
3 Fluid Flow

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CONTENTS

- 3.1 Introduction
- 3.2 Gas Properties
 - 3.2.1 Density
 - 3.2.1.1 Density at Actual and Standard Conditions
 - 3.2.1.2 Density of Mixtures
 - 3.2.2 Ratio of Specific Heat of Gas Mixtures
 - 3.2.3 Fuel Heating Value
- 3.3 Fluid Dynamic Concepts Commonly Used in the Burner Industry
 - 3.3.1 Laminar and Turbulent Flow
 - 3.3.2 Free Jet Entrainment
 - 3.3.3 Eduction Processes
 - 3.3.4 Pressure
 - 3.3.4.1 Definition and Units of Pressure
 - 3.3.4.2 Atmospheric Pressure
 - 3.3.4.3 Gage and Absolute Pressure
 - 3.3.4.4 Furnace Draft
 - 3.3.4.5 Static, Velocity, and Total Pressure
 - 3.3.4.6 Pressure Loss
- 3.4 Calculating the Heat Release from a Burner
 - 3.4.1 Definition of Heat Release
 - 3.4.2 Fuel Gas Orifice Calculations
 - 3.4.2.1 Discussion of Sonic and Subsonic Flow
 - 3.4.2.2 Equations for Calculating Fuel Flow Rate
 - 3.4.2.3 Discharge Coefficient
 - 3.4.3 Example Calculation: Heat Release from a Burner
- 3.5 Combustion Air Flow Rate through Natural-Draft Burners
- References

3.1 INTRODUCTION

Fluid dynamics is one of the most important areas of engineering science. Over the past 100 years, research into the science of fluid dynamics has grown at an exponential rate. Today, there are hundreds of papers published and dozens of conferences and symposia held every year devoted to the subject. Fluid dynamics is a broad subject because it is an important tool used in many engineering fields, for example, turbulence, acoustics, and aerodynamics. There are many good textbooks on the subject; those by Panton,¹ White,² Fox and McDonald,³ Vennard and Street,⁴ Hinze,⁵ Schlichting,⁶ and Hughes and Brighton⁷ are a few examples.

When designing combustion equipment, engineers might perform several fluid dynamic calculations. For example, it is common for engineers to size the fuel ports using orifice calculations or determine the size of the burner using pressure drop equations. A good familiarity with fluid dynamics is essential to the burner engineer. The purpose of this chapter is to give the reader a fundamental understanding of some of the fluid dynamic concepts that are important in combustion systems and show how they are commonly applied to burner design.

3.2 GAS PROPERTIES

The properties of a gas describe its physical characteristics and are commonly used in burner design calculations. Table 3.1 contains a list of various pure-component gases with their corresponding properties. This section presents a brief description of these properties and describes how the properties of a mixture are calculated. For a list of additional gas properties, the reader is referred to Geerssen,⁸ Baukal,⁹ Turns,¹⁰ and Bartok and Sarofim.¹¹

TABLE 3.1
Properties of Various Gases

Gas Composition	Molecular Weight	Density (lb/scf)	Ratio of Specific Heat*	LHV** (BTU/scf)	HHV*** (BTU/scf)
Methane (CH ₄)	16.043	0.04238	1.31	909.40	1010.00
Ethane (C ₂ H ₆)	30.070	0.07943	1.19	1618.70	1769.60
Propane (C ₃ H ₈)	44.097	0.11648	1.13	2314.90	2516.10
n-Butane (C ₄ H ₁₀)	58.123	0.15352	1.10	3010.80	3262.30
Pentane (C ₅ H ₁₂)	72.150	0.19057	1.08	3706.90	4008.90
n-Hexane (C ₆ H ₁₄)	86.177	0.22762	1.06	4403.80	4755.90
Cyclopentane (C ₅ H ₁₀)	70.134	0.18525	1.12	3512.10	3763.70
Cyclohexane (C ₆ H ₁₂)	84.161	0.22230	1.09	4179.70	4481.70
Ethylene (C ₂ H ₄)	28.054	0.07410	1.25	1498.50	1599.80
Propene (C ₃ H ₆)	42.081	0.11115	1.15	2181.80	2332.70
Butene (C ₄ H ₈)	56.108	0.14820	1.11	2878.70	3079.90
Pentene (C ₅ H ₁₀)	70.134	0.18525	1.08	3575.00	3826.50
Butadiene (C ₄ H ₆)	54.092	0.14288	1.12	2789.00	2939.90
Carbon Dioxide (CO ₂)	44.010	0.11625	1.29	0.00	0.00
Water (H ₂ O)	18.015	0.04758	1.33	0.00	0.00
Oxygen (O ₂)	31.999	0.08452	1.40	0.00	0.00
Nitrogen (N ₂)	28.013	0.07399	1.40	0.00	0.00
Sulfur Dioxide (SO ₂)	64.060	0.16920	1.27	0.00	0.00
Hydrogen Sulfide (H ₂ S)	34.080	0.09002	1.32	586.80	637.10
Carbon Monoxide (CO)	28.010	0.07398	1.40	320.50	320.50
Ammonia (NH ₃)	17.031	0.04498	1.31	359.00	434.40
Hydrogen (H ₂)	2.016	0.00532	1.41	273.80	324.20
Argon (Ar)	39.944	0.10551	1.67	0.00	0.00
Acetylene (C ₂ H ₂)	26.038	0.06878	1.24	1423.20	1473.50
Benzene (C ₆ H ₆)	78.144	0.20633	1.12	3590.90	3741.80

*At 59°F (standard temperature)

**LHV (Lower Heating Value) at 59°F and 14.696 psia

***HHV (Higher Heating Value) at 59°F and 14.696 psia

3.2.1 DENSITY

3.2.1.1 Density at Actual and Standard Conditions

Atmospheric air is a mixture of many gases plus water vapor and other pollutants. Aside from pollutants, which may vary considerably from place to place, the composition of dry air is relatively constant; the composition varies slightly with time, location, and altitude.¹² The *ASHRAE Handbook of Fundamentals*¹³ provides the following approximate composition of dry air on a volume fraction basis:

Nitrogen	0.78084
Oxygen	0.20948
Argon	0.00934
Carbon dioxide	0.00031
Neon, helium, methane, sulfur dioxide, hydrogen, etc.	0.00003

In 1949, a standard composition of dry air was fixed by the International Joint Committee on Psychrometric Data,¹⁴ as shown below.

Constituent	Molecular Weight	Volume Fraction
Oxygen	32.000	0.2095
Nitrogen	28.016	0.7809
Argon	39.944	0.0093
Carbon dioxide	44.010	0.0003
	28.965	1.0000

Density is defined as the mass per unit volume of a fluid and is usually given the Greek symbol ρ (rho). The units of density, for example, can be written as lbm/ft³ or kg/m³. Based on the composition of dry air given above, the density is 0.0765 lbm/ft³ or 1.225 kg/m³. This is the density of air at standard temperature, standard pressure, and is usually denoted as STSP (59.0°F or 15°C, 14.696 lbf/in.² or 101.325 kPa).

The density of a gas can change with temperature and pressure. For example, the density of dry air at 59.0°F and at 14.696 psia is 0.0765 lbm/ft³ STSP. If we compress the air and cool it down from standard conditions, its density will increase. We can calculate the density of a gas at any condition using the following equation:

$$\rho_{\text{actual}} = \rho_{\text{STSP}} \left(\frac{460 + 59}{460 + T(^{\circ}\text{F})} \right) \left(\frac{14.696 + P(\text{psig})}{14.696} \right) \quad (3.1)$$

Example 3.1

Determine the density of dry air at 5 psig and 40°F. Substituting the appropriate values into Equation 3.1 gives:

$$\rho_{\text{actual}} = 0.0765 \left(\frac{519}{460 + 40} \right) \left(\frac{14.696 + 5}{14.696} \right) = 0.1064 \text{ lbm/ft}^3 \text{ ATAP}$$

In this example, the density of the air has increased from 0.0765 at STSP to 0.1064 lbm/ft.³ This is the density at actual conditions and is usually denoted as ATAP (actual temperature, actual pressure). Notice from Equation 3.1 that as the temperature of the gas increases, the density decreases;

but as the pressure increases, the density increases. The density of any gas at ATAP conditions can be calculated using Equation 3.1 if one knows the density of that gas at STSP conditions.

If only the molecular weight of a gas is known, one can calculate the density of that gas at ATAP conditions. First, the density of the gas at STSP is calculated using the following equation:

$$\rho_{\text{gas,STSP}} = \rho_{\text{air,STSP}} \left(\frac{MW_{\text{gas}}}{MW_{\text{air}}} \right) = 0.0765 \frac{\text{lbm}}{\text{ft}^3} \left(\frac{MW_{\text{gas}}}{28.965} \right) \quad (3.2)$$

where MW_{gas} is the molecular weight of gas, MW_{air} is the molecular weight of air, and $\rho_{\text{gas,STSP}}$ is the density of the gas at STSP conditions. After the density of the gas has been calculated at STSP conditions, the density at actual conditions can be determined using Equation 3.1.

Example 3.2

Find the density of methane at 100°F and 30 psig.

