Air System Basics

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One of the fundamental concepts that apply to air systems is the fluid mechanics of conveying air. This article presents the basic principles that form the groundwork for the design, installation, operation, and troubleshooting of these systems.

Introduction

Air handling systems have been designed, and installed in nearly every building currently in use and will be utilized in almost every building design still on the drawing board today. Air systems in buildings are used principally as a means to transport energy or to remove contaminants although their functions are actually much more complex than these restricted definitions. The energy required for the movement of air for these purposes represents one of the largest categories of electrical consumption in most modern facilities. Thus, a fundamental knowledge of the physics underlying the function and operation of air handling systems is vital to every practicing HVAC engineer today.

However, even to deal with the fundamentals of air systems requires the HVAC engineer to be well grounded in an extensive group of fundamental scientific concepts – physics, fluid mechanics, heat transfer, psychrometries, combustion, and human physiology, to name only a few. A brief list of the functions of air systems clearly illustrates this point.

Air systems may be used to:

- Provide a sufficient quantity and temperature of air to maintain the space dry bulb temperature at a desired condition.
- Provide a sufficient quantity and dew point temperature of air to maintain the space specific humidity at a desired condition.
- Provide for space air circulation to promote heat and moisture transfer for human comfort and to facilitate proper mixing to obtain uniform space conditions.
- Provide ventilation air from outdoors to replenish oxygen, to dilute space contaminants, or to provide combustion air.
- Clean outdoor and re-circulated air for human health and protect the system heat transfer equipment.
- Provide a means for the exhaust of specific contaminants from the space.
- Provide a source of exhaust or supply pressurization to contain the spread of smoke in a fire situation within a building.

The fundamental concepts underlying these functions span the gamut of the scientific disciplines indicated above. This article will discuss only one portion of the basics of air systems-fluid mechanics of conveying air. Basic principles will be presented that form the groundwork for the design, installation, operation, and troubleshooting of such systems. The content has been arranged as a refresher for seasoned practitioners and also as a bridge from theory to practice for entry-level individuals.

Since air is a fluid, the four basic principles of fluid mechanics all have application in the conveying of air in an effective and energy-conserving manner. These concepts are fluid statics, conservation of mass, conservation of energy, and conservation of momentum. These will be treated separately before their practical application to air systems is presented.

Fluid Statics

Unlike the other concepts of fluid mechanics, the concept of fluid statics is based on the physics of fluids at rest rather than in motion. Though it is unusual in HVAC work to consider air that is not in motion, the concept of fluid statics provides the basis for manometers and the resulting set of units necessary to measure air pressure. Consider the vessel shown in *Figure 1A*, filled with fluid as indicated. The fluid statics principle indicates that the pressure read on the pressure gauge attached to the vessel would be:

$$P = \rho h \qquad (1)$$

where

P = pressure, lb per sq ft

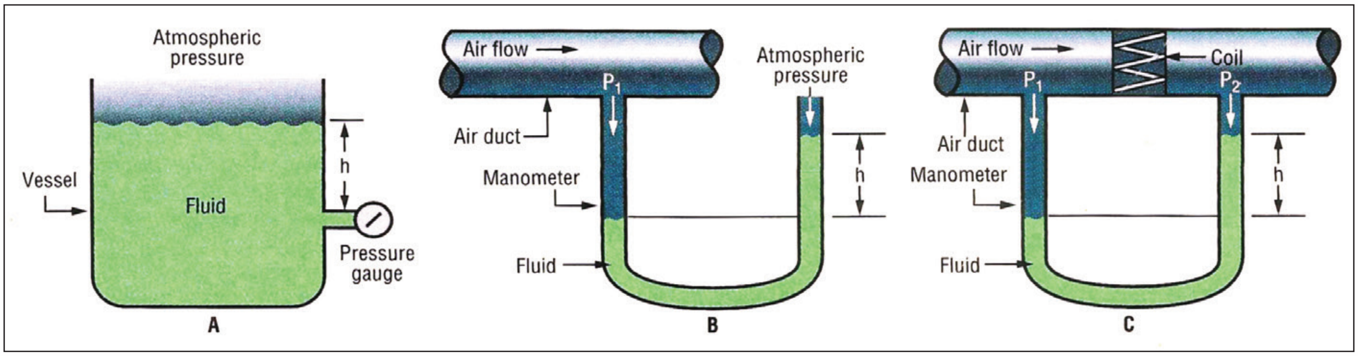

Figure 1: View A - measurement of static pressure, h. View B – measurement of static pressure, h, within a duct. View C – measurement of static pressure drop, h, across a heating or cooling coil.

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ρ = density of fluid in vessel, lb per cu ft

h = height of fluid level above pressure gauge, ft

The fluid creates a pressure on the gauge in excess of atmospheric pressure based on the height of the fluid column and the density of the fluid in the vessel. This principle permits the conversion of pressure to a unit of fluid "head" if the density of the fluid is known. Based on the density of water at 70 F (62.3 lb per cu ft), the fluid statics principle indicates a conversion factor of 1 psi equal to 2.31 ft of water.

Using the same principle, *Figure 1B* extends the concept of the measurement of air pressure in a duct using the fluid statics equation. Here, an air pressure, P_1, in excess of atmospheric pressure is exerted on one side of the manometer while the other side is exposed 'to atmospheric pressure'. The deflection of the fluid column indicates the head difference (and therefore the pressure difference) between the air pressure at the sensed point and atmospheric pressure. This deflection for air systems is commonly expressed as inches water gauge (in. WG) or inches water column (in. WC) as this represents a convenient unit for the range of air pressures typically encountered in HVAC work. Since the pressure measured in this case was above atmospheric pressure (with the deflected column rising toward the atmosphere), the pressure is termed positive. If the air pressure at P_1 had been below atmospheric pressure, the fluid column would have deflected in the opposite direction, and the pressure would be termed negative. Incidentally, the fluid statics equation and the standard density of air (0.075 lb per cu ft) would yield the following relationship: 1 in. WG = 0.036 psi = 69.22 ft of air.

