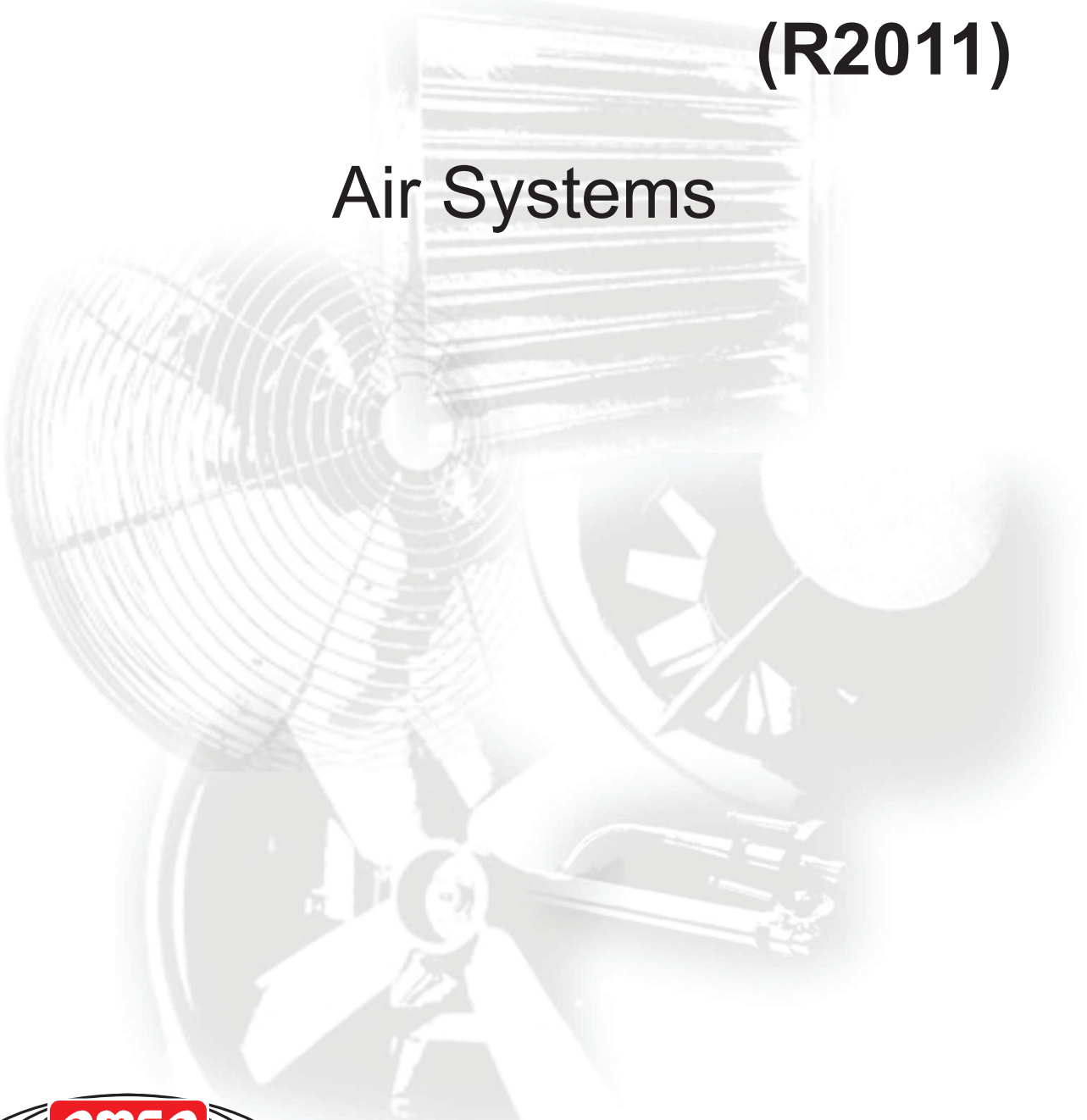
AMCA Publication 200-95 (R2011)

Air Systems



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Air Systems

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Authority

AMCA Publication 200-95 was adopted by the membership of the Air Movement and Control Association International, Inc. in 1995, and it was reaffirmed in 2000, 2007, and 2011.

Foreword

This publication is intended to provide basic information needed to design effective and energy efficient air systems. in those cases where the system handles a gas other than air, the design data must be modified to allow for the different physical properties of the gas being used.

Discussion is limited to systems where there is a clear separation of the fan inlet and outlet and does not cover applications in which fans are used only to circulate air in an open space.

The design of most air sytems is based on airflow velocities which result in turbulent airflow. Some applications require very low velocities, resulting in laminar flow conditions; others may require very high velocities, approaching the speed of sound (Mach 1). The information given in this publication applies only to turbulent flow conditions and not to these special systems using very low or very high flow velocities.

The flow through an air system will stabilize when the total pressure provided by the fan is exactly equal to the total pressure losses in the system.

To achieve the desired airflow in the system the diesgner must have complete information on:

a) System Pressure Losses: The total pressure loss due to friction losses, shock losses, dissipation of velocity pressure at the system discharge, and static pressure differences between the entry and discharge openings. System pressure losses are discussed in detail in Section 4.

b) Fan Performance Characteristics: The relationship of the total pressure rise and the volume flow generated by the fan. Fan performance characteristics are reviewd in Section 5. More complete information is contained in AMCA Publication 201, *Fans and Systems*.

c) System Effect: The effect on the performance of the fan resulting from the difference between the fan inlet and outlet connections to the installed system and the standardized connections used in laboratory tests to obtain fan performance ratings. A practical approach to estimating System Effects is explained in AMCA Publication 201, *Fans and Systems*.