Step 1: Using Equation 3.2, calculate the density of methane at STSP conditions.

$$\rho_{\text{methane,STSP}} = \rho_{\text{air}} \left(\frac{MW_{\text{methane}}}{MW_{\text{air}}} \right) = 0.0765 \frac{\text{lbm}}{\text{ft}^3} \left(\frac{16.043}{28.965} \right) = 0.0424 \frac{\text{lbm}}{\text{ft}^3} \text{ STSP}$$

Step 2: Using Equation 3.2, calculate the density of methane at ATAP conditions.

$$\rho_{\text{methane,ATAP}} = \rho_{\text{methane,STSP}} \left(\frac{460 + 59}{460 + T(^{\circ}\text{F})} \right) \left(\frac{14.696 + P(\text{psig})}{14.696} \right)$$

$$\rho_{\text{methane,ATAP}} = 0.0424 \left(\frac{519}{460 + 100} \right) \left(\frac{14.696 + 30}{14.696} \right) = 0.1195 \frac{\text{lbm}}{\text{ft}^3} \text{ ATAP}$$

Notice that the density is higher than at STSP conditions. The reason is because we are calculating the density at a pressure higher than standard conditions, thus resulting in an increase in density.

3.2.1.2 Density of Mixtures

The molecular weight and density of a gas mixture can be determined by the following equations, respectively:

$$MW_{\text{mixture}} = MW_1 x_1 + MW_2 x_2 + \cdots + MW_n x_n \quad (3.3)$$

$$\rho_{\text{mixture,STSP}} = \left(\frac{\rho_{\text{air,STSP}}}{MW_{\text{air}}} \right) [MW_1 x_1 + MW_2 x_2 + \cdots + MW_n x_n], \quad (3.4)$$

where MW_n and x_n is the molecular weight and volume fraction of the n th component, respectively.

Example 3.3

Determine the molecular weight and density of a gas mixture containing 25% hydrogen and 75% methane (volume basis).

We know the following variables:

$$MW_1 = 2.0159$$

$$MW_2 = 16.043$$

$$x_1 = 0.25$$

$$x_2 = 0.75$$

$$\rho_{\text{air,STSP}} = 0.0765 \text{ lbm/ft}^3$$

$$MW_{\text{air}} = 28.965$$

Substituting these values into Equations 3.3 and 3.4 gives:

$$MW_{\text{mixture}} = MW_1x_1 + MW_2x_2 = 2.0159 \times 0.25 + 16.043 \times 0.75 = 12.536$$

$$\rho_{\text{mixture,STSP}} = \left(\frac{\rho_{\text{air,STSP}}}{MW_{\text{air}}} \right) [MW_1x_1 + MW_2x_2 + \dots + MW_nx_n]$$

$$\rho_{\text{mixture,STSP}} = \left(\frac{0.0765}{28.965} \right) [2.0159 \times 0.25 + 16.043 \times 0.75] = 0.0331 \frac{\text{lbm}}{\text{ft}^3} \text{ STSP}$$

3.2.2 RATIO OF SPECIFIC HEAT OF GAS MIXTURES

The *ratio of specific heat*, denoted by the letter k , is a fluid property used in many engineering calculations and is mathematically defined as:

$$k = \frac{c_p}{c_v} \quad (3.5)$$

where c_p and c_v is the specific heat at constant pressure and volume, respectively. For most hydrocarbons, the ratio of specific heat varies between a value of nearly 1.0 to 1.7 and exhibits a strong dependence on the gas temperature and composition. For example, methane at standard temperature (59°F) has a k value of approximately 1.31, while propane has a value of 1.13. However, at 200°F, the ratios of specific heat of methane and propane decrease to a value of approximately 1.27 and 1.10, respectively. Values of k for several pure-component gases at standard temperature (59°F) are listed in Table 3.1.

One can think of the ratio of specific heat as being a value that relates to the compressibility of a gas. How much a gas compresses when a pressure is applied to it is an important property that influences the flow rate of a gas through an orifice. Later in this chapter we show how the ratio of specific heat is utilized in orifice calculations.

For a mixture of gases, at a given temperature, the ratio of specific heat can be determined with the following equation:

$$k_T = \frac{\frac{k_1x_1}{(k_1-1)} + \frac{k_2x_2}{(k_2-1)} + \dots + \frac{k_nx_n}{(k_n-1)}}{\frac{x_1}{(k_1-1)} + \frac{x_2}{(k_2-1)} + \dots + \frac{x_n}{(k_n-1)}} \quad (3.6)$$

where k_n is the ratio of specific heat, x_n is the volume fraction, and MW_n is the molecular weight of n th component.

Example 3.4

Determine the ratio of specific heat of the following fuel mixture (percentage by volume) at a gas temperature of 59°F:

- 75% Hydrogen
- 14% Methane
- 11% Propane

The volume fraction x of each gas component is:

$$x_1 = 0.75 \text{ (hydrogen)}$$

$$x_2 = 0.14 \text{ (methane)}$$

$$x_3 = 0.11 \text{ (propane)}$$

The molecular weight and ratio of specific heat of each gas component can be obtained from Table 3.1.

$$MW_1 = 2.0159 \text{ (hydrogen)}$$

$$MW_2 = 16.043 \text{ (methane)}$$

$$MW_3 = 44.097 \text{ (propane)}$$

$$k_1 = 1.41 \text{ (hydrogen)}$$

$$k_2 = 1.31 \text{ (methane)}$$

$$k_3 = 1.13 \text{ (propane)}$$

Substituting the above values into Equation 3.5 and solving gives the ratio of specific heat of the fuel mixture at 59°F.

$$k_T = \frac{\frac{1.41 \times 0.75}{(1.41 - 1)} + \frac{1.31 \times 0.14}{(1.31 - 1)} + \frac{1.13 \times 0.11}{(1.13 - 1)}}{\frac{0.75}{(1.41 - 1)} + \frac{0.14}{(1.31 - 1)} + \frac{0.11}{(1.13 - 1)}} = 1.32$$

As mentioned, the ratio of specific heat also varies with the temperature of the gas. Typically, as the gas temperature increases, the ratio of specific heat decreases as shown in Table 3.2 for several pure gas components. To determine the ratio of specific heat of a fuel mixture at various temperatures, one must determine the ratios of specific heat for each gas component at that given temperature. Using these ratios of specific heat values, Equation 3.5 can be used to determine the ratio of specific heat of the mixture at the given temperature.

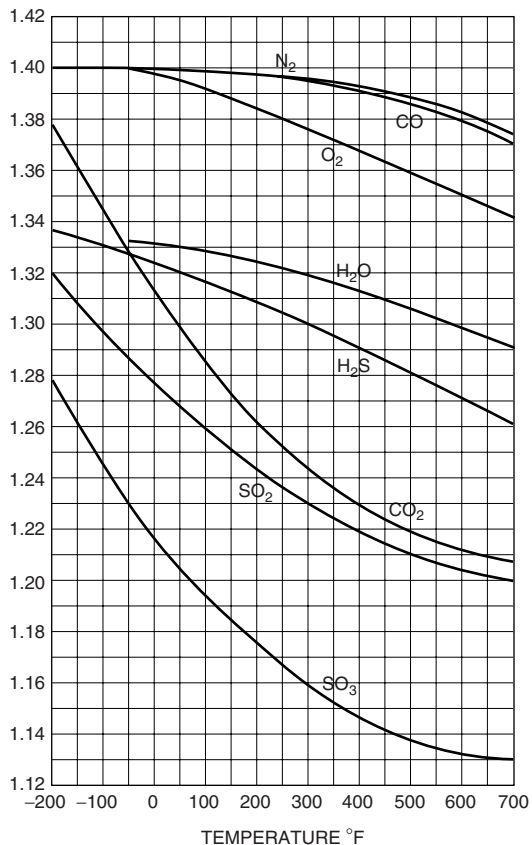
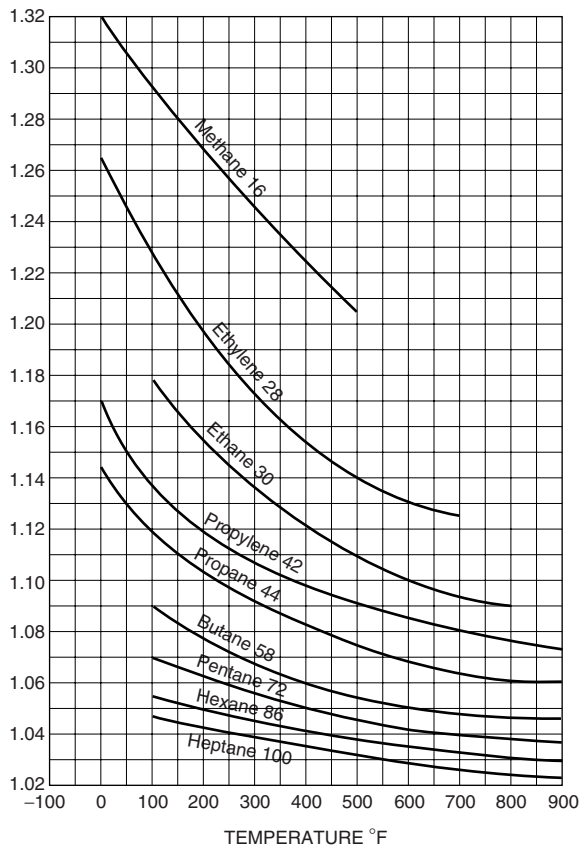
3.2.3 FUEL HEATING VALUE

The amount of energy released when a fuel is burned depends on its composition. For example, suppose we take 1 cubic foot each of hydrogen, methane, and propane. Which fuel will output the most energy when burned? The answer is propane. Propane has a heating value of 2590 BTU per cubic foot while methane and hydrogen have 1013 and 325 BTU per standard cubic foot, respectively. So, one can see that the heating value of a fuel depends on the components in the fuel blend. The heating values just mentioned are based on what is referred to as the *Higher Heating Value* (HHV) of the fuel. Fuels can also be related to their *Lower Heating Value* (LHV).

