 56H	3½	4⅞	3 3&5(‡)	2¾	11/32 slot	1⅞(†)	⅝(†)	—	3/16(†)	3/16(†)	1⅜(†)	56 56H
56HZ	3½	**	**	**	**	2¼	⅞	2	3/16	3/16	1⅜	56HZ
66	4⅛	5⅞	5	3⅛	13/32 slot	2¼	¾	—	3/16	3/16	1⅞	66
143T 145T	3½	5½	4 5	2¼	11/32 dia.	2¼	⅞	2	3/16	3/16	1⅜	143T 145T
182 184	4½	7½	4½ 5½	2¾	13/32 dia.	2¼	⅞	2	3/16	3/16	1⅜	182 184
182T 184T			4½ 5½			2¾	1⅛	2½	¼	¼	1¾	182T 184T
203# 204#	5	8	5½ 6½	3⅛	13/32 dia.	2¼	¾	2	3/16	3/16	1⅜	203# 204#
213 215	5¼	8½	5½ 7	3½	13/32 dia.	3	1⅛	2¾	¼	¼	2	213 215
213T 215T			5½ 7			3⅜	1⅜	3⅛	5/16	5/16	2⅜	213T 215T
224# 225#	5½	9	6¾ 7½	3½	13/32 dia.	3	1	2¾	¼	¼	2	224# 225#
254#	6¼	10	8¼	4¼	21/32 dia.	3⅜	1⅛	3⅛	¼	¼	2⅜	254#
254U 256U	6¼	10	8¼ 10	4¼	17/32 dia.	3¾	1⅜	3½	5/16	5/16	2¾	254U 256U
254T 256T			8¼ 10			4	1⅝	3¾	⅜	⅜	2⅞	254T 256T
284#	7	11	9½	4¾	21/32 dia.	3¾	1¼	3½	¼	¼	2¾	284#
284U 286U	7	11	9½ 11	4¾	17/32 dia.	4⅞	1⅝	4⅝	⅜	⅜	3¾	284U 286U
284T 286T			9½ 11			4⅝	1⅞	4¾	½	½	3¼	284T 286T
324# 326#	8	12½	10½ 12	5¼	21/32 dia.	4⅞	1⅝	4⅝	⅜	⅜	3¾	324# 326#
324U 326U	8	12½	10½ 12	5¼	21/32 dia.	5⅝	1⅞	5⅜	½	½	4¼	324U 326U
324T 326T			10½ 12			5¼	2⅛	5	½	½	3⅞	324T 326T
326TS			12			3¾ (▲)	1⅞ (▲)	3½ (▲)	½	½	2 (▲)	326TS
364# 364S# 364T 365# 365T	9	14	11¼ 11¼ 11¼ 12¼ 12¼	5⅞	21/32 dia.	5⅝ 3¼ 5⅞ 5⅝ 5⅞	1⅞ 1⅝ 2⅜ 1⅞ 2⅜	5⅜ 3 5⅝ 5⅜ 5⅝	½ ⅜ ⅝ ½ ⅝	½ ⅜ ⅝ ½ ⅝	4¼ 1⅞ 4¼ 4¼ 4¼	364# 364S# 364T 365# 365T
364U 365U	9	14	11¼ 12¼	5⅞	21/32 dia.	6⅜	2⅛	6⅛	½	½	5	364U 365U
404T 405T	10	16	12¼ 13¾	6⅝	13/16 dia.	7¼	2⅞	7	¾	¾	5⅝	404T 405T

(*) Dimension D will never be greater than the above values on rigid mount motors, but it may be less so that shims up to 1/16" thick may be required for coupled or geared machines.
(‡) Dayton motors designated 56H have two sets of 2F mounting holes—3" and 5".
(▲) Standard short shaft for direct-drive applications.
(#) Discontinued NEMA frame.
(**) Base of Dayton 56HZ frame motors has holes and slots to match NEMA 56, 56H, 143T and 145T mounting dimensions.
(†) Certain NEMA 56Z frame motors have ½" dia. x 1½" long shaft with 3/64" flat. These exceptions are noted in this catalog.
(§) Dimension "V" is shaft length available for coupling, pinion or pulley hub—this is a minimum value.

NEMA LETTER DESIGNATIONS FOLLOWING FRAME NUMBER

C Face mount; see next page.
H Has 2F dimension larger than same frame without H suffix.
J Face mount for jet pumps; see next pg.
K Has hub for sump pump mounting; see next page for dimensions.
M, N Flange mount for oil burner; see next pg.
T, U Integral HP motor dimension standards set by NEMA in 1964 and 1953.
Y Non-standard mounting; see manufacturer's drawing for mounting dimensions.
Z Non-standard shaft (NW, U dimensions).

nameplate horsepower. If the motor is designed for multiple voltages, the multiple amperage rating will also be given. This rating should not be exceeded, or damage and shortened motor life can be expected.

i. SERVICE FACTOR (S.F.)

The service factor is a safety allowance built into the motor that will allow the motor to operate beyond its nameplate rating. The rating shown is based on the

Table 3-3 A.C. MOTOR CHARACTERISTICS

Motor Type	HP Rating	Speed Characteristics	Full Voltage Starting Torque	Full Voltage Starting Current	Remarks
POLYPHASE					
Squirrel-cage induction	Small to large	Constant and multi-speed	High to normal	Low to normal	Most widely used for constant speed service
Wound-rotor	All	Constant or variable	High	Low	For applications requiring high starting torque and low starting current, or limited variation in speed control
Synchronous	Medium to large	Strictly constant	Normal to low	Low to normal	For constant speed service and where power factor correction is required
SINGLE-PHASE					
Capacitor-start, induction-run	Small*	Constant	High	Normal	General purpose
Capacitor-start, capacitor-run	Small*	Constant	High	Low	High efficiency
Split-phase	Fractional	Constant	Normal	Normal	Least expensive of higher starting torque types, general purpose
Permanent split-capacitor	Fractional and small integral	Constant or adjustable varying	Low	Normal	Quiet, efficient; low running current; poor starting torque
Shaded-pole	Fractional	Constant or adjustable varying	Low	—	Inexpensive; poor starting torque; least efficient; high running current

*Up to 7.5 hp.

motor operating at the design voltage and temperature. Operating a motor in the service factor range will increase the operating temperature and reduce the operating life of the motor. It is not recommended that motors be operated continuously in the service factor range.

Although other data may be given on the nameplate, NEBB requires only that data listed above on most NEBB Test Report Forms.

j. AMBIENT TEMPERATURE

Most motors are designed to operate in a space where the surrounding or ambient temperature will not exceed 40°C (105°F). If this temperature is to be exceeded, a motor with a higher insulation class should be used.

F MOTOR CONTROLS

1. Introduction

A simple "on-off" toggle switch, a safety switch, or an individual circuit breaker in an electrical power panel is *not* an overload protection device for a motor. Many "ordinary looking" toggle switches *do* contain overload protection for smaller single-phase motors. Many small motors do have built-in overload protection, and do not need additional protection. The circuit breaker only provides overload protection for the wiring circuits, but not any connected motor(s).

The electric current to a motor must be switched off and on to stop and start the motor (manually or automatically). The switching device is commonly called a *motor starter.* This is not to be confused with a "safety switch", which is a device that *must* be placed in the "off" position before any work is done on a motor or electrical equipment. This prevents the motor from accidently starting from remote control devices.

There are a large number of different types of starters, each with various advantages and limitations. In most cases a specific type of starter is required by a particular type of motor. For example, a full voltage *magnetic starter* usually is used with an induction motor. *Reduced voltage or reduced current* starters, while more expensive than a magnetic starter, often must be used with larger horsepower motors to prevent disruption (by producing large drops in line voltage) of marginally adequate power services. Many electrical utility companies have mandatory requirements for these starters above a certain horsepower (this varies with the type of equipment and voltage). The TAB technician need not concerned with all the various possible combinations of starters and motors.

The motor starter or the safety switch is the main source of access to motor terminal leads for measurement of voltage and amperage. The starter also can contain holding coils, auxiliary contacts, control transformers, and a push-button station or a "hand-off-automatic" selector switch. This last item is useful in troubleshooting. If the switch is turned to the "hand" position, the motor should run if each phase line is hot, unless there is trouble in the motor. This is because the "hand" portion of the switch bypasses the various controls in this circuit. The "automatic" portion of the switch is connected to the circuit containing auxiliary devices such as, thermostats, safety lock-outs and other external switches used to control or turn off the motor. If the motor runs on "hand", but not on automatic", one of the control or safety interlocks usually is open.

Emphasizing safety, the TAB technician can find many different voltages around starters and starter combinations (see Figure 3-10). If a remote room thermostat was added in series with the push button station of the "A" unit, the 110 volt control circuit might have been required to be 24 volts. It is not good

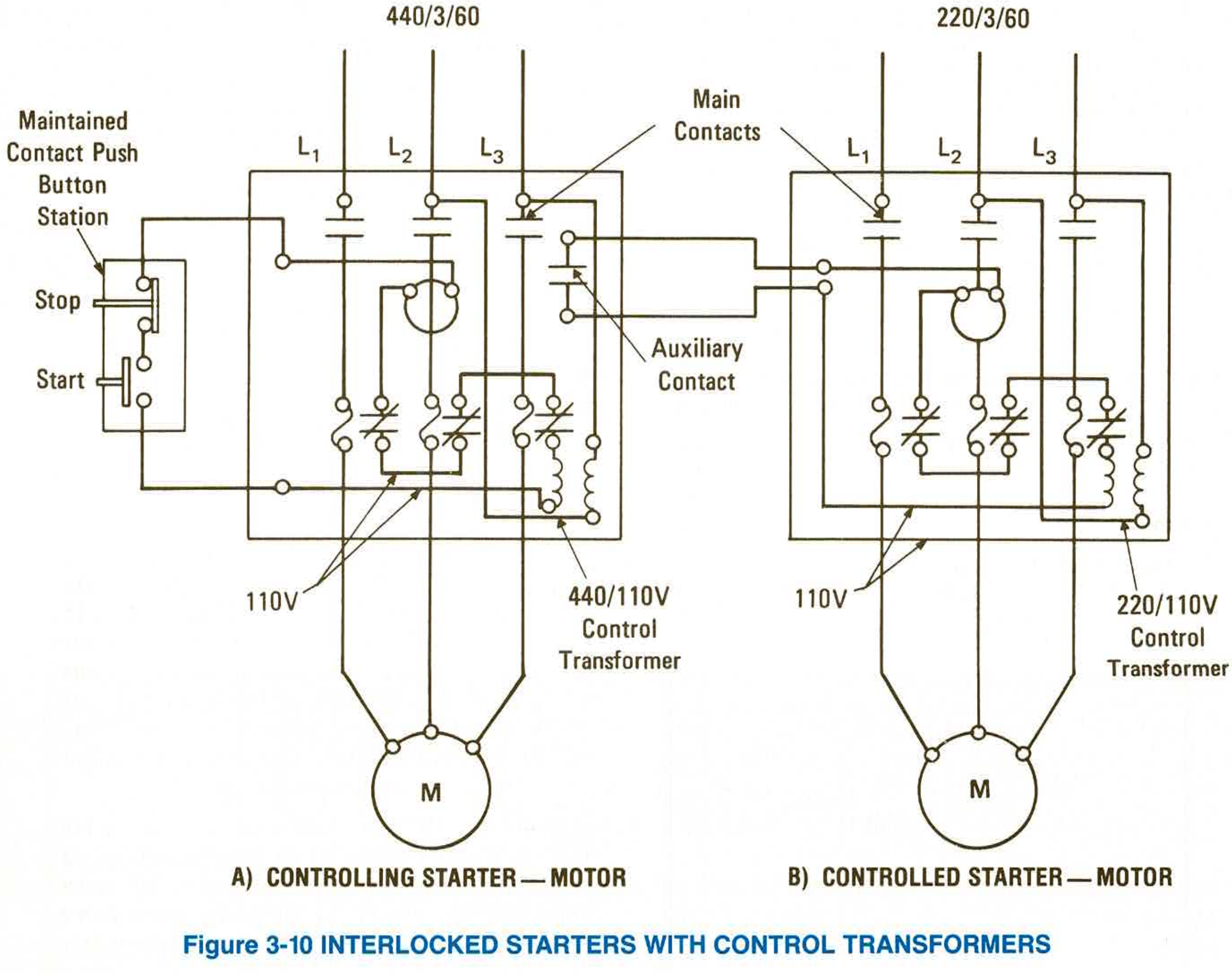


Figure 3-10 INTERLOCKED STARTERS WITH CONTROL TRANSFORMERS

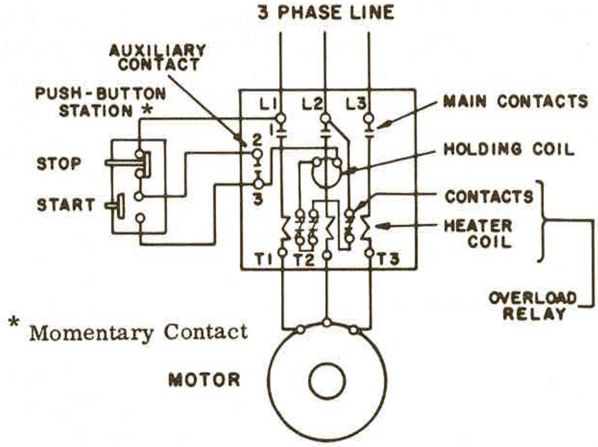


Figure 3-11 THREE-WIRE MAGNETIC STARTER CONTROL

practice to use line voltages (110 volt or higher) for control circuits, but to save dollars, 240 volt control circuits are not uncommon.

There are two basic types of bush button stations—the "maintained" contact station and the "momentary" contact station. The important point to remember is that after an interruption of the motor current with the momentary contact station, the motor will not restart until the "start" button is pushed. The differences in the wiring of the same magnetic starter can be seen by comparing Figures 3-10 and 3-11.

2. Motor/Control Operation

If current passes through a motor for any length of time, that is considerably greater than the full load rating, the windings will be overheated and damage may occur to the insulation, resulting in a burned up motor. Motors must be provided with protections to prevent this. Smaller single phase motors often have built in protection. A device is included in the motor that senses the overload and breaks the circuit and stops the motor. A manual reset button is usually provided to restart the motor or it may reset automatically. Other single phase and three phase motors must have external protection.

A motor will draw considerably more current while starting than when running at full speed and load. This current "Inrush" may be from 300 percent to 1000 percent greater than the full load amperage (FLA) rating for the motor. It will depend on the type motor, load, and how long it takes to get up to speed. The starter has to allow for the starting current inrush without tripping out, and still provide protection against exceeding the motor FLA during continuous service.

This is done by putting a small electric heater in series with the phase line and locating it by a bi-metalic, heat sensitive switch. Since the heaters will take a short time to heat up the bi-metalic switch, the motor will have time to get started and up to speed. These heater coils are available in different sizes or amperage ratings. They should be matched to the rating of the motor being protected. If the motor is overloaded, the heater coils will remain hot and the bi-metalic switch will shortly trip out, breaking the circuit to the magnetic holding coil and shuting down the motor.

When fuses are used with motors, they must be "dual element fuses". These are the only type that allow for large "inrush" currents without blowing out.

3. Heater Coil Sizing

Heater coil sizing also is affected by the ambient or surrounding temperature at the starter location. If this surrounding temperature is appreciably higher than the motor operating temperature, one size larger heaters may be required. A motor operating right at full load amperage may go into the allowable "service factor" range occasionally due to low line voltage or intermittent loads. To prevent nuisance tripping, heater coils one size larger may be required. The TAB technicians should not replace heater coils with oversize ones unless directed to by the persons responsible for the equipment. Never "jump out" or bypass heaters or fuses to run motors. With no protection, you may be held responsible for any malfunctions or damage that may occur.

On larger motors, the starter may have provisions for *reduced voltage starting*. This allows a motor to start running at a lower speed before full voltage is applied. An adjustable timer is used to determine how long the reduced voltage is applied.

The problem most likely encountered by the TAB technician will be improperly sized heater coils. Quite often, the motor will not be drawing its full load amperage, but will be tripping out the starter. All heater coils have a number on them. A chart is often located in the starter cover which will tell you how many amps various heaters are rated for. Verify that the heater coils are large enough to handle the full load amperage of the motor. If not, the proper heater will have to be installed. Of course, if the heater coils are too large, the motor will not be properly protected and this should be corrected also.

G ELECTRICAL TAB WORK

Most electrical measurements by the TAB technician normally will be limited to voltage and amperage readings. Occasional continuity and resistance readings may be required if you become involved in troubleshooting. There are some tests that the TAB technician should be able to perform and calculate to determine the performance of various components.

1. Electrical Testing Equipment

There are many types and degrees of electrical testing equipment: ammeters, voltmeters, ohmmeters, described in Chapter II; Section E. An accurate clamp-on volt-ammeter is the *only* electrical testing instrument required by the Certified NEBB TAB Procedures. These devices have several scales for different ranges of volts and amps. Amperage is measured by clamping the probes around a *single* conductor. Figure 3-12 shows a method which doubles the amperage reading when there is a low current flow. This allows a more accurate reading (if there is enough slack in the wire), but the amperage reading *must* be divided by two.

Voltage readings are taken by using two test leads connected to the instrument which contain probes or clamps on the ends. Test measurements should always be taken with the highest scale first, to avoid damage to the instrument by excessive voltage or amperage. Gradually select each successively lower scale until a mid-scale reading (or as close as possible) is obtained. Test instruments have the greatest accuracy in mid-scale readings.

2. Test Measurements

One of the functions of the TAB technician is to measure and record the actual "running" voltages and the amperages of motors, driving fans, pumps, etc. This information is necessary and valuable, but it becomes critical when troubleshooting. Almost all motor data measurements by the TAB technician will be taken within the motor starter. If line phase terminals need to be changed to correct for polyphase motor rotation, the changes can be made either in the starter or at the motor terminal connections. Amperage and voltage readings must be taken "live" in the vicinity of voltages up to 480 volts. EXTREME CARE MUST BE EXERCISED WHEN USING THE TEST INSTRUMENT INSIDE A "LIVE" ELECTRICAL JUNCTION BOX OR STARTER. One wrong move could cause serious burns or electrocution.

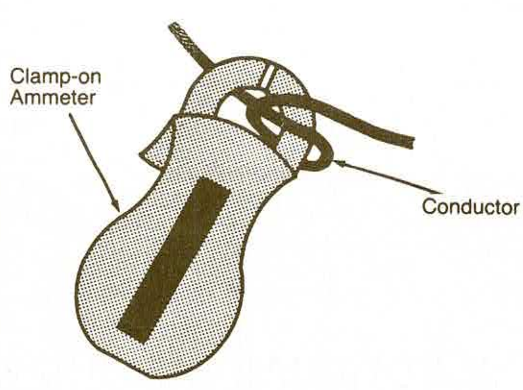


Figure 3-12 LOOPED CONDUCTOR DOUBLES AMMETER READING

To take amperage readings (see Figure 3-10), the clamp-on ammeter would be placed around L1, then L2, then L3 (one line at a time). The amperages, which are recorded, seldom have the same values, but should be within 10 percent of one another. The two voltage test probes are placed across the different pairs: L1/L2, L1/L3, and L2/L3. Again, the voltages should be recorded and the values should be within 10 percent of each other. When the voltages or amperages are not within 10 percent of one another, there is an indication of trouble. If the voltage and amperage readings of one phase line is zero, "single phasing" could be occurring, and the equipment must be shut down *immediately.*

Single phasing results when one phase of a polyphase circuit is broken or open. Motors can continue to run under this condition with a lower power output and possible overheating. Motors usually will not start under this condition, but will produce a loud "hum".

a. BRAKE HORSEPOWER

The following equations can be used to obtain an accurate (but not exact) brake horsepower by measuring motor amperages and voltages under no load and full load conditions.

Equation 3-5

$$\text{Actual F.L. amps} = \frac{\text{F.L. amps}^* \times \text{voltage}^*}{\text{Actual voltage}}$$

Equation 3-6

$$\text{Bhp} = \text{HP}^* \times \frac{(\text{Motor operating amps}) - (\text{No load amps} \times 0.5)}{(\text{Actual F.L. amps}) - (\text{No load amps} \times 0.5)}$$

*Nameplate ratings

Example 3C

A fan has a 3 HP, 220 volt, 3 phase motor that actually draws 6.2 amps at 210 volts. The full load amperage shown on the nameplate is 8.16 amps and the "no load" measurement is 4.7 amps. What is the approximate fan brake horsepower?

Solution

(a) Using Equation 3-5:

$$\text{Actual F.L. amps} = \frac{8.16\text{ A} \times 220\text{V}}{210\text{V}}$$

$$\text{Actual F.L. amps} = 8.55\text{ amps}$$

(b) Using Equation 3-6:

$$\text{Bhp} = 3 \times \frac{(6.2) - (4.7 \times 0.5)}{(8.55) - (4.7 \times 0.5)}$$

$$\text{Bhp} = 3 \times \frac{6.2 - 2.35}{8.55 - 2.35}$$

$$\text{Bhp} = 3 \times \frac{3.85}{6.20} = 1.86$$

On direct drive fans and pumps, it is impossible to obtain the "no load" amperage readings. Equation 3-7 may be used to obtain an approximate equipment brake horsepower.

Equation 3-7

$$\text{Bhp} = \frac{(\text{Motor operating amps}) \times \text{HP*}}{\text{F.L. Amps*}}$$

Example 3D

Calculate the Bhp using Equation 3-7 from data in Example 3C and compare the results.

Solution

$$\text{Bhp} = \frac{6.2\text{ A} \times 3\text{ HP}}{8.16\text{ A}}$$

$$\text{Bhp} = 2.28\text{ (approx.)}$$

Equation 3-7 gives a higher approximate Bhp than using Equation 3-5 and 3-6 (2.28 vs 1.86)

Example 3E

Motors are rated to deliver a specific HP at an input of a specific voltage and amperage. If a 5 HP rated motor is using its full rated amperage of 13.2 amps at its rated voltage of 220 volts, it will deliver 5 HP. But suppose the actual voltage is 230 volts, then what would the amperage be for delivering the same 5 HP?

Solution

Using Equation 3-5:

$$\text{Actual F.L. amps} = \frac{13.2\text{ A} \times 220\text{V}}{230\text{V}}$$

$$\text{Actual F.L. amps} = 12.63\text{ amps}$$

Therefore with a higher voltage, the motor will deliver 5 HP with less amperage.

b. KILOWATT/HEAT OUTPUT

There are occasions when testing an electric heating coil or furnace, that you will need to calculate the heat output (Btu). Since one kilowatt (kW) equals 3414 Btu, the Btu being delivered by an electric heating coil can be determined by using Equations 3-2 (a or b).

Example 3F

A 5 kW heating coil is rated at 18.0 amps using a 230 volt, single-phase service. Calculate the heat output when the voltage is measured at 235 volts and the amperage at 18.1 amps. If the airflow across the coil is 670 cfm, calculate the temperature rise.

Solution

$$P = E \times I = 235\text{V} \times 18.1\text{ A}$$

$$P = 4253.5\text{ Watts} = 4.25\text{ kW}$$

$$\text{Heat output} = 4.25\text{ kW} \times 3414 = 14{,}509.5\text{ Btuh}$$

$$Q = 1.08 \times \text{cfm} \times \Delta t;\ \Delta t = \frac{Q}{1.08 \times \text{cfm}}$$

$$\Delta t = \frac{14{,}509.5}{1.08 \times 670} = 20.05°\text{F}$$

H TRANSFORMERS

Going back to the transformer diagram illustrated in Figure 3-1, for the voltage reduction indicated, the number of turns shown on the primary side should be twice the number of turns shown on the secondary side. Voltage is "transformed" by the transformer stepping down the voltage to one half the original voltage. By swapping the primary and secondary connections, this same transformer could step up the voltage from 440 volts to 880 volts. The ballasts in fluorescent lights in buildings step up the voltages from 115, 220, or 277 volts to voltages near 2500 volts, the required voltage to produce light in the tubes.

The function of the center tap of the transformer is illustrated in Figure 3-13. If a 220 volt difference exists between the legs of the secondary side, it is logical that a 110 volt difference would exist between one leg and a center tap. Most single-phase residential transformers have high voltages on the primary side, but the secondary voltages use a "center tap" (the ground) to furnish two 110 volt circuits along with the 220 volt power (Figure 3-5). This size transformer, which looks like a large can, is usually attached to a pole near the residence. It can supply power to several residences or buildings, or just to a single building.

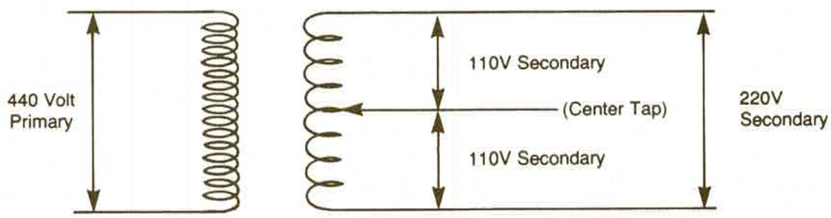


Figure 3-13 TRANSFORMER WITH TAPPED SECONDARY

Larger transformers are ground-mounted, and if outside, are usually in a large metal housing. These transformers can be either single-phase or three-phase. The "tap" (or neutral) cannot be used with hot line "L2" for 110 volt single-phase loads (Figure 3-7) from a three-wire, 220 volt circuit. However, all three hot line or phase wires may be used for 120 volt loads in four-wire, 208 volt circuits (Figure 3-8).

Many small transformers, such as 110V/24V or 220V/24V, are used in control circuits for HVAC equipment. When working in electric panels, be aware that transformers often are powered from external electrical circuits and may be "hot," even though the main switch has been pulled for the equipment.

I POWER FACTORS

A low or unsatisfactory *power factor* (pf) can be an indication of overall electrical system inefficiency. It is caused by the use of inductive (magnetic) devices, particularly induction motors (55 to 90% pf) since there are so many of them in use and since they are usually run at less than full load. Other devices responsible for the low power factor include:

- non-power factor corrected fluorescent and HID lighting fixture ballasts (40 to 80% pf)
- arc welders (50 to 70% pf)
- solenoids (20 to 50% pf)
- induction heating equipment (60 to 90% pf)
- lifting magnets (20 to 50% pf)
- small "dry-pack" transformers (30 to 95% pf).

It takes special meters (as shown in Figure 3-14) or recording equipment to measure the data needed to calculate power factors and the correction needed. The work is not part of the NEBB TAB procedures.

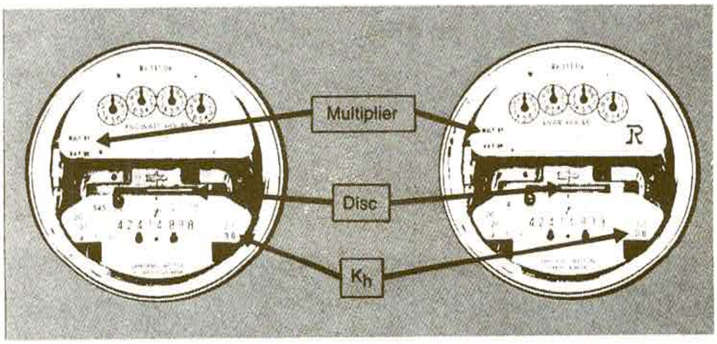


Figure 3-14 UTILITY COMPANY METERS (WATTHOUR AND VAR-HOUR)

Table 3-4 APPROXIMATE MOTOR AMPERAGES

Motor HP	Recommended Starter Size Three Phase		Polyphase A-C Squirrel Cage-and Wound Rotor (Induction Type)						Recomd. Starter Size Single Phase	Single Phase A-C		
			230V		460V		600V					
	230V	460V	3 PH	2 PH 4 Wire	3 PH	2 PH 4 Wire	3 PH	2 PH 4 Wire	230V	115V	230V	460V
1/6			...	....	...	......	...	...		3.2	1.6	...
1/4			0.96	0.83	0.48	0.42	...	...		4.6	2.3	...
1/2			1.68	1.47	0.84	0.74	0.8	0.8		7.4	3.7	...
3/4			2.33	2.02	1.17	1.01	1.1	1.0		10.2	5.1	...
1			3.05	2.64	1.53	1.32	1.4	1.3		13.0	6.5	...
1½			4.28	3.70	2.14	1.85	2.0	1.8		18.4	9.2	...
2			5.76	4.98	2.88	2.49	2.6	2.2	0	24.0	12.0	...
3	0		8.29	7.17	4.14	3.59	4.0	3.2		....	17	...
5		0	13.2	11.4	6.60	5.71	6	6	1	....	28	...
7½	1		19.3	16.7	9.7	8.4	9	8		....	40	21
10		1	25.2	21.8	12.6	10.9	11	10	2	....	50	26
15	2		38.1	33	19.1	16.5	16	14	3	....	....	...
20			50.5	43.7	25.3	21.9	21	18		....	....	...
25		2	62.7	54.3	31.3	27.1	26	22		....	....	...
30	3		72.8	63	36.4	31.5	31	27		....	....	...
40			98	85	49.0	42.5	41	35		....	....	...
50	4	3	121	105	60.5	52.4	50	43		....	....	...
60			143	124	71.5	62	60	52				
75			178	154	89.0	77	74	63				
100	5	4	186	202	93.2	100	100	86				
125			230	250	115	125	120	104				
150			346	300	173	150	148	128				
200	6	5	460	398	230	199	168	144				

NEBB

CHAPTER 4 AIR SYSTEMS

A INTRODUCTION

In this chapter, the various components of air systems will be presented, along with their purpose and how they operate. Starting with the supply air flow, which is the prime mover in any air system, and ending with terminal devices or air outlets, the many basic system designs and how they are used will be covered.

The TAB technician does not need to get involved in too much theory or in other items that are not encountered in everyday TAB work. This chapter concentrates on what the TAB technician really will find and need to know about the air side of HVAC systems while performing TAB work. Those interested in an indepth study of HVAC air systems should read SMACNA and ASHRAE publications on the subject, and the NEBB "Environmental Systems Technology" textbook.

B FANS

1. Types of Fans

There are three main categories of fans: a) centrifugal, b) axial and c) special designs. In the centrifugal fan classification, there are four types: a) forward curved, b) backward curved or backward inclined, c) air foil and d) radial. (Figure 4-1) Of these four, the radial type seldom is used in the HVAC industry. Forward curved fans are used primarily in low pressure HVAC systems in equipment such as furnaces and packaged air conditioners involving small horsepower.

Airfoil and backward curved fans are used with large horsepower motors on large HVAC systems, which generally have been built-up from individual components on the jobsite. This type of fan is the most efficient when used with higher horsepowers, and has the added advantage of being nonoverloading, that is, when approaching free delivery, the horsepower decreases from that of the design maximum. If these airfoil and backward curved fans are correctly sized, the motors will not overload regardless of the airflow quantity that is established.

Under the axial flow category, there are three types: propeller, tube axial and vane axial fans. The most well-known axial fan is usually used in low pressure exhaust applications as a propeller fan. This type of fan does not work well when ductwork is added, particularly if the ductwork has been added on the suction side. The airflow (cfm) drops rapidly with a *slight* increase in static pressure, although some are designed to handle small amounts of resistance. Therefore, it is very difficult to test these fans for static pressure. If there is no ductwork, there is no differential static pressure. Even airflow is difficult to measure, but it can be done with a rotating vane anemometer, if the application is a wall-type propeller fan (with or without a discharge louver). Airflow of rooftop propeller fans in low silhouette-type housings cannot be measured from the roof side, and often are difficult to measure because of low velocities on the inlet or room side.

Tube axial fans and vane axial fans are found most often in medium and high pressure applications respectively. The vane axial fan has been used in high

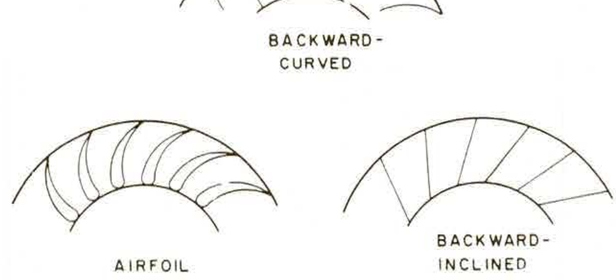


Figure 4-1 BASIC FAN BLADE CONFIGURATIONS

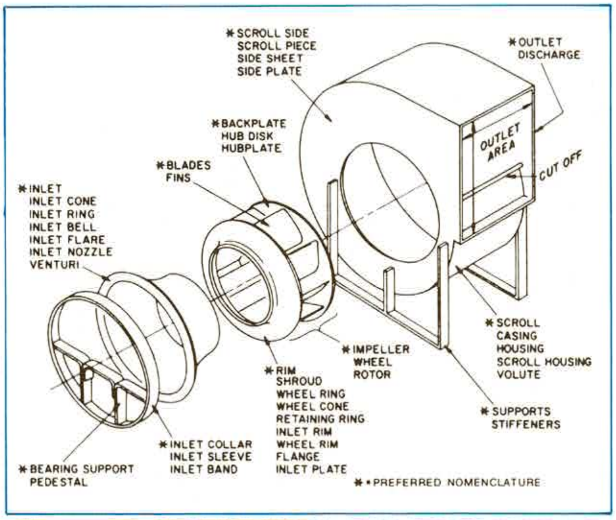


Figure 4-2 CENTRIFUGAL FAN COMPONENTS

pressure type systems, such as the variable volume system which uses mixing boxes for terminal control. Tube axial fans often have been used as return air fans for these same systems. Certain manufacturers provide volume control to these fans using variable pitch blades.

a. FORWARD CURVED BLADES (FC)

The *forward curved blade* fan usually is used to produce large air volumes at relatively low static pressures. Their main advantages are low cost and low operational speeds. Disadvantages include their inability to develop high static pressures and a curve that will usually overload the motor if the static pressure gets too low. The static pressure (SP) is usually around twenty percent of the total pressure (TP) with about eighty percent being velocity pressure (V_p).

b. BACKWARD INCLINED BLADES (BI)

The *backward inclined blade* fan will develop considerably more static pressure than a forward curved fan, but they run about twice the speed of FC fans. The static pressure (SP) is usually about seventy percent of the total pressure (TP) with about thirty percent being velocity pressure (V_p). Advantages include stable operation and a "non-overloading" fan curve. Disadvantages include a higher operating rpm which requires larger bearings and generally heavier construction.

c. AIRFOIL BLADES

Fans with *airfoil blades* are a refinement of the backward inclined blade fan that results in higher static efficiency and lower noise levels.

d. RADIAL BLADES

Fans with *radial blades* are used primarily for material handling. They will develop high static pressures but are noisy. Their blade configuration makes them good for waste collection, etc.

e. TUBULAR CENTRIFUGAL

Tubular centrifugal fans (Figure 4-3) generally consist of a single width airfoil wheel arranged in a cylinder to discharge air radially against the inside of the cylinder. Air is then deflected parallel with the fan shaft to provide straight-through flow. Vanes are used to recover static pressure and to straighten the air flow. The main advantage is space savings. They have a lower static efficiency and a higher noise level than a conventional airfoil or backwards inclined fan.

f. PROPELLER

Propeller fans are well-suited for moving air at very low or no static pressure. They are best located in partitions or walls (rather than in duct systems) to move large quantities of air from open spaces.

g. VANEAXIAL AND TUBEAXIAL

Tubeaxial and *vaneaxial* fans (Figure 4-4) are simply propeller fans mounted in a cylinder and are similar

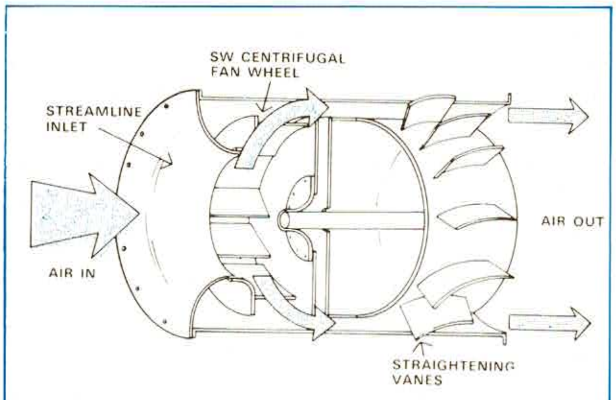


Figure 4-3 TUBULAR CENTRIFUGAL FAN

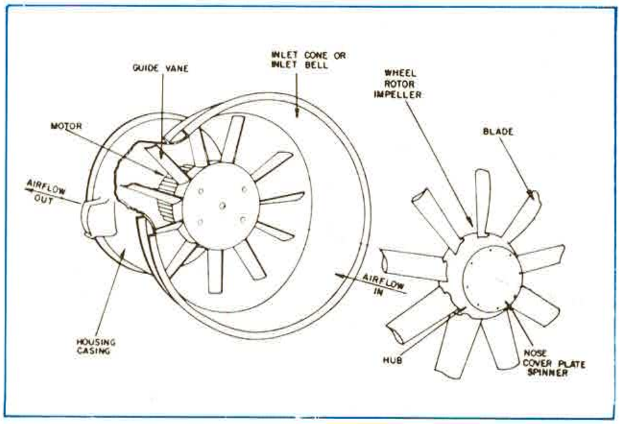


Figure 4-4 AXIAL FAN COMPONENTS

Table 4-1 TYPICAL FAN RATING TABLE

VOL	OUT VEL	VEL PRES	0.125	S.P.	0.250	S.P.	0.375	S.P.	0.500	S.P.	0.625	S.P.	0.750	S.P.	0.875	S.P.	1.000	S.P.	1.250	S.P.	1.500	S.P.	1.750	S.P.	2.000	S.P.
CFM	FPM	IN. H₂O	RPM	BHP	RPM	BHP	RPM	BHP	RPM	BHP	RPM	BHP	RPM	BHP	RPM	BHP	RPM	BHP	RPM	BHP	RPM	BHP	RPM	BHP	RPM	BHP
2264	800	0.04	398	0.10	456	0.15	507	0.21	557	0.26	608	0.32	656	0.40	703	0.47	747	0.55								
2547	900	0.05	434	0.13	487	0.19	536	0.25	578	0.30	624	0.37	669	0.44	712	0.51	755	0.60	835	0.78						
2830	1000	0.06	472	0.17	519	0.23	565	0.29	608	0.36	645	0.42	686	0.49	727	0.57	767	0.65	843	0.83	916	1.03				
3113	1100	0.08	510	0.21	552	0.27	595	0.34	636	0.42	675	0.49	708	0.56	745	0.63	782	0.71	855	0.89	924	1.10	991	1.31		
3396	1200	0.09	549	0.26	587	0.33	627	0.40	666	0.48	702	0.56	738	0.64	788	0.71	802	0.79	870	0.97	936	1.17	999	1.39	1062	1.63
3679	1300	0.11	589	0.32	624	0.39	661	0.47	697	0.55	731	0.64	765	0.73	798	0.81	825	0.89	888	1.07	950	1.26	1012	1.48	1070	1.71
3962	1400	0.12	629	0.39	662	0.46	695	0.54	729	0.63	762	0.72	794	0.81	826	0.91	856	1.01	909	1.17	967	1.37	1026	1.58	1083	1.82
4245	1500	0.14	668	0.46	700	0.54	730	0.62	762	0.72	794	0.81	825	0.91	854	1.01	884	1.12	936	1.30	989	1.50	1043	1.71	1097	1.94
4528	1600	0.16	709	0.55	739	0.63	767	0.72	796	0.81	827	0.91	856	1.02	884	1.13	912	1.23	967	1.46	1013	1.64	1063	1.86	1114	2.09
4811	1700	0.18	749	0.65	778	0.74	805	0.83	832	0.92	860	1.03	888	1.14	915	1.25	942	1.36	994	1.59	1044	1.82	1087	2.01	1134	2.25
5094	1800	0.20	790	0.75	818	0.85	843	0.95	868	1.05	894	1.15	921	1.26	948	1.38	973	1.50	1023	1.74	1073	1.99	1115	2.21	1157	2.43
5377	1900	0.23	830	0.88	857	0.98	882	1.08	906	1.19	930	1.29	955	1.40	980	1.53	1005	1.65	1053	1.90	1100	2.16	1146	2.42	1185	2.64
5660	2000	0.25	872	1.01	897	1.12	921	1.23	944	1.33	966	1.44	989	1.56	1014	1.68	1038	1.81	1084	2.08	1129	2.34	1173	2.61	1217	2.89
5943	2100	0.27	913	1.16	937	1.27	960	1.39	982	1.50	1004	1.61	1025	1.73	1048	1.85	1071	1.99	1116	2.26	1160	2.54	1202	2.82	1245	3.12
6226	2200	0.30	954	1.32	977	1.44	999	1.56	1021	1.68	1042	1.80	1062	1.91	1083	2.04	1104	2.17	1148	2.46	1191	2.75	1231	3.04	1272	3.34
6509	2300	0.33	995	1.50	1017	1.62	1039	1.75	1059	1.87	1080	1.99	1100	2.12	1119	2.24	1139	2.38	1181	2.67	1222	2.97	1262	3.28	1301	3.58
6792	2400	0.36	1037	1.70	1057	1.82	1079	1.95	1099	2.08	1118	2.21	1137	2.34	1156	2.47	1175	2.60	1215	2.90	1255	3.20	1293	3.52	1331	3.84
7358	2600	0.42	1120	2.13	1139	2.26	1159	2.40	1178	2.55	1196	2.68	1214	2.82	1231	2.97	1248	3.10	1284	3.40	1321	3.72	1358	4.06	1393	4.40
7924	2800	0.49	1204	2.64	1221	2.78	1239	2.93	1257	3.08	1274	3.23	1291	3.38	1308	3.53	1324	3.69	1356	3.99	1389	4.32	1424	4.67	1458	5.03
8490	3000	0.56	1287	3.23	1303	3.38	1320	3.53	1337	3.70	1353	3.86	1370	4.02	1385	4.18	1401	4.34	1431	4.67	1461	5.00	1492	5.35	1525	5.73
9056	3200	0.64	1371	3.90	1386	4.05	1401	4.21	1417	4.39	1433	4.56	1448	4.74	1464	4.91	1478	5.08	1507	5.43	1535	5.77	1563	6.13	1593	6.51
9622	3400	0.72	1455	4.66	1469	4.82	1483	4.99	1498	5.16	1513	5.35	1528	5.54	1542	5.72	1556	5.91	1583	6.27	1611	6.64	1637	7.00	1664	7.39
10188	3600	0.81	1539	5.51	1552	5.68	1566	5.86	1579	6.04	1594	6.24	1608	6.43	1621	6.63	1635	6.82	1661	7.20	1687	7.59	1713	7.99	1737	8.37
10754	3800	0.90	1623	6.46	1636	6.64	1648	6.82	1661	7.01	1674	7.21	1688	7.42	1701	7.63	1714	7.84	1740	8.25	1764	8.65	1788	9.06	1813	9.48
11320	4000	1.00	1707	7.52	1719	7.70	1731	7.89	1743	8.09	1755	8.29	1769	8.52	1781	8.74	1794	8.95	1818	9.39	1841	9.80	1865	10.24	1888	10.68

Pressure class limits:

Class	Maximum RPM
I	1550
II	2140

except for vane-type straighteners on the vaneaxial. These vanes remove much of the swirl from the air and improve the efficiency. Thus, a vaneaxial fan is more efficient than a tubeaxial and can reach higher pressures. Note that with axial fans the Bhp is at maximum at the blocktight static pressure. With centrifugal fans, the Bhp is at minimum at blocktight static pressure.

Tubeaxial and vaneaxial fans are generally used for handling large volumes of air at low static pressures.

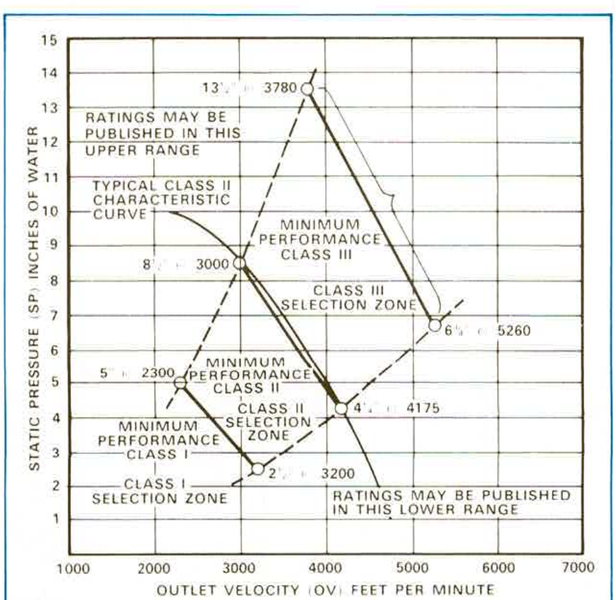


Figure 4-5 FAN CLASS STANDARDS (SW BI FANS)

Advantages of tubeaxial and vaneaxial flow fans are the reduced size and weight and the straight-through air flow which frequently eliminates elbows in the ductwork. Disadvantages include their lower efficiency and higher noise levels than centrifugal fans.

2. Fan Classifications and Arrangements

Fans are built to industry standards that determine certain construction features such as the thickness of the metal, types and amount of bracing, shaft and bearing size, etc. (Figure 4-5). *Class I* is standard for relatively light duty operation. When higher cfm, static pressures and fan rpm are needed, a *Class II or III* fan would be used. The fan classification usually is shown on the manufacturer's performance tables.

The TAB technician usually has little control over what class fan has been provided, but he will need to be aware of this if it becomes necessary to speed up the fan. *Caution: Always check with the published ratings of the fan equipment to make sure that revised operating conditions do not require a different class fan.* Often this type of change also could change the pressure classification of part or all of a duct system. AMCA has developed standard fan drive arrangements for various bearing and drive locations (Figure 4-6) for all centrifugal blower fans.

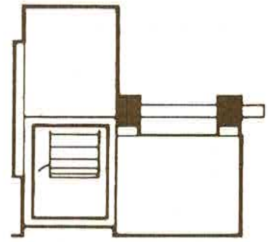
In-line fans are designated in much the same way as standard centrifugal type fans. Standard arrangements are:

SW – Single Width DW – Double Width
SI – Single Inlet DI – Double Inlet

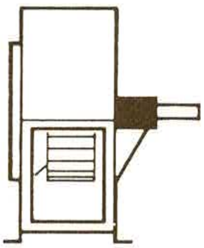
Arrangements 1, 3, 7 and 8 are also available with bearings mounted on pedestals or base set independent of the fan housing.

For designation of rotation and discharge, see Figure 3-16

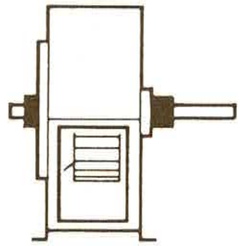
For motor position, belt or chain drive, see Figure 3-15



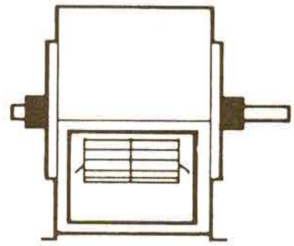
ARR. 1 SWSI For belt drive or direct connection. Impeller overhung. Two bearings on base.



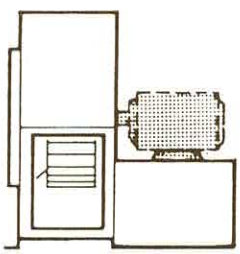
ARR. 2 SWSI For belt drive or direct connection. Impeller overhung. Bearings in bracket supported by fan housing.



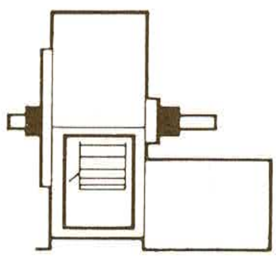
ARR. 3 SWSI For belt drive or direct connection. One bearing on each side and supported by fan housing.



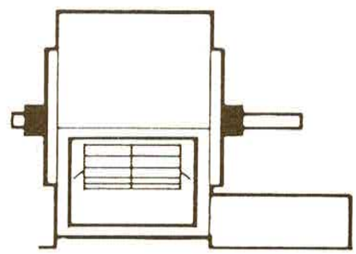
ARR. 3 DWDI For belt drive or direct connection. One bearing on each side and supported by fan housing.



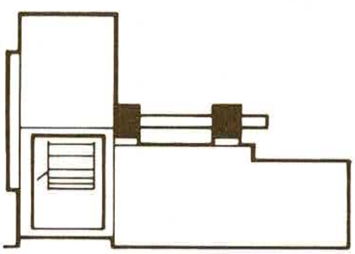
ARR. 4 SWSI For direct drive. Impeller overhung on prime mover shaft. No bearings on fan. Prime mover base mounted or integrally directly connected.



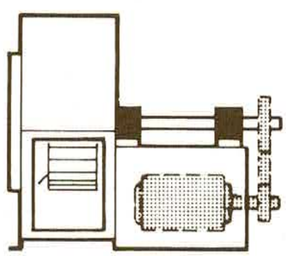
ARR. 7 SWSI For belt drive or direct connection. Arrangement 3 plus base for prime mover.



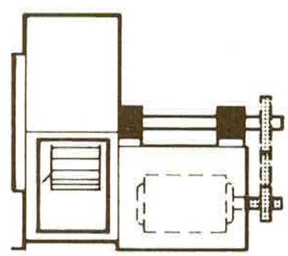
ARR. 7 DWDI For belt drive or direct connection. Arrangement 3 plus base for prime mover.



ARR. 8 SWSI For belt drive or direct connection. Arrangement 1 plus extended base for prime mover.



ARR. 9 SWSI For belt drive. Impeller overhung, two bearings, with prime mover outside base.



ARR. 10 SWSI For belt drive. Impeller overhung, two bearings, with prime mover inside base.

Figure 4-6 DRIVE ARRANGEMENTS FOR CENTRIFUGAL FANS

Arrangement 1—belt drive with motor mounted independent of fan casing—typically used for motors too large for fan casing (Figure 4-7).

Arrangement 4—direct drive with wheel overhung on motor shaft (Figure 4-8).

Arrangement 9—belt drive with motor located on periphery of casing in one of eight standard locations designated by the letters beginning with A at the top and proceeding clockwise at eight equal intervals through the letter H when viewing the fan from the discharge (Figure 4-9).

Vertical units are designated as either upblast or downblast and generally are available only in Arrangements 4 and 9. Motor location is specified at W, X, Y and Z. This motor location is always determined by facing the fan drive sheave. It is independent of the discharge or rotation (Figure 4-10).

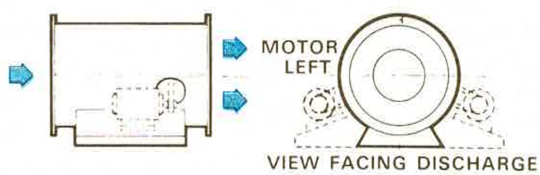


Figure 4-7 ARRANGEMENT 1—IN-LINE FANS

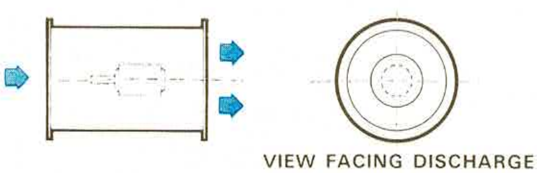


Figure 4-8 ARRANGEMENT 4—IN-LINE FANS

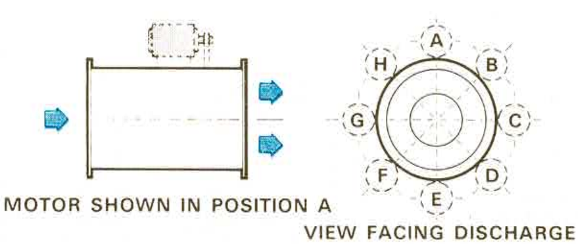
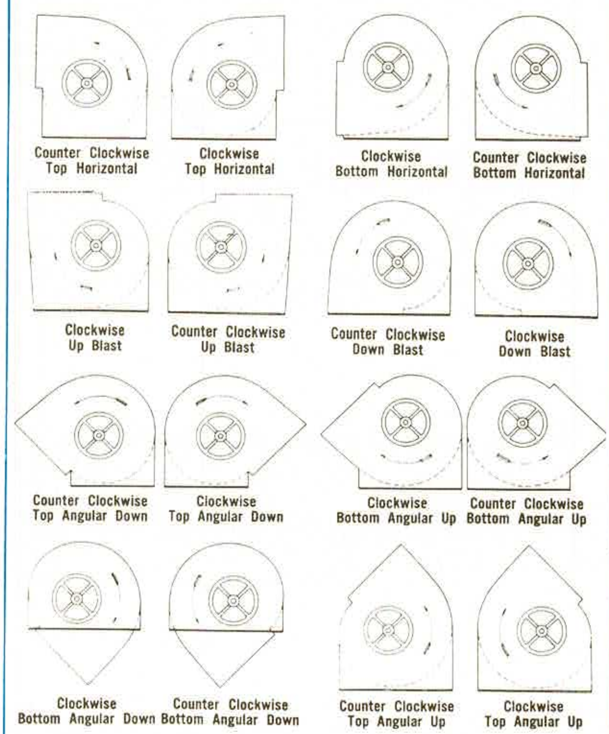


Figure 4-9—ARRANGEMENT 9—IN-LINE FANS

Figure 4-10 CENTRIFUGAL FAN MOTOR LOCATIONS



Direction of rotation is determined from the drive side. On single inlet fans, drive side is considered as opposite inlet, regardless of actual drive location.

Figure 4-11 DIRECTION OF ROTATION AND DISCHARGE

3. Fan Operation

To better understand the operation of fans, some definitions will be reviewed:

a. FAN AIR VOLUME

The cubic feet per minute (cfm) of air produced by a fan in HVAC systems is independent of the air density, as *a fan is a "constant volume machine"*.

Cfm: cubic feet per minute of air handled by a fan at any density.

Scfm: cubic feet per minute of *standard air* (0.075 lb./cu.ft.) handled by a fan.

b. FAN TOTAL PRESSURE (TP)

Fan total pressure (Figure 4-12) is the difference between the total pressure measured at the fan outlet and the total pressure measured at the fan inlet. The fan total pressure is the measure of the total mechanical energy added to the air or gas by the fan.

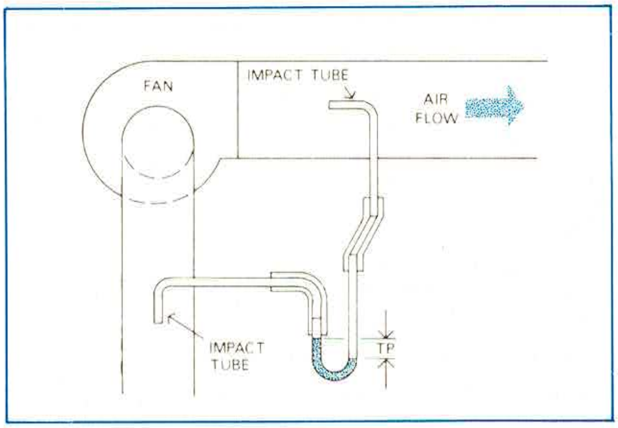


Figure 4-12 FAN TOTAL PRESSURE (TP)

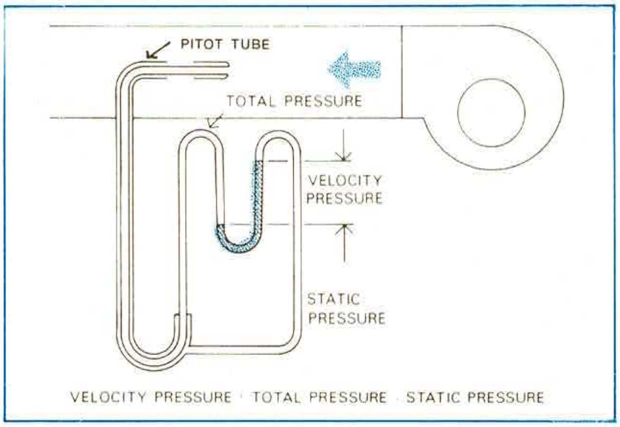


Figure 4-14 FAN VELOCITY PRESSURE (V_p)

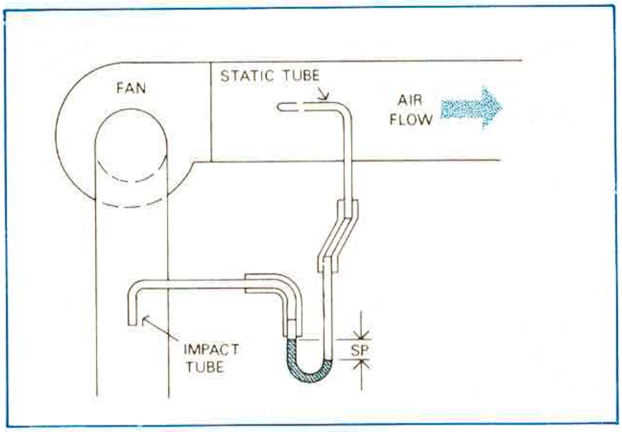


Figure 4-13 FAN STATIC PRESSURE (SP)

c. FAN STATIC PRESSURE (SP)

Fan static pressure (Figure 4-13) is the fan total pressure less the fan velocity pressure. It can be calculated by subtracting the total pressure at the fan inlet from the static pressure at the fan outlet. This is a source of some confusion within the industry, but, by definition:

Fan SP = Fan TP (outlet) − TP (inlet) − V_p (outlet)

Also:

TP (outlet) − SP (outlet) = V_p (outlet)

and, substituting, results in

Fan SP = SP (outlet) − TP (inlet)

In taking field measurements, care must be taken when a duct inlet is employed to measure the total pressure at the inlet rather than just static pressure. However from a practical viewpoint, it is almost impossible to get an accurate reading because of the high pressure losses of the fittings at the fan and the absence of straight sections of inlet and discharge ducts.

d. FAN VELOCITY PRESSURE (V_p)

Fan velocity pressure (Figure 4-14) is the pressure corresponding to the fan outlet velocity. It is the kinetic energy per unit volume of flowing air.

e. FAN OUTLET VELOCITY

This is the theoretical velocity of the air as it leaves the fan outlet, and is calculated by dividing the air volume in cfm by the fan outlet area in square feet. However, all fans have a non-uniform outlet velocity, that is, velocity varies over the cross-section of the fan outlet (see Figure 4-15). Therefore, outlet velocity as calculated above is only a theoretical value that could occur at a point removed from the fan.

As shown in Figure 4-15, almost all of the airflow occurs at the side of the outlet farthest from the fan shaft. Velocity readings taken at the side nearest the shaft may actually indicate airflow from the discharge duct back into the fan.

f. BRAKE HORSEPOWER

Brake horsepower (bhp) is the actual horsepower required to drive the fan. It is greater than a theoretical "air horsepower" because it includes loss due to turbulence and other inefficiencies of the fan, plus bearing losses. Brake horsepower is an important value to the TAB technician because it is the power furnished by the fan motor (less drive losses).

4. Fan Laws

Fan performance at various speeds (rpm) and when handling air of various densities can be predicted when the performance is known at a specific condition. The different conditions can be calculated by using the fan laws.

a. FAN LAW #1—CFM VARIES IN DIRECT PROPORTION TO THE RPM

If you increase the rpm of a fan by 10%, the cfm will also increase by 10%.

Equation 4-1

$$\frac{cfm_2}{cfm_1} = \frac{rpm_2}{rpm_1}$$

Where:

cfm_2 = new or required airflow

cfm_1 = actual or existing airflow

rpm_2 = new or required fan speed

rpm_1 = actual or existing fan speed

Example 4A

A fan is operating at 875 rpm, and is delivering 6550 cfm. Find the fan speed at which the fan can deliver 7875 cfm to the system.

Solution

Using Equation 4-1:

$$\frac{cfm_2}{cfm_1} = \frac{rpm_2}{rpm_1}$$

$$rpm_2 = \frac{rpm_1 \times cfm_2}{cfm_1} = \frac{875 \times 7875}{6550}$$

$$rpm_2 = 1052$$

Therefore, increasing the fan speed from 875 rpm to 1052 rpm should establish a new system air volume of 7875 cfm.

Example 4B

Suppose a fan is turning at 620 rpm and delivering 8875 cfm. You increase the fan speed to 690 rpm. Calculate the new fan airflow.

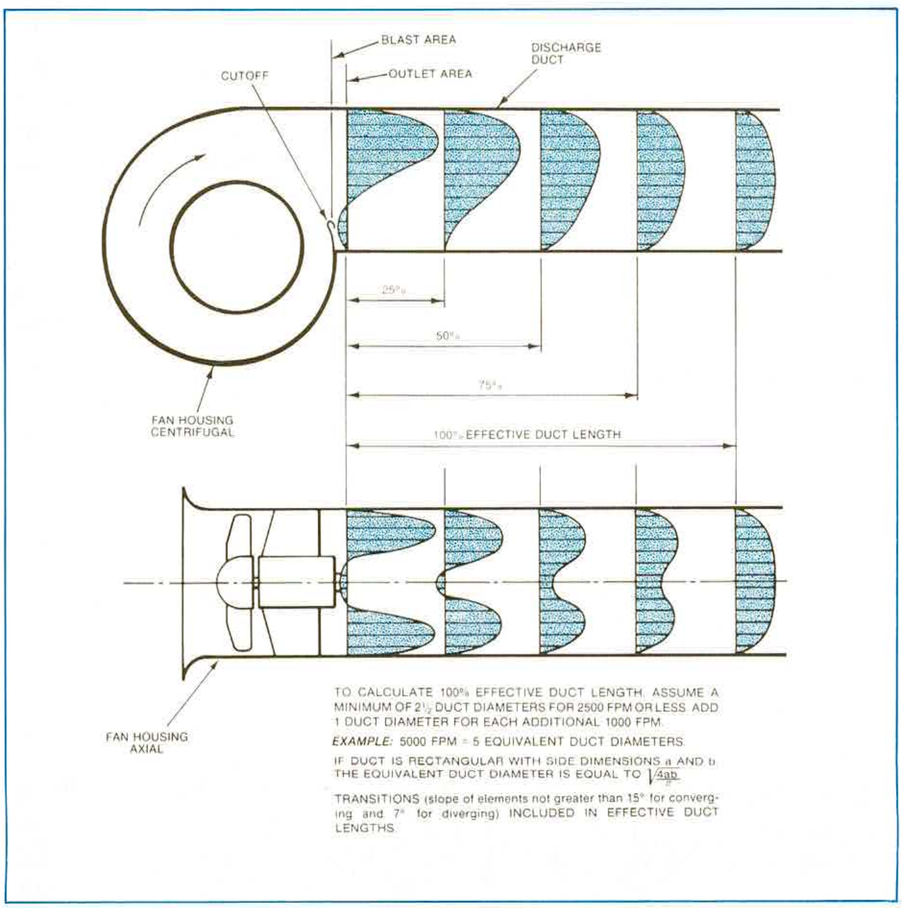


Figure 4-15 CONTROLLED DIFFUSION AND ESTABLISHMENT OF A UNIFORM VELOCITY PROFILE IN A STRAIGHT LENGTH OF OUTLET DUCT

Solution

$$\text{New cfm} = \text{Original cfm} \times \frac{\text{New rpm}}{\text{Original rpm}}$$

$$\text{New cfm} = 8875 \times \frac{690 \text{ rpm}}{620 \text{ rpm}} = 9877$$

By speeding up the fan from 620 rpm to 690 rpm, you have increased the air volume from 8875 cfm to 9877 cfm.

b. FAN LAW #2—STATIC PRESSURE VARIES BY THE SQUARE OF THE RPM

Therefore, if you decrease the rpm of a fan by 10%, the static pressure will decrease by the square of 10%.

Equation 4-2

$$\frac{SP_2}{SP_1} = \left(\frac{rpm_2}{rpm_1}\right)^2$$

Where:

SP_2 = new or required static pressure

SP_1 = actual or existing static pressure

rpm_2 = new or required fan speed

rpm_1 = actual or existing fan speed

Example 4C

Using the figures from Example 4A, increase the speed of the fan from 875 rpm to 1052 rpm. The fan static pressure at 875 rpm is 0.68 in.w.g. Calculate the new S.P. at 1052 rpm.

Solution

Using Equation 4-2:

$$\frac{SP_2}{SP_1} = \left(\frac{rpm_2}{rpm_1}\right)^2$$

$$SP_2 = SP_1 \left(\frac{rpm_2}{rpm_1}\right)^2 = 0.68 \times \left(\frac{1052}{875}\right)^2$$

$$SP_2 = 0.98 \text{ in.w.g.}$$

By speeding up the fan from 875 rpm to 1052 rpm, you have increased the fan static pressure from 0.68 in.w.g. to 0.98 in.w.g.

Since fan rpm and cfm are directly proportional, cfm can readily be substituted for rpm in Equation 4-2 to provide the new cfm when a fan speed has been changed.

Equation 4-3

$$\frac{SP_2}{SP_1} = \left(\frac{cfm_2}{cfm_1}\right)^2$$

Example 4D

A fan is handling 8500 cfm at a S.P. of 0.75 in.w.g. The fan speed is increased and the S.P. is now 1.15 in.w.g. Find the new airflow.

Solution

Using Equation 4-3:

$$cfm_2 = cfm_1 \times \sqrt{\frac{SP_2}{SP_1}}$$

$$cfm_2 = 8500 \sqrt{\frac{1.15}{0.75}} = 10{,}525$$

c. FAN LAW #3—HORSEPOWER VARIES BY THE CUBE OF THE RPM

Equation 4-4

$$\frac{bhp_2}{bhp_1} = \left(\frac{rpm_2}{rpm_1}\right)^3$$

Where:

bhp_2 = new or required brake horsepower

bhp_1 = actual or existing brake horsepower

rpm_2 = new or required fan speed

rpm_1 = actual or existing fan speed

Example 4E

Using figures from Example 4A, you have increased the speed of a fan from 875 rpm to 1052 rpm. The fan was using 12.25 bhp when running at 875 rpm. What bhp is it now using when running at 1052 rpm?

Solution

Using Equation 4-4:

$$bhp_2 = bhp_1 \left(\frac{rpm_2}{rpm_1}\right)^3$$

$$bhp_2 = 12.25 \times \left(\frac{1052}{875}\right)^3 = 21.3$$

With a 22.5 percent increase in fan speed, the fan brake horsepower has increased 73.8 percent!

Example 4F

A fan is running at 723 rpm and is using 11.4 bhp. You need to speed up the fan to get as much air as possible with the existing 15 HP motor installed. How fast can you run this fan with this motor?

Solution

$$rpm_2 = rpm_1 \times \sqrt[3]{\frac{bhp_2}{bhp_1}}$$

$$rpm_2 = 723 \sqrt[3]{\frac{15}{11.4}} = 792$$

Therefore, you should be able to speed this fan up to about 792 rpm and have the motor fully loaded to its rated 15 HP.

d. FAN LAW #4—FAN VOLUME IN CFM WILL NOT CHANGE WITH A CHANGE IN DENSITY

A fan is a *constant volume machine* and will handle the same *airflow* (cfm) regardless of the system fluid or air density. The bhp and static pressure (SP) will vary in direct proportion to the density. Handling more dense or heavier air will produce more pressure and require more horsepower. Fan performance data is based on *standard air* which has a density of 0.075 lbs. per cubic foot.

Equation 4-5

$$\frac{SP_2}{SP_1} = \frac{d_2}{d_1}$$

Equation 4-6

$$\frac{bhp_2}{bhp_1} = \frac{d_2}{d_1}$$

Where:

SP_2 = new or required static pressure

SP_1 = actual or existing static pressure

bhp_2 = new or required brake horsepower

bhp_1 = actual or existing brake horsepower

d_2 = new or required density

d_1 = actual or existing density

Example 4G

A tested fan is handling 12,500 cfm at 0.93 in.w.g. and is using 4.8 bhp. The density of the air being handled is 0.070 pounds per cubic foot. To relate these figures to the manufacturer's published data will require correcting the SP and bhp to what they would be if the fan was handling standard air. Since the fan is a constant volume device, the airflow will still be 12,500 cfm.

Solution

$$SP_2 = SP_1 \times \frac{d_2}{d_1}$$

$$SP_2 = 0.93 \times \frac{0.075}{0.070} = 0.996 \text{ in.w.g.}$$

$$bhp_2 = bhp_1 \times \frac{d_2}{d_1}$$

$$bhp_2 = 4.88 \times \frac{0.075}{0.070} \times 5.14$$

At standard air conditions, the fan will handle 12,500 cfm at 0.996 in.w.g. SP and 5.14 bhp.

The TAB technician should be cautioned that when balancing a system where the fan is handling hot and therefore less dense air most of the time, a check of the horsepower being used should be made with standard air (70°F) going through the system to insure that the motor isn't overloaded during start up or cool down. At these times, the fan would be handling a denser air and would be using more brake horsepower.

5. Fan Curves

Fan curves can be very useful to the TAB technician, particularly when a problem arises. Fan curves (Figure 4-16) are usually laid out so that the static pressure (SP) is on the left vertical side of the graph and the airflow (cfm) is along the bottom horizontal side. Horsepower curves are plotted from upper left to lower right. Point A on Figure 4-16 is the selected fan capacity of 1140 cfm at 1.7 in.w.g. SP. This selection would require a fan speed of 1700 rpm and a ¾ HP meter.

The fan curves shown in Figures 4-17 to 4-22 illustrate the various characteristics of the basic types of fans. Generally toward the right side of the fan curves, it can be seen that the fans are able to deliver more cfm if the static pressure or resistance is lowered or decreased. In fact, a point can be found on a fan curve in which the static pressure is high and yet quite a bit of cfm is still being delivered. This is a point near the peak static efficiency of the fan where it will produce the least noise for the particular size and is shown in the figures as the "static efficiency curve." Knowing the performance characteristics of each type of fan is very important to TAB technicians, as each and every job will involve the use of this knowledge.

If you are not getting proper results from a fan, obtain a copy of the correct fan curve and plot your test results on the curve and compare them. If you are low on airflow (cfm), you may find that you are operating at a static pressure higher than design, which will put you higher up on the chart.

There will be times when you won't be able to obtain a fan curve. You may be able, however, to get a copy of the fan rating table (Table 4-1). This can also be of some assistance but the data is usually less complete. Again, interpolation will be necessary.

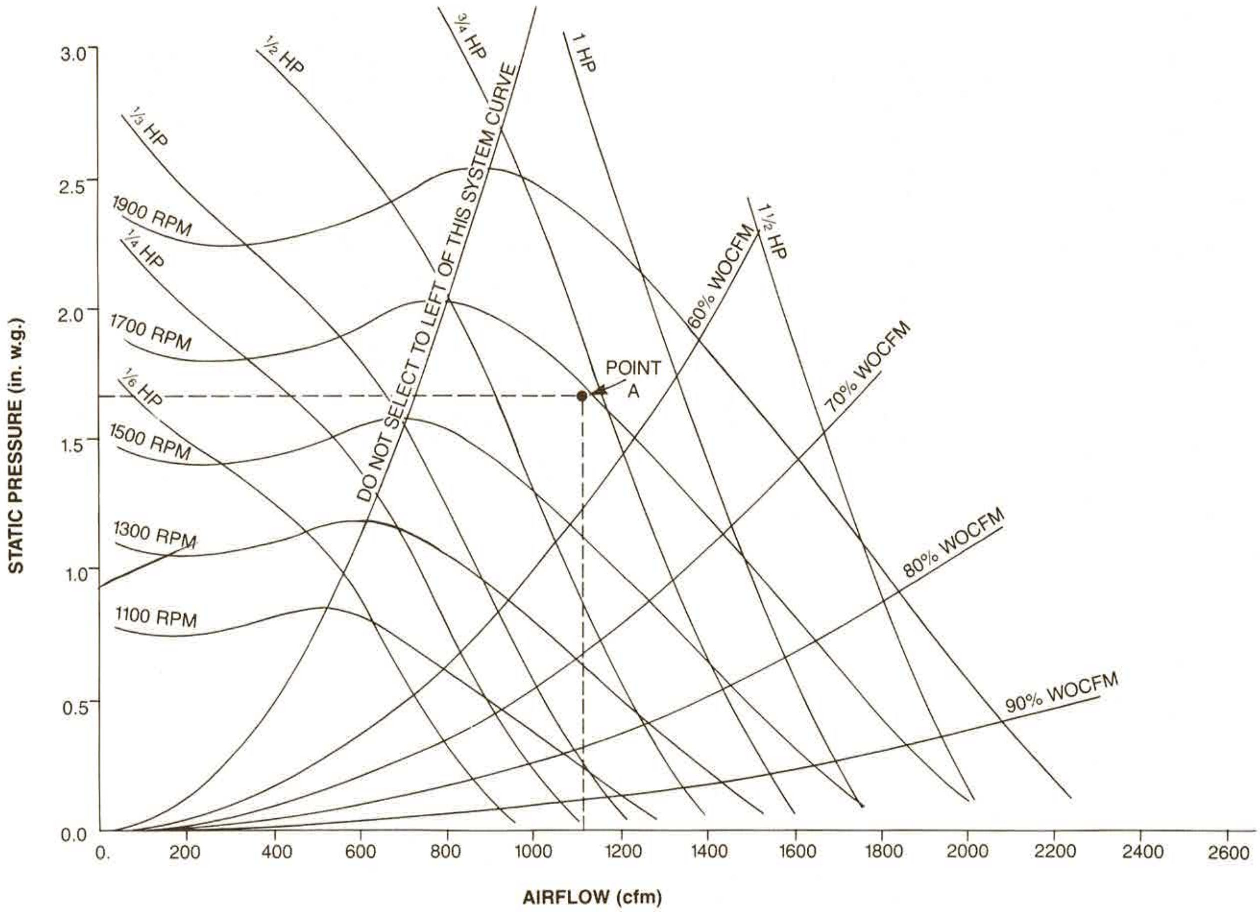


Figure 4-16 TYPICAL FC FAN CURVES

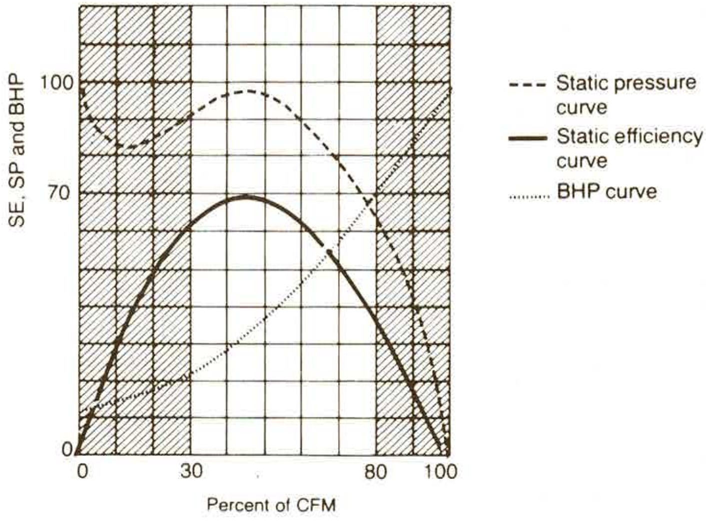


Figure 4-17 FORWARD CURVED FAN WHEEL

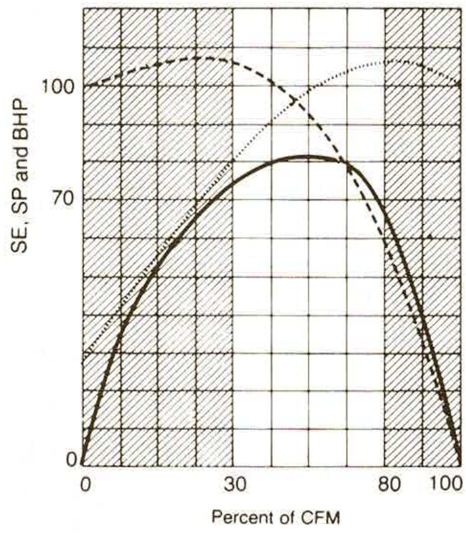


Figure 4-18 BACKWARD INCLINED FAN WHEEL

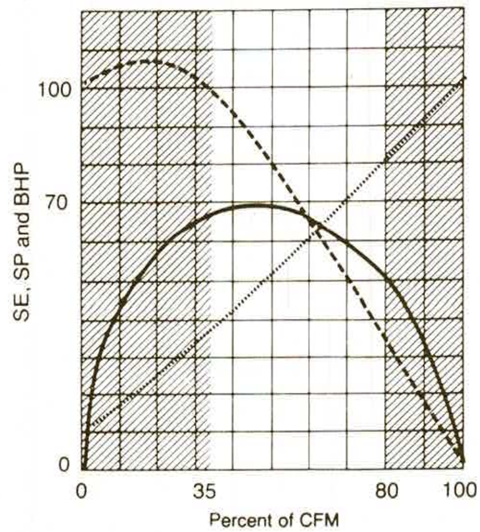


Figure 4-19 RADIAL BLADED FAN WHEEL

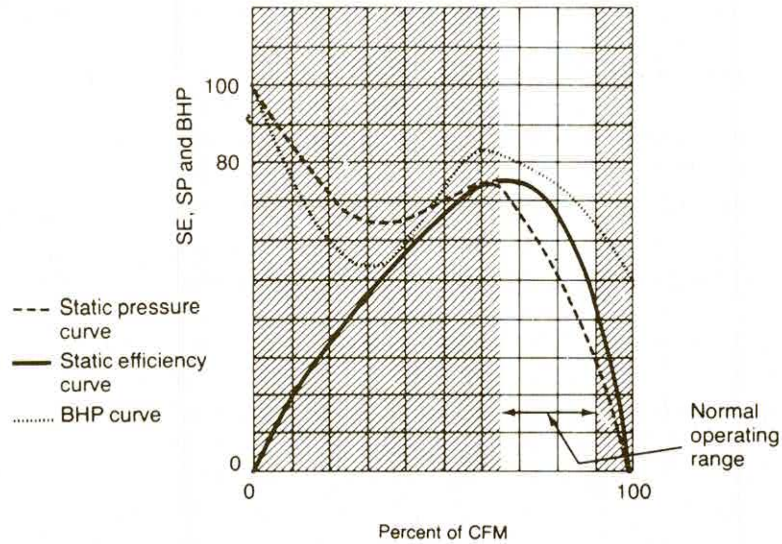


Figure 4-20 CHARACTERISTIC CURVES FOR VANEXIAL FANS (HIGH PERFORMANCE)

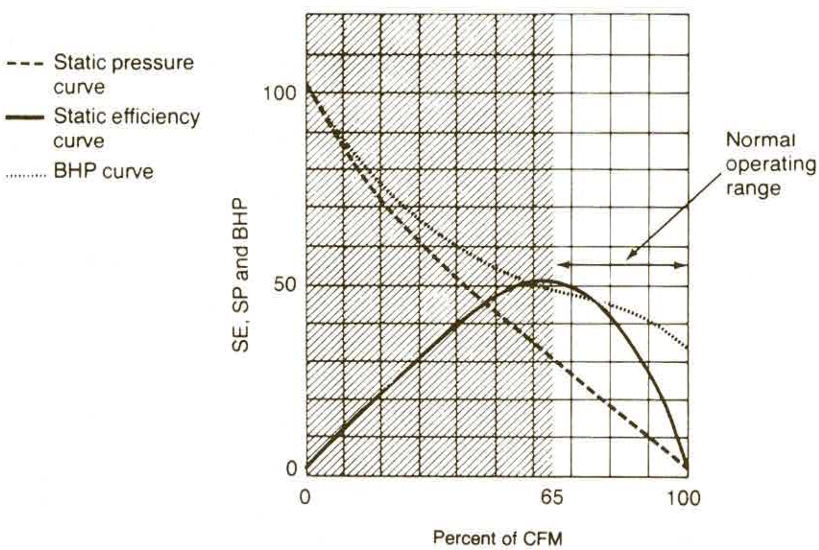


Figure 4-21 CHARACTERISTIC CURVES FOR PROPELLER FANS AND TUBEAXIAL FANS

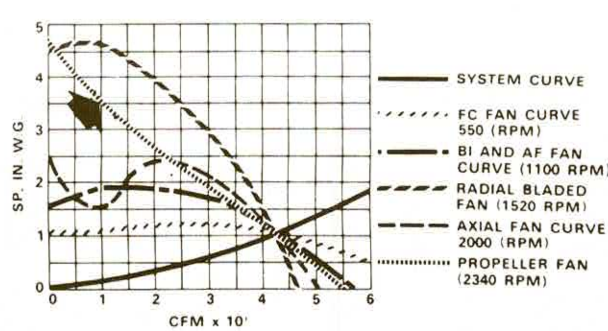


Figure 4-22 COMPARISON OF 20″ DIAMETER FAN CURVES

6. Fan/System Curve Relationships

In a given duct system with a known airflow rate and when the positions of all dampers are stable, a specific, measurable static pressure resistance to the airflow can be determined or measured.