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Related AMCA Standards and Publications

Publication 200 ***AIR SYSTEMS***

System Pressure Losses
Fan Performance Characteristics
System Effect
System Design Tolerances

Air Systems is intended to provide basic information needed to design effective and energy efficient air systems. Discussion is limited to systems where there is a clear separation of the fan inlet and outlet and does not cover applications in which fans are used only to circulate air in an open space.

Publication 201 ***FANS AND SYSTEMS***

Fan Testing and Rating
The Fan "Laws"
Air Systems
Fan and System Interaction
System Effect Factors

Fans and Systems is aimed primarily at the designer of the air moving system and discusses the effect on inlet and outlet connections of the fan's performance. System Effect Factors, which must be included in the basic design calculations, are listed for various configurations. AMCA 202 and AMCA 203 are companion documents.

Publication 202 ***TROUBLESHOOTING***

System Checklist
Fan Manufacturer's Analysis
Master Troubleshooting Appendices

Troubleshooting is intended to help identify and correct problems with the performance and operation of the air moving system after installation. AMCA 201 and AMCA 203 are companion documents.

Publication 203 ***FIELD PERFORMANCE MEASUREMENTS OF FAN SYSTEMS***

Acceptance Tests
Test Methods and Instruments
Precautions
Limitations and Expected Accuracies
Calculations

Field Performance Measurements of Fan Systems reviews the various problems of making field measurements and calculating the actual performance of the fan and system. AMCA 201 and AMCA 202 are companion documents.

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TABLE OF CONTENTS

1. Introduction 1

1.1 Air system components 1

2. Symbols and Subscripts 3

2.1 Symbols and subscripted symbols 3

2.2 Subscripts 3

3. Properties of Air 4

3.1 Properties of gases 4

4. Airflow 5

4.1 Flow conditions 5

4.2 Flow about immersed bodies 6

4.3 Ducted flow 6

4.4 System losses 7

4.5 *System Effects* 11

5. The Flow System 13

5.1 Concepts of pressure 13

5.2 Examples of pressures in duct systems 13

5.3 Conservation of energy 17

5.4 Fan total and static pressure 20

5.5 The total system 22

5.6 Types of fan system 26

5.7 System resistance factors 27

5.8 System design and loss calculations 29

5.9 Density effects in air systems 31

6. System Design and Tolerances 34

6.1 Point of operation 34

6.2 Fan performance 35

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Full document details: AMCA 200 2011

6.3 Effects of system changes 36

6.4 Variable systems 36

Annex A. SI / I-P Conversion Table 40

Annex B. Standard Atmospheric Data Versus Altitude Charts 41

Annex C. Psychrometric Density Tables 43

Annex D. Friction Charts 47

Annex E. Air Density Correction Factor Charts 49

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Air Systems

1. Introduction

An air system is any assembly of ducts, filters, conditioning devices, dampers, louvers, fans, etc., the main purpose of which is to move air from one place to another in a controlled fashion. Most air systems draw air from one space and discharge it into another.

Air systems are often required to operate satisfactorily in a wide range of environmental conditions. The conditions which will be encountered must be considered in the design of the ducts, pipes, etc., which will contain the airflow and constitute the boundary of the system.

1.1 Air system components

A typical air system may contain one or more of the following (see Figure 1):

a) System inlet
b) Distribution system
c) Fan
d) Control device
e) Conditioning device
f) System outlet

1.1.1 System inlet. An air system usually includes devices such as louvers, filters, screens, guards, grilles, etc., where the air enters the system. These are used for safety reasons as well as to inhibit the entry of rain, dust, and other unwanted matter. Their appearance may be important as they are usually visible on the exterior of a structure.

1.1.2 Distribution system. Most air systems are made up of ducts specially designed and constructed to convey air from the system inlet(s) to the system outlet(s). In some cases, enclosed spaces in the structure such as plenums above ceilings or holes in walls may be used to confine and direct the flow.

1.1.3 Fan. Understanding the design and opera-tion of air systems begins with an understanding of the various types of fans, their performance characteristics, and their applications.

A fan is required in order to produce the pressure differential which results in the flow of air through a system. The fan must be carefully selected to meet the specified airflow and pressure for proper system operation. Different fan designs produce different pressure-volume and fan power relationships, which are critical to air system operation. Refer to Figure 4.2, AMCA Publication 201-90.

1.1.4 Control devices. In many air systems it is necessary to regulate and control the flow through the system in response to some monitoring signal, usually temperature or pressure. It may be also necessary to regulate the flow in the individual branches of the system. Control devices such as dampers function by controlling the amount of airflow. In some cases, the output of the fan can be varied by other methods (variable speed motor, variable inlet vanes, variable pitch impeller, etc.)

1.1.5 Conditioning device. Most air systems are designed to take air from the inlet and change its condition before discharging it at the outlet. Changes may include the temperature, humidity, pressure, contaminant level and cleanliness, etc., of the air. Many conditioning devices require outside energy sources, for example, heating and cooling coils; other components such as filters are passive devices and have no external energy connection. All conditioning devices increase the pressure drop across the system and this effect must be considered in the selection of the fan.

1.1.6 System outlet. An air system usually includes a special component at the termination of the system or at the end of each of the system's branches, such as a simple screen or louver. In many cases the distribution of the air at the outlet to the receiving space is very important, e.g., in an occupied air conditioned room. These systems require carefully selected outlets and diffusing devices to achieve desirable air motion and temperature conditions in the conditioned space. Typical devices are ceiling diffusers and grilles. In some cases these may incorporate control devices such as dampers and mixing boxes.