The HHV of a fuel is based on the energy released assuming that the water vapor in the combustion products condenses to liquid. When the water vapor condenses to liquid form, additional energy is released. This energy is referred to as the heat of condensation or latent heat of vaporization.

TABLE 3.2

Plots of Ratio of Specific Heat (k) for Various Pure-Component Gas at Different Temperatures



The HHV represents the total heat obtained by first burning a fuel and then cooling the products to standard temperature. If the water vapor in the products is not condensed, the total amount of energy released from the fuel is less than the HHV and is referred to as the LHV. So, for example, the LHV and HHV of propane are 2385 and 2590 BTU per standard cubic foot, respectively.

Heating values are also provided on a mass basis. For example, the LHV of propane, methane, and hydrogen are 19,944, 21,520, and 51,623 BTU per pound, respectively. Notice that hydrogen, on a pound basis, has a much higher energy output than propane and methane; however, on a volume basis, the energy output is much lower due to its low density. Typical hydrocarbons, such as methane, ethane, butane, etc., have an LHV of approximately 20,000 BTU per pound.

To determine the heating value of a mixture of gases, it is necessary to know the heating value of each compound and the composition of the gases in terms of volume or mass fractions. The following equations are used to calculate heating value of a gas mixture:

$$HV_{\text{mixture,vol}} = HV_1x_1 + HV_2x_2 + \cdots + HV_nx_n \quad (3.7)$$

$$HV_{\text{mixture,mass}} = HV_1y_1 + HV_2y_2 + \cdots + HV_ny_n \quad (3.8)$$

where $HV_{\text{mixture,vol}}$ = heating value of mixture, volume basis; HV_n = heating value of n th component, volume basis; x_n = volume fraction of n th component; $HV_{\text{mixture,mass}}$ = heating value of mixture, mass basis; HV_n = heating value of n th component, mass basis; and y_n = mass fraction of n th component.

Example 3.5

Determine the LHV of the following mixture:

Mixture Content, by Volume	LHV (BTU/scf)
40% hydrogen	275
55% methane	913
5% propane	2385

Substituting the appropriate values into Equation 3.7 gives:

$$LHV_{\text{mixture,vol}} = LHV_1x_1 + LHV_2x_2 + LHV_3x_3$$

$$LHV_{\text{mixture,vol}} = (0.4)(275) + (0.55)(913) + (0.05)(2385) = 731.4 \text{ BTU/scf}$$

3.3 FLUID DYNAMIC CONCEPTS COMMONLY USED IN THE BURNER INDUSTRY

3.3.1 LAMINAR AND TURBULENT FLOW

If you have ever watched the smoke trailing from a cigarette or incense burner in a calm room, you probably noticed that the flow of the smoke is initially smooth and then quickly transitions into a chaotic motion. The smooth part of the flow is called *laminar flow*; that is, the flow of the smoke is laminated. The chaotic or random motion is called *turbulent flow*. Turbulent flow is very easy to find in nature; flow from a volcano, wind, flow of rivers, exhaust from a jet engine are a few examples. Laminar flow, however, is more difficult to find in nature.

Flames are typically classified as either *laminar* or *turbulent*. [Figure 3.1](#) contains photographs of a burning match and a forest fire. One can easily recognize the forest fire as being a turbulent flame and the burning match as a laminar flame. Notice that the flame front on the burning match



FIGURE 3.1 Photographs showing examples of turbulent (left) and laminar (right) flames.

is smooth while the forest fire is chaotic. For detailed discussions of laminar and turbulent flames, the reader is referred to Kuo¹⁵ and Lipatnikov and Chomiak.¹⁶

In the burner industry, one will see both types of flames. Sometimes, a burner engineer will design a burner that produces a flame that is laminar in order to prevent a phenomenon called flashback — an event that occurs in premix burners when the flame propagates back through the burner tip. Sometimes, burners are designed to create turbulence to enhance mixing of the fuel with combustion air and furnace flue gases. The rate at which these gases are mixed is critical to the design of a burner because it can affect the flame shape and size.

A common method used to determine if the flow of a fluid is laminar or turbulent is to relate several of the flow parameters to a nondimensional index called the Reynolds number. Mathematically, the Reynolds number (Re) is defined as:

$$\text{Re} = \frac{VD}{\nu} \quad (3.9)$$

where V is the velocity of the flowing fluid, D is a length variable in the flow field (e.g., pipe diameter), and ν is the kinematic viscosity. The flow of a fluid in a pipe is turbulent, for example, if the Reynolds number is greater than 4000; but if the Reynolds number is less than 2300, the flow is laminar. The Reynolds number is used in many fields of fluid dynamics. In the burner industry, the Reynolds number is used, for example, to scale orifice discharge coefficients, relate the pressure drop of a fluid flowing through an obstruction, and calculate heat transfer rates.

3.3.2 FREE JET ENTRAINMENT

When a fluid emerges from a nozzle, it will interact with the surrounding fluid. This is called a *free jet* and is common in combustion systems; for example, fuel gas exiting a burner nozzle. Figure 3.2a is a photograph showing a gas exiting a nozzle. The white streak-lines in the photograph are small particles that have been released in the vicinity of the jet. The photograph clearly shows that the particles are pulled toward the flow path of the free jet. These particles are pulled into the jet because, as the gas exits the nozzle, a low-pressure region is created downstream of the exit. This low-pressure region causes the surrounding gas to be pulled in. As the surrounding gas is pulled in, turbulence causes the free jet and surrounding gas to mix as shown in Figure 3.2b. As the free jet captures the surrounding gas, the jet diameter increases.

The rate at which a free jet entrains and mixes with the surrounding gas is a critical parameter when designing burners. For example, sometimes an engineer designs a burner to mix large quantities of furnace flue gas with the fuel prior to combustion in order to reduce the flame

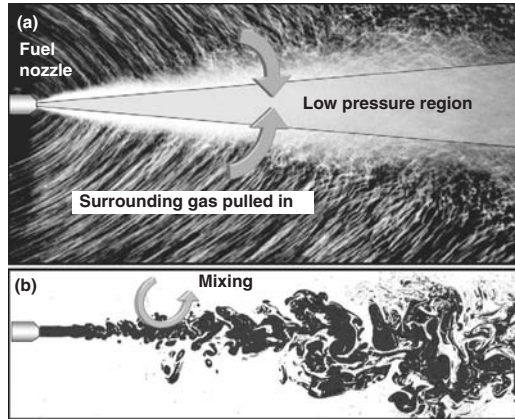


FIGURE 3.2 Photographs showing the mixing and entrainment of a free jet.

temperature and lower NO_x emissions. However, when designing a burner for low BTU gases that are difficult to burn, the engineer may want to delay the mixing of the fuel with the furnace flue gases and air in order to provide a stable flame.

The engineer controls the rate of mixing of fuel with combustion air and furnace flue gases by designing the fuel nozzle to utilize a certain amount of pressure. For example, suppose we have two nozzles passing the same mass flow-rate of fuel (same heat release); one of the nozzles is 0.0625 in. in diameter and the other nozzle is 0.25 in. in diameter. Obviously, the smaller diameter nozzle must operate at a higher pressure than the larger diameter nozzle in order to pass the same amount of fuel. If we measure the amount of surrounding gas entrained by each nozzle at any given location downstream of the nozzle exit, which one would have more surrounding gas mixed with the fuel? The smaller diameter nozzle would have more surrounding gas mixed with the fuel because it exits the nozzle with more energy. Because more work is required to compress the fuel to a higher pressure for the smaller nozzle than for the larger nozzle, one should expect more work from the fuel as it exits the nozzle; this additional energy results in better mixing.

The amount of surrounding gas entrained by a turbulent jet can be estimated¹⁷ from the following equation:

$$\frac{m_{\infty}}{m_f} = 0.32 \left(\frac{\rho_{\infty}}{\rho_f} \right)^{1/2} \left(\frac{x}{d} \right) \quad (3.10)$$

where m_{∞} is the mass of the surrounding gas, m_f is the mass of the fuel, ρ_{∞} is the density of the surrounding gas, ρ_f is the density of the fuel, x is the distance downstream of the nozzle exit, and d is the diameter of the port. It should be noted that this equation is valid for $x/d > 18$.

Example 3.6

Suppose two nozzles are discharging methane into ambient air. The temperature of both the methane and the air is 60°F. One nozzle has a port diameter equal to 0.0625 in. and the other nozzle has a port diameter equal to 0.25 in. Determine the mass ratio of air-to-fuel for each nozzle at a location 8 in. downstream of the exit.

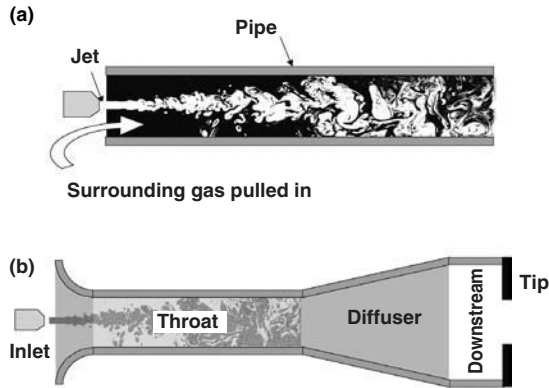


FIGURE 3.3 Eductor systems: (a) simple version, and (b) more complex version.