But if the flow rate is increased, the duct system resistance is increased. That is, if the cfm increases, the system static pressure increases. System resistance is the sum of all pressure losses through filters, coils, dampers and ductwork. The system resistance curve or *system curve* is a plot of the pressure that is required to move air through the system (Figure 4-23). For fixed systems, that is, with no changes in damper settings, etc., system resistance varies as the square of the airflow (cfm).

Equation 4-7

$$\frac{P_2}{P_1} = \left(\frac{Q_2}{Q_1}\right)^2 \text{ or } \frac{SP_2}{SP_1} = \left(\frac{cfm_2}{cfm_1}\right)^2$$

Where:

P = System pressure (in.w.g.)

Q = System airflow (cfm)

The resistance curve for any system is represented by a single curve. The example, consider a system handling 1,000 cfm with a total resistance of 1 inch SP (in.w.g.). If the airflow is doubled, the static pressure resistance will increase by that ratio squared to

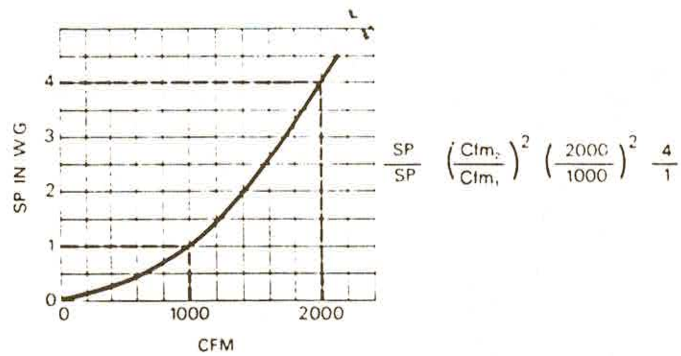


Figure 4-23 SYSTEM RESISTANCE CURVE

4 in.w.g. shown. This curve changes, however, as filters load with dirt, coils start condensing moisture, or when outlet dampers are changed in position.

The operating point at which the fan and system will perform is determined by the intersection of the system curve and the fan performance curve. Every fan operates *only* along its performance curve. If the designed system resistance is not the same as the resistance in the installed system, the operating point will change and the static pressure and volume of air delivered will not be as calculated.

Looking at the right side of the two fan curves in Figure 4-24, if the duct system airflow (cfm) increases, the static pressure decreases. This seemingly contradictory statement comes from the fact that

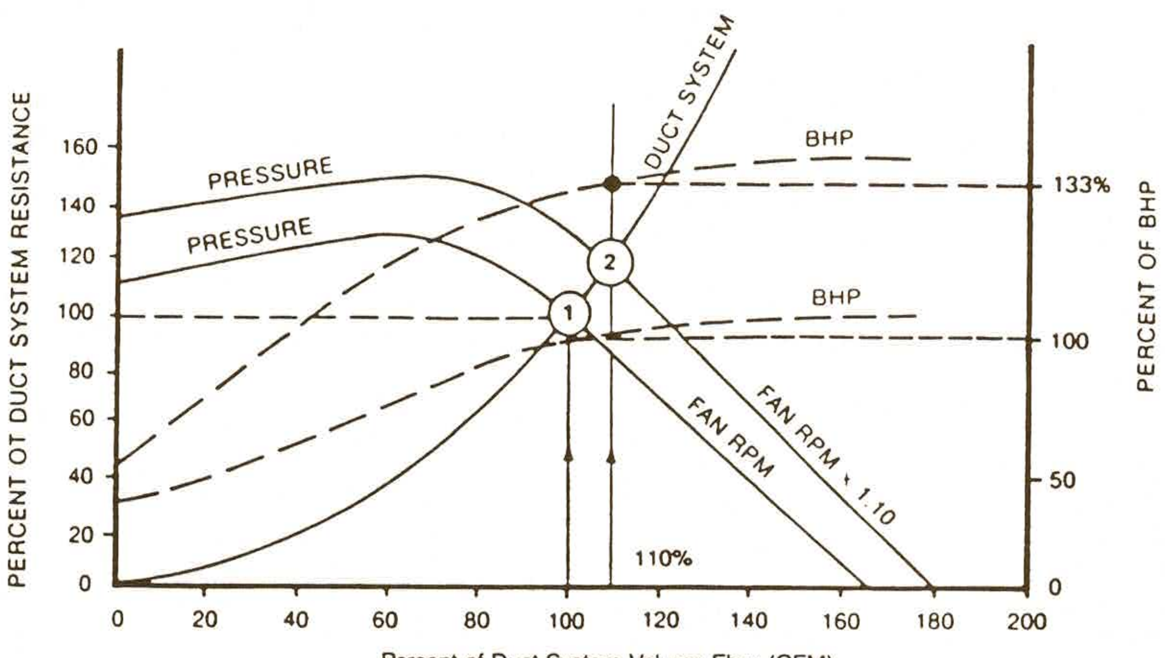


Figure 4-24 EFFECT OF 10% INCREASE IN FAN SPEED

"apples" and "oranges" are being compared, that is, a fan curve with a duct system curve. The fan curve statement is correct because it "looks at" the fan. What is said about the system is also correct because it is "looking at" the system. In fact, Equation 4-7 was written for the system curve. The application of Equation 4-7 says that system pressure increases proportionally to the square of the system airflow, as stated earlier.

At any given *constant speed and horsepower,* it is found that as the fan curve progresses to the right, the cfm *increases.* But in order for it to increase, the static pressure must *decrease.* When a duct system curve is drawn on the same fan curve, it moves up as it progresses to the right as shown in Figure 4-24. The duct system curve also predicts performance, as the cfm increases the same as the fan curve, and at some point, the two curves will intersect. This intersection point will predict the performance of *this* fan on *this* system. The duct system curve does enforce the fundamental statement on duct systems, that is, as the cfm or airflow increases, the system static pressure also increases.

What this means is that each duct system has its own unique system curve, and that each fan has a fan curve for each speed or rpm, and wherever the two intersect will be the operating point of both the fan and duct system. It now can be seen that both theories of static pressure versus cfm are correct. In the system, the static pressure does rise as the cfm increases. In the fan, however, the static pressure usually decreases as the cfm increases (at least in the right hand portion of the fan curves).

With fan laws (Equations 4-1 to 4-6), the fan curves, and the system curves, any change to any component can be calculated and graphically portrayed. Looking at the intersection point of system curve and the fan curve in Figure 4-24, suppose the cfm is increased, that is, move this point to the right. Would the static increase, decrease, or stay the same? Either answer may be correct because it depends on how the point is moved to the right.

To move the point, a physical change must be made to the fan or to the system. It could be to the fan, that is, a change in the sheave size to speed up the fan. This will increase the cfm and increase the static pressure because the system curve will remain unchanged. The fan curve (because that is where the physical change is made) will go to a higher parallel curve. *The intersection point will travel along the system curve to the new fan speed.*

If, however, physical changes were made to the system, the static pressure may decrease. Assume that a balancing damper on a large duct branch was discovered to be closed and was then opened. This would create a new system curve to the right. The new system would have more free area, that is, would have more square area of cross-section and therefore be able to handle more cfm.

In both of the system changes, it was the system curve that was changed, not the fan curve. *In this case the intersection point moved along the existing fan curve to the new system curve.* While the cfm was increasing, the static pressure was decreasing. The question "If the cfm increases, will the static pressure increase or decrease?" becomes ambiguous. More facts need to be known in order to answer the question. Are the physical changes taking place in the fan or in the system?

The same type of change also could have happened if the system was installed with rather abrupt constrictions in order to pass under a deep beam. After having gone through the proper channels of communication, corrective measures were authorized. The old section of ductwork came down and new ductwork was erected in a different location eliminating the constriction and allowing freer air delivery (decreasing the resistance and increasing the cfm).

The use of the fan curve charts can be used as an aid in trouble-shooting this type of problem in a system. It is possible to physically plot the changes on the fan curve, determine the new points, and determine other data such as horsepower, etc., by interpolation or graphic means. This method is not recommended to achieve the *exact* answer. In order to achieve exact answers, the calculations must be made using the appropriate fan law equations. Notice that these are entitled "fan laws," as these equations will handle *only* the physical changes made to the fan, such as changing the fan speed.

Don't be surprised when test measurements do not fall right into place on the curves. More often than not, there will be differences. This can be due to many different or combined reasons, some of which follow:

(a) Manufacturer's published data is from tests performed under ideal laboratory conditions with a controlled environment. This is not the case in the field. Duct configurations and unmeasureable effects from the way the ductwork is connected to the fan can have a marked impact on fan performance. Normal field tests will not detect these deratings of the fan performance. Turbulence and "spin" are quite common effects covered in the NEBB publication "Environmental Systems Technology."

(b) Although TAB instruments are very accurate, the measurements are dependent on the availability of good locations in which to take them.

Quite often, these locations are not accessible or do not exist.

(c) For fan tests, the most accurate measurement will be the rpm reading. Brake horsepower calculations should be fairly accurate. System airflow measurements will be very dependent on having good locations for Pitot tube traverses. If several diameters of straight duct are available, the results should be good. Static pressure readings are the most difficult readings to get accurately in the field, so they are usually the least accurate.

7. Fan Drives

A fan may be driven either directly from a motor shaft or through a drive consisting of pulleys and V-belts (Figure 4-25). Quite often, the belt drive type have an adjustable pitch pulley on the motor so the speed of the fan can be adjusted. By changing the size of the pulleys, various speeds can be obtained for the fan. Usually the drive motor is mounted on an adjustable base so that the belts can be properly tensioned. The TAB technician primarily is concerned with alignment, tensioning and speed adjustment if provided. Some general items are:

(a) Alignment: The pulley should be in parallel alignment and the shafts should also be parallel (see Figure 4-26).

(b) Tensioning: Belts that are not tight enough will slip, causing belt and drive wear; or they even may fly off. Belts that are too tight will put undue strain on the bearings, possibly even causing premature failure. Tension gauges should be used for making correct adjustments. Slack in belts can be noticed when they are running. If properly sized belts squeal when starting up, they are too loose and must be tightened. Never force belts over the pulley grooves; always loosen the adjustments enough to install or remove them without force.

(c) Speed Adjustment: Many belt drives use an adjustable pitch drive pulley. The TAB technician often will need to make adjustments to obtain the correct fan speed and airflow. The following drive equations will let you determine what size pulleys will deliver the correct speed (Figure 4-29).

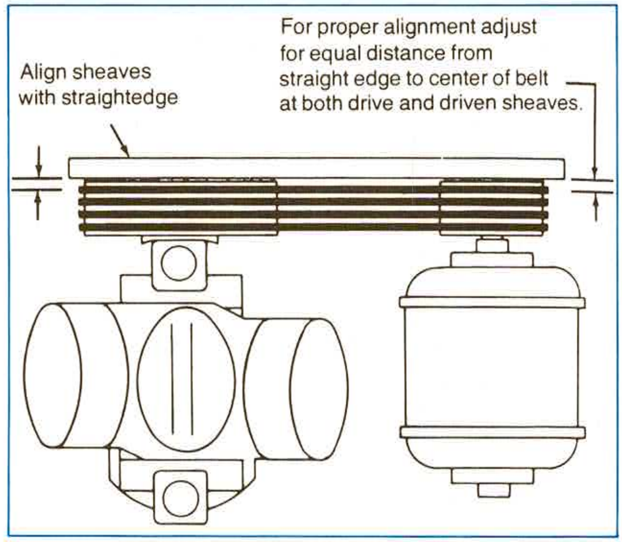


Figure 4-26 CORRECT V-BELT DRIVE ALIGNMENT

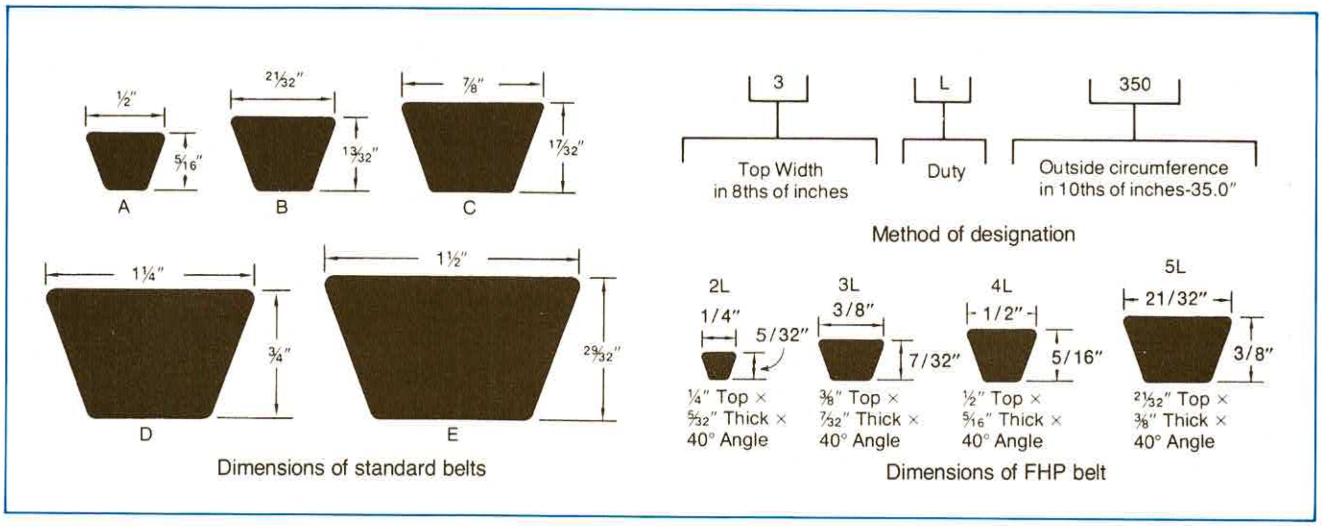


Figure 4-25 DESIGNATION OF V-BELT SIZE

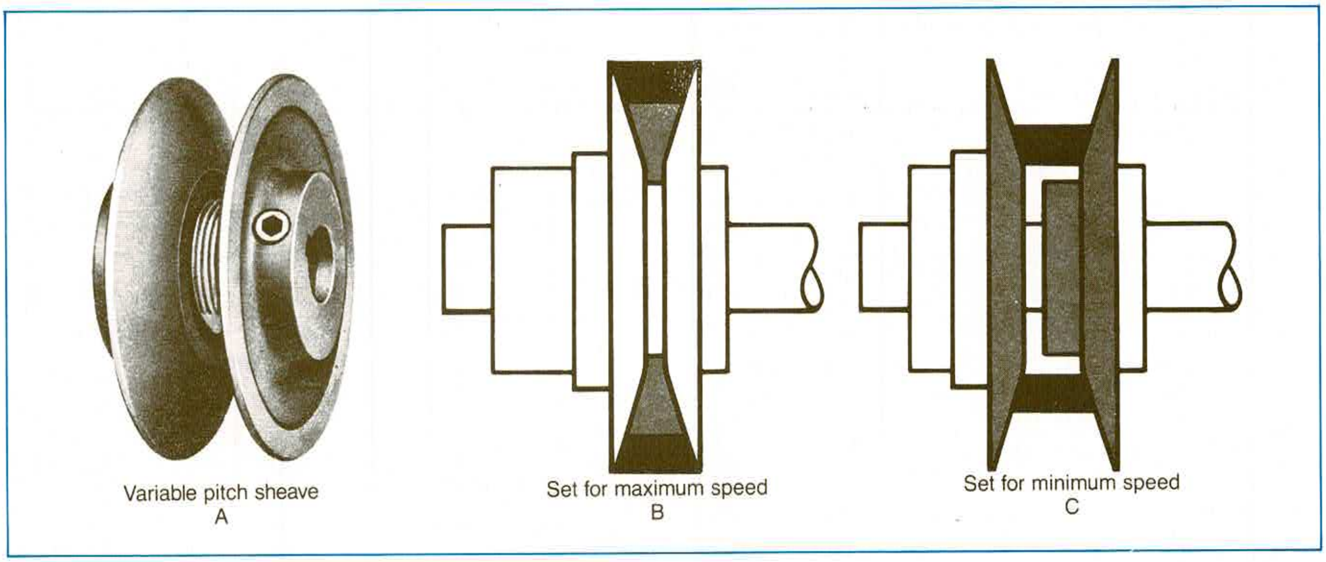


Figure 4-27 VARIABLE PITCH SHEAVE FOR SINGLE-BELT DRIVE

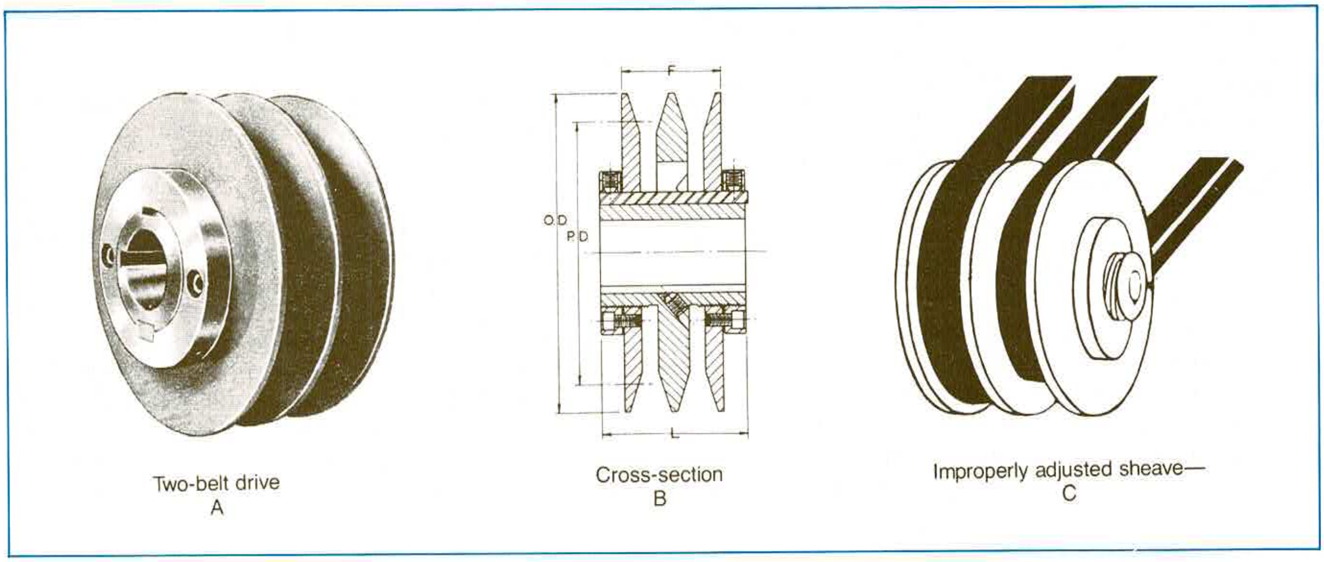


Figure 4-28 VARIABLE PITCH SHEAVE FOR TWO-BELT DRIVE

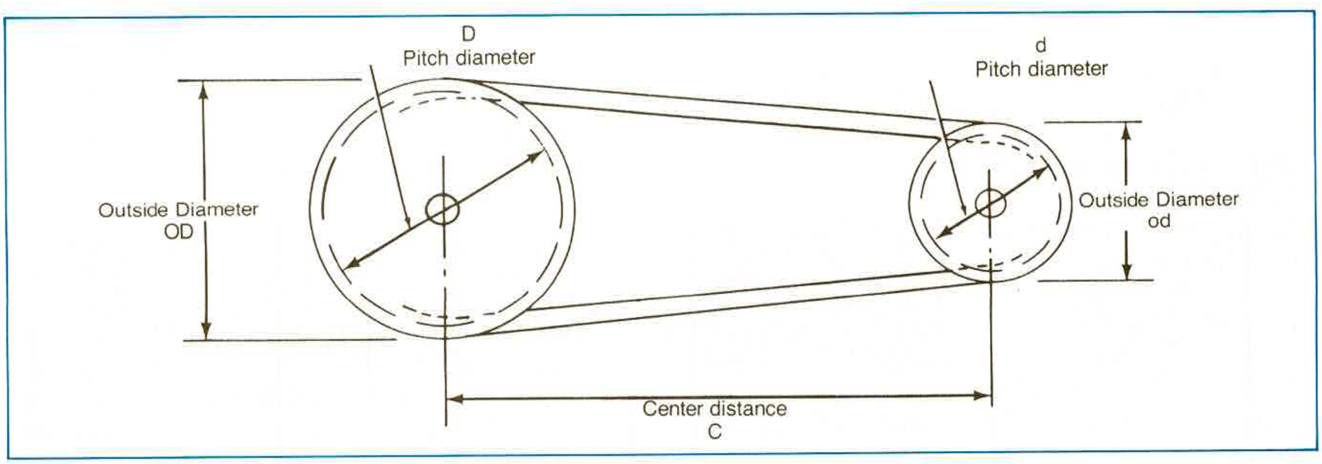


Figure 4-29 V-BELT DRIVE

Equation 4-8

$$\frac{\text{rpm (Fan)}}{\text{rpm (Motor)}} = \frac{\text{Pulley pitch diameter (motor)}}{\text{Pulley pitch diameter (fan)}}$$

Example 4H

A fan has an 8″ diam. motor pulley and a 10″ diam. fan pulley. The motor is turning 1750 rpm. What speed is the fan turning?

Solution

Using Equation 4-8:

$$\text{Fan rpm} = \frac{\text{Motor rpm} \times \text{pulley diameter (motor)}}{\text{Pulley diameter (fan)}}$$

$$\text{Fan rpm} = \frac{1750 \times 8''}{10''} = 1400$$

The fan will be running at 1400 rpm.

Example 4I

A fan is supposed to be running at 1370 rpm. The motor pulley is an adjustable type. What should be the adjusted pitch diameter of the motor pulley to obtain a fan speed of 1370 rpm?

Solution

$$\text{Motor Pulley diam.} = \frac{\text{Required Fan rpm} \times \text{Fan Pulley Diameter}}{\text{Motor rpm}}$$

$$\text{Motor Pulley diam.} = \frac{1370 \times 10''}{1750} = 7.83''$$

This motor pulley will need to be adjusted to 7.83″ pitch diameter.

Drive manufacturers work with pitch diameters, which are approximately the centerline of the belts, but most drive manufacturers now also list the outside diameters (O.D.) in their catalogs. The TAB technician usually will be measuring outside diameters in the field. So for TAB work, the O.D. may be used for calculating sizes, as the results are practically identical.

8. V-Belts

HVAC equipment belts in general use are of V-belt type. They are made in several sizes of belt cross-section, some typical ones being shown in Figure 4-25. In this illustration, the group of sizes *A* through *E* are standard industrial-type belts, while those designated *2L* through *5L* are light-duty belts, sometimes called FHP belts (for fractional horsepower belts). Belts of the FHP class are more flexible than industrial belts of equivalent cross-section and therefore can be used on sheaves of smaller diameter than would be recommended for industrial belts, and are popularly used for driving fans of smaller size ranges.

Belts are rated by horsepower per belt, and in order to transmit a given horsepower from a motor or engine to a driven machine such as a fan, it may be necessary to use two, three, or more belts to avoid excessive belt stress. Belts are also rated as to minimum diameter of radius of curvature; that is, the minimum size of sheave with which they should be used, because excessive bending of the belt will cause excessive wear and early failure of the belt.

With larger belt cross-sections and with the larger sheaves that will be required, a greater center distance between shafts also will be required. As it usually is desirable to make machinery as compact as possible, the practice is to use several belts of smaller cross-section rather than one belt of large cross-section. Further referring to Figure 4-25, note that *A* belts and *4L* belts have the same cross-sectional dimensions and that *B* and *5L* belts have the same cross-sectional dimensions. However, the *4L* and *5L* belts are more flexible than the comparable standard belts, and so can be satisfactorily used with sheaves of smaller diameter. Thus, if belts have to be changed, it is important to avoid a change in the type of belt used.

9. Drive Alignment and Tension

If several belts are used in one drive, the belts should be matched in length, so that they will all have approximately the same tension and therefore each will carry its proportionate share of the load. The belts will stretch with use, and when it is necessary to change one belt in a multi-belt drive, all belts in the drive should be changed, replacing them with a matched set. Otherwise, upon installing a new set of belts, one or more belts may be too tight or too slack as compared to the others. If this variance does not result in improper operation such as the slipping of one or more belts, there may in time be enough stretching of the tighter belts until all belts match. Belt tension should be checked and adjusted several times during the first few days of operation of such a drive.

The motor and the fan (or other driven machine) must be properly aligned to avoid excessive belt wear and the chance of belts jumping off the sheaves.

All belt drives must be adjusted for proper belt tension. There must be enough tension to prevent excessive slippage between the belt or belts and the sheaves. However, excessive tension is to be avoided as it can result in early belt failure, excessive wear on shaft bearings, and possible overload of a drive motor. With practice, proper belt tension can be judged fairly well by simply pushing on the belt, estimating that tension is about right if each belt can be depressed about one-half to three-quarters of an inch under normal pressure from the thumb. For more accurate results, belt tension testers are available. These are tools which operate on the principle of measuring the force in pounds needed to depress or deflect the belt a given amount; the amount of deflection and the deflection force in pounds being taken from belt manufacturer's data.

Although excessive belt tension is undesirable, it is also undesirable to have any slack in the belts as this causes belt slippage, a loss in power transmitted, and excessive belt wear.

For new belt drives, tension should be rechecked after initial operating periods, such as the first 12 hours and 72 hours of operation, making such adjustments of tension and alignment as may be indicated.

C AIR SYSTEM COMPONENTS

1. Cooling Coils

Coils described herein are used for cooling an air stream under forced convection. This equipment may consist of a single coil section or a number of individual coil sections built-up into banks. Coils are also used extensively as components in central station type air handling units, room terminals, and in factory assembled self-contained air conditioners.

Coils are used for air cooling with or without dehumidification. Precooling coils using relatively high temperature water usually do not dehumidify the air; however, a major portion of coil equipment is designed to provide, simultaneously, both sensible cooling and dehumidification.

a. COIL CONSTRUCTION

Coils are basically of two types, those consisting of *bare tubes or pipe,* and those having *extended or finned* surfaces. The design and arrangement of a coil, constructed with extended type surface on the air side, involve consideration of such items as: materials, fin size and spacing, ratio of extended surface area to that of the tube area, tube nesting center dimensions, use of staggered or in-line tube arrangement, and use of turbulators. The design and surface arrangement have a great effect on the air-film heat transfer resistance and associated air-side pressure drop. Figure 4-32 illustrates several arrangements.

Cooling coils most frequently have aluminum fins and copper tubes, although copper fins on copper tubes are also used and the combination of aluminum fins on aluminum tubes is finding usage. There are many makes of cooling coils of the lightweight extended-surface type for both heating and cooling with tubes commonly 1/4″, 3/8″, 1/2″, 5/8″, 3/4″, and 1″ outside diameter, and with fins spaced three per inch up to fourteen per inch. The tube spacing generally varies from about 5/8″ to 2-1/2″ on centers, depending upon the width of individual fins and on other considerations of performance.

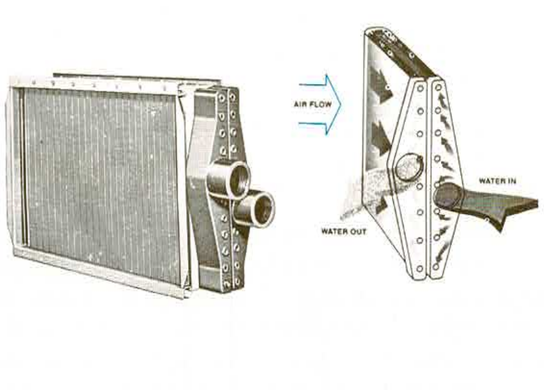


Figure 4-30 HYDRONIC HEATING COIL

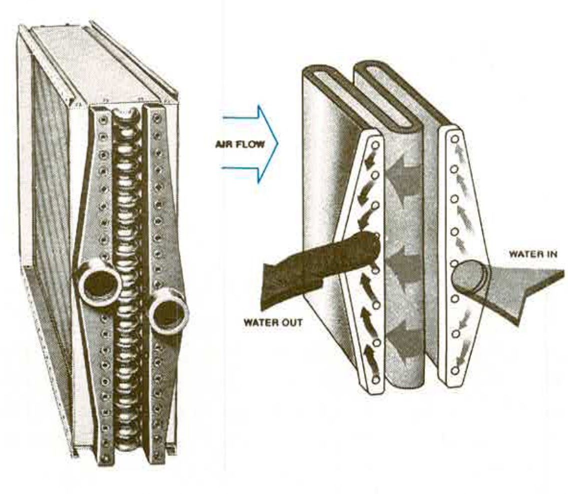


Figure 4-31 HYDRONIC COOLING COIL

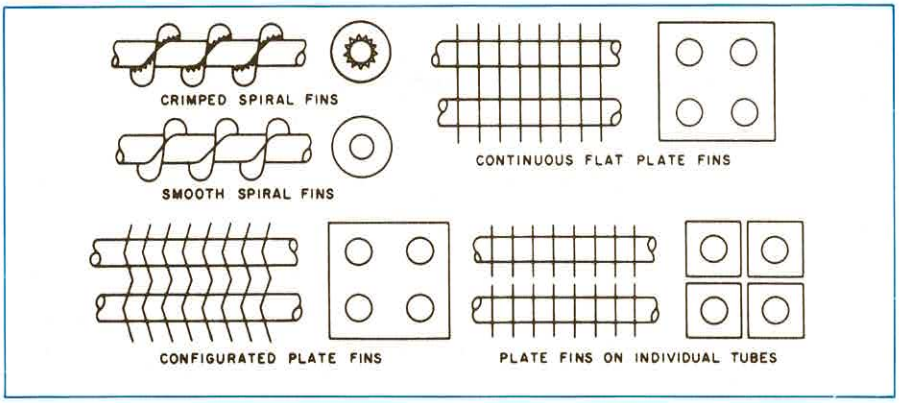


Figure 4-32 TYPES OF FIN-COIL ARRANGEMENTS

b. WATER COILS

The performance of water coils depends on the elimination of air from the water circuit and proper distribution of water. Unless vented, air may accumulate in the coil tube circuits, causing a reduction in thermal performance and possibly noise or vibration in the piping system. Air vent connections are usually provided on the coil water headers. Depending upon performance requirements, the water velocity inside tubes usually ranges from approximately 1 to 8 feet per second and the design water pressure drop across coils varies from about 5 to 50 feet head of water.

c. DIRECT EXPANSION COILS

Direct expansion coils (DX coils) present more complex problems of cooling fluid distribution than water or brine coils. It is desirable that the coil be effectively and uniformly cooled throughout, and it is necessary that the compressor be protected from entrained, unevaporated refrigerant. Direct expansion coils are used on two types of refrigeration systems, flooded systems and dry-expansion systems.

The flooded system is used mainly where evaporator coil performance for low temperature applications at a small temperature difference between the air and refrigerant is advantageous. However, a relatively large volume of refrigerant is required, together with extra appurtenances such as surge tank and interconnecting piping. Other applications of direct expansion coils generally use dry expansion. For dry-expansion systems, two of the most commonly used refrigerant liquid metering devices are the capillary tube and the thermostatic expansion valve.

2. Heating Coils

Coils used for air heating are generally of the extended-surface type. The heating medium is generally steam or heated water.

a. STEAM COILS

For proper performance of steam heating coils, condensate and air or other noncondensables must be rapidly eliminated and the steam must be uniformly distributed to the individual tubes. Noncondensable gases, such as carbon dioxide, remaining in the coil cause chemical corrosion and result in early coil failure.

Uniform steam distribution is accomplished by different methods such as:

(1) Individual orifices in the tubes.

(2) Distributing plates in the steam headers.

(3) Special perforated small diameter inner steam distributing tubes extending into the larger diameter tubes of the primary surface.

Coils of the perforated inner tube type are constructed with different arrangements such as:

(1) Supply and return on one end, with the incoming steam used to heat the leaving condensate (Nonfreeze type).

(2) Supply and return on opposite ends.

(3) Supply and return on one end and a supply on the opposite end (nonfreeze type).

Properly designed and selected steam distribution tube coils distribute the steam throughout the entire length of all primary tubes, even when the leaving air temperature is controlled by modulating the steam supply through a steam-metering valve. Thus, more uniform leaving air temperatures are produced over the entire length and face of the coil than would result when using a single-tube coil.

Particular care in piping, controls, and installation is necessary to protect the coils from freezeup due to

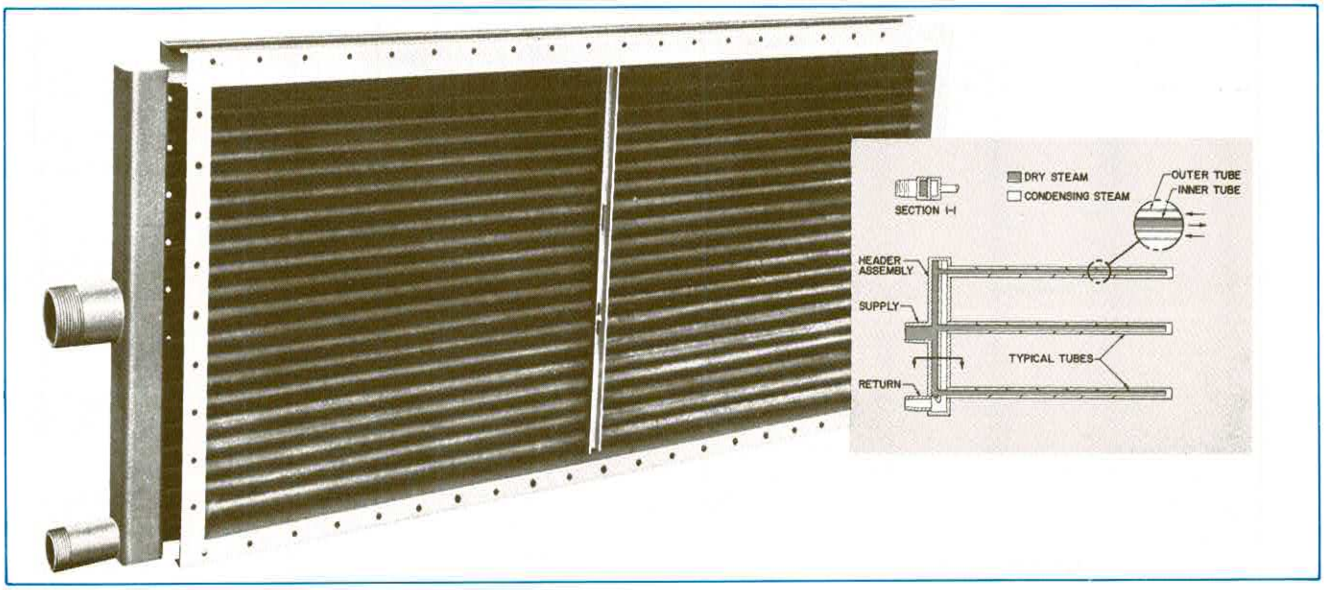


Figure 4-33 STEAM HEATING COIL (NON-FREEZE TYPE)

incomplete draining of condensate. When the entering air temperature is 32°F or below, it is best not to modulate the steam supply to the coil.

Coils located in series in the airstream, with each coil sized to be on or completely off in a specific sequence depending upon the entering air temperature, produce a system in which there is little likelihood for a freezeup.

The use of bypass dampers is also common. When less than full load conditions occur, air is bypassed around the steam coil with full steam being kept on the coil. In this system, care must be taken that high velocity jets of low temperature air do not impinge on the coil when the face dampers are in a partially closed position.

b. HOT WATER COILS

Comfort heating systems employing hot water usually require not more than one or two rows of tubes in direction of airflow in order to produce the desired heating capacity. To produce the most efficient capacity without excessive water pressure drop through the coil, various circuit arrangements are used.

When hot water coils are used with entering air temperatures below freezing, consideration should be given to piping the coil for parallel flow rather than counterflow. This will provide the highest water temperature on the entering air side. Coils piped for counterflow have the water entering the coil in the tube row on the leaving air side of the coil. Coils piped for parallel flow have the water entering the tube row on the entering air side of the coil.

c. COIL RATINGS

Steam and hot water coils are usually *rated* within these limits which may be exceeded for special applications:

(1) Air face Velocity. Between 200-1500 fpm, based on air at standard density of 0.075 lb per cu.ft.

(2) Entering Air Temperature. 20°F to 100°F for steam coils; 0°F to 100°F for hot water coils.

(3) Steam Pressures: From 2 to 250 psi at the coil steam supply connection (pressure drop through the steam control valve must be considered).

(4) Hot Water Temperatures. Between 120°F and 250°F.

(5) Water Velocities. From 0.5 to 8 fps.

d. COIL HEAT TRANSFER

In HVAC system design and TAB work, it is practical to consider that in a heating coil or heat exchange device that the heat rejected by one fluid is equal to the heat absorbed by the other fluid; and that the amount lost to the surroundings in negligible. Using the specific heat of water (C_p = 1.0 Btu/lb°F), the amount of heat transferred per hour can be obtained by using Equations 1-7 and 1-11 found in Chapter I. ($Q = 1.08 \times cfm \times \Delta t$ and $Q = 500 \times gpm \times \Delta t$)

e. COIL MEASUREMENTS

The air in HVAC systems is usually tempered by passing through a coil or series of coils, both heating and cooling, or it could pass through a heat exchanger such as a furnace or duct heater. The fan may either blow the air through the exchanger or draw it through. The primary concern of the TAB technician is to insure that the resistance or pressure loss across these units is not excessive and is not restricting the airflow. When these conditions are found, it is usually the result of dirt accumulation from running the system without any or with inadequate filters.

The TAB technician will need to obtain the design air pressure drop of the air across the coils from the manufacturer's submittal data. When dealing with cooling coils, the published air pressure drop will usually be for a wet coil, due to the formation of condensate on the cold surfaces. If the coil is dry the pressure drop will be considerably less.

3. Direct-fired Heat Exchangers

Most exchangers of this type that the TAB technician will encounter, will be duct furnaces, where hot gases from combustion are routed through sealed tubes, channels or a combination of both; although electric coils (Figure 4-34) fall into this category. These heat exchangers tend to have considerably less air pressure drop and therefore less tendency to clog up with dirt. There is also a type used for energy recovery systems where exhaust air going out of a space passes through one set of passages of the exchanger and outside air passes through another set of passages. The outside air is either pre-cooled or preheated by the exhaust air, whichever the case may be.

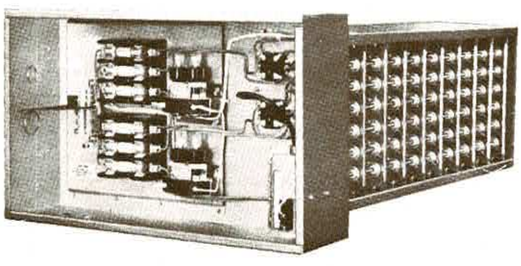


Figure 4-34 ELECTRIC DUCT COIL

4. Filters

All supply systems and some elaborate exhaust systems are equipped with filters. Their purpose is to filter out dirt and debris, and to protect the environment as well as protect the HVAC equipment, primarily the coils. They are available in many types and efficiencies. The less efficient type are usually a fibre glass mesh which may be coated with oil. The more efficient types are usually made of a fine blanket-like material which may be pleated or bag shaped. Since these are much more expensive, they are usually protected by using fibre glass pre-filters which trap most of the larger particles of dirt before they get to the secondary filter.

The TAB technician's main concern about filters is that they are clean and not creating a high pressure drop. The design pressure drop must be obtained from the manufacturer. Quite often, a "clean" and "dirty" pressure drop will be given. Also, some specifications will call for balancing with the filters in a dirty condition. Check the specifications when in doubt.

Checking the filters is an item that must be done before any testing is started. Quite often the filters will be clogged from construction dirt. Also, it is quite common to use some type of temporary blanket filter during construction to save the final filters. Visually verify that the filters are installed and are the proper type.

5. Volume Dampers

A large percentage of the TAB technicians time will be spent by adjusting volume dampers as they are one of the main ways to control the flow of air. A damper is a primary element in the duct system and is used for controlling airflow rates by introducing a selective resistance to airflow in the system. In older high pressure systems, dampers also were used as "pressure reducing valves."

Volume control or balancing dampers should be installed in each branch duct or zone duct. Single leaf dampers which are an integral part of a manufactured air grille do not meet the requirements. Opposed blade dampers which are a part of a manufactured register can be used as a "last resort" if there is not enough room for a regular balancing damper, and if sufficient space is provided behind the grille face for proper operation of the damper. Otherwise a balancing damper should be installed in the branch duct to the register. It should be accessible from the grille or diffuser opening, or a quadrant should be used and access provided.

Volume dampers installed in branch ducts where the total estimated static pressure is less than 0.5 in.w.g.

can be of a single leaf type. Volume dampers installed in ductwork where the total estimated system static pressure exceedsd 0.5 in.w.g. should be an opposed blade type.

a. MULTI-BLADE DAMPERS

Figure 4-35 shows two types of multiple blade dampers—the parallel blade and opposed blade dampers. The terms parallel and opposed refer to the movement of the adjacent blades. In the parallel blade damper, all of the blades move in parallel. The opposed blade damper has a linkage which causes the adjacent blades to move in opposite directions.

Partial closing of a damper increases the resistance of the duct system to airflow. The reduction in airflow with closure of the damper may or may not be proportional to the amount of adjustment of the damper. That is, closing the damper half-way does not necessarily mean that the air volume will be reduced to fifty percent of that volume which flows through the damper when it is wide open. The relation between the position of the damper and the percent of air that flows through the damper with respect to the airflow through the wide open damper is termed the "flow characteristic."

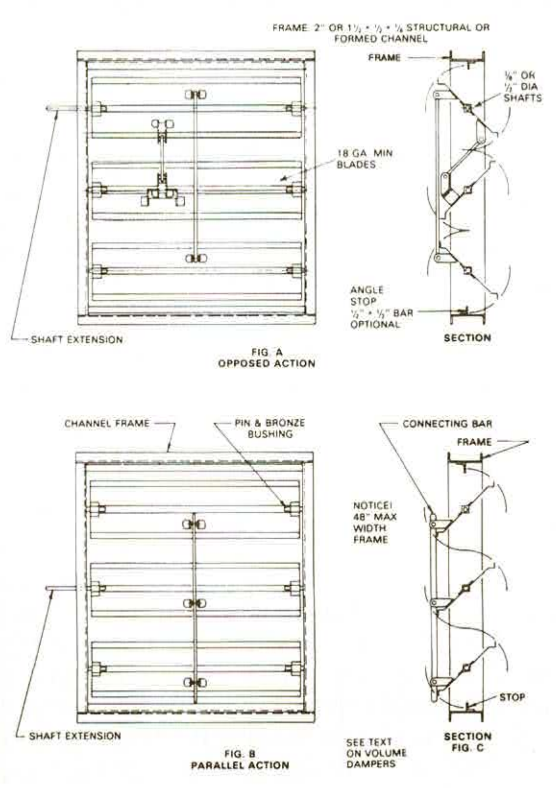


Figure 4-35 MULTI-BLADE VOLUME DAMPERS

Typical flow characteristic curves for the parallel blade and opposed blade dampers are shown in Figures 4-36 and 4-37. In Figure 4-36, the flow characteristic curves for the parallel blade damper show that as the damper is closed the flow reduction may be proportional to the closure of the damper as is shown by curve J, or partial closure of the damper may have little effect on the flow as is shown by Curve A.

The manner in which the damper reacts in the duct system is determined by how complicated the system is. If the system is very simple and the damper makes up a major part of the resistance in the system then any movement of the damper will change the resistance of the entire system and a good control of the airflow will result. If the damper resistance is very small in relation to that of the entire system, a poor flow characteristic such as curve A in Figures 4-36 and 4-37 result. In Table 4-2 typical ratios of damper to system resistance are shown for each flow characteristic curve.

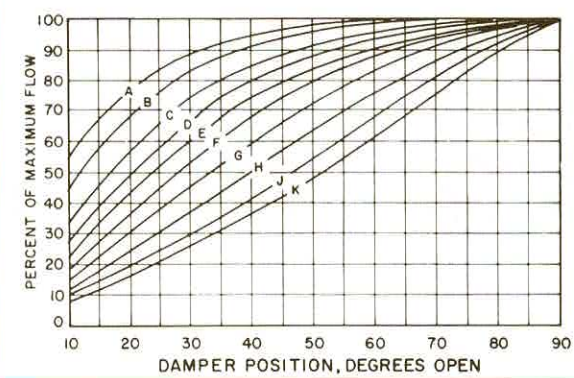


Figure 4-36 FLOW CHARACTERISTICS FOR A PARALLEL BLADE DAMPER

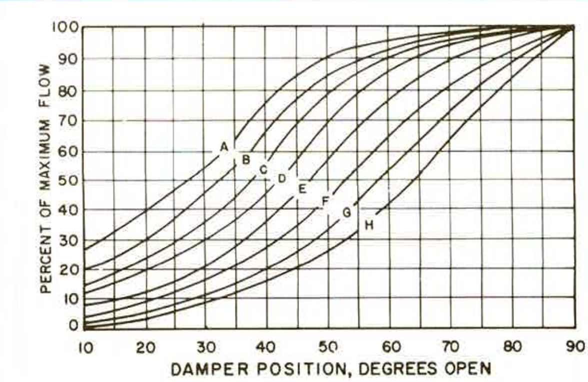


Figure 4-37 FLOW CHARACTERISTICS FOR AN OPPOSED BLADE DAMPER

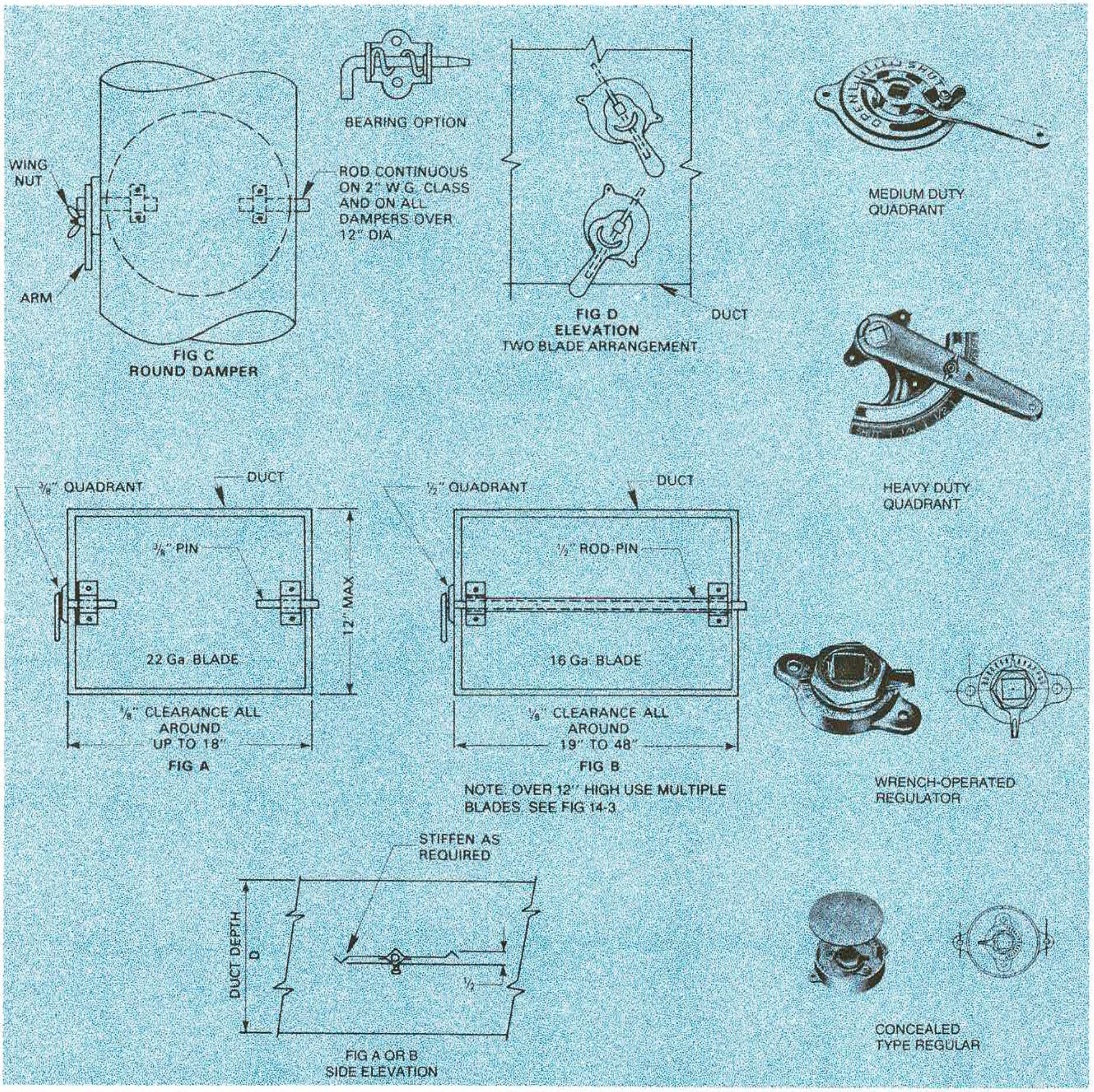


Figure 4-38 VOLUME DAMPERS

The set of curves for the opposed blade damper (Figure 4-37) show that for a given ratio of damper to system resistance a better flow characteristic usually results than with the parallel blade damper (Figure 4-36). The improved characteristic results from the fact that as the opposed blade damper is closed it introduces more resistance to airflow for a given position than does the parallel blade damper.

In balancing systems it should be realized that the flow characteristics of a damper are not constant and will vary from one system to another. The actual effect of closing the damper can only be determined by measurements in the particular system unless the system designer has taken into account the damper flow characteristics in his system design.

It is important to understand the airflow patterns of multi-blade dampers. The parallel blade damper has a tendency to throw the air toward one side of the

Table 4-2 TYPICAL RATIOS OF DAMPER TO SYSTEM RESISTANCE FOR FLOW CHARACTERISTIC CURVE

Parallel-leaf dampers		Opposed-leaf dampers	
Open damper resistance, percent of system resistance	Flow character-istic curve	Open damper resistance, percent of system resistance	Flow character-istic curve
0.5- 1.0	A	0.3- 0.5	A
1.0- 1.5	B	0.5- 0.8	B
1.5- 2.5	C	0.8- 1.5	C
2.5- 3.5	D	1.5- 2.5	D
3.5- 5.5	E	2.5- 5.5	E
5.5- 9.0	F	5.5-13.5	F
9.0-15.0	G	13.5-25.5	G
15.0-20.0	H	25.5-37.5	H
20.0-30.0	J		
30.0-50.0	K		

duct. This uneven pattern may adversely affect coil or fan performance or airflow into branch ducts if the damper is located closely upstream of any system component.

These flow patterns should be noted when it becomes necessary to measure airflow in a duct near a damper. Where possible, make any measurements upstream rather than downstream of the damper.

b. QUADRANTS AND LINKAGES

When dampers are located within ducts and are manually controlled, they are usually secured in place with regulators or quadrants. Varying in strength and locking ability, they should be of suitable size for the size damper with which they are used. When setting a damper, the regulator or quadrant must be securely tightened to assure that the damper remains as set.

Do not always accept the position of the regulator pointer as an indication of the actual position of the damper blade. If there is any doubt, inspect the end of the damper rod at the face of the regulator. A groove, as cut by hacksaw, will indicate that the damper blade runs in the same direction as the cut. If still in doubt, verify visually or by taking static pressure readings while moving the quadrant or lever.

Where dampers should have tight shutoff when closed, the linkage between blades must be properly adjusted. Damper motor linkage also must be properly adjusted. Cold deck and hot deck dampers, as used in multi-zone units, must close tightly, as must face and by-pass dampers used in some air handling units.

D TERMINAL DEVICES

1. Terminal Units

A terminal unit is a device or unit, often a box, that is located where the supply duct or branch duct terminates and the air is introduced into the space to be conditioned. There is a wide variety of terminal units; some contain only air control dampers or valves, while others may have cooling or heating coils. They are designed for various functions, such as to regulate the quantity of air, to regulate its temperature, or both.

Terminal units can be single duct, dual duct, or induction type units, and can be located either in or above the conditioned space. The terminal unit has several functions. It must supply air at a proper temperature to take care of the load in the conditioned space. This is done in response to a room thermostat located in the space. The unit also contains some type of device to regulate the airflow to the space. Pressure is reduced in the terminal unit to a level where the air can be introduced into the space. Any noise that is generated within the unit in the reduction of the pressure must be attenuated within the terminal unit. Terminal boxes are rated to deliver a specified airflow at a specified pressure drop.

Each terminal box has a specific static pressure range required to overcome the box pressure losses plus the losses of any discharge ducts, outlets, etc. The minimum inlet static pressure requirement will be the total of published minimum pressure drop and the downstream supply duct static pressure requirements. When testing a system or terminal box to find out if enough static pressure is available to deliver the specified airflow, the calculated or measured supply duct static pressure drop must always be taken into account and added to the published minimum pressure drop of the terminal.

These units or boxes may be pressure dependent, pressure independent, constant volume, variable volume, fan powered or a combination of two or more. In other words, it's not uncommon to have a single duct, pressure independent, variable volume box with a reheat coil. Listed below is a description of each of these possible features.

a. SINGLE DUCT UNITS

This is a terminal box with a single inlet duct. They are usually supplied with cold air and use reheat coils if heat is needed. They may be constant volume or variable volume and they may be pressure dependent or pressure independent.

b. DUAL DUCT UNITS

These boxes have a hot duct and a cold duct entering the box. A mixing damper arrangement is controlled by a space thermostat. This gives the system the ability to deliver hot or cold air, or a combination of both as needed. These boxes are always pressure independent and may be constant or variable air volume (VAV).

c. PRESSURE DEPENDENT UNITS

A pressure dependent terminal box is usually a box with only one damper in it. The damper may be manual or it may be controlled automatically by a thermostat, as in a variable air volume (VAV) application. The damper will not change positions unless done manually or by the thermostat signal. Therefore, if the inlet static pressure to the box changes, the airflow passing through the box will also change. Inlet static pressure will vary considerably on dual duct and VAV systems. With a pressure dependent terminal box, the amount of air passing through the box is dependent on the static pressure at the box inlet. Pressure dependent boxes are usually only used on single duct applications. They may be constant or variable volume.

d. PRESSURE INDEPENDENT UNITS

The term "pressure independent" refers to a terminal box that will pass the same quantity of air, regardless of the static pressure at the inlet duct to the box (within design limits). Suppose a manufacturer rates a pressure independent box for 500 cfm with a minimum pressure drop of 0.25 in.w.g. and a maximum pressure drop of 6.0 in.w.g. This means that as long as the inlet static pressure is within these limits, the

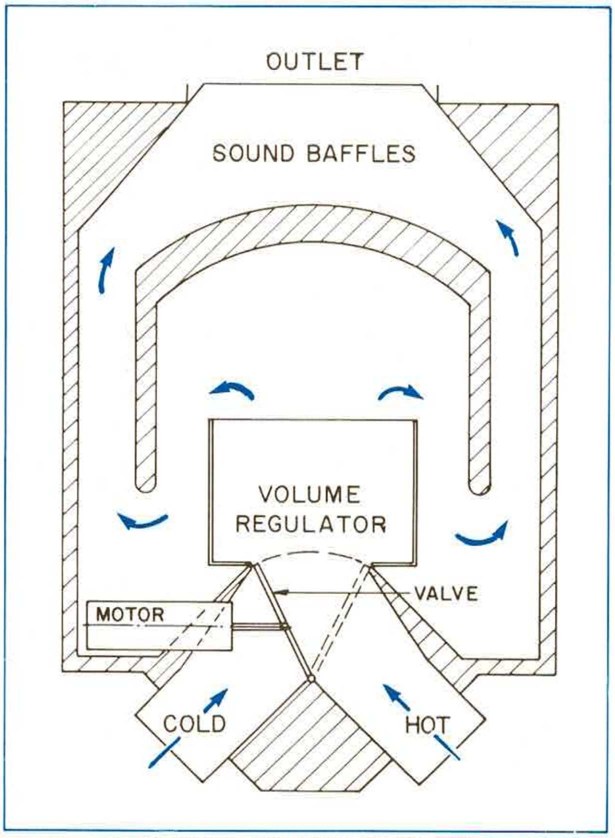


Figure 4-39 DUAL DUCT UNIT WITH SINGLE MOTOR ACTUATED BY THE ROOM THERMOSTAT TO SUPPLY WARM OR COOL AIR

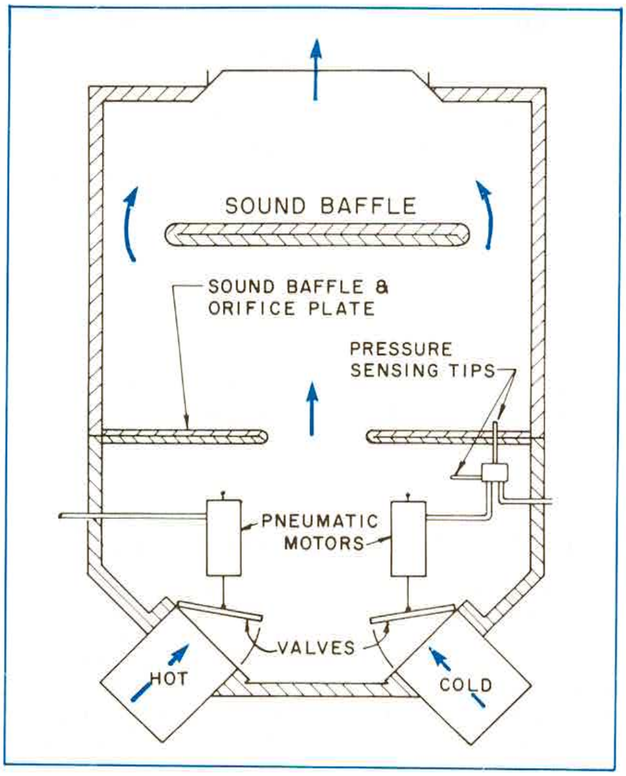


Figure 4-40 DUAL DUCT UNIT WITH TWO MOTORS POSITIONED TO MAINTAIN THE SPACE TEMPERATURE BY A PNEUMATIC RELAY

box will deliver a constant 500 cfm. This task is accomplished by an automatic constant volume regulator, which also can be variable if desired. Hence, the term, "pressure independent" meaning that the terminal box will deliver the same airflow independently of the inlet static pressure.

Pressure independent boxes are usually preset for the correct cfm at the factory. Experience has shown that many will need resetting in the field. The airflow regulator may be system powered or have pneumatic, electric, or electronic controls. The system powered controllers use the velocity and/or the static pressure in the duct system to operate the regulator. A choice of curtains, flapper doors, or bellows may be used in a combination with springs. This combination will form a type of damper that will react and close down as the velocity increases and maintains a constant airflow through the box. The pneumatic, electric and electronic airflow regulators operate in a similar fashion, but have different power mediums.

A velocity sensor is located in the box, usually near the inlet. It can be similar to a Pitot tube or an electronic type using a thermistor or pressure transducer. A signal is sent from the sensor which is monitoring the quantity of air passing through the box to the controller. The controller has adjustments to determine the design airflow. The controller compares the actual cfm signal from the sensor to the pre-set point. If the two are not the same, the controller will send a signal to the damper to either open or close, whichever is required, to obtain the required cfm. On VAV systems, the controller will have a minimum and a maximum setting, and a signal from the space thermostat will determine how much cfm will be delivered within the design airflow limits.

e. CONSTANT VOLUME UNITS

As the term implies, a constant volume terminal box delivers the same airflow (cfm) constantly. The box may be dual duct or single duct.

f. VARIABLE AIR VOLUME (VAV) UNITS

This is currently the most used type of terminal unit. They are available in most any combination of pressure dependent, pressure independent, single duct, dual duct, induction and fan powered. The basic VAV box will have a single duct and a damper. In the pressure dependent version, the damper will be just thermostatically controlled. The pressure independent version will have a regulator to limit the maximum/minimum cfm to within a pre-set limit.

The main feature of the VAV box is its ability to vary the delivered airflow as required by the heat load in the space. The cfm will vary from a design maximum cfm down to a minimum cfm which may be as low as zero (shutoff) or as high as 80%. Common practice is a low of somewhere around 25% to 30%. This is especially true when reheat coils are used on the box discharge.

The dual duct versions are available with many sequences of operation. Most have the cold duct throttle down to under 50% before the hot duct begins to open. With the many sequences available, you must obtain the correct data and know how the terminal box is supposed to work before you will be able to properly balance it.

Additional information on the operation of VAV boxes (both pressure dependent and pressure independent) may be found in Section E—"HVAC Duct Systems" of this chapter.

g. FAN POWERED UNITS

These boxes have a fan and an inlet duct from the space, which is usually from the return plenum ceiling. The fan recirculates air from the return inlet and into the supply system. Primary air, usually cool air, is introduced from the main system, either on the inlet or discharge side of the fan, to mix with the recirculated air. The primary air is usually pressure independent, with the space thermostat controlling how much primary air is introduced. A system of dampers, backdraft and/or motorized, control the airflow and mixing of the air. The fan may run all the time or it may shut off or vary in speed as the primary air cfm varies.

The most common sequence will have the primary air at 100% and the fan off when full cooling is called for. When there is a demand for heat, the thermostat signal will begin to reduce the primary air. As this happens, the fan will start and return air will mix with the reduced primary air. The primary air will decrease to its minimum setting (which may be shut off), but the fan will keep recirculating air into the spaces.

The most common application of these boxes is around the perimeter of large buildings where they include a reheat coil on the box discharge. Stagnation is eliminated and heating is available even if the primary air unit is off, such as during unoccupied times. Here again, the TAB technician must obtain the correct sequence of operation and fully understand the design mode of operation before the fan powered units can be balanced.

The latest version of these boxes have a built-in digital computer. Temperature, airflow and fan speed are controlled electronically. Each job is furnished a plug-in adaptor for reviewing the computer program and to assist in setting up the box.

h. INDUCTION BOXES

The induction box uses primary air forced through a venturi at a high velocity into a plenum. A low pressure area forms around the venturi discharge which will induce the air around the venturi to mix with the primary air coming through the venturi for a total cfm greater than the primary air. The primary air is usually pressure independent and may be constant or variable volume. The induced air is usually controlled with an automatic damper which is thermostatically controlled and the damper may be interlocked with the primary air on VAV type systems. Once again, the TAB technician must have the correct operating sequence before the balancing can be performed.

i. TERMINAL INDUCTION UNITS

The induction unit (Figure 4-41) is usually a single duct unit which is supplied with primary air at a constant temperature. The primary air, at high pressure, is supplied to a nozzle within the unit. The air discharges from the nozzle behind a coil and creates a low pressure region. The low pressure region causes air to be induced from the room and this is pulled through a water coil mounted on the unit. The water coil, in response to a room thermostat, is supplied with heated or chilled water to maintain a mixture of primary and induced air that will satisfy the load within the space and maintain a constant air temperature in the conditioned space.

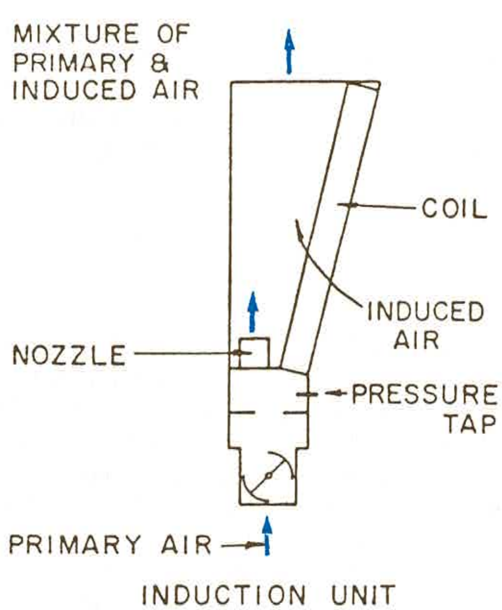


Figure 4-41 SINGLE DUCT INDUCTION UNIT SUPPLIED WITH PRIMARY AIR AT CONSTANT TEMPERATURE

A damper or pressure reducing valve in the inlet of the unit is adjusted at the time the system is balanced to maintain the proper quantity of primary air. A pressure tap is provided on the unit so that the quantity of primary air can be determined from a pressure measurement which is referred to a calibration curve supplied by the manufacturer. The unit is also provided with a means for sound attenuation. Induction units are usually used around the perimeter of a building, preferably under windows where they handle most of the heating load.

j. BYPASS BOXES

These boxes are usually used on smaller less expensive systems. The primary air comes into the box and can exit either into the supply system to the conditioned space or be discharged out into the plenum return ceiling. The diverting damper will be closed or opened on demand by the space thermostat. These boxes can be used with a conventional constant volume HVAC unit, but still provide VAV to the conditioned space. Manual balancing dampers are usually provided on the box inlet or the outlet and bypass opening.

2. Air Outlets and Inlets

The purpose of the HVAC system is to introduce and circulate conditioned air while removing the stale air, with the objective of making the environment more healthful and comfortable for the inhabitants. Supply air outlets are the final step on the path that the air travels to the conditioned space, and they have much influence on the comfort level within the space. Supply air outlets must distribute the air into the space without causing objectionable drafts and be reasonably quiet while doing so. They must be able to mix the supply air with the air already in the space and maintain reasonably even temperature throughout.

Furnishing the proper type and size supply air outlet and correctly adjusting and balancing it, is necessary for a satisfactory job. Since TAB technicians will have little to do with the selection or location of the outlets, they must confine themselves with the proper adjusting and balancing of the various outlets that are encountered. More detailed information on room air distribution devices may be found in the NEBB "Environmental Systems Technology" textbook.

TAB technicians will be required to adjust the quantity of air (cfm) from the outlet and to adjust the pattern of the airflow. The required cfm usually is shown on the mechanical drawings. The direction of airflow may or may not be shown.

TAB technicians will have to try to keep the following in mind when determining air patterns. The basic rule

Table 4-3 SUPPLY AIR OUTLET PERFORMANCE

				Characteristics	
Group	Type	Mounting	Discharge Direction	Cooling	Heating
A	High Sidewall Grilles Sidewall Diffusers Ceiling Diffusers Slot Diffusers (Parallel Flow)	Ceiling High Sidewall	Horizontal	Good mixing with warm room air. Minimum temperature variation within room. Particularly suited to cooling applications	Large stagnant air area near floor. In interior zones where loading is not severe, stagnant air area is practically non-existent
B	Floor Grilles Baseboard Units Fixed Bar Grilles Linear Grilles	Floor Low Sidewall Sill	Vertical Non Spreading Air Jet	Small stagnant air area generally above occupied zone	Smaller stagnant air area than Group A outlets
C	Floor Grilles Adjustable Bar Grilles Linear Diffusers	Floor Low Sidewall Sill	Vertical Spreading Air Jet	Larger stagnant air area than Group B outlets	Smaller stagnant air area than group B outlets — particularly suited to heating applications
D	Baseboard Units Grilles	Floor Low Sidewall	Horizontal	Large stagnant air area above floor in occupied zone — not recommended for comfort cooling	Uniform temperature throughout area. Recommended for process applications
E	Ceiling Diffusers Linear Grilles Grilles Slot Diffusers (Vertical Flow) Sidewall Diffusers	Ceiling High Sidewall	Vertical	Small stagnant air area near ceiling. Select for cooling only applications	Good air distribution. Select for heating only applications
F	Variable Area Grille Variable Area Diffuser	Ceiling High Sidewall	Horizontal Specially adapted for variable volume systems	Maintain design air distribution characteristics as air volume changes	Maintain design air distribution characteristics as air volume changes

about "hot air rising because it is lighter than cool air" must be remembered. Cooled air is best distributed from high in conditioned spaces, such as the ceiling. If it is distributed from low sidewall or floor registers, the outlets should be adjusted to direct the air up as much as possible, since it will naturally start falling as soon as it loses its velocity.

The opposite is true for hot air when being distributed from high locations. Since most systems handle both hot and cold air as the seasons change, a compromise must be made, and it usually favors the cooling season and the designer uses ceiling supply outlets. For most installations of this type, the TAB technician usually will adjust the outlets for a horizontal flow pattern. When high ceilings are involved, it may be necessary to use a vertical pattern in order to force the hot air down to the occupied level of the space. When linear diffusers are used around the perimeter of a building, it is common to have at least one slot of the diffuser blowing air down along the vertical window or wall surface. The other slots, if used, may blow either way, depending on the designers specifications.

The TAB technician undoubtably will be called back to the job occasionally for complaints of too much air or not enough air. Although the airflow may be cor-

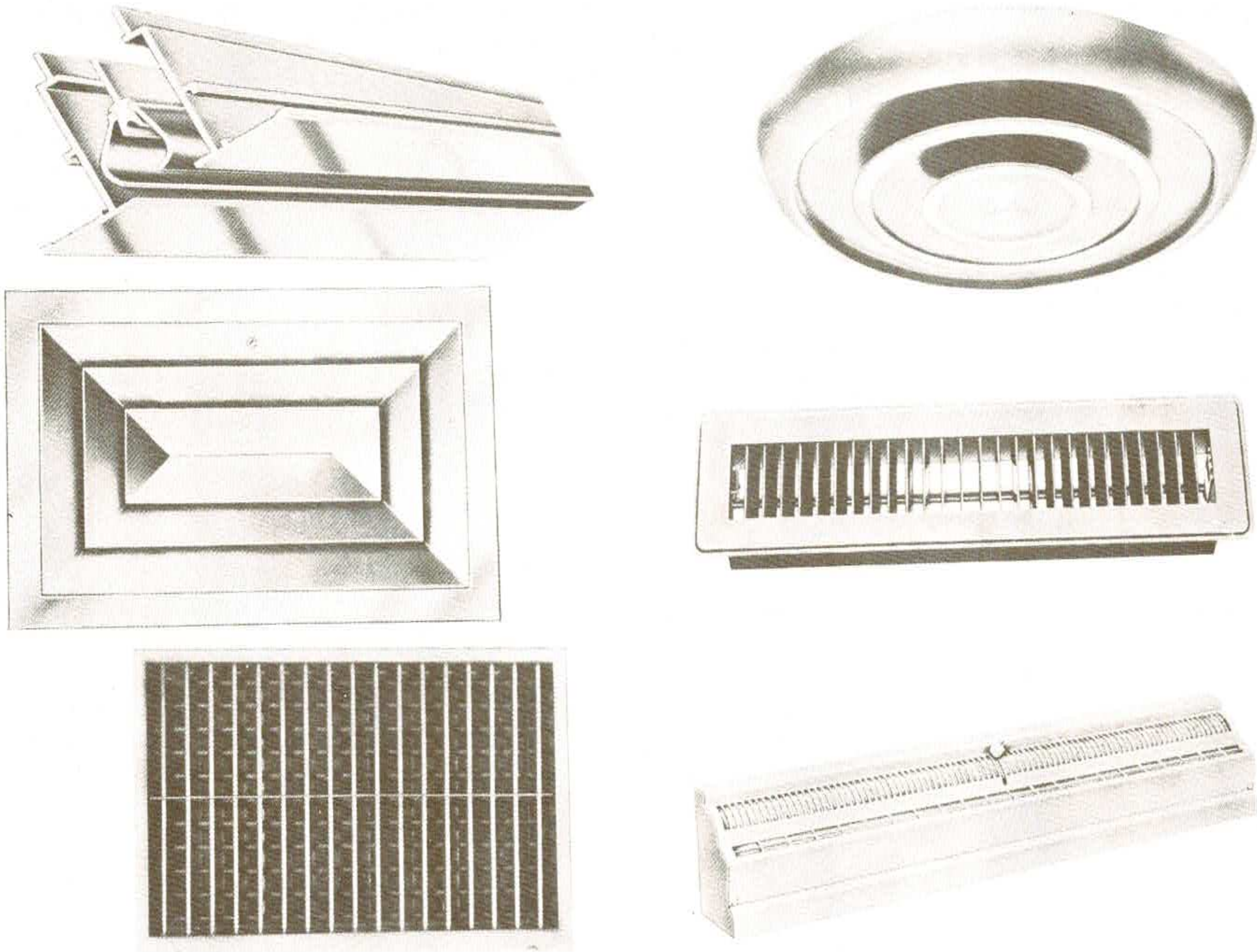


Figure 4-42 TYPICAL SUPPLY OUTLETS

rect, the occupant may be uncomfortable due to drafts or stagnation. By making simple adjustments to the air pattern, quite often you will be able to make the occupant comfortable.

Return inlets are used to remove air from the space. They may be used either for return air or exhaust air. Some supply outlets are used as inlets, and it is quite common to use the ceiling as a return plenum. Then slots in the lights, egg crate openings, perforated panels or grilles may be mounted in the ceiling as a path for the air to get to the ceiling plenum. Return inlets tend to be less numerous and their locations can be less critical than supply outlets in some areas of the building.

Inlets and outlets are available either with or without dampers built in or attached. If none are provided, dampers should be installed in the connecting ducts. Listed below are some of the various types of inlets and outlets with some of their features:

a. REGISTERS AND GRILLES

A register is a grille with a built-in or attached damper. They are available with horizontal bars, vertical bars or bars running both ways. Quite often these bars are adjustable so that the flow pattern can be directed either or both ways. Return and exhaust registers usually have bars only one way (single deflection) and quite often they are fixed in a 45° position. Registers and grilles are used in ceilings, walls and floors, both high and low, although they are not usually recommended for supply use in the ceiling. They are suited for the delivery of larger quantities of air, and with double deflection bars, they are extremely flexible and well suited for sidewall installations. Heavily constructed models are available for use in floors or sills. When balancing supply registers, the TAB technician must be aware of the airflow pattern adjustments, as they are more critical than with ceiling diffusers.

b. CEILING DIFFUSERS

A round diffuser is especially well suited for air delivery equally in all directions. Square and rectangular diffusers tend to direct the air in the four directions, but they still usually perform quite well. Square and rectangular diffusers are readily adaptable to either one-way, two-way or three-way blow, which is advantageous in many locations. They also are more compatible with many of the current ceiling systems, particularly the lay-in tile systems.

Ceiling diffusers do a good job of providing a horizontal flow air pattern that tends to flow along the ceiling before disbursing and dropping. This is advantageous for cooling systems, but it does tend to induce "smudges". As the air discharges from the diffuser, it will entrain surrounding room air which may be dirty. This dirty air then is deposited as smudge on the ceiling. Anti-smudge rings are available to help eliminate this problem. Many of these diffusers are available with deflectors that are adjustable and can be raised or lowered so that a vertical or horizontal discharge pattern can be used.

c. PERFORATED FACE TERMINALS

Perforated face diffusers are one of the more popular types in use today. They are especially well suited

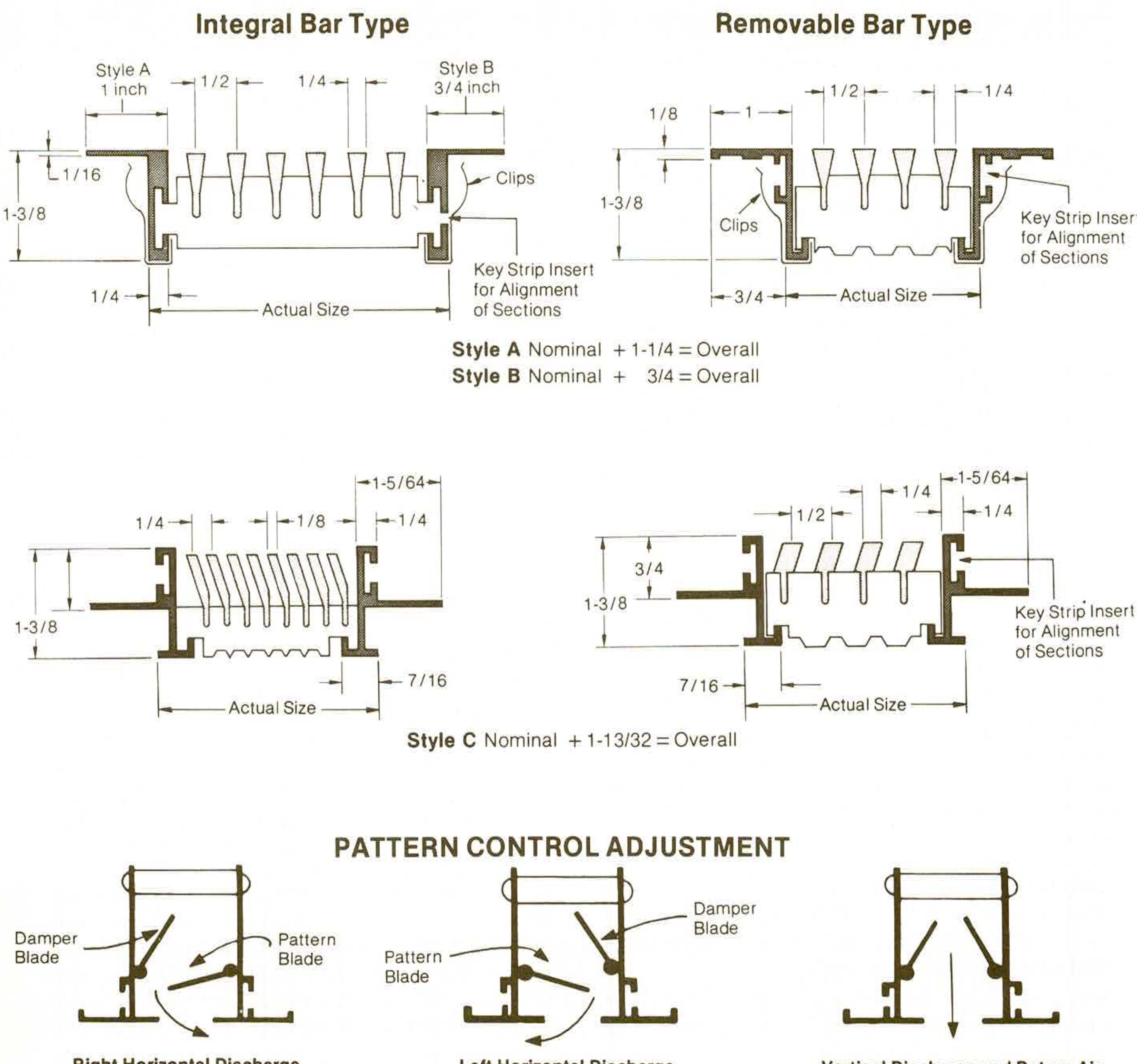


Figure 4-43 EXAMPLES OF SLOT TYPE DIFFUSERS

for lay-in tile ceilings and they are considered to be more attractive by many architects. In a supply application, they are quite similar to a conventional flush mounted diffuser with the added perforated face installed over it. They do not deliver a true horizontal pattern as conventional diffusers do. However, the perforated face allows the use of supply, exhaust or return terminals that match and look alike for a more uniform ceiling.

For ceiling return plenums using lay-in tile ceilings, panels made of perforated metal are available for use as return inlets. Although most of the manufacturer's publish AK factors as well as procedures for testing perforated terminals, field experience has shown that the only consistently accurate way to test perforated diffusers in the field, is with a flow measuring hood.

d. LINEAR AND SLOT DIFFUSERS

The longer linear diffusers have been around for years and are well suited to match various architectural schemes. With the increasing use of VAV systems using very low minimum airflow, the use of four foot individual slot diffusers with built in plenums arranged for flex duct connections have become popular. Other lengths are also available, but the four foot length is most compatible with many ceiling systems. The long horizontal discharge has an advantage called the "Coanda effect." The air discharges along the ceiling and entrains room air, which reduces the "dumping" or rapidly falling action of cold air that happens when the supply air leaves the diffusers at very low velocities. Linear and slot diffusers are available with any number of slots and in one or two-way blow patterns.

e. LIGHT TROFFERS

Light troffers are devices that fit over a florescent lamp fixture and deliver air through a slot along the edges of the lamp assembly. They have a relatively inexpensive first cost.

They are available as either single or double units, meaning that supply air is available on either or both sides of the lamp fixture. The supply air units are not usually installed on every lamp fixture, and the units without the duct connections serve as a path for the return air to flow into a ceiling return plenum. Architecturally, they are relatively unnoticable. For the TAB technician, they make it difficult to get accurate airflow readings. They tend to fit loosely to the lamps and leak air back into the ceiling space.

f. PLENUM CEILING SYSTEMS

A supply air duct system will discharge air into a ceiling plenum. Openings in the form of slots in the ceiling tile will be used along with regular ceiling diffusers. In theory, the air should travel throughout the ceiling from the ducts to the outlets and flow into the space. In actual practice, it has been found that it is very difficult to construct these plenums airtight. Leaks are very common and poor air distribution is the result. TAB technicians will have to balance the duct system to obtain the correct cfm into the ceiling plenum. This is usually done from a duct system with several openings and dampers above the ceiling. TAB technicians will then have to extensively check for leakage if the system does not perform properly. Usually, pressure drops across the ceiling will be the only way to determine if there are leaks. The title manufacturer's provide data to determine the airflow through the tile slots based on pressure differential. The popularity of these systems has diminished considerably in the last few years.