The same fluid statics principle would apply in the use of a manometer to measure an air pressure difference across an obstruction to flowing air, such as the coil indicated in *Figure 1C*. Here, the pressure drop due to the energy level reduction of the air as it passes through the coil is registered by the deflection of the column. The pressure drop is always considered positive when the pressure reduces in the direction of air flow. Notice that no reference to atmospheric pressure is made in this measurement – only the pressure change that has occurred through the process.

Thus, the fluid statics principle provides us with the universal unit of inches water gauge to predict or measure pressures or pressure differences in air systems. Whether field readings are made with a U-tube or inclined-tube manometer, whether the fluid is water or gauge oil (with a lower density to increase deflections and make readings more accurate), or even if the gauge does not function on a manometer principle at all (as in the case of a Magnehelic gauge), the principle of fluid head as established by the fluid statics concept is the basis for all air pressure calculations and measurements in the HVAC industry.

Conservation of Mass

The law of conservation of mass, simply stated, is that fluid mass flowing in an enclosed duct is conserved – it is neither created nor destroyed as it passes through the system. In fluid mechanics, this concept is described by the continuity equation as displayed in *Figure 2A*. If air is flowing from left to right in this duct at a constant flow rate (steady flow), the mass flow rate would be

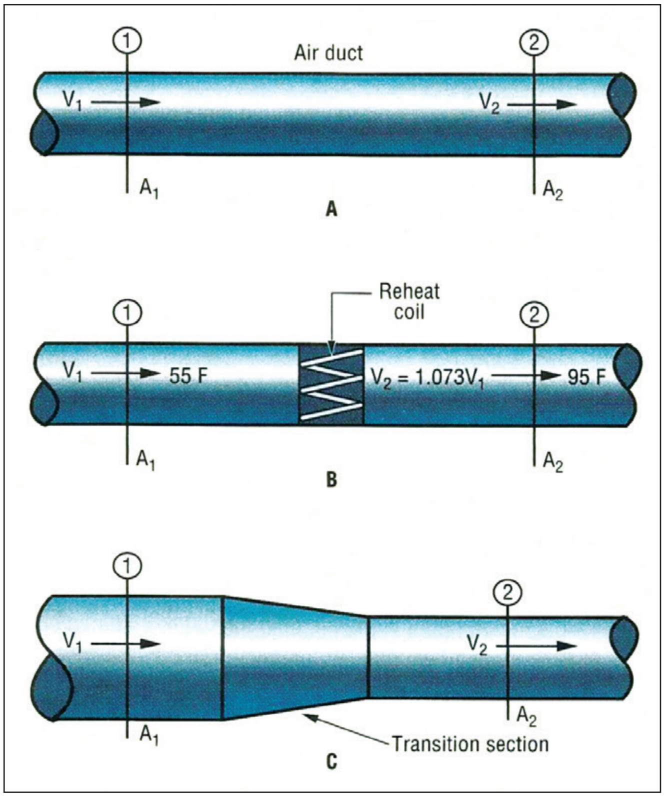

Figure 2: View A – classical form of the continuity equation for air systems is $Q = V_1A_1 = V_2A_2$, where Q = volume, V = velocity, and A = area. View B – change in density between 1 and 2 above is –7.3 percent. If $A_1 = A_2$, $V_2 = 1.073V_1$. Therefore, the assumption of incompressibility of a flowing air stream is correct within an accuracy limit of ±10 percent for typical HVAC processes. View C – When duct size changes, velocity also changes: $V_2 = V_1(A_1/A_2)$.

given by the following equation:

$$m = \rho VA = \text{constant} \quad (2)$$

where

m = mass flow rate, lb per min

ρ = air density, lb per cu ft

V = air velocity, fpm

A = duct area, sq ft

Referring to *Figure 2A*, writing the mass flow equation between Stations 1 and 2 would yield the following equation:

$$\rho_1 V_1 A_1 = \rho_2 V_2 A_2 \quad (3)$$

If the density were assumed to be constant between these two stations, then the classical form of the continuity equation for air systems would be derived:

$$Q = V_1 A_1 = V_2 A_2 \quad (4)$$

where

Q = volumetric flow rate, cfm

This equation forms the basis for the calculation of accelerating and decelerating flows, which will have significant application to all areas of air system technology. But before exploring this concept more fully, let us spend just a moment on the assumption of constant density that was made to derive Equation 4.

The assumption of constant density for a flowing air stream (or to put it another way, the assumption that the flow can

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be considered incompressible, as for a water stream) is only an approximation. Even if the duct is fully insulated with no temperature change in the air stream, any real duct system will experience pressure losses in the direction of air flow due to frictional and dynamic effects. This reduction in air pressure at a constant temperature will, according to the perfect gas law, reduce the density of the air slightly as it passes down the duct stream flowing with losses. For a constant area duct with a pressure drop from Station 1 to Station 2 of 10 in. WG, the density of the air at Station 2 will decrease by 2.4 percent and, based on Equation 3, the velocity will increase by 2.4 percent. If we superimpose on this the effect of temperature, the deviation becomes more extreme, as indicated in *Figure 2B*.

Assume that a duct-mounted reheat coil with a pressure drop of 0.5 in. WG heats an air stream from 55 to 95°F. Under these conditions, the density of the air at Station 2 will decrease by 7.3 percent and, assuming $A_1 = A_2$, the velocity at Station 2 will increase by 7.3 percent. Therefore, the assumption of incompressibility of a flowing air stream is correct within an accuracy limit of approximately ±10 percent for typical HVAC processes. Since the accuracy is acceptable for most measurements that are made in the field , the assumption is valid. However, when evaluating the necessity of a safety factor in calculating pressure requirements for the selection of a fan, remember this margin of error is inherent in the incompressible fluid mechanics on which most of our calculations are based.

The real power of the continuity equation comes from its ability to predict the volumetric flow rate based on the duct area and velocity and its ability to predict duct velocities based on changes in the geometry of the duct system. *Figure 2C* indicates the use of the continuity equation for a simple process. Air flowing from Station 1 to Station 2 passes through a transition fitting. Since the area of the duct at Station 2 is smaller than at Station 1, it is intuitively obvious that the flow has accelerated between these two stations. The continuity equation permits the exact calculation of this effect based on the use of Equation 4:

$$V_2 = V_1(A_1 / A_2) \quad (5)$$

Knowing the velocity entering the fitting and the ratio of the flow areas, we can calculate the velocity leaving the fitting.