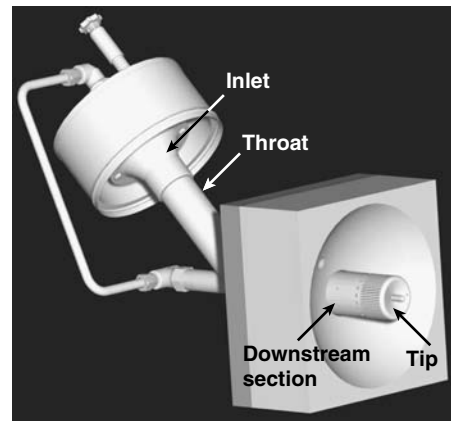


FIGURE 3.4 Radiant wall, eductor-style burner.

First, notice that the value of x/d is greater than 18 for both nozzles; therefore, Equation 3.10 is valid. The values of each variable in this example are: $\rho_{\infty} = 0.0765 \text{ lbm/ft}^3$, $\rho_f = 0.0422 \text{ lbm/ft}^3$, $x = 8 \text{ in.}$, $d = 0.0625 \text{ in.}$ and 0.25 in. Substituting these values into Equation 3.10, the mass ratio of air to fuel for the 0.0625-in. and 0.25-in. diameter nozzles are 55.1 and 13.8, respectively. Notice that the smaller port has entrained four times more ambient air than the larger port. Again, this illustrates that burner engineers can control the rate of mixing of fuel with the surrounding gas by drilling fuel nozzles to a particular diameter.

3.3.3 EDUCTION PROCESSES

If a pipe is placed just downstream of a free jet as illustrated in Figure 3.3, a low-pressure zone will be created inside the pipe, causing the surrounding gas to be pulled in through the inlet. This system is sometimes referred to as an *eductor* or *jet pump* and is commonly used throughout the combustion industry. These systems consist of two main parts: a primary nozzle and a pipe commonly referred to as a mixer or venturi. In the burner industry, it is common to see more complex venturi designs as illustrated in Figure 3.4. These more complex designs usually consist of five components: inlet, throat, diffuser, downstream section, and tip. Each component plays a major role in the entrainment performance of the eductor system.

The inlet typically consists of a well-rounded bell design. The purpose of a bell inlet is to reduce the pressure losses as the secondary gas enters into the eductor system. By reducing pressure

losses, the engineer can improve the entrainment performance; sometimes this is critical, especially when designing premixed burners. Located just downstream of the venturi inlet is the throat.

The design of the venturi throat is critical to the entrainment performance of an eductor system. Both the diameter of the throat and the ratio of the length to the diameter are the critical design parameters. Eductor systems can be designed with a specific throat diameter that will provide maximum entrainment performance. The throat diameter that provides the maximum entrainment performance depends on the momentum of the motive gas and the overall momentum losses as the gas flows through the eductor system. The length-to-diameter ratio (“L over D” ratio) is also a critical design parameter and has an optimum value that provides maximum entrainment performance. If the “L over D” ratio is too large, additional momentum will be consumed by friction losses as the gas flows along the throat wall. If the “L over D” ratio is too small, the motive gas jet will not impinge the wall of the venturi throat, thus resulting in a reduction in entrainment performance. Values of “L over D” ratios that provide optimized entrainment performance typically vary from 5 to 7. Located just downstream of the venturi throat is the diffuser section.

The diffuser is conical in shape and provides a transition from the venturi throat to the downstream section. Typically, diffusers are designed with small transition angles to provide smooth flow in order to reduce the pressure losses as gas flows from the throat to the downstream section.

The design of the downstream section can be as simple as a straight pipe or quite complex, consisting of a variety of fittings. Usually, in the burner industry the outlet of the downstream section consists of a tip. There are a variety of tips used in the burner industry, depending on the design application. The pressure loss associated with the flowing gas through the downstream section and tip can have a major influence on the design and performance of an eductor system.

Due to the large number of variables involved, it can be challenging for burner engineers to properly predict and optimize the performance of eductor systems. Burner designers usually rely on experiments or computer models to determine eductor performance. Some of the first theoretical and experimental studies on entrainment performance of an eductor system began in the early 1940s.¹⁸ Since that time, a lot of work has been devoted to understanding the mechanisms governing the performance of these systems.¹⁹ The trends shown in Figure 3.5 are a typical representation of how the pressure of the primary jet influences the entrainment performance of an eductor system. It is convenient for engineers to analyze these trends plotted with the ratio of

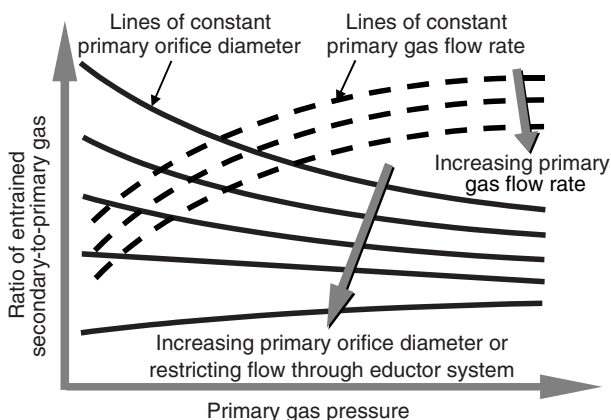


FIGURE 3.5 A typical representation of how the motive energy of the primary fuel jet affects the entrainment performance of an eductor system.

entrained secondary-to-primary gas. These general trends can provide valuable insight to engineers who design eductor systems and can be summarized as follows:

1. At a constant primary gas pressure, increasing the orifice diameter results in a decrease in the entrainment performance.
2. At a constant primary gas flow rate, increasing the primary gas pressure increases the entrainment performance.

3.3.4 PRESSURE

3.3.4.1 Definition and Units of Pressure

Pressure is created by the collision of molecules on a surface and is defined as the force exerted per unit area on that surface. Pressure is generally measured in units of Pascals ($\text{Pa} = \text{N/m}^2$); however, burner system designers will use a variety of pressure units because the choice of units will vary, depending on the customer. For example, customers in the United States will typically use units of psi (pounds per square inch) or inches of water column (inches of WC), whereas in Europe and Asia the units are usually Pa or millimeters of water column (mm of WC). The conversion of these pressure units are as follows: $1 \text{ lb/in.}^2 = 27.68 \text{ inches of water column} = 6895 \text{ Pa} = 703.072 \text{ mm of water column}$.

3.3.4.2 Atmospheric Pressure

Atmospheric pressure refers to the pressure created by a column of air that extends from the surface of the Earth to several miles above it. Initially, engineers developed a *Standard Atmospheric Pressure* so that the performance of aircraft and missiles could be evaluated at a standard condition. The idea of a standard atmospheric pressure was first introduced in the 1920s.²⁰ In 1976, a revised report was published that defined the U.S. standard atmosphere that is the currently accepted standard. This standard is an idealized representation of the mean conditions of the Earth's atmosphere in one year. The standard atmospheric pressure at sea level is equal to 14.696 psi (407 inches of WC or 101,325 N/m^2).

The atmospheric pressure varies with elevation. As one moves away from the surface of the Earth, the atmospheric pressure decreases because there is less atmosphere overhead to create pressure. For example, in Denver, Colorado, the elevation is about 1 mile above sea level. The atmospheric pressure at this elevation is approximately 12.1 psi. The atmospheric pressure in the troposphere, defined as the layer between sea level and 10,769 m, can be estimated using the following equation:²¹

$$p = 101,325 \left[\frac{T_o - 6.5 \times 10^{-3}(H)}{T_o} \right]^{5.259} \quad (3.11)$$

where p is the pressure in N/m^2 , T_o is the temperature at sea level in Kelvin, and H is the height above sea level in meters.

Example 3.7

If the temperature at sea level is 60°F, what is the atmospheric pressure at an elevation 1 mile above sea level?

First, let's determine the temperature at sea level in units of Kelvin (K).

$$T_o(\text{K}) = [T_o(^{\circ}\text{F}) - 32]/1.8 + 273.15 = [60 - 32]/1.8 + 273.15 = 288.71 \text{ K}$$

Next, let's convert the height above sea level (H) to units of meters.

$$H(\text{m}) = 1 \text{ mile} = 5280 \text{ ft} \times 1 \text{ m}/3.281 \text{ ft} = 1609.27 \text{ m}$$

Substituting these values into Equation 3.11 gives $p = 83,450.77 \text{ N/m}^2 = 12.10 \text{ psi}$.

3.3.4.3 Gage and Absolute Pressure

If the Earth was in a perfect vacuum, there would be no column of air above the surface; hence, the atmospheric pressure would be zero. The *absolute pressure* is measured relative to a perfect vacuum. Therefore, when a pressure measurement is taken at the surface of the Earth, the absolute pressure is equal to the atmospheric pressure. When writing the units for pressure, it is customary to designate absolute pressure with the letter “a” or “abs” after the units; for example, psia, psi (abs), or kPa (abs). The absolute pressure can never be less than zero; however, the gage pressure can.

The *gage pressure* is always measured relative to the atmospheric pressure. A gage pressure of less than zero can exist. For example, suppose there is a sealed container that holds a vacuum at 10 psia at sea level. The gage pressure, which is measured relative to the absolute pressure, would be $10 \text{ psia} - 14.7 \text{ psia} = -4.7 \text{ psig}$. The letter “g” after the pressure units represents gage pressure. Now suppose the container is pressurized to 20 psia. The gage pressure will then be $20 \text{ psia} - 14.7 \text{ psia} = 5.3 \text{ psig}$. Thus, the gage pressure can either be a positive or negative number and is just the difference in pressure between the atmospheric pressure and the pressure at interest.