E HVAC DUCT SYSTEMS

1. Single-Zone Systems

The simplest form of the air duct system has a single fan-coil unit serving a single temperature-control zone. The unit may be installed within or remote from the space is serves, and may operate with or without distributing ductwork. Well-designed systems can maintain temperature and humidity closely and efficiently and can be shut down when desired without affecting the operation of adjacent areas.

The air is distributed directly from the unit into a single duct system that supplies all of the terminal outlets. The air may be heated and/or cooled. The return air fan shown in Figure 4-44 adds the capability of using 100 percent outside air for "free cooling" when the outside air temperatures are appropriate and cooling is desired. The return air fan then serves as an exhaust air fan by exhausting the return air to the outside. In usual practice, the outside air coming into the HVAC unit is blended with the return air in the correct proportion to provide mixed air at a predetermined temperature. This is called an "Economizer" cycle. On other systems, the return air fan will be omitted entirely. Usually, the quantity of outside air will then be constant at all times or as in residential systems, the outside air feature may not be used at all, and the system will only recirculate air. If an economizer cycle is provided on a system without a return air fan, a relief damper must be provided, either in the return duct or in the space.

Control of the single-zone system can be affected by varying the quantity of cooling medium, providing reheat, face and bypass dampers or a combination of

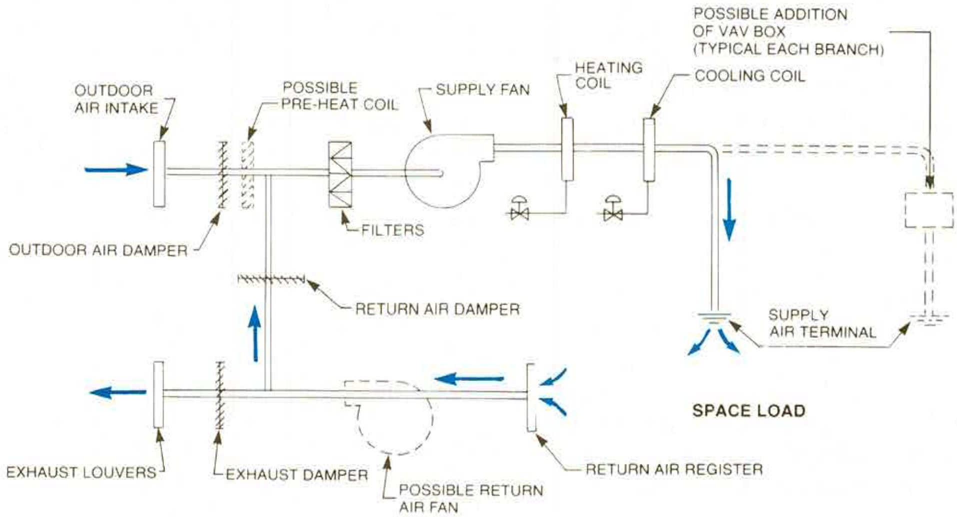


Figure 4-44 SINGLE DUCT SYSTEM

these. The single-duct systems with reheat satisfy variations in load by providing independent sources of heating and cooling. When a humidifier is included in the system, humidity control completely responsive to space needs is available. Since control is directly from space temperature and humidity, close regulation of the system conditions may be achieved. Single-duct systems without reheat offer cooling flexibility but cannot control summer humidity independent of temperature requirements.

2. Terminal Reheat Systems

The reheat system is a modification of the single-zone system. Its purpose is: (1) to permit zone or space control for areas of unequal loading, (2) to provide heating or cooling of perimeter areas with different exposures, or (3) to promote process or comfort applications where close control of space conditions is desired. As the word *reheat* implies, the application of heat is a secondary process being applied to either preconditioned primary air or recirculated room air.

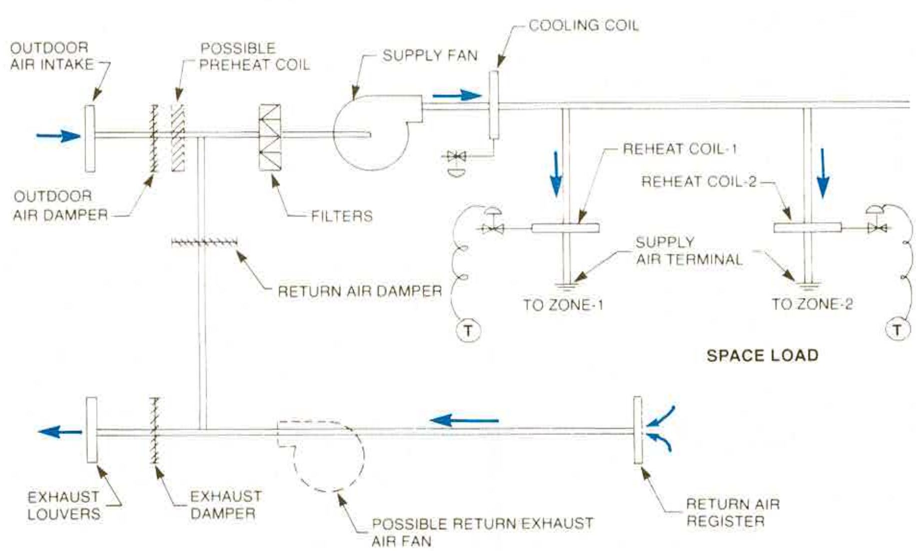


Figure 4-45 TERMINAL REHEAT SYSTEM

In terminal reheat systems (Figure 4-45), the main supply air duct will split into two or more zones, and each zone will have its own reheat coil and space thermostat. The supply air from the unit is discharged at a lower temperature than needed in the space. The air is then reheated in each zone to provide the correct temperature air to temper the space in each zone. Each zone will usually have its own balancing damper and the balancing procedure will be identical to other constant volume systems. Due to the inefficiencies involved in first cooling air and then reheating it, these systems are not being used in new buildings except where humidity control is required.

3. Multizone Systems

The multizone system is used to heat and/or cool a number of zones from a single HVAC unit. The requirements of the different zones are met by mixing cold air and warm air through zone dampers at the central unit in response to zone thermostats. The mixed hot and cold air is distributed throughout the building by a system of single-zone ducts. Either packaged units complete with all components or field-fabricated apparatus casings may be used. The return air is usually handled in a conventional manner (as shown with the single-zone system).

A multizone unit uses one fan that can blow air through two paths, usually a cooling coil and a heating coil, before being discharged from the unit. The air passes through each coil into a cold air plenum or a hot air plenum, respectively. From these plenums, air passes through mixing dampers into two or more zones serving various spaces. Each zone is connected to both the cold and hot air plenums, with an automatic damper at each connection. These dampers are controlled in such a way that one is fully open when the other is fully closed (Figure 4-46).

When the thermostat on any particular zone calls for cooling, the cold air damper opens and the hot air damper closes. As the zone thermostat becomes satisfied, the cold air damper will begin to close and the hot air damper will begin to open, allowing the air to mix to satisfy the requirements. As the zone calls for maximum heat, the cold air damper will close completely and the hot air damper will open fully to allow only hot air into the space.

Each multizone HVAC unit will have two or more zones coming off the plenum, serving different spaces. In each zone duct, usually close to the unit, a manual volume damper must be used also. This damper is used to balance the airflow to each zone, since the mixing dampers are not capable of controlling total airflow quantity. Some units, often called "Texas multizone units," will not have a heating coil, and will bypass the air through a resistance of some type when cooling is not needed.

Multizone systems normally are balanced with all of the zones in a cooling position. There are exceptions, such as when the cooling coil is designed for less air than the fan delivers and the system requires. This is done because the building will not normally need full cooling in all zones at the same time. This diversity

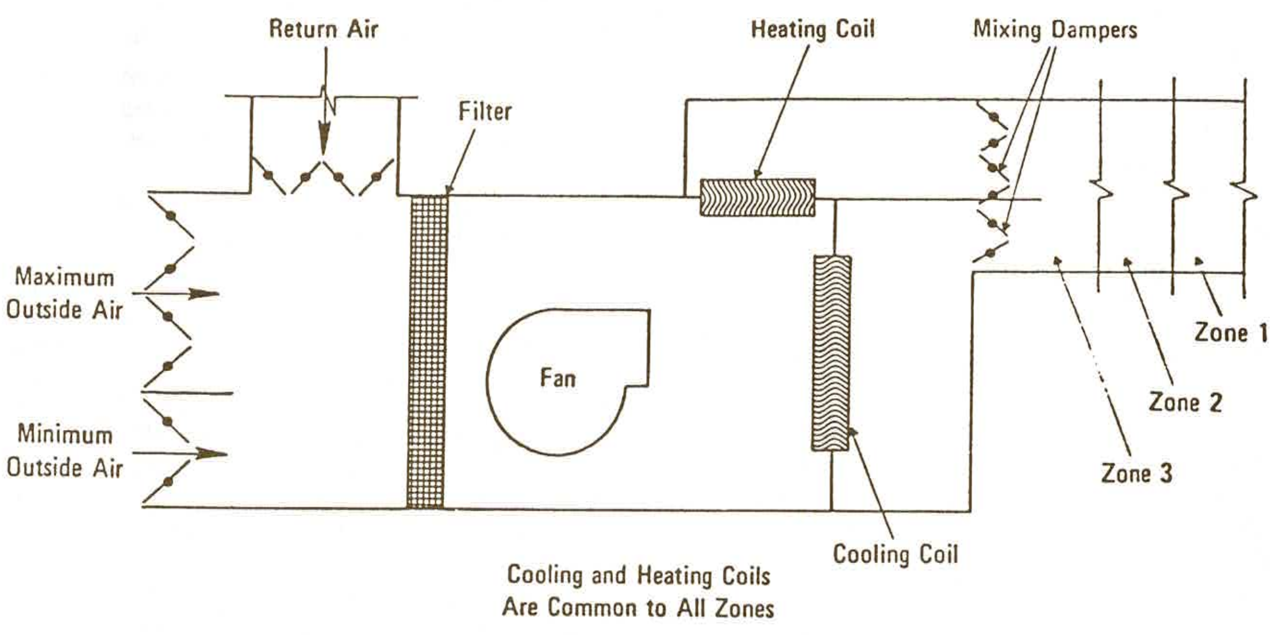


Figure 4-46 MULTIZONE UNIT

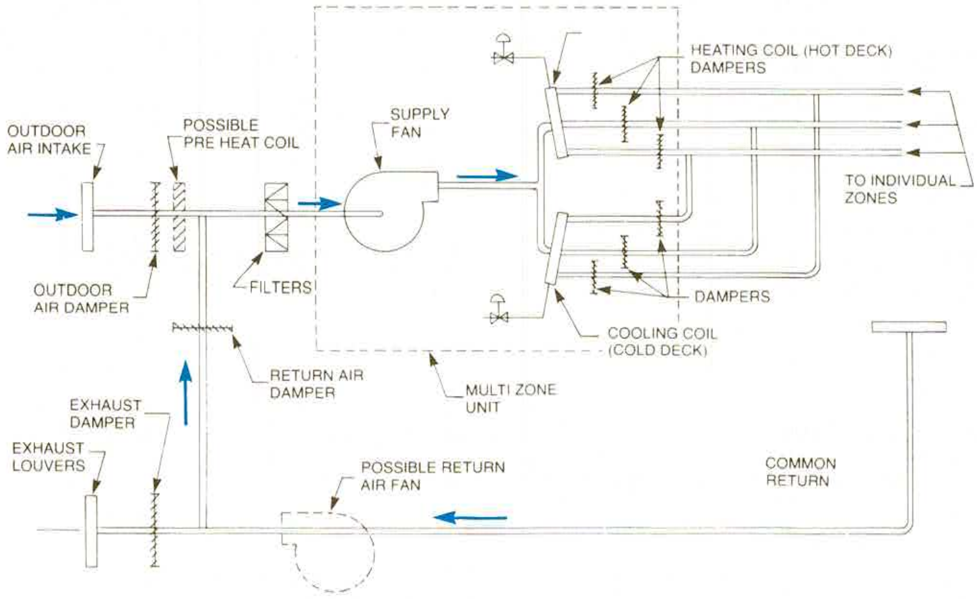


Figure 4-47 MULTIZONE SYSTEM

is caused by sun load changing from east to west, etc. It will be necessary to check the manufacturer's data to determine if the cooling coil is sized for full flow or if diversity has been designed into the system, so you will know what procedure to take while balancing.

4. Constant Volume Systems with Terminal Boxes

Constant volume systems use both pressure dependent and pressure independent boxes. Pressure dependent constant volume boxes are essentially an attenuator box containing a manual damper. The system may operate at low, medium or high pressures. The supply air comes from the unit and is ducted to two or more boxes. The boxes may have reheat coils, in which case the system would become a terminal reheat system. The system air pressure is reduced in the boxes and then the air is distributed through the downstream duct system to the outlets.

Constant volume systems that use pressure independent boxes are identical to those with the pressure dependent boxes, except that instead of using manual dampers, an automatic constant volume regulated damper is utilized. The regulated damper will maintain the same constant air flow, regardless of the box inlet pressure, as long as the system air inlet pressure is within the box manufacturers published limits.

Pressure independent boxes usually are used in systems with special applications. Their advantages are that they can be mixed into a variable air volume (VAV) system and still maintain constant volume where needed; and that in some critical flow situations, like laboratories, they are capable of delivering a constant airflow volume, even as the system filters get dirty and regardless of whether a cooling coil is wet or dry.

5. Variable Air Volume Systems

(See Section D—"Terminal Devices" of this chapter for details of terminal units or boxes.)

To maintain control of the temperature in a space, many system designers are now using the variable air volume (VAV) concept. This means that the airflow (cfm) can be varied (as the load varies), all of which is controlled by a thermostat in the space. As the VAV boxes vary the airflow to the spaces, control of the fan airflow must be provided to match the system cfm and static pressure to the total system requirements.

Most VAV systems use a single duct to supply the boxes, but there are variations which use dual ducts (hot and cold) and these are covered later in this chapter. Fan configurations used are similar to that described for other low pressure systems with one major difference, that being the ability to control the system airflow volume. Airflow control can be accomplished with inlet or discharge dampers, variable speed drives or variable speed motors. A static pressure sensor, preferably located about two-thirds of the way down in the duct system, senses the duct static pressure and sends a signal back to the ap-

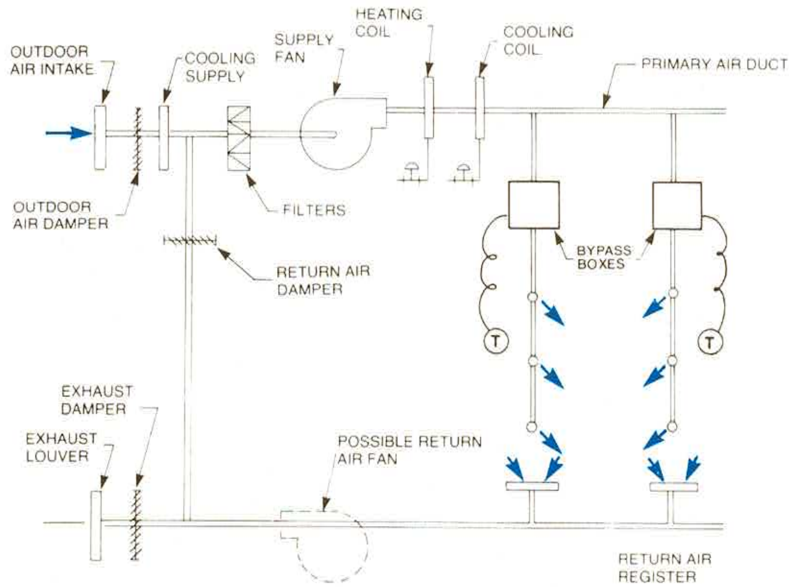


Figure 4-48 VARIABLE VOLUME SYSTEM

paratus controlling the fan volume. The controls are set to maintain a constant static pressure at the sensor location, as the airflow varies up and down. This variation of airflow is a result of the load demand caused by the terminal boxes to satisfy the zone space temperatures.

Each VAV box has some form of automatic volume control damper that is controlled by a thermostat sensing space temperature. The fan must track the system airflow demand within the same cfm range that the fan is designed for. The total terminal airflow requirements should equal the fan design cfm; and this can be accomplished by setting thermostats controlling each box well below the space temperatures. If the system has a "diversity", meaning that the total terminal airflow (cfm) requirements exceed the fan capacity, you then must set only those terminal boxes into a full flow condition that have a total airflow that equals the design cfm of the fan. This means that some of the terminal boxes on the system will be in a minimum flow condition. Be sure to spread those terminal boxes set at minimum flow throughout the system and not have them all on one portion of the system. After ascertaining that the system (as operating) airflow is equal to the fan design cfm, the fan can be tested using normal procedures.

What is known to the trade as a conventional VAV system has VAV terminal units or boxes. The function of these boxes is to reduce the system air pressure from the inlet duct to a lower pressure at the discharge duct, and to vary the cfm to meet the demand of the space served. A thermostat usually located in the space, senses the space temperature and varies the cfm discharging from the box. VAV systems are available with many different features and new items are appearing daily. Already they have become the most popular type of system for commercial work. So it is a system with which the TAB technician must be well versed and kept up to date.

a. PRESSURE INDEPENDENT VAV SYSTEMS

Pressure independent boxes have the ability to maintain a constant maximum airflow and usually a constant minimum airflow also, as long as the box inlet static pressure is maintained within the design range of the VAV box. A feature of the box is that the inlet static pressure can vary within the design limits of the box, while the discharge cfm will remain the same under any given condition. The manufacturers publish data for their boxes which details the static pressure operating range and the minimum static pressure drop across each terminal box for a given cfm. Consult this data to verify that an adequate static pressure is available for the terminal box to function properly.

A VAV system using pressure independent boxes, when properly designed, installed and balanced, will have the ability to provide the design maximum cfm or minimum cfm, or any cfm in between, whenever the space thermostat demands it, and do this regardless of the VAV box inlet static pressure as long as the static pressure is within the box design limits.

(1) System Powered: The term *system powered* means that the operating controls of the boxes are powered by the system static pressure and/or the system velocity pressure which ultimately results from the fan. System powered boxes usually have a higher required minimum inlet static pressure. Although the higher static pressure may not be needed for the air quantity, it will be needed to operate the controls. In these days of lower energy consumption, system powered equipment is being used less, mainly because of higher static pressure requirements placed on the fan that increases operating costs. Their big advantage is that they do not need a separate pneumatic or electric control system, which simplifies installation and keeps first costs down. Many of them are completely independent units with built-in thermostats, etc. Some however, use a combination of system powered cfm regulators with pneumatic or electronic thermostatic action and VAV control.

(2) Non-System Powered: *Non system powered* VAV box systems use VAV box controls that are powered from external sources, instead of from the system pressure. These controls usually are part of the pneumatic or electric control system. Otherwise, the controls provide the same service as the system powered controls and the VAV boxes function similarly.

b. PRESSURE DEPENDENT VAV SYSTEMS

A pressure dependent VAV box is essentially a pressure reducing attenuator box with a motorized damper that is controlled by a thermostat. A VAV system using these boxes will have a lower first cost and can function well if properly designed, maintained and preferably not used in large or extensive duct systems. Since the airflow of these boxes is in direct relation to the box inlet static pressure, it is possible for the boxes closest to the HVAC unit to get more air than required, with the result that the boxes down the line will be getting little or no air. Where a large diversity is involved, the problem is worse. To obtain satisfactory results, the control system will require tamper-proof thermostats so that some occupants cannot get their boxes to open fully and take all the available system air. Also, the unit must be operated with a low enough discharge air temperature to satisfy all space requirements. The VAV box will then be able to throttle down so that adequate static pressure is available throughout the system.

The term *slave unit* has been applied to some pressure dependent units that have their damper controlled from a nearby pressure independent VAV box. This eliminates the cost of having pressure independent controls on each box. One box in an area will be pressure independent and one or more pressure dependent boxes or "slaves" will be controlled so that their dampers are opened to the same degree as the damper in the pressure independent "master" unit. This system will work fine as long as all inlet static pressures are the same. However, if the static pressure is different at the slave boxes, their cfm also will be different.

c. FAN POWERED VAV SYSTEMS

Except at the VAV box itself, fan powered box VAV systems function exactly like a pressure independent, non system powered VAV system. The air going into the fan powered VAV box from the HVAC unit is called *primary air*. It is a pressure independent, variable air volume system and is controlled by a space thermostat. The big difference is that the fan can recirculate return air through the system and mix it with the primary air as per the design operating sequence. Quite often, these boxes are used on the perimeter of a building and in conjunction with reheat coils.

d. INDUCTION BOX VAV SYSTEMS

These systems operate similarly to any pressure independent, non system powered VAV system except at the box. Here, primary air is used to induce air, usually from the return air ceiling plenum, and mix it with the primary tempered air from the system. This enables more air to be circulating than would be with just primary air, especially as the primary air is reduced to its minimum position. Reheat coils are often used. (Note that induction boxes also may be used on constant volume systems. These boxes will usually be pressure dependent.)

e. BYPASS BOX VAV SYSTEMS

The fan and other unit components on a bypass box system are essentially of a conventional constant volume type. The difference is that the bypass box has the ability to divert the supply air in two directions. A space thermostat signals the box to send air into the duct system serving the space, or to divert it into a ceiling return plenum instead, where it will return to the unit. The system operates and is balanced similar to constant volume, pressure dependent box system. The system is constant volume from the fan to the box. It is variable volume from the box and into the space being served.

f. SYSTEMS WITH COMBINATIONS OF BOXES

Many designers will mix different operating types of boxes on the same systems. Balancing some of these systems can be difficult. Some examples include mixing pressure independent and pressure dependent boxes, constant and variable volume boxes and dual duct and single duct boxes on the same systems.

6. Dual Duct Systems

The dual duct system conditions all the air in a central apparatus and distributes it to conditioned spaces through two parallel supply air ducts. One duct carries cold air and the other warm air. This allows the conditioned spaces to have both heating or cooling at all times. In each conditioned space or zone, a mixing box with damper responsive to a room thermostat mixes the warm and cold air in proper proportions to satisfy the prevailing heat load of the space. Needless to say, these systems are not usually energy efficient!

The mixing boxes may be constant volume or variable volume. They are usually pressure independent and they may be system powered or otherwise. Most systems are designed so that the cold duct system can handle 100% of the total airflow, but that the hot duct system only handles about 75% of the fan and system design airflow. The mixing box satisfies the space temperature requirements by mixing the hot and cold air as demanded by the space thermostat and maintaining the correct cfm. The mixed air is discharged into a conventional low pressure duct system and outlets.

These systems have a lot of flexibility and are usually able to control space temperatures quite well. The combinations of controls and sequences available are many and the balancing technician must review the sequence of operations on each job before determining the proper balancing procedure.

a. CONSTANT VOLUME

Dual duct constant volume systems deliver the same total cfm to the space at all times by utilizing a constant volume fan and dual duct system along with constant volume dual duct boxes. The boxes mix hot and cold air as required, but maintain the same discharge airflow at all times.

b. VARIABLE AIR VOLUME

The VAV dual duct system was derived from a very flexible combination of systems. First, the fan and system must be capable of varying the air volume. It also must be of dual duct design so that both hot and cold air is available to the boxes. The boxes must be capable of mixing the hot and cold air to obtain the temperature required as well as varying the volume to maintain satisfactory space temperatures. These boxes are always *pressure independent.* The combinations of sequences are unlimited and many different schemes are used. The design sequence must

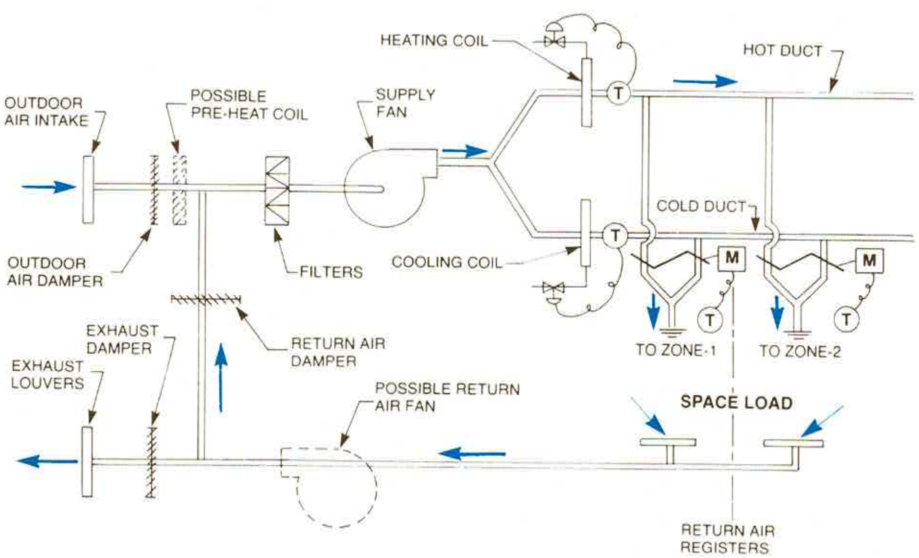


Figure 4-49 DUAL DUCT LOW VELOCITY SYSTEM

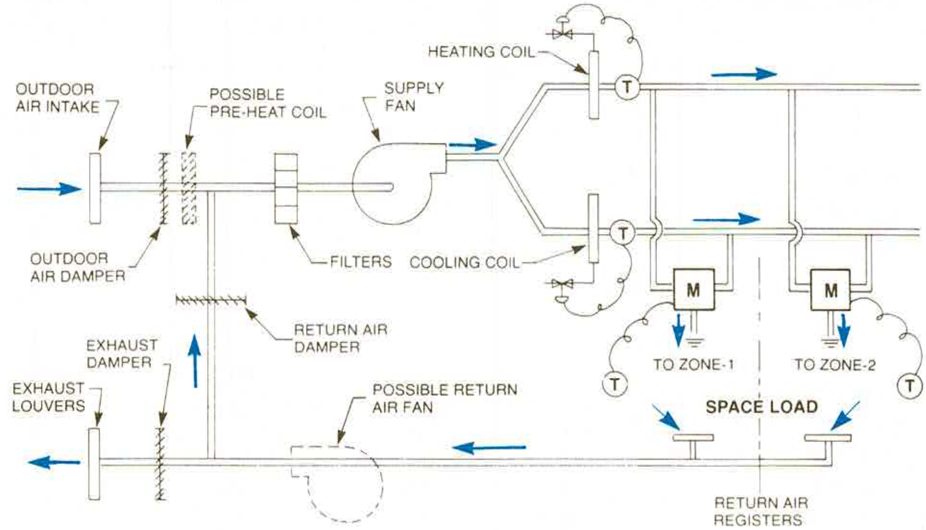


Figure 4-50 DUAL DUCT HIGH VELOCITY SYSTEM

be provided to the TAB technician so that the proper balancing procedure can be determined.

c. LOW PRESSURE DUAL DUCT SYSTEMS (NO MIXING BOXES)

These are relatively simple pressure dependent dual duct systems. Instead of a mixing box, branch supply air ducts are run from the hot and cold main supply air ducts with each branch duct having its own control damper. The branch hot and cold ducts are then connected together and supply the space airflow. A thermostat controls the dampers which are inter-connected so that one is opened while the other one is closed, and vice versa. Being pressure dependent, the cfm to the space will vary when the static pressure in the system varies. The cfm also will change as the space load changes from hot to cold. These systems are relatively expensive to install, and they are not being designed for use in new buildings because of energy demands.

7. Induction Unit Systems

The induction-type reheat unit is shown schematically in Figure 4-51. Full cooling capacity is provided in the higher pressure primary airstream supplied by the central HVAC unit. Zone control is accomplished by heating or cooling the secondary or induced airstream with a water coil or electric coil. This type of terminal is used when it is desirable to introduce supply air to the space at a higher (or lower) temperature, or permit higher space air movement without increasing the quantity of primary air over the amount of air required for cooling.

The primary air is discharged from nozzles arranged to induce room air into the induction unit at a rate approximately 4 times the volume of the primary air. The induced air is cooled or heated by a *secondary* water coil. The water coil may be supplied by a 2-pipe system where either chilled water or heated water is available, but not simultaneously; or by a 3-pipe system where separate supplies of hot or chilled water are continuously available and, after passing through the unit, are mixed into a common return; or by a 4-pipe system, where separate supply and return systems of hot water and chilled water are both continuously available.

Induction-type units are generally located under the window to offset winter downdrafts. Overhead installations are limited, since ductwork connections carrying induced air have limited static pressure available, thereby decreasing induction air volume and unit capacity. When installed under the window, this unit has the advantage of providing gravity heating during off-hour operation, permitting shutdown of the primary air system.

Induction unit nozzles may be worn through many years of cleaning and operation, resulting in increased primary air quantity at lower air velocities with lower induced air volumes. Check the induction units before the TAB work commences. If the nozzles are worn, have the owner either repair the nozzles or replace them before attempting any balancing work on the system.

The induction system primary supply air fan operates at high static pressures which require high horsepower input. Energy can be saved on existing sys-

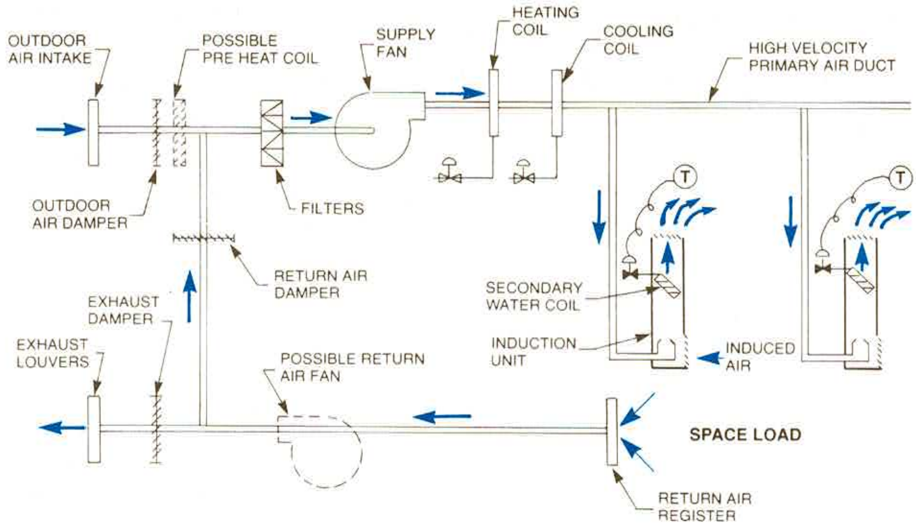


Figure 4-51 INDUCTION SYSTEM

tems by making a careful analysis to reduce the primary airflow volume and system static pressure to the minimum required to operate the induction terminal units.

8. Systems with Hoods

Systems with hoods are found in various types of ventilating systems. Kitchen hoods for restaurants and institutions are the type most frequently encountered. Most kitchen hoods are designed for a free velocity of about 100 feet per minute at the hood entrance. *Capture velocity* is necessary to insure entrainment of steam and grease laden vapors. Some municipalities have ordinances setting minimum requirements for face velocities at kitchen hoods and further require that an authorized representative be present at the time of balancing.

(a) Kitchen make-up air systems must be in operation when the balancing takes place. Sometimes make-up is achieved by means of relief grilles from adjoining areas.

(b) Fume hoods are frequently found in laboratory buildings and hospitals. Here experiments are carried on in confined areas that are designed to prevent the escape of toxic or noxious fumes. Balancing should be done with the fume door set in the normal operating position. Some hoods have a built-in make up air system to minimize the loss of conditioned air from the laboratory. This must also be accurately balanced. When toxic experiments are to be performed by the occupants, a smoke candle test should be made by the balance crew to ensure that vapors do not escape.

(c) Factory exhaust systems with hoods fall into two categories. One group, similar in many respects to laboratory fume hoods, is used in conjunction with dip tanks and plating tanks. Exhaust hoods are often placed at the opposite side. This permits vapors to be swept from the tank surface but still leaves the top open for overhead handling equipment. Balancing procedures are the same as for fume hoods.

(d) A second group of factory exhaust systems is used to remove and convey solid materials. Sawdust, wood chips, paper trimmings, etc., are transported through exhaust systems. These systems must be balanced so that velocities do not fall below predetermined transport velocities, at which point the materials would drop out.

CHAPTER 5 HYDRONIC SYSTEMS

A INTRODUCTION

Hydronics may be defined for TAB work as the process of heating and cooling through the use of liquid fluids. *Hydronic systems* may be described as the piping systems used to transport the fluid from one source, such as a boiler or chiller, by means of a pump to an outlet or a terminal heat exchange unit, such as a fan-coil unit or an air handling unit. Hydronic systems also may contain more than one source and one outlet in addition to several methods of piping. It therefore becomes essential that TAB technicians become acquainted with the various types of systems, piping applications, and pumps and system components, so that a complex system, or combination of systems, may be properly analyzed and the most efficient method of testing and balancing be developed.

As fans are to air systems, so pumps are to hydronic systems. Both are "prime movers" and the "hearts" of the systems. The TAB technician will find that pumps generally are less complicated than fans, and that hydronic systems are more "forgiving" than air systems in TAB work.

B PUMPS

Pumps used in hydronic systems are liquid pumps having the specific purpose of producing sufficient pressure to overcome system resistence at the required flow rate. Although it will be seen that the pump laws are essentially the same as the fan laws and are applied in the same manner, the problems encountered in the selection and operation of liquid pumps are somewhat different, mainly because of the need to move "solid," practically incompressible fluids.

Although not true of fan data, pump performance curves are commonly available in manufacturers' catalogues, and there is generally no need to develop them from tabulated information. Therefore, the selection of pumps at the most desirable characteristic or operating point is less difficult, and investigations of field operating conditions can be simplified considerably.

The flow rate is usually given in gallons per minute of the fluid pumped. However, some few curves indicate pounds per minute or pounds per hour. Use of curves established from flow rates other than gpm should be avoided to minimize the possibility of having to correct for temperature and density.

The pressure coordinate is most useful if presented in total feet of head of the fluid pumped. If this coordinate is given in pounds per square inch (psi), temperature and density corrections may again be necessary. Pressure, even if measured in feet, may be given in total feet, dynamic feet, or may have no identification at all other than "feet." In the following discussion it will be established that pump differential pressure requirements will be determined by a combination of elevation, friction, and velocity heads. Therefore, total head is most desirable even if elevation and/or velocity heads are determined not to be significant.

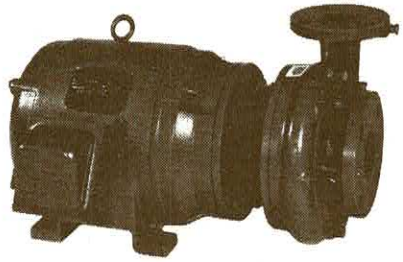
Because of the physical weight of the liquid to be circulated, elevation within the system is a major factor in the selection and construction of hydronic pumps. Insufficient positive pressure on the suction of a pump can produce undesirable consequences, even to the point where the pump may fail to operate. Neither of these two conditions is normally of concern in the application of fans.

The pumps used in HVAC systems fall into two major categories, positive displacement pumps and centrifugal pumps, with the centrifugal pumps being more widely used.

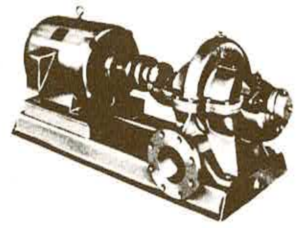
1. Types of Pumps

a. POSITIVE DISPLACEMENT PUMPS

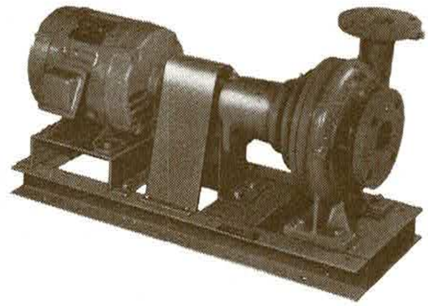
The usual types of positive displacement pumps in environmental systems are the piston pump (reciprocating), the rotary pump, and the screw pump. One



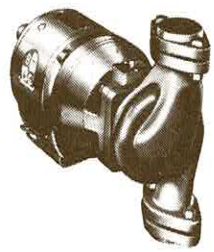
(A) End-suction close-coupled



(C) Horizontal Split Case—double suction



(B) End-suction base mounted, with flexible coupling



(D) In-line Hot Water Circulator

Figure 5-1 TYPICAL CENTRIFUGAL PUMPS

of the characteristics that is common to all positive displacement pumps is their ability to overcome excessive pressures. These pumps are called positive displacement pumps because they take a discreet quantity of fluid at a low pressure and transform it into the same quantity of fluid at a higher pressure. This implies very little slippage within the pump (very little leakage around the impeller) and high pressures at the pump outlet.

The pump curve for the positive displacement pumps is nearly linear in a vertical direction that is, at a given speed it will pump the same volume (gpm) against whatever pressure is connected at the outlet (usually from zero to several hundred pounds). Many of these pumps are constructed with a built-in spring loaded relief valve which will automatically by-pass liquid internally from the discharge side to the side to the suction side in the event of an excessive pressure build-up. This prevents damage to the pump or to the pump packings or seals.

Even though these pumps are positive displacement and are close to being "perfect" pumps, they still cannot lift a fluid on the suction side higher than the equivalent of 33.9 feet of water. Since oil is lighter than water, the suction height would be higher if oil was being pumped; or drastically lower if a heavy fluid such as mercury was being pumped. This is because all pumps depend upon the atmosphere pressure to push the fluid into suction side. Large oil transfer pumps are a good example of a common application or rotary positive displacement pumps in HVAC work.

b. CENTRIFUGAL PUMPS

Centrifugal pumps are constructed with less stringent tolerances than positive displacement pumps. There is more fluid slippage, which means that as the pressures go up, more fluid slips past the impeller and less fluid is delivered to the outlet of the pump. This makes the pump curve similar to that shown in Figure 5-3. The impellers of these pumps are classified according to the configuration of the vanes: plain (radial) flow, mixed flow, axial flow, etc. The differences are due to the angles at which the fluid enters and leaves the impeller.

Staging of the impellers is one of the ways in which centrifugal pumps overcome their inability to pump against high heads. Pumping through two or more

Table 5-1 CHARACTERISTICS OF CENTRIFUGAL PUMPS

Type	Impeller Type	Number of Impellers	Casing	Motor Connection	Motor Mounting Position
Circulator	Single suction	One	Volute	Flexible coupled	Horizontal
Close coupled, end suction	Single suction	One or two	Volute	Close coupled	Horizontal
Frame mounted, end suction	Single suction	One or two	Volute	Flexible coupled	Horizontal
Double suction, horizontal split case	Double suction	One	Volute	Flexible	Horizontal
Horizontal split case, multistage	Single suction	Two to five	Volute	Flexible coupled	Horizontal
Vertical inline	Single suction	One	Volute	Flexible or close	Vertical
Vertical turbine	Single suction	One to twenty	Diffuser	Flexible coupled	Vertical

stages multiplies the head capacity. A variety of drives are available including direct drives and varible speed drives. The most common pump drive uses flexible couplings between the motor and the pump assembly. The alignment or mis-alignment of the coupling will not effect the performance of the pump nor the speed at which it is driven. However, the misaligned coupling will have an amazingly short life as the elastic interface begins to break down. This sometimes will allow the speed of the pump to vary and effect its output.

In many cases the elastic will deteriorate to such a point that the pump will stop even though the motor will continue to run. This lack of flow condition is difficult to control except by using flow switches to protect other equipment such as chillers. Condenser water systems normally do not have the flow switches. The problem here is that the motor can be running, but the pump has become uncoupled. Electrically, all interlocks operate because the motor is running. Functionally, there is no water flow at the cooling tower.

Table 5-2 CHARACTERISTICS OF COMMON TYPES OF PUMPS

	Positive displacement pumps		Centrifugal pumps		
Characteristics	Rotary	Piston	Radial	Mixed flow	Axial flow
Flow	Even	Pulsating	Even	Even	Even
Effect of increasing head:					
on flow	Negligible decrease		Decrease	Decrease	Decrease
on bhp	Increase	Increase	Decrease	Small decrease to large increase	Large increase
Effect of decreasing head:					
on flow	Negligible increase		Increase	Increase	Increase
on bhp	Decrease	Decrease	Increase	Slight increase to decrease	Decrease
Effect of closing discharge valve:					
on pressure	Can destruct unless relief valve is used		Up to 30% increase	Considerable increase	Large increase
on bhp	Increase to destruction		Decrease 50%-60%	10% decrease to 80% increase	Increase 80%-150%

2. Pump Construction

Pressure, produced by the pump or system, plays a major role in the design and selection of a pump. Fans are relatively unaffected by the pressures produced by the height of the system since air has an extremely low density. The fluids moved by liquid pumps are sufficiently dense to produce significant weight and pressure as system height increases. These pressures are high enough to require consideration in the pump design even when a system is not operating. Consequently, the complete pump housing and accessories must be designed to withstand all system pressures in addition to whatever pressure the pump itself can produce.

For example, assume that the maximum system pressure to overcome friction losses is 25 psig. If the selected pump is at the bottom of a water system with an elevation equivalent to 75 psig, the maximum pressure on the pump would be approximately 100 psig and a standard 125 psig pump casing design would be satisfactory. If the pressure required to overcome friction were the same but system elevation were equivalent to 200 psig, the required standard casing design pressure would be 250 to 300 psig. The pump performance does not change. At the same time the pump seals, flanges, and all other accessories exposed to the pressure would necessarily be rated at the same higher value.

Although the most common liquid pumped in an environmental system is water, there are times when

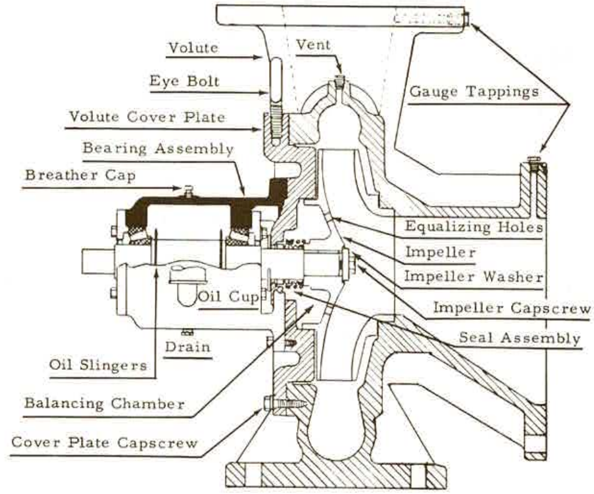


Figure 5-2 TYPICAL CONSTRUCTION OF A CENTRIFUGAL PUMP

refrigerants, brines, oil, or other thermal fluids are required. In some instances these may be handled by water pumps with minor modifications to some pump features but without major design changes. In other cases, pumps commonly used to pump water are not satisfactory. The pumps discussed here are centrifugal types commonly used with water.

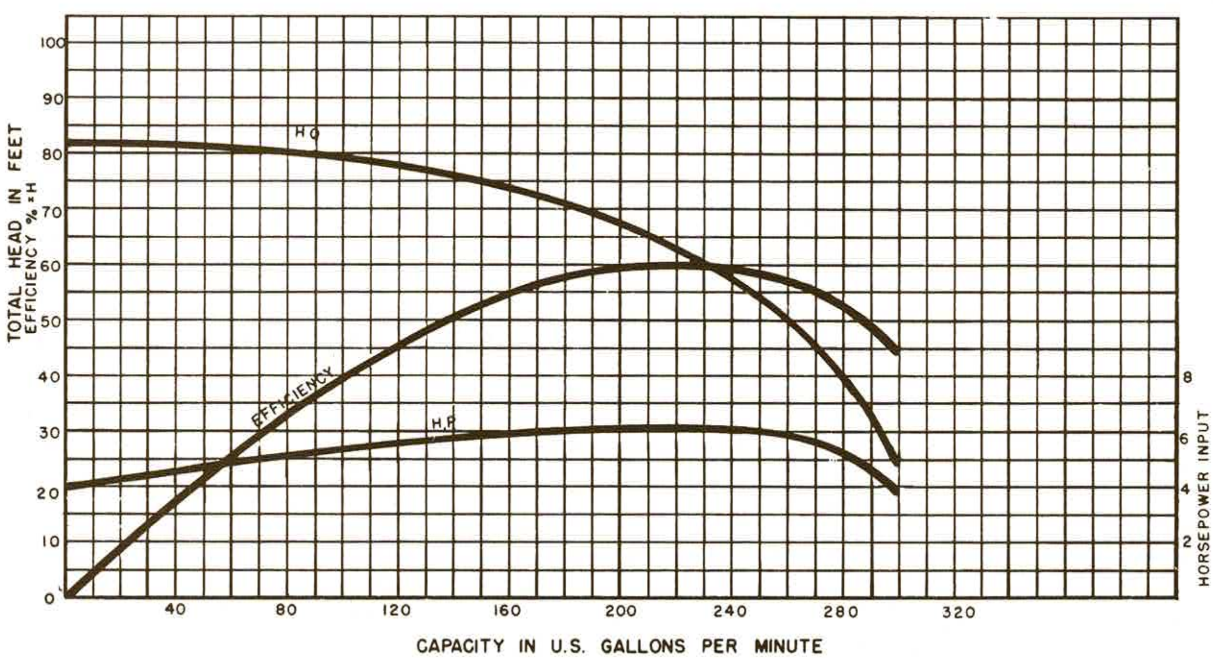


Figure 5-3 TYPICAL PUMP PERFORMANCE CURVES

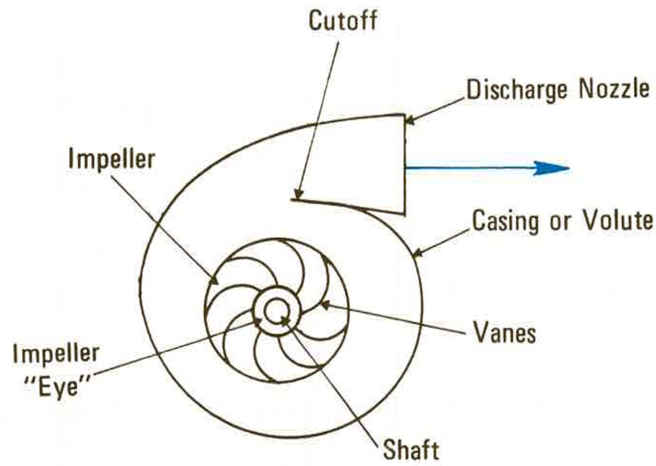


Figure 5-4 TYPICAL CENTRIFUGAL PUMP CROSS-SECTION

Several of the pump construction features are listed below with some of the considerations required for their application noted. Refer to the cross-section and pump diagrams in Figure 5-2, 5-4, and 5-5 for location of the pump parts.

a. IMPELLER

The diagram in Figure 5-5 indicates a double suction pump and with a double suction impeller. The casing is constructed so that the water entering the inlet or suction connection is directed to both sides of the impeller where it flows into two impeller inlets or "eyes." For many applications, both single suction and double suction pumps are available. The single suction pump (Figure 5-2) is generally less expensive, but the double suction pump generally will be easier to service without dismantling the pump from the piping.

b. CASING

In Figure 5-5, the two halves of the casing are flanged with the flanged faces parallel to and at the same elevation as the centerline of the shaft. In this way, the top half may be unbolted and removed as a "lid," exposing the top half of impeller bearings, packing (or seals), and the machined flange face of the bottom half of the casing. This may be accomplished without disturbing the bottom half of casing, base, shaft, coupling (or other drive), and motor (or other drive). All of the exposed components are then accessible for inspection and removal of foreign materials. Casings for close-coupled or single suction pumps are often not constructed with the split feature. Consequently, it is often necessary to dismantle the entire assembly for repairs, especially in the smaller sizes.

c. PRESSURES

The casing must be designed to withstand the highest system pressure whether produced by the pump,

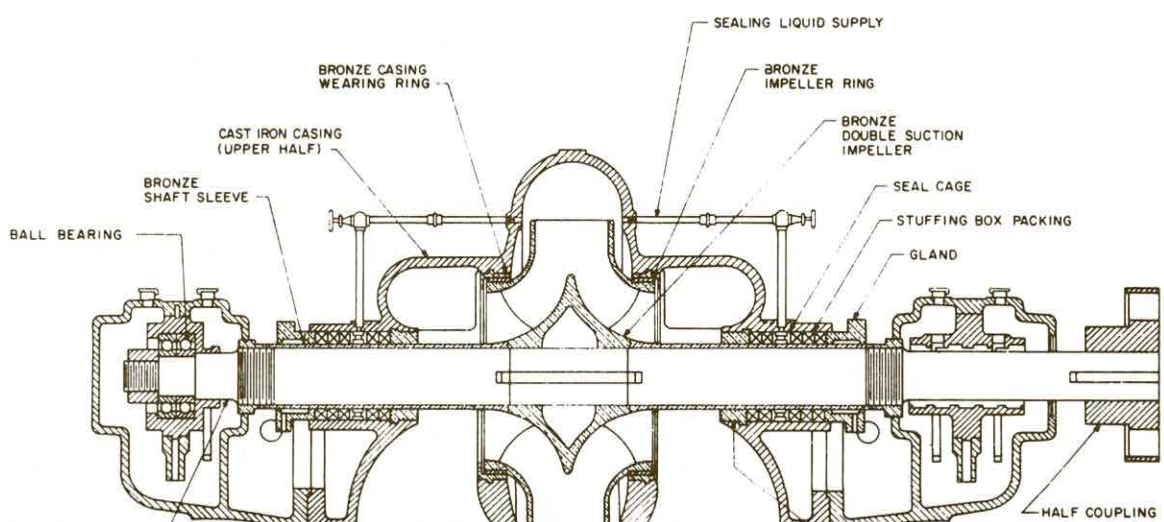


Figure 5-5 MAJOR PARTS OF A CENTRIFUGAL PUMP

the system elevation, or other pumps in the system. In addition, packing or seals must be designed to withstand these pressures without "blowing-out." The small piping shown in Figure 5-5 which connects the top of the casing on each side to an intermediate ring of packing, is in reality a pressure equalizing device. The packing rings toward the impeller have approximately the same pressure exerted on both sides; and the other rings absorb the major pressure differential.

In applications where pressures are not high or where mechanical seals are used, this piping is eliminated and the external connections are plugged. When the connection at the shaft is internal, a special ring is inserted with the packing to virtually seal off the opening.

d. STAGES

By enlongating the shaft and casing, additional impellers may be placed on the shaft much in the same manner as a turbine. Each impeller, or stage, successively pumps into the next in series with fixed vanes between impellers to direct the flow. In this way, pressure may be increased internally at a given flow rate in the multistage pump. This result also can be accomplished by connecting separate single stage pumps in series.

e. PUMP ARRANGEMENTS

The physical arrangement of the pump is related to the shaft position. A horizontal pump has a horizontal shaft and a vertical pump has a vertical shaft. Some pumps may be operated horizontally or vertically as in the case of the "in-line" type mounted directly in the piping. Single or double suction, as previously noted, is a function of the casing and impeller design.

Vertical pumps may also be designed with a volute type of casing or may be submerged directly as in a deep well where the casing is provided by the well wall. Some submerged pumps are built with an extended shaft so that the pump and casing are submerged, but the motor and coupling are mounted on a plate or structure above the floor of the mechanical space. Other submerged pumps are built with pump and motor sealed in a housing which together are submerged in the liquid.

f. SUCTION AND DISCHARGE

Single suction pumps are usually constructed with the inlet at the end of the impeller and shaft with the casing arranged so that the discharge may be rotated to any position allowed by the bolt configuration.

Double suction pumps do not have this flexibility, the suction and discharge being fixed and usually below the shaft centerline, each at a slightly different elevation. In either case, the suction connection is normally one or two pipe sizes larger than the discharge connection.

g. ROTATION

Rotation of any pump is fixed by the configuration and type of vanes and the suction and discharged connections. An arrow to indicate proper direction is often cast directly into the casing metal. In addition to proper position of the pump in the piping, rotation is also dependent upon the motor or driver rotation. Rotation of motor and pump must be tested prior to operation. However, *pumps with mechanical seals must not be run dry, even for "bumping" to determine rotation.*

h. DRIVES

A pump may be driven by any appropriate means. For the most part, environmental system pumps are motor driven, and the shafts of the motor and pump are connected end-to-end by some type of coupling. In the case of some small pumps, the pump may be mounted directly on the extended motor shaft without a coupling. Some pumps, usually those employed in pumping fuel oil, are belt driven. In this case the belts may be used as a protective device, either slipping or breaking before damage to the pump can occur in the event of overload.

The couplings between motor shaft and pump shaft are made in two pieces or "halves." By removing bolts, springs, or some other final connecting device, the two coupling halves may be disconnected for removal of the pump without disturbing the motor, for running the motor independently of the pump, or for removal of the motor without disturbing the pump.

The coupling also serves as a means of adjustment of the pump and motor shaft alignment. The ideal alignment condition is that both shafts are in a straight line and concentric under all conditions of operation and shut down. Because of changes of liquid temperature and operational temperature of the pump and motor, an unequal expansion of parts causes a change of alignment during operation. Alignment is usually a compromise, therefore, between the two extremes of operation, the hottest and coldest. Perfect alignment produces the least coupling and bearing wear, but actual conditions require operation within the alignment tolerances set forth by the pump, motor, and coupling manufacturers.

Some pumps are factory aligned and guaranteed for the operating application. Some small pumps do not require field alignment. Base mounted pumps, especially in larger sizes, require at least an alignment check in the field. This may be done in a superficial but often satisfactory way with a straight edge since the outside perimeters of the coupling halves are machined to the same diameter and, because of the installation means, are perpendicular to each shaft. Centerlines and coupling faces must be true. (See Figure 5-6.)

To provide accurate alignment, a dial indicator is employed. This is a small gauge-like instrument, which, when properly applied, measures alignment differences in thousandths of an inch, tolerances which are impossible with a straight-edge.

Once aligned, the pump base is anchored into place and may be grouted in with cement to maintain the established equipment positioning. Standing on a pump base, even after grouting, can change alignment enough to be measurable. However, grouting may minimize distortion. Once alignment is set, care should be taken to prevent disturbing the conditions. The coupling can accommodate small variations, but its purpose is coupling, not compensating for misalignment.

i. SHAFTS AND SHAFT SLEEVES

The basic requirement of the pump shaft is structural strength. Although steel is a common material used

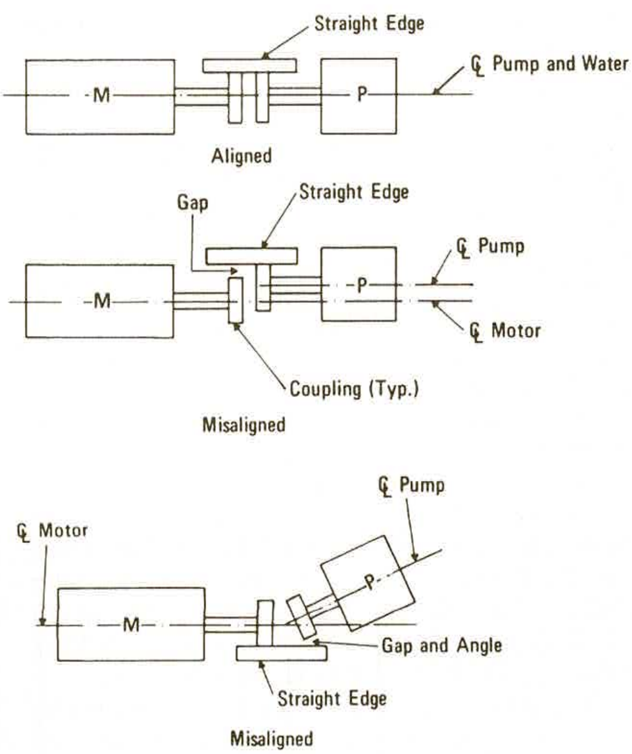


Figure 5-6 COUPLING ALIGNMENT WITH STRAIGHT EDGE

for this purpose, stainless steel and bronze will not rust when exposed to water and will not be subject to additional forms of corrosion in the presence of other liquids. To eliminate the corrosion problem and to provide a wearing surface for the packing described later, sleeves are often installed over steel shafts. The sleeve material will be selected to accommodate any special requirement of the application. In the event of breakdown, the sleeve may be replaced with relative ease without disturbing the shaft, which may be a press fit on either impeller or bearings, or both.

j. PACKING AND SEALS

Figures 5-2 and 5-5 illustrate packed pumps. A specified number of rings of packing, selected from the many types available, are slipped over the shaft without undue pressure. The packing gland is installed last, with considerable care that it is not cocked. Bolts on the packing gland or the packing gland nut are tightened *evenly around* until the recommended position on the shaft or sleeve is reached. The pump is started, and the packing gland tightened *evenly around* until one drop of liquid per second is leaking, or until other manufacturers recommendations are met. After operation for an hour, the leakage is checked, and the gland is adjusted as required. This checking procedure should be followed with diminishing frequency until the required leakage is maintained.

Mechanical seals are devices used in place of the packing to retain the system liquid within the pump at the shaft penetration. On some pumps this is the only such means available, room for packing and stuffing boxes not being provided. There are two major parts to a mechanical seal, a stationary disc and a rotating disc held in contact by a spring arrangement. One disc may be ceramic and the other carbon. Running causes a wearing-in process to maintain a complete seal. Foreign matter between the faces will destroy this seal. **Running or "bumping" a pump without water in the system will damage mechanical seals.**

While a packed pump may normally allow operation until shutdown for convenience, a seal failure may cause a complete blowout and require immediate system shutdown and perhaps replacement if there is no spare pump. Use of packing or seals is determined by what is available for the pump selected, somewhat by the application, and a great deal by previous experience.

k. BEARINGS

Bearings may be sleeve, roller or ball type. The most common are ball bearings, but other types are available when required for the application.

l. MATERIALS

Pump casings are generally cast iron. Trim may be bronze, stainless steel, monel, or other metals. Small circulators used in domestic water systems are often bronze. Special pumps are available in almost any material, including plastics.

m. BASES

The base on which the pump and motor are mounted will depend upon the type of pump. The variations are numerous. It is worth noting that horizontal pumps are usually fitted with a cast iron bed plate which is arranged to collect dripping fluid. The drip trough is fitted with connections to pipe away the fluid which drips from the packing.

3. Pump Pressures or Heads

The purpose of a pump for HVAC work is to establish fluid flow and produce sufficient pressure to overcome the resistance of a system and the system components at the design flow rate.

a. PUMP "HEAD" DEFINITIONS

When working with pumps, the word "head" will often be used to define pressure. Definition of these and other common "head" terms are noted here, even though some may be defined again under other discussions:

Friction head is the pressure in psi or feet of the liquid pumped which represents system resistance that must be overcome.

Velocity head is the pressure needed to accelerate the liquid being pumped. (For practical purposes, the velocity head is insignificant and usually can be ignored in HVAC system calculations.)

Static suction lift is the distance in feet between the pump centerline and the source of liquid below the pump centerline.

Suction lift is the combination of static suction lift and friction head in the suction piping when the source of liquid is below the pump centerline.

Suction head is commonly the positive pressure on the pump inlet when the source of liquid supply is above the pump centerline.

Static suction head is the positive vertical height in feet from the pump centerline to the top of the level of the liquid source.

Dynamic suction lift is the sum of suction lift and velocity head at the pump suction when source is below pump centerline.

Dynamic suction head is positive static suction head minus friction head and minus velocity head.

Dynamic discharge head is static discharge head plus friction head plus velocity head.

Total dynamic head is dynamic discharge head (static discharge head, plus friction head, plus velocity head) plus dynamic suction lift, or dynamic discharge head minus dynamic suction head.

b. PRESSURE RELATIONSHIPS

For the pressure relationship to the inlet and suction side of the pump, the discharge pressure is higher. In the process of establishing this head, the impeller produces a lower or relatively negative pressure on the suction side.

It is important to note the deliberate use of the term "relative" in a discussion of the pressures which pumps produce. The system elevation static pressure is of major consequence in the liquid system. During the time when the pump is running, there is a redistribution of pressures in the system because of the combination of the elevation pressures and the pressures produced by the pump. However, when the pump is shut down, the system pressures return to the same values of elevation static as before the pump was started. The pressures produced by the pump merely add to or subtract from the initial shutdown pressures.

Since the operating discharge pressure produced by the pump is an *increase,* this value is added to the shutdown pressure in the discharge piping. Since the operating suction pressure produced by the pump is a *decrease,* this value is subtracted from the shutdown pressure in the suction piping. Assuming that the system piping and equipment losses were properly calculated and that the pump was properly selected to overcome those losses and to withstand the system static and dynamic pressures, it would be expected that the pump would produce the required fluid flow. However, this may not be the case because of the pump's sensitivity to the pressure conditions on its inlet (the discharge conditions of the pump do not generally present a problem).

c. CAVITATION

The fluid being pumped, usually water, generally contains some entrained air which has been absorbed as a result of atmospheric pressure when the fluid was exposed to the atmosphere prior to being introduced into the system. This air is released because of an increase in fluid temperature, a decrease in fluid pressure, or because of the fluid vapor pressure.

Therefore, there may be a point in the environmental fluid pumping system where air may be released from the fluid being pumped if the pressure is low enough and/or the liquid may change to a gas. Should these conditions occur, the pump, which has been designed to move liquid, is generally unable to cope, and the flow of liquid is either greatly reduced or stopped completely.

However, at some point within the pump where the impeller produces sufficient pressure, the bubbles of gaseous liquid will be re-liquefied and the bubbles of air will be re-absorbed. This transition occurs suddenly and is accompanied by crackling or explosive noises often described as "marbles going through the pump." The phenomenon is called *cavitation* and may cause destructive pitting and wearing of the impeller and casing as well as noise and vibration. Any one or all of these conditions will reduce pump performance and life.

d. NET POSITIVE SUCTION HEAD (NPSH)

To eliminate the problem of cavitation, it is necessary to maintain a minimum suction pressure at the inlet side of the pump.

The actual value in psi or ft. w.g. of internal pump losses depends on the pump size and design, and the volume of water being pumped. This must be determined by the pump manufacturer, and is given by numerical values of *net positive suction head,* abbreviated NPSH. *Required NPSH,* sometimes designated NPSHR, can be considered to be the amount of pressure, in excess of the vapor pressure, required to overcome internal pump losses and so keep water flowing into the pump. For a given pump, the required NPSH increases as capacity increases. Each system, as a result of design and physical limitations, will produce an *available NPSH* sometimes designated NPSHA. When the available NPSH is greater than the required NPSH, the problems of air release, vaporization, and cavitation will not arise.

The required NPSH for a specific pump is available from the manufacturer, either in catalogue data or upon request. Although usually given as a single number, the value varies with flow and head. For any pump, the full range of values for each impeller size and operating speed is expressed as a curve (see Figure 5-7). The TAB technician must remember, however, that for satisfactory pump operation, NPSHA, must always exceed the NPSHR; if it does not, bubbles and pockets of vapor will form in the pump. The results will be reduction in capacity, loss of efficiency, noise, vibration and cavitation.

The net positive suction head (NPSH) of a pump increases as the flow rate (gpm) increases (see Figure 5-7). Notice that in Figure 5-8 they are usually much lower in head pressure than the delivery of the pump. In this chart, the net positive suction head varies from 5 feet to 20 feet whereas discharge heads vary from 110 feet to 360 feet.

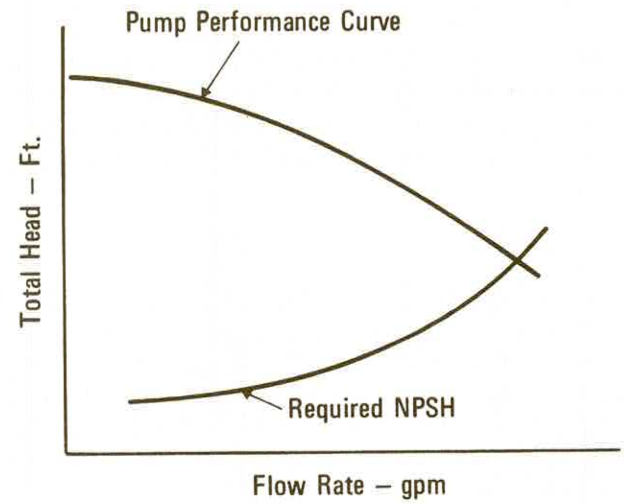


Figure 5-7 TYPICAL REQUIRED NPSH CURVE

NPSH is normally not a consideration in closed systems, especially where the pump is at the bottom of a rise. It is also not ordinarily a factor in most open systems unless pumping hot fluids, or if there is a considerable suction lift, or if there is considerable friction in the pump suction pipe. In unusual considerations of excessive suction line friction, there could be insufficient NPSHA. Such a condition could exist because of an undersized pipe, or too many fittings, or if a valve in the suction line was throttled, or if a fine mesh strainer on the suction side of the pump should become clogged.

4. Pump Curves

Pump curves are similar to fan curves except that most pumps are direct connected to their motors, so the pump speed (rpm) remains almost constant. However, pump impellers can be changed or machined down in size to a specific size. Otherwise flows, pressures, horsepowers, and efficiencies on the fan curves and pump curves are read in the same manner.

In Figure 5-8, the efficiency curve is a measure of the promised pump output versus the output of a theoretically perfect pump which would use 100% of its energy of horsepower. Many design engineers attempt to pick pumps at peak efficiency. Efficiency has little or no value to the TAB technician. Whether the pump that was actually purchased is as efficient as another pump which was specified, or if the furnished pump horsepower draw is higher than the specified, is of no concern to the TAB technician. Your responsibility extends only to seeing that the horsepower required of the pump is being delivered at the designed resistance.

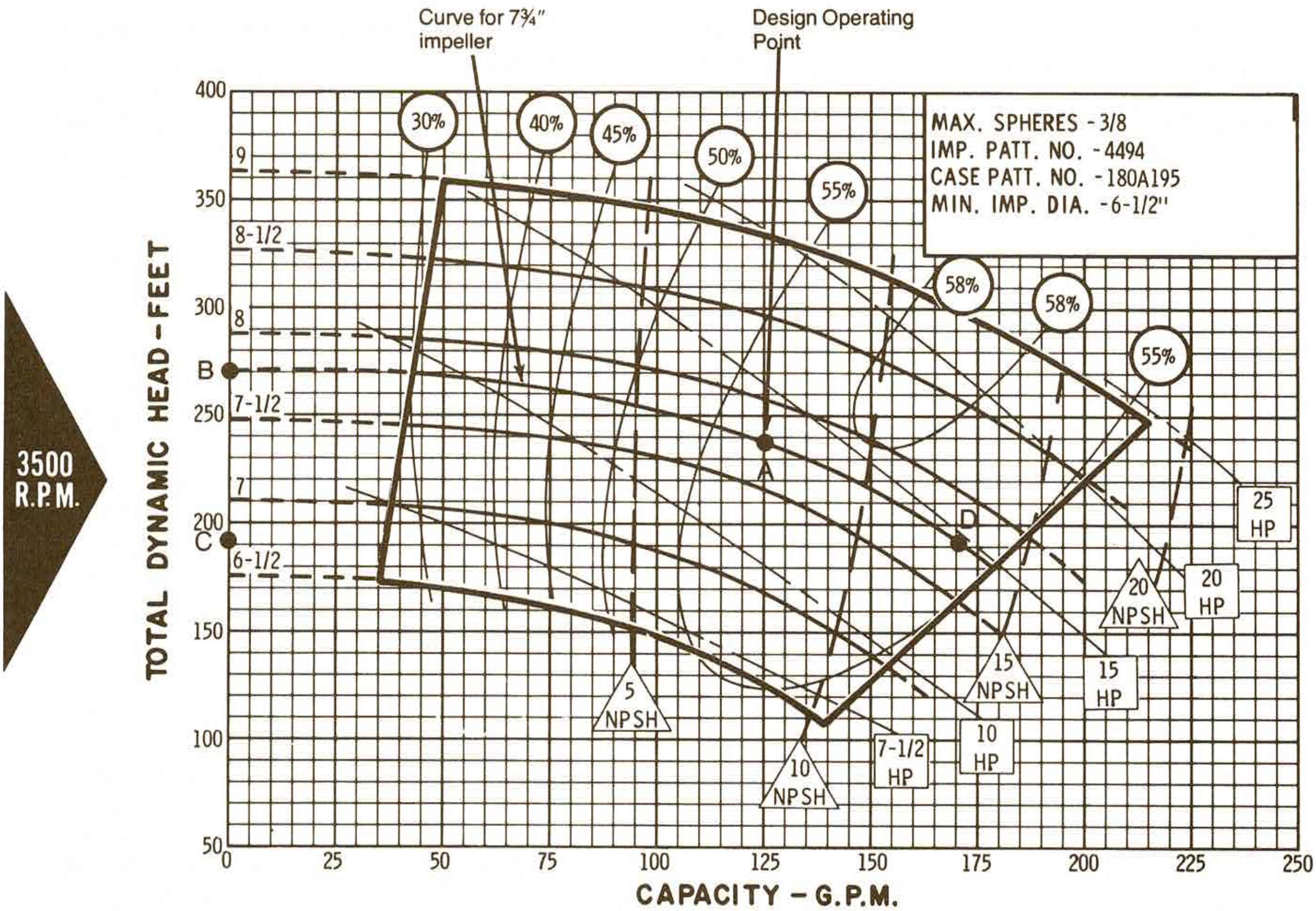


Figure 5-8 TYPICAL PUMP PERFORMANCE CURVE

Referring again to Figure 5-8, the pump horsepowers indicate that an operating point to the left of that line will require no more than the listed amount of horsepower (HP): 7½, 10, 15, 20, etc. Because of the variable slopes of the curves, the pump with a 7 inch impeller will never exceed 10HP, even though it seems to come close to it. When an 8½ inch impeller is used, the 15HP and the 20HP curves are both crossed. For this reason caution is needed, as increasing the flow can exceed the limits of the pump motor provided for the application. This depends not only on the variety of pumps available, but also on how the system fits the specific pump curve.

When you are told to fully open the discharge value to determine the maximum amperage draw, there can be many instances where the pump motor will be overloaded. If the pump motor starter heater coils are oversized, there is a chance that the motor could be damaged before the amperage measurements are taken. You must check the size of the heater coils before starting equipment for testing.

5. System Curves

a. CLOSED SYSTEM CURVES

The hydronic system curve is simply a plot of the change in energy head resulting from a flow change in a fixed piping circuit. System curve construction methods differ between open and closed piping circuits.

From the pipe size and design flow rate, a calculated energy head pressure drop is determined. It should be particularly noted that system static height is of no importance in determining energy head pressure drop. This is because the static heights of the supply and return legs are in balance; the energy head required to raise water to the top of the supply riser is balanced by the energy head regain as water flows down the return riser.

Example 5A

A design flow rate of 200 gpm establishes 30 foot pressure drop in a typical system. This particular point can be plotted on a foot head versus gpm pump curve as shown in Figure 5-9. What pressure drop would occur were the flow changed to 125 gpm through the piping circuit?

Solution

Another calculation would indicate that 11.8 ft. of head is needed. The same procedure carried out for 75 gpm flow rate would result in a 4.2 foot pressure drop. These points can also be plotted on the foot head versus gpm chart as shown in Figure 5-9. Connection of these three points describes a "system curve." The system curve is a statement of the change in pipe friction drop with water flow change for a fixed piping circuit. This is a most important working tool for pump application.

The calculations described above are not needed to establish a system curve. This is because the pipe friction drop varies in a mathematical ratio with the change in water flow rates. The head will change as the square of the water flow rate change.

Equation 5-1

$$\frac{H_2}{H_1} = \left(\frac{gpm_2}{gpm_1}\right)^2$$

Where:

H = Head (ft. w.g.)

gpm = Gallons per minute

The operation of the pump in Figure 5-9 on the piping circuit described by the system curve **must** be at the intersection of the pump curve with the system curve.

In a closed system, the system pressure can be regulated or limited by the pressure relief valve (safety valve) and the automatic water make-up valve (lines pressure regulator valve). The pressure in various parts of the system will vary from top (less) to bottom (more) due to the static head whether the pump(s) is running or not. If the pump is located at the top of the system rather than at the bottom, the suction pressure of the pump will be lower.

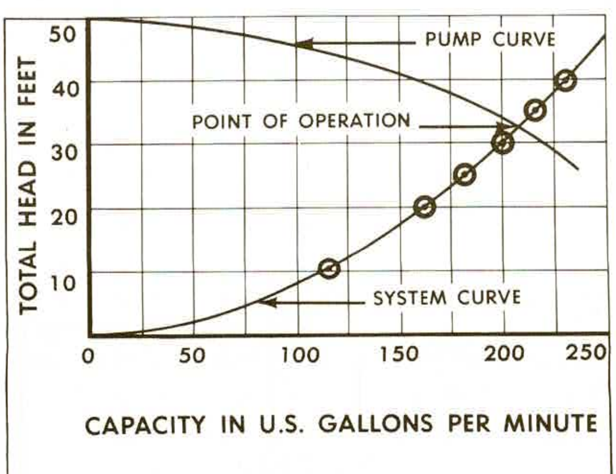


Figure 5-9 SYSTEM CURVE PLOTTED ON PUMP CURVE

When the pump is started, the discharge pressure gauge and the suction pressure gauge (which had similar readings) draw apart, with the discharge pressure gauge going to a higher reading and the suction pressure gauge going to a lower reading. If the make-up water valve is connected to the suction side of the pump, whenever the pump goes on the pressure decrease created by the pump will cause the system to fill and build the pressure back up to the pressure regulator setting. Then when the pump is shut off, the pressure on the suction side will increase to a higher "pump off" reading.

b. OPEN SYSTEM CURVES

This same process takes place in open systems where the fill valve is regulating the water level in a sump or basin which is connected directly to the suction side of the pump. When the pump draws down the water level on start-up, the make-up water flows until the original water level is achieved. When the pump stops, the excess water drains back into the sump, raising the level on the suction side above its neutral position. So there are not many differences between open and closed system characteristics.

The friction factor tables which are used to calculate the piping friction loss during the design phase are different, although some sources used closed system table values times a 1.7 "aging factor". If a closed system which had been installed were opened to the atmosphere in such a way that no additional fluid entered or escaped (that is, water did not leak out due to an excess of pressure at that point; nor was air sucked in due to a pressure deficit at that point), then there would be no change in the actual system curve. This is due to the flow relationship being identical, and pressure measurements taken at any point remaining the same as before. So at this point in time, there would be no difference between the open system and the closed system.

In plotting the system curve for an open system the statics of the system must be analyzed in addition to the friction loss. The different static conditions are illustrated in Figure 5-10.

A typical cooling tower application is illustrated in Figure 5-11. In this system, the pump is drawing water from the tower sump and discharging it through the condenser to the tower nozzles, at a 10 foot higher elevation than the sump level.

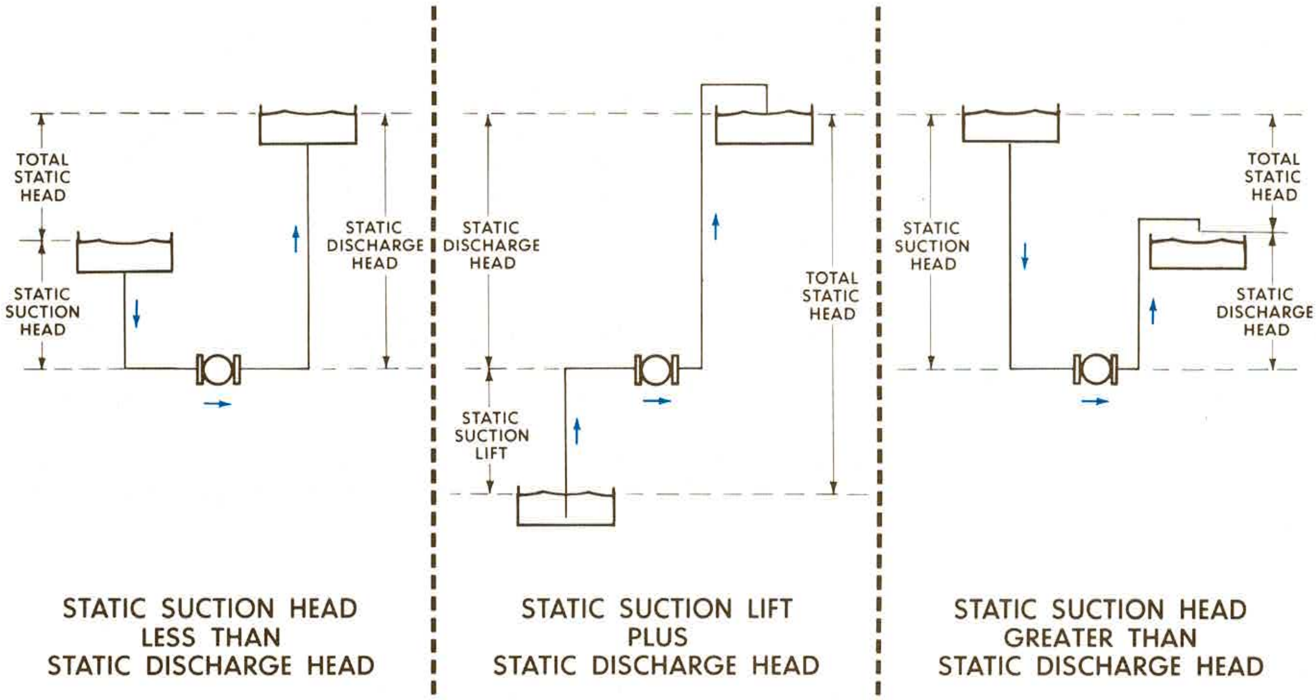


Figure 5-10 TYPICAL OPEN SYSTEMS

Total friction loss (suction & discharge piping, condenser, nozzles, etc.) is 30 foot at a design flow rate of 200 gpm, the change in piping pressure drop for a change in water flow rates is determined and plotted to develop a system curve.

This system curve **cannot** be applied directly to the pump curve and the intersection taken as the accurate pumping point for the open system. A false evaluation using this criteria, but without evaluating the static height of the tower, is shown in Figure 5-12.

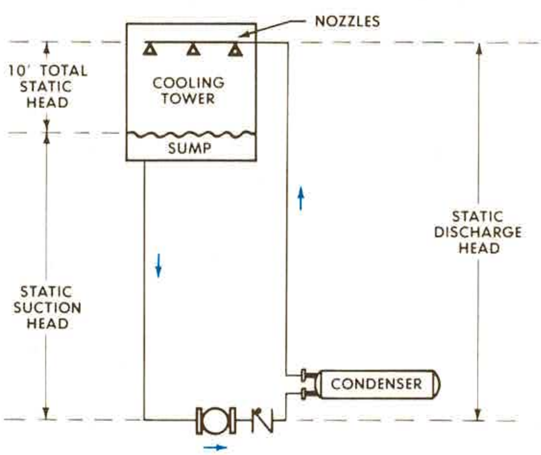


Figure 5-11 TYPICAL COOLING TOWER APPLICATION

The illustration is false because the pump must also provide the necessary energy to raise water from the tower sump to the spray nozzles. In this case, the pump must raise each pound of water 10 ft. in height, or it must provide 10 ft. of energy head due to the static difference in height between the water levels.

The static difference of 10 ft. must be added to the piping pressure drop to provide total required head for each of the gpm points previously noted. the revised gpm versus total required head is shown in Table 5-3.

The correct procedure for plotting a system curve for the circuit shown in Figure 5-11 is illustrated in Figure 5-13.