Conservation of Energy

The law of conservation of energy is based on the principle that energy is neither created nor destroyed as it passes through a system. The steady flow energy equation that is developed from this concept sets up an accounting format to keep track of all of the energy forms. It is obtained from the first law of thermodynamics, which considers both mechanical and thermal forms of energy. It states that the amount of heat added to the air stream as it passes through a system is equal to the change in energy content of the air stream plus any work done by the air stream. The steady flow equation written between Stations 1 and 2 in *Figure 2A* per pound of air flowing would have the following form:

$$W_{1\text{-}2} + (V_1^2 / 2g) + JU_1 + JQ_{1\text{-}2} + P_1\nu_1 + Z_1 =$$

$$(V_2^2 / 2g) + JU_2 + P_2\nu_2 + Z_2 \quad (6)$$

where

$W_{1\text{-}2}$ = work done on the air stream between Stations 1 and 2 (this value would be negative if it represented work done by the air stream), ft -lb per lb of air flowing or ft

V = air stream velocity, fps

g = acceleration of gravity, 32.2 fps^2

J = mechanical equivalent of heat, 778.2 ft-lb per Btu

U = internal energy, Btu per lb

$Q_{1\text{-}2}$ = heat transfer into the system between Stations 1 and 2 (this value would be negative if heat were transferred from the system), Btu per lb

P = air pressure, lb per sq ft

ν = fluid specific volume, cu ft per lb

Z = elevation of air above a datum plane, ft

This is an energy statement since each component represents the energy of the air per unit of mass flowing (ft-lb per lb or simply ft). The various energy components would generally be referred to as follows:

W = work energy

$V^2 / 2g$ = kinetic energy

JU = internal energy

JQ = heat energy

$P\nu$ = flow work energy

Z = potential energy

Fortunately, when we use this equation for air systems, several of the terms can usually be eliminated, thus simplifying things a bit. If we assume that the air system does not receive or dissipate heat, then $Q_{1\text{-}2} = 0$. If the duct system is horizontal, then $Z_1 = Z_2$, and the steady flow energy equation for an air system can be re stated as follows :

$$W_{1\text{-}2} + (V_1^2 / 2g) + JU_1 + P_1\nu_1 + = (V_2^2 / 2g) + P_2\nu_2 \quad (7)$$

Three of the terms of this equation are quite important and deserve further discussion. The term $W_{1\text{-}2}$ relates to the work done by a fan in moving air through a system. It refers to the shaft energy imparted by the fan energy source (electric motor, steam turbine, etc.) to the air stream to provide the motivating energy for air movement. It is the *only* point within an air system (except at heating sources) when the air stream energy level increases. At all other locations, the net energy level is decreasing. The terms $P\nu$ and $V^2/2g$ are illustrated in *Figure 3A* since their use forms the basis of almost all measurements and analysis work done on HVAC air systems. $P\nu$ is called flow work but is referred to as static pressure (SP) or static head in air systems. $V^2/2g$ is called kinetic energy but is referred to as velocity pressure (VP) or velocity head in air systems. The numerical sum of these two energy components ($P\nu + V^2/2g$) is referred to as total pressure (TP) or total head. All of these components are normally referred to in units of inches water gauge as registered by the manometer arrangements shown in *Figure 3A*.

With these two terms now defined, Equation 7 can be applied to two common situations that require analysis in typical HVAC

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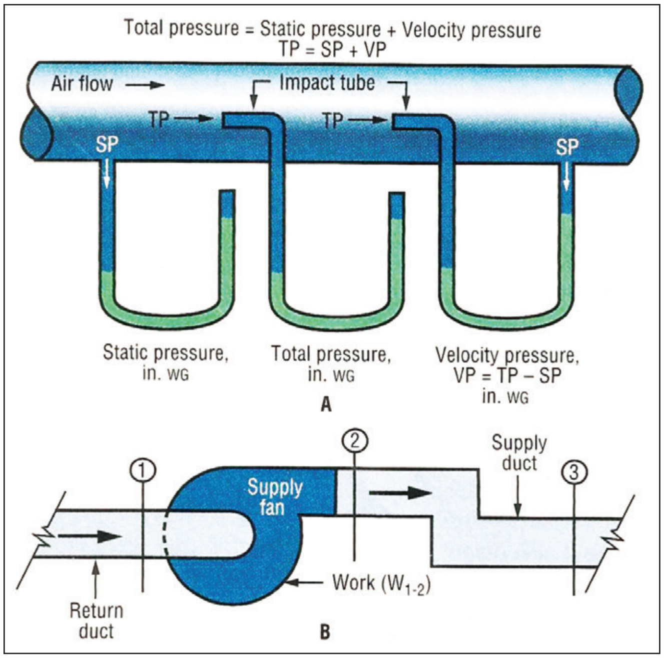

Figure 3: View A – illustration of Pv and V²/2g. Flow work energy, Pv, is referred to as static pressure (SP). The kinetic energy term V²/2g is referred to as velocity pressure (VP). View B – addition of a fan to an air duct system increases the flow work, kinetic energy, and internal energy of the air stream.

duct systems. *Figure 3B* shows a section of an air duct system containing a supply fan and a section of supply air ductwork. Equation 7 can be rewritten into a similar form to determine the outcome of the work performed by the fan:

$$W_{1\text{-}2} = [(V_2^2 - V_1^2) / 2g] + P_2 v_2 - P_1 v_1 + J(U_2 - U_1) \quad (8)$$