3.3.4.4 Furnace Draft

The term “draft” is commonly used to describe the pressure inside a heater.²² Draft is defined as the pressure difference between the atmosphere and the interior of the heater *at a particular elevation*. Because both the atmospheric pressure and the pressure inside the heater vary with elevation, it is important to make these pressure measurements at the same elevation.

Draft is established in a heater because the hot flue gases in the stack have a lower density than the surrounding air. The difference in density of the flue gas and air creates a buoyant force, causing the flue gases to flow vertically upward as illustrated in Figure 3.6; this effect is similar to why a hot-air balloon rises. As this column of hot flue gases rises from the furnace, it creates a negative pressure inside the furnace, drawing combustion air in through the burners. In the burner industry, furnace draft can be as low as 0.0036 psi. This low pressure is usually measured using a sensitive pressure transmitter or an inclined manometer.

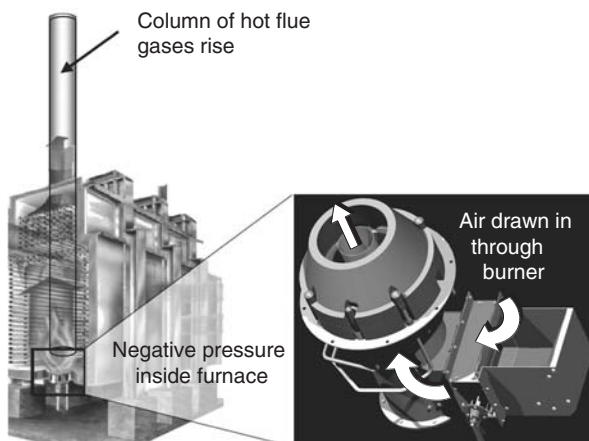


FIGURE 3.6 Description of furnace draft.

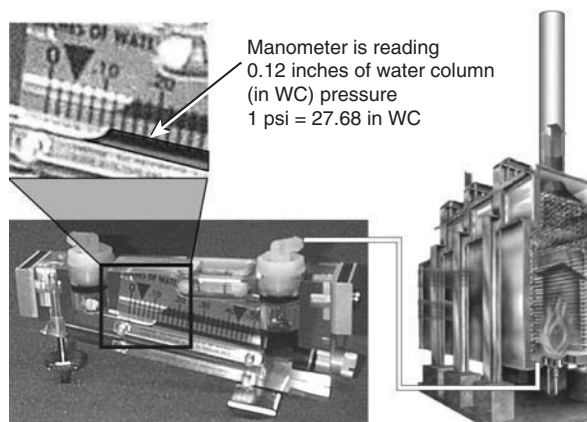


FIGURE 3.7 Inclined manometer.

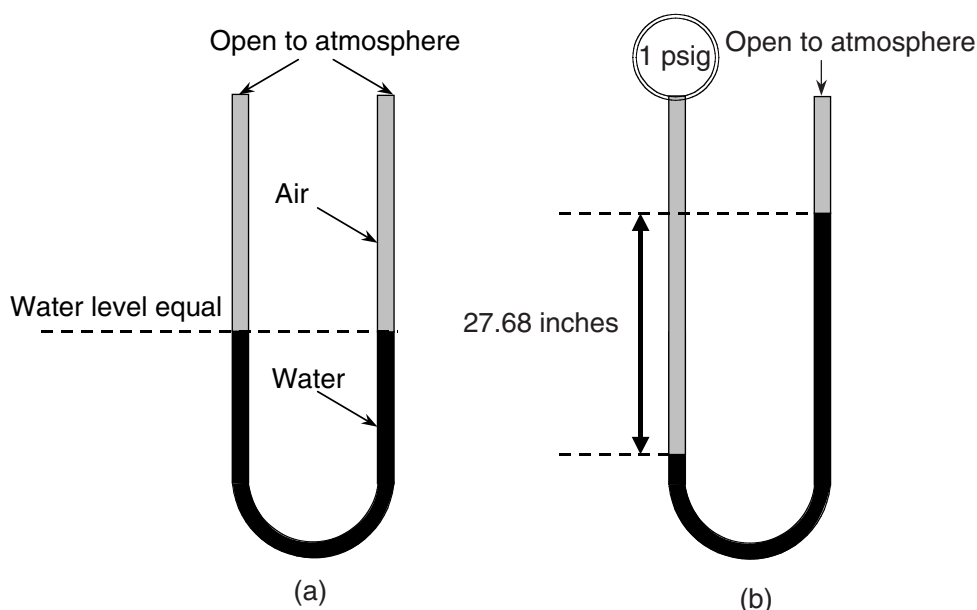


FIGURE 3.8 U-tube manometer illustrating the principle of water column pressure.

An inclined manometer consists of a clear tube positioned at a slight angle relative to the horizontal as shown in [Figure 3.7](#). The tube is typically filled with a light, red oil with a specific gravity of 0.8. One end of the tube is connected to the furnace while the other end is left open to the atmosphere. The negative pressure inside the heater pulls the oil down the tube. A marked gage located parallel to the tube is calibrated to provide the pressure in a particular unit.

The unit of draft typically used in the burner industry is expressed in inches or millimeters of water column (in. WC or mm WC). The unit of pressure, *inches of WC*, can be described with a simple example. Suppose a tube, shaped in the form of a U, is partially filled with water, as illustrated in [Figure 3.8a](#). If both ends of the tube are open to the atmosphere, the water will equalize to the same level on each side of the tube. If a pressure of 1 psi is applied to one end of the tube, the difference in water level will now be 27.68 in., as illustrated in [Figure 3.8b](#). That is, a pressure

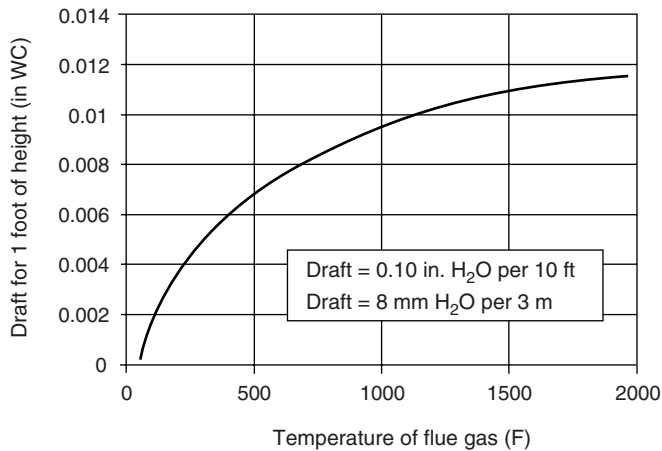


FIGURE 3.9 Plot of flue gas temperature versus draft per 1 foot of height.

of 1 psi will create a pressure head equal to 27.68 in. of water column. Typically in the burner industry, the furnace draft varies between 0.1 and 1 in. of water column. The draft can be estimated using the following equation:

$$P = \frac{27.68 \times H \times (\rho_{\text{flue}} - \rho_{\text{ambient}})}{144} \quad (3.12)$$

where P represents the furnace draft in inches of WC, H is the stack height in units of feet, and ρ_{flue} and ρ_{ambient} are the density of the flue gas and ambient air in lbm/ft^3 , respectively.

Example 3.8

A heater located at sea level has a stack that is 160 ft tall with an average flue gas temperature of 800°F (assuming a molecular weight of 28). If the ambient temperature is 60°F, estimate the draft that can be achieved by the stack at these conditions.

Using Equations 3.1 and 3.2, the density of the flue gas and ambient air can be calculated: $\rho_{\text{flue}} = 0.0305 \text{ lbm/ft}^3$ and $\rho_{\text{ambient}} = 0.0765 \text{ lbm/ft}^3$. Substituting these values into Equation 3.12 gives a draft level equal to 1.41 inches of WC. As a quick reference for determining the draft, refer to [Figure 3.9](#).

As mentioned, the draft does not remain constant throughout the furnace. For example, [Figure 3.10](#) is an illustration showing the draft in a heater at various elevations. First, notice that the atmospheric pressure represented by the dashed line decreases with elevation as discussed earlier. If this furnace were located at sea level, the atmospheric pressure would be 14.696 psi or 407 inches of WC. At the top of the stack, an elevation of 160 ft above sea level, the atmospheric pressure is approximately 404.7 inches of WC (calculated using Equation 3.11 at 60°F). The difference in the atmospheric pressure at grade and at an elevation of 160 ft is the maximum draft that can be achieved for this particular heater. That is, 2.3 inches WC (407 – 404.7) is the maximum achievable draft. This would be the draft if the density of the gas inside the heater was equal to zero and there were no pressure losses as the flue gases flow through the heater and stack. In reality, however, the density of the flue gases can never equal zero and there are always pressure losses associated with the flow of the flue gases through the furnace. Notice in [Figure 3.10](#) that the pressure inside the furnace continually varies with elevation. Again, the draft is defined as the pressure difference between the atmosphere and the interior of the heater *at a particular elevation*. In this particular example, the draft available for the burners is approximately 0.8 inches of WC