Table 5-3 GPM VS TOTAL REQUIRED HEAD (COOLING TOWERS APPLICATION)

GPM	0	115	165	185	Design 200	215	230
Pipe & Valve Pressure Drop	0	10	20	25	30	35	40
+ Static Energy Head	10	10	10	10	10	10	10
Total Req'd Head for Flow	10	20	30	35	40	45	50

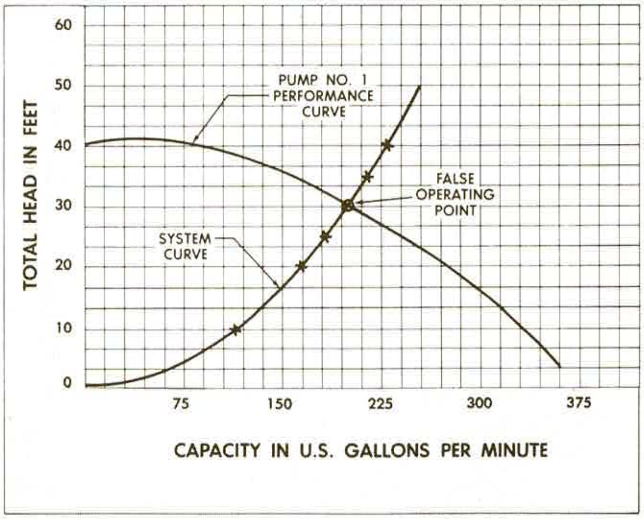


Figure 5-12 SYSTEM CURVE FOR OPEN CIRCUIT—FALSE OPERATING POINT

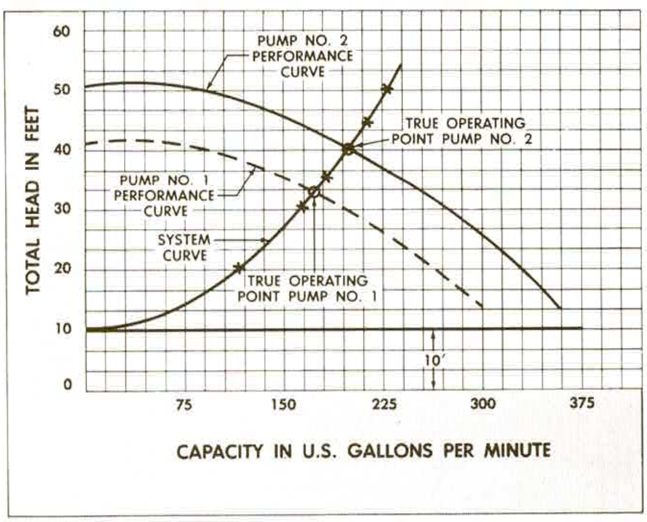


Figure 5-13 SYSTEM CURVE FOR OPEN CIRCUIT—TRUE OPERATING POINT

c. MULTIPLE PUMPS

Multiple pumps piped in parallel are not a common installation. Figure 5-14A describes two pumps piped in parallel, while Figure 5-15 includes a system head curve as well as the head-capacity curves for single-pump and two-pump operation. Figure 5-14 illustrates two pumps piped in series with bypasses for single-pump operation. Figure 5-16 indicates the use of series pumping on a hydronic system with a system head curve consisting of a large amount of system friction.

6. Pump Installation Criteria

a. PRESSURE GAUGE LOCATION

To eliminate the effect of pipe friction, fittings, valves, and other obstructions, the most desirable guage location for accuracy would be at the pump flanges. However, this is not usually practical. Gauges should be located as close to the flanges as possible as shown in Figure 5-17.

To eliminate an evaluation static head correction, the gauges on suction and discharges should be at the same height with respect to the pump centerline. If this precaution is not taken, the difference in gauge elevation, even though usually of small numerical value, must be accounted for in the gauge differential.

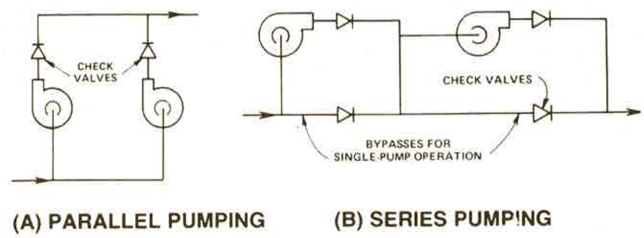


Figure 5-14 MULTIPLE PUMPS

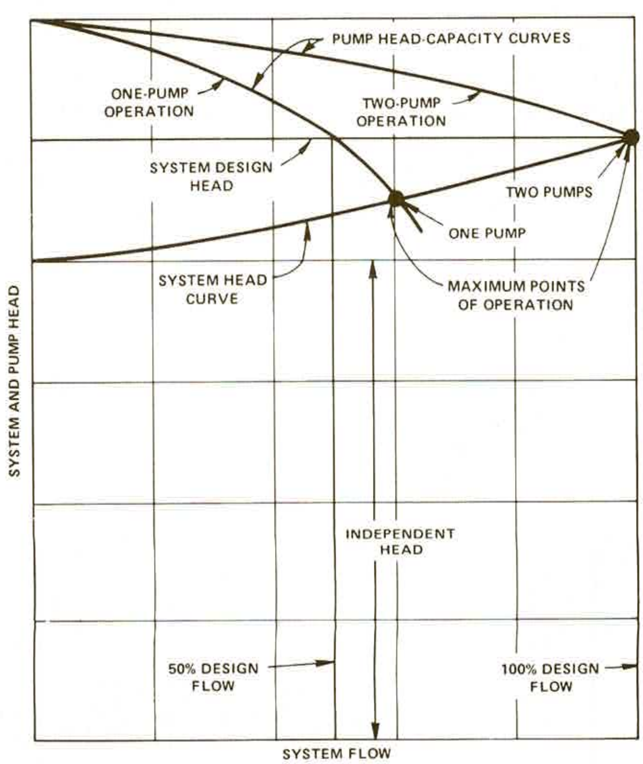


Figure 5-15 PUMP AND SYSTEM CURVES FOR PARALLEL PUMPING

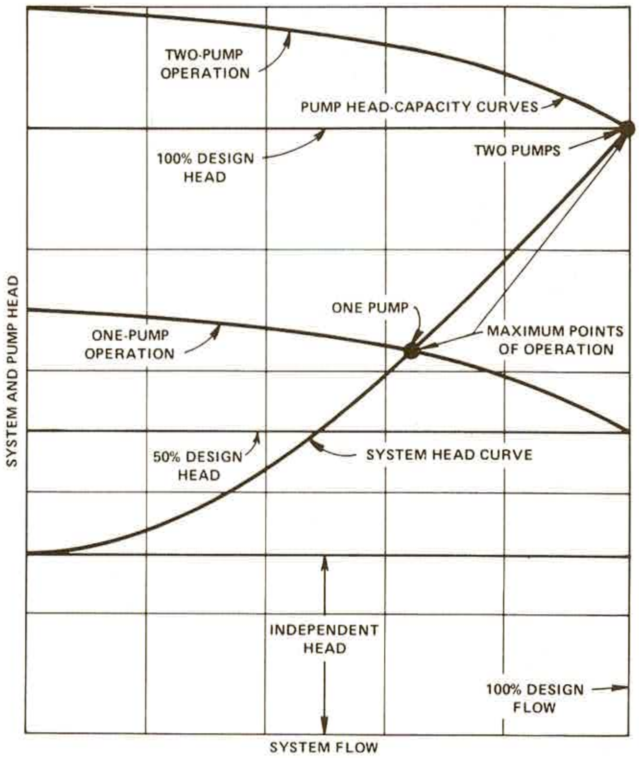


Figure 5-16 PUMP AND SYSTEM CURVES FOR SERIES PUMPING

In Figure 5-18 there is a physical difference in height of 2 feet. If the gauge pressure, when converted, measured 50 feet on the discharge and 30 feet on the suction, subtraction alone would indicate a differential of 20 feet. However, with respect to the discharge gauge which is two feet lower in the piping, the suction gauge reads two feet of head too little, and at the same elevation as the discharge gauge would read 32 feet. A similar analysis would apply if the positions were reversed or if one or both gauges were located below the horizontal pipe or pump centerline.

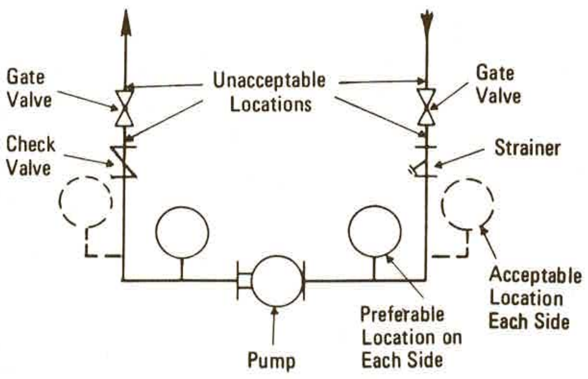


Figure 5-17 GAUGE LOCATION

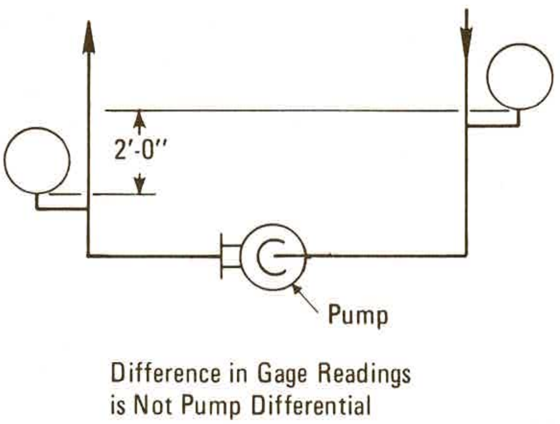


Figure 5-18 RELATIVE GAUGE ELEVATIONS

b. FLUID VISCOSITY

It should be noted that as long as the head-gpm curve is based on feet of head, no correction need be made for temperature or density since feet of head and gallons per minute account for these factors. However, density does increase the pump power requirements. The horsepower curves for water applications are developed at near maximum density (at approximately 85°F). Since density decreases as temperature rises, pump water horsepower will decrease, but the change is usually ignored. Viscosity can change the pump impeller head-capacity curve provided the change in viscosity is greater than the change of water viscosity between 40°F and 400°F. The effect on the curve is illustrated in Figure 5-19.

c. INSTALLATION CRITERIA

Some of the important points for the TAB Technician to observe in a well designed pump installation are:

(1) Suction piping should be air tight and free of air traps.

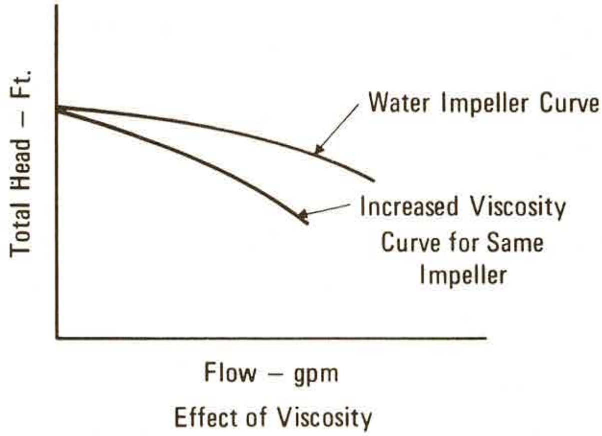


Figure 5-19 EFFECT OF VISCOSITY

(2) Piping should provide a smooth flow into the suction without unnecessary elbows.

(3) Suction pipe should be one or two sizes larger than pump inlet (eccentric reducer or reducing elbow to connect inlet to piping).

(4) There should not be any restrictions at the pump suction.

(5) Piping should be supported independently of the pump casing.

(6) Use of a radial ("silent") check valve in the pump discharge in multipump installations.

(7) Manual air vent in pump casing and piping.

(8) Pressure gauges on suction and discharge at the same elevation.

7. Pump Laws and Equations

Pump laws apply equally to open and closed systems. These laws show only relationships and cannot be used to calculate pump curves. They can be used to calculate only system curves so maybe they should have been called "system curve laws"! If one point of a system curve is known, the pump laws can be utilized to increase or decrease the flow up and down that system curve. As the flow is changed, pressure and horsepower will change. Even the laws which seemingly depend upon the pump really govern only the system curve. If impeller diameters are changed, the new flow characteristics move the operating point along the system curve to a new operating point.

When the system curve is changed (by opening or closing the discharge valve or cleaning the strainers, etc.), the operating point is moved along the pump curve to the new system curve. Naturally, by achieving the proper balance between the system and the pump, the desired result can be achieved.

In practice, it is much easier to change the system by using a balancing valve than it is to make changes to the pump. When trouble occurs and adjusting the discharge valve is not sufficient, then the pump must be involved. The system curve also will be changed by cleaning strainers.

a. AFFINITY LAWS OR PUMP LAWS

- Flow (capacity) varies *directly* as the *speed* or *impeller diameter.*
- *Head* varies as the *square* of the *speed* or *impeller diameter.*
- *Bhp* varies as the *cube* of the *speed* or *impeller diameter.*

b. PUMP EQUATIONS (BASED ON THE PUMP LAWS)

Equation 5-2

$$\frac{gpm_2}{gpm_1} = \frac{rpm_2}{rpm_1}$$

Equation 5-3

$$\frac{gpm_2}{gpm_1} = \frac{D_2}{D_1}$$

Equation 5-4

$$\frac{H_2}{H_1} = \left(\frac{rpm_2}{rpm_1}\right)^2$$

Equation 5-5

$$\frac{H_2}{H_1} = \left(\frac{D_2}{D_1}\right)^2$$

Equation 5-6

$$\frac{bhp_2}{bhp_1} = \left(\frac{rpm_2}{rpm_1}\right)^3$$

Equation 5-7

$$\frac{bhp_2}{bhp_1} = \left(\frac{D_2}{D_1}\right)^3$$

Where:

gpm = Flow (gallons per minute)

rpm = Revolutions per minute

D = Impeller diameter (inches)

H = Head (ft.w.g.)

bhp = Brake horsepower

c. HYDRONIC EQUATIONS

Some of the following hydronic equations can be found elsewhere in this manual, but they are repeated for convenience.

Equation 5-8

$$Q = 500 \times gpm \times \Delta t$$

Where:

gpm = Gallons per minute

Δt = Temperature differential (°F)

Q = Heat flow (btuh)

Equation 5-9

$$\frac{\Delta P_2}{\Delta P_1} = \left(\frac{gpm_2}{gpm_1}\right)^2$$

Where:

ΔP = Pressure differential (psi)

gpm = Gallons per minute

Equation 5-10

$$\Delta P = \left(\frac{gpm}{C_v}\right)^2$$

Where:

ΔP = Pressure differential (psi)

gpm = Gallons per minute

C_v = Valve constant

Equation 5-11

P = F/A

Where:

P = Pressure (psi)

F = Force (pounds)

A = Area (sq.in.)

Equation 5-12

$$H = \frac{fLv^2}{2gD}$$

Where:

h = Head loss (feet)

f = Friction factor (Moody)

L = Length of pipe (feet)

v = Velocity (fps)

g = Gravity (32.2 ft/sec^2)

D = Internal diameter (feet)

Equation 5-13

$$whp = \frac{gpm \times H \times Sp.\ Gr.}{3960}$$

Where:

whp = Water horsepower

H = Head (ft.w.g.)

Sp. Gr. = Specific gravity (use 1.0 for water)

d. HYDRONIC EQUIVALENTS

- One gallon water = 8.33 pounds
- Specific heat (C_p) water = 1.00 btu/lb °F @ 68°F)
- Specific heat (C_p) water vapor = 0.45 btu/lb °F (@ 68°F)
- One ft. of water = 0.433 psi
- One ft. of mercury (Hg) = 5.89 psi
- One cu.ft. of water = 62.4 lb = 7.49 gal.
- One in. of mercury (Hg) = 13.6 in.w.g. = 1.13 ft.w.g.
- Atmospheric Pressure = 29.92 in.Hg = 14.696 psi
- One psi = 2.31 ft.w.g. = 2.04 in.Hg

Water Temperature	Ft. head differential per in.Hg. differential
60°F	1.046
150°F	1.07
200°F	1.09
250°F	1.11
300°F	1.15
340°F	1.165

8. Pump Location

Pump location varies with the size and type of system. Figures 5-22, 5-23, and 5-26 illustrate pumps in the supply main from the boiler or chiller, while Figure 5-21 has the pump in the return piping. A pump in the boiler return is acceptable for small systems when pump head is low (12 foot head or less), the compression tank is on the boiler (or a nearby main), and the highest piping and radiation is maintained at a static pressure greater than full pump head. These conditions apply to most residential systems.

When pump head is equal to or greater than the difference between boiler fill and relief valve discharge pressures, or when highest piping or radiation can be at a static pressure less than total pump head, the pump must be located on the supply side of the boiler, with the compression tank at the pump inlet, as illustrated in Figure 5-26. This assures that pump cycling will not cause excessive pressure variations in the boiler and will not cause subatmospheric pressure at topmost system points to produce air leakage into the system. Pump cavitation is prevented by locating a properly sized compression tank near the pump inlet.

C HYDRONIC SYSTEM COMPONENTS

1. Heating and Cooling Sources

a. BOILERS

A boiler is a cast-iron or steel pressure vessel heat exchanger, designed with and for fuel burning devices and other equipment to (1) burn fossil fuels (or use electric current) and (2) transfer the released heat to water (in water boilers) or to water and steam (in steam boilers). Boiler heating surface is the area of fluid-backed surface exposed to the products of combustion, or the *fire-side* surface. Various codes and standards define allowable heat transfer rates in terms of heating surface. Boiler design provides for connections to a piping system which delivers heated fluid to the place of use and returns the cooled fluid to the boiler.

b. HEAT EXCHANGERS

Heat exchangers or converters are used as heat sources for many hot water heating systems. Heat exchangers may be of three general types: (1) steam-to-water; (2) water-to-water; or (3) water-to-steam (generators).

Steam-to-water exchangers usually take the form of shell-and-tube units. Steam is admitted to the shell, and water is heated as it circulates through the tubes. Steam-to-water converters are useful where an addition is to be made to an existing steam system and where hot water heating is desired. They are also widely used in areas where district steam is available and individual buildings are to be heated with a hot water system. High-rise buildings can be zoned vertically by using steam distribution and installing converters at various levels to serve several floors, thus limiting maximum operating pressures in the zone.

Water-to-water heat exchangers (generally shell-and-tube units) are used in high temperature water (HTW) systems to produce lower temperature water for certain zones or in process water or domestic water services.

Water-to-steam heat exchangers generally consist of a U-tube bundle installed in a tank or pressure vessel to provide space for the release of steam. They are used in HTW systems to provide process steam where required.

c. WATER CHILLERS

The source of cooling in a chilled water or a dual-temperature system is a water chiller. There are three general types of water chillers: (1) reciprocating, (2) centrifugal, and (3) absorption.

d. HEAT PUMPS

A heat pump may serve as a source for both hot water and chilled water in a dual-temperature system. Water temperatures available are generally low in winter (about 90°F to 130°F) and terminal heat transfer must be designed for operation under these conditions. In some cases, a supplementary heat source is used to raise temperature levels.

2. Terminal Heating and Cooling Units

a. GENERAL

Many types of terminal units are available for central water systems. Some are suited to only one type of system and others may be used in all types of systems. Terminal units may be classified in several ways:

(1) *Natural convection units,* including cast-iron radiators, cabinet convectors, baseboard and finned tube radiation. These units are used in heating systems.

(2) *Forced convection units,* including unit heaters, unit ventilators, fan-coil units, induction units, air handling units, heating and cooling coils in central station units, and most process heat exchangers. Fan-coil units, unit ventilators, and central station units can be used for heating, ventilating, and cooling.

(3) *Radiation*, including panel systems, unit radiant panels, and certain special types of cast-iron radiation. All transfer some heat by convection. Such units are generally used for heating in low temperature water (LTW) systems. However, special designs of overheat radiant surfaces, both tubular and panel, are being used in medium and high temperature water systems to take advantage of the lowered surface requirements achieved through the use of high surface temperatures. Panel cooling is applied in conjunction with control space humidity to maintain the space dew point below the panel surface temperature.

In any single circuit having similar loads and a single control point, the terminal units should be of similar response types. Cast iron radiators should not be installed in the same controlled circuit as baseboard or convector units. Caution should be exercised when including fan-operated units with natural convection units on the same pumping circuit.

b. RADIATORS AND CONVECTORS

Cast-iron radiation and cabinet convectors are widely used in LTW systems. Ceiling-hung radiators are frequently used where floor space may not be available for other units. Pressure limitations must be considered for cast iron radiation. Convectors are used extensively in areas where high output is needed and limited space is available, and where linear heat distribution is not desired. Typical areas heated include corridors, entries, toilet rooms, storage areas, work rooms, and kitchens.

c. BASEBOARD AND FIN TUBE RADIATION

Baseboard and fin tube radiation permits the blanketing of exposed surfaces or maximum comfort. Baseboard and fin tube elements are generally rated at various average water temperatures and at one or more water velocities. Velocity corrections may be applied. Many designers feel that these units are thus limited to systems designed to a 20°F TD. However, careful selection can result in successful application with temperature drops much higher than 20°F.

d. UNIT VENTILATORS

Unit ventilators, originally developed for specific application in school classrooms, are being used today in a much wider range of applications. Unit ventilators consist of a forced convection heating or cooling unit with dampers permitting introduction of controlled amounts of outdoor air to provide a complete cycle of heating, ventilating, ventilation cooling, or mechanical cooling as required. Condensation may be a problem during summer operation unless chilled water flow is stopped when fans are not operating. Condensate drains are necessary. Comparatively low supply temperature and rise may be required.

e. FAN-COIL AND INDUCTION UNITS

Fan-coil units are generally used, with or without outdoor air, in dual-temperature water systems. The same coil is often used for both heating and cooling. Individual control is usually achieved by the use of valves, or by using intermittent or multispeed fan operation. Hot water ratings are usually based on flow rates or temperature drops at various entering water and air temperatures. Temperature drops of 40°F to 60°F are frequently used. Induction units are similar to fan-coil units except that air circulation is provided by a central air system which handles part of the load, instead of a blower in each cabinet.

f. UNIT HEATERS

Unit heaters are available in several types: horizontal propeller fan, downblow, and cabinet. They are used where high output in a small space is required, and where no cooling is to be added. Cabinet units are frequently applied in corridors and at entrances to blanket doors which are frequently opened. Normally, unit heaters do not provide ventilation air.

g. CENTRAL STATION HVAC SYSTEMS

Central station systems have chilled water coils and/or hot water coils, and are available in a variety of sizes and types. Air capacities may range from a few hundred to many thousand cfm delivery. Single zone units may be used for heating, ventilating, or cooling, from a single dual-temperature system with a single-duct air distribution system.

Multizone systems use separate heating and cooling coils discharging air simultaneously through separate duct systems. Temperature in each system is separately controlled. Central station HVAC units may be complete factory-assembly units or may be built up on the job from components selected and matched by the designer. See Chapter IV for detailed information on HVAC duct systems.

3. Compression/Expansion Tanks

Compression or expansion tanks are used in closed hydronic systems for the following reasons:

(1) Allows the expansion and contraction of the system fluid due to heating and cooling.

(2) Maintains a minimum safe operating pressure as set by the system pressure reducing valve (make-up water).

(3) Provides a safe location within the system for air to be stored (air removed from system by air separation device) until purged.

Some systems utilize bladder tanks in which the air charge within the tank is separated from system water with the use of a rubber diaphragm. Care should be used to confirm proper bladder pressure in addition to installation of an external system air control (automatic air vent).

4. Piping System Components

a. AIR CONTROL AND VENTING

If air and other gases are not eliminated from the flow circuit, they may cause air binding in the terminal heat transfer elements and noise in the piping circuit. High points in piping systems and terminals units should be vented with manual or automatic air vents. As automatic air vents may malfunction, valves should be provided at each vent to permit service without draining the system. The discharge of each vent should be piped to a point where water can be wasted into a drain or container. If a plain expansion tank is used, free air contained in the circulating water should be removed from the piping circuit and trapped in the expansion tank by a boiler dip tube or other air separation devices. If a diaphragm-type tank is used, all air should be vented from the system.

b. DRAINS AND SHUTOFFS

All low points should be equipped with drains. Provisions should be made for separate shutoff and drain of individual equipment and circuits so that the entire system does not have to be drained for service of a particular item.

c. BALANCE FITTINGS

Balance fittings should be applied as needed to permit balancing of individual terminals and major subcircuits. Such fittings should be placed at the circuit return when possible.

d. PITCH

Piping need not pitch but can be run level, providing flow velocities in excess of 1.5 feet per second are maintained.

e. STRAINERS

Strainers should be used where necessary to protect the elements of a system. Strainers placed in the pump suction need to be analyzed carefully to avoid cavitation. Large separating chambers are available which serve as main air venting points and dirt strainers ahead of pumps. Automatic control valves or spray nozzles operating with small clearances require protection from pipe scale, gravel, welding slag, etc., which may readily pass through the pump and its protective separator. Individual fine mesh strainers may therefore be required ahead of each control valve. Condenser water systems without water regulating valves do not necessarily require a strainer. If a cooling tower is used, the strainer provided in the tower basin will be usually adequate.

f. THERMOMETERS

Thermometers or thermometer wells should be installed to assist the system operator and the TAB technician, and to use for troubleshooting. Permanent thermometers with correct scale range and separable sockets should be used at all points where temperature readings are regularly needed. Thermometer wells should be installed where readings will be needed only during start-up and balancing.

g. FLEXIBLE CONNECTORS

Flexible connectors are sometimes installed at pumps and machinery to reduce pipe vibration. Vibrations are transmitted through the water column across a flexible connection and reduce the effectiveness at the connector. Flexible connectors, however, prevent damage caused by misalignment of equipment piping flanges.

h. GAUGES

Gauge cocks should be installed at points where pressure readings will be required. Note that gauges permanently installed in the system will deteriorate due to vibration and pulsation, and will not be reliable when needed.

D HYDRONIC PIPING SYSTEMS

1. General

A hydronic or all-water system is one in which hot or chilled water is used to convey heat to or from a conditioned space or process through piping connecting a boiler, water heater, or chiller with suitable terminal heat transfer units located at the space or process.

All-water systems may be classified by:

(1) Temperature

(2) Generation of flow

(3) Pressurization

(4) Piping arrangement

(5) Pumping arrangement

In terms of flow generation, hot water heating systems are of two types (1) the *gravity* system, in which circulation of the water is due to the difference in weight between the supply and return water columns of any circuit or system; and (2) the *forced* system in which a pump, usually driven by an electric motor, maintains the necessary flow. Water systems can be either once-through or recirculating systems.

2. Temperature Classifications

Water systems may be classified according to operating temperature as follows:

a. LOW TEMPERATURE WATER SYSTEM (LTW)

A hot water heating system operating within the pressure and temperature limits of the ASME boiler construction code for low pressure heating boilers. The maximum allowable working pressure for low pressure heating boilers is 160 psi with a maximum temperature limitation of 250°F. The usual maximum working pressure for boilers for LTW systems is 30 psi, although boilers specifically designed, tested, and stamped for higher pressures may frequently be used with working pressures to 160 psi. Steam-to-water or water-to-water heat exchangers also are used.

b. MEDIUM TEMPERATURE WATER SYSTEM (MTW)

A hot water heating system operating at temperatures of 350°F or less, with pressure not exceeding 150 psi. The usual design supply temperature is approximately 250° to 325°F, with a usual pressure rating for boilers and equipment of 150 psi.

c. HIGH TEMPERATURE WATER SYSTEM (HTW)

A hot water heating system operating at temperatures over 350°F and usual pressures of about 300 psi. The maximum design supply water temperature is 400° to 450°F, with a pressure rating for boilers and equipment of about 3000 psi. It is necessary that the pressure-temperature rating of each component be checked against the design characteristics of the particular system.

d. CHILLED WATER SYSTEM (CW)

A chilled water-cooling system operating with a usual design supply water temperature of 40° to 55°F, and normally operating within a pressure range of 125 psi. Antifreeze or brine solutions may be used for systems (usually process applications) which require temperatures below 40°F. Well water systems may use supply temperatures of 60°F or higher.

e. DUAL-TEMPERATURE WATER SYSTEM (DTW)

A combination hot water heating and chilled water cooling system which circulates hot and/or chilled water to provide heating or cooling using common piping and terminal heat transfer apparatus. They are operated within the pressure and temperature limits of LTW systems, with usual winter design supply of water temperatures about 100°F to 150°F and summer supply water temperatures 40°F to 55°F.

3. Types of Hydronic Systems

Generally, the most economical distribution system layout has mains that are run by the shortest and most convenient route to the terminal equipment having the largest flow rate requirements, and branch or secondary circuits are then connected to these mains.

Water distribution mains are most frequently located in corridor ceilings, above hunt ceilings, wall-hung along a perimeter wall, or in pip trenches, crawl spaces, or basements. Water system piping need not be run at a definite level or pitch, but may change up or down as required by architectural or structural needs. Water system piping may be divided into two arbitrary classifications:

(1) Pipe circuits suitable for complete small systems or for terminal or branch circuits on large systems.
  (a) Series loop.
  (b) One-pipe.
  (c) Two-pipe *reversed*-return.
  (d) Two-pipe *direct*-return.

(2) Main distribution piping used to convey water to and from the terminal units or circuits in large system.
  (a) Two-pipe *direct*-return.
  (b) Two-pipe *reversed*-return.
  (c) Three-pipe.
  (d) Four-pipe.

a. SERIES LOOP SYSTEMS

A series loop is a continuous run of pipe or tube from supply connection to return connection. Terminal units are a part of the loop. Figure 5-20 shows a system of two series loops on a supply and return main *(split series loop)*. One or many series loops may be used in a complete system. Loops may connect to mains, or all loops may run directly to and from the boilers. Water temperature drops progressively as each radiator transfers heat to the air, the amount of drop depending on radiator output and water flow rate.

A decrease in loop water flow rate increases temperature drop in each unit and in the entire loop. Average water temperature shifts downward progressively from first to last radiator in series. Unit output gradually lowers from first to last on the loop. Consequently, comfort cannot be maintained in separate spaces heated with a single series loop if water flow rate is varied. Control of output from individual terminal units on a series loop is impractical except by control of heated airflow. Manual dampers can be used on natural convection units; automatic fan or face-and-bypass damper control can be used on forced air units.

b. ONE-PIPE SYSTEMS (DIVERTING FITTING)

One-pipe circuits (Figure 5-21) use a single loop main. For each terminal unit, a supply and a return tee are installed on the same main. One of the two

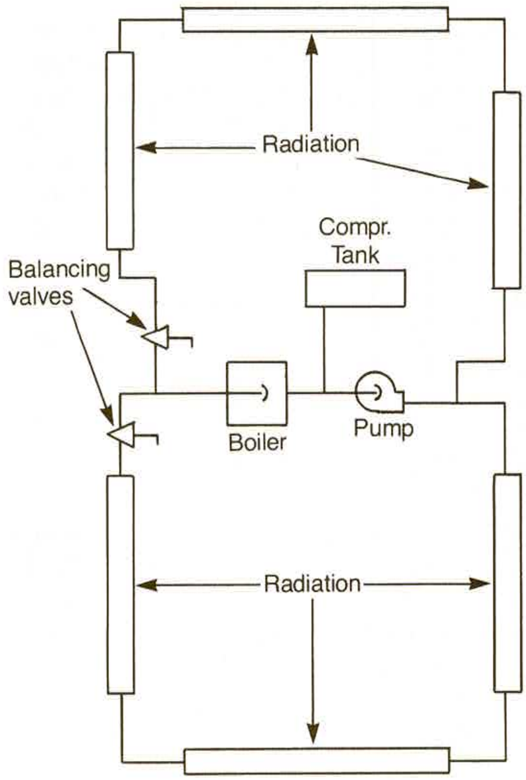


Figure 5-20 A SERIES LOOP SYSTEM—TWO CIRCUITS

tees is a special diverting tee which creates a pressure drop in main flow to divert a portion of main flow to the unit. One (return) diverting tee is usually sufficient for upfeed (units above main) systems. Two special fittings (supply and return tees) are usually required for downfeed units to overcome thermal head. Special tees are proprietary; consult manufacturer's literature for flow rates and pressure drop data.

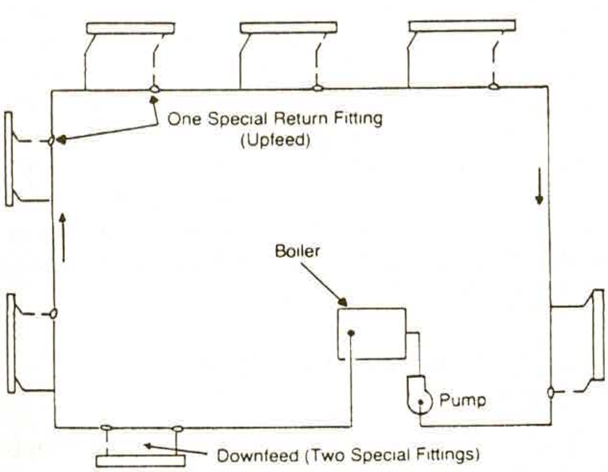


Figure 5-21 A ONE-PIPE SYSTEM

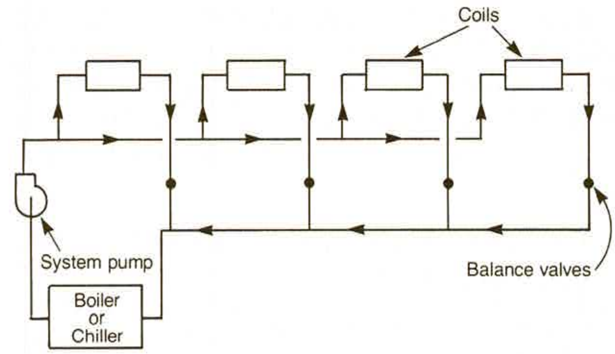


Figure 5-22 DIRECT RETURN TWO-PIPE SYSTEM

One-pipe circuits allow manual or automatic control to flow to individual connected heating units. On-off rather than flow modulation control is advisable because of the relatively low pressure and flow diverted. Length and load imposed on a one-pipe circuit are usually small because of the limitations listed.

c. TWO-PIPE SYSTEMS

Two-pipe circuits (Figures 5-22 and 5-23) may be direct-return (return main flow direction is opposite supply main flow; return water from each unit takes the shortest path back to the boiler) or reverse-return (return main flow is in the same direction as supply flow; after the last unit is fed, the return main returns all water to the boiler). The direct-return system is popular because less main pipe length is required; however, circuit balancing valves usually are required on units or subcircuits. Since water flow distance from and to the boiler is virtually the same through any unit on a reverse-return system, balancing valves are seldom adjusted. Operating (pumping) cost is likely to be higher with direct return because of the added balancing fitting pressure drops at the same flow rate.

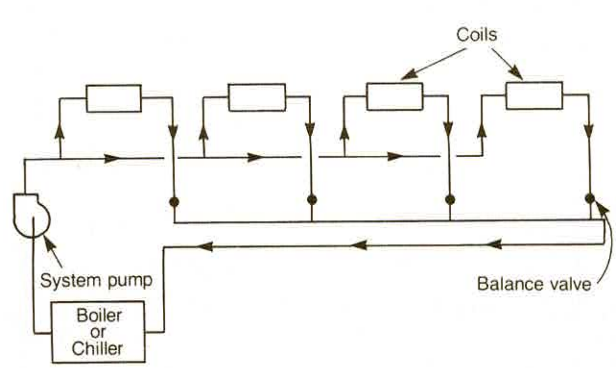


Figure 5-23 REVERSE RETURN TWO-PIPE SYSTEM

d. THREE-PIPE SYSTEMS

The three-pipe system (Figure 5-24) satisfies variations in load by providing independent sources of heating and cooling to the room unit in the form of constant temperature primary air and secondary chilled or hot water.

The unit contains a single secondary water coil. A three-way valve at the inlet of the coil admits the water from either the hot or cold water supply, as required. The water leaving the coil is carried in a common pipe to either the secondary cooling or heating equipment. The usual room control for three-pipe systems is a special three-way modulating valve which modulates either the hot or the cold water in sequence, but does not mix the streams. The primary air is cold and at the same temperature year-round.

During the period between seasons, if both hot and cold secondary water is available, any unit can be operated within a wide capacity range from maximum cooling to maximum heating within the limits set by the temperature of the secondary chilled or hot water. Any unit in the system can be operated through its full range of capacity without regard to the operation of any other unit in the system, recognizing the operating cost penalty that will result from simultaneous heating and cooling loads. All units are selected on the basis of their peak capacity requirements.

The return mix three-pipe room unit is provided with a single coil which receives either hot or cold water. A modulating three-way valve at the inlet to the unit admits either hot water or cold water to the secondary coil (see Fig. 5-25). The three-way valves are a special design in which the hot port gradually moves from open to fully closed, and the cold port gradually moves from fully closed to open. The valves are constructed so that at mid-range there is an interval in which both ports are completely closed. Room control action is the same during all seasons.

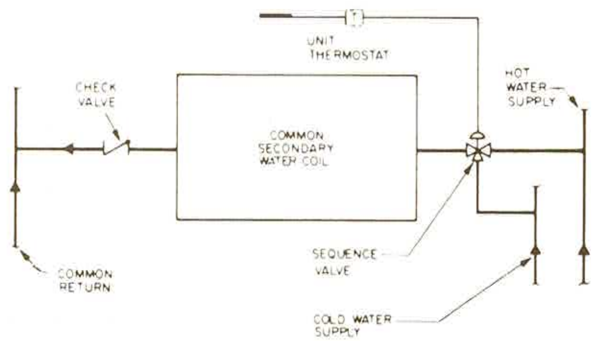


Figure 5-25 RETURN MIX SYSTEM/ROOM UNIT CONTROLS

e. FOUR-PIPE SYSTEMS

Four-pipe systems for induction, fan-coil, or radiant panel systems derive their name from the four pipes to each terminal unit. As noted before, these pipes are a cold water supply, cold water return, warm water supply, and warm water return. The four-pipe system satisfies variation in cooling and heating to the room

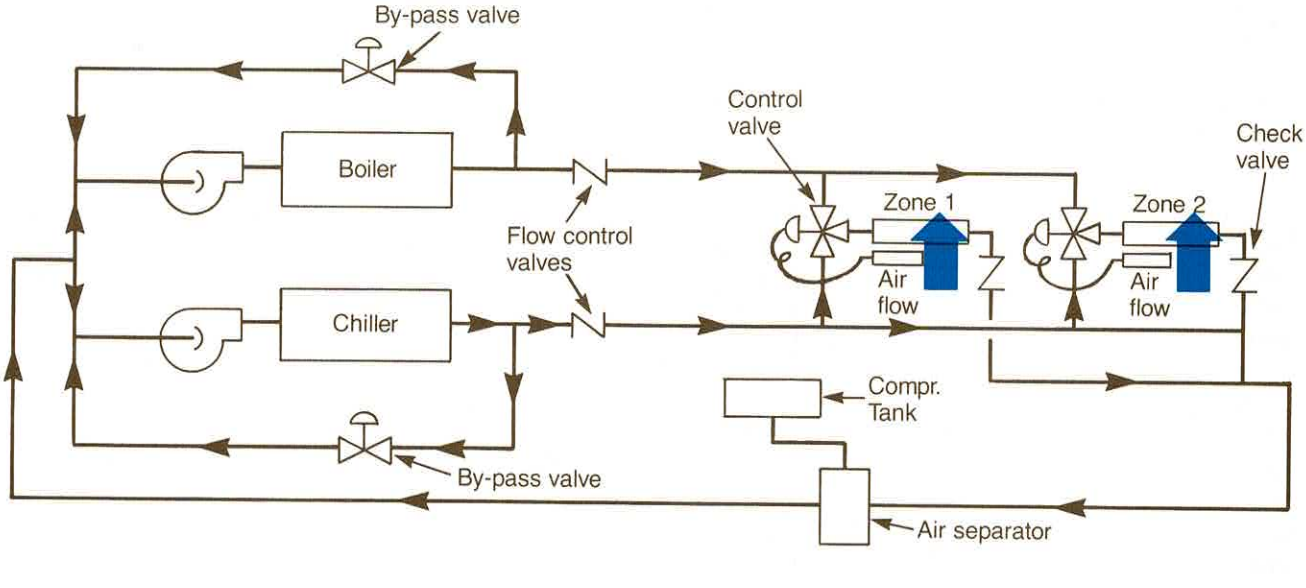


Figure 5-24 THREE-PIPE SYSTEM

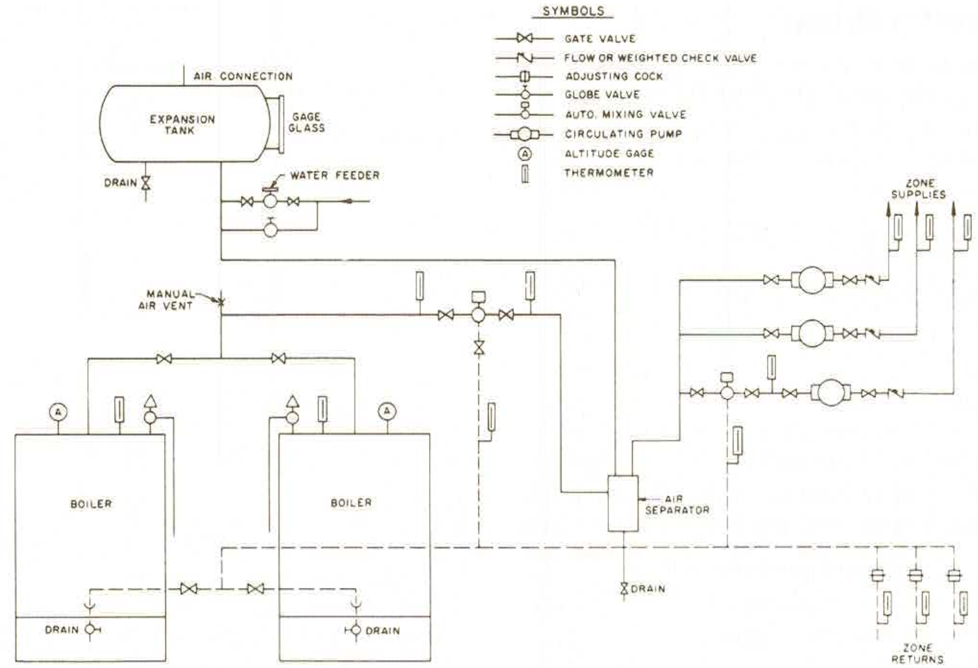


Figure 5-26 BOILER PIPING FOR A MULTIPLE-ZONE MULTIPLE-PURPOSE HEATING SYSTEM

unit in the form of constant temperature primary air, secondary chilled water, and secondary hot water.

The terminal unit (Figure 5-27A) is usually provided with two independent secondary water coils, one served by hot water, the other by cold water. The primary air is cold and remains at the same temperature year-round. During peak cooling and heating, the four-pipe system performs in a manner similar to the two-pipe system, with essentially the same operating characteristics. During the period between seasons, any unit can be operated at any capacity level from maximum cooling to maximum heating, if both cold water and warm water are being circulated. Any unit can be operated at or between these extremes without regard to the operation of any other unit.

Figure 5-27B illustrates another unit and control configuration which is sometimes used. A single secondary water coil is provided at the unit, and three-way valves located at the inlet and leaving side of the coil admit the water from either the hot or cold water supply, as required, and divert it to the appropriate return pipe. This arrangement requires a special three-way modulating valve, originally developed for one form of the three-pipe system, which controls the hot or cold water selectively and proportionally but does not mix the streams. The valve at the coil outlet is a two-position valve open to either the hot or cold water return, as required.

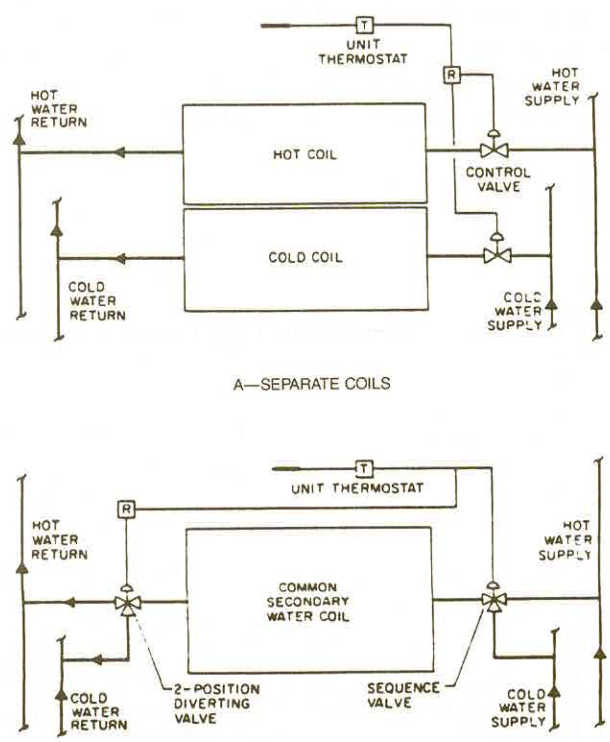


Figure 5-27 FOUR-PIPE SYSTEM/ROOM UNIT CONTROLS

f. COMBINATION PIPING CIRCUITS

The four basic arrangements exist only to describe function; one type can grade into another; a piping system can contain from one to all four types and thus cannot be described as a particular type. Figure 5-28 illustrates a *primary circuit* and two *secondary circuits.* As pipe lengths and number of units vary, and as circuit types are combined, basic names for piping circuits become meaningless; flow, temperatures, and head must be determined for each circuit and for the complete system.

Figure 5-29 shows primary and secondary water circuits completely separated hydraulically. Secondary water cooling is accomplished by a water-to-water heat exchanger, which receives primary water from the chiller to provide the secondary cooling. This arrangement makes it impossible for either the primary or secondary circuit to be affected hydraulically by the operation of the other circuit. In large buildings where several secondary water circuits are served by a single primary system, there may be advantages to separating the water circuits. In high-rise buildings, the secondary water distribution should be divided horizontally into two or more zones in order to limit system pressures caused by height. The zones are isolated from each other hydraulically, with one or more using a primary-to-secondary heat exchanger for secondary cooling.

g. SUMMER-WINTER SYSTEM

A summer-winter system utilizes one coil in each terminal unit to accomplish both the heating and cooling requirements of the space. Chilled water is supplied in the summer and hot water in the winter to each terminal unit through summer-winter changeover valves (manual or automatic) normally located at the primary chilled and hot water heat exchange equipment. Summer-winter systems do not offer the flexibility of simultaneous operation of hot and chilled water operation like a four-pipe system, but are an economical and widely used piping arrangement utilized in hotels, garden apartments, and other installations where seasonal control is satisfactory.

4. System Flow

a. CONSTANT VOLUME SYSTEMS

A constant volume piping system is where the required system flow rate does not change during normal system operations. Constant volume systems may be classified as straight-through or 3-way.

(1) Straight-Through

A system that does not contain any automatic temperature control valves in the piping circuit. Flow through the source, outlets, and piping remains constant. Temperature control is accomplished by the modulation of system water temperature, or the use

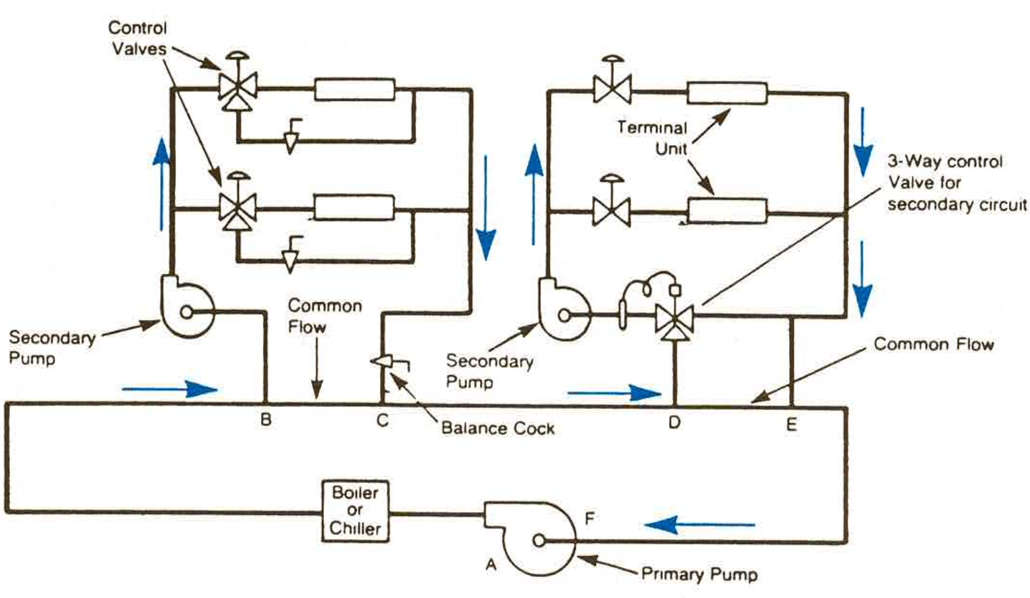


Figure 5-28 EXAMPLE OF PRIMARY AND SECONDARY PUMPING CIRCUITS

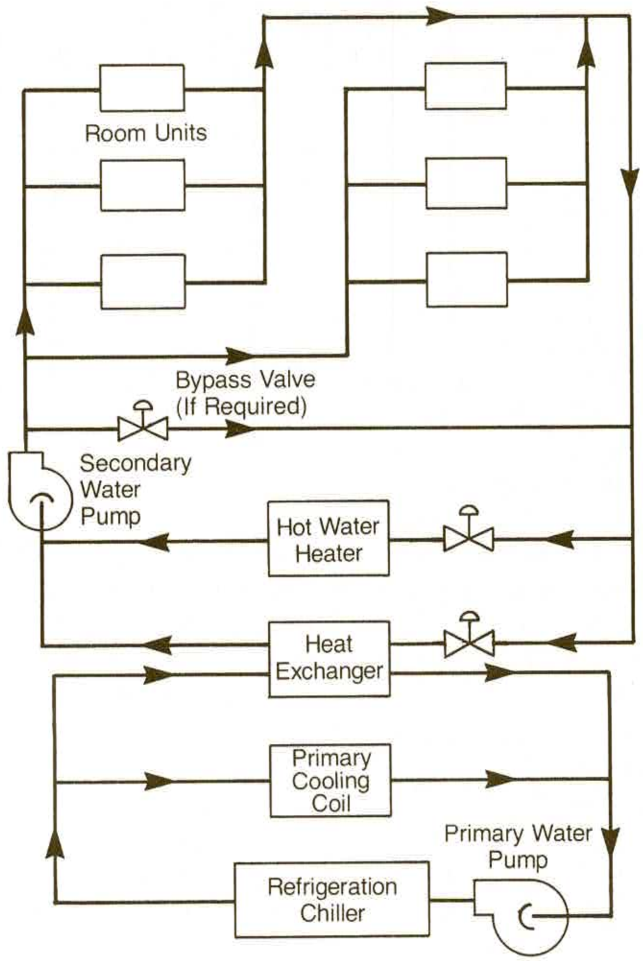


Figure 5-29 TWO-PIPE SYSTEM WATER DISTRIBUTION WITH SEPARATED CIRCUITS

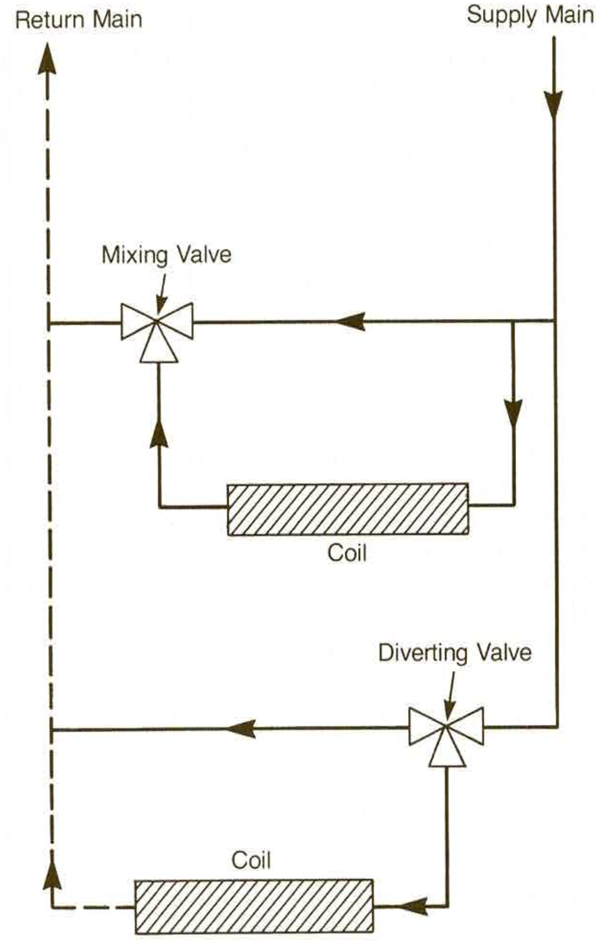


Figure 5-30 THREE-WAY VALVES CONTROLLING COIL FLOW RATE

of dampers at terminal units. An application of this system is constant volume finned tube radiation with system water temperature reset through an outside air controller (system water temperature increases with a decrease in outside air temperature).

(2) Three-Way Valves

Constant system flow may also be achieved with the use of the three-way automatic temperature control valves at terminal units (Figure 5-30). Three-way (or bypass) valves allow circulation around a terminal unit when heat transfer is not required. System water temperatures may remain constant with systems utilizing three-way control valves allowing closer temperature control than is found in a straight-through system. Two types of three way valves are used, diverting and mixing.

DIVERTING VALVE—A diverting valve is primarily used in two position control applications (open & closed) where the flow is required to be diverted from one pipe to another. The valve therefore has one inlet and two outlets (see Figure 5-31). In the case of terminal units, the valve diverts supply water from the coil to the bypass.

MIXING VALVE—Similar to a diverting valve in appearance, a mixing valve has three ports, but that is where the similarity ends. Mixing valves are primarily used where proportional, or modulating control, is required. The valve contains two inlets and one outlet (see Figure 5-32), allowing the mixture of two fluids. In the case of terminal units, the valve allows the mixing of bypass and coil return flow to the common return piping of the system.

Note: Mixing valves may be required for two-position control applications, in essence, "diverting" flow. This does not constitute nor require a diverting valve! Mixing and diverting valves have been designed and subsequently named for their internal construction and flow handling capabilities.

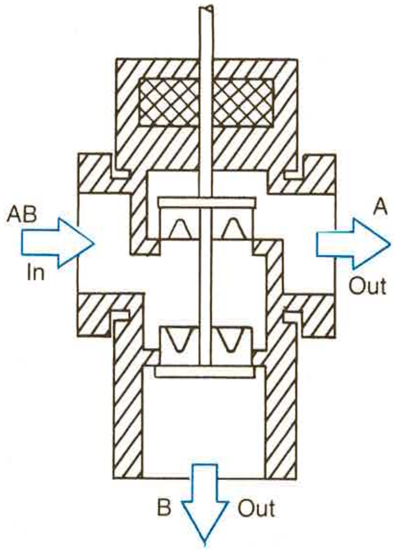


Figure 5-31 DIVERTING VALVE

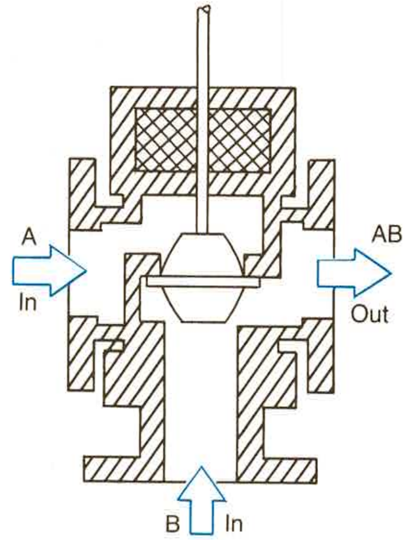


Figure 5-32 MIXING VALVE

b. VARIABLE VOLUME SYSTEMS

A variable volume piping system is where the required system flow rate varies during normal system operation. Variable volume systems have been created with the use of two-way automatic temperature control valves at terminal units.

A variable volume piping system is where the required system flow rate varies during normal system operation. Variable volume systems have been created with the use of two-way automatic temperature control valves at terminal units.

(1) Two-Way Valves

Two-way control valves may either operate with two-position or proportional control, both methods reducing flow rate through a terminal unit when heat transfer is not required. Figures 5-33 and 5-34 show the construction of typical two-way control valves. The use of the two-way valves at system terminal units results in varied system flow rates and pressures during normal operation. A possibility of extremely low flow rates at pumps and primary heat exchange equipment, in addition to system differential pressures that may cause leakage or failure of automatic control valves, is present when a majority of units are isolated by two-way control valves. Two methods are commonly used to maintain minimum flow rates and pressures required to maintain satisfactory system operation, differential pressure control valves and variable speed pumping.

DIFFERENTIAL PRESSURE CONTROL VALVE (DPCV)—As the two-way control valves at terminal units close, system differential pressure increases and is monitored by a differential pressure controller. The controller modulates the DPCV to maintain a

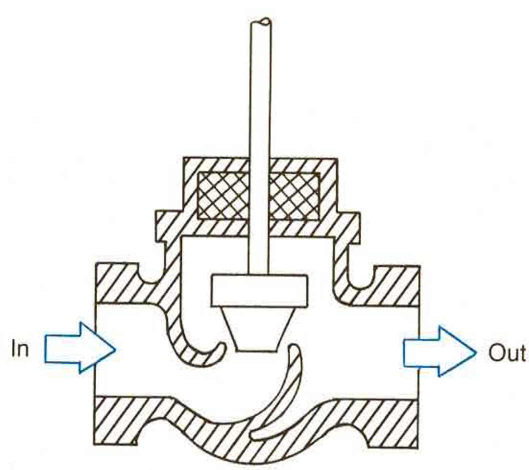


Figure 5-33 SINGLE SEATED TWO-WAY VALVE

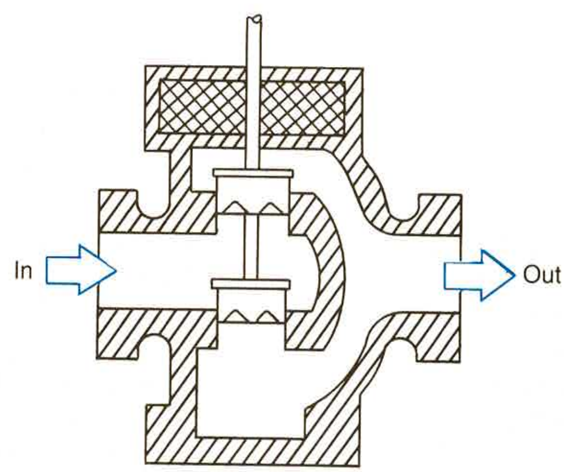


Figure 5-34 DOUBLE SEATED TWO-WAY VALVE

Table 5-5 HYDRONIC TROUBLE ANALYSIS GUIDE

Complaint	Possible Cause	Recommended Action
Pump or system noise	Shaft misalignment	• Check and re-align.
	Worn coupling	• Replace and re-align.
	Worn pump/motor bearings	• Replace, check manufacturer's lubrication recommendations. • Check and re-align shafts.
	Improper foundation or installation	• Check foundation bolting or proper grouting. Check possible shifting due to piping expansion/contraction. • Re-align shafts.
	Pipe vibration and/or strain caused by pipe expansion/contraction	• Inspect, alter or add hangers and expansion provision to eliminate strain on pump(s).
	Water velocity	• Check actual pump performance against specified and reduce impeller diameter as required. • Check for excessive throttling by balance valves or control valves.
	Pump operating close to or beyond end point of performance curve	• Check actual pump performance against specified and reduce impeller diameter as required.
	Entrained air or low suction pressure	• Check expansion tank connection to system relative to pump suction • If pumping from cooling tower sump or reservoir, check line size. • Check actual ability of pump against installation requirements. • Check for vortex entraining air into suction line.

Complaint	Possible Cause	Recommended Action
Inadequate or no circulation	Pump running backwards (3-phase)	• Reverse any two-motor leads.
	Broken pump coupling	• Replace and re-align.
	Improper motor speed	• Check motor nameplate wiring and voltage.
	Pump (or impeller diameter) too small	• Check pump selection (impeller diameter) against specified system requirements.
	Clogged strainer(s)	• Inspect and clean screen.
	System not completely filled	• Check setting of PRV fill valve. • Vent terminal units and piping high points.
	Balance valves or isolating valves improperly set	• Check settings and adjust as required.
	Air-bound system	• Vent piping and terminal units. • Check location of expansion tank connection line relative to pump suction. • Review provision for air elimination.
	Air entrainment	• Check pump suction inlet conditions to determine if air is being entrained from suction tanks or sumps.
	Low available NPSH	• Check NPSH required by pump. • Inspect strainers and check pipe sizing and water temperature.

system differential pressure satisfactory for pump and primary heat exchange equipment operation. A DPCV is normally found in smaller variable volume systems.

VARIABLE SPEED PUMPING—Similar in control as described for a DPCV, the differential pressure controller resets the pump speed reducing system differential pressure. This control application is utilized in larger variable volume applications due to higher installation costs, but results in lower operating costs than with the use of a DPCV.

5. A Final Note

As few systems tested are as straight forward as those pictured within this chapter, you do not need to become frustrated. The fact remains that *ALL* systems contain the three basic components:

(1) Source

(2) Outlet

(3) Piping to & from

With this knowledge and an understanding of basic piping applications, no system will be too difficult to understand and to balance. Table 5-5 is a handy chart for you to use when balancing a hydronic system and/or when trouble is encountered. This list can be expanded easily as your experience dictates.

NEBB

CHAPTER 6 TEMPERATURE CONTROL SYSTEMS

A HVAC SYSTEM CONTROL BASICS

The proper operation of automatic temperature control (ATC) systems lies with the temperature control contractor or the installing contractor, and is not the responsibility of the TAB contractor. However, one should now realize how important automatic control valves and dampers are to good system operation. A TAB technician must know how changes in control settings can effect the operation of a unit or of a whole system. Without this knowledge, testing and balancing cannot be accomplished, and troubleshooting becomes an impossibility.

For example, using the three-pipe system with terminal units shown in Figure 6-1, each terminal unit is controlled by a three-way "blending" valve. To balance the chilled water side of this system, all three-way valves must be opened to the chilled water supply main. These valve position settings then become important to the TAB technician. One valve in a system left indexed to the hot water main could invalidate the entire set of readings, causing the work to be done again. This is but one example to show that some knowledge of automatic temperature control systems is necessary for TAB technicians.

Automatic temperature controls are used in HVAC systems to maintain design conditions within an occupied space. Maintaining design conditions involves the control of temperature, humidity, and pressure. Other automatic controls provide for the safe and economical operation of the entire HVAC system.

1. Types of ATC Systems

Control systems are classified by the source of power. Four basic types of control systems are used:

- Electric controls
- Electronic controls
- Pneumatic controls
- Self-Contained controls

a. ELECTRIC CONTROLS

Electric controls use low voltage and line voltage current to modulate control devices such as dampers or

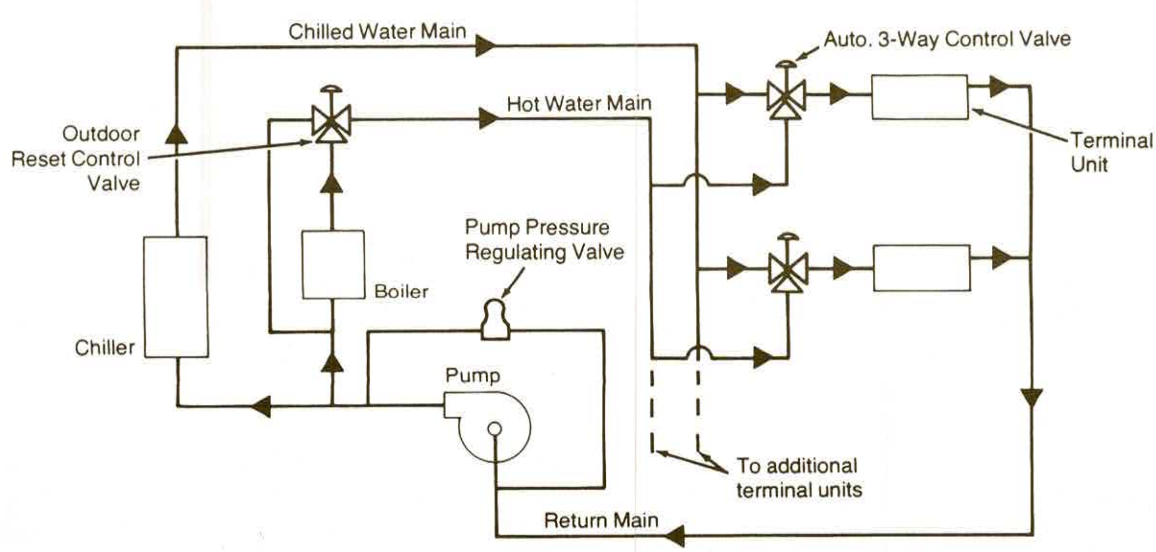


Figure 6-1 ARRANGEMENT OF A THREE-PIPE HYDRONIC SYSTEM

valves. Electric controls are usually two position and are used in smaller ATC applications. Larger and more complex systems normally use pneumatic or electronic control systems.

b. ELECTRONIC CONTROLS

Electronic control systems use electricity as a source of power. Small electric currents are used to transport signals to amplifiers where the signal is strengthened and used to operate a control device similar to those used in electric systems. Direct Digital Control (DDC) systems fall under this category.

c. PNEUMATIC CONTROLS

Compressed air is used as the source of this control system. This system provides proportional, or modulating control, of diaphragm or piston type control devices.

d. SELF-CONTAINED CONTROLS

Self-contained controls differ from all of the previous control systems in that they do not use an external source of power. Power to control is developed internally, or from pressures within the system. Condenser water regulating valves and thermostatic expansion valves are examples of self-contained controls.

Even though there are four distinct classifications of control systems, it is possible and likely that a project may utilize a combination of the above systems to satisfy the scope of control requirements.

2. Control Loops

No matter which type of control system is used, or whether the purpose is to control cooling, heating, humidity, or pressure, all control applications must involve a fundamental *control loop*. A control loop consists of three components:

- a controller (thermostat, receiver-controller)
- a controlled device (valve, damper)
- a sensing device (transmitter, bi-metal strip)

The loop is never-ending, just the way it should be if continuous control is desired.

For example, a sensing device (remote bulb) monitors the temperature of a supply air duct and sends a signal to the controller (see Figure 6-2).

The controller monitors the signal as sent by the sensing device, and reacts by either opening or closing a controlled device (the steam valve). As a result of more or less of the controlled agent (steam in the duct heating coil), the action of the controlled device creates a change in the air temperature of the duct

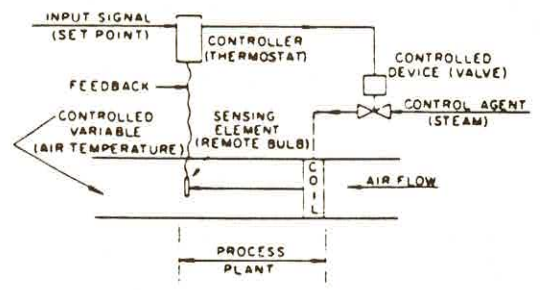


Figure 6-2 DISCHARGE AIR TEMPERATURE CONTROL

and causes the sensing device to change the signal once again to the controller.

The operation of the control loop is never-ending during normal operation of the HVAC system.

3. Types of Control Action

Controllers (such as thermostats, humidistats, and receiver controllers) have two possible sets of control actions:

- Modulating or Two-position
- Direct or Reverse Acting

a. MODULATING/TWO-POSITION

Modulating control (also called proportional control) is obtained when the control signal sent by the controller to the controlled device is constantly changing in small increments to gradually increase or decrease the capacity of a terminal unit to suit the load conditions. *Two-position control* (which also can be on-off) may only assume two positions; fully open or fully closed. Two-position controllers are normally used with equipment that may only operate in an on-off application. Equipment such as gas valves or small hermetic air conditioning compressors would fall under this category.

b. DIRECT/REVERSE ACTING

Pneumatic controllers may increase or decrease the output (branch) control pressure with changes in space conditions monitored by the sensing element. A *direct acting* controller will increase the output (branch) control pressure as the controlled variable (temperature, humidity, pressure) increases. A *reverse acting* controller increases control pressure as the controlled variable decreases. The action of the pneumatic controller must be properly matched with the control device, or the control loop will produce unexpected results or those opposite of that desired.

The position of a controlled device when de-energized is considered the *normal position.* Control devices (such as valves or dampers) are either *normally open* (N.O.) or *normally closed* (N.C.). Some electric devices also contain switches that are normally open or normally closed until moved to the opposite position by a controller.

4. Safety Controls

As stated earlier, control systems maintain design conditions within a space, but also provide for the safe and economical operation of the HVAC system. The devices discussed up to now are considered operating controls. The other type of control that the TAB technician must be aware of is the safety or limit control. Safety or limit controls are used to provide safe system operation and may interrupt the operating controls at any given time to insure safe system operation. Examples of safety controls are freezestats, firestats, flow switches, smoke detectors, and refrigeration high-low pressure cutouts.

B AUTOMATIC TEMPERATURE CONTROL SYSTEMS

1. Introduction

Controlled devices, such as valves or dampers, are also a form of balancing devices. These are usually modulated by a sensor control to maintain some stable condition. Using the hydronic system in Figure 6-1, as the load of each terminal unit varies, the control valve on each unit responds accordingly. If the system designer wishes to maintain a set temperature in the air discharge of a unit or maintain any other set condition, he can assign a sensing control to reset the control valve to do this.

In some systems, equipment is stopped and started when certain limits are reached. For example, a chiller is set to stop when the outdoor temperature is below 50°F. To form a complete cycle, the chiller must be turned back on. A higher temperature is usually selected for this setting to avoid rapid switching from on to off when the temperature is exactly 50°F. A temperature setting of 53°F might be selected for the chiller to restart. This will allow a "dead band" or "differential" of 3°F so that only a significant change in the outdoor temperature will activate the equipment or stop it.

The thermostatic controller (outdoor temperature) actually would be connected to a relay in the starter circuit of the chiller. The relay is electrically actuated, but the sensing element may be either pneumatic, electric, or electronic. If pneumatic controls are used, another device must be used to transform the pneumatic (air actuated) signal to an electrical signal which causes the relay to actuate the chiller. "PE" switches are the devices that convert pneumatic signals to electric or electronic control circuit signals.

Pneumatic control systems use pneumatic devices exclusively, but are usually hybrids of both pneumatic and electric devices. The term "pneumatic", "electronic", or "electric" is used to identify the *type* of control system depending upon the primary control devices that the majority of the system contains. Pneumatic systems are used primarily when the installation is large enough to support the cost of an air compressor and small enough that extremely long runs in pneumatic piping are not encountered. Electronic systems often are used when the sensing elements and the actuating elements are some distance from one another. In some instances, one timeclock in a central location in a city theoretically could control equipment in buildings throughout the city with separate day and night schedules of operation. In this instance, the electronic signals would be transmitted to each building by telephone lines.

2. Control Diagrams

In a typical job specification, there are general descriptions of various types of control applications which the automatic temperature control contractor would translate into a set of drawings called "control diagrams". These control diagrams, after approval by the system designer, are used by the automatic temperature control (ATC) contractor for control system installation in coordination with the HVAC system contractor. The data found in these diagrams is extremely important to the TAB contractor. These diagrams frequently are the only description of how the HVAC system will operate.

Control system diagrams also can be used to assist the TAB technician in troubleshooting. For example, when the "hand-off-automatic" switch of a fan motor starter is in the "automatic" position, it is found that the fan will not run. But it is found that the fan will run when the switch is in the "hand" position. This indicates that some type of automatic temperature control devices or safety switch is preventing the fan from running.

A review of the system control diagrams and the sequence of operation indicates that there are auxiliary contacts and a low limit safety control in the fan control circuit. The two devices are inspected to verify if one or both is responsible for the "open" in the control circuit. If the auxiliary contacts are found to be in the "closed" position, then the low limit (freeze-stat) control is the next item to be inspected. If the outdoor temperature is well above freezing, and if pushing the reset button allows the fan to start running, the indication is that the air temperature sensed by the control has been cold enough to activate it or that the control has malfunctioned. The operation of this particular safety device then should be checked frequently for another malfunction. If nuisance "tripouts" continue to occur, the ATC contractor should adjust or replace the control. There is also a possibility that another device, such as an outside air intake damper, is not set properly, allowing low temperature air to reach the sensing device when the outdoor air temperature is below freezing.

If the outside air intake damper is not operating properly, the TAB technician should ask the ATC contractor to adjust the necessary controls to cause the damper to change position. Then the action of the damper, linkages, and actuator can be observed. Sometimes the dampers or linkages may slip on the shafts. If slipping has occurred, the linkage may need adjusting to allow the damper to arrive at the correct minimum and maximum settings. Controller or sensor output pressure also may be incorrect in pneumatic systems. Many times more than one component of a control system may be found to be out of adjustment or to be faculty. In other words, an improperly adjusted or malfunctioning damper could be the cause of the "freeze-stat" being activated.

If problems with the outside air intake damper were corrected, or there was no problem found at the damper, the safety device could still be stopping the fan because of an improper sensor location. If the sensor bulb is located where the sun could cause a sensor bulb temperature of 35°F when 30°F outdoor air is flowing into the HVAC unit, the coils could freeze before the "real" problem is discovered. This is after the problem with the fresh air dampers "had been solved". In this situation, an installation or design error was made when the outdoor sensor bulb was located where the sun could affect its operation. The TAB technician needs to have enough knowledge about different types of control systems and to be able to read control diagrams so that an improperly used or an improperly located device can be found before a system is damaged, or time is wasted in an attempt to do the TAB work under fluctuating conditions.

3. Control Relationships

All control system sensors and controllers are generally *linear* (which means "straight line"). Figure 6-3 indicates the error induced by non-linear control devices such as dampers and valves. Linear control in pneumatic systems may translate to one degree of temperature change from one psi of air pressure change. One psi of fluid pressure change on the discharge side of a pump also could result in a two or three psi of control pressure change. These systems are linear as long as each increment of "controlled variable" produces the same increment of "signal". The system would be non-linear if different amounts of signals emanate from a fixed increment of the controlled variable. For example, a system is non-linear if at 70°F, a one degree change produces a one psi control signal; but at 90°F, a one degree change produces a two psi control signal.

Most actuators are linear, that is, if an actuator has a signal range of ten psi from fully open to fully closed, a five psi signal will cause a 50% travel. However, if

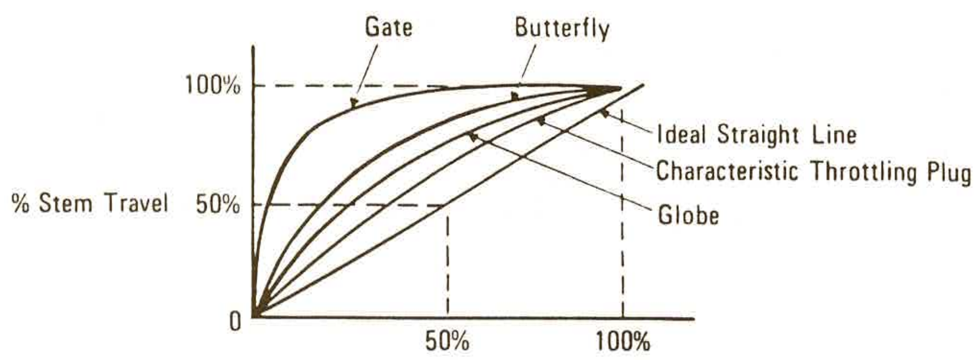


Figure 6-3 VALVE THROTTLING CHARACTERISTIC COMPARISON

the actuator device is used on a valve or damper, an actuator change of 50% will seldom change the fluid flow by the same 50%. From Figure 6-3 one can see that a 50% stem travel of a gate valve from wide open will have little effect on the fluid flow.

Equation 6-1

$$C_v = Q\frac{(2.3)^{1/2}}{(H)^{1/2}} = Q\sqrt{\frac{2.3}{H}}$$

Equation 6-2

$$\frac{H_1}{H_2} = \frac{Q_1^2}{Q_2^2} = \frac{\Delta P_1}{\Delta P_2}$$

Equation 6-3

$$\Delta P = \left(\frac{Q}{C_v}\right)^2$$

Where:

C_v = Flow coefficient or valve constant

Q = Flow rate of the fluid (gpm)

H = Head loss or pressure drop (feet of fluid)

ΔP = Pressure different (psi)

Equations 6-1 and 6-2 indicate that the pressure drop across the valve is proportional to the square of the fluid flow rate. This relationship is indicated by the general curve shown in Figure 6-3 which can apply to most systems, although the numbers may vary. The non-linearity of the controlling device is apparent with this curve. In order to minimize the resulting control inaccuracies, the controller and the controlled device must be carefully matched so that an average linearity is achieved. This cannot be done across the entire range of the device, therefore, the devices are matched for a "normal operating range", which is a matter of judgment of the system designer or the ATC Contractor.

Example 6A

A control valve with a C_v of 40 has a flow rate of 70 gpm. Find the pressure drop across the valve in ft. w.g. and psi.

Solution

(a) Using Equation 6-1 with $C_v = 40$ and $Q = 70$:

$$C_v = Q\frac{(2.3)^{1/2}}{(H)^{1/2}} \text{ or } H = \frac{2.3}{(C_v/Q)^2}$$

$$H = \frac{2.3}{(40/70)^2} = \frac{2.3}{0.327}$$

$$H = 7.04 \text{ ft. w.g.}$$

(b) Using Equation 6-3:

$$\Delta P = \left(\frac{Q}{C_v}\right)^2 = \left(\frac{70}{40}\right)^2$$

$$\Delta P = 3.06$$

Example 6B

If the heat loss curve was $C_v = 1$ in Figure 6-4, what would the pressure drop across the valve be at

(a) 1 gpm, (b) 2 gpm, and (c) 3 gpm?

Plot the results in Figure 6-4.

Solution

(a) $$C_v = Q\frac{(2.3)^{1/2}}{(H)^{1/2}} \text{ or } H = \frac{2.3}{(C_v/Q)^2}$$

$$H = \frac{2.3}{(1/1)^2} = \frac{2.3}{1} = 2.30 \text{ ft. w.g.}$$

(b) $$H = \frac{2.3}{(1/2)^2} = \frac{2.3}{0.25} = 9.20 \text{ ft. w.g.}$$

(c) $$H = \frac{2.3}{(1/3)^2} = \frac{2.3}{0.111} = 20.72 \text{ ft. w.g.}$$

Example 6C

Checks parts (a) and (c) of the answers of Example 6B using Equation 6-2.

Solution

$$\frac{H_1}{H_2} = \frac{Q_1^2}{Q_2^2};\ H_1 = \frac{H_2 Q_1^2}{Q_2^2}$$

$$H_1 = \frac{20.72 \times (1)^2}{(3)^2} = \frac{20.72}{9} = 2.30 \text{ ft. w.g.}$$

(Checks with "H" answer of part (a) of Example 6B).

4. Valves and Dampers

Control valves and dampers generally fall into two categories, "modulating" and "two-position" (open or closed). Modulating means that the device can assume infinite positions of control as required by the sensor. In Figure 6-5, either the two-way valve "A" or the three-way valve "B" can be two-position or modulating. If valve "A", for example, is only to stop and start the flow to the terminal unit, it would be a two-position valve. The designer might not want this terminal unit to be used during the cooling season due to load characteristics of the building. For heating only, controls would sense cooling water in the system and close valve "A". The control system would again open valve "A" when heating water was sensed in the system.