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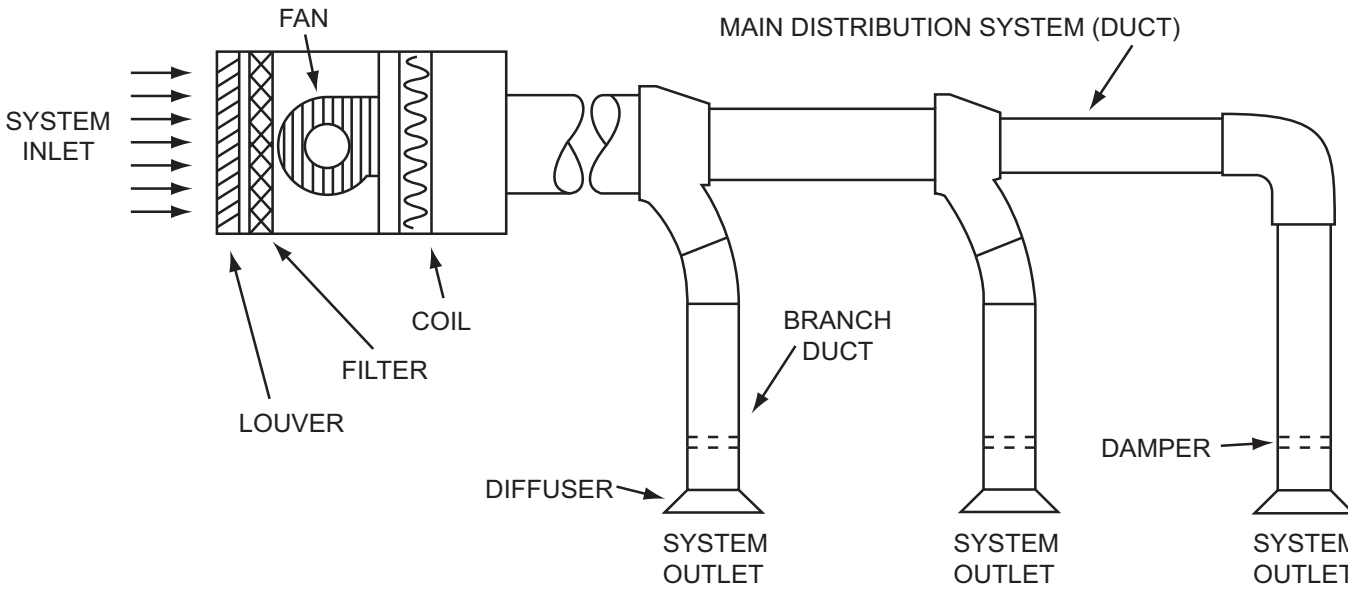

Figure 1 - Typical Air System

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2. Symbols and Subscripts

2.1 Symbols and subscripted symbols

Symbol	Description	SI	(I-P)
A	Area	m^2	(ft^2)
A_e	Area-Orifice Equivalent to System	m^2	(ft^2)
A_o	Area-Nozzle with no loss	m^2	(ft^2)
ah	Absolute Humidity	kg/m^3	(lb/ft^3)
C	Dynamic Loss Coefficient	Dimensionless	
C_d	Coefficient of Discharge	Dimensionless	
C_n	Coefficient of Nozzle Discharge	Dimensionless	
c	Speed of Sound	m/s	(ft/s)
D	Duct Diameter and Equivalent Diameter	m	(ft)
E	System Resistance Curve	Dimensionless	
ε	Absolute Surface Roughness Height	m	(ft)
f	Friction Coefficient	Dimensionless	
g	Gravity	m/s^2	(ft/s^2)
γ	Ratio of Specific Heats	Dimensionless	
K	System Effect Factor (System)	Dimensionless	
L	Length	m	(ft)
μ	Air Viscosity, Absolute	$N\text{-}s/m^2$	(lbm/ft-s)
ΔP	Pressure Differential	Pa	(in. wg)
P	Pressure	Pa	(in. wg)
P_s	Static Pressure	Pa	(in. wg)
P_{sx}	Static Pressure at Plane x	Pa	(in. wg)
P_t	Total Pressure	Pa	(in. wg)
P_{tx}	Total Pressure at Plane x	Pa	(in. wg)
P_v	Velocity Pressure	Pa	(in. wg)
p	Atmospheric Pressure	Pa	(in. Hg)
Q	Airflow Rate	m^3/s	(cfm)
Q_x	Airflow Rate at Plane x	m^3/s	(cfm)
R	Gas Constant	J/kg-K	(ft-lb/lbm-°R)
Re	Reynolds Number	Dimensionless	
rh	Relative Humidity	%	(%)
ρ	Air Density	kg/m^3	(lbm/ft^3)
ρ_x	Air Density at Plane x	kg/m^3	(lbm/ft^3)
SEF	System Effect Factor (Fan)	Pa	(in. wg)
S_R	System Resistance Factor	m^{-4}	(ft^{-4})
sh	Specific Humidity (_/_ dry air)	kg/kg dry air	(lb/lb) dry air
t	Temperature	°C	(°F)
t_d	Dry-Bulb Temperature	°C	(°F)
t_w	Wet-Bulb Temperature	°C	(°F)
V	Average Velocity	m/s	(ft/min)
v	Velocity - At any Point	m/s	(ft/min)
Y	Expansion Factor	Dimensionless	
Z	Altitude	m	(ft)
~	Is Proportional to	Dimensionless	

2.2 Subscripts

Subscript	Definition	Subscript	Definition
a	Element a	n	Reference to Nozzle
b	Element b	O	Plane of System Outlet
c	Element c - Combined	x	Plane 0, 1, 2,...as appropriate
d	Discharge	x,x'	Between Planes x and x'
E	Plane of System Entry	1	Plane of Fan Inlet
F	Fan	2	Plane of Fan Discharge

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3. Properties of Air

Atmospheric air is a mixture of several gases, water vapor, and impurities. The relative amounts of the important constituents for dry, sea level air are given in Table 3.1. This table may be considered representative of air at any altitude.