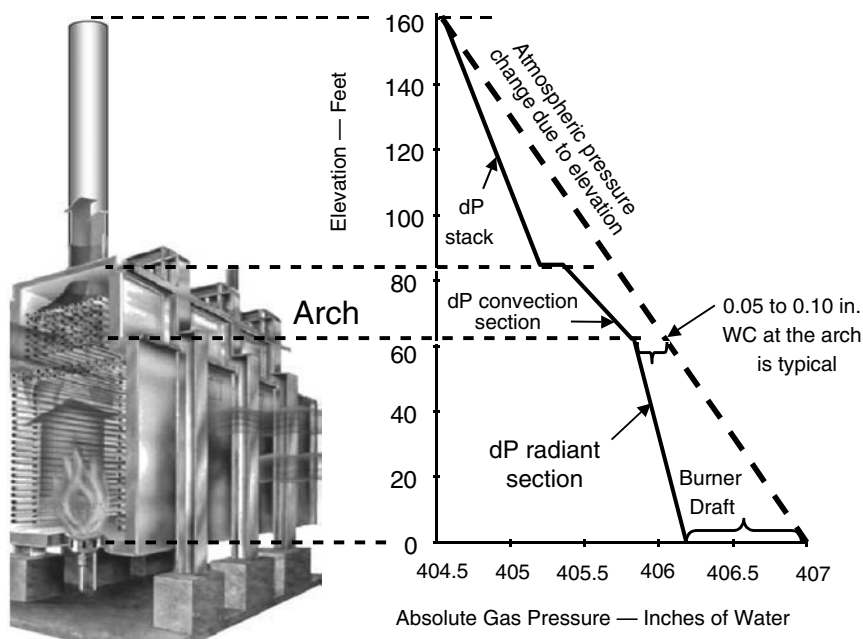


FIGURE 3.10 Furnace draft at various elevations.

(406.2 to 407). Also notice that minimum draft occurs just after the radiant section (arch). In most industrial furnaces, this is typically the location of minimum draft and is the most important location to measure and control draft (also called target draft). Typically, draft levels at this location are maintained at a pressure of 0.05 to 0.1 inches of WC. Maintaining a slight negative pressure at this location normally ensures a negative pressure throughout the heater, which is desirable.

3.3.4.5 Static, Velocity, and Total Pressure

The *total pressure* is defined as the velocity pressure plus the static pressure:

$$P_T = P_V + P_S \quad (3.13)$$

where P_T is the total pressure, P_V is the velocity pressure, and P_S is the static pressure. The *velocity pressure* is defined as the pressure created by a flowing gas impacting a surface. The *static pressure* is defined as the pressure created by gas molecules impacting a surface at a location where the velocity of the gas is equal to zero. For example, suppose a sealed vessel contains a gas at a pressure of 1 psig as illustrated in Figure 3.11. What is the static pressure inside the vessel? We know that the total pressure inside the vessel is equal to 1 psig. Because the velocity of the gas inside the vessel is zero, we can say that the pressure created by a flowing gas, or the velocity pressure, inside the vessel is equal to zero. Therefore, from Equation 3.13 we can conclude that the static pressure inside the vessel is equal to the total pressure of 1 psig.

Now suppose that we open a valve to a pipe located on the vessel. If we assume that there are no pressure losses as the gas flows from the vessel to a location in the pipe just downstream of the vessel, then we can say that the total pressure inside the pipe is still equal to 1 psig. Because the gas is moving, we now have a pressure created by flowing gas, or a velocity pressure. We can measure the total pressure at any traverse location in the pipe using an impact probe. The impact probe will measure the static pressure inside the pipe plus the impact pressure, or velocity pressure, of the flowing gas, as illustrated in Figure 3.11. Regardless of where we traverse the impact probe inside the pipe,

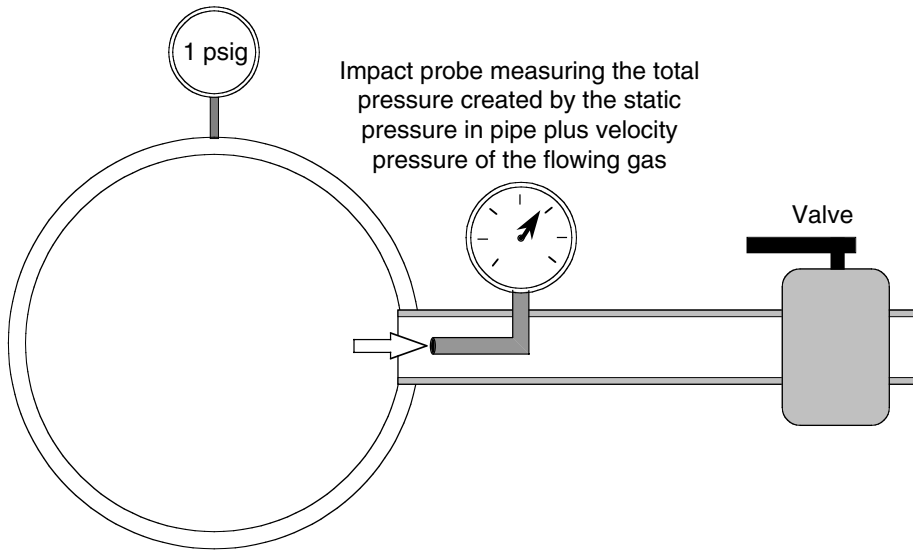


FIGURE 3.11 The difference between static, velocity, and total pressure.

we will get a total pressure reading of 1 psig. For example, if we place the impact probe directly on the pipe wall where the velocity is zero, we will read a total pressure of 1 psig. Because the velocity of the gas at the pipe wall is equal to zero, we can conclude that the static pressure at the wall is equal to 1 psig. If we locate the impact probe at the center of the pipe, we will still read a total pressure of 1 psig. At this location the static pressure will be lower because we now have a velocity pressure at this point. The velocity pressure can be calculated using the following equation:

$$P_v = \frac{\rho V^2}{2g_c} \quad (3.14)$$

where ρ is the density of the flowing gas, V is the velocity, and g_c is the gravitational constant.

Example 3.9

Suppose air is flowing inside a pipe with a velocity at the centerline equal to 200 feet per second and a density equal to 0.0765 lbm/ft³. What is the velocity pressure at the centerline of the pipe?

The value of $\rho = 0.0765$ lbm/ft³, $V = 200$ ft/s, $g_c = 32.2$ lbm-ft/lbf-s². Substituting these values into Equation 3.14 yields $P_v = 47.5$ lbf/ft² = 0.33 psig. Suppose at the centerline of the pipe an impact probe measures a total pressure of 1 psi; what is the static pressure at this location. From Equation 3.13 we can calculate the static pressure at the centerline of the pipe as $P_s = P_T - P_v = 1 - 0.33 = 0.67$ psig.

3.3.4.6 Pressure Loss

Obstructions within the flow stream of a pipe or duct can alter the flow direction and pattern of a fluid. An obstruction, for example, might consist of a fitting such as an inlet, elbow, tee, contraction, or expansion.

When a fluid flows through an obstruction, a reduction in the total pressure will occur. Pressure losses are the result of additional turbulence and/or flow separation created by sudden changes in the fluid momentum. This section discusses the general procedure for estimating the pressure drop through various fittings.

A complete theoretical analysis for calculating the flow through fittings has not yet been developed. Thus, the pressure drop is based on equations that rely heavily on experimental data. The most common method used to determine the pressure loss is to specify the *loss coefficient* K_L , defined as follows:

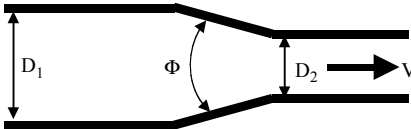
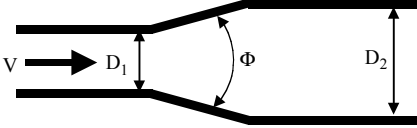
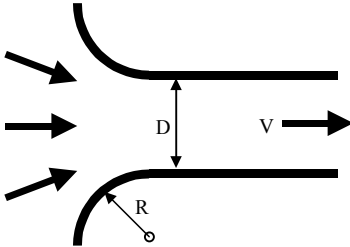
$$K_L = \frac{\Delta P}{\frac{1}{2}\rho V^2} \tag{3.15}$$

where ΔP is the pressure drop through the fitting, ρ is the approaching fluid density, and V is the approaching fluid velocity. Notice that the loss coefficient is dimensionless and is defined as the ratio of the pressure drop through a fitting to the approaching velocity pressure of the fluid stream. Solving Equation 3.15 for ΔP relates the pressure drop through a fitting:

$$\Delta P = K_L \frac{\rho V^2}{2g_c} \tag{3.16}$$

If the loss coefficient is equal to 1.0, then the pressure loss through that fitting will equal the velocity pressure of the approaching fluid stream, $\rho V^2/2$. The loss coefficient is strongly dependent on the geometry of the obstruction and the Reynolds number. The loss coefficient for turbulent flow through various fittings is given in Table 3.3. For more detailed information on loss coefficients through various fittings, refer to Idelchik²³ and Crane.²⁴

TABLE 3.3
Loss Coefficient for Various Fittings

Description	Sketch	Additional Data	K_L	
Contraction $\Delta P = K_L V^2 / (2 g_c)$		D_2/D_1	$\Phi = 60^\circ$	$\Phi = 180^\circ$
		0.0	0.08	0.50
		0.20	0.08	0.49
		0.40	0.07	0.42
		0.60	0.06	0.32
		0.80	0.05	0.18
		0.90	0.04	0.10
Expansion $\Delta P = K_L V^2 / (2 g_c)$		D_1/D_2	$\Phi = 10^\circ$	$\Phi = 180^\circ$
		0.0		1.00
		0.20	0.13	0.92
		0.40	0.11	0.72
		0.60	0.06	0.42
		0.80	0.03	0.16
Pipe Entrance $\Delta P = K_L V^2 / (2 g_c)$		R/D	$\Phi = 60^\circ$	
		0.0	0.50	
		0.10	0.12	
		> 0.20	0.03	

Example 3.10

Combustion air flows through the entrance of an eductor system. Compare the pressure drop through the entrance of a well-rounded inlet with a radius of 0.4 in. to a straight pipe inlet ($r = 0$). The diameter of the downstream pipe is 2 in. and the air velocity and density is 100 ft/s and 0.0765 lbm/ft³, respectively.