Analysis of this equation indicates that the work applied to the fan wheel creates in the air stream an increase in kinetic energy (or velocity pressure), an increase in flow work energy (or static pressure), and an increase in internal energy. Recognizing the thermodynamic property of enthalpy, we can write:

$$h = (Pv / J) + U \quad (9)$$

where

h = air enthalpy, Btu per lb

Substituting into Equation 8 yields the following form:

$$W_{1\text{-}2} = [(V_2^2 - V_1^2) / 2g] + [J(h_2 - h_1)] \quad (10)$$

Since the enthalpy rise for a perfect gas is given by the following:

$$\Delta h1 = C_p \Delta t \quad (11)$$

where

Δh = air stream enthalpy change, Btu per lb

C_p = specific heat capacity of air at constant pressure, Btu per lb - F

Δt = air stream temperature change, F

it follows that a large portion of the work put in at the fan shows up as a heat rise across the fan. A more rigorous treatment of this phenomenon indicates that all of the shaft work done by the fan appears as heat (otherwise known as fan heat rise or simply fan heat). Since the term in Equation 8 for the rise in internal energy is related to inefficiencies in the fan pressure increase process, the temperature rise for standard air can be approximated by the following:

$$\Delta t_{fan} = \Delta TP / 2.7\eta_f \quad (12)$$

where

Δt_{fan} = air temperature rise across fan, F

ΔTP = total pressure increase across fan, in. WG

η_f = fan efficiency, decimal

Another important concept identified by Equation 8 is that the fan imparts its energy rise to the air stream not only in the form of static pressure increase but also in the form of velocity pressure increase. Thus, the total pressure rise across the fan is the only truly meaningful method of assessing fan performance and power required. Our industry's fondness with static pressure and ratings based on fan static pressure cannot negate the principles of fluid mechanics on which Equation 8 is based.

Referring again to *Figure 3B*, consider a second application of Equation 7 as it relates to duct system losses. If we consider the run of ductwork between Stations 2 and 3, our experience tells us that for any real duct system, a pressure drop will occur in the direction of air flow between these two points. As we will discuss in a later section, these losses are generally characterized as frictional losses, which relate to fluid viscosity and the roughness of the confining duct walls, or to dynamic losses, which relate to duct stream turbulence or obstructions to straight-ahead flow. For the purpose of the current discussion, though, we will refer to these effects collectively as losses to see what insight the steady flow energy equation can yield into this phenomenon. If no work is done on the system (no fan is in this section of ductwork) and no heat is added to or dissipated from the air system, Equation 7 reduces to the following:

$$(V_2^2 / 2g) + P_2 v_2 + JU_2 = (V_3^2 / 2g) + P_3 v_3 + JU_3 \quad (13)$$

Substituting Equation 9 would yield the following:

$$(V_2^2 / 2g) + Jh_2 = (V_3^2 / 2g) + Jh_3 \quad (14)$$

Our first realization from the equation then relates to temperature conditions as the pressure is progressively lost. It can be shown[1] that the processes generating duct losses approximate a classical throttling process wherein the enthalpy remains constant. Referring to Equation 11 and recognizing that in an equal area duct the velocity would not change between Stations 2 and 3 (except for the slight variations in air density described in the section on the continuity equation), one can see that the temperature of the air stream does not change as the air passes through pressure drops. Only at the location of a fan (or a heating or cooling coil) does the temperature of the air stream change. Since the steady flow energy equation should be valid for flow with or without frictional and dynamic loss effects, our second and more important recognition from Equation 13 relates to duct air stream energy levels and the source of the losses. Rearranging Equation 13 yields the following:

$$(V_2^2 / 2g) + P_2 v_2 = (V_3^2 / 2g) + P_3 v_3 + J(U_3 - U_2) \quad (15)$$

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$J(U_3 - U_2)$ represents fluid friction and turbulence in the form of kinetic energy in eddies transformed into thermal energy. This heat addition occurs internally in an irreversible process as a result of frictional dissipation of mechanical energy into internal heat in the gas stream. (This internal heat addition counteracts the reduction in temperature that would have occurred in a reversible polytropic gas process conducted in a reversible manner for this reduction in pressure.[2]) Stated in words, Equation 15 reveals that the static pressure plus the velocity pressure at Station 2 are equal to the static pressure plus the velocity pressure at Station 3 plus the pressure loss that has occurred between these two stations. Using the relationship that the total pressure is equal to the static pressure plus the velocity pressure, the following equation is derived:

$$TP_2 = TP_3 + (TP_{loss})_{2-3} \quad (16)$$

Three very important observations are apparent from this equation:

- In any duct section without a fan, the total pressure is constantly dropping in the direction of air flow. Such a statement cannot necessarily be made with respect to static pressure or velocity pressure.
- The measure of the energy level in an air stream at any point is uniquely given by the total pressure only. Reference to static pressures or velocity pressures alone in this regard can be quite misleading.
- The losses in duct systems occurring due to frictional and dynamic effects must, based on the equation, be losses in total pressure. Any other evaluation of losses must be qualified by assumptions regarding the duct system geometry (such as that the area of the duct does not change).

Conservation of Momentum

Newton's first law of motion states that a body will maintain its state of rest or uniform motion (at constant velocity) along a straight line unless compelled by some unbalanced force to change that state. The momentum of a body, given by the product of its mass and its velocity, will thus tend to be conserved. Momentum is a vector quantity whose direction is that of its velocity so that a change of direction must be caused by an unbalanced force. Conservation of momentum concepts in fluid mechanics are usually used to calculate the dynamic forces exerted by moving fluids on fixed obstructions to the flow path.

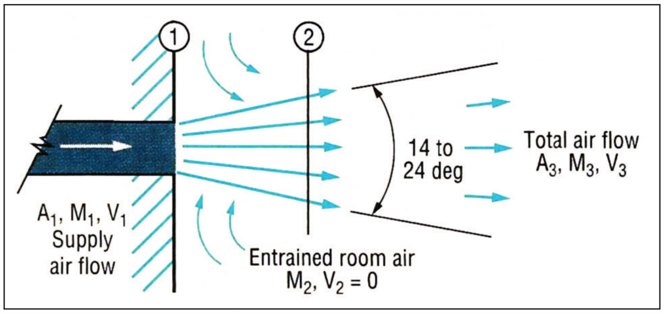

Figure 4: Discharge pattern of air leaving a rectangular sidewall grille in both the horizontal and vertical planes.