Table 3.1 - Dry Air Composition, Fraction

Component	Volume	Weight
Nitrogen	0.7809	0.7552
Oxygen	0.2095	0.2315
Argon	0.0093	0.0128
Carbon Dioxide	0.0003	0.0004
Also slight traces of neon, hydrogen, helium, krypton, ozone and others		

Although the gas composition of air can be considered essentially constant, the amount of water vapor contained in the air can vary greatly. The properties of moist air are dependent upon the relative amount of water vapor and dry air, therefore, in defining the properties of moist air, this relative amount must be defined (see Section 3.1.5 Humidity).

The impurities in the air are of various forms, but basically can be divided into two categories: a) particulates which can be either solid or liquid, and b) mixtures, which can be either gas or vapor. The distribution of these impurities is not uniform on an atmospheric scale, but can be considered uniform for the purposes of air system design. Since air is a mixture of several gases, the behavior of air under varying conditions can be best understood by first reviewing the behavior of pure gases.

3.1 Properties of gases

A gas may be defined as a compressible substance which has no free surfaces and occupies all portions of its container. The important properties of an ideal gas are listed below.

3.1.1 Density. The density of a gas is defined as the total mass of the molecules in a unit volume. In the SI system density is given in kilograms per cubic meter (kg/m^3); in the I-P system, density is given in pounds per cubic foot (lbm/ft^3).

For purposes of uniformity, standard air has been defined as air with a density of 1.2 kg/m^3 (0.075 lbm/ft^3) and an absolute viscosity of 18.19 × 10^{-6} N-s/m^2 (1.222 × 10^{-5} lbm/ft-s). This is substantially equivalent to air at a temperature of 20°C (68°F), 50% relative humidity, and a barometric pressure of 101 kPa (29.92 inches mercury) at sea level. The ratio of specific heats, (γ), is taken to be 1.4, which is the expected value for a perfect diatomic gas.

The temperature and barometric pressure of atmospheric air vary widely with weather conditions and geographical location, most noticeably altitude. In order to simplify design, standard atmospheric conditions have been defined which give the variation of atmospheric pressure, temperature, and, therefore, density with altitude. Annex B lists these variations.

3.1.2 Pressure. In an air system, pressure is the force exerted by the air molecules on the surfaces which make up the system. Since air molecules are always in motion, they continuously collide with other air molecules or a solid surface. All these collisions are considered to be perfectly elastic and, in the case when a molecule strikes a surface, the surface experiences a force equal and opposite to the time rate of change of momentum of the rebounding molecule. This force causes the gas to exert an overall pressure on an immersed body and this force per unit area is referred to as the pressure. In air system work, the units of pressure are given in terms of force per unit area. The unit of measure for pressure in the SI system is the Pascal (Pa); in the I-P system the units are inches of water gauge (in. wg).

3.1.3 Temperature

3.1.3.1 Thermal relationships. The kinetic energy of gas molecules increases with increasing temperature. The important effects of this fact are stated in Boyle's Law and Charles' Law, which state that the volume of a perfect gas varies inversely with absolute pressure and directly with absolute temperature, respectively. The total effect is more properly stated by the equation of state:

$$PV = m\mathrm{R}T \qquad \text{Eq. 3.1-1}$$

or

$$P = \rho \mathrm{R}T \qquad \text{Eq. 3.1-2}$$

Where:

P = Pressure
V = Volume
m = mass
R = Gas Constant
T = Absolute Temperature
ρ = m/V = density

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In the design of most air systems, it is acceptable to assume that the gas is incompressible, therefore, the air density may be considered constant, and therefore, the absolute pressure and absolute temperature are directly proportional.

3.1.3.2 Dry-bulb, wet-bulb and dew point temperature. Unless otherwise specified, the temperature of an air-water vapor mixture is that temperature which is indicated by an ordinary or dry-bulb thermometer. This dry-bulb temperature is the temperature of both the air and the water vapor in the mixture. The wet-bulb temperature may be determined by exposing a wetted bulb in a moving air-water vapor mixture until equilibrium is obtained. The wet-bulb temperature will be lower than the dry-bulb temperature as long as evaporation continues. If no evaporation is possible, the mixture is saturated and the wet and dry-bulb temperatures for this condition will be identical. The dew point temperature of an air-water vapor mixture is the saturation temperature corresponding to the absolute humidity of the mixture. The dew point temperature may also be considered as that temperature at which condensation begins when the mixture is gradually cooled.

3.1.4 Viscosity. A non-perfect gas, such as air, is capable of exerting a force parallel to the surface of a body which is moving with respect to the gas. The magnitude of the force parallel to the surface is used to define an important property of non-perfect gases - viscosity. The effects of viscosity on the behavior of real gases cause resistance to flow; the resistance is proportional to the velocity gradients which exist in the gas. The absolute viscosity (μ) is defined as the shearing stress for a unit rate of change of velocity. The absolute viscosity has units of newton-sec per meter squared (N-s/m^2) in the SI system and pound mass per foot-second (lbm/ft-s) using I-P units.

3.1.5 Humidity. The density of atmospheric air is also a function of the humidity. Although the change in density due to a change in humidity is not large, it is often significant and air system designers should be aware of these changes. Remember that increasing humidity lowers the density since water vapor is lighter than dry air. The density of saturated air for various barometric and hygrometric conditions is shown in Annex C.