From Table 3.3, the loss coefficient for the well-rounded inlet ($R/D = 0.4/2 = 0.2$) is 0.03, and for the straight pipe inlet ($R/D = 0$) the loss coefficient is 0.5. Substituting the appropriate values into Equation 3.16, the pressure loss can be determined as:

$$\Delta P_{\text{bell}} = K_L \frac{\rho V^2}{2g_c} = 0.03 \frac{0.0765 \frac{\text{lbm}}{\text{ft}^3} \times (100)^2 \frac{\text{ft}^2}{\text{s}^2}}{2 \times 32.2 \frac{\text{lbm} \cdot \text{ft}}{\text{lbf} \cdot \text{s}^2}} = 0.356 \frac{\text{lbf}}{\text{ft}^2} = 0.069 \text{ in WC}$$

$$\Delta P_{\text{pipe}} = K_L \frac{\rho V^2}{2g_c} = 0.5 \frac{0.0765 \frac{\text{lbm}}{\text{ft}^3} \times (100)^2 \frac{\text{ft}^2}{\text{s}^2}}{2 \times 32.2 \frac{\text{lbm} \cdot \text{ft}}{\text{lbf} \cdot \text{s}^2}} = 5.94 \frac{\text{lbf}}{\text{ft}^2} = 1.143 \text{ in WC}$$

Notice that the straight pipe inlet has a pressure drop that is approximately 16.5 times greater than the well-rounded-bell inlet. Typically in the burner industry, one will see well-rounded inlets to burner appurtenances in order to reduce the pressure drop.

3.4 CALCULATING THE HEAT RELEASE FROM A BURNER

3.4.1 DEFINITION OF HEAT RELEASE

When shopping for a light bulb, one makes a selection depending on the lighting condition that is needed. For example, one might buy a 25-W bulb for a small reading lamp; but to light an entire room, one would use a bulb with more power output, such as a 100-W bulb. Power is a measure of how much energy is released in a given amount of time. In SI units, power is usually written as watts (W) or kilowatts (kW), and in English units as BTU/hr or MMBTU/hr (MM = millions of BTU/hr) (1 W = 3.41 BTU/hr). The amount of power output from a burner, referred to as *burner heat release*, depends on how much fuel the burner consumes and how much chemical energy the fuel has (heating value), which is referred to as the heating value of the fuel and can be written mathematically as:

$$HR = \dot{m} \times HV \quad (3.17)$$

where HR is the heat release of the burner, $\dot{m}$ is the mass flow rate of the fuel, and HV is the heating value of the fuel.

The heat release of a burner is used throughout many areas of interest in combustion. For example, burner manufacturers will sometimes relate the length of a flame to the heat release of a burner. One might describe the length of a flame, for example, as 2 ft per MMBTU/hr. Burner manufacturers will often relate emissions from a burner to the heat release. Sometimes, one will refer to the emissions from a burner as the pounds of NO_x emitted per MMBTU/hr. Also, burner manufacturers will size the burner based on the heat release. For example, a 20-MMBTU/hr burner will have larger physical dimensions than a 4-MMBTU/hr burner operating under the same furnace conditions.

In the burner industry, the heat release typically varies between 1 and 20 MMBTU/hr. If one were to compare the power output from a burner to the power output of a light bulb, one would find that a burner operating at 20 MMBTU/hr has a power output equivalent to approximately 60,000 light bulbs at 100 W each.

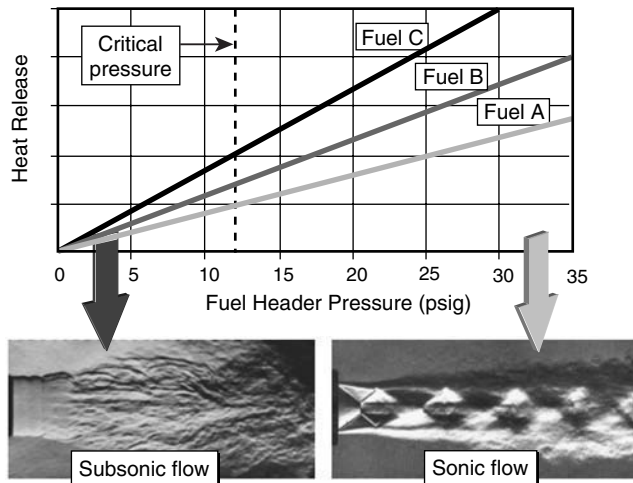


FIGURE 3.12 A typical fuel capacity curve for subsonic and sonic flow.

3.4.2 FUEL GAS ORIFICE CALCULATIONS

3.4.2.1 Discussion of Sonic and Subsonic Flow

Burner manufacturers typically provide customers with curves that show the heat release of a burner at various fuel pressures and compositions. For example, Figure 3.12 shows a typical capacity curve. The y-axis represents the heat release of the burner in MMBTU/hr and the x-axis represents the fuel pressure. Sometimes, several curves will be displayed, each representing a different fuel mixture with a different heating value.

Notice the shape of the capacity curve in Figure 3.12: for a fuel pressure ranging from zero to approximately 12 psig, the shape of the curve is bent; but above 12 psig, the shape becomes linear. The shape of the curve changes from a bent shape to a linear shape at a point called the *critical pressure*. The critical pressure usually varies from 12 to 15 psig, depending on the fuel composition and the temperature. Below the critical pressure, the burner heat release varies as the square root of the fuel pressure. Above the critical pressure, the burner heat release varies linearly with the fuel pressure. Why does this phenomena occur?

The region below the critical pressure is referred to as the *subsonic* region while the region above the critical pressure is referred to as the *sonic* region. In the subsonic region, the fuel exits the orifice at a velocity less than the speed of sound in the fuel gas. However, when the fuel pressure reaches the critical pressure, the fuel exits the orifice at a velocity equal to the speed of sound. The speed of sound in the fuel gas is the limiting velocity of the fuel at the orifice exit. That is, at fuel pressures above the critical pressure, the velocity of the fuel remains constant at the speed of sound (also called choked flow). If the velocity of the gas remains constant at fuel pressures above the critical pressure, then how does the burner heat release (or mass flow rate of fuel) increase as one increases the fuel pressure?

At a fuel pressure above the critical pressure, a marked change occurs in the structure of the fuel jet. At fuel pressures above the critical pressure, shock waves form at the orifice exit plane and downstream, as shown in Figure 3.12. The shock wave at the orifice exit plane consists of a thin layer of compressed fuel that acts as a barrier causing the fuel to compress upstream of the orifice exit. This, in turn, increases the density of the fuel at the orifice exit, allowing for an increase in the fuel mass flow rate.

3.4.2.2 Equations for Calculating Fuel Flow Rate

When designing a burner, burner engineers must determine the correct area of the fuel orifice. If they design the fuel orifice area too large, the burner will operate at a low fuel pressure. This could result in the fuel not properly mixing with the combustion air creating flames that produce soot or that could impinge on the process tubes. If the burner engineer designs the orifice area too small, the burner will not be able to achieve the required heat release at the customer's design pressure. How does a burner engineer determine the orifice size?

Burner engineers use equations based on the ideal gas law and assumptions of ideal flow to calculate the flow rate of fuel through an orifice. The first step in calculating the amount of fuel gas discharging through an orifice is to determine if it is operating above or below the critical pressure. This can be determined by calculating the critical pressure ratio defined as follows:

$$P_c = \left[\frac{2}{k+1} \right]^{k/(k-1)} \quad (3.18)$$

where P_c is the critical pressure ratio and k is the ratio of specific heats of the fuel.

If $P_c > P_b/P_t$, then the fuel exits the orifice at sonic conditions. If $P_c < P_b/P_t$, then the fuel exits the orifice at subsonic conditions. The P_b and P_t terms represent atmospheric pressure and fuel pressure in absolute, respectively.

The second step is to determine the mass flow rate of the fuel through the orifice. If the fuel exits the orifice at sonic conditions, Equation 3.19 is used to determine the mass flow rate:

$$\dot{m} = \frac{c_d P_t g_c A}{\sqrt{T_t \bar{R} g_c / MW}} k^{\frac{1}{2}} \left[\frac{2}{k+1} \right]^{\frac{k+1}{2(k-1)}} \quad (3.19)$$

where c_d is the orifice discharge coefficient (to be discussed later in this chapter), A is the area of the orifice, T_t is the total temperature of the fuel gas, $\bar{R}$ is the universal gas constant equal to 8314.34 J/kmol/K = 1545.32 (ft-lbf)/(lb-mole-R), MW is the molecular weight of the fuel, and g_c is the gravitational constant equal to 32.2 (lbm · ft)/(lbf · s²) = 1.0 (kg · m)/(N · s²).