Though this is not a concern in most HVAC applications, the law of conservation of momentum does play into certain types of processes and dynamic loss effects.

Consider a rectangular sidewall supply air outlet, as shown in *Figure 4*, delivering air into a room at the same temperature as the room air (delivering air above or below room temperature introduces buoyancy effects that are superimposed on the momentum effects, thus complicating the analysis). Air emerging from the grille will entrain room air. This induction effect, according to the law of conservation of momentum, will decrease the velocity of the total air stream while increasing its volume. If M_1 and V_1 are the mass and velocity of the supply air, M_2 and V_2 are the mass and velocity of the entrained room air, and M_3 and V_3 are the mass and velocity of the total air mixture, then the law of conservation of momentum yields the following equation:

$$M_1V_1 + M_2V_2 = M_3V_3 \quad (17)$$

Since V_2, for all practical purposes, may be considered zero prior to the acceleration of the entrained air and since $M_3 = M_1 + M_2$, Equation 17 becomes:

$$M_1V_1 = (M_1 + M_2)\, V3 \quad (18)$$

The velocity of the mixture stream, V_3, has been reduced from V_1 since $M_1 + M_2$ must be greater than M_1. Furthermore, as the geometry of the grille changes, the induction ratio, $(M_1 + M_2) / M_1$, changes, entraining varying amounts of air (with long, narrow outlets having a significantly higher induction ratio than square outlets). The law of conservation of momentum, therefore, can predict the spread of the air stream as it exits the opening. To do so, since the density of the air exiting the opening is the same as the room air, one can substitute the volumetric flow rate for the mass flow rate, yielding the following:

$$Q_1V_1 = (Q_1 + Q_2)\, V_3 \quad (19)$$

With Q_1 being the volumetric flow rate of the exiting supply (primary) air and Q_2 being the volumetric flow rate of the entrained (secondary) air. Assume an opening of 1 sq ft (A_1) discharges 1000 cfm at 1000 fpm, and at Station 2 on *Figure 4*, an additional 1000 cfm of secondary air is entrained into the air stream. Solving Equation 19 indicates that the velocity at Station 2 is reduced to 500 fpm. Application of Equation 4 indicates that the flow area increased to 4 sq ft. This stream spread has occurred as a matter of course due to the basic principles of the law of conservation of momentum and the continuity equation. Further analysis would indicate that for an opening as shown in *Figure 4*, the spread of the air stream in both planes will open at between 14 and 24 deg, giving a spread in either direction of approximately 1 ft in every 5 to 8 ft of throw. It should be remembered that discharging air, whether into a room or into a plenum, will exhibit this same geometry unless the air stream is coerced to do otherwise.

Though the principle of conservation of momentum is not one of the areas most emphasized in air system technology, its importance to the design of duct fittings and duct configurations and their resulting dynamic losses is profound. As we will see later, failure to consider the law of conservation of momentum and

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the dynamic losses generated as a direct result of its effects has contributed to many problems in operating systems.

The four basic concepts of fluid mechanics discussed in the previous paragraphs form the groundwork for the technology of air systems as it is currently employed for HVAC work. The following sections will deal with a few selected application areas in which these principles become important, sometimes uniquely but more often as a combination of more than one of the concepts mentioned.

Flow Processes, Measurements

The concepts of static pressure (SP), velocity pressure (VP), and total pressure (TP) were developed and discussed earlier with regard to the law of conservation of energy and are depicted graphically in *Figure 3A*. Static pressure exerts itself in all directions in the duct, irrespective of whether the air is in motion or at rest. Positive static pressure indicates pressure attempting to burst the duct while negative static pressure attempts to collapse the duct. It is a measure of potential energy of the air stream since this pressure can create flow (kinetic energy) if it is released. Velocity pressure, on the other hand, exists only due to the velocity of a moving air stream and exerts itself only in the direction of air flow. It is a measure of the kinetic energy possessed by the air due to its velocity. Total pressure is the sum of these two energy types, representing in one statement both the sum of these two energy levels and the total energy possessed by the air stream at the point of measurement.

The properties of the energy equation and the fluid statics principle can be combined to determine a useful concept with respect to air measurement. The kinetic energy term, $V^2/2g$, from the energy equation represents velocity pressure in terms of ft-lb per lb of air flowing or simply head in terms of ft of air. Setting the kinetic energy term equal to the velocity pressure head and with appropriate conversions for ft of air to inches water gauge, one can derive the following equation:

$$VP = (V / 1096)^2 \rho_a \qquad (20)$$

where

VP = velocity pressure, in . WG

V = air velocity, fpm

ρ_a = density of air, lb per cu ft

For standard air (sea level pressure, 75 F, and a resulting density of 0.075 lb per cu ft), the equation reduces to:

$$VP = (V / 4005)^2 \qquad (21)$$

This is one of the simplest yet most powerful equations in all of air system technology since it permits the measurement of flow quantities in operating ductwork and therefore permits the field balancing of air systems. Without this principle, flow tests in operating air systems would be much more difficult. *Figure 3A* indicated a field test arrangement to measure velocity pressure utilizing the properties of total, static, and velocity pressures. In the early 1700s, a French physicist named Henri Pitot invented a device to simplify measurements of this type through a single duct insertion point. This device, shown in *Figure 5*, is known today as a pitot-static tube and is used almost universally for field velocity measurement of duct air flow. Devices of very similar configuration are utilized in flow hoods to measure the air delivered to rooms through diffusers. Measurement of the velocity pressure permits the direct calculation of the duct velocity, using Equation 20, if the air density is known. Use of the continuity equation (Equation 4) then permits the determination of the air flowing in the duct or out the flow hood if the flow area is known.