Partially saturated air contains vapor that is superheated, that is, the temperature of the mixture and, therefore, that of the vapor is higher than the saturation temperature for the existing vapor pressure.

The relative humidity (*rh*) of an air-water vapor mixture is defined as the ratio of the vapor pressure existing compared to the vapor pressure at saturation for the same dry-bulb temperature. This is also equal to the ratio of the mole fractions under the same condition. Relative humidity is always expressed as a percent.

Specific humidity (*sh*) is the actual mass (weight) of the water vapor existing per unit mass (unit weight) of dry air or gas. Absolute humidity (*ah*) may be expressed in kilograms (pounds) of water vapor per cubic meter (cubic foot) of mixture. The humidity of an air-water vapor mixture is often expressed by giving either relative humidity or a wet-bulb depression.

4. Airflow

The flow of any fluid between two points is caused by the existence of a pressure differential between the two points. It is the purpose of this section to explain the parameters that may affect the flow of a gas between two points.

4.1 Flow conditions

Most air systems are designed in the incompressible range. Where compressibility is a factor, Mach number and Reynolds number must be considered.

The magnitude of these parameters gives an indication of the effects which can be expected from the deviations in the non-perfect gas behavior from that of a perfect gas.

4.1.1 Mach number. Mach number, for our purposes here, is the ratio of the velocity of an airstream to the speed of sound in that airstream.

Mach number = V/c

Where:

V = velocity of air, m/s (ft/s)
c = speed of sound in air, m/s (ft/s)

The speed of sound is a function of temperature and is the speed at which very small pressure disturbances are propagated throughout the gas. The speed of sound is proportional to the square root of the absolute temperature, and for standard air is approximately 345 m/s (1130 ft/s). If the Mach number is small and no large static pressure changes are introduced by mechanical means, the flow may be considered incompressible, that is, the density is everywhere constant. Air can be considered incompressible if the fan total pressure rise is less than 2980 Pa (12 in. wg).

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4.1.2 Reynolds number. The ratio of the inertia force to the viscous force caused by changes in velocity within the fluid element is known as the Reynolds number.

$$\mathrm{Re} = DV\left(\frac{\rho}{\mu}\right) \qquad \text{Eq. 4.1-1A SI}$$

$$\mathrm{Re} = DV\left(\frac{\rho}{60\mu}\right) \qquad \text{Eq. 4.1-1A I-P}$$

$$= \left(\frac{DV}{\gamma}\right)$$

For standard air:

$$\mathrm{Re} = 65970.3DV \qquad \text{Eq. 4.1-1B SI}$$

$$\mathrm{Re} = 102.3DV \qquad \text{Eq. 4.1-1B I-P}$$

Where:

D = Any convenient reference dimension, m (ft)
V = Velocity, m/s (ft/min)
μ = Absolute viscosity, N-s/m^2 (lbm/ft-s)
γ = Kinematic viscosity, m^2/s (ft^2/s)
ρ = Density, kg/m^3 (lbm/ft^3)

For flow about immersed bodies, D is normally taken as the length of body in the direction of flow. In ducted flow, D is normally taken as the diameter of the duct; in unducted flow, D is normally taken as the diameter of the opening through which the flow passes.

For a fan, D is equal to the impeller tip diameter and is only proportional to conventional Reynolds numbers. The Reynolds number provides a convenient non-dimensional means of comparing two flows.

4.2 Flow about immersed bodies

If a solid body is immersed in a flowing stream of a gas, the direction of flow of the gas will be parallel to the surface of the solid body. The changes in the direction of the molecules close to the body exert forces on the body which when taken over the entire body, are perpendicular to the direction of the gas flow.

A non-perfect gas will also exert a force parallel to the direction of the velocity, due to the viscosity of the gas. This force, usually called drag, is due to two effects. The first is the shearing force set up within the molecules of the gas resulting from the molecules decelerating from the gas velocity to zero velocity when in contact with the body. This is called skin friction drag, and, for streamlined bodies closely aligned with the flow, represents the entire drag force. For blunt bodies, which may be streamlined bodies at large angles to the flow, profile drag exists. Profile drag is caused by the inability of the flow, due to its viscous effects, to follow the body shape. The inability to follow the body shape creates a wake of very turbulent flow which in effect creates the profile drag force. These wake effects are the predominant cause of flow losses in systems.

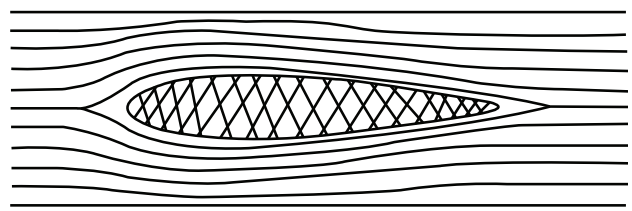

Figure 4A - Skin Friction Drag

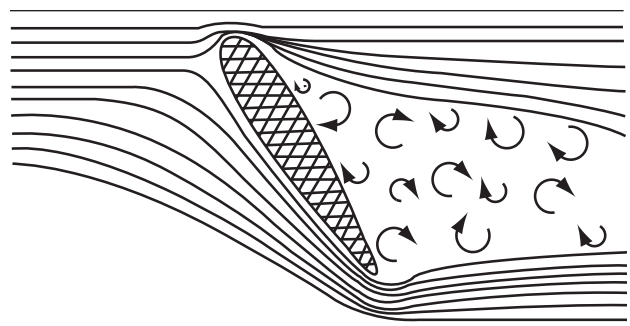

Figure 4B - Profile Drag

Figures 4A and 4B illustrate skin friction drag and profile drag.