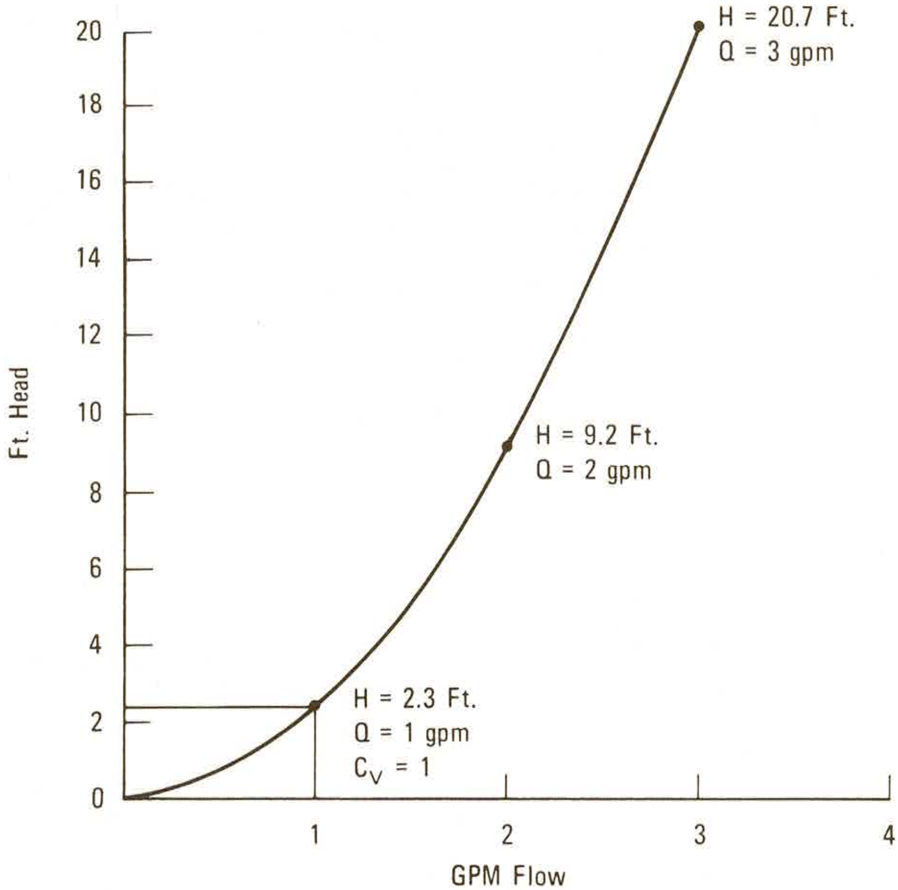


Figure 6-4 VALVE FLOW RATE-HEAD LOSS CURVE WHEN C_v = 1

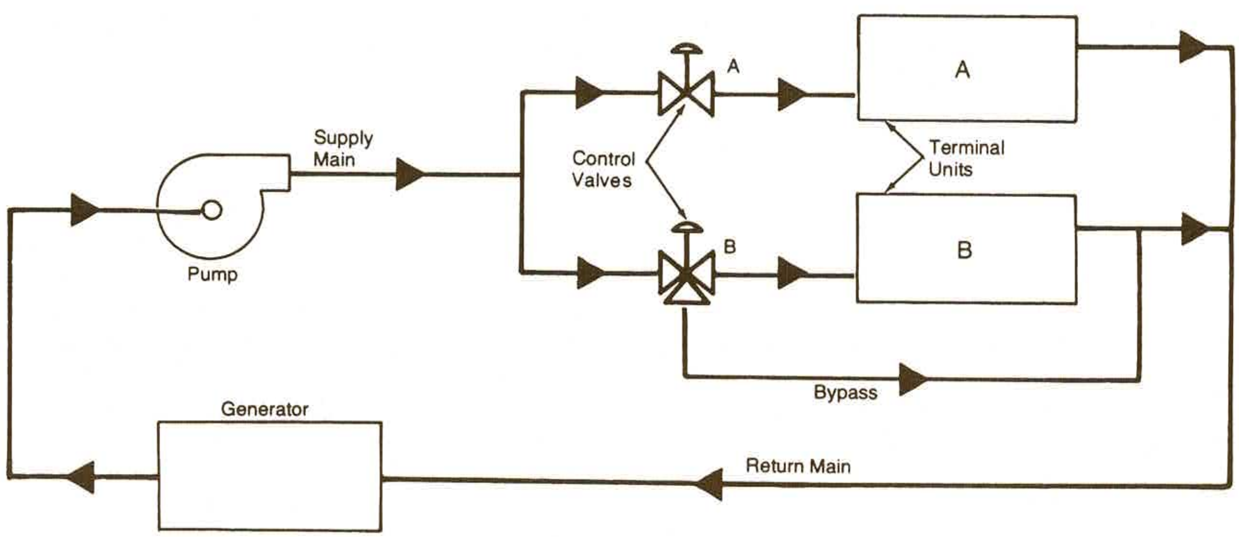


Figure 6-5 THE COMPONENTS OF A HYDRONIC SYSTEM

If valve "B" was a modulating valve to be used for both heating and cooling, it would open to allow just enough heating or cooling water to pass through the terminal unit to satisfy the space load. A changeover signal from a sensing device would reverse the action of valve "B" so that it can gradually open on a decrease in space temperature in the winter, and gradually open with an increase in space temperature in the summer. The same changeover system can be applied to an "economizer cycle" in both the spring and the fall. In Figure 6-5, the generator can actually be two devices or a single device that can be a cold generator (or chiller) for summer and a heat generator (or boiler) for winter.

A safety interlock prevents a chiller from being damaged if it is operated with very low fluid flow volumes of condensor or chilled water. This safety interlock usually is accomplished by using flow switches in the condensor water and chilled water circuits that are connected electrically to a relay in the chiller starter. This interlock delays operation of the chiller until sufficient flow in the condensor water and chilled water lines has been proven, that is, the pumps circulate an adequate volume of water to cause the flow switches to activate or close. This action then electrically closes the relay and the chiller starter, allowing the chiller to run. There are a number of such control interlock applications on each job.

Referring again to Figure 6-5, the valves may be controlling condensor water to "self-contained" classroom type units. Then both control valves would be two-position with a piping bypass installed to keep the pump from running against a high head or closed circuit. The terminal units would have built-in controls to operate their individual coils and/or compressors to maintain the space temperatures. When each unit has its own factory installed controls, the terminology is "unitary control". The "central control" system in this diagram would control valves "A" and "B", Central control systems generally serve more than one HVAC unit or system from a central location.

The examples used above did not indicate how complicated some control systems can be. Very complicated mechanical systems generally require very complicated control systems, while simple mechanical systems normally require only a simple control system. It is also apparent that the more complicated control systems are more difficult to troubleshoot. This also applies to the mechanical systems. The failure rate of controls and control circuits of very complicated systems increases geometrically with its size. Therefore, the experienced designer uses the most simple control system that will suffice for the application.

Although control systems and control applications can be very sophisticated, they often will not work as intended. The intent of the designer and the operation of the system often is not clearly spelled out in the specifications. If the controls and control systems do not function as designed, the mechanical systems also will not function properly. It is part of the responsibility of the TAB technician to point out these errors so that corrective action can be taken by the ATC contractor.

5. Control System Adjustment and Calibration

In most control systems, controllers can be adjusted in range, set points, sensitivity, and differential. This often is done with the ATC system control panels, which like motor starters, are the heart of the control systems. There is usually some calibration and control adjustment in sensing devices and actuators which may be located in other areas. The TAB technician is *not* authorized to make changes to the control system. It would be advantageous to everyone concerned with HVAC systems to have the ATC contractor on the job to make adjustments during the system balancing process. Initial position settings of valves and dampers can be given to the ATC contractor prior to the TAB work. Coordination and cooperation between the TAB contractor and the ATC contractor is the key to a smooth running, well-balanced HVAC system.

6. A Final Note

To properly balance any HVAC system, knowledge of it's automatic temperature control system will be required. However, it is recommended that TAB technicians become acquainted with all types of control systems that are installed on the type of jobs that they balance, even though they should request assistance from the ATC contractor when settings for automatic dampers and valves must be changed to perform the TAB work. The NEBB "Environmental Systems Technology" textbook has a comprehensive chapter on all types of ATC systems and components, and how they work.

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CHAPTER 7 PRELIMINARY PROCEDURES

A PROCUREMENT OF DATA

Since testing, adjusting, and balancing of HVAC systems can best be accomplished by following systematic procedures, the entire TAB process should be thoroughly organized and planned. Proper preparation and organization during the initial phases of a project may be the determining factor between success and failure. TAB preparatory work accomplished in the office may be broken down into three categories:

(1) Procurement of data

(2) System review and analysis including the agenda and preliminary paperwork

(3) Planning and scheduling TAB field procedures

1. Contract Drawings

The TAB contractor must obtain a complete set of up-to-date contract drawings, including all the latest revisions and addendas. If the job is a typical rental office building, the latest tenant drawings also will be needed. Shop drawings are included in the list as some are usually closer to "as-built" drawings than any others available, and they will indicate where dampers, valves, etc. are actually located. The temperature control system drawings will show how the HVAC systems operate.

Independent balancing contractors often must keep pursuing the above items to continually stay on top of things or they will end up with outdated material. Many general contractors and mechanical contractors do not realize how important these items are to the TAB contractor and many have a habit of neglecting to adequately provide up-to-date data.

2. Specifications

If the TAB contractor has trouble obtaining the drawings, he probably will have the same problem obtaining the specifications, and they are a *must*. Specifications usually spell out exactly what data and testing is required, and what guidelines or balancing procedures are desired. They also may include the sequence of operation for the control systems, which normally are found on the temperature control system drawings. All addendas and revisions must be included. Quite often the TAB work is an oversight in the original design and is added or intensified in revised specifications.

3. Submittal Data

Obtain all applicable approved equipment submittals. Pay close attention to the word "approved." Many projects have submittals rejected for not meeting specification requirements. Be aware of extraneous or erroneous data not applicable to the project. Particular attention should be paid to the following equipment submittals.

a. FANS

Included with listing of performance data and physical characteristics, fan curves should be provided. Fan performance data must relate to the actual job requirements and include such items as inlet vanes, and altitude and temperature corrections if applicable. Verify that all components (motor, drives, belts) comply with contract requirements. Pay close attention to the external and total static pressure ratings of the fans. Ratings given are at times confusing, use caution when reviewing submittals. Do not read into the submittal what is wished to be seen.

b. AIR TERMINALS

Included in manufacturer's data should be air pressure loss at design flow conditions, sound pressure data, air pattern adjustment, and recommended testing procedure. Note whether the terminal has the means for airflow adjustment, or if auxiliary dampers must be used.

c. AIR DISTRIBUTION DEVICES

In addition to listing of components and rated capacities, air distribution devices such as variable volume boxes, static pressure dampers, and constant volume regulators should have manufacturer's recommended test procedures. Automatic temperature control diagrams provided by the unit manufacturer should be checked with ATC diagrams provided by the temperature control contractor for compatibility.

d. PRIMARY HEAT EXCHANGE EQUIPMENT

Performance data for equipment such as boilers, chillers, cooling towers, and heat exchangers should be examined to ensure that unit capacity and pressure losses are within acceptable tolerances as set by contract documents. Be aware of flow rates listed that deviate from design requirements to comply with maximum friction loss requirements. Also be aware of the units (psi or in.w.g.) in which pressure is expressed.

e. TERMINAL HEAT EXCHANGE EQUIPMENT

Performance data for equipment such as HVAC unit coils, reheat coils, fan coil units and unit heaters should be reviewed to ensure temperature and pressure ratings comply with design requirements.

B REVIEW AND ANALYSIS OF SYSTEMS

After all preliminary data has been collected, a study of each HVAC system may be performed. Two basic reasons for the importance of system review and analysis are (1) to isolate any discrepancies in the data or drawings that may prevent the proper balancing and performance of a system, and (2) to develop an agenda to establish the best approach for testing and balancing. An agenda allows an opportunity for the design engineer, architect, and owner to review what procedure will be taking place during the actual testing and balancing of their systems. The agenda also offers an excellent vehicle to present any discrepancies found during the system review and analysis.

The preparatory work described in this chapter will give the agenda preparer a tremendous insight into the layout and operation of the systems and the job. Some contractors as a matter of routine use a TAB Supervisor or others in this capacity to prepare the paperwork and set up all TAB work. Many times this person does not always go on the job, but does direct competent TAB technicians to do the work. In this case, a lot of time spent being familiarized with the job is not totally used after the preparation is complete.

It is becoming more common to use TAB technicians, who will be doing the balancing and will be on the job site and directly responsible for the execution of the work, to do the preliminary office work, including the agenda and paperwork. This gives them a tremendous advantage when they walk onto the job. They will already be familiar with the building layout and the design and operation of the system. This decision varies with each TAB contractor and will be influenced by the abilities of the personnel available.

1. System Components and Types

Review the plans, specifications and equipment data. Examine each system noting such things as the types and locations of the areas served, types of system, types of components used such as fans, pumps, boilers, chillers, coils, VAV boxes, etc. Note such things as primary and secondary systems, interconnected or interlocked systems, possible tie-ins to existing systems and the location of motor control centers, breakers, etc. Review the equipment schedules, the temperature control drawings and particularly review in detail the operating sequences of each system. You must know how a system is designed to operate before you will be able to successfully balance it.

When reviewing plans always read all the notes. System designers have a way of slipping little notes in that can easily be missed if a thorough review isn't done. Look for possible additional fans or other equipment that may not be listed in the schedules. Now is the time to "familiarize" yourself with the project, and particularly the HVAC systems that you will be working with while performing the TAB work.

2. Schematic Diagrams

It is recommended that a schematic layout of each duct and piping system be prepared (see Figure 7-1). All components that pertain to balancing of the system should be indicated (e.g. dampers, air terminals, terminal units, regulating devices, etc.). Show velocities and flow rates (cfm or gpm) for main and branch circuits. Identify air and water terminals with an appropriate numbering system that may be easily identified with by both field personnel and the person who will ultimately review the report.

3. Test Report Form Preparation

Now that you have reviewed the systems, equipment, and prepared the schematic layouts, you should proceed to prepare the test report forms. Although many TAB technicians prefer to start with the equipment forms and work into the system, you will find that you

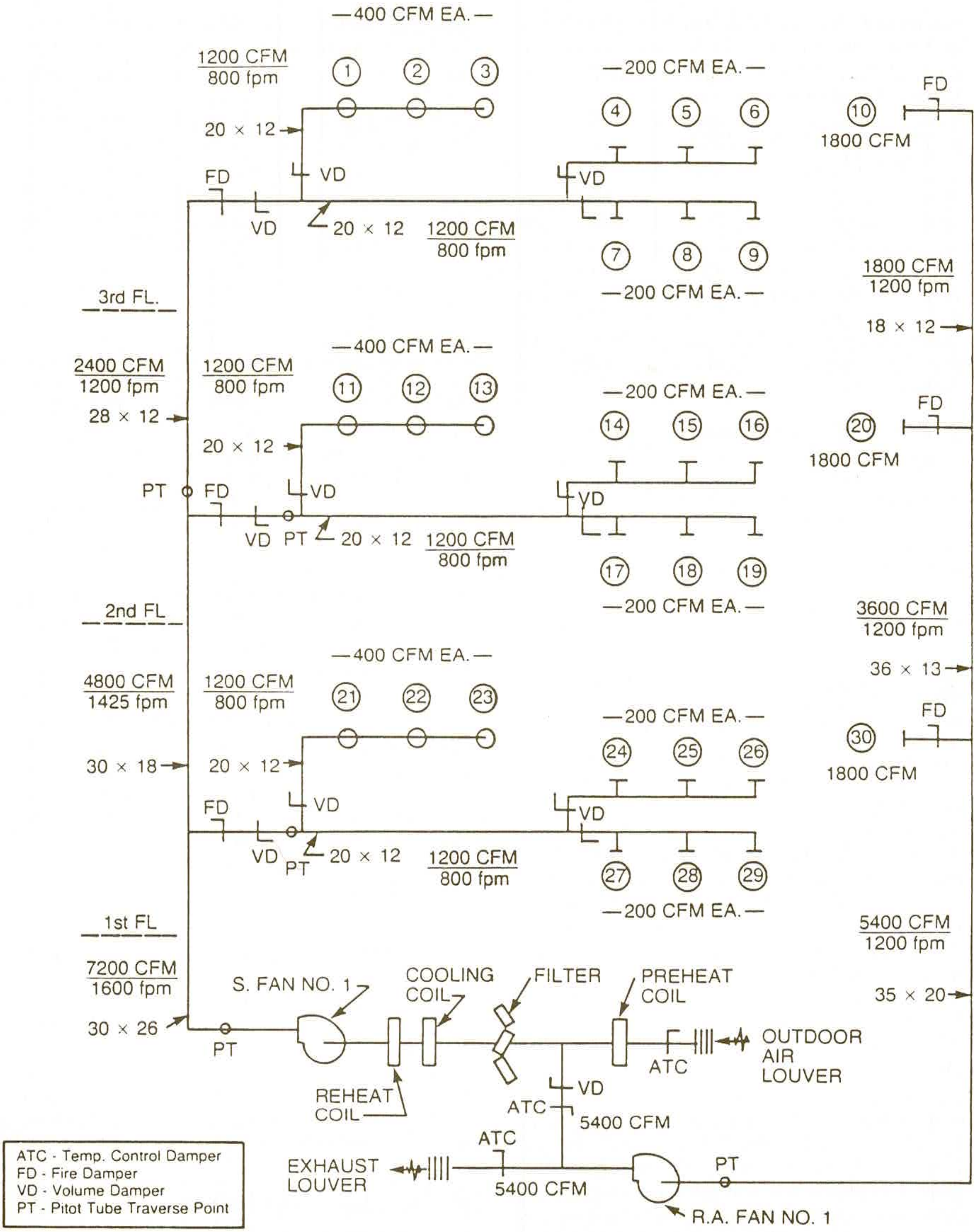

Figure 7-1 SCHEMATIC DUCT SYSTEM LAYOUT

can't complete many of them until the system terminals have been tabulated. Many times it is preferable, and particularly on the air side, to start with the "Air Outlet Test Report."

The preparation of these forms provides a valuable educational process about the job. While going through each system and numbering the terminals, you will become familiar with where dampers are located, where dampers may be needed, how to divide the system into zones for performing Pitot tube traverses, and uncovering discrepancies that should be brought to the attention of the proper people. A brief description of each NEBB Test Report Form follows, described in an order that you might want to use in setting up a typical job. The actual order, however, will depend on the equipment used in the HVAC systems.

a. CERTIFIED TEST, ADJUST AND BALANCE REPORT (NEBB TAB 1-83)

The Certified Test, Adjust and Balance Report is used as a cover sheet for all final report forms which are submitted. The TAB Supervisor who has managerial responsibility for the TAB work must complete this form with the required information.

b. CERTIFICATION (NEBB TAB 2-83)

The NEBB TAB Contractor who performs and/or supervises the TAB work is to complete this form. If there is more than one certified contractor performing the work, i.e. an air performance contractor and a hydronic performance contractor, the NEBB TAB Contractor, who has the responsibility for the project, shall have his Supervisor apply his Certification Seal.

c. SYSTEM DIAGRAM (NEBB TAB 3-83)

This form is to be used primarily for a schematic layout of air distribution systems, but it may be used for hydronic systems as well. A single line system diagram is highly recommended to insure systematic and efficient procedures.

A sample single line schematic (Figure 7-1) depicts a typical system diagram. Be sure to show quantities of outside air, return air and relief air, sizes and cfm for main ducts, sizes and cfm of outlets and inlets, and all dampers, regulating devices and terminal units. All outlets should be numbered before filling out the Outlet Test Report. While diagrams are suggested, the use of this form is not mandatory. If appropriate, a large (similar) diagram sheet may be used in the report. These diagrams are extremely helpful in preparing balancing procedures and in locating discrepancies.

The following forms will be necessary when preparing the final TAB Certified Report for submission.

d. AIR APPARATUS TEST REPORT (NEBB TAB 4-83)

Obtain the data for this form from the schedules on the plans, and verify each applicable item with the submittal data. Any discrepancies should be shown on this form in the blank space provided. Discrepancies should be reported in writing (to the proper people) as well as detailed in the agenda. Diversity and total system airflow should be shown when they are different from the fan design airflow (cfm). External static pressure should be entered when applicable.

e. APPARATUS COIL TEST REPORT (NEBB TAB 5-83)

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. Confirm that totaled gpm for the coils matches the pump design gpm.

f. GAS/OIL FIRED HEAT APPARATUS TEST REPORT (NEBB TAB 6-83)

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 start-up equipment report, but could be used as a supplement. All available design data should be reported. The "HP/RPM, F.L. AMPS/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. Procedures used for the Air Apparatus Test Report form apply here also.

g. ELECTRIC COIL/DUCT HEATER TEST REPORT (NEBB TAB 7-83)

This form is to be used for electric furnaces or for electric coils installed in built-up units or in branch ducts. "Min.AirVel." is the manufacturers recommended minimum airflow velocity.

h. FAN TEST REPORT (NEBB TAB 8-83)

This form is to be used with supply air, return air, or exhaust air fans. Since housings for various types of fans may have many different shapes and arrangements, not all entry blanks will be needed for testing a particular fan. The performance of up to three fans may be reported on this sheet. Procedures are similar to those for the Air Apparatus Test Report form.

i. RECTANGULAR DUCT TRAVERSE REPORT (NEBB TAB 9-83)

This form is to be used as a worksheet for recording the results of a Pitot tube traverse in a rectangular duct. It is recommended that the velocity pressure be recorded in one-half of each of the spaces provided and converted to velocities in the other half of each space at a later time. **The airflow velocities shall be averaged (not the velocity pressures).**

While preparing the outlet sheets, you can total the cfm requirements for the individual Pitot tube traverses that will be required. You will need to obtain a system total cfm as well as supplementing Pitot tube traverses. The total airflow of the fan should be a single Pitot tube traverse if possible. Otherwise, try to keep the total Pitot tube traverses required to a minimum for the system total airflow (cfm). The supplementary Pitot tube traverses should be taken on each floor on multi-story buildings and anywhere that a major zone breaks off from a main line duct. Locate the Pitot tube traverse in the best section of straight duct available. Don't locate them downstream and close to any elbows, dampers, transitions or offsets. Proposed locations made in the office will have to be field verified, as quite often duct changes and/or Pitot tube obstructions will prevent using the original choice of locations.

j. ROUND DUCT TRAVERSE REPORT (NEBB TAB 10-83)

Record the results of a Pitot tube traverse in a round duct on this worksheet type form. Spaces shown are for velocity pressures and velocities taken at points across two diameters of the duct. Procedures and locations for this form are similar to the Rectangular Duct Traverse Report form. Only the shape of the duct is different.

k. AIR OUTLET TEST REPORT (NEBB TAB 11-83)

As this form can be used as both a worksheet and a final report form, TAB technicians are encouraged to record all readings on this test report form. However, it is not necessary to record preliminary velocity readings on the final sheet unless requested by the system designer. If more than two sets of preliminary readings are necessary or required, the data can be entered in the two blank columns between "Preliminary" and "Final." The outlet number refers to the number assigned on the schematic layout. The column entitled "Type" is to be used for recording the type or model number of the air outlet.

If the final adjusted cfm of any air outlet varies by more than ± 10% from the design cfm, 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.

Completing these sheets for any one system will enable the TAB technician to total the terminals and determine the system cfm requirements. Quite often, the total design cfm of the terminals will not be the same as the fan design cfm as listed on the equipment schedule and the submittal data. If the discrepancy is small, you can usually just note it and proceed to balance it as is. If there is a sizable difference, the proper people should be notified in writing, as well as including this data in the Agenda.

One exception to this, of course, is if you have a diversity type VAV system where the fan is purposely sized for less cfm than the totaled outlets.

l. TERMINAL UNIT COIL CHECK REPORT (NEBB TAB 12-83)

This form is used as a worksheet to check the water coil of terminal units. Any of the three alternate methods for determining water flow or heat transfer indicated on the test report form is acceptable. Locate and number these coils so that each circuit and zone total gpm requirement can be determined and compared to the pump design gpm.

m. PACKAGED CHILLER TEST REPORT (NEBB TAB 13-83)

This form may be used as a check sheet to record the control settings and the entering and leaving conditions at the chiller. Since the TAB contractor is not necessarily responsible for start-up 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 system designer from the data contained therein.

This form or the manufacturer's form should be substantially completed and verified by the manufacturer's representatives and/or the installing contractor before the HVAC distribution systems are balanced. Temperature and pressure (drop) readings of the chiller unit evaporator and condenser should be entered during the TAB procedures. Verify that the design gpm match up with the pumps, the total system flow requirements, and the cooling tower flow requirements (condenser water). Report any discrepancies and include in the agenda.

n. PACKAGE ROOFTOP/HEAT PUMP/AIR CONDITIONING UNIT TEST REPORT (NEBB TAB 14-83)

Test data from package units of all types is to be recorded on this form, with most of the data being

NEBB

CERTIFIED TEST, ADJUST, AND BALANCE REPORT

DATE ____________

PROJECT ____________
ADDRESS ____________
____

ARCHITECT ____________
____

ENGINEER ____________
____

HVAC CONTRACTOR ____________
____

NEBB TAB CONTRACTOR ____________
ADDRESS ____________
____

TAB 1-83

National Environmental Balancing Bureau

NEBB

CERTIFICATION

PROJECT ____________
ADDRESS ____________
____

THE DATA PRESENTED IN THIS REPORT IS AN EXACT RECORD OF SYSTEM PERFORMANCE AND WAS OBTAINED IN ACCORDANCE WITH NEBB STANDARD PROCEDURES. ANY VARIANCES FROM DESIGN QUANTITIES WHICH EXCEED NEBB TOLERANCES ARE NOTED THROUGHOUT THIS REPORT.

THE AIR DISTRIBUTION SYSTEMS HAVE BEEN TESTED & BALANCED AND FINAL ADJUSTMENTS HAVE BEEN MADE IN ACCORDANCE WITH NEBB "PROCEDURAL STANDARDS FOR TESTING—ADJUSTING-BALANCING OF ENVIRONMENTAL SYSTEMS" AND THE PROJECT SPECIFICATIONS.

NEBB CONTRACTOR ____________

REG. NO. ________ CERTIFIED BY ____________ (Air TAB Supervisor) DATE ________

THE HYDRONIC DISTRIBUTION SYSTEMS HAVE BEEN TESTED & BALANCED AND FINAL ADJUSTMENTS HAVE EN MADE IN ACCORDANCE WITH NEBB "PROCEDURAL STANDARDS FOR TESTING—ADJUSTING-BALANCING OF ENVIRONMENTAL SYSTEMS" AND THE PROJECT SPECIFICATIONS.

NEBB CONTRACTOR ____________

REG. NO. ________ CERTIFIED BY ____________ (Hydronic TAB Supervisor) DATE ________

SUBMITTED & CERTIFIED BY:

NEBB CONTRACTOR ____________
TAB SUPERVISOR ____________
REG. NO. ____________
ATE ____________

TAB 2-83

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SYSTEM DIAGRAM

PROJECT ____________ SYSTEM ____________
LOCATION ____________ DATE ____________

TAB 3-83

National Environmental Balancing Bureau
PAGE ____ OF ____

NEBB

AIR APPARATUS TEST REPORT

PROJECT ____________ SYSTEM/UNIT ____________
LOCATION ____________

UNIT DATA	
Make/Model No.	
Type/Size	
Serial Number	
Arr./Class	
Discharge	
Make Sheave	
Sheave Diam/Bore	
No. Belts/make/size	
No. Filters/type/size	

MOTOR DATA	
Make/Frame	
H.P./RPM	
Volts/Phase/Hertz	
F.L. Amps/S.F.	
Make Sheave	
Sheave Diam/Bore	
Sheave ℄ Distance	

TEST DATA	DESIGN	ACTUAL
Total CFM		
Total S.P.		
Fan RPM		
Motor Volts T_1T_2, T_2T_3, T_3T_1		
Motor Amps T_1 T_2 T_3		
Outside Air CFM		
Return Air CFM		

TEST DATA	DESIGN	ACTUAL
Discharge S.P.		
Suction S.P.		
Reheat Coil Δ S.P.		
Cooling Coil Δ S.P.		
Preheat Coil Δ S.P.		
Filters Δ S.P.		
Vortex Damp. Position		
Out. Air Damp. Position		
Ret. Air Damp. Position		

REMARKS:

TEST DATE ____________ READINGS BY ____________

TAB 4-83

National Environmental Balancing Bureau
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NEBB

APPARATUS COIL TEST REPORT

PROJECT ______

COIL DATA	COIL NO.		COIL NO.		COIL NO.		COIL NO.	
System Number								
Location								
Coil Type								
No. Rows-Fins/In.								
Manufacturer								
Model Number								
Face Area, Sq. Ft.								
TEST DATA	DESIGN	ACTUAL	DESIGN	ACTUAL	DESIGN	ACTUAL	DESIGN	ACTUAL
Air Qty., CFM								
Air Vel., FPM								
Press. Drop, In.								
Out. Air DB/WB								
Ret. Air DB/WB								
Ent. Air DB/WB								
Lvg. Air DB/WB								
Air Δ T								
Water Flow, GPM								
Press. Drop, PSI								
Ent. Water Temp.								
Lvg. Water Temp.								
Water Δ T								
Exp. Valve/Refrig.								
Refrig. Suction Press.								
Refrig. Suction Temp.								
Inlet Steam Press.								

REMARKS:

TEST DATE ______ READINGS BY ______

TAB 5-83

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PAGE ______ OF ______

NEBB

GAS/OIL FIRED HEAT APPARATUS TEST REPORT

PROJECT ______

UNIT DATA	UNIT NO.		UNIT NO.		UNIT NO.	
System						
Location						
Make/Model						
Type/Size						
Serial Number						
Type Fuel/Input						
Output						
Ignition Type						
Burner Control						
Volts/Phase/Hertz						
H.P./RPM						
F.L. Amps/S.F.						
Drive Data						
TEST DATA	DESIGN	ACTUAL	DESIGN	ACTUAL	DESIGN	ACTUAL
CFM						
Ent./Lvg. Air Temp.						
Air Temp. Δ T						
Ent./Lvg. Air Press.						
Air Press. Δ P						
Low Fire Input						
High Fire Input						
Manifold Press./CFH						
High Limit Setting						
Operating Set Point						

REMARKS:

TEST DATE ______ READINGS BY ______

TAB 6-83

National Environmental Balancing Bureau
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PAGE ______ OF ______

NEBB

ELECTRIC COIL/DUCT HEATER TEST REPORT

PROJECT ______ MANUF/MODEL ______

DESCRIPTION		RATED DATA							TEST DATA										
														VOLTAGE			AMPERAGE		
COIL NO.	SYSTEM / LOCATION	KW	STAGES	VOLTS PHASE	AMPS	CFM	FACE AREA, FT.2	MIN. AIR VEL. FPM	KW	VEL. FPM	COIL CFM	EAT	LAT	T1-T2	T2-T3	T3-T1	T1	T2	T3

REMARKS:

TEST DATE ______ READINGS BY ______

TAB 7-83

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NEBB

FAN TEST REPORT

PROJECT ______

FAN DATA	FAN NO.		FAN NO.		FAN NO.	
Location						
Service						
Manufacturer						
Model Number						
Serial Number						
Type/Class						
Motor Make/Style						
Motor H.P./RPM/Frame						
Volts/Phase/Hertz						
F.L. Amps/S.F.						
Motor Sheave Make/Model						
Motor Sheave Diam./Bore						
Fan Sheave Make						
Fan Sheave Diam./Bore						
No. Belts/Make/Size						
Sheave ℄ Distance						
TEST DATA	DESIGN	ACTUAL	DESIGN	ACTUAL	DESIGN	ACTUAL
CFM						
Fan RPM						
S.P. In/Out						
Total S.P.						
Voltage T_1T_2 T_2T_3 T_3T_1						
Amperage T_1 T_2 T_3						

REMARKS:

TEST DATE ______ READINGS BY ______

TAB 8-83

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NEBB

RECTANGULAR DUCT TRAVERSE REPORT

PROJECT ______ SYSTEM/UNIT ______
LOCATION/ZONE ______ SERVICE ______
ALTITUDE ______ DENSITY ______ CORR. FACTOR ______

DUCT	REQUIRED	ACTUAL
S.P. ______ AIR TEMP ______ °F SIZE ______ SQ. FT. ______	SCFM ______ FPM ______ CFM ______	SCFM ______ FPM ______ CFM ______

DISTANCE FROM BOTTOM	POSITION	1	2	3	4	5	6	7	8	9	10	11	12	13
	1													
	2													
	3													
	4													
	5													
	6													
	7													
	8													
	9													
	10													
	11													
	12													
	13													
DISTANCE FROM DUCT EDGE														
VELOCITY SUB-TOTALS														

NOTE: Take readings with air blowing toward the observer.

REMARKS:

TEST DATE ______ READINGS BY ______

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NEBB

ROUND DUCT TRAVERSE REPORT

PROJECT ______ SYSTEM/UNIT ______
LOCATION/ZONE ______ SERVICE ______
ALTITUDE ______ DENSITY ______ CORR. FACTOR ______

DUCT	REQUIRED	ACTUAL
S.P. ______ AIR TEMP ______ °F SIZE ______ SQ. FT. ______	SCFM ______ FPM ______ CFM ______	SCFM ______ FPM ______ CFM ______

(SEE REVERSE SIDE FOR INSTRUCTIONS)

Vert. Subtotal ______
Horiz. Subtotal ______
Total ______
No. of Points ______
Average ______

REMARKS:

TEST DATE ______ READINGS BY ______

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PAGE ______ OF ______

NEBB

AIR OUTLET TEST REPORT

PROJECT ______ SYSTEM ______
OUTLET MANUFACTURER ______ TEST APPARATUS ______

AREA SERVED	OUTLET				DESIGN		PRELIMINARY				FINAL		REMARKS
	NO.	TYPE	SIZE	AK	CFM	VEL	VEL. OR CFM	VEL. OR CFM			VEL	CFM	

REMARKS:

TEST DATE ______ READINGS BY ______

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TERMINAL UNIT COIL CHECK REPORT

PROJECT ______ SYSTEM ______ MANUFACTURER ______

ROOM NO.	RISER NO.	UNIT SIZE	DESIGN GPM	ALTERNATE NO. 1			ALTERNATE NO. 2			ALTERNATE NO. 3			ΔP	ΔT
				DESIGN ΔP	ENT. WTR. PR.	LVG. WTR. PR.	DESIGN ΔT	EWT	LWT	DESIGN ΔT	EAT	LAT		

NOTE: USE ONE OF THE ABOVE ALTERNATE METHODS

REMARKS: WATER SUPPLY TEMP. ______ OUT. AIR TEMP. ______

TEST DATE ______ READINGS BY ______

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PACKAGED CHILLER TEST REPORT

PROJECT ____________________ UNIT ____________________

LOCATION ____________________

MANUF. __________ MODEL __________ SERIAL NO. __________

CAPACITY __________ REFRIG. __________ STARTER __________ HEATER SIZE __________

EVAPORATOR	DESIGN	ACTUAL
Evaporator Press./Temp.		
Ent./Lvg. Water Press.	xxxxx	
Water Press. Δ P		
Ent./Lvg. Water Temp.		
Water Temp. Δ T		
GPM		

CONDENSER	DESIGN	ACTUAL
Condenser Press./Temp.		
Ent./Lvg. Water Press.	xxxxx	
Water Press. Δ P		
Ent./Lvg. Water Temp.		
Water Temp. Δ T		
GPM		

COMPRESSOR	DESIGN	ACTUAL
Make/Model		
Serial Number		
Suction Press./Temp.		
Dischg. Press./Temp.		
Oil Press./Temp.		
Voltage T_1T_2, T_2T_3, T_3T_1		
Amps T_1 T_2 T_3		
KW Input		
Crankcase Htr. Amps		
Ch.W. Control Setting		
Cond. W. Control Setting		
L.P. Cutout Setting		
H.P. Cutout Setting		

REFRIGERATION	DESIGN	ACTUAL
Oil Level Checked	xxxxx	
Oil Failure Sw. Diff.		
Refrig. Level Checked	xxxxx	
Relief Valve Setting		
Unloader Set Points		
% Cylinders Unloaded		
Purge Operation Checked		
Bearing Temperature		
Vane Position		
Demand Limit		
Low Temp. Cutout Setting		

REMARKS:

TEST DATE __________ READINGS BY __________

TAB 13-83 (Old Form TAB 16-79)

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PACKAGE ROOFTOP/HEAT PUMP/AIR CONDITIONING UNIT TEST REPORT

PROJECT ____________________ SYSTEM/UNIT ____________________

LOCATION ____________________

UNIT DATA	
Make/Model Number	
Type/Size	
Serial Number	
Type Filters/Size	
Fan Sheave make	
Fan Sheave Diam./Bore	
No. Belts/make/size	
Type Heating Section*	

MOTOR DATA	
Make/Frame	
H.P./RPM	
Volts/Phase/Hertz	
F.L. Amps/S.F.	
Make Sheave	
Sheave Diam./Bore	
Sheave ℄ Distance	

TEST DATA EVAPORATOR	DESIGN	ACTUAL
Total CFM		
Total S.P.		
Discharge S.P.		
Suction S.P.		
Out. Air CFM		
Out. Air DB/WB		
Ret. Air CFM		
Ret. Air DB/WB		
Ent. Air DB/WB		
Lvg. Air DB/WB		
Fan RPM		
Voltage T_1T_2, T_2T_3, T_3T_1		
Amperage T_1 T_2 T_3		

TEST DATA CONDENSER	DESIGN	ACTUAL
Refrigerant/Lbs		
Compr. Mfr./Number		
Compr. Model/Ser. Number		
Low Amb. Control		
Suction Press./Temp.		
Cond. Press./Temp.		
Crankcase Htr. Amps.		
Compr. Volts T_1T_2, T_2T_3, T_3T_1		
Compr. Amps T_1 T_2 T_3		
L.P./H.P. Cutout Setting		
No. of fans/fan RPM		
Cond. Fan HP/CFM		
Cond. fan Volts/Amps/ϕ		

*Use TAB 6-83 or TAB 7-83 for heating section test report

REMARKS:

TEST DATE __________ READINGS BY __________

TAB 14-83

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COMPRESSOR AND/OR CONDENSER TEST REPORT

PROJECT ____________________

UNIT DATA	UNIT NO.		UNIT NO.		UNIT NO.	
LOCATION						
Unit Manufacturer						
Unit Model/Ser. Number						
Compressor Manufacturer						
Compr. Model/Ser. Number						
Refrigerant/Lbs.						
Low Amb. Control						
TEST DATA	DESIGN	ACTUAL	DESIGN	ACTUAL	DESIGN	ACTUAL
Suction Press./Temp.						
Cond. Press./Temp.						
Oil Press./Temp.						
Voltage T_1T_2, T_2T_3, T_3T_1						
Amps T_1 T_2 T_3						
KW Input						
Crankcase Htr. Amps						
No. of Fans/Fan RPM/CFM						
Fan Motor Make/Frame/H.P.						
Fan Motor Volts/Amps						
Duct Inlet/Outlet S.P.						
Ent./Lvg. Air D.B.						
Cond. Wtr. Temp. In/Out						
Cond. Wtr. Press. In/Out						
Control Setting						
Unloader Set Points						
L.P./H.P. Cutout Setting						

REMARKS:

TEST DATE __________ READINGS BY __________

TAB 15-83

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COOLING TOWER OR EVAPORATIVE CONDENSER TEST REPORT

PROJECT ____________________ SYSTEM ____________________

LOCATION ____________________

MANUF. __________ MODEL __________ SERIAL NO. __________

NOM. CAPACITY __________ REFRIG. __________ WATER TREAT. __________

FAN DATA	
No. of Fan Motors	
Motor Make/Frame	
Motor H.P./RPM	
Volts/Phase/Hertz	
Motor Sheave Diam./Bore	
Fan Sheave Diam./Bore	
Sheave ℄ Distance	
No. Belts/Make/Size	

PUMP DATA	
Make/Model	
Pump Serial No.	
Motor Make/Frame	
Motor H.P./RPM	
Volts/Phase/Hertz	
GPM	

AIR DATA	DESIGN	ACTUAL
Duct CFM		
Duct Inlet S.P.		
Duct Outlet S.P.		
Avg. Ent. W.B.		
Avg. Lvg. W.B.		
Ambient W.B.		
Fan RPM		
Voltage T_1T_2, T_2T_3, T_3T_1		
Amperage T_1 T_2 T_3		

WATER DATA	DESIGN	ACTUAL
Ent./Lvg. Water Press.		
Water Press. Δ P		
Ent./Lvg. Water Temp.		
Water Temp. Δ T		
GPM		
Bleed GPM		
Voltage T_1T_2, T_2T_3, T_3T_1		
Amperage T_1 T_2 T_3		

REMARKS:

TEST DATE __________ READINGS BY __________

TAB 16-83 (Old Form TAB 23-76)

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NEBB

HEAT EXCHANGER/CONVERTER TEST REPORT

PROJECT ______________________

UNIT DATA	UNIT NO.		UNIT NO.		UNIT NO.	
Location						
Service						
Rating, BTU/Hr.						
Manufacturer						
Model Number						
Serial Number						
TEST DATA	DESIGN	ACTUAL	DESIGN	ACTUAL	DESIGN	ACTUAL
Steam: Pressure, PSI						
Steam: Flow, Lbs./Hr.						
Primary Water: Ent./Lvg. Temp.						
Primary Water: Temp. Δ T						
Primary Water: Ent./Lvg. Press.						
Primary Water: Press. Δ P						
Primary Water: GPM						
Secondary Water: Ent./Lvg. Temp.						
Secondary Water: Temp. Δ T						
Secondary Water: Ent./Lvg. Press.						
Secondary Water: Press. Δ P						
Secondary Water: GPM						
Control Set Point						
Exchanger Circuiting						

REMARKS:

TEST DATE ______________ READINGS BY ______________________

TAB 17-83 (Old Form TAB 24-78)

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NEBB

PUMP TEST REPORT

PROJECT ______________________

DATA	PUMP NO.	PUMP NO.	PUMP NO.	PUMP NO.	PUMP NO.
DESIGN: Location					
DESIGN: Service					
DESIGN: Manufacturer					
DESIGN: Model Number					
DESIGN: Serial Number					
DESIGN: GPM/Head					
DESIGN: Req. NPSH					
DESIGN: Pump RPM					
DESIGN: Impeller Diam.					
DESIGN: Motor Mfr./Frame					
DESIGN: Motor HP/RPM					
DESIGN: Volts/Phase/Hertz					
DESIGN: F.L. Amps/S.F.					
DESIGN: Seal Type					
ACTUAL: Pump Off Press.					
ACTUAL: Valve Shut Diff.					
ACTUAL: Act. Impeller Diam.					
ACTUAL: Valve Open Diff.					
ACTUAL: Valve Open GPM					
ACTUAL: Final Dischg. Press.					
ACTUAL: Final Suction Press.					
ACTUAL: Final Δ P					
ACTUAL: Final GPM					
ACTUAL: Voltage T_1T_2, T_2T_3, T_3T_1					
ACTUAL: Amperage T_1 T_2 T_3					

REMARKS:

TEST DATE ______________ READINGS BY ______________________

TAB 18-83 (Old Form TAB 25-78)

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NEBB

BOILER TEST REPORT

PROJECT ______________________

UNIT DATA	UNIT NO.		UNIT NO.		UNIT NO.	
Location						
Manufacturer						
Model Number						
Serial Number						
Type/Size						
Fuel/Input						
No. of Passes						
Ignition Type						
Burner Control						
Volts/Phase/Hertz						
TEST DATA	DESIGN	ACTUAL	DESIGN	ACTUAL	DESIGN	ACTUAL
Operating Press./Temp.						
Ent./Lvg. Temp.						
No. Safety Valves/Size						
Safety Valve Setting						
High Limit Setting						
Operating Contr. Setting						
High Fire Set Point						
Low Fire Set. Point						
Voltage T_1T_2, T_2T_3, T_3T_1						
Amperage T_1 T_2 T_3						
Draft Fan Volts/Amps						
Manifold Press.						
Output—MBH						
Safety Controls Check						

REMARKS:

TEST DATE ______________ READINGS BY ______________________

TAB 19-83 (Old Form TAB 26-78)

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NEBB

INSTRUMENT CALIBRATION REPORT

PROJECT ______________________

INSTRUMENT/SERIAL NO.	APPLICATION	DATES OF USE	CALIBRATION TEST DATE

REMARKS:

TEST DATE ______________ READINGS BY ______________________

TAB 20-83 (Old Form TAB 27-78)

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furnished and verified by the installing contractor of the HVAC system. If the unit has components other than the evaporator fan, DX coil, compressor and condenser fan(s), use the appropriate test report form. Procedures used for the Air Apparatus Test Reports apply here also.

o. COMPRESSOR AND/OR CONDENSER TEST REPORT (NEBB TAB 15-83)

The same comments apply to this form as for the "Packaged Chiller Test Report." 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.

p. COOLING TOWER OR CONDENSER TEST REPORT (NEBB TAB 16-83)

This form should be substantially completed and verified by the installing HVAC system contractor before the system is balanced. The "pump data" section is to be used for the recirculating pump in evaporative condensers, not the condenser water system pump used with cooling towers.

q. HEAT EXCHANGER/CONVERTER TEST REPORT (NEBB TAB 17-83)

This form is designed to record final conditions for up to three steam or hot water heat exchangers. Verify that the gpm etc., match up with the requirements of the system and the pump capacities. If not, notify the proper people in writing and note same in agenda.

r. PUMP TEST REPORT (NEBB TAB 18-83)

This report form may be used as a worksheet. However, the final data on each pump performance must be recorded on this form. The actual impeller diameter entry is that indicated by plotting the head curve or by actual field measurement where possible. Net positive suction Head (NPSH) is important for pumps in open circuits and for pumps handling fluid at elevated temperatures. NPSH defines the required pressure excess above the fluid flash point at the impeller eye. The total gpm should be compared to the total system flow requirement. Any discrepancies should be reported in writing and noted in the agenda.

s. BOILER TEST REPORT (NEBB TAB 19-83)

This form may be used by the installing HVAC system contractor to substantially verify the data on this test report, particularly when factory start-up services are involved. A flue gas analysis is beyond the scope of TAB procedures, but data could be added in the "remarks" section if available and required by the system designer. Verify that the boiler specifications are compatible with the associated systems.

t. INSTRUMENT CALIBRATION REPORT (NEBB TAB 20-83)

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.

4. Processing the Report Forms

With the schematic diagrams and the preliminary paperwork now complete, the report forms may be processed. Prepare the paperwork into logical sections that may be easily understood by the TAB field personnel and that may be processed easily for the final reports. When all test report forms have the necessary preliminary data recorded, review them again for possible missing data.

Locate where Pitot tube traverses of duct mains and branches are to be made. Determine the number of readings to be taken, calculate the required velocities, and set up the duct traverse test reports. Record any notes found during the system review that may assist TAB personnel in the testing and balancing of the systems.

Do not wait until you get on the job to perform all paperwork entries and calculations that may be accomplished in the office. It is considerably easier and more efficient to calculate required terminal velocities etc. in the office where working conditions are likely to be better.

Assemble the test report forms in a manner that will make them easier to use in the field. Remember, at the end of a project, the installing HVAC system contractors have physical proof of their work in the duct and piping of a project, but TAB contractors only have a stack of paper to show for their efforts! Proper preparation and care of paperwork is extremely important.

With all the preliminary data and schematic drawings in hand, it is time to study the systems for possible discrepancies and to establish the balancing procedures. Actual methods for balancing air and water systems are extensively reviewed in Chapters VIII and IX, and should be used in conjunction with Chapters IV and V to develop the best possible methods and procedures for testing and balancing the HVAC systems.

5. The Agenda

An agenda should be prepared where possible and submitted on all jobs, prior to the installation of any of the HVAC systems. The agenda should include a preliminary reporting of any discrepancies that would prevent the proper balancing of the project. It should include brief descriptions about the system and their operation. It should include all proposed balancing procedures and note any items excluded. This is the time to request any needed additional data and/or clarifications.

Agendas are now being required by owners and system designers on many jobs, and they must be submitted and approved before any TAB work can begin. They soon probably will be required for all jobs. Essentially, you are telling the owner/designer before you start, exactly what you propose to do, how you propose to proceed to do it, and what you will need from them in turn to do the job. The important thing is that you submit the agenda before it's too late to make any needed changes, and that the owner/designer will have time to review the agenda and either agree or disagree with you before the systems are installed and covered up. Now is the time to bring out any misunderstandings or misinterpretations before time and money have been spent performing TAB work that may not be accepted. The agenda should include detailed information on the following items.

a. BALANCING DEVICES

Review the project drawings, schematics, and details to insure all necessary balancing devices such as volume dampers and balancing valves are provided to facilitate the balancing procedure. If more dampers or any other devices are necessary to properly balance the job, request them now. Sometimes it may be necessary to explain why these items are needed in these locations. Cost conscious owners and system designers are often reluctant about approving expenses for the purpose of balancing. It is important that you try to obtain these devices by including the request in the agenda and by a separate letter or change order. If the request for these additional items is denied, you will still have the documentation that they were properly notified when you previously requested them. This procedure may save you many problems later when the system cannot be balanced satisfactorily. Also bring to attention any balancing devices that may be inaccessible.

b. SYSTEM CAPACITY REVIEW

Check that the total flow requirements of all system terminal units equals the design fan or pump capacities. If a total system capacity exceeds the design of the fan or pump by more than ten percent, a *diversity procedure* should be established and clearly defined in the agenda. Be sure to verify that the system is capable of variable volume operation and control if a diversity balancing procedure is utilized.

Also, look for obvious static pressure and head discrepancies. Review the scheduled pressure drops of components compared to the fans and pumps for each system. Here again, time is of the essence. If a fan is undersized, you should be notifying the system designer and purchaser or owner before the fan is purchased and not after it's installed and operating unsatisfactorily.

c. LOCATIONS OF FLOW MEASUREMENT DEVICES

Determine the desired locations for the measurement of air and water flow quantities, and if access is readily available. Unfortunately, some of these locations (particularly Pitot tube traverse hole locations) will end up being relocated later due to unforeseen or unpredictable conditions which may prevent swinging or inserting the Pitot tube. Many times, test plugs will have to be installed before the ducts are externally insulated. However, even the best procedures can fail. When you come back to test, you may find the ceiling contractor has installed a tee-bar directly under the duct test holes. Such is life for a TAB technician!

d. SEQUENCE OF OPERATION

Examine the temperature control system diagrams to determine how to set the HVAC system components (ATC dampers, terminal boxes, etc.) and whether full heating or cooling is required for testing. Also be aware of any possible sequences that may result in an unbalanced system performance during normal operation.

Be sure that you fully understand the intent of all phases of the sequence of operation. You cannot correctly balance a system until you understand fully and exactly how a system is designed to operate. If there are questions on items that may need clarification, or possible deficiencies are found, they must be reported *now*. Consultation with the temperature control contractors' personnel may clarify some problems. Otherwise, also send them a copy of the agenda.

e. PROPOSED BALANCING PROCEDURE

Chapters VIII and IX give extensive detailed procedures for balancing different types of air and hydronic systems. By now the TAB technician is going to be very familiar with the system on the job. If there are any questions, a review of Chapters IV and V on types of systems may be in order and is recommended. After identifying the types of systems, determine which balancing procedure in Chapter VIII and IX will be used. As stated in these chapters, every system is slightly different and small modifications may be necessary to a procedure. Include in this section of the agenda, the exact procedure you intend to use. If you are excluding anything such as existing terminals with no given design cfm or gpm, additional dampers or drive changes, describe it here.

The agenda is an important and valuable submission for both the TAB contractor and the owner/designer team. It will demonstrate to the owner/designer that the TAB contractor has, so to speak, "done their homework" regarding the project.

The owner/designer may not agree with the TAB contractor on each item, but it makes them aware that you have taken more than just a casual glance at the plans and that you are concerned about performing professional TAB work that will result in satisfactory operation of the HVAC systems and provide comfort for the occupants. This will help open the doors of communication between the two parties, so that any differences may be settled promptly and pave the way for resolving additional problems which may arise later during the balancing process.

6. Instrumentation

Develop a list of test instruments that will be needed for the testing and balancing of the system components. Refer to Chapter II, "TAB Instrumentation and Use," for the proper application of test instruments. Where two or more identical instruments are to be used on the same job, it is advisable to compare the readings of both. Although both instruments may be within calibration limits, if one is on the high side and the other one on the low side, the difference can be enough to complicate the balancing process if their use is intermixed. Most instruments should read within ±5% of each other.

Review the instrument list to be sure that the correct instruments will be available when needed. Be sure that the instruments requiring calibration are up to

Figure 7-2 INSTRUMENTS SELECTED FOR A TAB JOB

date and that their certificates are readily available. Some instruments require considerable time to have recalibrated. Now is the time to get this done. Investigate all instruments to be sure that all accessories are intact and that the instruments are in good physical condition. It's very embarrassing to show up on the job to start balancing and find a needed accessory missing from an instrument or that the instrument is not operating properly.

C PLANNING TAB FIELD PROCEDURES

Prepare a schedule and plan of attack for the balancing of a project. Do not be caught off-guard with a phone call stating, "My ten-story office building is ready for balancing, and the tenants are moving in the day after tomorrow. May I pick up the TAB report tomorrow afternoon?" Review the progress schedule with your supervision for the project, and know when the systems are expected to be ready for balancing. Project how many TAB technicians should be in your crew and what instruments will be required to accomplish the testing and balancing in the most efficient manner for all parties involved.

Although the TAB contractor is not usually the cause of the delays, the TAB contractor usually is pressured to hurry up and get finished because the final payments of the installing contractors often are held up waiting for the balancing report. If you keep on top of a job before it's ready to start, you will have a good feel for when you will be able to start the TAB work. Many jobs run late and get behind. Then when the completion date gets close and liquidated damages are involved, everyone gets in a rush. The TAB contractor, by the nature of the work, is one of the last on the job and will be pressured.

To help protect yourself from last minute pressures, determine how much time it will take to perform the TAB work and notify all contractors involved in writing that you will need a specific period of time to perform your work after the job is complete and ready for balancing. Spell out what items you will need, such as all doors, windows, ceilings, thermostats, diffusers and registers and all controls complete and in automatic operation, before you can start. This way they know what they have to do and how long you will need before it's too late. If you get this letter out early, it may help you considerably later when the job may be running late.

If partial occupancy is planned, find out which sections of the building will be needed. Determine if it's possible to perform the TAB work for this area only. If not, a temporary balance may be required, and this should be negotiated as an extra cost item with a written change order.

D PRELIMINARY AIR SYSTEM TAB FIELD PROCEDURES

1. Readiness Check

Now that the preliminary TAB work in the office has been completed, you should have a good idea of what is involved as you proceed to the job. You will need to meet with foremen from the other trades involved, such as the pipefitters, sheet metal workers, electricians and temperature control system technicians. These people can inform you about what work is complete and operable. Also, they can be a wealth of information about the little peculiarities that usually exist on most jobs.

Although it's always desirable that the HVAC systems be 100 percent complete and in operation before the TAB work begins, this is often not the case. So you must find out what is ready and where you will be able to start. It's always a good idea to walk the entire job and observe the general condition of all the conditioned spaces and in particular, the mechanical equipment areas. Be on the lookout for missing terminals, incomplete controls, missing ductwork, missing thermostats and incomplete electrical wiring. Notice if the necessary architectural items are installed to obtain the normal HVAC air paths as designed. This would include ceiling plenums, doors, windows, partitions, etc. If items are missing that will be necessary for the balancing work, the proper people should be notified promptly.

Being one of the last trades on the job usually means that there is a lot of pressure to finish by the completion date and before possible liquidated damages begin. You should always promptly notify, in writing, the proper people about any items that are delaying you, so that you can protect yourself from being held responsible for the job not being completed on time. A check list similar to that in Figure 7-3 can be used to verify that the building and its systems are ready for the TAB work.

After completing the above inspections, systematically follow the steps listed below. Note that the responsibility will be considered different if the TAB contractor is also the installing contractor. If the TAB contractor is only doing TAB work, the installing contractor should be responsible for getting the equipment started and operating properly.

CHECK LIST

	Ready Yes	Ready No	Date Corrected
1. HVAC Units and Built-up Units			
a) *General*			
Louvers installed			
Manual dampers open and locked			
Automatic dampers set properly			
Housing construction-leakage			
Access doors-leakage			
Condensate drain piping and pan			
Free from dirt and debris			
Nameplate data			
b) *Filters*			
Type and size			
Number			
Clean			
Frame-leakage			
c) *Coils (Hydronic)*			
Size and rows			
Fin spacing and condition			
Obstructions and/or debris			
Airflow and direction			
Piping leakage			
Correct piping connections and flow			
Valves open or set			
Airvents or steam traps			
Provisions made for TAB measurements			
d) *Coils (Electric)*			
Size and construction			
Airflow direction			
Duct connections			
Safety switches			
Obstructions			
Free from debris			
Contactors and disconnect switches			
Electrical service and connections			
Nameplate data			
e) *Fans*			
Rotation			
Wheel clearance and balance			
Bearing and motor lubrication			
Drive alignment			
Belt tension			

	Ready Yes	Ready No	Date Corrected
e) *Fans (continued)*			
Drive set screws tight			
Belt guard in place			
Flexible duct connector alignment			
Starters and disconnect switches			
Electrical service and connections			
Nameplate data			
f) *Vibration Isolation*			
Springs and compression			
Base level and free			
2. Duct Systems			
a) *General*			
Manual dampers open and locked			
Access doors closed and tight			
Fire dampers open and accessible			
Terminal units open and set			
Registers and diffusers open and set			
Turning vanes in square elbows			
Provisions made for TAB measurements			
Systems installed as per plans			
Ductwork sealed as required			
b) *Architectural*			
Windows installed and closed			
Doors closed as required			
Ceiling plenums installed and sealed			
Access doors closed and tight			
Air shafts and openings as required			
3. Pumps			
a) *Motors*			
Rotation			
Lubrication			
Alignment			
Set screws tight			
Guards in place			
Tank level and controls			
Starters and disconnect switches			
Electrical service and connections			
Nameplate data			
b) *Piping*			
Correct flow			
Correct connections			

Figure 7-3 SYSTEMS READY TO BALANCE

CHECK LIST

	Ready Yes	Ready No	Date Corrected
b) *Piping (continued)*			
Leakage			
Valves open or set			
Strainer clean			
Air vented			
Flexible connectors			
Provisions made for TAB measurements			
Cavitation possibilities			
c) *Bases*			
Vibration isolation			
Grouting			
Leveling			
4. Hydronic Equipment a) *Boilers*			
Operating controls and devices			
Safety controls and devices			
Lubrication of fans and pumps			
Draft controls and devices			
Piping connections and flow			
Valves open or set			
Water make-up provisions			
Blowdown provisions			
Electrical connections			
Nameplate data			
b) *Heat Exchangers*			
Correct flow and connections			
Valves open or set			
Airvents or steam traps			
Leakage			
Provisions made for TAB measurements			
Nameplate data			
c) *Cooling Towers/ Evaporative Condensers*			
Correct flow and connections			
Valves open or set			
Leakage			
Provisions made for TAB measurements			
Sump water level			
Spray nozzles			
Fan/pump rotation			
Motor/fan lubrication			
Drives and alignment			
Guards in place			

	Ready Yes	Ready No	Date Corrected
c) *Cooling Towers/ Evaporative Condensers (continued)*			
Starters and disconnect switches			
Electrical connections			
Nameplate data			
5. Refrigeration Equipment			
Crankcase heaters energized			
Operating controls and devices			
Safety controls and devices			
Valves open			
Piping connections and flow			
Flexible connectors			
Oil level and lubrication			
Alignment and drives			
Guards in place			
Vibration isolation			
Starters, contactors and disconnect switches			
Electrical connections			
Nameplate data			
6. Hydronic Piping Systems			
Leak tested			
Fluid levels and make-up			
Relief or safety valves			
Compression tanks and air vents			
Steam traps and connections			
Strainers clean			
Valves open or set			
Provisions made for TAB measurements			
Systems installed as per plans			
7. Control Systems			
Data centers			
Outdoor/return Air/reset			
Economizer			
Static pressure			
Room controls			
8. Other Checks			
a) *Other trades or personnel notified of TAB work requirements*			
b) *Preliminary data complete*			
c) *Test report forms prepared*			

Figure 7-3A SYSTEMS READY TO BALANCE (CONT.)

2. Fans

(a) Locate the electric starter, disconnect switches and main circuit breaker. Familiarize yourself with any interlocked equipment and circuit interrupting equipment such as freezestats and firestats, etc. Confirm that thermal and electrical overload devices such as fuses and heater coils are installed and are the proper size.

(b) Confirm that electrical power is available for start-up and testing and that the voltage and equipment needs are compatible.

ALWAYS be alert for safety hazards. Before touching any equipment, be sure that it cannot be started while you are where you can be injured. **Be sure the electrical disconnect for the fan is OFF and tagged or locked.** The moving parts of fans can be very dangerous.

(c) Record all required data on the NEBB Test Report Forms. Include motor nameplate data, sheave data and belt sizes. This data is usually only available in the field.

(d) Compare the data on the fan nameplate to the manufacturer's submittal data. This confirms that the correct equipment has been furnished and is installed in the correct location.

(e) Investigate the inside of the fan and associated ductwork where possible, for any debris that might get into the fan and do damage when the fan is started.

(f) Check the alignment of the drives, using a good straight-edge or a string line. Always line up the straight-edge or string line from the fan pulley, which is fixed. If the fan and motor shafts are not parallel, the motor will have to be repositioned. Adjust the belts to the proper tension using a tension gauge for best results.

(g) Investigate the fan wheel for proper alignment and clearance in the housing. This will affect the fan performance. Investigate for any foreign items on the wheel such as plaster, mud, sealer, etc. These items can cause unbalance and vibration.

(h) Check all setscrews on the wheel, drives and bearings to be sure they are tight. Be sure keyways fit tight. (On certain types of sheaves and bushings, keys can be loose, even when the setscrews are tight.)

(i) Check the mounting bolts for tightness on the fan, motor, bearings, belt guard (if applicable), duct connections, etc.

(j) Confirm that the fan and motor bearings have been lubricated.

(k) Turn the fan slowly by hand and observe any rubbing of the wheel or the drives.

(l) If vibration isolators or bases are used, check for the proper adjustment, collapsed spring coils, etc., and that the equipment is level. If there are any abnormalities, verify that the correct isolators have been furnished and are installed in the right locations. (A heavier spring isolator is usually used under the motor location.)

(m) Check the alignment and tightness of the flexible connections with the duct. If they are now stretched tight, they may be damaged when the fan is started.

(n) Check for freedom of operation of any vortex dampers that may be provided.

(o) On roof top units and fans, confirm that there is an airtight seal between the fan and the connecting duct and/or curb.

(p) Locate and verify the size of electrical equipment such as disconnect switches, starters, heater coils, and interlocking controls.

(q) "Bump" the motor for rotation direction check. This is best done jointly with the electrician. If the rotation is incorrect, it will need to be reversed before proceeding. (Note that many fans will continue to move air even when running backwards. Therefore, rotation must be visibly checked.) Rotation arrows are furnished on many fans, but the TAB technician should observe the fan and its arrangement to verify what direction is proper.

3. Air Handling Units

(a) Complete the checks listed for fans.

(b) *Filters*

(1) Confirm that the correct size & type of filters are installed for the TAB work. (Sometimes different filters are used temporarily during startup and construction.)

(2) If the permanent filters are to be used, confirm that they are the correct size and type of filter by comparing them with the submitted data and the NEBB Test Report Form.

(3) Check the filters for cleanliness. If they aren't clean, have them replaced before starting TAB work.

(4) Confirm that the filters and filter frames are properly installed and are airtight. Any leaks must be corrected.

(5) If the unit has been running with no filters or very dirty filters, be alerted for possible dirty or clogged coils, etc.

(c) *Dampers*

Familiarize yourself with all unit and related dampers and their sequence of operation (includes fire and smoke dampers). Proper damper position during startup and TAB work is very critical. If dampers are closed, restricting the airflow, serious damage can be done to casings, housings and ductwork. It is generally best to work with a temperature control technician during startup to insure that all ATC dampers are positioned properly. However, it may be necessary to manually secure some dampers into position before starting the fan. This can be done with blocks of wood or by tying them with wire or rope. Great care must be taken to be sure that secured dampers cannot come loose and slam shut, causing damage to the damper and/or the housing and duct systems. When blocking dampers, confirm that the actuators will not move and twist or damage the secured dampers. It is safest to disconnect the actuators when dampers are manually secured.

The dampers should be in a position to insure the desired path for air to travel through the correct components of the system and not cause a choked or blocked condition. For startup and balancing the following damper settings are recommended for the most common systems.

(1) Outside air (O.A.) damper: Where separate minimum and maximum O.A. dampers are used, the minimum damper should be opened 100 percent. Where a single O.A. damper is used in conjunction with a minimum position controller, the damper should be opened approximately to a percentage equal to the percentage of minimum outside airflow. If the system uses 100 percent outside air, the damper will have to be fully open. (Note: Don't leave O.A. dampers open when the unit is not in use, especially during cold weather when freezeups may occur.)

(2) Return air dampers should be opened.

(3) Exhaust air dampers are usually left closed. Some systems may require opening them to a percentage equal to the minimum outside air damper setting. This is where a minimum relief or exhaust cfm is specified.

(4) Face and Bypass dampers should be set so that the airflow is through the coil. Multizone dampers should be set so the air will flow through the cooling coil. Confirm that the coils are sized for a cfm equal to the fan design. Occasionally coils are sized for less cfm than the fan. In this case, the bypass damper should be left open an amount equal to the excess fan cfm so that total cfm will not be restricted.

(5) Fire and smoke dampers will vary due to location in the system. Generally, they must be in the open position. More complex systems use a variety of damper configurations, particularly where multiple fans and sequences are involved. In this case, the sequence of operation must be studied and dampers must be open or closed accordingly.

(6) Vortex and other fan limiting dampers are often used with variable air volume (VAV) systems. It is safest to start the systems with these dampers throttled somewhat and then open them up slowly, observing the static pressure and amperage accordingly.

(7) Check for any type of temporary blockage over the outside air inlet opening and the exhaust air discharge openings, (such as polyethylene, cardboard or plywood), that may have been put there during construction. Look for any other types of debris or blockage in both the outside air and the return air duct systems.

(8) Check the coils for cleanliness and straightness of the fans. Confirm that the piping connections are correct.

(9) Confirm that condensate drains from the cooling coil drain pans have been provided and that they are properly trapped and functioning. Improper drain traps are a common cause of leakage, flooding and moisture carryover.

(10) Check the entire unit and the internal components for proper leak sealing. Leaks will cause whistling, possible moisture carryover and short circuiting of the air. Particularly check around pipes and panel holes. Have the leaks sealed.

(11) If the system has spray systems, they should be clean and operating.

4. Duct System Checks

(a) Walk the system from the fan on out to the last terminal. Observe the following:

(1) Is the ductwork complete and installed correctly? Are there any openings in the ductwork or any endcaps missing? Are all the access doors closed and secured tight?

(2) Are all the terminals, boxes, reheat coils, etc. installed?
(3) Does the installation match the plans?
(4) Are all the necessary architectural items installed, such as doors, partitions, ceilings and ceiling plenums, windows, etc.?
(5) Is the system *really* ready for balancing? This is important because the TAB contractor often is pressured by the owner, general contractors, mechanical contractors, or system designer to start the TAB work before the building and/or the system is ready. The TAB contractor must resist this pressure and educate these parties as to why they should wait; such as explaining why they will face additional expenses later when rebalancing may be required.

(b) Confirm that adequate balancing dampers have been provided and that all volume dampers, fire dampers and smoke dampers are installed at the correct locations. Verify that they are wide open and that adequate access has been provided.

(c) Confirm that any terminal boxes, such as VAV boxes and mixing boxes are installed and accessable. Confirm that the controls are energized and in operation. Thermostats must be installed and operable. For startup and full flow testing, most boxes will have to be set in the full cooling position by setting down the thermostat. This will insure maximum airflow and the least amount of system restriction.

Caution: *Some boxes are furnished with normally closed dampers. Starting a complete system with closed terminal box dampers will result in excessive system static pressures and possible duct system damage.*

(d) Verify that all terminal devices are installed and that their dampers are open. Confirm that the terminals are of the same size and type as specified. Frequently, substitutions are used due to space configurations or availability. This will usually change the A_k or K factor. (The size and type check can be performed while making the preliminary readings at the terminals.)

(e) Inspect the system for leakage. Specifically check access doors and hardware, coils, humidifiers, pipe penetrations, duct connections, flex duct and terminal connections. Confirm that any specified sealing has been done correctly.

(f) Where plenum ceilings are utilized, confirm that they are airtight. Pipe penetrations and any other holes should be sealed. Any air barriers must be well sealed. *This is extremely important on supply ceiling plenums.*

(g) Confirm that any required openings between partitions etc, have been installed and are open to insure proper air passage.

(h) The locations for Pitot tube test holes were covered in the preliminary office procedures section. Now these locations must be located in the field to confirm that they are accessible. Confirm that the actual duct installation matches the plans and that adequate straight sections of duct work are available for the tests. Look for obstructions that will hamper swinging the Pitot tube, such as pipes, ceiling supports, lights, etc. Pitot tube test holes must be sealed or capped when not being used. Some jobs will specify a metal test hole extension with a threaded gasketed cap. Otherwise, plastic snap-in plugs are available. Duct tape is a poor choice and should be avoided.

If the duct is insulated, any insulation removed for the test will have to be replaced and resealed. Use caution when removing insulation so that a neat repair can be easily made.

If the Pitot tube test holes and caps have already been installed by others, confirm that they are in the correct ducts and at satisfactory locations.

E PRELIMINARY HYDRONIC SYSTEM TAB FIELD PROCEDURES

1. Hydronic Piping System Checks

(a) Confirm that the system has been hydrostatically tested and is free of leaks.

(b) Trace the system piping from the source (e.g. boiler, chiller, heat exchanger) to all terminal units to determine:
(1) Completeness and integrity of the installation.
(2) Any variations between actual installation and design.
(3) That all required valves (Manual and Automatic) outlined in the agenda are open.
(4) That accessibility is readily available for testing of all balancing devices, flow meters, and terminal units.

(c) Confirm that the system has been cleaned, flushed, filled, and all air purged. Verify that strainer baskets have been cleaned and construction baskets (if used) have been removed. Maintaining system cleanliness is a full-time job, especially during initial system operation. Monitor the system cleanliness during the TAB work and be aware of any debris build-up that may affect the final system balance.

(d) Verify that the system water level and pressures are correct for the height of the highest terminal units. Procedures for open and closed systems are outlined below:

(1) *Open System*—Confirm that the system water level is correct and verify the operation of the make-up water device. Open systems with low static heads require special care on initial start-up to prevent inducing air into the pump. A good practice is to initially start the pump with the *discharge valve* partially closed so that the pump volume drawn from the sump will not exceed the make-up water capacity. Monitor the sump level and *slowly open the discharge valve* until the system is in full operation.

(2) *Closed System*—Inspect the pressure reducing valve(s) (PRV) for proper installation and operation. The setting of the PRV should always maintain a minimum 4 PSI static pressure at the highest point of the system. Check the water level of the expansion or compression tank. Generally, 1/3 air and 2/3 water is the proper ratio of a cold expansion tank.

(e) *Verify the proper installation of piping safety devices!* Do not attempt to operate any system without backflow prevention, or a closed system without the proper pressure relief valves.

2. Pumps

(a) Record all required data on the NEBB Test Report Forms. Include pump nameplate and motor nameplate data.

(b) Compare the nameplate data with manufacturer's submittal data and design requirements. Note any discrepancies.

(c) Verify that the pumps have been properly aligned, grouted, anchored, and lubricated.

(d) If vibration isolation is used, check for proper installation and adjustment. All piping should be supported independent of the pump housing. Operation of a pump without proper support may result in damage to the volute casing.

(e) Confirm that the drive guards are in place.

(f) Verify that access has been provided for accurate pressure drop readings.

(g) Verify the cleanliness of any strainer(s) serving the pumps.

(h) Purge all air from volute casings. **Caution:** *Operation of a pump with air may cause cavitation resulting in damage to impeller and volute casing.*

(i) Locate the electrical starter, disconnect switches, and main circuit breaker. Confirm that overload devices such as fuses and heater coils are installed and are the proper size.

(j) Confirm that electrical power is available for start-up and that the voltage and equipment needs are compatible.

(k) "Bump" the motor for rotation check. Be sure the pump is *completely* full of water if mechanical seals are used. Operation of a dry pump with mechanical seals, including "Bumping" will result in seal damage. If a packing gland is used, adjust for proper shaft lubrication.

3. Boilers

(a) Record all required data on the NEBB Test Report Forms.

(b) Compare the nameplate data with manufacturer's submittal data and design requirements. Note any discrepancies.

(c) Verify that the boilers have been started and tested for proper and safe operation in accordance with the manufacturer's recommendation.

(d) Locate and confirm that all combustion air openings and barometric or draft control dampers are the proper size.

(e) Confirm that the settings of all operational and safety controls for both temperature and pressure are correct.

(f) Confirm the proper operation of and lubrication of boiler equipment, motors, pumps, and boiler feed equipment.

(g) Verify that the water levels of steam boilers are steady and that the boiler(s) have been properly cleaned, flushed and blown-down.

4. Heat Exchangers

(a) Record all required data on the NEBB Test Report Forms.

(b) Compare the nameplate data with manufacturer's submittal data and design requirements. Note any discrepancies.

(c) Verify that the equipment is properly piped, including all required valves, vents, and safety devices. **Caution:** *Steam to hot water heat exchangers must contain a vacuum breaker on the steam piping to prevent exchanger damage.*

(d) Confirm that provisions are available for the required temperature and pressure measurements.

(e) Confirm the proper installation and operation of automatic temperature control devices.

5. Refrigeration Equipment

(a) Record all required data on the proper NEBB Test Report Sheets.

(b) Compare the nameplate data with the manufacturer's submittal data and design requirements. Note any discrepancies.

(c) Verify that the equipment has been started and tested for proper and safe operation in accordance with the manufacturer's recommendations.

(d) Confirm that the settings of all operational and safety controls for both temperature and pressure are correct.

(e) Confirm that provisions are available for the required temperature and pressure measurements.

(f) Confirm that all compressor crankcase heaters are in operation. **Caution:** *Do not attempt to start refrigeration equipment until crankcase heaters have been in operation at least 24 hours. Larger systems may require more time; check manufacturer's data for recommendations.*

(g) Check air cooled condenser fans and motors for proper rotation, voltage and phase. Confirm that dampers and all controls are functioning properly.

6. Cooling Towers

(a) Record all required data on the NEBB Test Report Forms.

(b) Compare the nameplate data with manufacturer's submittal data and design requirements. Note any discrepancies.

(c) Check the cooling tower sump water level and make-up water device. Adjust as required.

(d) Confirm that provisions have been made for condenser water bleed off and chemical treatment.

(e) Inspect the cooling tower nozzles for cleanliness and blockage.

(f) Check the cooling tower fans and motors for proper rotation, voltage, and phase.

(g) Confirm the proper settings and operation of all controls, dampers, and freeze protection devices.

7. Coils/Terminal Units

(a) Verify that the units are piped correctly including required valves, vents, and safety devices.

(b) Confirm that provisions are available for making the required temperature and pressure measurements.

(c) Confirm the proper installation and operation of automatic temperature control devices.

(d) Purge all air from terminal units on hydronic systems.

(e) Verify the proper airflow direction and fan rotation.

(f) Inspect the coils on both sides for fin damage and blockage.

(g) Confirm voltages and phases on electrical equipment of terminal units.

NEBB

CHAPTER 8 AIR SYSTEM TAB PROCEDURES

A THE SYSTEM FAN

Upon completion of the procedures detailed in the previous chapters, you are ready to proceed with the actual TAB work in the field. To use these procedures will require referrals to previous areas of this manual for specific details on the use of the instruments.

1. Preparation

(a) Assemble the previously prepared paperwork, drawings, etc. for the system that you are starting on.

(b) Verify that all preliminary procedures have been performed (see Chapter 7).

(c) If you are doing the initial start up on the fan, have a temperature control technician and an electrician work with you if at all possible, particularly on the larger and more complex systems.

(d) Take a last minute check of the dampers and drives. Verify that nobody is working in, on or around the equipment that could be injured upon start up.

2. Fan Startup

(a) Verify that the fan rotation is correct by "bumping" the motor. "Bumping" is a procedure where the motor is energized for a fraction of a second while someone watches the drive to see what direction in which it rotates. If the rotation is incorrect, it will have to be changed. This normally should be done by the job electrician, as this eliminates the possibility of the TAB technician being held responsible for any possible damage done to the motor or equipment. Reversing the rotation in three phase motors involves interchanging any two of the three phase lines coming into the motor. This is most easily done at the electrical starter or disconnect switch. On single phase motors the change must be made at the terminal junction box on the motor itself. There is usually a diagram, either on the back of the terminal junction cover or on the nameplate showing which wires to interchange. **DANGER: Electrical shock can be fatal.** Use caution before handling any wiring. Confirm that electrical disconnect switches are off and tagged. They should be locked, if possible, so that they aren't turned on by others while you are working on the equipment.

(b) You are now ready to start the fan. On larger, built-up systems, it is desirable to have an observer watch for any possible mishaps such as the throwing off of belts, excessive fan movement, duct bulging and any unusual noises indicating possible mechanical problems. Be ready to turn the fan off immediately if anything isn't right.

(c) Upon starting the fan, quickly check for the obvious mechanical malfunctions. Then measure the amperage and voltage of the motor. Voltages should be within 10 percent of the motor nameplate rating. There should be very little difference of voltage and amperage between each leg on three phase motors. Amperage should not exceed the nameplate ratings. If the amperage is high, stop the fan immediately.

(d) If everything is operating correctly, you should proceed to start any associated equipment related to or interlocked with this unit (such as a return airfan), so that you have a complete system in operation and ready to balance.

(e) Quickly go to each automatic damper that hasn't been blocked or disconnected and confirm that the damper is being controlled automatically and is in the correct position. If the system is equipped with an economizer cycle, the mixed air temperature will be controlling the position of the outside air (O.A.), return air (R.A.) and exhaust air (E.A.) dampers accordingly. The same goes for face and bypass dampers. There will be some effect on the airflow when these dampers are "hunting." This is un-

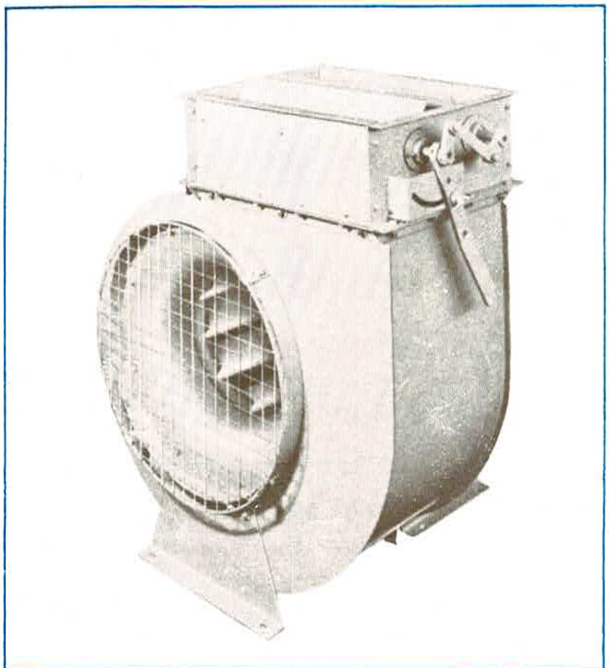

Figure 8-1 FAN WITH INLET SCREEN AND OUTLET DAMPER

desirable while doing air balancing. Therefore, the dampers or their controls should be blocked out to keep them in the desired position. All dampers should be set for a full flow cooling condition.

3. Fan Tests

a. AMPERAGE

Check the amperage using a clamp-on ammeter. For the most accuracy, the test should be done at the motor junction box. However, it is usually easier and safer, to test it at the disconnect switch or at the starter. Be sure to take amperage readings at the output leads of a starter rather than the input leads because quite often the controls in the starter are energized from the power coming into the starter and add some amperage to the circuit total. For the most accuracy, the closer to the motor the reading is taken, the better.

If the amperage readings are over the nameplate rating, the fan should be stopped immediately. If the amperage is considerably over, the starter should trip-out quickly and stop the fan before doing any damage to the motor. If the amperage is high, you will need to find out why. The most common cause with new unbalanced systems is that the fan is turning too fast. This usually can be corrected by adjusting the motor sheave to reduce the fan rpm.

If the amperage is considerably over the nameplate rating, examine the wiring hookup at the terminal junction. There should be a wiring diagram somewhere on the motor or access cover. Verify that the leads are all connected properly for the voltage in use. If you find the amperage considerably below the rated full load amperage (FLA), suspect a low airflow in the system. *Always recheck the amperage whenever any rpm change or major damper setting change is made.*

b. FAN SPEED

Test and record the speed (rpm) of the fan using one of the instruments described in Chapter 2. If a direct contact tachometer can't be used, then use one of the electronic tachometers. Follow the instructions carefully, because on some types it's very easy to get false readings if you aren't familiar with using them. The fan rpm should also be checked mathematically by using Equation 4-8 from Chapter 4:

$$\text{Fan rpm} = \frac{\text{motor rpm} \times \text{motor pulley diameter}}{\text{fan pulley pitch diameter}}$$

For utmost accuracy when using the equation, the motor rpm should be tested because it's usually a little different than the nameplate rpm. For most general TAB work, however, this won't be necessary. On some installations, calculation may be the only method available to obtain the fan rpm.

c. BRAKE HORSEPOWER (Bhp)

From the amperage and voltage readings, the approximate Bhp can be determined using the procedures and equations outlined earlier in the manual. The Bhp should compare favorably with the manufacturers' submitted data. The Bhp is one of the more accurate field tests and is a valuable figure when working with fan curves.

d. STATIC PRESSURE (SP)

Static pressure tests are necessary and can be very helpful when trouble shooting. Unfortunately, they are usually the most difficult readings to obtain accurately in the field. There are many reasons for this, but the most common ones are turbulence of the air at the fan inlet and outlet and the inability to take the readings where they should be taken to test the pressure exactly as the fan senses it.