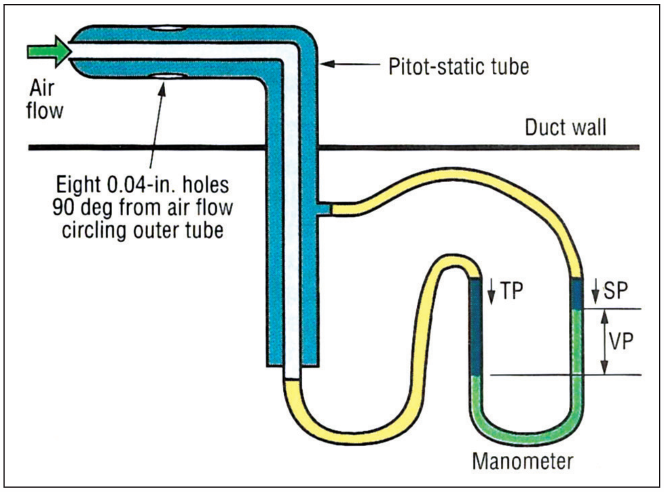

Figure 5: The pltot-statlc tube (a modified pitot tube) is almost universally used in the field to determine the total, static, and ve locity pressures of a fluid stream. When VP is known, the fluid velocity can be calculated.

Another concept relating fluid statics, the continuity equation, and the energy equation relates to the interchangeability of static and velocity pressures in air systems. Consider the duct section shown in *Figure 6*. Air flow is from left to right , moving from Station 1 to 2 in a duct that is assumed, for the purpose of this illustration, to be without losses. It therefore follows that if there are no losses, the total pressure (TP) relating to the overall air stream energy level would be unchanged between these two stations. This is indicated by the plot of total pressure shown below the duct section, with pressures given as positive values with respect to surrounding air pressure (ρ_a). At Station 1, which represents a relatively high velocity duct section, the total pressure TP_1 is composed of static pressure SP_1 and velocity pressure VP_1. Since the flow area at Station 1 is smaller than that at Station 2, the continuity equation (Equation 4) indictates that the velocity at Station 2 must decrease. Based on Equation 21, this dictates that the velocity pressure VP_2 must decrease compared to VP_1. However, assuming no losses occur between these two stations, the energy equation (Equation 13) dictates that the static pressure at Station 2 must increase. Notice, referring again to *Figure 6*, that a trade off has occurred between static pressure and velocity pressure based on straight forward application of basic fluid mechanics concepts and the geometry of the system with no actual losses occurring.

The phenomenon depicted here is called static pressure regain, and it is a very important principle of air system design. A similar principle explains the basis of operation of a centrifugal fan. One of the main purposes of a fan is not only to move air

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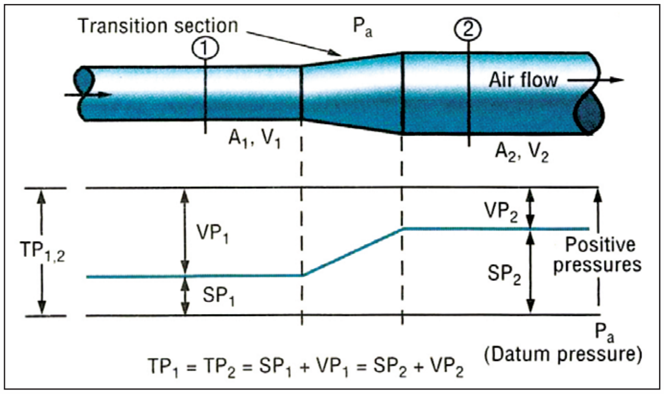

Figure 6: Interchangeability of static and velocity pressures in air systems. Graph below the duct shows the change in VP and SP as the fluid flows from the smaller duct through the transition section and into the larger duct.

but to increase the pressure of the air to overcome losses in the remainder of the system. As air passes through the fan, kinetic energy is imparted to the air stream and the velocity of the air is increased. As the air passes at high velocity from the relatively small blade passages to the connecting ductwork, the velocity is decreased, as at Station 2 in *Figure 6*, and the static pressure is increased. Application of this principle is vital to the operational performance of fans. Likewise, any ductwork geometry (such as discharge into an open plenum) that does not allow this regain to occur at the fan discharge in a controlled manner will degrade the capacity and performance of the fan.

Figure 6 also depicts the difficulty in field measurements using only static pressure. Connecting a simple manometer between Stations 1 and 2 and reversing the air flow (right to left from Station 2 to Station 1) would indicate that a large pressure drop had occur red between these two points. This is deceptive, however, since by definition for this example, no true energy losses have been considered. The static pressure drop measured between Stations 2 and 1 due to a velocity pressure increase would represent no real loss and no additional burden to be overcome by the fan. Thus, the energy forms of static pressure and velocity pressure are interchangeable, with the only truly significant term being their sum, the total pressure.

Losses and System Curves

Any real duct system operating with air flow will experience losses based on the term $J(U_3 - U_2)$ from the energy equation (Equation 15). These losses will be a function of the velocity of the air flow, the configuration of the duct system, and several other variables. For the purposes of analysis and calculations, these losses are neatly divided into two categories: frictional losses and dynamic losses. Though convenient for these purposes, the separation of these two types of losses becomes somewhat difficult in field testing, where frequently the effects are superimposed. For the purpose of this discussion, however, we will treat each of these effects separately.

Frictional losses occur in straight ductwork measured from centerline to centerline when fittings are involved. These losses occur due to the dissipation of mechanical energy from fluid viscosity and momentum interchange between particles moving at different velocities within the duct. Frictional losses can be calculated through the use of the Darcy-Weisbach equation, stated here for systems conveying air at standard conditions:

$$\Delta h_f = f(L / D)(V / 4005)^2 \qquad (22)$$

where

Δh_f = frictional loss of total pressure, in. WG

f = friction factor, dimensionless

L = duct length, ft

D = duct diameter , ft

V = air velocity, fpm

One important observation initially is that the equation predicts the loss in total pressure, not static pressure as is often the assumption. If, however, we are dealing with a sect ion of ductwork where the velocity remains constant and therefore the velocity pressure is constant, the change in total pressure will equal the change in static pressure. In this special case only will Equation 22 predict the static pressure drop due to frictional losses.

Another observation has to do with the relationships among the parameters affecting frictional losses. The equation clearly shows that the losses are directly proportional to the friction factor, the length, and the square of the velocity and are inversely proportional to the diameter of the duct. Furthermore, using the continuity equation (Equation 4), we see that if the area of the duct remains constant, the loss is directly proportional to the square of the volumetric flow rate or cfm. The last relationship is particularly important in the establishment of system curves, as we will see shortly.