4.3 Ducted flow

When air flows through a duct of constant cross-section, the average velocity remains constant and is parallel to the center line of the duct. Due to friction, the velocity at the duct wall is zero and the average velocity profile can be defined as either of two conditions:

a) Laminar Flow: Flow in which the air velocity vectors are parallel to the duct wall. This type of flow is described as smooth.

b) Turbulent Flow: Flow in which air velocity vectors at various points across the duct are at various angles, up to and including reverse flow.

Except for extremely low air velocities, laminar flow does not exist and all duct flow involving air can be considered to be in the transition region between laminar and fully turbulent flow. The transfer of

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energy from the high velocity section in the center of the duct to the low velocity section near the duct wall causes a marked resistance to the flow. This resistance varies linearly with the length of the duct and approximately with the square of the average velocity in the duct. The resistance is also a function of the Reynolds number of the flow, which is calculated using the average velocity in the duct, the duct diameter, and the surface roughness of the duct wall.

The velocity profiles in a duct system for fully developed flow will vary depending on whether the flow is laminar or turbulent and the degree of duct roughness. Velocity profiles of various flow conditions are shown in Figure 4C.

The absolute velocity of the air stream will vary substantially over the cross-sectional duct area, but for duct systems the velocity used for determining the velocity pressure is always the average velocity given by:

V average = Q/A Eq. 4.3-1

Where:

V = Velocity, m/s (ft/min)
Q = Flow rate, m^3/s (cfm)
A = Area of the cross-section where the flow occurs, m^2 (ft^2)

The duct velocity profiles shown in Figure 4C are uniform along the length of the duct and symmetrical around the center line. Where there are disturbances in the ducts, such as turns, expansion or contraction, etc., the velocity profile across the duct can become very asymmetrical as shown in Figure 4D.

The flow will return to a normal velocity profile after a disturbance if there is sufficient length of straight duct to allow the velocity distribution to regain uniformity. A minimum of 2½ equivalent duct diameters of straight duct is required to attain a normal velocity profile for velocities of 12.7 m/s (2500 ft/min) or less. Add one duct diameter for each additional 5.08 m/s (1000 ft/min). See AMCA Publication 201-90, *Fans and Systems*.

4.4 System losses

The losses in total pressure for flow through a system are caused by two factors: **friction losses** due to viscosity as the air flows along the surface of ducts and other system elements, and **dynamic losses** due to the turbulent wake caused by changes in direction and separation of the flow around obstructions.

In addition to the losses in total pressure in a system caused by friction losses and dynamic losses, there are losses due to *System Effects*.

System Effects occur because of the differences between the fan inlet and outlet connections to the installed system and the standardized connections used in laboratory tests to obtain fan performance ratings. AMCA Publication 201, *Fans and Systems*, gives specific details on *System Effects* related to fans. *System Effects* related to series system elements are covered further in Section 4.5 of this publication.

4.4.1 Duct friction losses. In the normal range of air systems for HVAC and industrial applications, the flow falls into the transition region between laminar flow and complete turbulent flow. In this region the losses due to friction are a function of Reynolds number and the relative roughness of the duct wall. The pressure loss in the transition region will vary at slightly less than the square of the velocity. The pressure loss due to friction for flow in ducts may be calculated from the Darcy-Weisbach equation:

$\Delta P_t = f(L/D)\, P_v$ Eq. 4.4-1

Where:

ΔP_t = Total pressure loss due to friction, Pa (in. wg)
f = Friction factor, dimensionless
L = Length of duct, m (ft)
D = Diameter of pipe, m (ft)
P_v = Velocity pressure, Pa (in. wg)

In the transition flow range, the value of the friction factor cannot be calculated directly. It can be obtained from the Moody diagram or by iterative solution of the Colebrook equation. See the ASHRAE Handbook: *Fundamentals*, chapter on *Duct Design*, for a more complete discussion of duct friction losses.

The Moody diagram, Figure 4E, shows the relationship of the friction factor, Reynolds number and duct roughness (ε) in meters (feet). Most applications are in the transition region between laminar and full turbulent flow conditions.

Using duct friction charts (see Annex D) is the most common method of determining friction losses. These charts are based on ducts having average roughness and standard air density. Correction factors must be applied for ducts having different roughness, and for variations in air density and viscosity.

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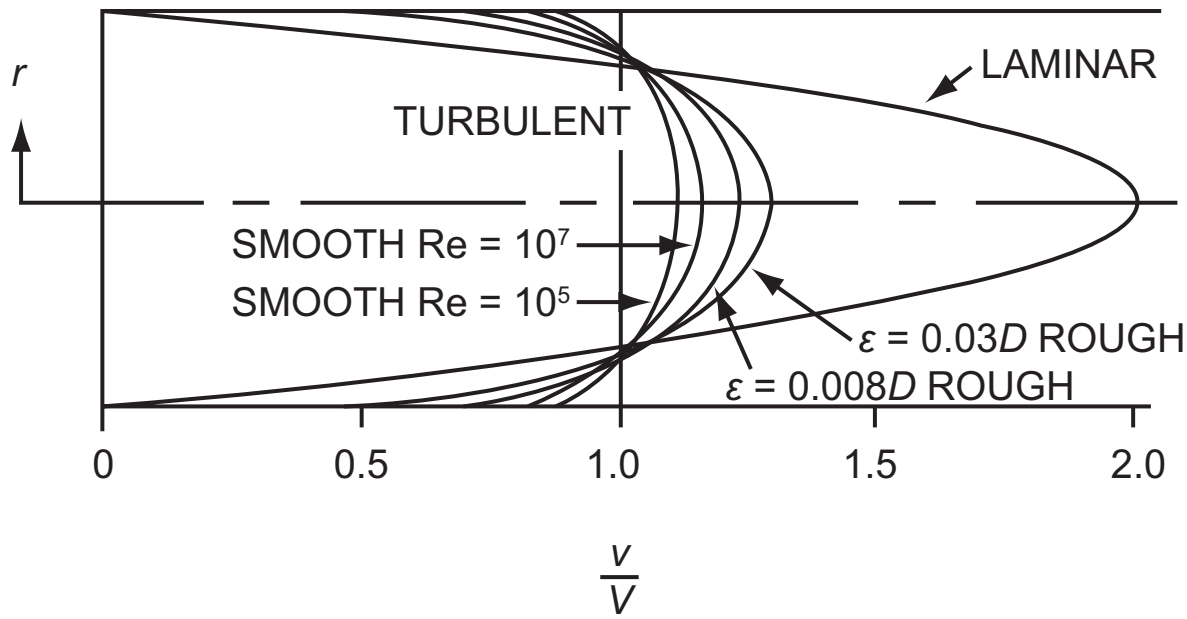