Static pressure and total pressure readings should be taken at the fan discharge and inlet. This is most easily done next to the flexible connections. This, however, is also one of the worst and most turbulent positions available. So the pressure readings should also be taken again further downstream, preferably in a section of straight ductwork. Be aware, however, that any transitions and elbows between the fan and the test point will have an effect on the readings. Static regain also can be taking place near the fan. The fan inlet SP can usually be taken in the plenum around the fan inlet, or with direct duct connections, at the flexible connections at the entry point. Pressure drops across the individual components are usually required and will be necessary when trouble shooting or comparing rated vs actual pressure drops.

A straight line sketch should be made showing each component and the pressures and pressure drops between each. Pressure drops across components can be taken as a single reading by using a two-hose hookup on the manometer from the high and low pressure sides of the component. The single reading on the manometer eliminates any pressure effects from the atmosphere immediately where the instrument is located.

Many fan rooms are also return air or outside air plenums. When taking readings in these rooms, it will be necessary to reference the manometer to the atmosphere. This is accomplished by running a length of hose from the opposite port of the manometer to the outside or atmosphere. Take advantage of existing possible openings available for static pressure readings such as access doors, handles, bolt holes etc. that may be removed to get the SP probe into the airstream. If none are available, you will have to drill your own. Be sure to close or cap them when finished.

e. AIRFLOW

The most accurate and accepted field test of airflow is by a Pitot tube traverse of the duct being tested. Anemometer traverses across coils and/or filters are a poor substitute and should only be used in an emergency and with considerable reservation. Field tests have shown that they will vary considerably. They usually tend to read higher than the actual system airflow, but no definite pattern has evolved.

A total of the terminal readings will be useful to compare with the Pitot traverse readings when system air leakage is suspected. There will be instances when they will be the only field readings available for the system total airflow. Fan curves can be used when other required data can be obtained, such as SP, rpm and Bhp. Experience has shown, however, that often all of the field readings will not fall into place on the fan and system curves. Therefore, it is best to make Pitot tube traverses whenever possible and use them in conjunction with the other test data and fan and system curves to tell what actually is happening.

The accuracy of a Pitot tube traverse is determined by the availability of a satisfactory location to perform the traverse. Reasonably uniform airflow through the duct is necessary. Ideally, you should have six to ten diameters of straight duct upstream from the test location. Realistically, you won't find this condition very often in the field. Therefore, you will have to use the best locations available. Avoid getting close to elbows, offsets, transitions or anything else in the duct that is creating turbulence.

While taking Pitot traverses, notice if the readings are fluctuating up and down. This indicates turbulence and reduces the accuracy of the test. When encountering turbulence, a larger manometer will read steadier. (For instance, a 1" to 10" manometer will read steadier than a straight, one inch manometer, due to the dampening effect of the added volume and weight of the fluid). When recording turbulent readings, try to get an average reading between the peaks. If the Pitot tube traverse indicates that the airflow is considerably different from one end or side of the duct to the other, the accuracy will be decreased. Where the readings across the duct aren't uniform, additional holes can be drilled and additional readings taken to improve the accuracy.

If the Pitot tube traverse readings are taken at a good location and the readings are reasonably steady and uniform, these readings are going to be your most accurate field measurement of the system airflow and should be used accordingly. When the readings are not steady and uniform, they should be used in conjunction with the other test data and the fan curves to make a determination. The fan curve and fan speed data, when used with the calculated brake horsepower, will give the most accurate field readings that can be relied on heavily. Static pressures will be the least accurate field readings along with airflow readings, depending on how and where they were taken. But with a combination of these readings, you should be able to make a reasonable determination of the performance of the fan.

Don't be surprised when all of this data doesn't fall into place on a fan curve. Field readings aren't that accurate, and fan curves do not reflect installed conditions. The fans were tested in a laboratory under ideal conditions. Accurate HVAC system airflow readings should be taken with a wet cooling coil. If this isn't possible, allow for some loss of cfm. When the coil is in use and wet, 5 to 15 percent difference in airflow and static pressure readings is common.

f. SYSTEM DEFICIENCIES

Compare the actual results of the above tests with the specified performance of the fan. If the fan airflow is not within 10 percent of design, try to find the reason for the difference. Determine if the pressure drops across the duct system components (such as coils, filters, sound attenuators, eliminator blades, etc.) agree with the manufacturer's ratings. Observe the duct system configurations at the inlet and discharge of the fans. Compare these with the contract drawings. Notice if any radical changes were made to the duct system layout during installation. If any corrections are needed, report this to the appropriate persons.

If there are no obvious deficiencies, and the airflow is high, the fan can be slowed down by adjusting the drives or making drive changes. When the airflow is low, the fan speed should be increased. Before doing this, determine if there is adequate motor horsepower available. The new airflow-horsepower relationship can be determined by use of the fan laws found in Chapter 4 or Chapter 7. The fan curves are a better reference, if available. If any sizable upward change is being made, the fan manufacturer's data should be checked for the maximum allowable rpm for this fan and its bearings. If the horsepower and static pressure is available and the fan speed can be increased, adjust the drives accordingly to obtain the desired cfm.

When new systems do not perform as designed, new drives and motors are often required. The financial responsibility for these items does not usually belong to the TAB contractor, but the submitted readings will have a lot to do with determining who is. Be sure to include your data together with explanations about how and where the readings were taken. The above steps also should be used for any return air or exhaust air fan associated with the HVAC system in question.

g. OUTSIDE AIR

Most systems are designed to operate with a minimum amount of outside air whenever the building is occupied. Everyone is more conscious of outside air quantities now that energy has become so expensive. The procedure for setting the outside air (O.A.) quantities will depend on the system and damper scheme.

Where a separate minimum and maximum O.A. damper are provided, start with the minimum O.A. dampers and the return air (R.A.) dampers open. The maximum O.A. dampers and exhaust air (E.A.) dampers should be closed. The airflow adjustment will be made by fan speed changes or minimum O.A. damper adjustment. Most systems use just one O.A. damper in conjunction with a minimum position controller, which will open the outside air and exhaust air dampers while closing the return air damper. Adjustment is made at the controller. Always ask the temperature control technician to assist if possible.

The quantities of outside air should be tested by making a Pitot tube traverse of the O.A. duct where possible. Otherwise, calculate the amount of outside air by subtracting the actual return air cfm from the actual supply air cfm. If either actual cfm test isn't possible, use the *temperature method* with the following calculations.

Equation 8-1

$$T_m = \frac{X_o T_o + X_r T_r}{100}$$

Where:

T_m = Temperature of mixed air (°F)

X_o = Percentage of outside air

T_o = Temperature of outside air (°F)

X_r = Percentage of return air

T_r = Temperature of return air (°F)

Example 8A

The supply air fan of the HVAC system furnishes 12,500 cfm, and the specified outdoor air quantity is 2500 cfm. Measurements indicate O.A. = 92°F and R.A. = 74°F. Calculate the mixed air temperature that would allow the correct amount of outside air.

Solution

$$\%\ \text{O.A.} = \frac{2{,}500\ \text{cfm}}{12{,}500\ \text{cfm}} \times 100 = 20\%$$

$$\%\ \text{R.A.} = 100\% - 20\% = 80\%$$

Using Equation 8-1:

$$T_m = \frac{X_o T_o + X_r T_r}{100} = \frac{20\% \times 92° + 80\% \times 74°}{100}$$

$$T_m = \frac{1840 + 5920}{100} = 77.6°\text{F}$$

Therefore, to obtain the correct amount of outside air (2500 cfm), the dampers will need to be adjusted to obtain a mixed air temperature of 77.6°F.

Quite often, the TAB technician is going to find that the mixed air temperature is very difficult to test accurately. The duct configuration of many systems creates a considerable amount of airstream stratification. Mixed air temperatures will vary considerably, depending on where the readings are taken. In this case, it will be necessary to take several temperature readings in the form of a thermometer traverse. Average the readings to obtain the correct mixed air temperature. This can be a time consuming process and a quick reading digital thermometer will speed things up.

A helpful way to lay out the temperature traverse is to use the center of each filter section. Readings downstream of filters are usually acceptable. Never, however, take mixed air temperature readings downstream of a coil (in use or not) or a fan as they usually will be unreliable. Drastic air stratification also can cause other problems such as coil freeze-up and bothersome freezestat tripping. If this is observed, it should be reported so that corrections can be made, such as using baffles.

h. ECONOMIZER SYSTEMS

If the HVAC system is equipped with an economizer control system (using outside air for "free" cooling), the dampers should be cycled from minimum O.A. to maximum O.A. and back while observing the fan motor amperage and the static pressures at the mixed air chamber, supply air fan, and the return air fan if applicable. Any changes in the amperage and/or static pressure indicates a changes in airflow quantities. Some variation is not unusual and is to be expected. Confirm that all motors in the system operate below or at the rated full load amperage. Confirm that all of the automatic dampers operate simultaneously and in the proportions of flow as designed. Damper operation lag is a common cause of reduced airflow due to the choking affect of one damper closing before the other one opens. If this is observed anywhere in the cycle, the temperature control contractor should be notified.

i. DATA ENTRY

By now you should have all the fans in the HVAC system and related systems operating and set up to the design airflow. All data should be entered on the NEBB Test Report Forms.

You are now ready to test the rest of the system.

B CONSTANT VOLUME SYSTEM PROCEDURES

1. General

The previous fan tests were made to determine that the system had the proper airflow and pressures available to continue with the system balancing and that all HVAC units and fans were operating properly. Balancing of the duct systems consists of measuring and adjusting the airflow to the desired rate of each terminal device. This process will require adjustment of volume dampers, and possible readjustment of fan speeds.

Attempting to use terminal device dampers (such as the damper blades of a register) for system balance can result in improper balance, excessive noise and poor air patterns. Individual branches or runouts should be balanced using the proper type of volume damper. The TAB technician will encounter many instances where there are not enough dampers provided. If the TAB contractor is also the installing contractor, this is not usually a problem. But if the TAB contractor is doing only the balancing work, the system designer should be advised if the proper types and locations of dampers are not shown or specified. Many times the specifications will call for dampers wherever the TAB contractor recommends them. This type of specification is wrong, as it can be difficult to get anyone to assume the financial responsibility for the additional dampers needed, but not shown.

2. Balancing Procedures

The procedure used for balancing a specific HVAC system will have to be tailored to that system. There usually are little differences on each system that will require some change of procedure from the ones listed below, which the TAB technician will have to recognize and take into account.

The following procedure is probably the most common one in use. Another procedure called the "Ratio Method" can be found in the NEBB "Procedural Standards for Testing, Adjusting, Balancing of Environmental Systems," Fourth Edition, 1983. Both methods obtain the same end results which is the correct airflow rate in cfm at each individual terminal, with the least noise, drafts and system electrical power consumption.

3. Pitot Tube Traverses

Pitot tube traverses have already been taken to obtain the total system airflow. It will now be necessary to make a Pitot tube traverse at each major branch duct. If the system covers more than one floor, a traverse is made at each duct where it comes off the riser. These locations should be determined when the initial paperwork is setup in the office. The necessary airflow data will then be ready on the appropriate NEBB Test Report Forms. Always record the static pressure at each Pitot tube traverse for future use in setting zone dampers and possible trouble shooting.

4. Zone Balancing

Using the acquired data from the Pitot tube traverses, determine which zones are high on cfm and which are low. By using Equation 4-3 from Chapter 4, you can determine the new required static pressure at the traverse location that will result in approximately the correct airflow (cfm).

Example 8B

A branch duct requires 1000 cfm of airflow. Pitot tube traverse measurements indicate 835 cfm at a static pressure (SP) of 0.13 in.w.g. Find the SP that will result in the required airflow.

Solution

$$\frac{SP_2}{SP_1} = \left(\frac{cfm_2}{cfm_1}\right)^2$$

$$SP_2 = SP_1 \left(\frac{cfm_2}{cfm_1}\right)^2 = 0.13 \left(\frac{1000}{835}\right)^2$$

$$SP_2 = 0.186 \text{ in.w.g.}$$

You can now adjust the zone damper on each zone that is high on airflow. Monitor the SP at the Pitot tube traverse location while closing the damper. Slowly close the damper until the SP comes down to the new required SP determined by the equation. You should now have approximately the correct cfm in this zone. This procedure should be used on each zone with high airflow, usually starting with the highest one first. Then remeasure the SP in all of the zones. There will usually be some interaction between the zones. Some of the adjusted zones may need adjusting again. The zones that were low in airflow should have increased, and now some of these may be high and may themselves need adjusting. It usually will take two or three passes of adjusting the dampers before all of the zones are correct. After the zones are adjusted to the new calculated SP, you can proceed to the terminal units. If you can't obtain the correct SP, check the following:

(a) Whenever you close a damper, you are increasing the SP upstream of that damper. Whenever there is an increase in the SP that the fan is working against, there will be a decrease in cfm from the fan. Therefore, when you throttle several zone dampers, you can expect an increase in the discharge SP (inlet SP on return air and exhaust air fans) at the fan and a reduction in total airflow and usually bhp from the fan. Therefore, it's good practice to allow a little extra cfm when setting up the fans to allow for the expected loss. A rule of thumb would be to allow ten percent extra, but be careful that you do not overload the motor. It may still be necessary to increase the fan speed after balancing the zones to get the total fan airflow up to design conditions.

(b) Although Equation 4-3 is a basic fan law, there will be instances where it will not give the correct results in the field. Usually this is because field readings of SP and cfm are not always completely accurate. Although they are the best readings available to the TAB technician, there will always be room for error. For this reason new Pitot tube traverses should be taken if there is any suspicion of a reading. Spot checks of traverses are not always reliable. Although you may have the correct total airflow, the velocity cross-patterns in the duct can change after an adjustment, particularly if the traverse is located near a damper.

(c) There will be times when the zone traverse will be upstream of the zone damper. Therefore, you won't be able to use the SP at the traverse location to readjust the damper. An easy solution is to take a reference SP downstream of the damper and calculate the new required SP at this location using Equation 4-3. Then adjust the damper using the SP at the reference location.

(d) When using Equation 4-3, it is absolutely necessary to be sure that no adjustments or changes are made to the system downstream of the zone damper from the start to finish of the damper adjusting process. Opening or closing a terminal damper or changing anything that will affect the cfm or SP that is different between the first actual SP and cfm reading and the final zone damper adjustment, will throw the process off by changing the system curve. For instance, if after taking the initial zone traverse and SP you inadvertently find a diffuser that has its damper closed, opening it now will increase the zone cfm, but decrease the zone SP. This reverse effect precludes using the equation without taking new zone SP and cfm readings after opening the damper.

(e) If a zone is low in airflow and doesn't seem to increase like it should, the damper should be checked as well as the duct for blockage and/or leakage. Quite often a damper handle or quadrant will indicate an open damper when the damper may be closed or partially closed. Mechanical problems such as improper installation, slipping setscrews, etc. can cause this. If possible, the damper should be visually checked. Otherwise, a SP drop across the damper can be taken. If the damper is wide open, the SP drop will be very low. The actual drop will vary depending on the velocity of the air passing through the damper.

Another method for rough setting a zone damper, is to pick one diffuser on the zone to be adjusted. Test and record this cfm reading. Then calculate the percentage that the entire zone needs to be reduced. Use this percent-

Figure 8-2 TERMINAL BALANCING

age figure to reduce the tested terminal. For example, if the total zone needs to be reduced twenty percent, close the zone damper until the airflow on the tested terminal has been reduced by twenty percent. Then recheck the zone cfm.

(f) There will be instances where a zone damper will need adjusting but there won't be any satisfactory location for Pitot tube traverse. In this instance, it will be necessary to take cfm readings at all the terminals on the zone and total them. Use this total cfm and take a reference SP as detailed earlier, and then proceed to balance the zone. Often this will result in a decrease in accuracy, but you should still be able to get the zone set close enough to proceed.

5. Terminal Balancing

Using the following procedure will allow you to balance the terminals starting from the fan out to the end of the system. You should have the NEBB Test Report Forms with all the terminals located and numbered. Using the appropriate instruments and procedures as delineated earlier, measure and record a preliminary reading at each terminal. This is a good time to also verify the size and type of terminal to be sure that the installed terminal is what was specified. If not, the A_k factor will surely be different. Unless a direct reading hood for airflow is being used, be absolutely sure that the manufacturer's published A_k and measurement procedure are being used. Many hours have been wasted searching for a problem when the wrong A_k was used. Direct reading hoods eliminate the need for A_k factors, special procedures, and also decrease the time needed for balancing.

After testing and recording all the terminals, total the readings on a zone by zone basis. Compare the totals to the comparable zone duct traverse reading and the required cfm. The terminal total should be close to the traverse reading for the zone. The terminal total will usually be a little lower due to some expected leakage found in unsealed low pressure ductwork. The accuracy of a good Pitot Tube traverse is usually considerably better than most terminal readings. On occasion, you will get a higher airflow total from the terminals than from the traverse reading. This is usually due to inaccuracies in the terminal readings.

If the readings indicate a loss of more than ten percent, you won't be able to balance the system properly. Investigate the duct joints and connections. Investigate the terminal connections and in particular, the plenums for linear diffusers. Also recheck for open access doors, holes in the ducts, etc. Repairs can usually be performed easily using sealer or tape. Notify the proper persons to have the leaks corrected. Attempting to balance a system with leaks usually results in wasted time and unsatisfactory results. As you close down terminal dampers, more air will be forced out through the leaks.

When it is determined that sufficient air is available in the system and at the terminals, proceed with the TAB work. Since the air will go first to the points of least resistance, you will usually find that the terminals closest to the fan are high on airflow, and that those near the end of the line are low on airflow. The results of your preliminary readings will tell you exactly what your system is doing and were the air is. Review the readings and pick out any terminals that are high on cfm. Look particularly for any that are excessively high and go to them first and reduce them by closing their dampers. On the first "adjusting pass" through the system, it usually helps to throttle the terminals to about ten percent under design cfm. This will allow for the possible buildup in the other terminals as adjustments are made. After reducing the high airflow terminals, proceed to make another pass through the entire zone or system. Adjust each terminal to the specified airflow when sufficient air is available. After two adjusting passes, most systems should be in good balance. An additional pass will probably be necessary to "trim" the system.

On unusual systems, more passes may be necessary. The final pass should be readings only, with no adjustments, and the readings are to be recorded in the final column of the NEBB Air Outlet Test Report. Often during the balancing process, the balancing results in an increase in the system static pressure to the extent that the fan airflow is reduced to where it is more than ten percent below design. In this case, it will be necessary to speed up the fan to overcome this loss in cfm. Follow the same precautions explained earlier in this chapter regarding the fan. *Don't overload the motor.*

The dampers that are furnished as part of the terminals (diffusers, registers, etc.) should only be used for minor "fine tuning" the system. Always use the balancing dampers in the ductwork where provided. Using the terminal dampers can result in excess noise as the damper is closed down. Quite often the damper can be closed all the way and still leak more air than the space requires. After making a terminal damper adjustment, the air pattern will often be altered to the extent that the A_k factor will be affected. This effect will vary with the damper type, location, air velocity and how far the damper is closed. Quite often, these dampers don't fit tight in the duct, so as they are closed, air will leak around the outside of the damper.

When the dampers on a register are closed, a pattern of airflow across the face of the register will develop where a high velocity will be jetting from the damper blade openings, with practically no airflow between the open areas. A velometer tip or any small tip reading instrument will be unsatisfactory for testing under these conditions. A direct reading flow measuring hood will read these terminals accurately. If the proper A_k factors are available, a rotating vane anemometer also may be used.

Most manufacturers of perforated diffusers publish an A_k factor and a procedure for testing. Field results have demonstrated that there still is a lot of difficulty obtaining accurate readings. The only way that consistent results have been obtained testing perforated diffusers has been by using the flow measuring hood.

If the balancing is resulting in noisy or other unsatisfactory results, the TAB technician should notify the proper people. If dampers are needed, it is not too late to recommend them. If after notification, no satisfactory response is obtained, the problem should be noted on the Test Report Forms.

6. Final Tests

After all balancing adjustments have been completed, final readings should be taken. They should include a retest of the amperage, voltage, static pressures and rpm of the supply air (and return air) fan. Final terminal readings should have already been taken. These results are to be submitted and represent the exact condition of the system when the TAB technicians completed the job. The system is considered balanced when the airflow of each terminal is ± 10 percent of the design airflow, unless there are conditions beyond the control of the TAB firm. All deficiencies should be outlined on the report ($\pm 10\%$ is the NEBB standard and is widely accepted. You must review the job specifications however, because there will be exceptions to this, where the tolerance will be lower.) At this point, all the dampers should be marked for ease of resetting in the event of tampering. (Spray paint works well.) Operation of automatic controls should be checked and verified.

If there are any obvious air drafts, or complaints of same exist, adjust the deflectors or diffuser cones, as provided, to eliminate any non-uniform room conditions. Generally, a horizontal air pattern is preferred, but be sure to check the contract documents. Quite often, desired air patterns are specified.

If there are areas of discomfort or temperature problems, it may be necessary to make adjustments to change the airflow from what is specified to serve these areas. This is something that TAB technicians should never do until they have been given approval by the appropriate people. The initial job of the TAB technician is to obtain the correct cfm shown on the plans for the terminals. If changes are necessary to obtain the desired space temperature, extra expenses are often incurred. You must refer to the individual job specifications to ascertain your obligation on each job.

After completing all of the actual field testing and balancing, prepare the NEBB Test Report Forms for submission to the office. Review them carefully to be sure that nothing has been omitted. Confirm that all readings are the latest ones taken. Include detailed notes, etc. on all problem areas, deficiencies and other abnormal conditions found. Be sure that all the necessary field data is recorded. This will eliminate unnecessary trips back to the field. Submit the field reports to the office for final preparation.

C PROCEDURES FOR OTHER SYSTEMS

1. Multizone Systems

A multizone unit (Figure 8-3) uses one fan that can blow air through two paths, usually a cooling coil and a heating coil or a perforated plate, before being discharged from the HVAC unit. After the air passes through each coil into a cold air plenum or a hot air (or neutral) plenum, the air then passes through mixing dampers into two or more zone ducts serving various spaces. Each zone duct, usually close to the unit, has a manual volume damper that is used to balance the cfm to each zone. These balancing dampers definitely are required, since the mixing dampers are not capable of controlling the total airflow quantity.

Some units will not have a heating coil, but will just bypass the return air-outside air mixture when cooling is not needed. However, multizone systems normally are balanced with all the zones in the full cooling position. There are, however, exceptions. The cooling coil may be designed for less air than the fan delivers and the total system requires, because the building normally will not need full cooling in all zones at the same time. This diversity is caused by the sun load of the spaces changing from east to west during the day. It will be necessary to check the manufac-

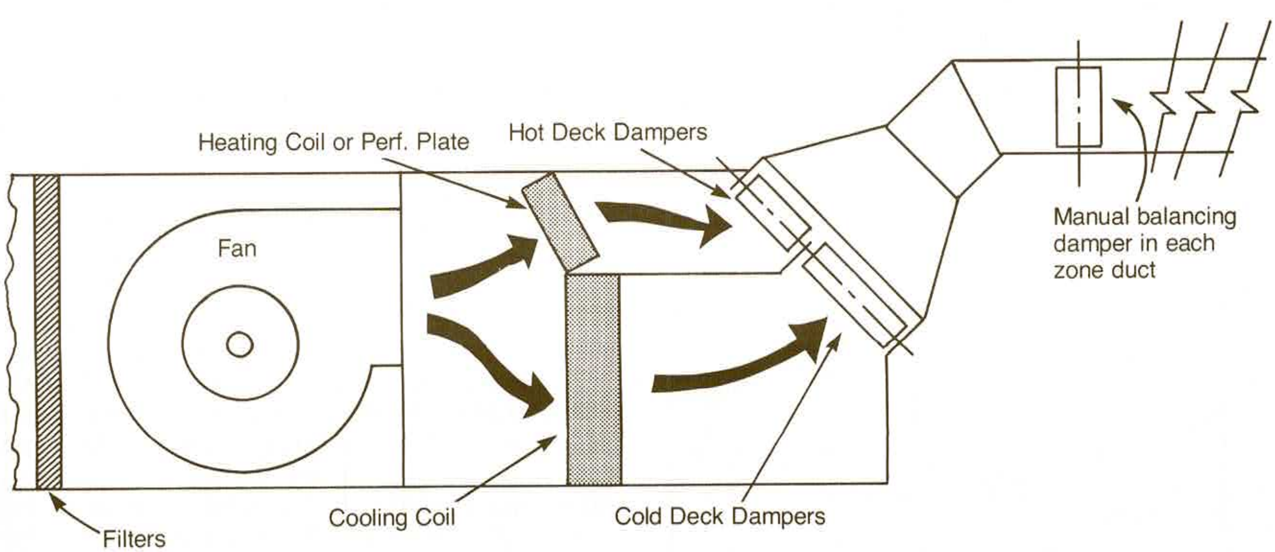

Figure 8-3 MULTI-ZONE SYSTEM

turer's data to determine if the cooling coil is sized for full airflow or if a diversity factor has been used. If there is a diversity, set enough zones into full cooling to equal the design cfm of the coil. The remaining air will then go through the heating coil or the bypass.

If the cooling coil is sized for the full fan airflow, put all zones into full cooling. This is most easily accomplished by setting each zone thermostat to its lowest point. The system is then balanced (similar to any low pressure, constant volume system detailed earlier in this chapter) as outlined below:

(a) Set up the fan and put the system into full flow through the cooling coil with allowance for diversity.

(b) Make a Pitot tube traverse of each zone and total the results.

(c) Make any required fan adjustments to obtain the design total cfm.

(d) Adjust each zone damper to obtain the proper cfm in each zone. This should be done by making a traverse if possible. This type of system usually can not be balanced satisfactorily without zone balancing dampers. If they are not installed, the TAB contractor should notify the proper people to have them installed.

(e) Once each zone has the correct airflow, the terminals can be balanced by using the previously described methods.

Some units (called "Texas multizones") will not use a heating coil but will bypass the air instead. Quite often, a manual damper will be provided in this bypass instead of a perforated plate. This damper should be set to provide the same total cfm whether the air is going through the coil or bypassing it.

During normal operations, there will be some variation in airflow as each zone satisfies its individual requirements. It is also not uncommon for the system to move less air when it is in the heating mode.

2. Variable Air Volume (VAV) Systems

a. GENERAL

VAV system are now used extensively in commercial buildings. Although they are available in various forms, most VAV systems use a single duct supply to the boxes; but there are variations which use dual ducts (hot and cold), and these are discussed later in this chapter.

The fan setup is similar to that described earlier for low pressure systems with one major difference, that being the ability to control the volume of cfm. Airflow control can be accomplished with inlet or discharge dampers, variable speed drives or variable speed motors. A static pressure sensor, preferably located about two-thirds of the way from the fan to the end of the duct system, senses the duct static pressure and sends a signal back to the apparatus controlling the fan airflow volume. The controls are set to maintain a constant static pressure at the sensor location, as the system cfm varies up and down.

To set up the fan requires that the system demand is the same cfm as the fan is designed to produce. To get the total of the terminal cfm requirements equal to the fan design cfm, set the thermostats controlling each box to the lowest temperature. If the system has a diversity, meaning that the totalled terminal cfm requirements are more than the fan design airflow volume, you must place a number of terminal boxes into a full flow condition, where the total airflow from the boxes equals the design cfm of the fan. This means that some of the terminal boxes will be in a minimum flow condition. Be sure to spread these terminal boxes throughout the system and not have them all on one major branch of the duct system. After ascertaining that the system demand cfm is equal to the fan design cfm, the fan can be tested using the normal methods previously described.

b. PRESSURE INDEPENDENT VAV SYSTEMS

Pressure independent VAV boxes have the ability to maintain a constant, maximum airflow and usually a constant, minimum airflow also, as long as the box inlet static pressure is within the design range of the VAV box. The manufacturer's published data gives the details for the static pressure operating range and the minimum static pressure drop across each terminal box for a given cfm. Use this data to verify that adequate pressure is available for the terminal box to function properly. The objectives of balancing pressure independent boxes is the same, regardless of the type of controls used. They must be adjusted to deliver the specified maximum and minimum airflows, (the minimum airflow can be zero in many instances).

For simplification, consider each pressure independent terminal box and its associated downstream ductwork to be a separate little system. If there is adequate static pressure and airflow available at the box inlet, the box and its associated outlets can be balanced. If adequate pressure is not available, any attempted balancing will result in an incorrect cfm, both now and later, when adequate static pressure becomes available.

Two common methods are usually available to verify that the box is delivering the correct airflow. First, most boxes have taps in the lines going to the regulator from the sensor inside the box. A manometer can be connected to these taps to read the pressure differential at the sensor. Most manufacturers provide data to convert these readings to the actual cfm. Quite often, this data is found on labels located on the side of the box. This method is used by most manufacturers to preset their boxes at the factory. Field testing, however, has proved that this method is not always accurate. The inlet duct configuration and the type of sensor will affect the signal sent to the regulator, causing erroneous readings. Through field experience, the terminal outlet readings must be totalled to verify that the terminal box is delivering the proper amount of air. This test will also aid in locating duct leakage.

To balance pressure independent boxes, first test the inlet static pressure. This static pressure test should be done with the box set at maximum airflow by setting the thermostat controlling the box down to a temperature well below the space temperature. Use the manufacturer's data to determine the required pressure drop across the box at the required cfm. Since this is only the pressure loss through the box, add the pressure required to overcome the discharge ductwork and the associated outlets. The total of these two pressures, the box pressure drop and the discharge duct system static pressure requirements, determine the static pressure at the box inlet, to allow it to be properly balanced.

With the box set at the maximum flow rate, measure the total airflow being delivered. If necessary, adjust the controller or regulator to deliver the specified cfm. Use the manufacturer's recommended procedures to do this. When the total cfm is correct, then balance the outlets in the normal manner for a low pressure system.

After the outlets are balanced, set the box for a minimum airflow, by setting the box thermostat up to a temperature higher than the space temperature. Check the total cfm and adjust the minimum setting on the box if necessary, following the manufacturer's recommendations. The individual outlets should be re-tested in the minimum position, but they should stay proportionally balanced. It is not unusual, however, for the terminals to be slightly out of balance in the minimum position. This condition, if found, should be reported, but leave the system balanced in the maximum airflow position. The terminal box and its associated outlets are now in balance and they should stay in balance as long as the inlet static pressure to the box stays within the design static pressure range given by the manufacturer.

Because of this *pressure independent* advantage, it usually is possible to balance all of the boxes on a system, even if the system pressure is low. When there is inadequate static pressure, you can try putting adjacent boxes in the minimum airflow position to increase the static pressure to the box you are testing. Usually each box and outlet on a system can be completely balanced to their maximum and minimum cfm, even though the fan isn't designed to or isn't able to deliver enough air for all the boxes to be calling for maximum cfm at the same time. This same procedure is used to balance a diversity system.

Once the VAV boxes are properly set and the outlets on each box downstream duct system are balanced, the HVAC duct system is balanced and will operate properly as long as adequate airflow and static pressure is available. At this time, the fan should be retested and necessary adjustments made. A determination of the static pressure required by the sensor can now be made and recorded. The static pressure regulator usually is set by or under the supervision of the temperature control contractor.

Diversity systems usually will require some compromise setting to operate the static pressure sensor. It will need to be high enough to satisfy the space requirements at the worst locations, which usually are the farthest from the fan; or it may be necessary to increase the setting to satisfy a certain area. But ideally, the static pressure setting should be as low as possible and still maintain the ability for the HVAC system fan to satisfy the space requirements.

(1) Non-system Powered Boxes

The VAV box controls are powered by electricity or pneumatic air from an independent source. The various pneumatic controllers usually have screws, thumbnuts, or sliding-type adjusters to set the box airflow cfm. The newer electronic boxes are being used more frequently because they have more versatility and are easily interfaced with central computer control panels. The adjustments are usually made with a potentiometer control. Many require the use of a millivoltmeter or a digital plug-in device to address the microprocessor. As the state of the art is changing rapidly, it is imperative that you have the manufacturer's data before you attempt to properly balance most of these boxes.

(2) System Powered Boxes

The term "system powered" means that the controls on the boxes are powered by the inlet static pressure and/or the velocity pressure found within the HVAC duct system. System powered boxes usually have a higher required minimum inlet static pressure. Although the higher static pressure may not be needed for the airflow quantity, it will be needed to operate the controls. Since most system powered boxes are normally open, it is possible to have a VAV box that is delivering the designed amount of air, but the controls will not operate due to low static pressure.

It is also possible to have the HVAC duct system not develop sufficient pressure to activate the box control systems upon startup, as a lack of static pressure leaves the boxes all wide open. With the boxes wide open, they deliver more air than designed to do; and with a system of wide open boxes, static pressure can not build up until the VAV boxes start controlling and close down. (Thus a "catch-22" situation develops.) The last two problems do not occur very often, but they do demonstrate how important static pressure is in any system using system powered boxes.

System powered boxes and systems should be balanced similarly to non-system powered systems detailed previously. You will need the manufacturer's data to determine where and how the adjustments are to be made.

(3) Satellite or Slave Units

Satellite boxes or *slave boxes* have been used to name two quite different types of terminals. To some terminal manufacturers, they are nothing more than linear slot diffusers which are usually connected by flex duct to a *plenum* or *octopus* on the discharge end of a terminal box. These are usually two or more round discharge outlets on the plenum, hence the term "octopus," connecting through flex ducts to the terminals or satellites. Balancing these satellites is similar to any other outlets. Once the VAV box is adjusted to deliver the correct total cfm, adjust the terminals with the dampers as provided.

Another type of satellite or slave box consists of a linear slot diffuser with a built-in automatic damper. The *slave boxes* receive a signal from the *master unit,* which is similar to the satellite unit except that it has a pressure independent volume regulator. The master unit therefore functions similar to any other pressure independent VAV device, and balancing is done in the same manner. The slave units receive a signal from the master which positions their dampers in the same position as the damper in the master. The satellite unit therefore will deliver the same airflow as the master as long as the inlet static pressure on the satellite is the same as the inlet static pressure on the master.

These slave units are actually pressure dependent (covered in the next section) and therefore the cfm being delivered will change if the inlet static pressure to the slave unit changes. The slave unit airflow quantity therefore will be different from the master cfm anytime the inlet static pressure for these units is different. Since the slave units have no balancing adjustments on them, measure the airflow being delivered and report it. The only adjustment possible is on the master unit which will affect the airflow from both the master unit as well as the connected slave units.

For this reason, system designers attempt to keep all slave units on the same duct system with the master unit, so that the inlet static pressure will be nearly the same on each terminal.

c. PRESSURE DEPENDENT VAV SYSTEMS

Pressure dependent VAV terminal boxes have no automatic volume controller to regulate the airflow as the inlet static pressure to the box changes. The cfm delivered by the box for any given condition will change at any time the inlet static pressure changes. Since the cfm delivered is dependent on the inlet static pressure furnished, the VAV boxes are considered *pressure dependent.*

These VAV boxes usually have a manual inlet balancing damper that is controlled by a thermostat. The automatic damper may or may not have a minimum position limiter to provide for adjustments of the minimum airflow. As these systems may or may not have diversity. the TAB technician must realize that every change in damper setting, either manual or automatic, is going to affect the adjacent VAV boxes in the system. Therefore, these systems constantly "hunt" while in normal operation.

(1) Non-diversity Systems

Non-diversity systems are balanced similar to a constant volume system.

(a) Put all of the boxes and the fan in a full flow condition.

(b) Test and adjust the fan close to design airflow using previously described procedures.

(c) With the duct balancing dampers provided, set each zone or branch to the proper cfm.

(d) Adjust the inlet dampers to each box to obtain the design total cfm from each.

(e) Balance the outlets downstream from each box to design cfm.

(f) Retest and adjust the fan cfm for final maximum readings.

(g) Test and record the operating static pressure at the sensor that controls the HVAC unit fan, if provided.

(h) Verify that the static pressure controller for the fan is operating and adjusted.

(i) If a minimum cfm is specified, put the system into a minimum flow mode. Verify that the fan is maintaining a constant static pressure if control is provided.

(j) Adjust each VAV box to deliver the correct minimum airflow.

(k) Test and record the values of the downstream terminals with minimum airflow.

Remember that any further retest will have to be made with the system in the exact same condition if repeatability of readings is to be expected.

(2) Diversity Systems

Diversity systems can be the most difficult VAV systems to balance satisfactorily. Any procedure used will be a compromise, and shortcomings will appear somewhere in the system under certain operating conditions. The TAB contractor can expect that some fine tuning will be necessary after the initial TAB work is complete.

To eliminate a lot of possible misunderstandings later, it is most important that an agenda with the proposed balancing procedures be submitted and approved by the system designer or authorized persons before the TAB work is started. This procedure is recommended for all jobs, but it is essential on jobs with these particular systems. Everyone must be aware before you start, just what you are going to do and the results that they can expect.

The objective of any HVAC system is to provide adequate airflow as needed to maintain the space requirements at any time. The balancing procedures usually will have to be tailored to each particular job, but the basic steps are listed below.

(a) Put the system into a mode where it will require approximately the same airflow as the maximum HVAC fan design airflow by placing the required number of VAV boxes in a minimum airflow position. Stagger the boxes set at minimum airflow so they all are not in one location or on one zone.

(b) Test and adjust the fan to deliver the design cfm, with the airflow control device set at "maximum." This device may be an inlet damper, discharge damper, variable speed drive, or variable speed motor.

(c) After the fan has been set to deliver the system design cfm, set all of the VAV boxes to *full air flow.*

(d) Starting at the fan end of the system, go to each VAV box and adjust the inlet damper to provide the correct cfm for that box, and then balance its downstream terminal outlets. Do this to each VAV box that has the correct airflow available with the system in this condition. These VAV boxes and their associated outlets are now balanced. Adjust the minimum airflow at this time, if required.

(e) The remaining VAV boxes will still be low on cfm due to the diversity in the system. Starting with the boxes that are closest to the design airflow, systematically place adjacent boxes to minimum airflow until just enough air is available to balance the VAV box you are testing. *Do not close* the inlet damper on this box. Leave it *wide open.* Balance the downstream terminal outlets and then set the box minimum cfm, if provided. Use these same procedures on each remaining box.

There may be some boxes that you can not readily obtain the required cfm. In this case, proportionally balance the outlets to as high a percentage of their design cfm as possible. Be sure to leave the box inlet damper *open.*

You essentially have balanced the boxes and terminals so that during the normal operation of the system, when more static pressure will be available, the required cfm should be delivered.

(f) You can now test and record the operating static pressure at the sensor (if provided). However, some experimentation will probably be necessary with the final setting. On these systems, the operating pressure usually will have to be increased some, to assure that all of the VAV boxes will have an adequate airflow available during normal operation.

If complaints are received later, you may have to do some testing when the system is in normal use at the time of day the complaints arise to determine what steps may be necessary to correct the problem.

Although they are not being installed currently, some of the early variations of VAV boxes used a system-powered, pressure independent, maximum controller and a thermostatically controlled inlet damper for minimum control. Since the minimum was pressure dependent, balancing procedures would be similar to those just described. Two characteristics the TAB technician should be aware of regarding these boxes is as follows.

(a) As the thermostat begins to close the inlet damper, the pressure independent "maximum" damper just downstream will open up trying to maintain the same cfm. There will be very little reduction in total cfm until after the "maximum" damper is fully open and the "minimum" damper continues to close.

(b) If there is enough inlet static pressure, the cfm with the damper minimum position may not be much less, if any, than in the maximum position.

(c) Most of these boxes were constructed in such a manner as to make access for adjustment of the minimum cfm very difficult.

d. FAN POWERED VAV BOXES

VAV boxes that contain individual supply air fans are becoming popular in many areas. The variations of operating sequences are numerous and it is imperative that the manufacturer's data be reviewed before proceeding with the TAB work. The actual balancing procedures may be included. Otherwise, you will have to develop your own TAB procedures based on a possible combination of procedures previously described.

e. INDUCTION VAV BOXES

Induction VAV boxes use primary air from a central fan system to create a low pressure area within the box by discharging the primary air at high velocities into a plenum. This low pressure area usually is separated from a ceiling return air plenum by an automatic damper. The induced air from the ceiling is mixed with the primary air, so that the actual cfm being discharged from the box is considerably more than the primary air cfm. Most of these induction boxes are for VAV operation, but they still are available for constant volume.

Study the manufacturer's data before attempting to do the TAB work, because many operating sequences are available. Balancing will consist of setting the primary airflow, both maximum and minimum. The discharge air is a total of the primary air and the induced air. Some boxes have adjustments for the induction damper setting. After the box is set, the downstream air outlets can be balanced in the conventional manner.

f. COMBINATION SYSTEMS

To complicate matters for the TAB technician, many system designers are now mixing pressure independent VAV boxes and pressure dependent VAV boxes on the same system, either with or without diversity. Balancing procedures will have to be tailored to each job, but its generally better to balance the pressure independent boxes first, since once they are balanced, they will not be affected by changing static pressures as the rest of the system is being balanced. You can usually build up an adequate system static pressure to accomplish this, by setting the boxes that are adjacent to the ones that you are working on to their minimum airflow position. If a system has many pressure dependent boxes, they may consume most of the system airflow and static pressure on the initial system start-up, since they will be wide open. Either set some of these boxes to a minimum airflow position or partially close the inlet dampers on some boxes to build up the static pressure in the system. After setting all of the pressure independent VAV boxes, use the procedures detailed previously for pressure dependent systems and balance the downstream air outlets in the conventional manner.

3. Dual Duct Systems

Dual duct systems use both a hot air duct and a cold air duct to supply air to a mixing box. The mixing box may operate in a constant air volume mode or in a variable air volume mode. They are usually pressure independent, but they may be either system powered or have external control systems. Dual duct systems will fall into several categories for testing: (1) high pressure, constant volume, (2) variable air volume and (3) low pressure systems.

a. CONSTANT VOLUME SYSTEMS

For many years, high pressure, dual duct systems were the most common type of dual duct systems. The mixing box has a thermostatically controlled mixing damper to satisfy the space temperature requirements. A mixture of the hot and cold air is then controlled by a system powered volume damper (also called a pressure reducing valve) that maintains a constant airflow to the space. Newer versions use pneumatic or electric regulators, similar to those found on VAV boxes to control the total cfm and they are usually non system powered.

The balancing procedures are as follows:

(1) Adjust the supply air fan using similar procedures for any conventional constant volume system. It is common practice to set all the mixing boxes to their full cold airflow position for setting the fan volume, but first verify that the cooling coil is designed to handle the same cfm as the HVAC duct system. It may be designed for less cfm creating a diversity that will require some mixing boxes to be set in a heating position for a total system flow test.

(2) An initial static pressure check should be made at the end of the longest duct run to ascertain that the required minimum static pressure is available. The minimum static pressure at the inlet to the box would be the manufacturer's published static pressure drop across the box added to the static pressure resistance of the downstream duct and terminal air outlets. If the boxes have been preset, a low reading at this location usually will indicate a need to increase the airflow from the fan. If adequate static pressure is available, proceed with the system TAB work.

(3) Pitot tube readings of the airflow quantities should be taken in both ducts (hot and cold) and totaled, even if all the mixing boxes are set on cold. There is usually some air leaking

through the hot deck of the HVAC unit, and quite often some boxes are hooked up backwards (in reverse from hot to cold). A significant flow of air through the hot duct will indicate that this is probably the case. A low airflow reading combined with low static pressure at the farthest box inlet will indicate a need to increase the cfm furnished by the fan. However, low cfm combined with high static pressure can indicate that the boxes may be set at an airflow rate that is too low which holds back the air in the system and develops a higher static pressure.

Testing and adjusting of the constant volume mixing boxes consists of the following:

(1) Verify the hot and cold operation by moving the thermostat adjustment back and forth and observing the temperature and volume change of the air. If the air temperature is reversed, the box duct connections or damper linkage will have to be changed.

(2) Verify that there is an adequate inlet static pressure before making volume adjustments. If adequate static pressure is not available in either the hot or cold position, put the box into the opposite position where enough pressure is available. Perform the following Steps (3) and (4) while the system is in this condition, which will insure adequate static pressure at the boxes. The boxes and terminals will then be in proper balance and will continue to be whenever adequate system pressure is available.

(3) Adjust the mixing box for the correct total cfm. This adjustment may be made by using a quadrant handle, a hex wrench or spring tensioning. The total cfm from the box may be obtained by making Pitot tube traverse readings, totaling the outlet cfm's, or if provided, by pressure tap readings at the box.

(4) The terminals can then be balanced conventionally.

(5) After the system has been satisfactorily balanced, recheck the static pressure at the end of the duct run. If it is much higher than required, the fan should be slowed down to decrease noise, mechanical wear and operating costs.

b. VARIABLE AIR VOLUME SYSTEMS

Variable air volume (VAV), dual duct mixing boxes are likely to be non system powered. As the box airflow changes from cold to hot, the quantity of the hot air discharged is increased as the cold air is reduced or shut off. The available sequences are numerous and it is imperative that the TAB technician review the operating sequence for the individual box being balanced.

(1) Testing and adjusting is similar to a dual duct constant volume system except that VAV capability is incorporated and will have to be taken into account.

(2) The mixing box adjustments will now incorporate a maximum and minimum adjustment and each need to be set instead of a single total volume adjustment.

(3) Outlets will still be adjusted conventionally after the boxes have been set for the correct airflow.

(4) It is not possible to cover all of the various operating sequences here, as each may require a different balancing procedure. You must obtain the manufacturer's complete data and follow the outlined operating sequence. If the manufacturer does not provide the proper procedure for adjusting the boxes, you will have to develop one using a combination of procedures detailed in this manual for VAV *and* dual duct boxes.

New innovations are constantly appearing, so do not hesitate to ask the manufacturer for data. Just as an example, some sequences allow the cold duct to be reduced to from 50% to 10% of the design cfm before the hot duct begins to open. Others start opening the hot duct as soon as the cold duct begins to close. Most sequences call for the maximum hot cfm to be lower than the maximum cold cfm, but the sequences and the controls can vary, so that the manufacturer's data must be obtained.

c. LOW PRESSURE SYSTEMS

Low pressure dual duct systems use no airflow controllers except for a possible zone static pressure regulator and control damper. They are pressure dependent at the terminals so therefore the cfm will vary with normal operation. Use the following balancing procedure:

(1) Set all the branch thermostats into a cooling position so that all the cold duct dampers are open and the hot duct dampers are closed. If static pressure controlled zone dampers are provided, verify that they are open.

(2) Test the fan and perform the Pitot tube traverses necessary to determine the total airflow.

(3) Make adjustments to the fan as required to obtain the correct system total cfm.

(4) If manual zone dampers are provided, perform Pitot tube traverses of all zones and adjust the balancing dampers to obtain the cor-

rect airflow in each zone. If static pressure regulated automatic dampers are provided, they should be set now to deliver the correct airflow.

(5) Proceed to balance the branch ducts and outlets. NOTE: Some systems use branch duct manual balancing dampers in both the hot and cold duct before they combine. Others use only one manual balancing damper located in the common duct after the hot and cold ducts are connected.

(6) If static pressure regulated automatic zone dampers have been utilized, they may need re-adjusting to obtain the desired cfm.

(7) After the system has been properly balanced in the cold position, set the thermostats up so that the entire system is on heating.

(8) Balance the zones as described in Step (4).

(9) If separate balancing dampers are provided in the hot branch ducts, adjust them for the proper cfm. If they are not, do not change the branch duct damper from their setting when balanced in cold position as this takes preference.

(10) Test the outlets. If the cfm is not reasonably close to what it was in the cold position, determine why it is not. Look for problems with the operation of the hot and cold mixing dampers. Verify that hot air is being delivered when the thermostat calls for heat and cold air when it calls for cold. Check for any other common malfunctions.

(11) These systems usually require several passes to obtain a satisfactory balance. After verifying satisfactory airflows on both hot and cold positions, record the final readings.

4. Induction Unit Systems

Induction unit systems use high or medium pressure fans to supply primary air to the induction units. Since they usually are located around the perimeter of the building, it is common practice to run many risers up and down the building and feed the units from a common header duct from the risers. As these systems use high or medium pressure, take extra precautions to avoid building up excessive system pressures and causing damage. Check to see that the induction unit dampers, as well as the system dampers, are wide open before starting the HVAC unit primary air fan.

Airflow readings at the induction units are taken by reading the static pressure at one of the nozzles and comparing it to the manufacturer's published data. The design static pressure and cfm will be shown on the manufacturer's submittal data for the various size units on the job. If the system is in an old building and this data is not available, you may have to contact the induction unit manufacturer. If you can obtain a given static pressure for any given cfm for a particular model, you can calculate the required static pressure for a different cfm by using Equation 4-3 from Chapter IV shown below:

$$\frac{SP_2}{SP_1} = \left(\frac{cfm_2}{cfm_1}\right)^2$$

Example 8C

You have obtained data that this induction unit delivers 150 cfm at a nozzle S.P. of 1.90 in.w.g. Find the nozzle S.P. required to adjust this induction unit to deliver 175 cfm.

Solution

$$SP_2 = SP_1 \left(\frac{cfm_2}{cfm_1}\right)^2$$

$$SP_2 = 1.90 \left(\frac{175}{150}\right)^2$$

$$SP_2 = 2.59 \text{ in.w.g.}$$

So if you adjust the damper on the induction unit to obtain 2.59 in.w.g. at the nozzle, the unit should be delivering 175 cfm.

a. BALANCING PROCEDURES

(a) Adjust the primary air fan using previously described methods for constant volume systems. With a new or wide open system, allow for a 10 percent loss in airflow while balancing.

(b) Using a dry type manometer, hose and a probe (which can be a length of metal tubing of the correct diameter), read the static pressure at the nozzle. By comparing it to the unit charts or data, determine the airflow being delivered. By taking readings of the top and bottom unit on each riser, you can determine which riser dampers will need to be throttled. Assuming that most of the risers are typical, the riser with the higher pressure should be throttled to build up the lower pressure ones, until all the riser static pressures are close to being equal. If the required airflow of each riser varies considerably, you may have to take Pitot tube traverses of the risers and adjust them in this manner.

(c) Proceed on your first pass to test and adjust each induction unit working from the fan outward. If this is a new wide open system, adjust the units closest to the fan to about 10 percent to 15 percent under design, anticipating some cfm build-up as the rest of the system is adjusted.

(d) On the first pass, you should find any blockages that may be in the system. If you find units with extremely low or zero static pressure, that usually indicates a duct obstruction. Since so many smaller size risers are used, it is quite common to have debris fall down into them while the building is under construction. The debris will usually stop in a transition (reducer) or elbow. Unfortunately, these risers are not usually very accessible and repairs to the system by the installing contractor can be time consuming.

(e) You will need to make two to three "adjusting" passes around the system. The final balancing pass should be just a reading pass of the units to record the results.

(f) As with all HVAC duct systems, the fan speed should be set so that the fan is delivering the correct cfm to the induction units located at the end of the longest duct run with their dampers wide open. The system balancing dampers should not be closed down to where there is excessive static pressure everywhere in the system. This uses excess horsepower, causes more mechanical wear, and creates noise problems. Wherever there is a partially closed damper, noise can be produced; and the higher the pressures involved, the higher the noise levels will be.

5. Systems with Hoods

a. EXHAUST AIR HOODS

Systems with hoods are found in various types of ventilating systems. Kitchen hoods in restaurants and institutions are the type most frequently encountered. Most kitchen hoods are designed for a face velocity of about 100 feet per minute at the hood entrance. *Capture velocity* is necessary to insure entrainment of steam and grease laden vapors from the equipment below. Some municipalities have ordinances setting minimum requirements for face velocities at kitchen hoods, and further require that an authorized representative be present at the time of the balancing work.

b. MAKE-UP AIR SYSTEMS

Kitchen make-up air systems must be in operation when the balancing takes place. Sometimes make-up is achieved by means of relief grilles from adjoining areas. The hot wire anemometer is a good instrument for measuring these low face velocities. Some swinging vane anemometers (velometer) can be used at velocities under 100 fpm using the low flow probe. A Pitot tube used with a micromanometer also can be used. When making a Pitot tube traverse of the duct from the hood, be sure to correct for air density, if required, due to temperature.

Most kitchen hood exhaust ducts are made of heavy gauge metal, and are covered with a thick fire resistant insulation. A Pitot tube traverse duct is the most accurate way to test, but the test holes will need to be plugged with moisture tight, fire resistant metal plugs or caps; and often, holes are *not allowed.* Avoid putting holes in the bottom of the duct where moisture can accumulate and/or leak out. If possible, put the test holes in the side of a riser. *Never use plastic or rubber test plugs in a kitchen exhaust duct.* Also, be aware that even if you have the correct cfm at the Pitot tube traverse, you still may not have the hood face velocity required by local ordinances. In this case you may have to speed up the fan, if the system designer approves.

Velocity readings across grease filters are not usually reliable. Accurate free area correction data is not usually available and it would be influenced by the condition of the filters.

c. FUME HOODS

Fume hoods are frequently found in laboratory buildings and hospitals. As experiments are conducted in confined areas, the hoods are designed to prevent the escape of toxic or noxious fumes. Make-up air must be provided from the HVAC system or from a separate system that some hoods have built-in to minimize the loss of conditioned air from the laboratory. These systems also must be accurately balanced.

When toxic experiments are to be performed by the occupants, a smoke candle test should be made by the TAB technician to ensure that vapors do not escape. Often the whole room is designed to be under a negative pressure, so the room also should be smoke tested. Some of the more sophisticated hoods have a built-in exhaust fan working in series with the system exhaust fan.

Most fume hoods are designed to exhaust the same amount of air with their door open or closed. Face velocity testing should be done with the door wide open. Readings are best taken with a hot wire anemometer and should be taken in equal area rectangles similar to a traverse. Be sure to keep your body well out of the airstream. Even then, air may enter the hood from other openings and not be accounted for. This is why a Pitot tube traverse of the exhaust duct is the preferred method for best accuracy. Here again, where toxic, noxious or corrosive fumes will be introduced, the test plug should be made of a compatible material.

A hand held smoke generator should be used to insure that the entire face of the hood is drawing air into the hood. Be sure that air is not swirling and escaping back out of the hood. A 30-second smoke candle should be set off in the hood also. This will insure that the hood will contain the exhaust fumes without escaping back into the space.

Fume hoods are used mostly in laboratories. When balancing laboratories, the TAB technician should carefully study the drawings, noting the cfm and pressure differentials between different areas and rooms. Balancing is critical in these areas and will usually require more precise readings and adjustments to obtain the correct positive and negative pressure relationship between spaces. Quite often, you may have the correct cfm readings, but the pressure relationships may be wrong. This is usually due to the inaccuracies of field airflow readings. You will usually have to make some adjustments to the airflows to obtain the correct pressures. If there is any question, you should consult the system designer.

d. INDUSTRIAL EXHAUST HOODS AND EQUIPMENT

Industrial exhaust systems with hoods fall into two categories. One group, similar in many respects to laboratory fume hoods, is used in conjunction with dip tanks and plating tanks. Exhaust hoods are often placed at one end of the tank and make-up hoods are placed at the opposite side. This permits vapors to be swept from the tank surface but still leaves the top open for overhead handling equipment. Often, an exhaust duct will be connected direct to a piece of equipment with no external hood. Other times, hoods may be used just to remove heat from equipment. Heat recovery systems are also being used more frequently. Here again, make-up air becomes critical and air density must be corrected in calculations.

The balancing procedure is still basically the same as any normal exhaust system. The differences are mainly in how to read the various inlet openings. Inlet velocities can vary from very low to very high. A Pitot tube traverse is the preferred method where possible. If an inlet must be read, you will probably have to obtain the free area opening yourself, by measuring and calculating. Quite often this will not be possible due to irregular shapes and/or obstructions.

A hot wire anemometer is a very valuable instrument for this type of work as the probe is small enough to get into obstructed places. But here again, consult the equipment manufacturer's data, as they often give the procedures for setting up and testing their equipment.

e. MATERIAL HANDLING EXHAUST SYSTEMS

A second group of factory exhaust systems is used to remove and convey solid materials. Sawdust, wood chips, paper trimmings, etc., are transported at high velocities through these exhaust systems. These systems must be balanced so that velocities do not fall below predetermined transport velocities, at which point the materials would drop out.

Balancing of these systems is done with blast gates which are installed in lieu of dampers and are used to temporarily shut off unused branches. In addition to velocity readings, static pressure readings of the pressure differential between the room and the hood should be recorded in a convenient reference point at each hood or intake device. This will permit easy future checks designed to spot any deviation in exhaust volumes from original volumes.

CHAPTER 9 HYDRONIC SYSTEM TAB PROCEDURES

Now that the HVAC air systems are in balance and all preliminary paperwork for the hydronic systems has been completed, the actual hydronic system balancing procedures may begin! As no two hydronic systems are exactly alike, an attempt to develop step by step procedures for each system would be very difficult, if not impossible. This chapter is separated into four sections that form a guideline for the testing, adjusting and balancing of any HVAC hydronic system encountered.

(1) Methods of Flow Measurement: a review of various methods used for the determination of flow during hydronic TAB work;

(2) Equipment TAB Work: a review of recommended test and balance procedures for various types of HVAC equipment;

(3) Specific Piping Applications: recommended test and balance procedures for specific piping applications; and

(4) Specific System Applications: recommended test and balance procedures for specific piping system applications.

A METHODS OF FLOW MEASUREMENT

The best possible method for flow measurement of hydronic systems cannot be determined without reviewing the systems. There are five basic methods available for measuring the flow quantity in a piping system: 1) with flow meters, 2) with calibrated balancing valves, 3) using the equipment pressure loss, 4) by heat transfer, and 5) using pump curves.

1. Flow Meter

A flow meter usually is deemed to be the most reliable method for measuring the system flow. Flow meters usually are permanently installed in the hydronic piping system and are used for the measurement and adjustment of flow to pumps, to primary heat exchange equipment, at each zone, and at terminal units. Flow meters such as the venturi, orifice, and annular types, require the use of a differential pressure gauge and flow charts provided by the manufacturer to calculate the system flow. Always verify that installation of the flow meter is in accordance with recommended practices given by the manufacturer. There must be adequate amounts of straight sections of piping upstream and downstream from the flow meter to prevent erroneous readings affecting final system balance.

NOTE: Verify that the pressure units of the differential pressure gauge and the pressure units found on the flow charts provided by the manufacturer are identical. If pressure units are not the same, (i.e. psi, in.w.g., ft.w.g.) pressure conversions will be required.

2. Combination Valve/Flow Meter

A calibrated balancing valve is another very reliable device used for measuring system flow. Two types of these combination valves are being used, self adjusting and field adjusted.

A self-adjusting valve/flow meter utilizes internal mechanisms that constantly change internal orifice openings to compensate for varying system differential pressures while maintaining a pre-set flow rate. No external adjustment is available with this device. Pressure taps, providing measurement of valve differential pressure, allow the TAB technician to measure the system flow.

Calibrated balancing valve/flow meters are field adjustable devices. Pressure loss of the valve is measured similar to that of a flow meter. A chart or graph, provided by the valve manufacturer, indicates actual flow rates at various valve positions and differential pressures. Unlike a flow meter, the flow coefficient of a calibrated balancing valve/flow meter changes with adjustment of the valve. Always be aware of the actual valve position when calculating the system flow.

3. Equipment Pressure Loss

Actual system flow rates may be established by using HVAC equipment pressure loss calculations, provided the following two items are available:

(a) certified data from the equipment manufacturer indicating rated flow and pressure losses;

(b) and an accurate means for determining the actual equipment pressure losses.

When the design criteria of the equipment and the actual pressure loss is known, the flow rate may be calculated by using Equation 5-9 from Chapter V:

$$gpm_2 = gpm_1 \sqrt{\frac{\Delta P_2}{\Delta P_1}}$$

Equation 5-9 may be used to determine the flow using any type of pressure units (psi, ft.w.g., in.w.g.). However, it is imperative that the pressure units of the equipment rating and the measured equipment pressure loss are the same. The flow rates of *all* items of equipment must then be totalled to obtain the system flow rate.

4. Heat Transfer

Actual flow rates may be established at heating and cooling terminal units by using measured heat transfer data and the proper equations. Although the equations used are slightly different, both methods are based upon the First Law of Thermodynamics (heat transfer) which simplified means: **HEAT LOSS = HEAT GAIN.** Each method determines the total heat transfer rate of the terminal unit at the time of testing, and then the flow rate is calculated based upon the fluid heat transfer rate (water temperature difference).

a. HOT WATER TERMINAL UNITS (SENSIBLE HEAT)

Coil total heat transfer (in Btuh) may be calculated when the entering and leaving dry bulb air temperatures and the coil airflow rate are known using Equation 1-7 from Chapter I: $Q = 1.08 \times cfm \times \Delta t$ (or) Btuh (Sensible) = (Coil cfm) × (1.08) × (leaving DB − entering DB). Coil gpm may be calculated when the actual heat transfer rate (Btuh) and the water temperature loss of coil are known using Equation 1-11:

$$Q = 500 \times gpm \times \Delta t; \; gpm = \frac{Q}{500 \times \Delta t}$$

(or)

$$gpm = \frac{\text{Coil Btuh}}{500 \times \text{Water } \Delta t}$$

b. CHILLED WATER TERMINAL UNITS (TOTAL HEAT)

Coil total heat transfer in terms of Btuh may be calculated when the entering and leaving wet bulb air temperatures and the coil airflow rate are known. Use of the following equations requires the understanding of Enthalpy (total heat). Enthalpy is the actual heat content of air, and may be determined directly by the wet bulb temperature using Equation 1-10 from Chapter 1. Table 9-1 shows actual enthalpy values corresponding to wet bulb temperatures (use the values in the last column). The values also can be found on psychrometric charts.

$Q\ (\text{Total}) = 4.5 \times cfm \times \Delta h$;
where Δh is the enthalpy difference.

The actual coil gpm may then be calculated in the same manner as described with hot water applications using Equation 1-11:

$$gpm = \frac{Q}{500 \times \Delta t}$$

Example 9A

The hot water coil of the office HVAC unit has an airflow rate of 4500 cfm with a temperature rise of 36°F. Water is entering the coil at 195°F and leaving at 175°F. Calculate the hot water flow at the HVAC unit.

Solution

Using Equation 1-7:

$$\begin{aligned} Q\ (\text{Sens.}) &= 1.08 \times cfm \times \Delta t \\ &= 1.08 \times 4500 \times 36 \end{aligned}$$

$Q\ (\text{Sens.}) = 174{,}960$ Btuh

Using Equation 1-11:

$$gpm = \frac{Q}{500 \times \Delta t} = \frac{174{,}960}{500 \times (195 - 175)}$$

$gpm = 17.50$ (hot water)

Example 9B

The cooling coil of the same HVAC unit has a change in wet bulb temperature of the airflow from 78°F to 56°F at the same 4500 cfm. If the chilled water is entering at 46°F and leaving at 56°F, calculate the chilled water flow at the HVAC unit.

Solution

Using Equation 1-10 and Table 9-1:

$$\begin{aligned} Q\ (\text{Total}) &= 4.5 \times cfm \times \Delta h \\ &= 4.5 \times 4500 \times (41.58 - 23.84) \end{aligned}$$

$Q\ (\text{Total}) = 359{,}235$ Btuh

Using Equation 1-11:

$$gpm = \frac{Q}{500 \times \Delta t} = \frac{359{,}235}{500 \times (56 - 46)}$$

$gpm = 71.85$ (chilled water)

Table 9-1 PROPERTIES OF MIXTURES OF AIR AND SATURATED WATER VAPOR
(Table Based on Barometric Pressure of 29.92 Inches.)

TEMP. F	HUMIDITY RATIO - WEIGHT OF SATURATED VAPOR PER POUND OF DRY AIR		ENTHALPY OF 1 LB. OF DRY AIR ABOVE O F IN BTU	ENTHALPY OF (SATU-RATED) VAPOR, BTU	ENTHALPY OF MIXTURE OF 1 LB. OF DRY AIR WITH VAPOR TO SATURATE IT IN BTU
	POUNDS	GRAINS			
0	0.000787	5.51	0.0	0.835	0.835
2	.000874	6.12	0.480	0.928	1.408
4	.000969	6.78	0.961	1.030	1.991
6	.001074	7.52	1.441	1.142	2.583
8	.001189	8.32	1.922	1.266	3.188
10	.001315	9.21	2.402	1.401	3.803
12	.001454	10.18	2.882	1.550	4.432
14	.001606	11.24	3.363	1.713	5.076
16	.001772	12.40	3.843	1.892	5.735
18	.001953	13.67	4.324	2.088	6.412
20	.002152	15.06	4.804	2.302	7.106
22	.002369	16.58	5.284	2.536	7.820
24	.002606	18.24	5.765	2.792	8.557
26	.002865	20.06	6.245	3.072	9.317
28	.003147	22.03	6.726	3.377	10.103
30	.003454	24.18	7.206	3.709	10.915
32	.003788	26.52	7.686	4.072	11.758
33	.003944	27.61	7.927	4.242	12.169
34	.004107	28.75	8.167	4.418	12.585
35	.004275	29.93	8.407	4.601	13.008
36	.004450	31.15	8.647	4.791	13.438
37	.004631	32.42	8.887	4.987	13.874
38	.004818	33.73	9.128	5.191	14.319
39	.005012	35.08	9.368	5.403	14.771
40	.005213	36.49	9.608	5.662	15.230
41	.005421	37.95	9.848	5.849	15.697
42	.005638	39.47	10.088	6.084	16.172
43	.005860	41.02	10.329	6.328	16.657
44	.006091	42.64	10.569	6.580	17.149
45	.00633	44.31	10.809	6.841	17.650
46	.00658	46.06	11.049	7.112	18.161
47	.00684	47.88	11.289	7.391	18.680
48	.00710	49.70	11.530	7.681	19.211
49	.00737	51.59	11.770	7.981	19.751
50	.00766	53.62	12.010	8.291	20.301
51	.00795	55.65	12.250	8.612	20.862
52	.00826	57.82	12.491	8.945	21.436
53	.00857	59.99	12.731	9.289	22.020
54	.00889	62.23	12.971	9.644	22.615
55	.00923	64.61	13.211	10.01	23.22
56	.00958	67.06	13.452	10.39	23.84
57	.00993	69.51	13.692	10.79	24.48
58	.01030	72.10	13.932	11.19	25.12
59	.01069	74.83	14.172	11.61	25.78
60	.01108	77.56	14.413	12.05	26.46
61	.01149	80.43	14.653	12.50	27.15
62	.01191	83.37	14.893	12.96	27.85
63	.01235	86.45	15.134	13.44	28.57
64	.01280	89.60	15.374	13.94	29.31
65	.01326	92.82	15.614	14.45	30.06
66	.01374	96.18	15.855	14.98	30.83
67	.01424	99.68	16.095	15.53	31.62
68	.01475	103.3	16.335	16.09	32.42
69	.01528	107.0	16.576	16.67	33.25
70	.01582	110.7	16.816	17.27	34.09
71	.01639	114.7	17.056	17.89	34.95
72	.01697	118.8	17.297	18.53	35.83
73	.01757	123.0	17.537	19.20	36.74
74	.01819	127.3	17.778	19.88	37.66
75	.01882	131.7	18.018	20.59	38.61
76	.01948	136.4	18.259	21.31	39.57
77	.02016	141.1	18.499	22.07	40.57
78	.02086	146.0	18.740	22.84	41.58
79	.02158	151.1	18.980	23.64	42.62
80	.02233	156.3	19.221	24.47	43.69
81	.02310	161.7	19.461	25.32	44.78
82	.02389	167.2	19.702	26.20	45.90
83	.02471	173.0	19.942	27.10	47.04
84	.02555	178.9	20.183	28.04	48.22
85	.02642	184.9	20.423	29.01	49.43
86	.02731	191.2	20.663	30.00	50.66
87	.02824	197.7	20.904	31.03	51.93
88	.02919	204.3	21.144	32.09	53.23
89	.03017	211.2	21.385	33.18	54.56
90	.03118	218.3	21.625	34.31	55.93
91	.03223	225.6	21.865	35.47	57.33
92	.03330	233.1	22.106	36.67	58.78
93	.03441	240.9	22.346	37.90	60.25
94	.03556	248.9	22.587	39.18	61.77
95	.03673	257.1	22.827	40.49	63.32
96	.03795	265.7	23.068	41.85	64.92
97	.03920	274.4	23.308	43.24	66.55
98	.04049	283.4	23.548	44.68	68.23
99	.04182	292.7	23.789	46.17	69.96
100	.04319	302.3	24.029	47.70	71.73
101	.04460	312.2	24.270	49.28	73.55
102	.04606	322.4	24.510	50.91	75.42
103	.04756	332.9	24.751	52.59	77.34
104	.04911	343.8	24.991	54.32	79.31
105	.0507	355.	25.232	56.11	81.34
106	.0523	366.	25.472	57.95	83.42
107	.0540	378.	25.713	59.85	85.56
108	.0558	391.	25.953	61.80	87.76
109	.0576	403.	26.194	63.82	90.03
110	.0594	416.	26.434	65.91	92.34
111	.0614	430.	26.675	68.05	94.72
112	.0633	443.	26.915	70.27	97.18
113	.0654	458.	27.156	72.55	99.71
114	.0675	473.	27.397	74.91	102.31
115	.0696	487.	27.637	77.34	104.98
116	.0719	503.	27.878	79.85	107.73
117	.0742	519.	28.119	82.43	110.55
118	.0765	536.	28.359	85.10	113.46
119	.0790	553.	28.600	87.86	116.46
120	.0815	570.	28.841	90.70	119.54
125	.0954	668.	30.044	106.4	136.44
130	.1116	781.	31.248	124.7	155.9
135	.1308	916.	32.452	146.4	178.9
140	.1534	1074.	33.655	172.0	205.7
145	.1803	1262.	34.859	202.5	237.4
150	.2125	1488.	36.063	239.2	275.3
155	.2514	1760.	37.267	283.5	320.8
160	.2990	2093.	38.472	337.8	376.3
165	.3581	2507.	39.677	405.3	445.0
170	.4327	3028.9	40.882	490.6	531.5
175	.5292	3704.4	42.087	601.1	643.2
180	.6578	4604.6	43.292	748.5	791.8
185	.8363	5854.1	44.498	953.2	997.7
190	1.099	7693.	45.704	1255.0	1301.0
200	2.295	16065.	48.119	2629.0	2677.0

5. Pump Curves

Flow may be established at circulating pumps through a series of differential pressure testing and pump curve analysis. Details of this testing procedure will be covered in the next section of this chapter. When circulating pumps are installed in series with any item of HVAC equipment, the equipment flow rate may be assumed to be equal to that of the pump.

B EQUIPMENT TAB WORK

This section reviews the basic testing, adjusting and balancing methods used for the hydronic portion of HVAC equipment. Methods for the determination of flow rates are listed in order of the highest reliability to the lowest reliability.

1. Pumps

Flow rates may be established at centrifugal pumps by using the following methods:

(a) flow meters or calibrated balancing valves,

(b) pump curves, or

(c) pressure losses of the connected equipment.

a. FLOW METERS OR CALIBRATED BALANCING VALVES

A properly installed flow meter or calibrated balancing valve serving a circulating pump is the fastest, easiest, and most reliable method for determining the pump flow rate. Although a flow meter reading is considered the most reliable method of measuring pump flow, a pump curve analysis should be performed as outlined in the next section to substantiate the flow meter readings.

b. PUMP CURVES

The following procedure outlines performance testing of a centrifugal pump including pump curve analysis used for the determination of flow:

(1) With the pump(s) off, observe and record the system static pressure.

(2) With the pump(s) running, *fully close* (slowly) the service (or balancing) valve located on the discharge of the pump. Record the pump discharge and suction pressures at the pump gauge taps located on, or as close as possible to, the pump volute housing. Pressure readings should be taken with one test gauge located at the same elevation (a good practice is to use a gauge manifold located at the pump base). Record the motor current and voltage. When all data is recorded, slowly open the discharge valve to the *full open position.*

Testing of pump operating pressures with the discharge valve fully closed is known as *NO-FLOW TESTING.* No-Flow testing is used to determine the actual impeller diameter and pump operating curve by establishing the pump operating head at a zero gpm flow rate. To determine the pump No-Flow head (in ft.w.g.), multiply the differential pressure (in psi units) of the pump by 2.31. Plot the intersection of the No-Flow head and the zero gpm line of the pump curve provided by the manufacturer and compare. Does the No-Flow head determined by testing fall at or close to the beginning of the design curve? If it does, proceed to the next step entitled "FULL-FLOW TESTING". If it does not, plot a new curve proportional to the other curves on the chart. Extreme care should be taken with No-Flow testing since an error in gauge reading, calculation of head, or interpolation of curve may result in confusion and additional TAB work.

(3) *FULL FLOW TESTING* is similar in procedure to *NO-FLOW TESTING.* With the discharge valve in the full open position, record the pump suction and discharge pressures, and motor current and voltage. To determine Full-Flow head (ft.w.g.), multiply the pump differential pressure (psi) by 2.31. Plot the intersection of the Full-Flow head and the established pump capacity curve to determine actual pump flow rate.

(4) If the actual flow rate is greater than acceptable tolerances, determine what differential pressure would be required to achieve a pump flow rate 10 percent above design, adjust the flow accordingly, by using the balancing valve located in the pump discharge piping.

(5) When the pump capacity is within acceptable tolerances, system testing and balancing may be initiated.

After all system TAB work has been performed, record the final pump operating pressures and motor current and voltage. Calculate the final pump operating head pressure and brake-horsepower. Plot the final pump operating point on the proper capacity curve and record the final system flow rate.

Refer to the NEBB Pump Test Report Form (TAB 18-83) located in Section VIII of the NEBB "Procedural Standards for Testing, Adjusting and Balancing of Environmental Systems" for the proper format of the test data.

c. PRESSURE LOSS OF CONNECTED EQUIPMENT

When flow meters are not available, and the circulating pump is piped in series with a piece of equipment with a known pressure loss, the actual flow rate may be established by equipment pressure loss testing. An example of this application would be an open condenser water system containing one pump, one condenser, and one cooling tower.

2. Primary Equipment

Flow may be established and adjusted at primary equipment such as chillers, boilers, condensers, and heat exchangers by the following methods:

(a) flow meters or calibrated balancing valves,

(b) pressure losses, or

(c) pump curves.

a. FLOW METERS OR CALIBRATED BALANCING VALVES

Certainly the most accurate method of flow determination, flow meters or calibrated balancing valves should be used to determine the flow when they are available. Even though flow metering is the most reliable method of establishing equipment flow rates, it is recommended that pressure loss and pump curve methods, if available, be used to verify the flow quantities established by the flow metering devices.

b. PRESSURE LOSSES

When flow metering is not available, flow quantity may be established using pressure loss calculations as described in this chapter. **Caution:** Measured pressure losses of equipment must be for the equipment only! Some installations contain pressure taps at poor locations that may result in erroneous calculations, because of added losses from piping, accessories, etc.

c. PUMP CURVES

If the equipment tested is solely served by a circulating pump, the equipment flow rate may be assumed to be equal that of the pump. Care should be taken to insure that equipment tested does in fact receive the total flow of the pump. Check for bypasses around the pump before assuming that the equipment flow rate is equal to the pump flow rate.

When the flow adjustment is accomplished, place the equipment in a full load operating condition and capacity test. Refer to the following NEBB Test Report Forms for required test data:

Test Report Form	Form Number
Gas/Oil Fired Heat Apparatus	TAB 6-83
Packaged Chiller Test Report	TAB 13-83
Compressor and/or Condenser	TAB 15-83
Heat Exchanger/Convector	TAB 17-83
Boiler Test	TAB 19-83

3. Cooling Towers and Evaporative Condensers

Flow may be established and adjusted at cooling towers and evaporative condensers by the following methods:

(a) flow meters or calibrated balancing valves,

(b) pressure losses or spray pressures, or

(c) pump curves.

a. FLOW METERS OR CALIBRATED BALANCING VALVES

As with the other types of primary heat exchange equipment, flow metering is considered the most reliable method of establishing a flow rate. However, it is recommended that other available methods for the determination of flow be used to verify flow quantity as established by the flow metering device.

b. PRESSURE LOSSES OR SPRAY PRESSURES

A cooling tower or evaporative condenser containing spray headers and nozzles may have flow quantity established and adjusted by nozzle pressure loss. Approved submittal data from the manufacturer will indicate what nozzle (or spray) pressure will be found at the required flow. Actual flow quantity may then be determined using Equation 5-9 from Chapter V:

$$gpm_2 = gpm_1 \sqrt{\frac{\Delta P_2 \text{ (nozzles)}}{\Delta P_1 \text{ (nozzles)}}}$$

NOTE: Even if flow metering or pump curves are used to determine total tower flow, nozzle pressure loss is an efficient method of adjusting individual tower sections or bays.

c. PUMP CURVES

When a tower or condenser is solely served by a circulating pump, the flow rate may be assumed to be equal that of the pump. The same cautions as described in earlier sections should be taken to insure that total pump flow is in fact serving the tower or condenser.

When flow to the tower is uniform and within plus or minus ten percent of the required quantity, performance testing may be completed. Refer to the NEBB Test Report Form (TAB 16-83) "Cooling Tower or Evaporative Condenser" for the required test data.

4. Terminal Units

Flow may be established and adjusted at terminal unit equipment (heating and cooling coils, unit heaters, unit ventilators, fan coil units, etc..) by the following methods:

(a) flow meters or calibrated balancing valves,

(b) pressure losses,

(c) heat transfer, or

(d) pump curves.

a. FLOW METERS OR CALIBRATED BALANCING VALVES

As true with other types of hydronic HVAC equipment, flow metering devices provide the most reliable method of establishing terminal flow rates. Again, it is recommended that other methods available be used to verify flow meter readings.

b. PRESSURE LOSSES

When flow metering is not available, terminal unit flow rate may be established and balanced by equipment pressure loss.

NOTE: The pressure loss testing method may be utilized on a control valve serving the terminal unit if the valve flow coefficient (C_v) is known. A valve C_v is the amount of flow that will pass through an open control valve and produce a pressure differential of 1 psi.

c. HEAT TRANSFER

Establishing a terminal unit flow rate by heat transfer is the result of several simple calculations performed on air and water test data recorded. Since temperature readings are required on NEBB Test Report Forms, determination of flow by heat transfer is highly recommended for verifying flow rates.

d. PUMP CURVES

If a terminal unit is solely served by a circulating pump, the terminal flow rate may be established by any of the acceptable methods available for pump flow rate testing.

When flow adjustment is accomplished, place equipment in a full load operating condition and capacity test (If flow was established by the heat transfer method, this step has already been accomplished). Refer to the following NEBB Test Report Forms for the required test data:

Test Report Form	Form Number
Apparatus Coil Test Report	TAB 5-83
Terminal Unit Coil Test Report	TAB 12-83

C SPECIFIC PIPING APPLICATIONS

This section reviews recommended procedures and methods used for the testing, adjusting and balancing of specific piping applications. As many systems installed use combinations of the piping applications outlined below, it is necessary to apply the balancing procedures correctly. This may require that a procedure (or system) be broken down into several steps which correspond to the source, outlet, and piping.

All of the balancing procedures outlined below have two things in common:

(1) Balancing of forced circulation systems starts at the pump. Pump testing and adjustment, as described in Section B, must be accomplished prior to any adjustments to system piping or terminal units.

(2) System terminal units are maintained in the full flow position (i.e. control valves open to coil and closed to bypass) during the entire balancing procedure.

1. One-Pipe Systems

a. SERIES LOOP

Upon adjustment of the source flow rate, balancing of a series loop is accomplished with the use of one balancing device located in the loop. Balancing valves are normally located at the end (return) of a series loop piping arrangement. Adjustment of more than one device within a loop will directly affect the flow and heat transfer of other terminals in the loop. Total loop flow is the primary concern in the adjustment of series loop systems.

b. ONE-PIPE (MONOFLO)

Upon adjustment of the source flow rate, balancing of a one-pipe (single main) system is initiated with the first terminal unit supplied by the source. Adjustment of individual terminal unit flow rates may be accomplished by any of the acceptable methods utilizing a balancing device normally located in the return piping. Adjustment of the system continues sequentially from the first to the last terminal unit served.