If the friction factor from Equation 22 were a constant, the value of many of the duct sizing programs and slide rules would be greatly diminished. However, the friction factor actually turns out to be a complex function of air velocity and viscosity, duct size, and duct surface roughness. At low air velocities approaching laminar flow, the friction factor has actually been shown to be inversely proportional to the velocity, which when substituted in Equation 22 makes the frictional pressure drop vary directly as the velocity or cfm, not as the square. Fortunately, this situation occurs in air systems at relatively few locations, usually limited to cooling coils, heating coils, and filter sections.

Data in the *ASHRAE Handbook or Fundamentals*[3] provides data on frictional losses based on air velocity, air quantity, and round duct size. It should be noted that many of the duct sizing programs and manual duct sizing slide rules relating flow, duct size, and frictional loss per unit length in use today are based on research done prior to 1950.[4] This research was done on a limited number of duct sizes and was based on duct materials and joining techniques that are quite different from those used commercially today. Data based on this research appeared in copies of the *ASHRAE Handbook of Fundamentals* up to and including 1985. New research, completed in 1987,[5] conclusively showed that the frictional losses then in use were conservative, particularly in smaller ductwork at higher velocities. *Table 1* is a comparison of friction loss per unit length based on results from the 1985 *ASHRAE Handbook of Fundamentals* utilizing the earlier data versus the 1993 edition of the same volume utilizing the results of the most recent friction loss testing."

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Table 1: Frictional pressure drop, in wg per 100 ft.

Condition	1985 ASHRAE Handbook of Fundamentals*	1993 ASHRAE Handbook of Fundamentals*
10 in. round duct 1000 rpm	0.16	0.155 (–3.1%)
10 in. round duct 3000 rpm	1.30	1.15 (–11.5%)
20 in. round duct 1000 rpm	0.068	0.066 (–2.9%)
20 in. round duct 3000 rpm	0.54	0.51 (–5.6%)
40 in. round duct 1000 rpm	0.029	0.029 (0%)
40 in. round duct 3000 rpm	0.24	0.23 (–4.2%)

* See the chapter 'Duct Design'.

Conversion equations also appear in the *ASHRAE Handbook of Fundamentals* for the determination of equivalent rectangular and flat oval ducts from the round ductwork described in Equation 22. This information is fairly straightforward and will not be discussed here. However, the basis of this conversion can reveal insight into the efficiency of various duct shapes at conveying air. It has been determined that ductwork flowing air at the same mean velocity will have the same frictional pressure drop per unit length if the ducts have the same ratio of cross-sectional area to wetted perimeter. This function is called the hydraulic radius and is designated as R_h. *Figure 7* shows a comparison of round ductwork versus square ductwork on the basis of this parameter. Notice that with equal flow areas, the round duct diameter D is larger than the square duct side X. Therefore, the hydraulic radius of the round duct (D/4) is larger than the hydraulic radius of the square duct (X/4) if their flow areas are equal. The result is that with the same flow velocity and the same flow area for each duct, the round duct has a larger hydraulic radius and therefore a lower pressure drop. On the basis of this relationship, the round duct is the more efficient means of conveying air between two points. The equivalent square duct with the same pressure drop per unit length will have a larger flow area with a lower mean velocity than the round duct. As the aspect ratio of the rectangular duct increases, the mean velocity must be reduced even further to provide an equal pressure drop per unit length, making high aspect ratio ducts more expensive to construct and more prone to temperature losses or gains due to their higher perimeter.

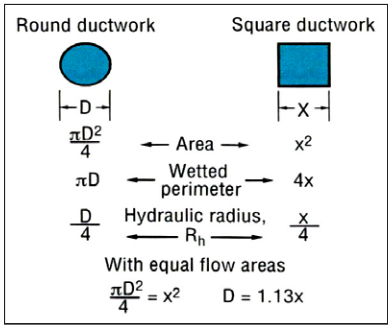

Figure 7: Comparison of a round and square duct on the basis of the hydraulic radius, R_h. The round duct is the more efficient means of conveying air.

The second type of duct system loss, dynamic loss, results from flow disturbances and turbulence caused by changes in flow direction, flow area, or obstructions to the path of the air flow. These losses, which are now considered the predominant losses in most air duct systems, occur due to the effects of the law of conservation of momentum and a related process termed flow separation. *Figure 8A* shows an elbow in a typical rectangular duct system. As indicated in the figure, as the air flow passes the heel of the elbow, the air has no intention of making the 90 deg left turn necessary to adhere to the duct wall. The momentum of the air stream causes the air to separate from the duct wall at this point. This separation effect causes a wake area beyond the elbow, with eddying flow causing drag and loss of energy. This energy loss appears as a reduction in air stream total pressure as given by the following equation for standard air conditions:

$$\Delta h_d = C_o (V / 4005)^2 \qquad (23)$$

where

Δh_d = dynamic loss of total pressure, in. WG

C_o = local loss coefficient, dimensionless

V = air velocity, fpm

The local loss coefficient, C_o, is cataloged in the *ASHRAE Handbook of Fundamentals*[3] and relates to the number of velocity heads lost in a given fitting configuration or flow geometry. The interesting thing about dynamic losses is that the momentum effects that contribute to these losses can cause far more damage than the simple pressure drop they create. Note how the air flow pattern is