D = Duct Diameter
ε = Duct Roughness
Re = Reynolds Number
v = Velocity at any Point
V = Average Velocity
r = Radius

Figure 4C - Velocity Profiles in a Round Duct for Various Reynolds Numbers and Duct Roughness

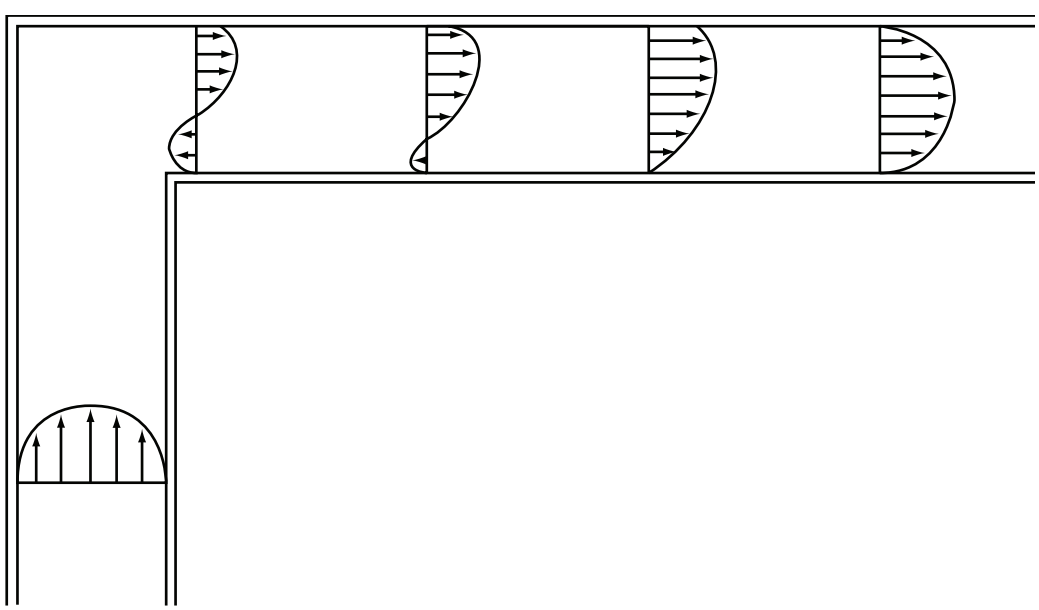

Figure 4D - Changing Velocity Profiles

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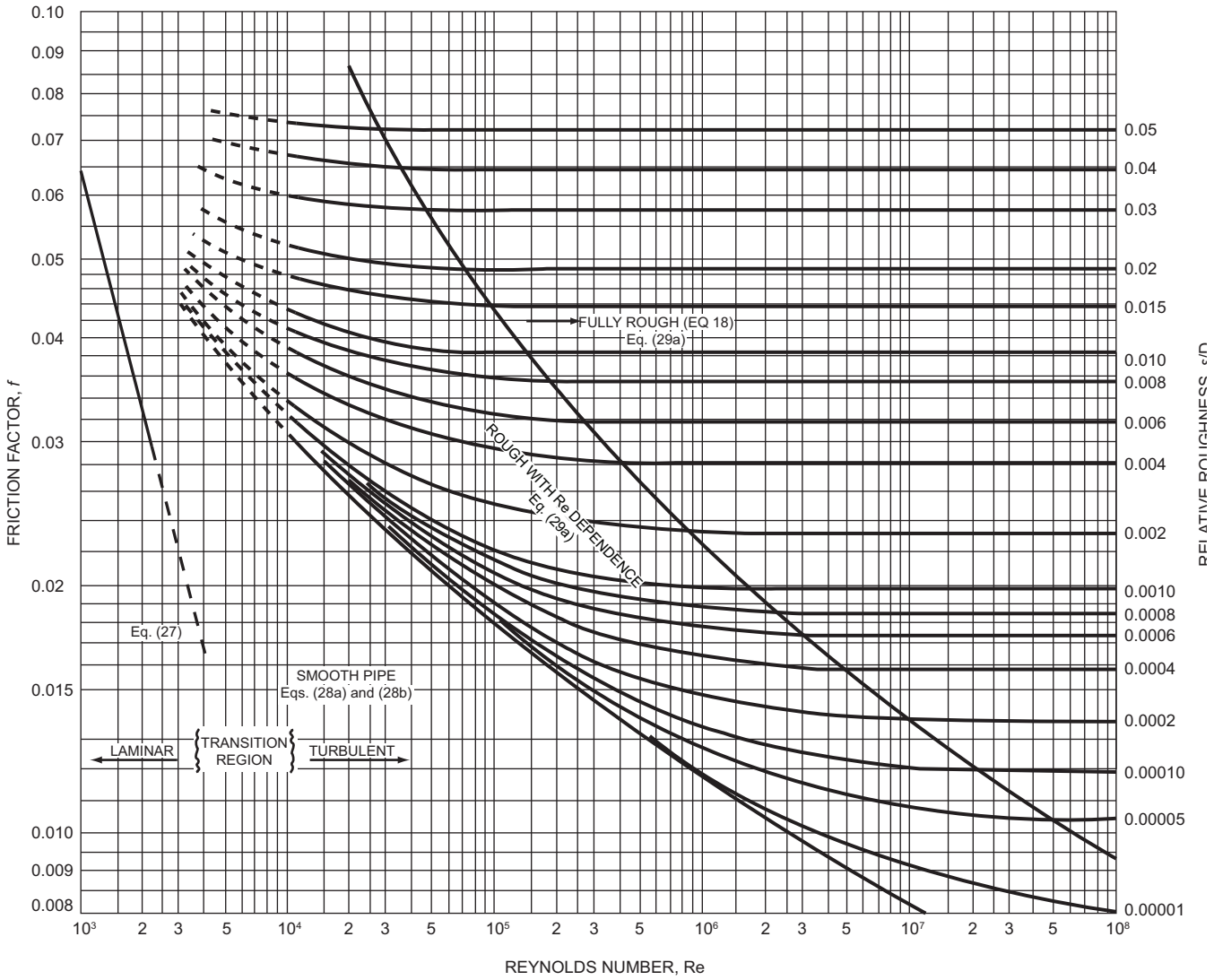