2. Two-Pipe Systems

a. DIRECT-RETURN

Upon adjustment of the source flow rate, balancing of a two-pipe, direct-return system is initiated with the first terminal unit supplied by the source. Adjustment of terminal unit flow rates may be accomplished by any of the acceptable methods as outlined in the previous section. Adjustment of system terminal units continues sequentially from the first to the last terminal unit served. Several passes of the system terminal units may be required to achieve balanced conditions. Therefore, on the first pass it is recommended that flow to the first 1/3 section of the system should be adjusted 10 percent below the design requirements. The reasoning for this is that the system differential pressure will increase as terminal unit flow rates are adjusted. On the second pass of balancing, flow rates for the first section of terminal units will have increased, but should remain within acceptable tolerances.

b. REVERSE-RETURN

Adjustment and balancing of two-pipe reverse-return systems is usually much easier due to the inherent equal system pressure differential resulting from the piping arrangement. For example, a properly sized reverse-return piping system serving ten terminal units of identical capacity and resistance may require adjustment to the total system flow rate only. Flow will then be equally distributed to the terminals as a result of the piping application. However, many reverse-return systems do not contain identical terminal units. A review of the terminal units with specific attention to unit resistance should be made. Adjust the terminal flow rates starting with the units of least resistance and work toward the units with the greatest resistance.

3. Three-Pipe Systems

Due to the nature of this unusual piping application, the heating and cooling water systems may be viewed as diversity operations. In other words, the required terminal unit flow rate may vary with actual load conditions. Balancing should be performed on a system in a *maximum load condition*. Therefore, source and terminal unit adjustment should be accomplished with all terminal units (or as many as stipulated by diversity procedure) in the wide-open position.

Although connected to one another, the cooling and heating piping systems should be balanced as independently as possible with the given system conditions. Be aware that incorrect piping of automatic temperature control valves or improper system pressurization may result in extreme difficulties while performing the balancing procedures. "Crossed piping" and/or incorrect valve applications are another common installation problem.

4. Four-Pipe Systems

Typical four-pipe systems are simply two independent two-pipe systems and should be addressed accordingly. Four-pipe systems may also be seen as the heart of summer-winter systems which will be reviewed in the next section.

D SPECIFIC SYSTEM APPLICATIONS

1. Primary-Secondary Systems

Primary-secondary systems may appear to be too complex when first reviewed, but a proper system analysis will result in a relatively simple balancing procedure. First, address the primary loop. The source (primary pump) may supply outlets (primary bridges) in any of the possible piping arrangements described previously. The duty of the primary pump is to supply proper circulation to the primary bridges and return water back to the source. Initial balancing should therefore be restricted to the primary loop and it's components. Note that secondary systems should be in full flow operation during primary loop balancing.

Upon adjustment of primary pump flow rate, primary bridge piping is adjusted using a procedure applicable to the piping arrangement of the loop. When primary loop flow rates have been adjusted to design quantities, testing of the secondary systems may begin. Testing of each secondary system should be accomplished independently with procedures applicable to the piping arrangement of the secondary loop.

2. Summer-Winter Systems

Characteristics of summer-winter piping applications dictate that initial system testing and balancing be accomplished in the summer mode of operation. Design terminal unit cooling flow rates are usually much greater than that required for heating. As the terminal unit may only be adjusted to satisfy one flow rate, that flow rate must be the greatest required, or normally that of the cooling application. The system piping should be analyzed and set to accommodate the requirements of summer operation prior to pump testing. Insure that no bypasses are open and that summer-winter changeover valves (manual or automatic) are functional and open to the cooling mode of operation. Proceed to test and balance the pump(s) required for the chilled water operation. Upon completion of the adjustments to the circulating pump(s), test and balance the terminal units of the system in accordance with recommended procedures outlined for the piping application.

Upon completion of system balancing in the cooling mode of operation, switch the system operation over to the heating mode. Balance applicable pumps and equipment unique to the hot water piping without disturbing the valve settings accomplished during the summer mode balancing procedure.

If necessary, balancing of summer-winter systems may be accomplished in the winter mode of operation provided system pump and terminal units are set to design chilled water flow rates.

3. Constant Volume Systems

Constant volume systems have the following characteristics:

(1) No system diversity is present (Pump and primary heat exchange capacity equals terminal unit capacity).

(2) Constant flow is maintained at system terminal units with the use of straight-through or 3-way control valve piping applications.

Constant volume systems are tested with terminal unit coils in the full flow position. When terminal unit flow has been balanced to the design requirements, place the terminal unit in the bypass condition and adjust the bypass flow rate accordingly. Proper adjustment of the terminal coil and bypass flow rates provides a steady system differential pressure and subsequently constant system flow and heat transfer capabilities.

4. Variable Volume Systems

Variable volume systems have the following characteristics:

(1) A system diversity is usually present (required terminal flow rate exceeds pump and primary heat exchange unit).

(2) Two-way control valves are utilized at terminal units creating a variable system flow rate and pressure differential.

(3) Some form of differential pressure control (such as variable speed pumping) normally is used to maintain system differential pressure requirements.

Variable volume systems are tested with simulated full load system conditions. This procedure usually requires the temporary isolation of portions of the system piping and terminal units. When circulating pump and terminal unit capacities are within acceptable tolerances, terminal unit balancing may be performed in accordance with procedures stipulated by the piping configuration of the system. Upon completion of balancing procedures with a portion of the system isolated, the isolated units are then opened and an equal capacity of units closed. Units isolated for the initial balancing procedure are then balanced to design flow rates.

Specific units and procedures involved in the diversity balancing procedure should be delineated in an agenda for approval prior to initiating field testing.

5. Summary

There is no doubt that the testing, adjusting and balancing of any HVAC system will require a combined knowledge of system components, piping applications, and system design. A proper analysis of the systems and their components will make determination and implementation of acceptable balancing procedures much easier.

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CHAPTER 10 TAB REPORTS

A INTRODUCTION

When all field testing, adjusting and balancing work for a project has been completed, all paperwork should be assembled for review before submission to your NEBB Supervisor. The report should be assembled in a logical manner with all appropriate notes, clarifications, and schematic drawings necessary for understanding the report. The NEBB Supervisor and the system designer should be able to obtain a visual concept of the HVAC systems from the report data. There are two important things to remember when preparing the TAB paperwork:

(1) A TAB report must be able to be understood by someone simply by review so that all potential questions regarding the report must be answered by notes or schematics.

(2) The TAB report must be complete. Any "loose ends" will only result in costly return trips to the project to obtain missed data.

Proper use of the NEBB Test Report Forms, both as worksheets and as a final report of actual HVAC system operating conditions, will provide the best method of insuring that the TAB procedures were correctly performed. Accuracy in preparing final TAB report forms is important for several reasons:

(1) The TAB report offers a permanent record of the operating conditions of all of the HVAC systems and equipment after final adjustments were made.

(2) The TAB report is verification that all TAB procedures were followed and properly executed.

(3) The TAB report will serve as a reference that may be used by the owner for maintenance and system modifications.

(4) The TAB report can serve as an aid to the system designer in an evaluation of the performance of the HVAC systems in diagnosing any problem areas.

B DATA REVIEW; REPORT ASSEMBLY

Prior to submission of the TAB report to your NEBB Supervisor for review, the following steps should be taken:

1. Prepare a Report Cover Sheet

Each report should begin with a cover sheet listing the following data:

(a) Project Name

(b) Project Address

(c) Names of architect and engineer

(d) HVAC contractor

(e) Date of report submission

(f) Names of TAB technicians responsible for the test data.

2. Prepare a System Review Sheet

Each test report should contain a listing of all systems (air and hydronic) balanced, with those systems highlighted that were found to be performing outside of design tolerances. The review sheet should also list any system deficiencies (Bad thermostats, loose dampers, etc. . .) noted during the TAB procedures.

3. Instrument Calibration Report

A listing of all test instruments and calibration dates is required in all NEBB test reports. List all of the instruments that were used on the project during the course of the TAB work on the NEBB "Instrument Calibration Report" (TAB 20-83) form. This includes flow measuring hoods and other related devices.

4. Air Systems

Prepare a TAB test report for each air system balanced, paying particular attention to the following:

a. SYSTEM DIAGRAM

Be sure that the system diagram includes the locations of all of the air terminals and Pitot tube traverse locations indicated in the test reports. The system diagram is also a convenient location to make the appropriate notes or static pressure readings taken while troubleshooting system deficiencies.

b. AIR APPARATUS OR FAN TEST REPORTS

Be sure all pertinent data is included on the test report forms. If test data could not be obtained or is not applicable, indicate such on the report forms. Indicate on the fan test report forms how the actual cfm was obtained. Pitot tube (traverse, total of outlet airflows, or a combination).

c. DUCT PITOT TUBE TRAVERSE REPORTS

Confirm that the Pitot tube traverse test reports are complete with actual temperatures and system pressures recorded at the time of testing.

d. AIR OUTLET TEST REPORTS

Confirm that the air outlet test reports are complete with all applicable A_k factors and terminal device sizes. If flow measuring hoods are utilized for outlet readings, indicate their use in the remarks column (and on the "Instrument Calibration Report").

5. Hydronic Systems

Prepare a TAB test report for each hydronic system that was balanced, paying particular attention to the following:

a. SYSTEM DIAGRAM

A schematic diagram including all heat exchange equipment and locations of flow measuring devices should be prepared.

b. PUMP TEST REPORT

Confirm all test data has been properly entered on the test report form. Attach the manufacturer's pump capacity curves, with the actual pump operating point plotted, to the test report form when available. Note how the actual pump flow rate was determined (flow meter, pump curve, etc. . .).

c. PRIMARY HEAT EXCHANGE EQUIPMENT

Confirm that all appropriate test data has been recorded for the chillers, boilers, heat exchangers, and other primary heat exchange equipment. Note how the actual flow rates of each item of equipment was determined or obtained.

d. TERMINAL HEAT EXCHANGE EQUIPMENT

Confirm that all required coil and terminal unit temperatures and pressures were recorded and entered properly on the TAB test report forms. Indicate how each terminal unit flow rate was determined.

C SUMMARY

After all data has been received and clarified with the appropriate notes and diagrams, submit the properly assembled TAB report to your NEBB Supervisor for his final review and formal submission of report to the proper parties. Remember, all TAB test reports must be complete and comprehensive prior to submission. It is the appearance of the testing, adjusting and balancing report and the exactness of the data that ultimately reflects the quality of work that a NEBB TAB technician produces.

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CHAPTER 11 GLOSSARY OF TERMS AND DEFINITIONS

Note: This chapter contains many terms and definitions, not found in the text material, but used by NEBB Supervisors and system designers, including those used in sound and vibration work.

A-Scale: A filtering system that has characteristics which roughly match the response characteristics of the human ear at low sound levels (below 55 dB Sound Pressure Level, but frequently used to gauge levels to 85 dB). A-scale measurements are often referred to as dB(A).

Absolute Pressure: Air at standard conditions (70°F air at sea level with a barometric pressure of 29.92 in.Hg.) exerts a pressure of 14.696 psi. This is the pressure in a system when the pressure gauge reads zero. So the *absolute pressure* of a system is the gauge pressure in pounds per square inch added to the atmospheric pressure of 14.696 psi (use 14.7 psi in *environmental system work*) and the symbol is "psia."

Acceleration: The time rate of change of velocity; i.e., the derivative of velocity with respect to time.

Acceleration Due to Gravity: The rate of increase in velocity of a body falling freely in a vacuum. Its value varies with latitude and elevation. The International Standard is 32.174 ft. per second per second.

Adiabatic Process: A thermodynamic process during which no heat is added to, or taken from, a substance or system.

Aerodynamic Noise: Also called generated noise, self-generated noise; is noise of aerodynamic origin in a moving fluid arising from flow instabilities. In duct systems, aerodynamic noise is caused by airflow through elbows, dampers, branch wyes, pressure reduction devices, silencers and other duct components.

Air, Ambient: Generally speaking, the air surrounding an object.

Air, Dry: Air without contained water vapor; air only.

Air, Outdoor: Air taken from outdoors and, therefore, not previously circulated through the system.

Air, Outside: External air; atmosphere exterior to refrigerated or conditioned space; ambient (surrounding) air.

Air, Recirculated: Return air passed through the conditioner before being again supplied to the conditioned space.

Air, Reheating of: In an air conditioning system, the final step in treatment, in the event the temperature is too low.

Air, Return: Air returned from conditioned or refrigerated space.

Air, Saturated: Moist air in which the partial pressure of the water vapor is equal to the vapor pressure of water at the existing temperature. This occurs when dry air and saturated water vapor coexist at the same dry-bulb temperature.

Air, Standard: Dry air at a pressure of 29.92 in. Hg at 69.8°F temperature and with a specific volume of 13.33 ft.^{3}/lb.

Airborne Sound: Airborne sound is sound which reaches the point of interest by radiation through the air.

Air Changes: A method of expressing the amount of air leakage into or out of a building or room in terms of the number of building volumes or room volumes exchanged.

Air Conditioning, Comfort: The process of treating air so as to control simultaneously its temperature, humidity, cleanliness and distribution to meet the comfort requirements of the occupants of the conditioned space.

Air Conditioning Unit: An assembly of equipment for the treatment of air so as to control, simultaneously, its temperature, humidity, cleanliness and distribution to meet the requirements of a conditioned space.

Air Diffuser: A circular, square, or rectangular air distribution outlet, generally located in the ceiling and comprised of deflecting members discharging supply air in various directions and planes, and arranged to promote mixing of primary air with secondary room air.

Air Washer: A water spray system or device for cleaning, humidifying, or dehumidifying the air.

Airborne Noise: Noise which reaches the observer by transmission through air.

Anemometer: An instrument for measuring the velocity of a fluid.

Anticipators: A small heater element in two-position temperature controllers which deliberately cause false indications of temperature in the controller in an attempt to minimize the override of the differential and smooth out the temperature variation in the controlled space.

Aspect Ratio: In air distribution outlets, the ratio of the length of the core opening of a grille, face, or register to the width. In rectangular ducts, the ratio of the width to the depth.

Aspiration: Production of movement in a fluid by suction created by fluid velocity.

Attenuation: The sound reduction process in which sound energy is absorbed or diminished in intensity as the result of energy conversion from sound to motion or heat.

Background Noise: Sound other than the wanted signal. In room acoustics, the irreducible noise level measured in the absence of any building occupants.

Barometer: Instrument for measuring atmospheric pressure.

Breakout Noise: The term used in Great Britain for the transmission or radiation of noise from some part of the duct system to an occupied space in the building. In the United States, the terms "flanking" and "duct radiation" are more frequently used; however, the term "breakout noise" seems more descriptive.

British Thermal Unit (Btu): The Btu is defined as the heat required to raise the temperature of a pound of water from 59° F to 60° F.

Bulb: The name given to the temperature sensing device located in the fluid for which control or indication is provided. The bulb may be liquid-filled, gas-filled, or gas-and-liquid-filled. Changes in temperature produce pressure changes within the bulb which are transmitted to the controller.

Bypass: A pipe or duct, usually controlled by valve or damper, for conveying a fluid around an element of a system.

Calibration: Process of dividing and numbering the scale of an instrument; also of correcting or determining the error of an existing scale, or of evaluating one quantity in terms of readings of another.

Capillary: The name given to the thin tube attached to the bulb which transmits the bulb pressure changes to the controller or indicator. The cross sectional area of the capillary is extremely small compared to the cross section of the bulb so that the capillary, which is usually outside of the controlled fluid, will introduce the smallest possible error in the signal being transmitted from the bulb.

Capillary Tube: The capillary tube is a metering device made from a thin tube approximately 2 to 20 feet long and from .025 to .090 inches in diameter which feeds liquid directly to the evaporator. Usually limited to systems of 1 ton or less, it performs all of the functions of the thermal expansion valve when properly sized.

Ceiling Outlet: A round, square, rectangular, or linear air diffuser located in the ceiling which provides a horizontal distribution pattern of primary and secondary air over the occupied zone and induces low velocity secondary air motion through the occupied zone.

Celsius (Formerly Centigrade): A thermometric scale in which the freezing point of water is called 0°C and its boiling point 100°C at normal atmospheric pressure (14.696 psi).

Coefficient of Discharge: For an air diffuser, the ratio of net area or effective area at vena contracta of an orificed airstream to the free area of the opening.

Coefficient of Expansion: The change in length per unit length or the change in volume per unit volume, per deg. change in temperature.

Coefficient of Performance (COP), Heat Pump: The ratio of the compressor heating effect (heat pump) to the rate of energy input to the shaft of the compressor, in consistent units, in a complete heat pump, under designated operating conditions.

Comfort Chart: A chart showing effective temperatures with dry-bulb temperatures and humidities (and sometimes air motion) by which the effects of various air conditions on human comfort may be compared.

Comfort Cooling: Refrigeration for comfort as opposed to refrigeration for storage or manufacture.

Comfort Zone: (Average) the range of effective temperatures over which the majority (50 percent or more) of adults feels comfortable; (extreme) the range of effective temperatures over which one or more adults feel comfortable.

Compressibility: The ease which a fluid may be reduced in volume by the application of pressure, depends upon the state of the fluid as well as the type of fluid itself. In TAB work, consider that water may not be compressed. Air is a compressible gas, but that factor is usually not considered during normal testing and balancing procedures.

Compressor: The pump which provides the pressure differential to cause fluid to flow and in the pumping process increases pressure of the refrigerant to the high side condition. The compressor is the separation between low side and high side.

Condensate: The liquid formed by condensation of a vapor. In steam heating, water condensed from steam; in air conditioning, water extracted from air, as by condensation on the cooling coil of a refrigeration machine.

Condenser: The heat exchanger in which the heat absorbed by the evaporator and some of the heat of compression introduced by the compressor are removed from the system. The gaseous refrigerant changes to a liquid, again taking advantage of the relatively large heat transfer by the change of state in the condensing process.

Condensing Unit, Refrigerant: An assembly of refrigerating components designed to compress and liquify a specific refrigerant, consisting of one or more refrigerant compressors, refrigerant condensers, liquid receivers (when required) and regularly furnished accessories.

Conductivity, Thermal: The time rate of heat flow through unit area and unit thickness of a homogeneous material under steady conditions when a unit temperature gradient is maintained in the direction perpendicular to area. Materials are considered homogeneous when the value of the thermal conductivity is not affected by variation in thickness or in size of sample within the range normally used in construction.

Connection in Parallel: System whereby flow is divided among two or more channels from a common starting point or header.

Connection in Series: System whereby flow through two or more channels is in a single path entering each succeeding channel only after leaving the first or previous channel.

Control: A device for regulation of a system or component in normal operation, manual or automatic. If automatic, the implication is that it is responsive to changes of pressure, temperature or other property whose magnitude is to be regulated.

Controlled Device: One which receives the converted signal from the transmission system and translates it into the appropriate action in the environmental system. For example: a valve opens or closes to regulate fluid flow in the system.

Controller: An instrument which receives the signal from the sensing device and translates that signal into the appropriate corrective measure. The correction is then sent to the system controlled devices through the transmission system.

Convection: Transfer of heat by movement of fluid.

Convection, Forced: Convection resulting from forced circulation of a fluid, as by a fan, jet or pump.

Convection, Natural: Circulation of gas or liquid (usually air or water) due to differences in density resulting from temperature changes.

Cooling, Evaporative: Involves the adiabatic exchange of heat between air and water spray or wetted surface. The water assumes the wet-bulb temperature of the air, which remains constant during its traverse of the exchanger.

Cooling Coil: An arrangement of pipe or tubing which transfers heat from air to a refrigerant or brine.

Core Area: The total plane area of that portion of a grille, included within lines tangent to the outer edges of the openings through which air can pass.

Corrosive: Having chemically destructive effect on metals (occasionally on other materials).

Counterflow: In heat exchange between two fluids, opposite direction of flow, coldest portion of one meeting coldest portion of the other.

Critical Velocity: The velocity above which fluid flow is turbulent.

Cycle: A complete course of operation of working fluid back to a starting point, measured in thermodynamic terms (functions). Also in general for any repeated process on any system.

Dalton's Law of Partial Pressure: Each constituent of a mixture of gases behaves thermodynamically as if it alone occupied the space. The sum of the individual pressures of the constituents equals the total pressure of the mixture.

Damper: A device used to vary the volume of air passing through an air outlet, air inlet or duct.

Decibel (dB): A decibel is a division of a logarithmic scale for expressing the ratio of two quantities proportional to power or energy. The number of decibels denoting such a ratio is ten times the logarithm of the ratio.

Degree Day: A unit, based upon temperature difference and time, used in estimating fuel consumption and specifying nominal heating load of a building in winter. For any one day, when the mean temperature is less than 65° F, there exist as many degree days as there are Fahrenheit degrees difference in temperature between the mean temperature for the day and 65° F.

Dehumidification: The condensation of water vapor from air by cooling below the dewpoint or removal of water vapor from air by chemical or physical methods.

Dehumidifier: (1) An air cooler or washer used for lowering the moisture content of the air passing through it; (2) An absorption or adsorption device for removing moisture from air.

Density: The ratio of the mass of a specimen of a substance to the volume of the specimen. The mass of a unit volume of a substance. When weight can be used without confusion, as synonymous with mass, density is the weight per unit volume.

Desiccant: Any absorbent or adsorbent, liquid or solid, that will remove water or water vapor from a material. In a refrigeration circuit, the desiccant should be insoluble in the refrigerant.

Design Working Pressure: The maximum allowable working pressure for which a specific part of a system is designed.

Dewpoint, Apparatus: That temperature which would result if the psychrometric process occurring in a dehumidifier, humidifier or surface-cooler were carried to the saturation condition of the leaving air while maintaining the same ratio of sensible to total heat load in the process.

Differential: The difference between the points where a controller turns "on" and "off." If a thermostat turns a furnace on at 68° and the differential is 3°, the burner will be turned off at 71°.

Diffuser: A circular, square, or rectangular air distribution outlet, generally located in the ceiling and comprised of deflecting members discharging supply air in various directions and planes, and arranged to promote mixing of primary air with secondary room air.

Discharge Stop Valve: The manual service valve at the leaving connection of the compressor.

Direct Acting: Instruments that increase control pressure as the controlled variable (such as temperature or pressure) increases; while *reverse acting* instruments increase control pressure as the controlled variable decreases.

Draft: A current of air, when referring to the pressure difference which causes a current of air or gases to flow through a flue, chimney, heater, or space; or to a localized effect caused by one or more factors of high air velocity, low ambient temperature, or direction of air flow, whereby more heat is withdrawn from a person's skin than is normally dissipated.

Drier: A manufactured device containing a desiccant placed in the refrigerant circuit. Its primary purpose is to collect and hold within the desiccant, all water in the system in excess of the amount which can be tolerated in the circulating refrigerant.

Drift: Term used to describe the difference between the set point and the actual operating or control point.

Drip: A pipe, or a steam trap and a pipe considered as a unit, which conducts condensation from the steam side of a piping system to the water or return side of the system.

Droop: Terms used to describe the difference between the set point and the actual operating or control point.

Drop: The vertical distance that the lower edge of a horizontally projected airstream drops between the outlet and the end of its throw.

Dry bulb, Room: The dry bulb (dewpoint, etc.) temperature of the conditioned room or space.

Dry Bulb Temperature: The temperature registered by an ordinary thermometer. The dry bulb temperature represents the measure of sensible heat, or the intensity of heat.

Duct: A passageway made of sheet metal or other suitable material, not necessarily leaktight, used for conveying air or other gas at low pressures.

Dynamic Discharge Head: Static discharge head plus friction head plus velocity head.

Dyanmic Suction Head: Positive static suction head minus friction head and minus velocity head.

Dynamic Suction Lift: The sum of suction lift and velocity head at the pump suction when the source is below pump centerline.

Dynamic Insertion Loss: The dynamic insertion loss of a silencer, duct lining or other attenuating device is the performance measured in accordance with ASTM E 477 when handling the rated airflow. It is the reduction in sound power level, expressed in decibels, due solely to the placement of the sound attenuating device in the duct system.

Effect, Cooling, Total: The difference between the total enthalpy of the dry air and water vapor mixture entering a unit per hour and the total enthalpy of the dry air and water vapor (and water) mixture leaving the unit per hour, expressed in Btu per hour.

Effect, Sun: Solar energy transmitted into space through windows and building materials.

Effective Area: The net area of an outlet or inlet device through which air can pass, equal to the free area times the coefficient of discharge.

Electro-Pneumatic (EP) Switches: Switches that open or close an air line valve from an electrical impulse.

Equal Friction Method: A method of duct sizing wherein the selected duct friction loss value is used constantly throughout the design of a low pressure duct system.

Enthalpy: The total quantity of heat energy contained in a substance, also called *total heat*; the thermodynamic property of a substance defined as the sum of its internal energy plus the quantity Pv/J, where P = pressure of the substance, v = its volume, and J = the mechanical equivalent of heat.

Enthalpy, Specific: A term sometimes applied to enthalpy per unit weight.

Entrainment: The capture of part of the surrounding air by the airstream discharged from an outlet (sometimes called secondary air motion).

Entropy: The ratio of the heat added to a substance to the absolute temperature at which it is added.

Entropy, Specific: A term sometimes applied to entropy per unit weight.

Equivalent Duct Diameter: The equivalent duct diameter for a rectangular duct with sides of dimensions a and b is $\sqrt{\frac{4ab}{\pi}}$. (for same cross-sectional area).

Evaporative Cooling: The adiabatic exchange of heat between air and a water spray or wetted surface. The water approaches the wet-bulb temperature of the air, which remains constant during its traverse of the exchanger.

Evaporator: The heat exchanger in which the medium being cooled, usually air or water, gives up heat to the refrigerant through the exchanger transfer surface. The liquid refrigerant boils into a gas in the process of the heat absorption.

Extended Surface: Heat transfer surface, one or both sides of which are increased in area by the addition of fins, discs, or other means.

Face Area: The total plane area of the portion of a grille, coil, or other items bounded by a line tangent to the outer edges of the openings through which air can pass.

Face Velocity: The velocity obtained by dividing the air quantity by the component face area.

Fahrenheit: A thermometric scale in which 32 (°F) denotes freezing and 212 (°F) the boiling point of water under normal pressure at sea level (14.696 psi).

Fan, Centrifugal: A fan rotor or wheel within a scroll type housing and including driving mechanism supports for either belt drive or direct connection.

Fan Performance Curve: Fan performance curve refers to the constant speed performance curve. This is a graphical presentation of static or total pressure and power input over a range of air volume flow rate at a stated inlet density and fan speed. It may include static and mechanical efficiency curves. The range of air volume flow rate which is covered generally extends from shutoff (zero air volume flow rate) to free delivery (zero fan static pressure). The pressure curves are generally referred to as the pressure-volume curves.

Fan, Propeller: A propeller or disc type wheel within a mounting ring or plate and including driving mechanism supports for either belt drive or direct connection.

Fan, Tubeaxial: A propeller or disc type wheel within a cylinder and including driving mechanism supports for either belt drive or direct connection.

Fan, Vaneaxial: A disc type wheel within a cylinder, a set of air guide vanes located either before or after the wheel and including driving mechanism supports for either belt drive or direct connection.

Filter: A device to remove solid material from a fluid.

Filter-Drier: A combination device used as a strainer and moisture remover.

Flanking Transmission (Sound): The reduction in apparent transmission loss of a wall caused by sound being carried around the wall by other paths. (Structure-borne, leaks, etc.)

Floating Action Controllers: Essentially two position type controllers which vary the position of the controlled devices but which are arranged to stop before reaching a maximum or minimum position.

Flow, Laminar or Streamline: Fluid flow in which each fluid particle moves in a smooth path substantially parallel to the paths followed by all other particles.

Flow, Turbulent: Fluid flow in which the fluid moves tranversely as well as in the direction of the tube or pipe axis, as opposed to streamline or viscous flow.

Fluid: Gas, vapor, or liquid.

Fluid Dynamics: Fluid Dynamics is used to describe the condition of motion of a fluid within a system. The velocity of a fluid is based upon the cross-sectional area and the volume of a fluid passing through it. The importance of this property is that volume may be determined for air or water systems when the area and velocity are known.

Fluid Statics: Fluid Statics as applied to TAB work, refers to a condition of a quantity of fluid at rest. It is the direct result of gravity and weight. Static pressure is used in both air and water testing to determine the potential for the movement of fluid within a system. Pressures in air systems are normally measured in units of inches of water (in.w.g.). A pressure unit of one inch of water is equivalent to the static pressure found at the base of a column of water one inch high. Pressures in water systems are normally measured in pounds per square inch (psi), but are converted to feet of water (ft.w.g.) for the purpose of evaluating pump and equipment performance.

Fluid, Heat Transfer: Any gas, vapor, or liquid used to absorb heat from a source at a high temperature and reject it to a lower temperature substance.

Force: The action on a body which tends to change its relative condition as to rest or motion.

Free Area: The total minimum area of the openings in the air outlet or inlet through which air can pass.

Free Delivery-Type Unit: A device which takes in air and discharges it directly to the space to be treated without external elements which impose air resistance.

Free Sound Field (Free Field): A free sound field is a field in a homogeneous, isotropic medium free from boundaries. In practice, it is a field in which the effects of the boundaries are negligible over the region of interest. In the free field, the sound pressure level decreases 6 dB for a doubling of distance from a point source.

Frequency: The number of vibrations, waves, or cycles of any periodic phenomenon per second. In architectural acoustics, the interest lies in the audible frequency range of 20 to 20,000 cps Hertz (cycles per second).

Friction: Friction is the resistance found at the duct and piping walls. Resistance creates a static pressure loss in systems. The primary purpose of a fan or pump is to produce a design volume of fluid at a pressure equal to the frictional resistance of the system and the other dynamic pressure losses of the components.

Friction Head: The pressure in psi or feet of the liquid pumped which represents system resistance that must be overcome.

Fumes: Solid particles commonly formed by the condensation of vapors from normally solid materials such as molten metals. Fumes may also be formed by sublimation, distillation, calcination, or chemical reaction wherever such processes create airborne particles predominantly below one micron in size. Such solid particles sometimes serve as condensation nuclei for water vapor to form smog.

Gas: Usually a highly superheated vapor which, within acceptable limits of accuracy, satisfies the perfect gas laws.

Gas, Inert: A gas that neither experiences nor causes chemical reaction nor undergoes a change of state in a system or process; e.g., nitrogen or helium mixed with a volatile refrigerant.

Gas Constant: The coefficient "R" in the perfect gas equation: PV = MRT.

Gradual Switches: Manual adjustment devices which proportion the control condition in accordance with the position of the switch.

Gravity, Specific: Density compared to density of standard material; reference usually to water or to air.

Grille: A louvered or perforated covering for an air passage opening which can be located on a wall, ceiling or floor.

Head, Dynamic or Total: In flowing fluid, the sum of the static and velocity heads at the point of measurement.

Head, Static: The static pressure of fluid expressed in terms of the height of a column of the fluid, or of some manometric fluid, which it would support.

Head, Velocity: In a flowing fluid, the height of the fluid or of some manometric fluid equivalent to its velocity pressure.

Heat: The form of energy that is transferred by virtue of a temperature difference.

Heat, Latent: Change of enthalpy during a change of state, usually expressed in Btu per lb. With pure substances, latent heat is absorbed or rejected at constant pressure.

Heat, Sensible: Heat which is associated with a change in temperature; specific heat exchange of temperature; in contrast to a heat interchange in which a change of state (latent heat) occurs.

Heat, Specific: The ratio of the quantity of heat required to raise the temperature of a given mass of any substance one degree to the quantity required to raise the temperature of an equal mass of a standard substance (usually water at 59° F) one degree.

Heat Capacity: The amount of heat necessary to raise the temperature of a given mass one degree. Numerically, the mass multiplied by the specific heat.

Heat Conductor: A material capable of readily conducting heat. The opposite of an insulator or insulation.

Heat Exchanger: A device specifically designed to transfer heat between two physically separated fluids.

Heat of Fusion: Latent heat involved in changing between the solid and the liquid states.

Heat of Vaporization: Latent heat involved in the change between liquid and vapor states.

Heat Pump: A refrigerating system employed to transfer heat into a space or substance. The condenser provides the heat while the evaporator is arranged to pick up heat from air, water, etc. By shifting the flow of air or other fluid, a heat pump system may also be used to cool the space.

Heat Transmission: Any time-rate of heat flow; usually refers to conduction, convection and radiation combined.

Heat Transmission Coefficient: Any one of a number of coefficients used in the calculation of heat transmission by conduction, convection, and radiation, through various materials and structures.

High Pressure Cutout: A pressure actuated switch to protect the compressor from pressure often caused by high condenser temperatures and pressure due to fouling and lack of water or air.

Hot Gas Bypass: The piping and manual, but more often automatic, valve used to introduce compressor discharge gas directly into the evaporator. This type of arrangement will maintain compressor operation at light loads down to zero by falsely loading the evaporator and compressor.

Hot Gas Piping: The compressor discharge piping which carries the hot refrigerant gas from the compressor to the condenser. Velocities must be high enough to carry entrained oil.

Humidifier: A device to add moisture to air.

Humidifying Effect: The latent heat of vaporization of water at the average evaporating temperature times the weight of water evaporated per unit of time.

Humidistat: A regulatory device, actuated by changes in humidity, used for the automatic control of relative humidity.

Humidity: Water vapor within a given space.

Humidity, Absolute: The weight of water vapor per unit volume.

Humidity, Relative: The ratio of the mol fraction of water vapor present in the air, to the mol fraction of water vapor present in saturated air at the same temperature and barometric pressure; approximately, it equals the ratio of the partial pressure or density of the water vapor in the air, to the saturation pressure or density, respectively, of water vapor at the same temperature.

Humidity, Specific: Weight of water vapor (steam) associated with 1 lb. weight of dry air, also called humidity.

Hunting: A condition which occurs when the desired condition cannot be maintained. The controller, controlled device and system, individually or collectively, continuously override or "overshoot" the control point with a resulting fluctuation and loss of control of the condition to be maintained.

Hygroscopic: Absorptive of moisture, readily absorbing and retaining moisture.

Inch of Water (in. w.g.): A unit of pressure equal to the pressure exerted by a column of liquid water 1 inch high at a temperature of 39.2°F.

Indicator: A term used to describe any device such as a thermometer or pressure gauge which is used to indicate the condition at a point in the system but which does not provide any controlling action or effect on the system operation.

Induction: The capture of part of the ambient air by the jet action of the primary airstream discharging from a controlled device.

Infiltration: Air flowing inward as through a wall, crack, etc.

Insertion Loss: The insertion loss of an element of an acoustic transmission system is the positive or negative change in acoustic power transmission that results when the element is introduced.

Isothermal: An adjective used to indicate a change taking place at constant temperature.

Lag: The condition which occurs when the control reacts too slowly, and the system never catches up with the variations of the controlled condition. The result will be poor control and hunting.

Latent Heat: Latent heat is any heat transfer that causes a change of state from a solid to a liquid, a liquid to a gas, or vice versa. Evaporation of water is an example of a latent heat transfer. Latent heat transfer at terminal coils may be defined as any process which humidifies or dehumidifies the air. Both processes result in a change of actual moisture content in the air.

Liquid Sight Glass: The glass ported fitting in the liquid line used to indicate adequate refrigerant charge. When bubbles appear in the glass, there is insufficient refrigerant in the system.

Liquid Solenoid Valve: The electrically operated automatic shut off valve in the liquid piping which closes on system shut down to close off receiver discharge when used in pump down cycle and which prevents refrigerant migration in any system.

Load: The amount of heat per unit time imposed on a refrigerant system, or the required rate of heat removal.

Low Temperature Cutout: A pressure or temperature actuated device with sensing element in the evaporator, which will shut the system down at its control setting to prevent freezing chilled water or to prevent coil frosting. Direct expansion equipment may not use this device.

Manometer: An instrument for measuring pressures: essentially a U-tube partially filled with a liquid, usually water, mercury, or a light oil, so constructed that the amount of displacement of the liquid indicates the pressure being exerted on the instrument.

Mass: The quality of matter in a body as measured by the ratio of the force required to produce given acceleration, to the acceleration.

Mass Law: The law relating to the transmission loss of sound barriers which says that in a part of the frequency range, the magnitude of the loss is controlled entirely by the mass per unit area of the barrier. The law also says that the transmission loss increases 6 decibels for each doubling of frequency or each doubling of the barrier mass per unit area.

Microbar: A unit of pressure equal to 1 dyne/cm^2 (one millionth of the pressure of the atmosphere).

Micron: A unit of length, the thousandth part of 1 mm or the millionth of a meter.

Modulating: Of a control, tending to adjust by increments and decrements.

Modulating Controllers: Constantly reposition themselves in proportion to the requirements of the system, theoretically being able to maintain an accurately constant condition.

Noise: Any undesired sounds, usually of different frequencies, resulting in an objectionable or irritating sensation.

Noise Reduction (NR): The difference between the average sound pressure levels of two spaces. Usually these two spaces are two adjacent rooms called, respectively, the source room and the receiving room.

Normally open (or Normally closed): The position of a valve, damper, relay contacts, or switch when external power or pressure is *not* being applied to the device. Valves and dampers usually are returned to a "normal" position by a spring.

Octave Band (O.B.): A range of frequency where the highest frequency of the band is double the lowest frequency of the band. The band is usually specified by the center frequency.

Offset: Term used to describe the difference between the set point and the actual operating or control point.

"On-off" Control: A two position action which allows operation at either maximum or minimum condition, or on or off, depending on the position of the controller.

Operating Point: The value of the controlled condition at which the controller actually operates. Also called control point.

Outlet, Ceiling: A round, square, rectangular, or linear air diffuser located in the ceiling, which provides a horizontal distribution pattern of primary and secondary air over the occupied zone and induces low velocity secondary air motion through the occupied zone.

Outlet, Slotted: A long, narrow air distribution outlet, comprised of deflecting members, located in the ceiling, sidewall, or sill, with an aspect ratio greater than 10, designed to distribute supply air in varying directions and planes, and arranged to promote mixing of primary air and secondary room air.

Outlet, Vaned: A register or grille equipped with vertical and/or horizontal adjustable vanes.

Outlet Velocity: The average velocity of air emerging from an opening, fan or outlet, measured in the plane of the opening.

Output: Capacity, duty, performance, net refrigeration produced by system.

Outside Air Opening: Any opening used as an entry for air from outdoors.

Overall Coefficient of Heat Transfer (thermal transmittance): The time rate of heat flow through a body per unit area, under steady conditions, for a unit temperature difference between the fluids on the two sides of the body.

Performance Factor: Ratio of the useful output capacity of a system to the input required to obtain it. Units of capacity and input need not be consistent.

Pneumatic-Electric (PE) Switches: Device that operates an electric switch from a change of air pressure.

Point of Duty: A statement of air volume flow rate and static or total pressure at the stated density and is used to specify the point on the system curve at which a fan is to operate.

Point of Operation: Used to designate the single set fan performance values which correspond to the point of intersection of the system curve and the fan pressure-volume curve.

Point of Rating: A statement of fan performance values which correspond to one specific point on the fan pressure-volume curve.

Potentiometers: Electric or electronic devices which position proportional controls in accordance with the position and resistance value of the potentiometer.

Pressure: The normal force exerted by a homogeneous liquid or gas, per unit of area, on the wall of its container.

Pressure, Absolute: Pressure referred to that of a perfect vacuum. It is the sum of gauge pressure and atmospheric pressure.

Pressure, Atmospheric: It is the pressure indicated by a barometer. Standard atmosphere is the pressure equivalent to 14.696 psi or 29.921 in. of mercury at 32° F.

Pressure, Gauge: Pressure above atmospheric.

Pressure, Hydrostatic: The normal force per unit area that would be exerted by a moving fluid on an infinitesimally small body immersed in it if the body were carried along with the fluid.

Pressure, Partial: Portion of total gas pressure of a mixture attributable to one component.

Pressure, Saturation: The saturation pressure for a pure substance for any given temperature is that pressure at which vapor and liquid, or vapor and solid, can coexist in stable equilibrium.

Pressure, Static (SP): The normal force per unit area that would be exerted by a moving fluid on a small body immersed in it if the body were carried along with the fluid. Practically, it is the normal force per unit area at a small hole in a wall of the duct through which the fluid flows (piezometer) or on the surface of a stationary tube at a point where the disturbances, created by inserting the tube, cancel. It is supposed that the thermodynamic properties of a moving fluid depend on static pressure in exactly the same manner as those of the same fluid at rest depend upon its uniform hydrostatic pressure.

Pressure, Total (TP): In the theory of the flow of fluids, the sum of the static pressure and the velocity pressure at the point of measurement. Also called dynamic pressure.

Pressure, Velocity (V_p): In moving fluid, the pressure capable of causing an equivalent velocity, if applied to move the same fluid through an orifice such that all pressure energy expended is converted into kinetic energy.

Pressure Drop: Pressure loss in fluid pressure, as from one end of a duct to the other, due to friction, dynamic losses, and changes in velocity pressure.

Primary Air: The initial airstream discharged by an air outlet (the air being supplied by a fan or supply duct) prior to any entrainment of the ambient air.

Psychrometer: An instrument for ascertaining the humidity or hygrometric state of the atmosphere.

Psychrometric Chart: A graphical representation of the thermodynamic properties of moist air.

Pyrometer: An instrument for measuring high temperatures.

Radiation (Acoustic): The process of turning structure-borne noise into airborne (or some other fluid-borne) noise.

Radius of Diffusion: The horizontal axial distance an airstream travels after leaving an air outlet before the maximum stream velocity is reduced to a specified terminal level; e.g., 200, 150 or 100 fpm.

Radiation, Thermal: The transmission of heat through space by wave motion; the passage of heat from one object to another without warming the space between.

Receiver: An auxiliary storage receptacle for refrigerant when the system is pumped down and shut down.

Register: A grille equipped with an integral damper or control valve.

Relative Humidity (RH): The ratio of water vapor in the air as compared to the maximum amount of water vapor that may be contained.

Relays: Switching devices which produce an appropriate reaction as a result of an originating action or signal. Electric relays, for example, may de-energize a device as a result of another device being energized. Relays may be electric, pneumatic, or a combination of both . PE and EP switches are relays.

Reset Controllers: Two controllers operating together; one sensing a condition other than that of the controlled space and changing the set point of the second controller, which is directly responsible for the result in the controlled space. The resetting controller is commonly called the master, and the controller being reset is commonly called the submaster (slave).

Resistance, Thermal: The reciprocal of thermal conductance.

Resistivity, Thermal: The reciprocal of thermal conductivity.

Return Air: Air returned from conditioned or refrigerated space.

Reverberation: The persistence of sound in an enclosed space after the sound source has stopped. In a reverberation room, it is characterized by the decay or dying away of the sound.

Room Dry Bulb The dry-bulb (dewpoint, etc.) temperature of the conditioned room or space.

Room Effect: The difference between the sound power level discharged by a duct (through a diffuser or other termination device) and the sound pressure level heard by an occupant of a room is called the Room Effect. The magnitude of the Room Effect depends upon the amount of sound absorption in the room (Sabins), the distance between the termination of the duct and the nearest observer and the directivity factor of the source.

Room Velocity: The residual air velocity level in the occupied zone of the conditioned space; e.g., 65, 50, 35 fpm.

Sabin: The unit of acoustic absorption. One sabin is one square foot of perfect sound-absorbing material.

Saturation, Degree of: The ratio of the weight of water vapor associated with a pound of dry air to the weight of water vapor associated with a pound of dry air saturated at the same temperature.

Secondary Air: The air surrounding an outlet that is captured or entrained by the initial outlet discharge airstream (furnished by a supply duct or fan).

Semi-Extended Plenum: A trunk duct that is extended as a plenum from a fan or HVAC unit to serve multiple outlets and/or branch ducts.

Sensible Heat: Sensible heat is any heat transfer that causes a change in temperature. Heating and cooling of air and water that may be measured with a thermometer is sensible heat. Heating or cooling coils that simply increase or decrease the air temperature are examples of sensible heat.

Sensible Heat Ratio, Air Cooler: The ratio of sensible cooling effect to total cooling effect of an air cooler.

Sensing Device: A device that keeps track of the measured condition and its fluctuations so that when sufficient variation occurs it will originate the signal to revise the operation of the system and offset the change. Example: a thermostat "bulb." A sensing device may be an integral part of a controller.

Sensitivity: The ability of a control instrument to measure and act upon variations of the measured condition.

Set Point: The value of the controlled condition at which the instrument is set to operate. The set point in the example in "differential" might be 69½°, the midpoint of the differential.

Solenoid Air Valves: EP switches with an electromagnetic coil in the valve topworks that opens or closes the valve from normal position. A spring returns the valve to the normal position when the coil is de-energized.

Sone: One sone is defined as the loudness of a 1000 Hz tone having a sound pressure of 40 dB. Two sones is twice as loud as the 40 dB reference sound of one sone, etc.

Sound Absorption: (1) The process of dissipating or removing sound energy. (2) The property possessed by materials, objects, and structures, such as rooms, of absorbing sound energy. (3) The measure of the magnitude of the absorptive property of a material, an object, or a structure, such as a room.

Specific Heat: Specific heat (C_p) is the amount of heat energy in Btu's required to raise the temperature of one pound of substance one degree Fahrenheit. The following are specific heat values at standard conditions:

water—C_p = 1.00 Btu/lb/°F
air—C_p = 0.24 Btu/lb/°F

Using these values in simple equations, gallons per minute or cubic feet per minute may be determined in a system if the Btu per hour and the temperature difference are known.

Specific Volume: The reciprocal of density and is used to determine the cubic feet of volume, if the pounds of weight are known. Both density and specific volume are affected by temperature and pressure. The specific volume of air under standard conditions is 13.33 cubic feet per pound and the specific volume of water at standard conditions is 0.016 cubic feet per pound.

Spread: The divergence of the airstream in a horizontal or vertical plane after it leaves the outlet.

Stagnant Air Area: An area within a space where the air velocity is less than 25 fpm.

Standard Air Density (d): Standard air density has been set at 0.075 lb/cu. ft. This corresponds approximately to dry air at 70° F and 29.92 in. Hg. In metric units, the standard air density is 1.2041 kg m^3 at 20° C and at 101.325 kPa.

Standard Conditions: The standard conditions referred to in environmental system work for air are: *dry* air at 70°F, and at an atmospheric pressure of 29.92 inches mercury (in.Hg.). For water, standard conditions are 68°F at the same barometric pressure. At these standard conditions, the density of air is 0.075 pounds per cubic feet and the density of water is 62.4 pounds per cubic foot.

Standard Rating: A rating based on tests performed at Standard Rating Conditions.

State: Refers to the form of a fluid, either liquid, gas or solid. Liquids used in environmental systems are water, thermal fluids such as ethylene glycol solutions, and refrigerants in the liquid state. Gases are steam, evaporated refrigerants and the air-water vapor mixture found in the atmosphere. Some substances, including commonly used refrigerants, may exist in any of three states. A simple example is water, which may be solid (ice), liquid (water), or gas (steam or water vapor).

Static Head: The pressure due to the weight of a fluid above the point of measurement.

Static Regain Method: A method of duct sizing wherein the duct velocities are systematically reduced, allowing a portion of the velocity pressure to convert to static pressure offsetting the duct friction losses.

Static Suction Head: The positive vertical height in feet from the pump centerline to the top of the level of the liquid source.

Static Suction Lift: The distance in feet between the pump centerline and the source of liquid below the pump centerline.

Stratified Air: Unmixed air in a duct that is in thermal layers that have temperature variations of more than five degrees.

Structure-Borne Noise: A condition when the sound waves are being carried by a solid material. Sound waves in this state are inaudible to the human ear, since they cannot carry energy to it. Airborne noise can be created from the radiation of structure-borne noise into the air. Structure-borne noise may be propagated by shear waves, tension-compression waves, bending waves, or complicated combination of waves.

Suction Head: The positive pressure on the pump inlet when the source of liquid supply is above the pump centerline.

Suction Lift: The combination of static suction lift and friction head in the suction piping when the source of liquid is below the pump centerline.

Suction Piping: The piping which returns gaseous refrigerant to the compressor. Sizes must be large enough to maintain minimum friction to prevent reduced compressor and system capacity but must be small enough to produce adequate velocity to return oil to the compressor.

Sun Effect: Solar energy transmitted into space through windows and building materials.

System: A series of ducts, conduits, elbows, branch piping, etc. designed to guide the flow of air, gas or vapor to and from one or more locations. A fan provides the necessary energy to overcome the resistance to flow of the system and causes air or gas flow through the system. Some components of a typical system are louvers, grilles, diffusers, filters, heating and cooling coils, air pollution control devices, burner assemblies, volume flow control dampers, mixing boxes, sound attenuators, the ductwork and related fittings.

System Curve: A graphic presentation of the pressure vs. volume flow rate characteristics of a particular system.

System, Central Fan: A mechanical, indirect system of heating, ventilating, or air conditioning, in which the air is treated or handled by equipment located outside the rooms served, usually at a central location, and conveyed to and from the rooms by means of a fan and a system of distributing ducts.

System Effect Factor: A pressure loss factor which recognizes the effect of fan inlet restrictions, fan outlet restrictions, or other conditions influencing fan performance when installed in the system.

System, Gravity Circulation: A heating or refrigerating system in which the heating or cooling fluid circulation is effected by the motive head due to difference in densities of cooler and warmer fluids in the two sides of the system.

System, Run-around: A regenerative-type, closed, secondary system in which continuously circulated fluid abstracts heat from the primary system fluid at one place, returning this heat to the primary system fluid at another place.

System, Unitary: A complete, factory-assembled and factory-tested refrigerating system comprising one or more assemblies which may be shipped as one unit or separately but which are designed to be used together.

Temperature, Absolute Zero: The zero point on the absolute temperature scale, 459.69 degrees below the zero of the Fahrenheit scale, 273.16 degrees below the zero of the Celsius scale.

Temperature, Dewpoint: The temperature at which the condensation of water vapor in a space begins for a given state of humidity and pressure as the temperature of the vapor is reduced. The temperature corresponding to saturation (100 percent relative humidity) for a given absolute humidity at constant pressure.

Temperature, Dry bulb: The temperature of a gas or mixture of gases indicated by an accurate thermometer after correction for radiation.

Temperature, Effective: An arbitrary index which combines into a single value the effect of temperature, humidity, and air movement on the sensation of warmth or cold felt by the human body. The numerical value is that of the temperature of still, saturated air which would induce an identical sensation.

Temperature, Mean Radiant (MRT): The temperature of a uniform black enclosure in which a solid body or occupant would exchange the same amount of radiant heat as in the existing non-uniform environment.

Temperature, Wet-bulb: Thermodynamic wet-bulb temperature is the temperature at which liquid or solid water, by evaporating into air, can bring the air to saturation adiabatically at the same temperature. Wet-bulb temperature (without qualification is the temperature indicated by a wet-bulb psychrometer constructed and used according to specifications.

Terminal Velocity: The maximum airstream velocity at the end of the throw.

Thermal Expansion Valve: The metering device or flow control which regulates the amount of liquid refrigerant which is allowed to enter the evaporator.

Thermocouple: Device for measuring temperature utilizing the fact that an electromotive force is generated whenever two junctions of two dissimilar metals in an electric circuit are at different temperature levels.

Thermodynamics: The science of heat energy and its transformations to and from other forms of energy.

Thermodynamics, Laws of: Two laws upon which rest the classical theory of thermodynamics. These laws have been stated in many different, but equivalent ways.

The First Law: (1) When work is expended in generating heat, the quantity of heat produced is proportional to the work expended; and, conversely, when heat is employed in the performance of work, the quantity of heat which disappears is proportional to the work done (Joule); (2) If a system is caused to change from an initial state to a final state by adiabatic means only, the work done is the same for all adiabatic paths connecting the two states (Zemansky); (3) In any power cycle or refrigeration cycle, the net heat absorbed by the working substance is exactly equal to the net work done.

The Second Law: (1) It is impossible for a self-acting machine, unaided by any external agency, to convey heat from a body of lower temperature to one of higher temperature (Clausius); (2) It is impossible to derive mechanical work from heat taken from a body unless there is available a body of lower temperature into which the residue not so used may be discharged (Kelvin); (3) It is impossible to construct an engine that, operating in a cycle, will produce no effect other than the extraction of heat from a reservoir and the performance of an equivalent amount of work (Zemansky).

Throttling Range: The amount of change in the variable being controlled to make the controlled device move through the full length of its stroke.

Throw: The horizontal or vertical axial distance an airstream travels after leaving an air outlet before the maximum stream velocity is reduced to a specified terminal level; e.g., 200, 150, 100 or 50 fpm.

Total Dynamic Head: Dynamic discharge head (static discharge head, plus friction head, plus velocity head) plus dynamic suction lift, or dynamic discharge head minus dynamic suction head.

Total Heat (Enthalpy): Total heat is the sum of the sensible heat and latent heat in an exchange process. In many cases, the addition or subtraction of latent and sensible heat at terminal coils appears simultaneously. Total heat also is called *enthalpy,* both of which can be defined as the quantity of heat energy contained in that substance.

Total Pressure Method: A method of duct sizing which allows the designer to determine all friction and dynamic losses in each section of a duct system (including the total system).

Transducer: The means by which the controller converts the signal from the sensing device into the means necessary to have the appropriate effect on the controlled device. For example, a change in air pressure in the pneumatic transmission piping.

Transmission: The means by which a signal is moved from the point of origin to the point of action.

Transmission, Heat, Coefficient of: Any one of a number of coefficients used in the calculation of heat transmission by conduction, convection, and radiation, through various materials and structures.

Transmittance, Thermal (U factor): The time rate of heat flow per unit area under steady conditions from the fluid on the warm side of a barrier to the fluid on the cold side, per unit temperature difference between the two fluids.

Vane Ratio: In air distributing devices, the ratio of depth of vane to shortest opening width between two adjacent grille bars.

Vapor: A gas, particularly one near to equilibrium with the liquid phase of the substance and which does not follow the gas laws. Usually used instead of gas for a refrigerant, and, in general, for any gas below the critical temperature.

Vapor, Water: Used commonly in air conditioning parlance to refer to steam in the atmosphere.

Vapor Barrier: A moisture-impervious layer applied to the surfaces enclosing a humid space to prevent moisture travel to a point where it may condense due to lower temperature.

Vapor Pressure: Vapor pressure denotes the lowest absolute pressure that a given liquid at a given temperature will remain liquid before evaporating into its gaseous form or state.

Velocity: A vector quantity which denotes, at once, the time rate and the direction of a linear motion.

Velocity Head: The pressure needed to accelerate the fluid being pumped.

Velocity, Outlet: The average discharge velocity of primary air being discharged from the outlet, normally measured in the plane of the opening.

Velocity Reduction Method: A method of duct sizing where arbitrary reductions are made in velocity after each branch or outlet.

Velocity, Room: The average, sustained, residual air velocity level in the occupied zone of the conditioned space; e.g., 65, 50, 35 fpm.

Velocity, Terminal: The highest sustained airstream velocity existing in the mixed air path at the end of the throw.

Ventilation: The process of supplying or removing air, by natural or mechanical means, to or from any space. Such air may or may not have been conditioned.

Viscosity: That property of semifluids, fluids, and gases by virtue of which they resist an instantaneous change of shape or arrangement of parts. It is the cause of fluid friction whenever adjacent layers of fluid move with relation to each other.

Volatility: Volatility, surface tension and capillary action of a fluid are incidental to environmental systems. *Volatility* is the rapidity with which liquids evaporate. Gasoline, for example, evaporates extremely rapidly and therefore is highly volatile.

Volume, Specific: The volume of a substance per unit mass; the reciprocal of density.

Wavelength: The distance between two similar and successive points on an alternating wave. The wavelength is equal to the velocity of the propagation divided by the frequency of the alternations.

Weight: The amount of force a substance exerts under pull by the earth's gravitational field and that force is measured in pounds in the United States.

Wet Bulb Temperature (WB): The temperature registered by a thermometer whose bulb is covered by a saturated wick and exposed to a current of rapidly moving air. The wet bulb temperature also represents the dew point temperature of the air, where the moisture of the air condenses on a cold surface.

Wet bulb Depression: Difference between dry bulb and wet bulb temperatures.

CHAPTER 12 EQUATIONS (U.S. UNITS)

A AIR EQUATIONS

a) $V = 1096 \sqrt{\frac{V_p}{d}}$

or for standard air ($d = 0.075$ lb/cu ft):

$V = 4005 \sqrt{V_p}$

$\left(d = 1.325 \frac{P_b}{T}\right)$

V = Velocity (fpm)

V_p = Velocity Pressure (in. w.g.)

d = Density (lb/cu ft)

P_b = Absolute Static Pressure (in. Hg) (Barometric pressure + static pressure)

T = Absolute Temp. ($460° + °F = °R$)

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b) $Q\text{ (sens.)} = 60 \times C_p \times d \times cfm \times \Delta t$

or for standard air ($C_p = 0.24$ Btu/lb – °F):

$Q\text{ (sens.)} = 1.08 \times cfm \times \Delta t$

c) $Q\text{ (lat.)} = 4750 \times cfm \times \Delta W$

d) $Q\text{ (total)} = 4.5 \times cfm \times \Delta h$

e) $Q = A \times U \times \Delta t$

f) $R = \frac{1}{U}$

g) $\frac{P_1 V_1}{T_1} = \frac{P_2 V_2}{T_2} = RM$

Q = Heat Flow (Btu/hr)

C_p = Specific Heat (Btu/lb • °F)

d = Density (lb/cu ft)

Δt = Temperature Difference (°F)

ΔW = Humidity Ratio (lb H_2O/lb dry air)

Δh = Enthalpy Diff. (Btu/lb dry air)

A = Area of Surface (sq ft)

U = Heat Transfer Coefficient (Btu/sq ft • hr • °F)

R = Thermal Resistance (sq ft • hr • °F/Btu)

P = Absolute Pressure (lb/sq ft)

V = Total Volume (cu ft)

T = Absolute Temp. ($460° + °F = °R$)

R = Gas Constant (ft/°R)

M = Mass (lb)

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h) $TP = V_p + SP$

i) $V_p = \left(\frac{V}{4005}\right)^2$

j) $V = V_m \left[\frac{d\text{ (other than standard)}}{0.075\ (d = \text{std. air})}\right]$

k) $cfm = A \times V$

l) $TP = C \times V_p$

TP = Total Pressure (in. w.g.)

V_p = Velocity Pressure (in. w.g.)

SP = Static Pressure (in. w.g.)

V = Velocity (fpm)

V_m = Measured Velocity (fpm)

d = Density (lb/cu ft)

A = Area of duct cross section (sq ft)

C = Duct Fitting Loss Coefficient

B FAN EQUATIONS

a) $\frac{cfm_2}{cfm_1} = \frac{rpm_2}{rpm_1}$

b) $\frac{P_2}{P_1} = \left(\frac{rpm_2}{rpm_1}\right)^2$

c) $\frac{bhp_2}{bhp_1} = \left(\frac{rpm_2}{rpm_1}\right)^3$

d) $\frac{d_2}{d_1} = \left(\frac{rpm_2}{rpm_1}\right)^2$

e) $\frac{\text{rpm (fan)}}{\text{rpm (motor)}} = \frac{\text{Pitch diam. motor pulley}}{\text{Pitch diam. fan pulley}}$

cfm = Cubic feet per minute

rpm = Revolutions per minute

P = Static or Total Pressure (in. w.g.)

bhp = Brake horsepower

d = Density (lb/cu ft)

C PUMP EQUATIONS

a) $\frac{gpm_2}{gpm_1} = \frac{rpm_2}{rpm_1}$

b) $\frac{gpm_2}{gpm_1} = \frac{D_2}{D_1}$

c) $\frac{H_2}{H_1} = \left(\frac{rpm_2}{rpm_1}\right)^2$

d) $\frac{H_2}{H_1} = \left(\frac{D_2}{D_1}\right)^2$

e) $\frac{bhp_2}{bhp_1} = \left(\frac{rpm_2}{rpm_1}\right)^3$

f) $\frac{bhp_2}{bhp_1} = \left(\frac{D_2}{D_1}\right)^3$

gpm = Gallons per minute

rpm = Revolutions per minute

D = Impeller diameter

H = Head (ft. w.g.)

bhp = Brake horsepower

Table 12-1 HYDRONIC EQUIVALENTS

a. One gallon water = 8.33 pounds
b. Specific heat (C_p) water = 1.00 Btu/lb °F (@ 68°F)
c. Specific heat (C_p) water vapor = 0.45 Btu/lb °F (@ 68°F)
d. One ft. of water = 0.433 psi
e. One ft. of mercury (Hg) = 5.89 psi
f. One cu.ft. of water = 62.4 lb = 7.49 gal.
g. One in. of mercury (Hg) = 13.6 in.w.g. = 1.13 ft. w.g.
h. Atmospheric Pressure = 29.92 in.Hg = 14.696 psi
i. One psi = 2.31 ft. w.g. = 2.04 in.Hg

D HYDRONIC EQUATIONS

a) $Q = 500 \times gpm \times \Delta t$

b) $\dfrac{\Delta P_2}{\Delta P_1} = \left(\dfrac{gpm_2}{gpm_1}\right)^2$

c) $\Delta P = \left(\dfrac{gpm}{C_v}\right)^2$

gpm = Gallons per minute

Q = Heat flow (Btu/hr)

Δt = Temperature diff. (°F)

ΔP = Pressure diff. (psi)

C_v = Valve constant (dimensionless)

d) $whp = \dfrac{gpm \times H \times Sp.\,Gr.}{3960}$

e) $bph = \dfrac{gpm \times H \times Sp.\,Gr.}{3960 \times E_p} = \dfrac{whp}{E_p}$

f) $E_p = \dfrac{whp \times 100}{bhp}$ (in percent)

whp = Water horsepower

gpm = Gallons per minute

bhp = Brake horsepower

H = Head (ft w.g.)

Sp. Gr. = Specific gravity (use 1.0 for water)

E_p = Efficiency of pump

g) $NPSHA = P_a \pm P_s + \dfrac{V^2}{2g} - P_{vp}$

h) $h = f \times \dfrac{L}{D} \times \dfrac{V^2}{2g}$

NPSHA = Net positive suction head available

P_a = Atm. press. (use 34 ft w.g.)

P_s = Pressure at pump centerline (ft w.g.)

$\dfrac{V^2}{2g}$ = Velocity head at point P_s (ft w.g.)

P_{vp} = Absolute vapor pressure (ft w.g.)

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

h = Head loss (ft)

f = Friction factor (dimensionless)

L = Length of pipe (ft)

D = Internal diameter (ft)

V = Velocity (ft/sec)

Table 12-2 CONVERTING PRESSURE IN INCHES OF MERCURY TO FEET OF WATER AT VARIOUS WATER TEMPERATURES

Water Temperature degrees F	60°	150°	200°	250°	300°	340°
Ft. head differential per in. Hg. differential	1.046	1.07	1.09	1.11	1.15	1.165

E ELECTRIC EQUATIONS

a) $\text{Bhp} = \dfrac{\text{I x E x P.F. x Eff.}}{746}$ (Single Phase)

b) $\text{Bhp} = \dfrac{\text{I x E x P.F. x Eff. x 1.73}}{746}$ (Three Phase)

c) $E = IR$

d) $P = EI$

e) $\dfrac{\text{F.L. Amps* x Voltage*}}{\text{Actual Voltage}} = \text{Actual F.L. Amps}$

f) $\text{Bhp} = \text{HP*} \times \dfrac{(\text{Motor operating amps}) - (\text{No load amps} \times 0.5)}{(\text{Actual F.L. amps}) - (\text{No load amps} \times 0.5)}$

I = Amps (A)
E = Volts (V)
P.F. = Power factor
R = ohms (Ω)
P. = watts (W)
Bhp = Brake horsepower

*Nameplate ratings

Table 12-3 AIR DENSITY CORRECTION FACTORS (U.S. UNITS)

Altitude (ft)	Sea Level	1000	2000	3000	4000	5000	6000	7000	8000	9000	10,000
Barometer (in.Hg)	29.92	28.86	27.82	26.82	25.84	24.90	23.98	23.09	22.22	21.39	20.58
Barometer (in.w.g.)	407.5	392.8	378.6	365.0	351.7	338.9	326.4	314.3	302.1	291.1	280.1
Air Temp. °F -40°	1.26	1.22	1.17	1.13	1.09	1.05	1.01	0.97	0.93	0.90	0.87
0°	1.15	1.11	1.07	1.03	0.99	0.95	0.91	0.89	0.85	0.82	0.79
40°	1.06	1.02	0.99	0.95	0.92	0.88	0.85	0.82	0.79	0.76	0.73
70°	1.00	0.96	0.93	0.89	0.86	0.83	0.80	0.77	0.74	0.71	0.69
100°	0.95	0.92	0.88	0.85	0.81	0.78	0.75	0.73	0.70	0.68	0.65
150°	0.87	0.84	0.81	0.78	0.75	0.72	0.69	0.67	0.65	0.62	0.60
200°	0.80	0.77	0.74	0.71	0.69	0.66	0.64	0.62	0.60	0.57	0.55
250°	0.75	0.72	0.70	0.67	0.64	0.62	0.60	0.58	0.56	0.58	0.51
300°	0.70	0.67	0.65	0.62	0.60	0.58	0.56	0.54	0.52	0.50	0.48
350°	0.65	0.62	0.60	0.58	0.56	0.54	0.52	0.51	0.49	0.47	0.45
400°	0.62	0.60	0.57	0.55	0.53	0.51	0.49	0.48	0.46	0.44	0.42
450°	0.58	0.56	0.54	0.52	0.50	0.48	0.46	0.45	0.43	0.42	0.40
500°	0.55	0.53	0.51	0.49	0.47	0.45	0.44	0.43	0.41	0.39	0.38
550°	0.53	0.51	0.49	0.47	0.45	0.44	0.42	0.41	0.39	0.38	0.36
600°	0.50	0.48	0.46	0.45	0.43	0.41	0.40	0.39	0.37	0.35	0.34
700°	0.46	0.44	0.43	0.41	0.39	0.38	0.37	0.35	0.34	0.33	0.32
800°	0.42	0.40	0.39	0.37	0.36	0.35	0.33	0.32	0.31	0.30	0.29
900°	0.39	0.37	0.36	0.35	0.33	0.32	0.31	0.30	0.29	0.28	0.27
1000°	0.36	0.35	0.33	0.32	0.31	0.30	0.29	0.28	0.27	0.26	0.25

Standard Air Density, Sea Level, 70°F 0.075 lb/cu ft at 29.92 in. Hg

F GEOMETRIC EQUATIONS

1. Right Triangle

$a^2 + b^2 = c^2$

$\text{Sine } \Theta = \dfrac{a}{c}$

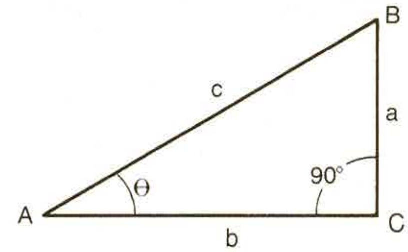
$\text{Cosine } \Theta = \dfrac{b}{c}$

$\text{Tangent } \Theta = \dfrac{a}{b}$

$\text{Cotangent } \Theta = \dfrac{b}{a}$

$\text{Area} = \dfrac{a \times b}{2}$

Circumference = sum of sides



NATIONAL TRIGONOMETRIC FUNCTIONS				
Angle/degrees	SIN	COS	TAN	COT
15°	0.2588	0.9659	0.2680	3.7320
30°	0.5000	0.8660	0.5774	1.7320
45°	0.7071	0.7071	1.0000	1.0000
60°	0.8660	0.5000	1.7320	0.5774
75°	0.9659	0.2588	3.7320	0.2680

2. Circle

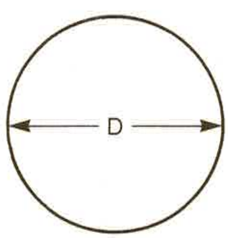
Circumference (C):

$C = \pi D$

$\pi = 3.1416$

Area (A):

$A = \dfrac{\pi D^2}{4}$



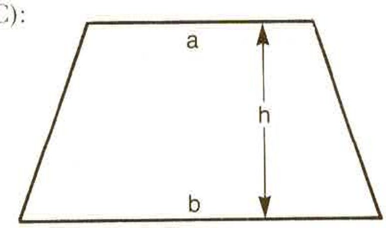
3. Trapezoid

Circumference (C):

Sum of sides

Area (A):

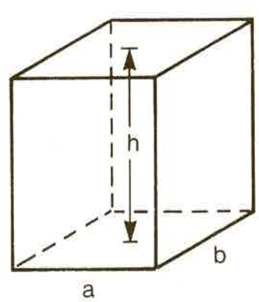
$A = \dfrac{h\,(a + b)}{2}$



4. Prism

Volume (V):

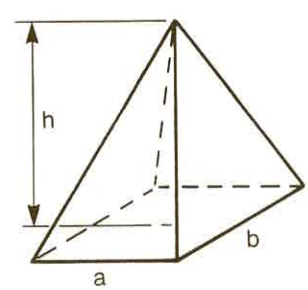
$V = a \times b \times h$



5. Pyramid

Volume (V):

$V = \dfrac{a \times b \times h}{3}$



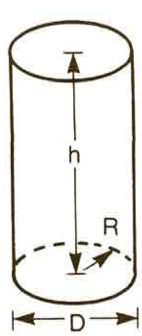
6. Cylinder

Volume (V):

$V = \dfrac{\pi D^2 h}{4}$

or

$V = \pi R^2 h$



7. Cone

Volume (V):

$V = \dfrac{\pi D^2 h}{12}$

or

$V = \dfrac{\pi R^2 h}{3}$

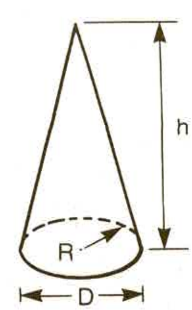


Table 12-4 VELOCITIES VS. VELOCITY PRESSURES

Velocity fpm	Velocity Pressure in. wg.	Velocity fpm	Velocity Pressure in. wg.	Velocity fpm	Velocity Pressure in. wg.	Velocity fpm	Velocity Pressure in. wg.	Velocity fpm	Velocity Pressure in. wg.
300	0.01	2050	0.26	3800	0.90	5550	1.92	7300	3.32
350	0.01	2100	0.27	3850	0.92	5600	1.95	7350	3.37
400	0.01	2150	0.29	3900	0.95	5650	1.99	7400	3.41
450	0.01	2200	0.30	3950	0.97	5700	2.02	7450	3.46
500	0.02	2250	0.32	4000	1.00	5750	2.06	7500	3.51
550	0.02	2300	0.33	4050	1.02	5800	2.10	7550	3.55
600	0.02	2350	0.34	4100	1.05	5850	2.13	7600	3.60
650	0.03	2400	0.36	4150	1.07	5900	2.17	7650	3.65
700	0.03	2450	0.37	4200	1.10	5950	2.21	7700	3.70
750	0.04	2500	0.39	4250	1.13	6000	2.24	7750	3.74
800	0.04	2550	0.41	4300	1.15	6050	2.28	7800	3.79
850	0.05	2600	0.42	4350	1.18	6100	2.32	7850	3.84
900	0.05	2650	0.44	4400	1.21	6150	2.36	7900	3.89
950	0.06	2700	0.45	4450	1.23	6200	2.40	7950	3.94
1000	0.06	2750	0.47	4500	1.26	6250	2.43	8000	3.99
1050	0.07	2800	0.49	4550	1.29	6300	2.47	8050	4.04
1100	0.08	2850	0.51	4600	1.32	6350	2.51	8100	4.09
1150	0.08	2900	0.52	4650	1.35	6400	2.55	8150	4.14
1200	0.09	2950	0.54	4700	1.38	6450	2.59	8200	4.19
1250	0.10	3000	0.56	4750	1.41	6500	2.63	8250	4.24
1300	0.11	3050	0.58	4800	1.44	6550	2.67	8300	4.29
1350	0.11	3100	0.60	4850	1.47	6600	2.71	8350	4.35
1400	0.12	3150	0.62	4900	1.50	6650	2.76	8400	4.40
1450	0.13	3200	0.64	4950	1.53	6700	2.80	8450	4.45
1500	0.14	3250	0.66	5000	1.56	6750	2.84	8500	4.50
1550	0.15	3300	0.68	5050	1.59	6800	2.88	8550	4.56
1600	0.16	3350	0.70	5100	1.62	6850	2.92	8600	4.61
1650	0.17	3400	0.72	5150	1.65	6900	2.97	8650	4.66
1700	0.18	3450	0.74	5200	1.69	6950	3.01	8700	4.72
1750	0.19	3500	0.76	5250	1.72	7000	3.05	8750	4.77
1800	0.20	3550	0.79	5300	1.75	7050	3.10	8800	4.83
1850	0.21	3600	0.81	5350	1.78	7100	3.14	8850	4.88
1900	0.22	3650	0.83	5400	1.82	7150	3.19	8900	4.94
1950	0.24	3700	0.85	5450	1.85	7200	3.23	8950	4.99
2000	0.25	3750	0.88	5500	1.89	7250	3.28	9000	5.05

$$\text{Velocity} = 4005\sqrt{V_p} \text{ (or) } V_p = \left(\frac{\text{Velocity}}{4005}\right)^2$$

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CHAPTER 13 EQUATIONS (METRIC UNITS)

A AIR EQUATIONS

a) $V = 1.414\sqrt{\frac{V_p}{d}}$

or for standard air ($d = 1.2\ kg/m^3$):

$V = \sqrt{1.66\ Vp}$

$\left(d = 3.48\ \frac{P_b}{T}\right)$

V = Velocity (m/s)

V_p = Velocity Pressure (pascals or Pa)

d = Density (kg/m^3)

P_b = Absolute Static Pressure (kPa)
(Barametric pressure + static pressure)

T = Absolute Temp. (273° + °C = °K)

---

b) $Q = C_p \times d \times l/s \times \Delta t$

or for standard air ($C_p = 1.005\ kJ/kg \cdot °C$)

$Q\ (sens.) = 1.23 \times l/s \times \Delta t$

c) $Q\ (lat.) = 3.0 \times l/s \times \Delta W$

d) $Q\ (total\ heat) = 1.20 \times l/s \times \Delta h$

e) $Q = A \times U \times \Delta t$

f) $R = \frac{1}{U}$

g) $\frac{P_1V_1}{T_1} = \frac{P_2V_2}{T_2} = RM$

Q = Heat Flow (watts or kilowatts)

C_p = Specific Heat ($kJ/kg \cdot °C$)

d = Density (kg/m^3)

Δt = Temperature Difference (°C)

ΔW = Humidity Ratio (g H_2O/kg dry air)

Δh = Enthalpy Diff. (kJ/kg dry air)

A = Area of Surface (m^2)

U = Heat Transfer Coefficient ($W/m^2 \cdot °C$)

R = Thermal Resistance ($m^2 \cdot °C/W$)

P = Absolute Pressure (kPa)

V = Total Volume (m^3)

T = Absolute Temperature (273° + °C = °K)

R = Gas Constant ($kJ/kg \cdot °C$)

M = Mass (kg)

---

h) $TP = V_p + SP$

i) $V_p = \left(\frac{V}{1.30}\right)^2 = \frac{d}{2} \times V^2$

j) $V = V_m\left[\frac{d\ (other\ than\ standard)}{1.2\ (d = std.\ air)}\right]$

k) $l/s = 1000 \times A \times V$

l) $TP = C \times V_p$

TP = Total Pressure (Pa)

V_p = Velocity Pressure (Pa)

SP = Static Pressure (Pa)

V = Velocity (m/s)

V_m = Measured Velocity (m/s)

d = Density (kg/m^3)

A = Area of duct cross section (m^2)

C = Duct Fitting Loss Coefficient

B FAN EQUATIONS

a) $\frac{l/s_2}{l/s_1} = \frac{m^3/s_2}{m^3/s_1} = \frac{rad/s_2}{rad/s_1}$

b) $\frac{P_2}{P_1} = \left(\frac{rad/s_2}{rad/s_1}\right)^2$

c) $\frac{kW_2}{kW_1} = \left(\frac{rad/s_2}{rad/s_1}\right)^3$

d) $\frac{d_2}{d_1} = \left(\frac{rad/s_2}{rad/s_1}\right)^2$

e) $\frac{\text{rad/s (fan)}}{\text{rad/s (motor)}} = \frac{\text{Pitch diam. motor pulley}}{\text{Pitch diam. fan pulley}}$

l/s = Litres per second
m³/s = Cubic metres per second
rad/s = Radians per second

P = Static or Total Pressure (Pa)
kW = Kilowatts
d = Density (kg/m³)

C PUMP EQUATIONS

a) $\frac{l/s_2}{l/s_1} = \frac{m^3/s_2}{m^3/s_1} = \frac{rad/s_2}{rad/s_1}$

b) $\frac{m^3/s_2}{m^3/s_1} = \frac{D_2}{D_1}$

c) $\frac{H_2}{H_1} = \left(\frac{rad/s_2}{rad/s_1}\right)^2$

d) $\frac{H_2}{H_1} = \left(\frac{D_2}{D_1}\right)^2$

e) $\frac{BP_2}{BP_1} = \left(\frac{rad/s_2}{rad/s_1}\right)^3$

f) $\frac{BP_2}{BP_1} = \left(\frac{D_2}{D_1}\right)^3$

l/s = Litres per second

m³/s = Cubic metres per second

rad/s = Radians per second

D = Impeller diameter

H = Head (kPa)

BP = Brake horsepower

D HYDRONIC EQUATIONS

a) $Q = 4190 \times m^3/s \times \Delta t$

b) $\dfrac{\Delta P_2}{\Delta P_1} = \left(\dfrac{m^3/s_2}{m^3/s_1}\right)^2 = \left(\dfrac{l/s_2}{l/s_1}\right)^2$

c) $\Delta P = \left(\dfrac{m^3/s}{C_v}\right)^2 = \left(\dfrac{l/s}{C_v}\right)^2$

Q = Heat flow (kilowatts)

Δt = Temperature difference (°C)

m^3/s (used for large volumes) = Cubic metres per second

l/s = Litres per second

ΔP = Pressure diff. (Pa or kPa)

C_v = Valve constant (dimensionless)

---

d) $WP\ (kW) = 9.81 \times m^3/s \times H(m) \times Sp.\ Gr.$

$WP\ (W) = \dfrac{l/s \times H(Pa) \times Sp.\ Gr.}{1002}$

e) $BP = \dfrac{WP}{E_p}$

f) $E_p = \dfrac{WP \times 100}{BP}$ (in percent)

WP = Water power (kW or (W)

m^3/s = Cubic metres per second

l/s = Litres per second

Sp. Gr. = Specific gravity (use 1.0 for water)

BP = Brake power (kW)

E_p = Efficiency of Pump

H = Head (Pa) or (m)

---

g) $NPSHA = P_a \pm P_s + \dfrac{V^2}{2g} - P_{vp}$

h) $h = f \times \dfrac{L}{D} \times \dfrac{V^2}{2g}$

NPSHA = Net positive suction head available

P_a = Atm. press. (Pa)
(Std. Atm. press. = 101,325 Pa)

P_s = Pressure at pump centerline (Pa)

$\dfrac{V^2}{2g}$ Velocity head at point P_s (m)

P_{vp} = Absolute vapor pressure (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)

E ELECTRIC EQUATIONS

a) $kW = \dfrac{I \times E \times P.F. \times Eff.}{1000}$ (Single Phase)

b) $kW = \dfrac{I \times E \times P.F. \times Eff. \times 1.73}{1000}$ (Three Phase)

c) $E = IR$

d) $P = EI$

e) $\dfrac{F.L.\ Amps^{*} \times Voltage^{*}}{Actual\ Voltage} = Actual\ F.L.\ Amps$

f) $Bhp = HP^{*} \times \dfrac{(Motor\ operating\ amps) - (No\ load\ amps \times 0.5)}{(Actual\ F.L.\ amps) - (No\ load\ amps \times 0.5)}$

*Nameplate ratings

kW = Kilowatts
I = Amps (A)
E = Volts (V)
P.F. = Power factor
R = ohms (Ω)
P. = watts (W)

TABLE 13-1 AIR DENSITY CORRECTION FACTORS (METRIC UNITS)

Altitude (m)	Sea Level	250	500	750	1000	1250	1500	1750	2000	2500	3000
Barometer (kPa)	101.3	98.3	96.3	93.2	90.2	88.2	85.1	83.1	80.0	76.0	71.9
Air Temp. °C 0°	1.08	1.05	1.02	0.99	0.96	0.93	0.91	0.88	0.86	0.81	0.76
20°	1.00	0.97	0.95	0.92	0.89	0.87	0.84	0.82	0.79	0.75	0.71
50°	0.91	0.89	0.86	0.84	0.81	0.79	0.77	0.75	0.72	0.68	0.64
75°	0.85	0.82	0.80	0.78	0.75	0.73	0.71	0.69	0.67	0.63	0.60
100°	0.79	0.77	0.75	0.72	0.70	0.68	0.66	0.65	0.63	0.59	0.56
125°	0.74	0.72	0.70	0.68	0.66	0.64	0.62	0.60	0.59	0.55	0.52
150°	0.70	0.68	0.66	0.64	0.62	0.60	0.59	0.57	0.55	0.52	0.49
175°	0.66	0.64	0.62	0.62	0.59	0.57	0.55	0.54	0.52	0.49	0.46
200°	0.62	0.61	0.59	0.57	0.56	0.54	0.52	0.51	0.49	0.47	0.44
225°	0.59	0.58	0.56	0.54	0.53	0.51	0.50	0.48	0.47	0.44	0.42
250°	0.56	0.55	0.53	0.52	0.50	0.49	0.47	0.46	0.45	0.42	0.40
275°	0.54	0.52	0.51	0.49	0.48	0.47	0.45	0.44	0.43	0.40	0.38
300°	0.51	0.50	0.49	0.47	0.46	0.45	0.43	0.42	0.41	0.38	0.36
325°	0.49	0.48	0.47	0.45	0.44	0.43	0.41	0.40	0.39	0.37	0.35
350°	0.47	0.46	0.45	0.43	0.42	0.41	0.40	0.39	0.38	0.35	0.33
375°	0.46	0.44	0.43	0.42	0.41	0.39	0.38	0.37	0.36	0.34	0.32
400°	0.44	0.43	0.41	0.40	0.39	0.38	0.37	0.36	0.35	0.33	0.31
425°	0.42	0.41	0.40	0.39	0.38	0.37	0.35	0.34	0.33	0.32	0.30
450°	0.41	0.40	0.38	0.37	0.36	0.35	0.34	0.33	0.32	0.31	0.29
475°	0.39	0.38	0.37	0.36	0.35	0.34	0.33	0.32	0.31	0.29	0.28
500°	0.38	0.37	0.36	0.35	0.34	0.33	0.32	0.31	0.30	0.28	0.27
525°	0.37	0.36	0.35	0.34	0.33	0.32	0.31	0.30	0.29	0.27	0.26

Standard Air Density, Sea Level, 20°C = 1.2041 kg/m³ at 101.325 kPa

F METRIC UNITS AND EQUIVALENTS

TABLE 13-2 METRIC UNITS (BASIC & DERIVED)

Unit	Symbol	Quantity	Equivalent or Relationship
ampere	A	Electric current	Same as U.S.
candela	cd	Luminous intensity	1 cd/m^2 = 0.292 ft lamberts
Celsius	°C	Temperature	°F = 1.8 °C + 32°
coulomb	C	Electric charge	Same as U.S.
farad	F	Electric capacitance	Same as U.S.
henry	H	Electric inductance	Same as U.S.
hertz	Hz	Frequency	Same as cycles per second
joule	J	Energy, work, heat	1 J = 0.7376 ft-lb = 0.000948 Btu
kelvin	K	Thermodynamic temperature	°K = °C + 273.15° $= \frac{°F + 459.67}{1.8}$
kilogram	kg	Mass	1 kg = 2.2046 lb
litre	l	Liquid volume	1 l = 1.056qt = 0.264 gal
lumens	lm	Luminous flux	1 lm/m^2 = 0.0929 ft candles
lux	lx	Illuminance	1 lx = 0.0929 ft candles
metre	m	Length	1 m = 3.281 ft
mole	mol	Amount of substance	—
newton	N	Force	1 N = kg m/s^2 = 0.2248 lb (force)
ohm	Ω	Electrical resistance	Same as U.S.
pascal	Pa	Pressure, stress	1 Pa = N/m^2 = 0.000145 psi = 0.004022 in. w.g.
radian	rad	Plane angle	1 rad = 57.29°
second	s	Time	Same as U.S.
siemens	S	Electric conductance	—
steradian	sr	Solid angle	—
volt	V	Electric potential	Same as U.S.
watt	W	Power, heat flow	1 W = J/s = 3.412 Btu/hr 1 W = 0.000284 tons of refrig.