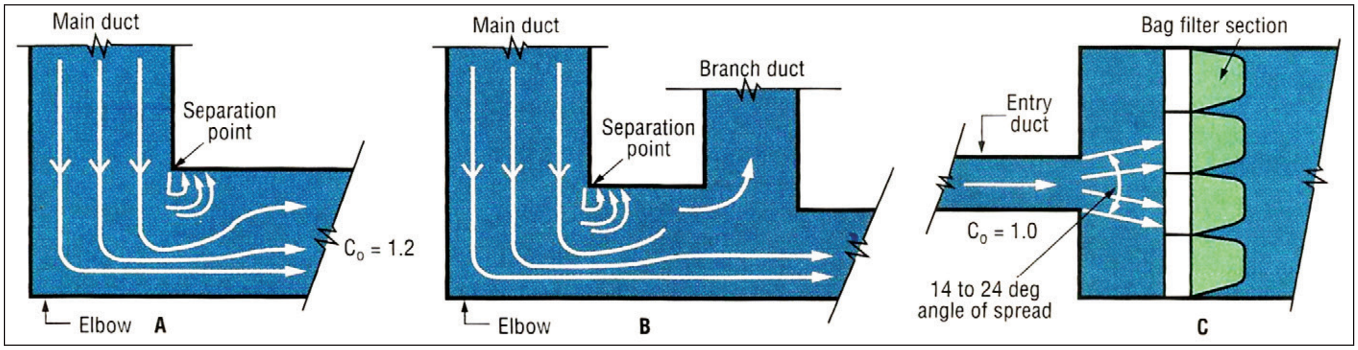

Figure 8: View A – loss coefficient, Co, relates to the number of velocity heads in a given fitting configuration or flow geometry. Note how the momentum effect bunches the air flow pattern toward the bottom of the duct in the elbow.The slight decrease in pressure at the top of the bend causes turbulence, resulting in a total pressure loss downstream. View B – flow pattern shown in View A has a serious effect on performance when a branch duct is placed too close to the elbow. Uniform ductflow has not yet recovered, and flow streamlines oppose entry into the branch duct. View C – lack of properly designed transition from a duct supplying a filter section causes uneven flow through filters (heavy in the center and lighter at the outer filters), resulting in uneven filter loading.

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bunched toward the bottom of the duct in *Figure 8A*. This situation is not a problem unless the branch duct of *Figure 8B* is added. Now the situation becomes very serious since the flow streamlines oppose entry into the branch duct shown. Such configurations can be found in numerous existing duct systems, and the poor performance of these branch ducts in terms of inadequate air flow is always the result. A similar situation can be seen in *Figure 8C*. Here, the pressure loss sustained from the free exit into the plenum is only part of the concern. The inability of the air stream to fill the plenum due to the momentum effects described earlier causes the succeeding filter section to load unevenly. Little air will flow across the outer filters, and the increased flow across the center filters will increase their expected pressure drop if the entire plenum was provided with a uniform entering air flow velocity. Failure to recognize such momentum effects increases dynamic losses and contributes to the types of problems shown here. By visualizing momentum effects during the design of a duct system, a designer can improve air separation geometries, thus reducing the local loss coefficient (C_o) for the fittings used, reducing pressure drops, and avoiding the types of problems shown in *Figure 8B* and *8C*.

Based on these discussions, the total pressure drop in a duct system would then be equal to the sum of the frictional and dynamic losses. The following equation can be derived by combining Equations 22 and 23:

$$\Delta h_t = \Delta h_f + \Delta h_d = f\,(L / D)\,(V / 4005)^2 + C_o\,(V / 4005)^2 \quad (24)$$

where

Δh_t = total pressure drop due to the combined effects of frictional and dynamic losses, in. WG

Combining terms, we can write this equation in the following form:

$$\Delta h_t = [f\,(L / D) + C_o]\,(V / 4005)^2 \quad (25)$$

Since for a fixed system the value of $f(L / D) + C_o$ would be composed of a set of constants, the total pressure drop would be proportional to the square of the velocity and (with the duct sizes set) to the square of the volumetric flow rate or cfm. In equation form, this relationship is stated as follows:

$$\Delta h_t = \text{constant x (cfm)}^2 \quad (26)$$

or

$$\Delta h_t \sim (\text{cfm})^2 \quad (27)$$

This equation is the basis of the system curve. It represents a plot of the pressures required to move air through the duct system analyzed at various flow rates. This parabolic relationship, shown as System Curve A in *Figure 9*, has become the basis of system analysis for air systems.

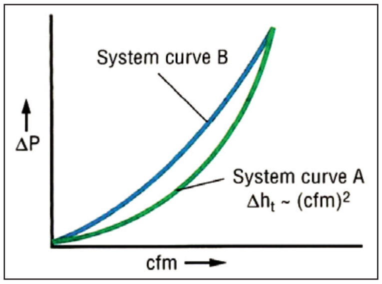

Figure 9: System curves are the basis for air system analysis. Curve A illustrates flow through a duct alone. Curve B shows the effects of overall system pressure drop, including ducts, coils, and filters.

Deviations from Equation 27 occur only where elements of the duct system, such as coils or filters, may exhibit near laminar flow characteristics, as discussed earlier. In this case, the pressure drop characteristics for these components should be separately evaluated using the following equation:

$$\Delta h_{c,f} = \text{constant x (cfm)}^N \quad (28)$$

where

$\Delta h_{c,f}$ = total pressure drop through coil or filter, in. WG

N = flow exponent (based on the mix of turbulent and laminar flow characteristics unique to that component)

To determine the flow exponent N for a given coil or filter, obtain two pressure drops, Δh_1 and Δh_2, from the manufacturer's rating information at two different flow rates, Q_1 and Q_2. The following equation can then be used to evaluate the exponent:

$$N = Log(\Delta h_1 / \Delta h_2) / Log\,(Q_1 / Q_2) \quad (29)$$

Flow exponents for typical coils range from 1.46 to 1.81 and for typical filters from 1.01 to 1.79.[6] System Curve B in *Figure 9* shows the effects of such components on the overall system pressure drop characteristics, including ductwork, coils, and filters. This type of curve would be derived by adding the system curve characteristics of the ductwork and fittings as given by Equation 26 to the component system curve characteristics of the coils and filters as given by Equation 28. This combined curve would represent the true pressure required to move air through an actual system composed not only of ductwork and fittings but also of heating coils, cooling coils, and filters. Air volume and system pressure drop are the same for both Curve A and B. The deviations created by the latter components may become important when performance changes are required in existing systems.[6]

Conclusion

This article has discussed only a few highlights of the basics of air systems related to the fluid mechanics of conveying air. The other aspects of the basics of air systems related to heat transfer, psychrometries, fans, and air distribution effects on human physiology are equally rich in theoretical background and deserve separate treatment.

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