Figure 4E - Moody Diagram

Reprinted by permission of the American Society of Heating, Refrigerating and Air-Conditioning Engineers, Atlanta, Georgia, from the 1993 *ASHRAE Handbook-Fundamentals*. (Moody 1944). Values on the chart are the same for both the SI and I-P systems. Equation numbers refer to equations in the source document.

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Loeffler[1] has developed simplified equations for the friction factor in the normal range of flow conditions found in industrial and HVAC air systems. The equations provide for direct calculation of duct friction total pressure losses. These equations yield results that are accurate within ±5% and are conservative over most of the range of flow.

For aluminum ducts, medium smooth:

ε = 0.0000457 m (0.00015 ft)

$$P_{t\ loss} = a\left(\frac{LQ^{1.863}}{D^{4.93}}\right)$$

Where:

$a = 1.764 \times 10^{-2}$ Eq. 4.4-2 SI

$a = 4.816 \times 10^{-9}$ Eq. 4.4-2 I-P

For galvanized steel ducts, average:

ε = 0.0001524 m (0.0005 ft)

$$P_{t\ loss} = a\left(\frac{LQ^{1.921}}{D^{5.066}}\right)$$

Where:

$a = 1.717 \times 10^{-2}$ Eq. 4.4-3 SI

$a = 3.534 \times 10^{-9}$ Eq. 4.4-3 I-P

For fiberglass ducts or lined ducts, fabric and wire flexible ducts (wire covered with fabric), medium rough,

ε = 0.00091443 m (0.003 ft)

$$P_{t\ loss} = a\left(\frac{LQ^{1.965}}{D^{5.208}}\right)$$

Where:

$a = 2.093 \times 10^{-2}$ Eq. 4.4-4 SI

$a = 3.64 \times 10^{-9}$ Eq. 4.4-4 I-P

Where:

$P_{t\ loss}$ = Total pressure loss, Pa (in. wg)
Q = Flow rate, m^3/s (cfm)
D = Duct diameter, m (ft) (or equivalent diameter of rectangular ducts) $D\ equivalent = (4ab/\pi)^{0.5}$ where a and b are the sides in m (ft)
L = Duct length in m (ft)

Correction factors for density and viscosity variations need to be applied for conditions other than standard air. See Annexes B and E.

4.4.2 Dynamic losses. Dynamic losses occur where there are changes in velocity or direction in the air system and are due primarily to the profile drag. Dynamic losses are proportional to the square of velocity, and therefore, are proportional to the velocity pressure.

$\Delta P_t \sim V^2 \sim P_v$

Dynamic pressure loss data are given in a number of forms such as pressure loss for given volume or velocity, equivalent length of duct, or velocity pressure multiplier, and is available from manufacturers' data and handbooks such as: ASHRAE, the Industrial Ventilation Guide and SMACNA.

Except at duct exits, dynamic losses occur along some length and cannot be separated from friction losses. For practical purposes the dynamic losses are assumed to be concentrated at one point and the friction losses are included as part of the duct friction. Dynamic loss coefficients for duct fittings are based on zero length. For friction loss calculations, the centerline length of the duct fitting is taken as the length of the fitting.

4.4.3 Dynamic loss coefficient. There are two common methods of expressing dynamic losses. These are: 1) The equivalent length of duct method, and 2) the loss coefficient method. The equivalent length of duct method replaces the dynamic loss of fittings (elbows, tees, branches, etc.) with a length of duct that will have an equivalent loss. The equivalent length of duct for all of the dynamic losses are added to the straight duct length. A friction chart showing the loss is then used to determine the total loss in the system.

The dynamic loss coefficient method is based on the fact that all losses in a system are functions of the velocity pressure and can be calculated by a corresponding dynamic loss factor multiplied by the velocity pressure. These losses are added to the straight duct friction loss to determine the total loss in the system.

The dynamic loss coefficient method is preferred because it is usually quicker and offers the advantage of faster recalculation when other branch duct sizes are tried.

Dynamic losses are proportional to the velocity pressure occurring in the system element and,

1. Loeffler, J. J., Simplified Equations for HVAC Duct Friction Factors, ASHRAE Journal, January, 1980

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