TABLE 13-3 METRIC EQUIVALENTS

Quantity	Symbol	Unit	U.S. Relationship
acceleration	m/s^2	metres per second squared	$1\ m/s^2 = 3.281\ ft/sec^2$
angular velocity	rad/s	radians per second	1 rad/sec = 9.549 rpm
area	m^2	square metre	$1\ m^2 = 10.76$ sq ft
atmospheric pressure	—	101.325 kPa	29.92 in Hg = 14.696 psi
density	kg/m^3	kilograms per cubic metre	$1\ kg/m^3 = 0.0624$ lb/cu ft
density, air	—	$1.2\ kg/m^3$	0.075 lb/cu ft
density, water	—	$1000\ kg/m^3$	62.4 lb/cu ft
enthalpy	kJ/kg	kilojoule per kilogram	1 kJ/kg = 0.4299 Btu/lb dry air
gravity		$9.8067\ m/s^2$	$32.2\ ft/sec^2$
heat flow	W	watt	1 W = 3.412 Btu/hr
length (normal)	m	metre	1 m = 3.281 ft
length (large)	km	kilometre	1 km = 0.6214 miles
linear velocity	m/s	metres per second	1 m/s = 196.9 fpm = 2.237 mph
mass flow rate	kg/s	kilograms per second	1 kg/s = 7936.6 lb/hr
moment of inertia	$kg{\cdot}m^2$	kilograms x square metre	$1\ kg \cdot m^2 = 23.73\ lb \cdot sq\ ft$
power	W	watt	1 W = 0.00134 hp
pressure	kPa	kilopascal	1 kPa = 0.296 in Hg = 4.015 in w.g.
specific heat-air (C_p)		1000 J/kg • °C	1000 J/kg·°C = 1 kJ/kg • °C = 0.2388 Btu/lb °F
specific heat-air (C_v)		717 J/kg • °C	0.17 Btu/lb° F
specific heat-water		4190 J/kg • °C	1.0 Btu/lb° F
specific volume	m^3/kg	cubic metres per kilogram	$1\ m^3/kg = 16.019$ cu ft/lb
thermal conductivity	$W \cdot mm/m^2 \cdot °C$	watt millimetre per square metre °C	$1\ W \cdot mm/m^2 \cdot °C$ $= 0.0069\ Btu \cdot in/ft^2 \cdot hr \cdot °F$
volume flow rate	m^3/s l/s	cubic metres per second litres per second $1 m^3/s = 1000$ l/s 1 ml = litres/1000	$1\ m^3/s = 2118.88$ cfm (air). 1 l/s = 2.12 cfm (air) $1\ m^3/s = 15{,}850$ gpm (water) 1 ml/s = 1.05 gph (water)

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CHAPTER 14 TAB MATHEMATICS AND EQUATIONS

A Introduction

This booklet provides the basic mathematical operations that must be used to make the calculations required for testing, adjusting, and balancing (TAB) work. You must be able to use and understand simple algebraic equations including ratios, proportions, percentages, powers and roots. Many of you may not have used them for a long time, so this will be considered a "refresher course."

However for some of you, this will be your first exposure to the step above everyday mathematics. So review each section thoroughly before going on to the next. Answers to the questions may be found at the back of the booklet.

1. CALCULATORS

A pocket calculator should be used to work the problems, as they save time and are very accurate if used correctly. Most have a memory and a square root ($\sqrt{\ }$) key.

After you get your calculator, be sure you understand all its uses. Read, study, and practice the manufacturer's instructions—all vary. You will save much time if you understand all the applications possible. Especially practice the memory functions, because they will certainly be helpful on the job. Most persons learn only the basic arithmetic functions and never learn all the other useful functions available. New calculators, small enough to carry on the job, are available with printing capabilities. Consider one of these because a printout of your calculations is often extremely valuable.

2. SLIDE RULES (Optional Instrument)

A slide rule to use in setting a proportion and determining many different values of that proportion. The slide rule can be set to the proportion, and any value of that proportion can be seen at the same time without any further setting. The slide rule is comparatively inexpensive and will take hard use. The rule should contain A, B, C, D, K, CF, and DF scales. The K scale will allow you to do cube root, which is normally not on a calculator. A good quality, 6″ slide rule in a leather case will perform all the calculations you need, and it can be carried in your shirt pocket. Round slide rules do the same job, but some find them harder to use on an occasional basis.

B Basic Mathematics

1. CONVERTING DECIMALS TO FRACTIONS

Converting decimals to fractions is a fundamental operation which every one must understand. It usually is easier to make mathematical calculations with decimals than it is with fractions. Also, if calculations are made with an electronic calculator, the answer is in a decimal. However, the measuring rule you use is divided into sixteenths or thirty-seconds of an inch—not into decimals of an inch. This section will show you how to convert any decimal into the nearest sixteenth of an inch—or into any fraction you wish. It will also show you how to convert a fraction to the equivalent decimal.

This method of converting a decimal to a fraction is approximate because you are finding the desired fraction that is closest to the decimal. For example, the correct fractional equivalent of 0.568 is $^{21}/_{37}$. But $^{21}/_{37}$ is a useless fraction, as far as measuring with a standard rule is concerned. However, 0.568 is equal to only slightly more than $^{9}/_{16}$, and $^{9}/_{16}$″ is usually close enough for our measuring.

To convert a decimal to a fraction:

(1) Multiply the decimal by the denominator (the number under the line) of the fraction desired (Figure 1).
(2) Round off to the nearest whole number.
(3) Use that number as the numerator (the number above the line) of the desired fraction.

EXAMPLE A

Convert 0.692 to the nearest sixteenth.

SOLUTION:

0.692×16 (desired denominator) $= 11.072$

11.072 rounded to a whole number $= 11$

Therefore, the fraction to the nearest sixteenth is $^{11}/_{16}$.

EXAMPLE B

Convert 2.771 to the nearest thirty-second.

SOLUTION:

$0.771 \times 32 = 24.67$

24.67 rounded off is 25

The fraction to the nearest thirty-second is $2^{25}/_{32}$.

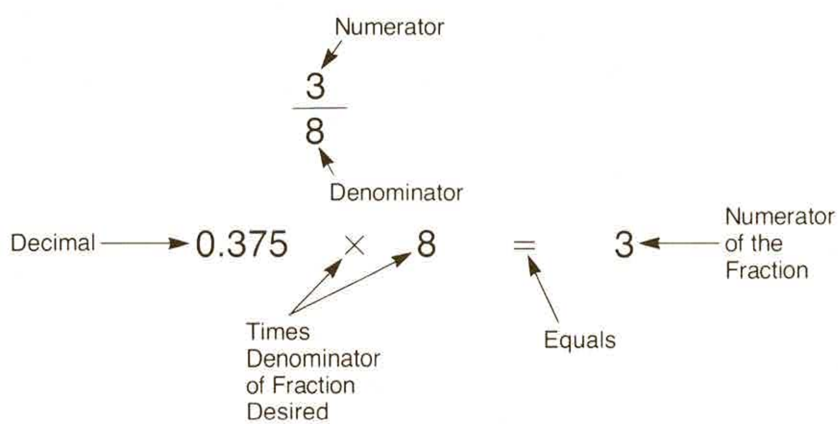


Figure 1. NUMERATOR AND DENOMINATOR

2. CONVERTING FRACTIONS TO DECIMALS

Fractions can be converted to decimals simply by dividing the bottom of the fraction into the top. For example, to convert ¼ to a decimal, divide 4 into 1:

$$\frac{1}{4} = 0.25$$

EXAMPLE C

Convert 7/16 into the decimal equivalent.

SOLUTION:

$7 \div 16 = 0.4375$

EXAMPLE D

Convert 2/9 to a decimal.

SOLUTION:

$2 \div 9 = 0.222\ldots$

Clearly, this calculation could go on indefinitely. This means that 2/9 is a repeating decimal. It will never end. The three dots mean "and so on to infinity." When using repeating decimals, they are generally rounded off to three places.

Here are some more examples of repeating decimals:

$\frac{1}{3} = 0.3333\ldots$ use 0.333

$\frac{2}{3} = 0.6666\ldots$ use 0.667

$\frac{5}{9} = 0.5555\ldots$ use 0.556

3. CIRCUMFERENCE

The circumference is the distance around a circle. The other parts of a circle are illustrated in Figure 2, which includes the diameter, the radius, and the constant π. The Greek letter Pi (π) stands for the constant 3.1416. A "constant" means that this number is the same under all conditions whenever the equation is used.

The constant π is the ratio of any circle's circumference to its diameter. In other words:

EQUATION 1

$$\pi = \frac{\text{Circumference (C)}}{\text{Diameter (D)}}$$

The circumference of a circle can be found by multiplying the diameter of a circle by π and then founding off the answer to two decimal places.

EQUATION 2

$C = \pi D$ (or)

$C = 3.1416 \times \text{Diameter}$

Note that 3.14 is often used when calculations are made with pencil and paper. However calculators are based on 3.1416 because this is the number programmed into the calculator. In this text we will use π as equal to 3.1416.

EXAMPLE E

Find the circumference of a circle with a 6″ diameter.

SOLUTION:

$C = 3.1416 \times 6''$

$C = 18.8496$ inches

When calculating the circumference of a circle, it is best to use decimals when multiplying and then convert them into fractions when fractions are needed.

$(18.8496'' = 18\tfrac{7}{8}'')$

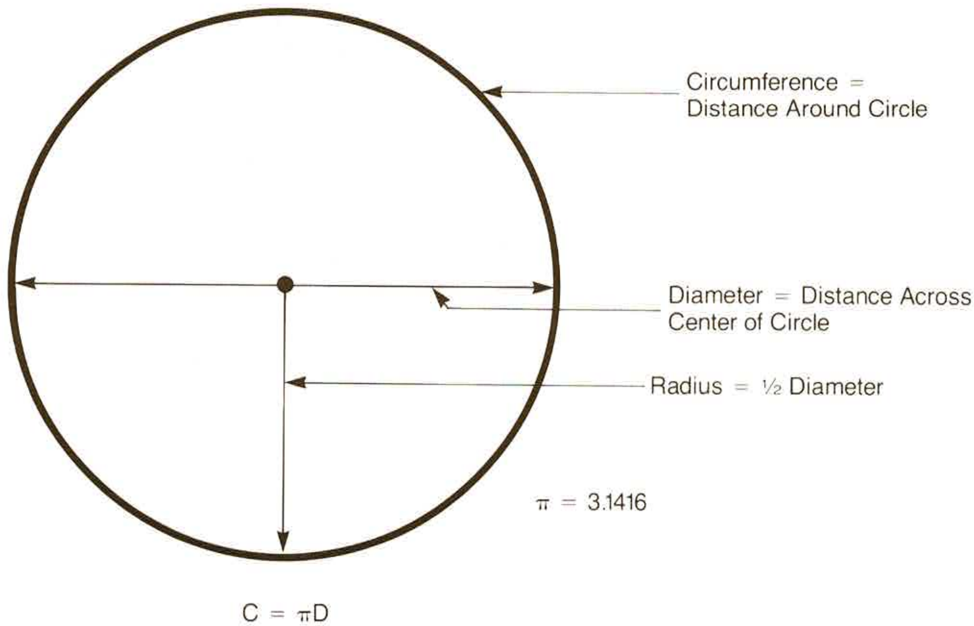


Figure 2 PARTS OF A CIRCLE

4. POWERS

Figure 3 illustrates basic terms which will be discussed in detail in the following pages.

Power means the number of times that a base number is multiplied by itself to get the product number.

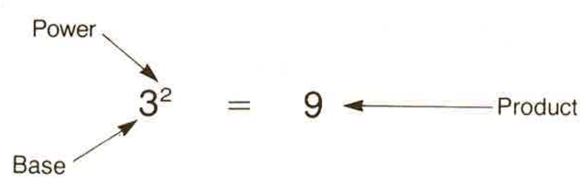


Figure 3 BASIC TERMS

a. Squared Numbers

To *square* a number means to multiply it by itself. Thus "three squared" is $3 \times 3 = 9$. Another way of saying this is "the square of three is nine." In equations, a squared number is indicated by a small two above and to the right of the numeral. For example 4^2 means 4 squared, or 4×4. Similarly, 7^2 means 7 squared, or 7×7.

This method of placing the smaller numeral over the larger numeral is used to show any number of multiplications. For example, 4^3 means four multiplied by itself three times ($4 \times 4 \times 4$). This is called either "four cubed" or "four to the third power."

Parentheses are also used with these powers, as shown below:

$$10 - 3^2 = 10 - 9 = 1$$

$$(10 - 3)^2 = (7)^2 = 49$$

The first equation states that 10 minus 3×3 equals 1.

The second equation states that $10 - 3$ is 7, and 7^2, or 7×7 equals 49. Remember, the work inside the parentheses must be done first.

Calculators are especially advantageous in figuring squares because of their speed and accuracy.

b. Powers

The *power* of a number means the times that a number is multiplied by itself. Eight squared (8^2) can also be stated as 8 to the power of 2. This is the same as 8^2 or $8 \times 8 = 64$.

Powers are written with a small number above and to the right of the number (the small number is called an exponent).

6^2 is 6 to the power of 2 (six squared)

$$6^2 = 6 \times 6$$

$$6^2 = 36$$

The power can be any number. The most commonly used are 2 (square) and 3 (cube).

- $8^3 = 512$ ($8 \times 8 \times 8$)
- 8^4 means "8 to the 4th power" ($8 \times 8 \times 8 \times 8$) = 4096
- The square of 8 (8^2) is 64
- The cube of 8 (8^3) is 512

Powers and roots are used in many equations. A simple example is calculating the area of round duct. The equation $A = \pi R^2$ means "the area of a circle equals 3.1416 times the radius squared.

The velocity pressure in a duct is calculated by the equation:

$$VP = \left(\frac{V}{4005}\right)^2$$

VP = Velocity Pressure in Inches Water Gage (in. w.g.)

V = Velocity in Feet per Minute (fpm)

This equation, rewritten to determine air velocity, is:

$$V = 4005 \sqrt{\text{Velocity Pressure}}$$

This means, "Velocity equals 4005 times the square root of the Velocity Pressure."

5. ROOTS

a. Square Root

Finding the square root of a given number is the process of determining what number, when multiplied by itself equals the given number. The symbol for square root is $\sqrt{\ }$. The statement $\sqrt{16} = 4$ means that 4 is the number which when multiplied by itself equals 16. Square root can be found on a slide rule or on an electronic calculator that has a $\sqrt{\ }$ key. The calculator makes finding square roots and performing calculations with them easy operations. Consult the manufacturer's instructions to learn how to find square root on your particular calculator, because the procedure varies according to the type of calculator.

Roots can be written as fractions. For instance,

$\sqrt{4} = 4^{(1/2)}$

$\sqrt{9} = 3$ (because $3 \times 3 = 9$)

$9^{(1/2)} = 3$ (because $3 \times 3 = 9$)

$\sqrt[4]{16} = 2$ (because $2 \times 2 \times 2 \times 2 = 16$)

$16^{(14)} = 2$ (because $2 \times 2 \times 2 \times 2 = 16$)

b. Cube Root

There is also cube root. Finding the cube root is the process of determining what number, when multiplied by itself three times equals the given number. The cube root of 27 is 3.

$(3 \times 3 \times 3 = 27)$

Cube root is shown by the symbol $\sqrt[3]{\ }$

$\sqrt[3]{27} = 3$

The process of finding the root is not limited to only square and cube. It may be necessary to determine the fourth root ($\sqrt[4]{\ }$), sixth root ($\sqrt[6]{\ }$) or higher; however, these procedures will not be covered in this text.

6. POWERS OF TEN (Scientific Notation)

To make calculations with very large or very small numbers easier, a shorthand system is used. Simplify numbers like 0.0000000284 and 23,000,000,000,000 without writing down all the zeros. For example, for 100 write $1 \times 10 \times 10$ or 1×10^2. This is read as "one times ten to the second power.' " Using powers of ten notation one may write 700,000,000,000,000 as 7×10^{14} and read it as "seven times ten to the fourteenth power." Note that the raised number (the exponent) above the ten indicates the number of zeros which follows the seven before the decimal place.

For negative powers, the decimal is moved to the left of the number to the number of places indicated by the exponent. The minus sign indicates that the number is smaller than a whole number. Therefore, 10^{-3} means 1×10^{-3} or .001 read as "ten to the minus third power." Likewise, 10^{-4} equals 1×10^{-4} or 0.0001 read as "ten to the minus fourth power."

Use the same method for more complicated numbers such as 3,640,000,000 or 0.02406. Simply find the standard form of the number and then count the number of places which the decimal was moved. The standard form of any number is a power of ten multiplied by a number 1 and 10, including 1. Take 3,640,000,000, for example. The standard form is 3.64 (a number between 1 and 10) and the decimal was moved 9 places to get the standard form. Therefore, 3.64×10^9 is the same as 3,640,000,000 (Figure 4). To change 0.02406 to a standrd number, move the decimal two places to the right, so you have 2.406. The standard method of writing 0.02406 is 2.406×10^{-2} (Figure 5).

3,640,000,000.00

To simplify complicated, long numbers into scientific notation,

3.640 000 000.00

Move the decimal point to the left, lowering the number to a value between 1 and 10,

1 2 3 4 5 6 7 8 9

3.640 000 000.00

Count the number of places that the decimal point was moved. In this example, the decimal was moved 9 places to the left.

$$3.64 \times 10^{9}$$

Power of Ten $= 10^{9}$

Figure 4 POWERS OF TEN (POSITIVE)

0.02406

To simplify a complicated decimal fraction into scientific notation,

0.02.406

Move the decimal point to the right, raising the number to a value between 1 and 10.

1 2

0.02.406

Count the number of places that the decimal point was moved. In this example, the decimal point moved 2 places to the right.

When dealing with decimal fractions, the power is written with a minus (−) sign in front of it.

$$2.406 \times 10^{-2}$$

Power of Ten $= 10^{-2}$

Figure 5 POWERS OF TEN (NEGATIVE)

7. RATIOS

You will use ratio calculations in comparing the existing fan RPM to a desired RPM in order to determine the proper sheave diameter.

Ratio means simply a comparison of two numbers as a fraction with the first number as the numerator and second number as the denominator.

For example, assume that two air outlets are on the same duct branch:

Outlet A = 300 CFM

Outlet B = 400 CFM

The ratio of the airflow of these outlets is:

$$\frac{300}{400} = \frac{3}{4}$$

This is said to be, "A ratio of 3 to 4."

In Figure 6, the ratio of the airflow outlets C to B is:

$$\frac{500}{750} = \frac{2}{3}$$

A ratio is reduced to the lowest numbers:

$$\frac{500}{750} = \frac{50}{75} = \frac{2}{3}$$

To reduce a ratio, divide each number in the ratio (the numerator and the denominator) by a common number.

8. PROPORTIONS

A proportion is a statement that two ratios are equal. The statement of:

$$\frac{50}{75} = \frac{2}{3}$$

is a simple proportion. In general, proportions are used to determine an unknown quantity when one ratio is known and only half of the other ratio is known. Knowledge of proportion will help you solve ratio problems. For example, in the proportion

$$\frac{x}{300} = \frac{2}{3}; \; x = \frac{300 \times 2}{3} = 200$$

Notice that $x = 200$ for the proportion to be true:

$$\frac{200}{300} = \frac{2}{3}$$

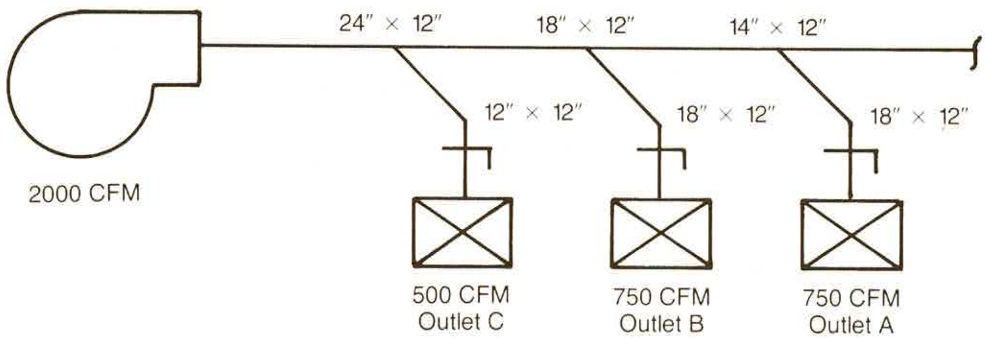


Figure 6. EXAMPLE SYSTEM

C Equations

Mathematical equations are really just a simplified and easy method of writing or reading a long statement about arithmetic processes. For instance, consider the following statement:

The circumference of a circle equals Pi times the diameter of the circle.

This is a long statement which is difficult to read and understand. This same statement is simplified when it is written as a mathematical equation:

$$C = \pi D$$

This equation says the same thing as the long statement. The equation is much shorter, and the use of symbols makes the mathematical relationships more obvious. Equations state problems and give directions for solving them in the quickest and easiest way (Figure 7).

When you learn to read equations, you will find many ways to make your work easier. For instance, you can carry many useful equations in a small notebook to use with a calculator. There usually is not enough time for hand calculations.

1. LETTERS IN EQUATIONS

Equations are simply a short way of stating an arithmetic problem. Consider the following:

Air Quantity = Cross Sectional Area
times the Average Velocity

Instead of writing out "Air Quantity" each time, CFM is used to mean air quantity (expressed in cubic feet per minute). Then the equation is:

CFM = Cross Sectional Area in Square Feet
times the Average Velocity in Feet per Minute

To further shorten the equation, Q is used to mean CFM, A is used to mean the cross sectional area. Also, V designates the average velocity. When these letters and symbols are used, the equation becomes:

$Q = A \text{ times } V = A \times V$

Remember that the *times sign* ($\times$) should be eliminated when using letters in an equation. Therefore, the equation becomes:

$Q = AV$

Since velocity is expressed in feet per minute, the equation is:

Q (CFM) = A (sq. ft.) times V (Feet per Minute)

Mathematical equations are short statements of arithmetic processes:

WRITTEN	ARITHMETIC	EQUATION
The circumference equals Pi times the diameter of the circle	Circumference = 3.1416 × diameter	$C = \pi D$
The circumference divided by two times Pi equals the radius of the circle	Radius of a circle = circumference ÷ 2 × 3.1416	$R = \frac{C}{2\pi}$

Figure 7 MATHEMATICAL STATEMENTS IN DIFFERENT FORMS

When an equation is developed, usually the first letter of a word is used to represent the whole word. However, any time that a new equation is given in a book, an explanation of the letters should follow. For example, a proper statement of the equation above would be:

Where:

$Q = AV$

Q = Air Quantity in Cubic Feet per Minute (CFM)

A = Area in Square Feet (Sq. Ft.)

V = Velocity in Feet per Minute (FPM)

2. READING EQUATIONS

Since equations are really short methods of writing mathematical statements, they require the use of a few special rules. Basically, these rules were developed to make it possible to write the shorter statement without losing any of the meaning and to make the statement as clear as possible.

a. Multiplication

The arithmetic sign for multiplication ($\times$) is rarely used in equations. This is because the letter $\times$ is often used in equations to stand for an unknown quantity. It would only lead to confusion to use it for multiplication as well. Letters, symbols, or numbers written next to each other are to be multiplied. In the equation $Q = AV$, the AV means "A times the V."

b. Parentheses

Parentheses are used in an equation to make it clear which operations are done first. The solution of the contents within the parentheses must be found before proceeding to solve that portion of the equation outside of the parentheses. For example, consider:

$$4 \times 6 - 2 = ?$$

If the 4 and 6 are first multiplied and then the 2 subtracted, the answer will be 22. However, if the 2 is first subtracted from the 6 and then the multiplication is done, the answer will be 16. In other words, the placement of parentheses changes the whole problem.

$$(4 \times 6) - 2 = 22 - 2 = 22$$

$$4 \times (6 - 2) = 4 \times 4 = 16$$

$$(4 - 2) \times 6 = 2 \times 6 = 12$$

$$(4 - 2)(6 - 1) = 2 \times 5 = 10$$

A number or letter written in front of a parenthesis is understood to be multiplication in the same way as πD is understood. Therefore, the second equation above is normally written as:

$$4(6 - 2) = 16$$

c. Division

The arithmetic sign for division ($\div$) is seldom used in equations. Instead, the statement is written as a fraction. Remember that the fraction ½ really means $1 \div 2$. To pose the problem $5 \div 4$, you would use 5/4 or $\frac{5}{4}$.

In Figure 7, some relationships are shown by the arithmetic method and by the equation method. In each case shown, both methods apply to exactly the same problem.

3. SUBSTITUTING NUMBERS OR LETTERS

Letters are used in equations to make the equation short and easy to read. But they are also used so that the equation may be applied to many different conditions. Consider again the equation:

$$A = \pi R^2$$

R is the symbol for the radius of the circle. It is replaced by the length of the actual radius. For a circle with a 3" radius, the equation would be:

$$A = \pi R^2$$

$$A = 3.1416 \times (3)^2$$

$$A = 3.1416 \times 9$$

$A = 28.2744$ sq. in. (Round off to 28.3 sq. in.)

EXAMPLE F

Using the following equation:

$A = 0.7854\ D^2$

Where:

A = Area of Circle

D = Diameter of Circle

Find the area of a 6″ diameter circle.

SOLUTION:

$A = 0.7854\ D^2$

$A = 0.7854 \times 6 \times 6$

A = 28.2744 sq. in. (Round off to 28.3 sq. in.)

4. REWRITING EQUATIONS

An equation is a short cut method for stating a mathematical relationship. If two or more letters are used in an equation, then the equation can be written in different forms. Being able to change equations into different forms will be a big advantage for you, since it means that you will not have to memorize so many different forms. For example, by now you probably know the equation for the airflow of a duct.

$Q = A \times V$

But what if the CFM is known and the velocity must be found? Then, the equation is changed to another form:

$$\frac{Q}{A} = V$$

If you do not know how to change equations you will have to remember both of these forms. If you know how to change equations you need only remember the common:

$Q = A \times V$

Another example is the equation for the area of a circle:

$A = \pi R^2$

If you remember this equation or if you have written it down, you can, if necessary, change the basic equation to make variations. If you cannot change equations, then you must also remember the equations below, which are variations of the same equation:

$$R^2 = \frac{A}{\pi}$$

$$R = \sqrt{\frac{A}{\pi}}$$

a. The Basics for Changing an Equation

The basic idea in changing an equation is to remember that the equal sign means that the total value on one side is equal to the total value on the other side. Think of the equal sign as the balance point on a scale (Figure 8). One side must balance the other side. If it does not, the equation is incorrect. In Figure 9, if a 2 lb. weight were removed from the left side, the scale would be out of balance.

Obviously, if 2 lbs. are taken from the left side, then 2 lbs. must also be taken from the right side in order to keep the scale in balance (Figure 10). The same is true of an equation. If an amount is taken from one side, an equal amount must be taken from the other side to keep the equation in balance.

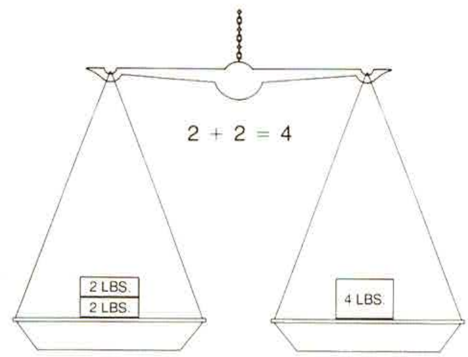


Figure 8 AN EQUAL SIGN MEANS THAT EACH SIDE BALANCES THE OTHER

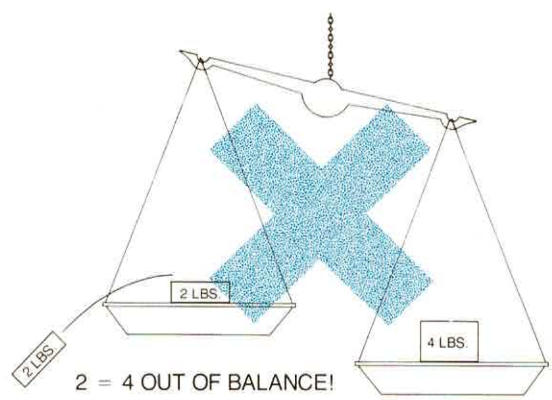


Figure 9 REMOVING A QUANTITY FROM ONE SIDE MAKES THE SCALE OUT OF BALANCE

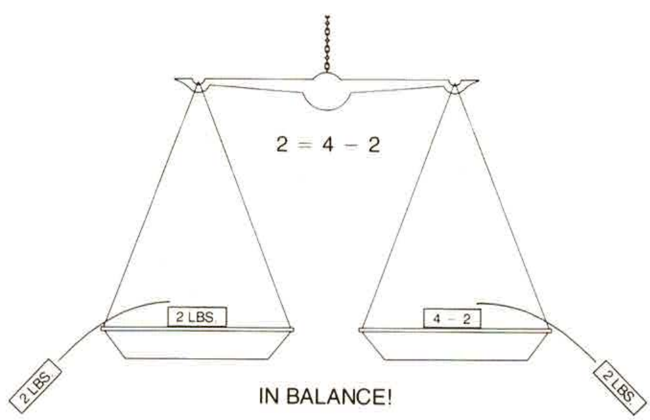


Figure 10 TO KEEP THE SCALE (OR THE EQUATION) IN BALANCE, IF 2 LBS. ARE REMOVED FROM ONE SIDE, THEN 2 LBS. MUST ALSO BE REMOVED FROM THE OTHER SIDE

b. Adding Values

The same idea of balancing that is used for subtracting values from an equation is also used for adding values. Obviously, if an amount is added to one side of the equation, the same amount must be added to the other side if the equation is to be kept in balance.

Rule: To keep an equation in balance, the amount that is added to one side of an equation must be added to the other.

Examples:

$$2 + 2 = 4$$
$$2 + 2 + 2 = 4 + 2$$
$$6 = 6$$
$$X + Y = 3$$
$$X + Y + Y = 3 + Y$$
$$X + 2Y = 3 + Y$$

c. Subtracting Values

Keep in mind the idea that **each side of the equal sign must balance** and consider the following rules for balancing equations:

Rule: To keep an equation in balance, the amount that is subtracted from one side must also be subtracted from the other side.

Examples:

(1) Change the equation so that only one of the 2's is on the left side.

$$2 + 2 = 4$$
$$2 + 2 - 2 = 4 - 2$$
$$2 = 2$$

(2) Change the equation so that only X is on the left side.

$$X + Y = 3$$
$$X + Y - Y = 3 - Y$$
$$X = 3 - Y$$

(3) Change the equation so that only Y is on the left side.

$$X + Y = 3 - X$$
$$X + Y - X = 3 - X$$
$$Y = 3 - X$$

Of course, you do not yet know why the equations were changed as they were. This is the next thing to learn. Assume that the equation $c = a + b$ is to be solved for a.

$$c = a + b$$

Since you are solving for **a**, this means that **a** must be alone on one side of the equal sign. To do this, **b** must be subtracted from that side of the equation. Remember, that to keep the equation in balance, whatever is done on one side must be done on the other. Therefore, **b** must be subtracted from both sides.

$$c - b = a + b - b$$

Since **b** − **b** equals zero (just as $2 - 2 = 0$), the equation can be cleaned up of unnecessary parts and written as:

$$c - b = a$$

or:

$$a = c - b$$

Look at the steps that were done to solve the equation for **a**:

(1) Decide the unknown for which the equation must be solved.
(2) Decide what must be done to remove all the other equation elements from the side of the unknown.
(3) Do the same things to both sides of the equation.
(4) Simplify the equation by removing unnecessary elements.

Example:

Given: $a^2 + b^2 = c^2$

Solve for b^2

(a) Determine the unknown.
The unknown is b^2

(b) What must be done to isolate b^2?
Subtract a^2

(c) Do the same things to both sides of the equation.
$\mathbf{a^2 + b^2 - a^2 = c^2 - a^2}$

(d) Simplify the equation
Eliminate $a^2 - a^2$
$\mathbf{b^2 = c^2 - a^2}$

In order to solve for b, simply find the square root of b^2. If $b^2 = c^2 - a^2$, then the square root of b^2 is $b = \sqrt{c^2 - a^2}$. Using numbers in place of the letters makes this process clear.

Given: $b^2 = c^2 - a^2$

Where: $a = 3$ and $c = 5$

Solve for b:

(a) $b^2 = 5^2 - 3^2$

(b) $b^2 = 25 - 9$

(c) $b^2 = 16$

(d) $b = \sqrt{16}$

(e) $b = 4$

d. Subtraction Process

Note that the same process applies to subtraction.

Given: $c = a - b$

Solve for a:

(a) Unknown is **a**

(b) What must be done?
Add b

(c) $c + b = a - b + b$

(d) Simplify
Eliminate $-b + b$
Therefore: $c + b = a$
or $a = c + b$

5. CHANGING EQUATIONS BY MULTIPLICATION AND DIVISION

Keep in mind that multiplying and dividing are nothing but fast methods of adding and subtracting. Therefore, the rule for balancing equations which states that if **anything is added or subtracted from one side of an equation, the same must be done to the other side,** also applies to multiplying or dividing. Keeping the balanced scale in mind and remember this general rule:

Rule: To keep an equation in balance, whatever is done to one side, must also be done to the other side.

Now look at the equation

$$c = 2\pi R$$

In the equation, R is known, and the equation will give the unknown quantity, C. Now, suppose that the circumference (C) is known and the radius (R) is unknown. Then, by dividing both sides of the equation by 2π, the equation is rewritten as:

$$R = \frac{C}{2\pi}$$

In this case, R is the unknown. This process of changing the equation is called **solving the equation for R.**

a. Multiplication Process

Given: $c = \frac{a}{b}$

Solve for a:

(a) Unknown is **a**

(b) Multiply both sides of the equation by **b**

(c)
$$c \times b = \frac{a}{\cancel{b}_{1}} \times \cancel{b}^{1}$$

(d) Simplify

$cb = a$

$a = cb$

b. Division Process

Given: $C = 2\pi R$

Solve for R:

(a) Unknown is **R**

(b) Because **R** is multiplied by 2π, both sides of the equation must be divided by 2π to isolate **R**:

$$\frac{\overset{1}{\cancel{2}}\pi R}{\underset{1}{\cancel{2}}\pi} = \frac{R}{2\pi}$$

You can then see how the 2π is eliminated on the left side of the equation by simple cancellation.

Simplify

$$\frac{C}{2\pi} = R$$

or: $R = \frac{C}{2\pi}$

6. NEGATIVE NUMBERS

Before going any further with changing equations, you must understand the idea of negative numbers. Consider the following equation, for example:

$$X - Y = 3$$

If you wished to move the "$-Y$" to the other side of the equation, you would **add** Y to each side.

$$X - Y + Y = 3 + Y$$
$$X + 0 = 3 + Y$$
$$X = 3 + Y$$

You may not see the logic of this yet, unless you already understand the idea of negative numbers. Very simply, positive numbers are **above** zero ($+2$, $+2$, etc.). Negative numbers are **below** zero (-1, -2, etc.).

Visualize a line with numbers marked along its length (Fig. 11). Zero is the base point or starting point. Every number above zero is a positive number. Every number below zero is a negative number. Look at the position of the numbers -1 and $+1$ in Fig. 11. They are **different** numbers and in different positions on the line. The difference between -1 and $+1$ is 2 units.

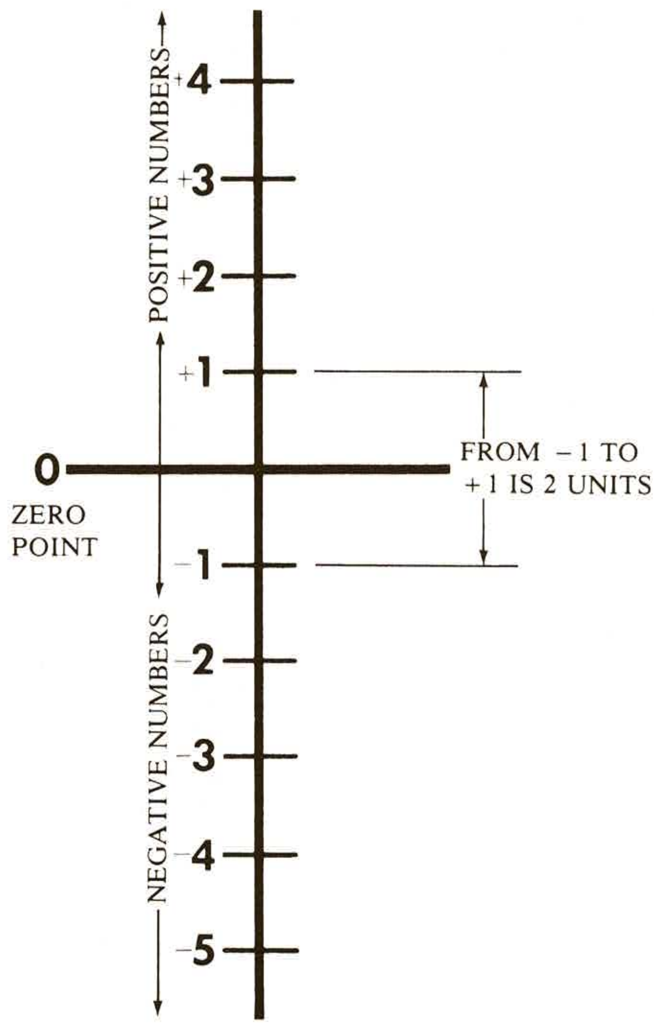


Figure 11 NEGATIVE AND POSITIVE NUMBERS ARE IN RELATION TO THE ZERO POINT

a. Adding or Subtracting Negative Numbers

Negative numbers can easily be seen on a thermometer (Figure 12). In Figure 12A the thermometer shows −10°F. This means the reading is 10°F below zero. If the temperature is raised 5°F, the temperature reading is then −5°F, (Figure 12B), or 5°F below zero. It is simple to see that 5°F added to −10°F results in −5°F. It is important to know how this is stated in numbers.

$-10°F + 5°F = -5°F$

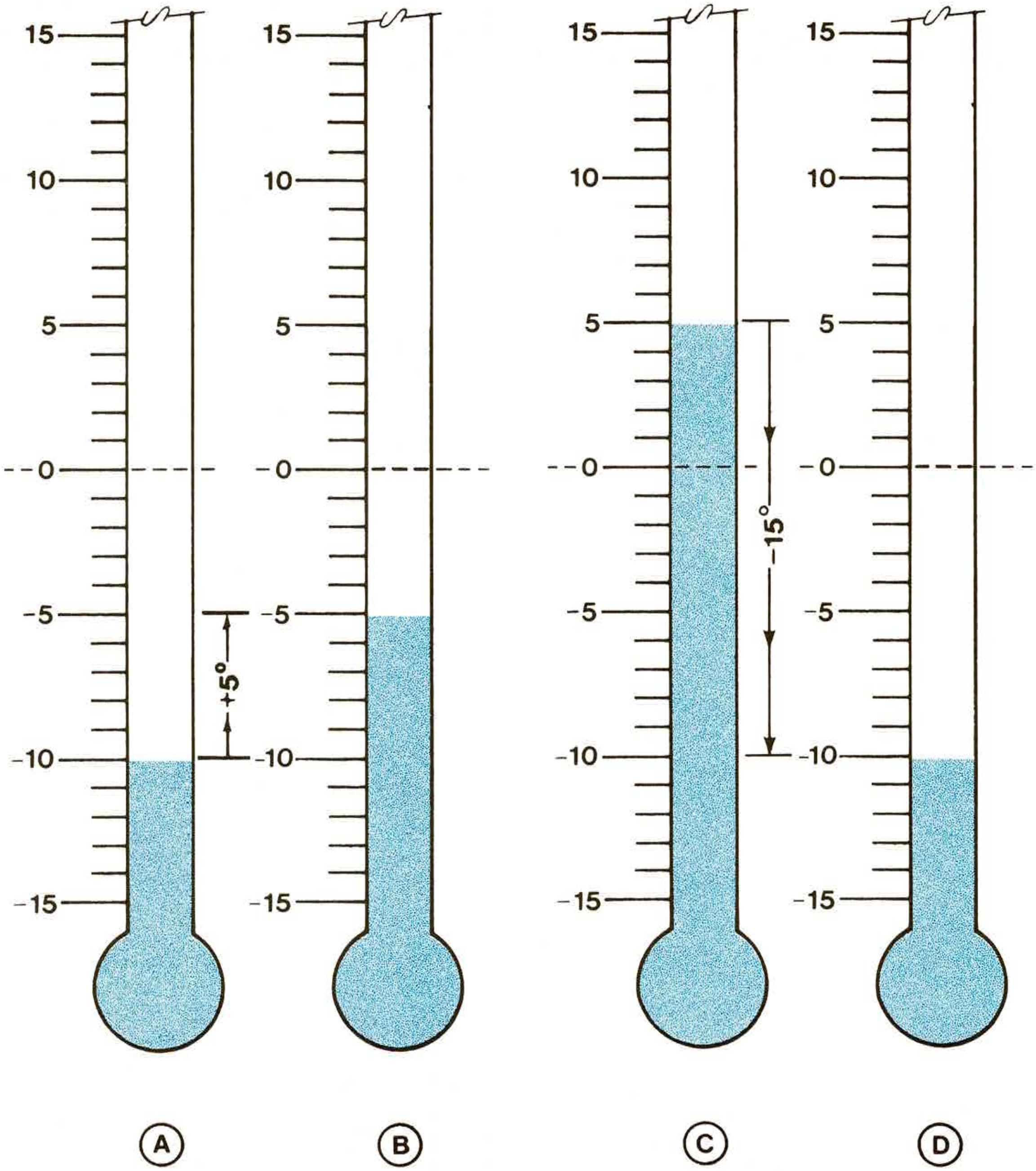


Figure 12 READING TEMPERATURES ON A THERMOMETER SCALE

Rule: If a negative and a positive number are added, the result is the difference between the two. The answer takes the sign of the larger number.

This relationship illustrates the first rule of negative numbers.

Study the numbers above, and the illustration and how the two show the same relationship.

Now consider the problem:

$+5 - 15 = ?$

The difference between the two numbers is 10 and since the larger number (15) has a negative sign, the answer is:

$+5 - 15 = -10$

On problems such as this, the plus sign is usually omitted because any number that does not have a negative sign is assumed to be positive. Thus, the previous problem is usually stated as follows:

$5 - 15 = -10$

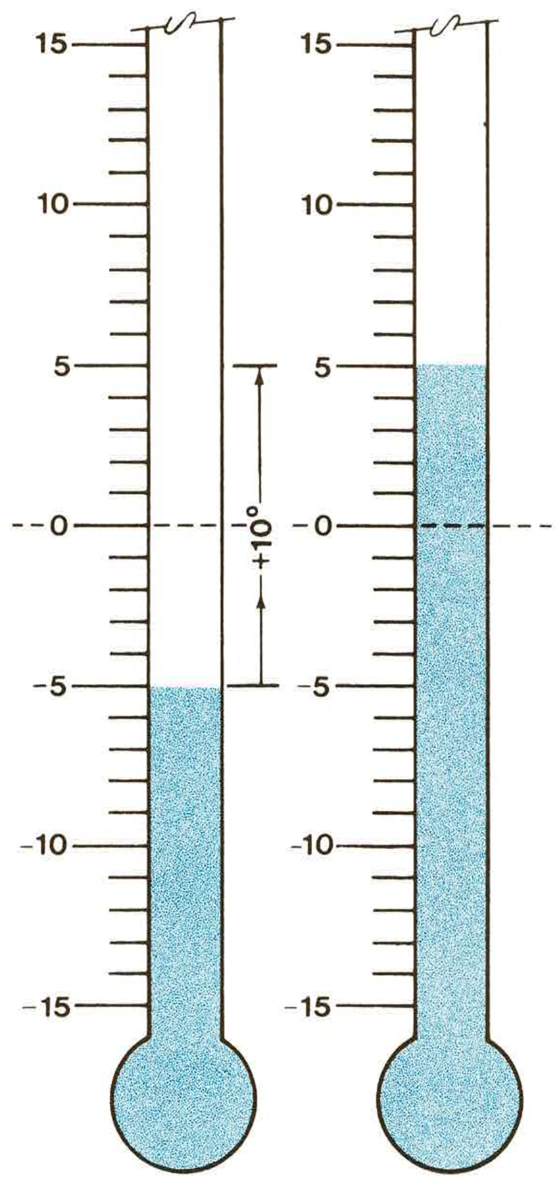


Figure 13 READING TEMPERATURES ON A THERMOMETER SCALE

Check Figure 12C ($+5°$) and Figure 12D ($-10°$) to see an illustration of this problem.

Apply the rule to the problem:

$-5 + 10 = ?$

The answer is $+5$ since the sign of the larger number (10) is positive. This is illustrated on the thermometer scales in Figure 13.

b. Multiplying and Dividing Negative Numbers

Since multiplying and dividing are really fast forms of adding and subtracting (2×10 really means ten twos added together), the same idea of adding and subtracting negative and positive numbers applies to multiplying and dividing. Remember this simple rule:

Rule: When multiplying or dividing numbers with opposite signs, the answer will be a negative (−) number. In all other cases, the answer will be a positive (+) number.

This rule means that:

$$-2 \times (+4) = -8$$
$$2 \times (-4) = -8$$
$$4 \div (-2) = -2$$
$$-4 \div (+2) = -2$$

In all of these cases, a positive (+) number is combined with a negative (−) number. Therefore, the answer is a negative number. When two positive numbers are combined, you know the answer must be positive.

$$2 \times 2 = 4$$
$$2 \div 2 = 1$$

It is also true that multiplying or dividing two negative numbers will give a positive answer.

$$-2 \times (-2) = 4$$
$$-2 \div (-2) = 1$$

The above can be summarized as follows (Figure 14):

+ times + = +	+ divided by + = +
− times − = +	− divided by − = +
+ times − = −	+ divided by − = −
− times + = −	− divided by + = −

Examples:

$$10 \times (-3) = -30$$
$$(-10) \times (-3) = 30$$
$$30 \div (-3) = -10$$
$$(-30) \div (-3) = 10$$

MULTIPLYING OR DIVIDING
NEGATIVE OR POSITIVE NUMBERS
+ WITH + GIVES +
− WITH − GIVES +
+ WITH − GIVES −
− WITH + GIVES −

Figure 14 RESULTS OF MULTIPLYING OR DIVIDING NEGATIVE AND POSITIVE NUMBERS

D Applied Mathematics

1. AREAS

Area is the measurement of the space on a plane surface within set boundaries. Areas are always stated in square inches or square feet in the English System.

a. Squares and Rectangles

EQUATION 3

Area = Length × Width

(or)

A = WH (can also be stated as W × H or W • H)

The cross-sectional area of a duct must be calculated to determine the amount of air flowing in a duct. Duct dimensions are always given in inches, but duct area must be expressed in square feet (Figure 16). This is because airflow is not simply a measurement of a plane, but is a measurement of volume, and volume within a duct is always stated in CFM (Cubic Feet per Minute).

To obtain the area in square feet, the duct dimensions (in inches) must be changed to feet. If the duct dimensions are left in inches, the result will be square inches. These square inches must then be converted to square feet by dividing the result of square inches by 144.

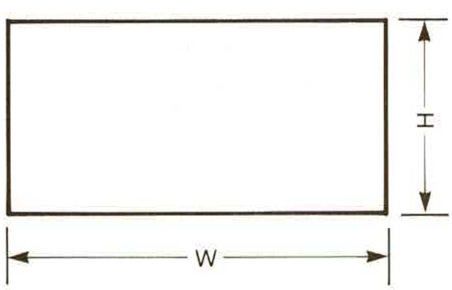
As an alternate method the area for the duct shown in Figure 16 can be found:

$$\frac{14''}{12''} = 1.167 \text{ feet}$$

$$\frac{20''}{12''} = 1.667 \text{ feet}$$

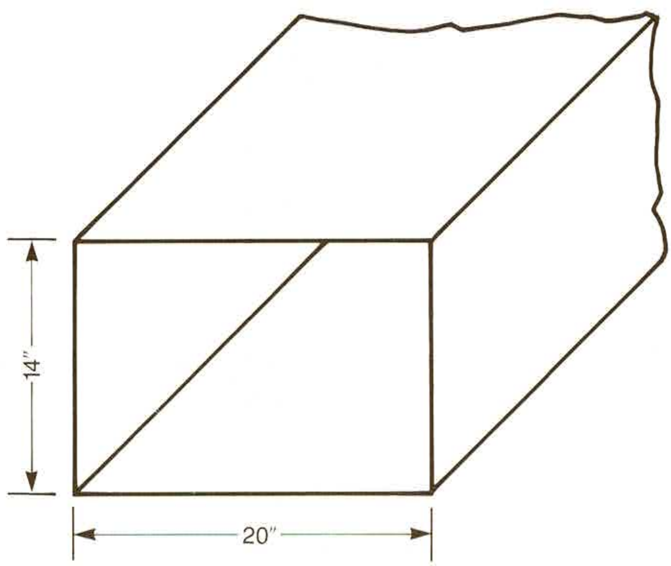
$$A = WH = 1.667 \times 1.167$$

$$A = 1.945 \text{ square feet}$$



W × H = Area

Figure 15 AREA OF RECTANGLE



$$\frac{14'' \times 20''}{144 \text{ Sq. In.}} = 1.94 \text{ Sq. Ft.}$$

Figure 16 DUCT CROSS SECTIONAL AREA

b. Circles

The mathematical equation for the area of a circle is:

EQUATION 4

$$A = \pi R^2$$

Where:

A = Area

R = Radius

EXAMPLE G

Find the area of a 6″ *radius* circle.

SOLUTION:

$$A = \pi R^2$$
$$A = 3.1416 \times 6'' \times 6''$$
$$A = 113.04 \text{ sq. in.}$$

EXAMPLE H

Find the area of a 6″ *diameter* circle.

SOLUTION:

$$A = \pi R^2$$
$$A = 3.1416 \times 3'' \times 3''$$
$$A = 28.27 \text{ sq. in.}$$

Often, only the circumference of a circle can be measured and, therefore, cannot see the equation to make the necessary calculations. Using the equation:

EQUATION 5

$$C = \pi D,$$

one may find the diameter of a circle with a 22 inch circumference:

$$C = \pi D;\ D = \frac{C}{\pi}$$

$$D = \frac{22''}{3.1416} = 7.00 \text{ inches}$$

c. Triangles

The area of a triangle is equal to one-half of the base times the altitude (Figure 17). The altitude is the height of the triangle.

EQUATION 6

$$A = \tfrac{1}{2}\, ab$$

Where:

A = Area

a = altitude (height)

b = base

The logic of this equation can be seen in Figure 17. The triangle is one half of the rectangle shown by the dashed lines. Since the area of the rectangle is A = ab, then the area of one triangle is ½ ab.

The area of the right triangle in Figure 18 is:

$A = \frac{1}{2} ab$

$A = \frac{1}{2} \times 6'' \times 8''$

A = 24 sq. in.

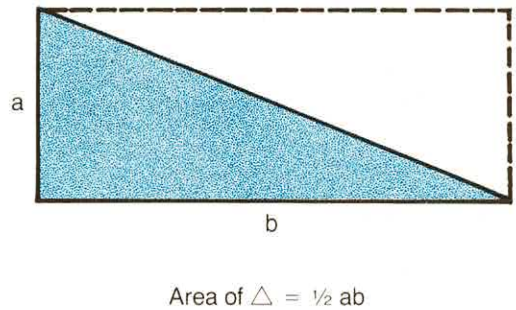


Figure 17 AREA OF A TRIANGLE

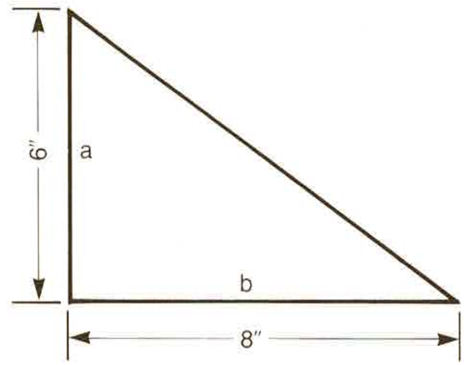


Figure 18 AREA OF A TRIANGLE

In this triangle, the altitude dimension is also the same dimension as one side.

The area of the triangle in Figure 19 is:

$A = \frac{1}{2} ab$

$A = \frac{1}{2} \times 10'' \times 6''$

A = 30 sq. in.

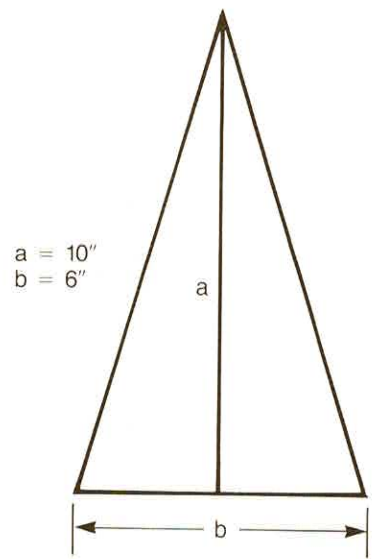


Figure 19 AREA OF A TRIANGLE

The area of the triangle in Figure 20 is:

$A = \frac{1}{2} ab$

$A = \frac{1}{2} \times 4 \times 6$

A = 12 sq. in.

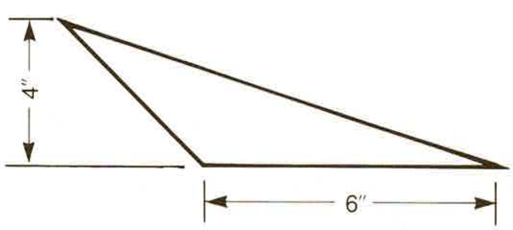


Figure 20 AREA OF A TRIANGLE

d. Parallelograms

The equation for the area of a parallelogram is the same as that for the area of a rectangle.

$A = bh$

This is because the parallelogram (Figure 21) may be seen as a rectangle if one cuts off one portion of the parallelogram and moves it to the other side.

If b = 12″ and h = 8″

Then

$A = 12'' \times 8'' = 96$ sq. in.

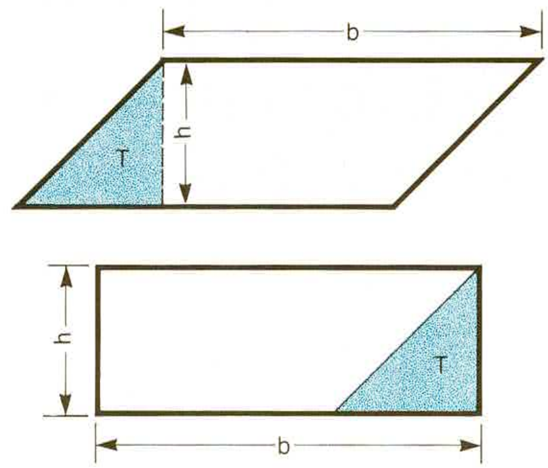


Figure 21 AREA OF PARALLELOGRAM

2. EQUIVALENT AREAS

Testing, Adjusting and Balancing Technicians must understand the principles of equivalent areas in order to determine if a problem exists in environmental systems. For example, what arrangement of pipes would be required to carry the fluid flow where the main pipe is 10 inches in diameter? Would two 5 inch pipes provide the equivalent area of the 10 inch diameter pipe? The answer is NO.

Figure 22 and 23 illustrate the equivalent area for a 10 inch diameter pipe. The area of a 10 inch diameter circle is 78.5 square inches, according to the equation $A = \pi R^2$. The combined area of two 5 inch diameter circles is 39.2 square inches. An 8 inch diameter pipe and a 6 inch diameter pipe connected to the 10 inch diameter pipe provide equal area. If the Technicians see branches that are not the equivalent area of the main line they should be noted and reported. It is the design engineer's or contractor's responsibility to determine whether they were intended to be these sizes for some special reason.

In SMACNA and ASHRAE manuals, tables can be found for "circular equivalents of rectangular ducts for equal friction and capacity". These tables are to be used *only* for determining the duct sizes and friction losses. They are *not to be used* to determine the area used for duct average velocities and for duct fitting losses.

Duct average velocities are obtained by dividing the airflow (CFM) by the cross-sectional area in square feet.

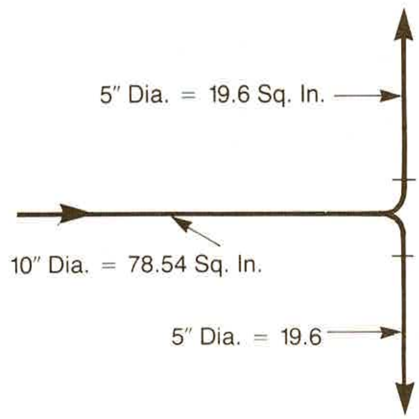


Figure 22 INCORRECT EQUIVALENT AREAS

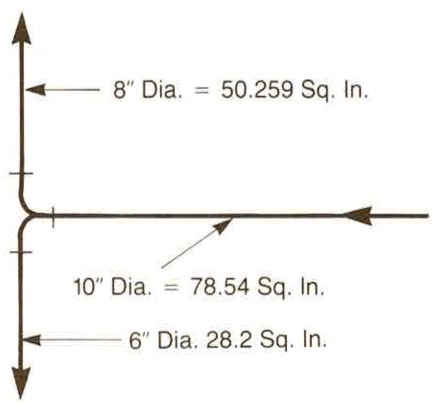


Figure 23 EQUIVALENT AREAS

EXAMPLE I

Find the average velocity of a 24″ × 18″ duct that has an airflow of 5400 CFM.

SOLUTION:

$$A = \frac{24'' \times 18''}{144 \text{ sq. in.}} = 3 \text{ sq. ft.}$$

$$V_{ave} = \frac{\text{CFM}}{\text{AREA}}$$

$$V_{ave} = \frac{5400}{3} = 1800 \text{ FPM}$$

3. VOLUMES

Volume in cubic measurement is stated in cubic inches or cubic feet (English System of Measurement).

A common measure for volume of airflow rate is CFM (Cubic Feet per Minute). Volumes of rooms and buildings must also be calculated.

For any shape with parallel sides, Volume equals the Area of the Base times the Height.

In Figure 24 the Area of the Base is:

8″ × 18″ = 144 sq. in.

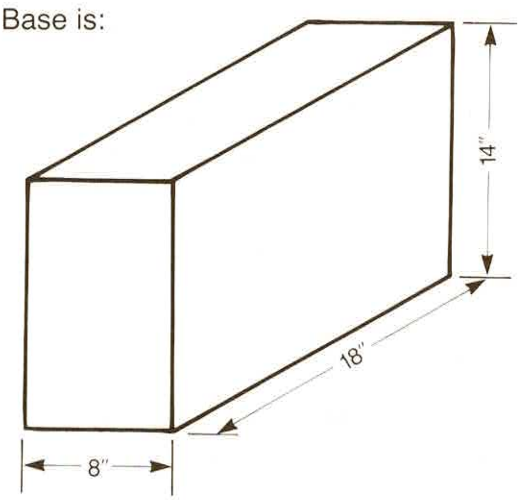


Figure 24 VOLUME

The Volume is the Area of the Base times the Height:

$144'' \times 14'' = 2016$ cubic inches

Cubic inches is converted to cubic feet by dividing by 1728 $(12'' \times 12'' \times 12'' = 1728)$.

2016 cu. in. $\div 1728 = 1.17$ cu. ft.

The Volume of a round tank is Area of the Base (πR^2) times the Height, or

$V = \pi R^2 H$

In Figure 25 the volume in cubic feet is:

$V = \pi R^2 H$

$V = \pi (47/12)^2\ 13$

$V = 626.52$ Cubic Feet

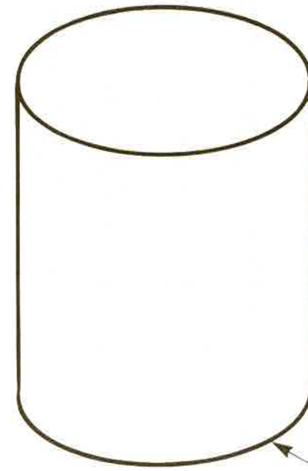


Figure 25 VOLUME

Most often in HVAC work, tank volumes are found in terms of gallons. To obtain the volume of the tank in Figure 24 in gallons, multiply the volume in cubic feet by 7.49 gallons per cubic foot.

$V = 626.52 \times 7.49$

$V = 4692.63$ gallons

4. PERCENTAGES

Percent means "parts of 100"

1% means 1 out of 100

50% means 50 out of 100

In order to understand percent, it is necessary to be able to read decimal places. As shown below, the first position to the right of the decimal point is the "tenths" position. The second position is the "hundreths" position. The third position to the right of the decimal is the "thousandths" position.

tenths ↘ ↙ thousandths

0.111

↖ hundreths

When you write 1%, 10%, and 1/10 of 1% as decimals, you write the following:

$0.01 = 1\%$

$0.1 = 10\%$

$0.001 = \frac{1}{10}$ of 1%

Percent is a ratio as compared to 100:

$$1\% \text{ means } \frac{1}{100}$$

$$50\% \text{ means } \frac{50}{100}$$

Percentages can be read to express the measured CFM of a duct outlet as compared to the design CFM.

A supply air outlet should deliver a design CFM of 100. It is measured at 75 CFM.

$$\frac{75}{100} = 0.75 \times 100 = 75\% \text{ of design CFM}$$

(You may convert a decimal to a percent by multiplying by 100.)

EXAMPLE J

The design CFM of an outlet is 850. It is measured at 700 CFM. What is the percent of design it is delivering?

SOLUTION:

$$\frac{700}{850} = 0.8235 \times 100 = 82\% \text{ of design}$$

EXAMPLE K

In a hydronic system, a pump is handling 825 GPM (Gallons per Minute). The design flow rate is 750 GPM. What is the percent of actual flow to design flow?

SOLUTION:

$$\frac{825}{750} = 1.10 \times 100 = 110\%$$

---

5. PROCEDURES

When calculating problems where the dimensions are in fractions and the answer must be in fractions instead of decimals (job conditions), use the following procedures in the example.

EXAMPLE L

The circumference of a circle is 40⅝". Find the diameter to the nearest $\frac{1}{32}$".

SOLUTION:

$$C = 40\tfrac{5}{8} = 40.625 \text{ inches}$$

$$C = \pi D \text{ or}$$

$$D = \frac{C}{\pi} = \frac{40.625}{3.1416}$$

$$D = 12.931 \text{ inches; } 0.931 \times 32 = 29.79 \text{ (use 30)}$$

$$D = 12\tfrac{30}{32} = 12\tfrac{15}{16} \text{ inches}$$

E

Review Questions

Answers may be found in Section F

1. Convert each of the numbers listed below to the nearest sixteenth and to the nearest thirty-second

	Change This Number	To the Nearest Sixteenth or Less	To the Nearest Thirty-second
a)	2.789		
b)	11.056		
c)	1.349		
d)	0.179		
e)	1.187		
f)	5.901		
g)	1.250		
h)	6.749		
i)	0.479		
j)	4.598		

2. Convert each of the fractions listed below to its decimal equivalent. Give your answer in three decimal places.

	Fraction	Decimal Equivalent
a)	$7/8$	
b)	$3/4$	
c)	$5/8$	
d)	$3/8$	
e)	$1/8$	
f)	$1/16$	
h)	$17\,17/32$	
i)	$2\,15/16$	
j)	$3/16$	

3. Solve the following:
   a) The square of 16 is ________.
   b) The cube of 9 is ________.
   c) $15^2 =$ ________.
   d) $8^3 =$ ________.
   e) $2^5 =$ ________.
   f) $\sqrt{25} =$ ________.
   g) $\sqrt[3]{125} =$ ________.
   h) $9^{(1/2)} =$ ________.
   i) $\dfrac{4\,(6 - 3)^2}{\sqrt{9}} =$ ________.
   j) $\dfrac{(4 + 3)\,(\sqrt{25} + 1)}{(3 \times 2)^2} =$ ________.

4. Convert the following numbers using powers of ten (scientific notion):
   a) 4,230,000.00 ____
   b) 0.00386 ____
   c) 78,400.00 ____
   d) 0.0415 ____
   e) 163,800.00 ____
   f) 0.00341 ____
5. Solve the following:
   a) $(2 \times 4) - 3 =$ ____ .
   b) $2\,(4 - 3) =$ ____ .
   c) $(5 - 2) + (6 - 3) =$ ____ .
   d) $(3 \times 8) - 4\,(3 - 8) =$ ____ .
   e) $5 + (5 \times 5)^2 \times 2 =$ ____ .
6. Rewrite the following equations solving for "X":
   a) $D = \frac{a - b}{x}$
   b) $A = \frac{x\,(c + b)}{A}$
   c) $x^2 + y^2 = z^2$
7. Find the areas of shaded portion of the figures.
   a) Figure 26 area is ____sq. in.
   b) Figure 27 area is ____sq. in.

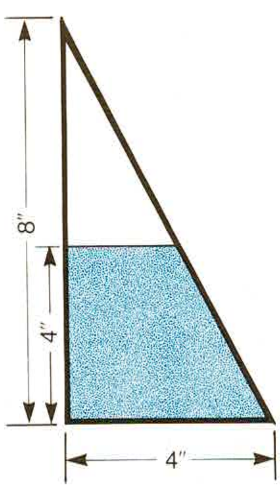


Figure 26 AREA PROBLEM

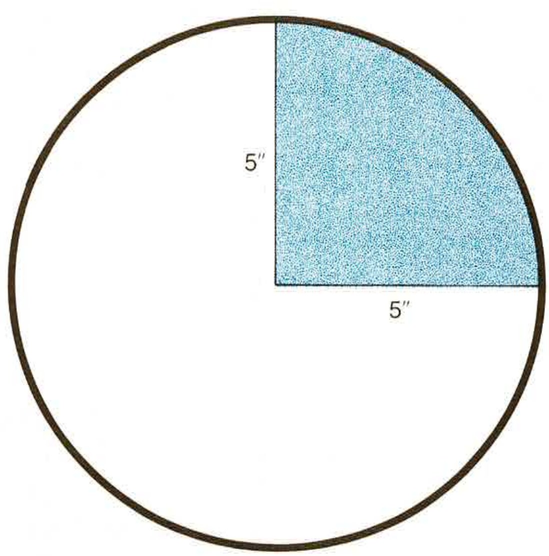


Figure 27 AREA PROBLEM

8. Find the areas:
   a) Figure 28 area is ________sq. ft.
   b) Figure 29 area is ________sq. in.
   c) Figure 30 area is ________sq. in.

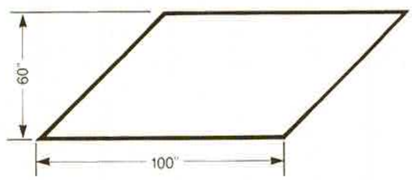


Figure 28

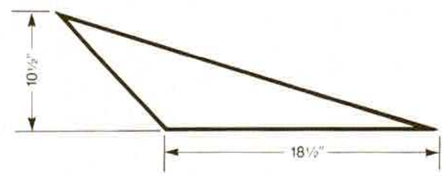


Figure 29

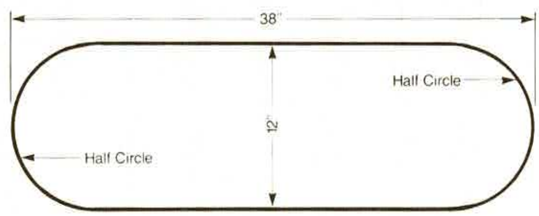
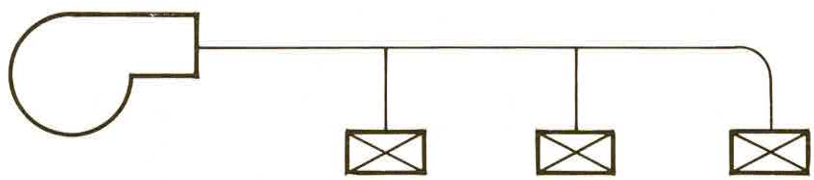


Figure 30

9. Fill in the percent of design for the three outlets (Figure 31).



	A	B	C
Design	875	900	650
Actual	800	850	750
% of Design	______	______	______

Figure 31

10. In a hydronic system, if a pump is handling 525 GPM and the design flow rate is 435 GPM, what is the percent of actual flow? ____________

11. The circumference of a pipe is 17 inches. What is its diameter? ________feet (use decimal)

12. A rectangular tank that is 32″ long, 24″ wide, and 61″ high will hold ____ gallons when filled to the top.

F

Answers to Review Questions

1. Convert each of the numbers listed below to the nearest sixteenth and to the nearest thirty-second

	Change This Number	To the Nearest Sixteenth or Less	To the Nearest Thirty-second
a)	2.789	**$2\frac{13}{16}$**	**$2\frac{25}{32}$**
b)	11.056	**$11\frac{1}{16}$**	**$11\frac{2}{32}$**
c)	1.349	**$1\frac{3}{8}$**	**$1\frac{11}{32}$**
d)	0.179	**$\frac{3}{16}$**	**$\frac{6}{32}$**
e)	1.187	**$1\frac{3}{16}$**	**$1\frac{6}{32}$**
f)	5.901	**$5\frac{7}{8}$**	**$5\frac{29}{32}$**
g)	1.250	**$1\frac{1}{4}$**	**$1\frac{8}{32}$**
h)	6.749	**$6\frac{3}{4}$**	**$6\frac{24}{32}$**
i)	0.479	**$\frac{1}{2}$**	**$\frac{15}{32}$**
j)	4.598	**$4\frac{5}{8}$**	**$4\frac{19}{32}$**

2. Convert each of the fractions listed below to its decimal equivalent. Give your answer in three decimal places.

	Fraction	Decimal Equivalent
a)	$\frac{7}{8}$	**0.875**
b)	$\frac{3}{4}$	**0.750**
c)	$\frac{5}{8}$	**0.625**
d)	$\frac{3}{8}$	**0.375**
e)	$\frac{1}{8}$	**0.125**
f)	$\frac{1}{16}$	**0.063**
h)	$17\frac{17}{32}$	**17.531**
i)	$2\frac{15}{16}$	**2.938**
j)	$\frac{3}{16}$	**0.188**

3. Solve the following:
   a) The square of 16 is **256.**
   b) The cube of 9 is **729.**
   c) $15^2 = \mathbf{225.}$
   d) $8^3 = \mathbf{512.}$
   e) $2^5 = \mathbf{32.\ (2 \times 2 \times 2 \times 2 \times 2 = 32)}$
   f) $\sqrt{25} = \mathbf{5.}$
   g) $\sqrt[3]{125} = \mathbf{5.}$
   h) $9^{(1/2)} = \mathbf{3.}$
   i) $\frac{4\,(6 - 3)^2}{\sqrt{9}} = \frac{\mathbf{4 \times 9}}{\mathbf{3}} = \frac{\mathbf{36}}{\mathbf{3}} = \mathbf{12.}$
   j) $\frac{(4 + 3)\,(\sqrt{25} + 1)}{(3 \times 2)^2} = \frac{\mathbf{7 \times (5 + 1)}}{\mathbf{36}} = \frac{\mathbf{42}}{\mathbf{36}} = \mathbf{1.167.}$

4. Convert the following numbers using powers of ten (scientific notion):
   a) 4,230,000.00 **4.23×10^6**
   b) 0.00386 **3.86×10^{-3}**
   c) 78,400.00 **7.84×10^4**
   d) 0.0415 **4.15×10^{-2}**
   e) 163,800.00 **1.638×10^5**
   f) 0.00341 **3.41×10^{-3}**

5. Solve the following:
   a) $(2 \times 4) - 3 =$ **$8 - 3 = 5.$**
   b) $2(4 - 3) =$ **$2 \times 1 = 2.$**
   c) $(5 - 2) + (6 - 3) =$ **$3 + 3 = 6.$**
   d) $(3 \times 8) - 4(3 - 8) =$ **$24 - 4 \times (-5) = 24 + 20 = 44.$**
   e) $5 + (5 \times 5)^2 \times 2 =$ **$5 + 25^2 \times 2 = 5 + 1250 = 1255$**

6. Rewrite the following equations solving for "X":
   a) $D = \frac{a - b}{x}$ **$x = \frac{a - b}{D}$**
   b) $A = \frac{x(c + b)}{A}$ **$x = \frac{A^2}{c + b}$**
   c) $x^2 + y^2 = z^2$ **$x = \sqrt{z^2 - y^2}$**

7. Find the areas of shaded portion of the figures.
   a) Figure 26 area is **12** sq. in.
   b) Figure 27 area is **19.64** sq. in.

a)

$$A_1 = \frac{8 \times 4}{2} = 16 \text{ (Total)}$$

$$A_2 = \frac{4 \times 2}{2} = 4 \text{ (Unshaded)}$$

$$A_3 = 16 - 4 = 12 \text{ sq. in.}$$

b) $A = \pi R^2 = \pi\, 25$
$A = 78.54$ **sq. in.**
$A/4 = 19.64$ **sq. in.**

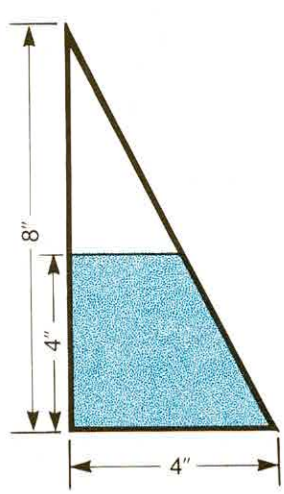


Figure 26 AREA PROBLEM

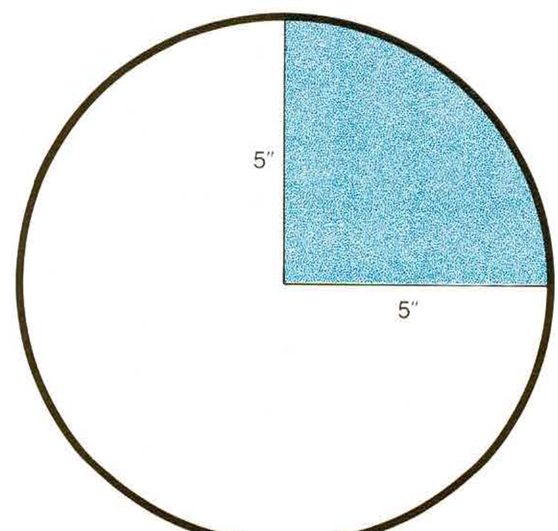


Figure 27 AREA PROBLEM

8. Find the areas:
   a) Figure 28 area is **41.67** sq. ft.
   b) Figure 29 area is **97.13** sq. in.
   c) Figure 30 area is **425.04** sq. in.

   a) $$\frac{100'' \times 60''}{144} = 41.67 \text{ sq. ft.}$$

   b) $$\frac{10.5 \times 18.5}{2} = 97.13 \text{ sq. in.}$$

   c) Area of 12″ diam. circle:
   $A_1 = \pi r^2 = \pi 36 = 113.04$ sq. in.
   Area of 26″ × 12 rectangle:
   $A_2 = 26'' \times 12'' = 312$ sq. in.
   Total Area = 312 + 113.04
   $A_T = 425.04$ sq. in.

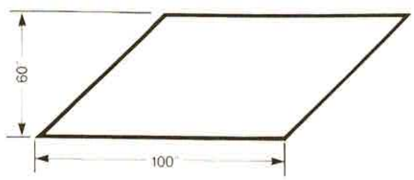


Figure 28

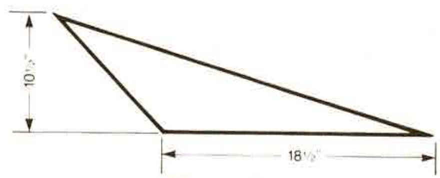


Figure 29

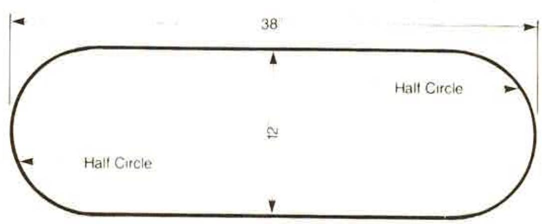
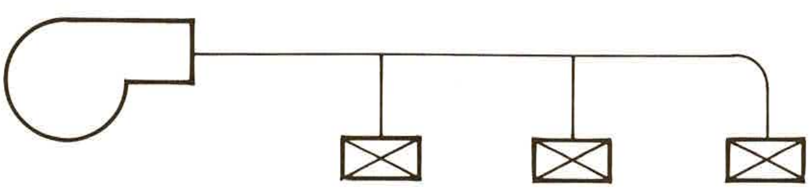


Figure 30

9. Fill in the percent of design for the three outlets (Figure 31).



	A	B	C
Design	875	900	650
Actual	800	850	750
% of Design	**91%**	**94%**	**115%**

Figure 31

10. In a hydronic system, if a pump is handling 525 GPM and the design flow rate is 435 GPM, what is the percent of actual flow? **121%**

11. The circumference of a pipe is 17 inches. What is its diameter?
0.45 feet (use decimal) **17″/π/12″ = 0.45′**

12. A rectangular tank that is 32″ long, 24″ wide, and 61″ high will hold **202.8** gallons when filled to the top.

$$\frac{\mathbf{32'' \times 24'' \times 61''}}{\mathbf{231\ cu.in./gal}} = \mathbf{202.8\ gal.}$$

NEBB

CHAPTER 15 INDEX