If the fuel exits the orifice at subsonic conditions, Equation 3.20 can be used to determine the mass flow rate:

$$\dot{m} = c_d \rho_e A M_e c_e \quad (3.20)$$

where

$$M_e = \sqrt{\frac{2}{k-1} \left[\left(\frac{P_t}{P_b} \right)^{\frac{k-1}{k}} - 1 \right]} \quad (3.21)$$

$$T_e = \frac{T_t}{1 + \frac{k-1}{2} M_e^2} \quad (3.22)$$

$$c_e = \left[\frac{k T_e \bar{R}}{MW} \right]^{\frac{1}{2}} \quad (3.23)$$

$$\rho_e = \frac{P_b}{\frac{T_e \bar{R}}{MW}} \quad (3.24)$$

The subscript e denotes the orifice exit, M_e is the Mach number of the fuel, T_e is the temperature of the fuel, c_e is the speed of sound in the fuel, and ρ_e is the density of the fuel.

3.4.2.3 Discharge Coefficient

Many different fuel nozzle types are used in the burner industry, the type depending on the application. Figure 3.13 shows various types of nozzles commonly employed in the industry. Each of these nozzles is designed with a different internal shape. The internal shape of the nozzle can significantly affect the mass flow rate of the fuel. For example, Figure 3.14 is a schematic that



FIGURE 3.13 Various burner fuel nozzles.

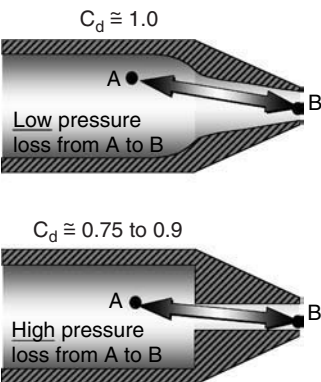
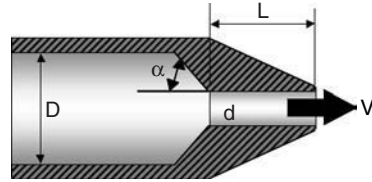


FIGURE 3.14 Orifice discharge coefficient explanation.



- Length-to-diameter ratio = L/d
- Reynolds number = $V \times d/\nu$
- Beta ratio = d/D
- Port angle = α

FIGURE 3.15 Variables affecting the orifice discharge coefficient.

shows the internals of two nozzles with the same port diameter. If these nozzles were operating under identical conditions, which one would flow more fuel? The upper one would, because it has less pressure drop as the fuel speeds toward the exit. The lower orifice design would create a small recirculation pattern just downstream from the entrance (called a *vena contracta*). This *vena contracta* creates a restriction in the flow, thus reducing the effective orifice area. To compensate for the results of the ideal equations and assumptions, a constant is introduced to account for the complexity of the flow that makes it nonideal. This constant is called the *orifice discharge coefficient*.

The discharge coefficient is defined as the ratio of the actual mass flow rate of a fluid through a nozzle to the ideal mass flow rate and is written as:

$$c_d = \frac{\dot{m}_{\text{actual}}}{\dot{m}_{\text{ideal}}} \quad (3.25)$$

The ideal mass flow rate is defined as the mass flow rate calculated using the ideal gas law and assumptions of ideal flow — no pressure losses due to the internals of the nozzle or tip. The value of the discharge coefficient, for a burner nozzle, is usually determined experimentally. Typically, in the burner industry, the discharge coefficient varies from about 0.75 to 0.95. Several factors that can affect the discharge coefficient include the (1) length-to-diameter ratio of the port, (2) Reynolds number of the fluid in the port, (3) beta ratio, (4) port angle, and (5) manufacturing tolerances.²⁵ See [Figure 3.15](#) for a description of these variables.

3.4.3 EXAMPLE CALCULATION: HEAT RELEASE FROM A BURNER

Example 3.11

A fuel is flowing through an orifice with the following values:

- LHV of fuel = 909 BTU/scf
- Orifice area = 0.5 in.²
- Molecular weight (MW) of fuel = 16
- Ratio of specific heat of fuel = 1.31
- Total fuel pressure = 35 psig
- Total fuel temperature = 60°F
- Atmospheric pressure = 14.7 psia.

Find the heat release of the burner.

Step 1: Determine if the flow is sonic or subsonic using Equation (3.18).

$$P_c = \left[\frac{2}{k+1} \right]^{k/(k-1)} = \left[\frac{2}{1.31+1} \right]^{1.31/(1.31-1)} = 0.544$$

$$\frac{P_b}{P_t} = \frac{14.7}{35+14.7} = 0.296$$

Because $P_c > P_b/P_t$, the fuel exits at sonic conditions.

Step 2: Determine the mass flow rate of fuel. Because the flow is choked, we will use Equation (3.19) to determine the mass flow rate:

$$\dot{m} = \frac{c_d P_t g_c A}{\sqrt{T_t \bar{R} g_c / MW}} k^{\frac{1}{2}} \left[\frac{2}{k+1} \right]^{\frac{k+1}{2(k-1)}}$$

$$\dot{m} = \frac{0.85 \times (35+14.7) \times 144 \times 32.2 \times \frac{0.5}{144}}{\sqrt{\frac{(60+460) \times 1545.32 \times 32.2}{16}}} (1.31)^{\frac{1}{2}} \left[\frac{2}{1.31+1} \right]^{\frac{1.31+1}{2(1.31-1)}}$$

$$\dot{m} = 0.35784 \frac{\text{lb}}{\text{s}} = 1288.2 \frac{\text{lb}}{\text{hr}}$$

Step 3: Determine the heat release of the burner. First determine the density of the fuel at standard conditions using Equation (3.2):

$$\rho = 0.0765 \frac{\text{lb}}{\text{ft}^3} \times \frac{16}{29} = 0.0422 \frac{\text{lb}}{\text{ft}^3}$$

$$HR = \dot{m} \left(\frac{\text{lb}}{\text{hr}} \right) \times \text{LHV} \left(\frac{\text{BTU}}{\text{lb}} \right) = \left(1288.2 \frac{\text{lb}}{\text{hr}} \right) \times \left(909 \frac{\text{BTU}}{\text{ft}^3} \times \frac{1}{0.0422} \frac{\text{ft}^3}{\text{lb}} \right) = 27.75 \frac{\text{MMBTU}}{\text{hr}}$$

Example 3.11 continued — If the fuel pressure is reduced to 10 psig, what is the burner heat release?

Step 1: Determine if the flow is sonic or subsonic.

$$P_c = \left[\frac{2}{k+1} \right]^{k/(k-1)} = \left[\frac{2}{1.31+1} \right]^{1.31/(1.31-1)} = 0.544$$

$$\frac{P_b}{P_t} = \frac{14.7}{10+14.7} = 0.595$$

Because $P_c < P_b/P_t$, the fuel exits at subsonic conditions.

Step 2: Determine the mass flow rate of fuel. Because the flow is not choked, we use Equation 3.20 to determine the mass flow rate.

$$M_e = \sqrt{\frac{2}{k-1} \left[\left(\frac{P_t}{P_b} \right)^{\frac{k-1}{k}} - 1 \right]} = \sqrt{\frac{2}{1.3-1} \left[\left(\frac{10+14.7}{14.7} \right)^{\frac{1.3-1}{1.3}} - 1 \right]} = 0.921$$

$$T_e = \frac{T_t}{1 + \frac{k-1}{2} M_e^2} = \frac{60 + 460}{1 + \left(\frac{1.3-1}{2} \right) (0.921)^2} = 461.31 \text{ R}$$

$$c_e = \left[\frac{k T_e \bar{R} g_c}{MW} \right]^{\frac{1}{2}} = \left[\frac{1.3 \times 461.31 \times 1545.32 \times 32.2}{16} \right]^{\frac{1}{2}} = 1365.67 \frac{\text{ft}}{\text{s}}$$

$$\rho_e = \frac{P_b}{\frac{T_e \bar{R}}{MW}} = \frac{14.7 \times 144}{\frac{461.31 \times 1545.32}{16}} = 0.0475$$

$$\dot{m} = c_d \rho_e A M_e c_e = 0.85 \times 0.0475 \times \frac{0.5}{144} \times 0.921 \times 1365.67$$

$$\dot{m} = 0.1763 \frac{\text{lb}}{\text{s}} = 634.8 \frac{\text{lb}}{\text{hr}}$$

Step 3: Determine the heat release of the burner using Equation 3.17.

$$HR = \dot{m} \left(\frac{\text{lb}}{\text{hr}} \right) \times \text{LHV} \left(\frac{\text{BTU}}{\text{lb}} \right) = \left(634.8 \frac{\text{lb}}{\text{hr}} \right) \times \left(909 \frac{\text{BTU}}{\text{ft}^3} \times \frac{1}{0.0422} \frac{\text{ft}^3}{\text{lb}} \right) = 13.67 \frac{\text{MMBTU}}{\text{hr}}$$

3.5 COMBUSTION AIR FLOW RATE THROUGH NATURAL-DRAFT BURNERS

Figure 3.16 shows a plot of a typical capacity curve that many burner manufacturers use for sizing burners. Capacity curves describe the airside pressure drop through burners of various sizes at different heat releases. These curves are usually generated based on experimental data. The curves shown in this particular example are based on burners operating in the natural-draft mode with 15% excess air in the furnace at an atmospheric temperature and pressure of 59°F (15°C) and 14.696 psia (1 bar), respectively.

When burners operate at different ambient conditions and excess air levels, the value of the airside pressure drop obtained from the capacity curves must be corrected. The equation used to correct for the airside pressure drop can be derived as follows. The airside pressure drop through a burner is proportional to the velocity pressure of the air and can be written as:

$$\Delta P \propto \rho V^2 \quad (3.26)$$

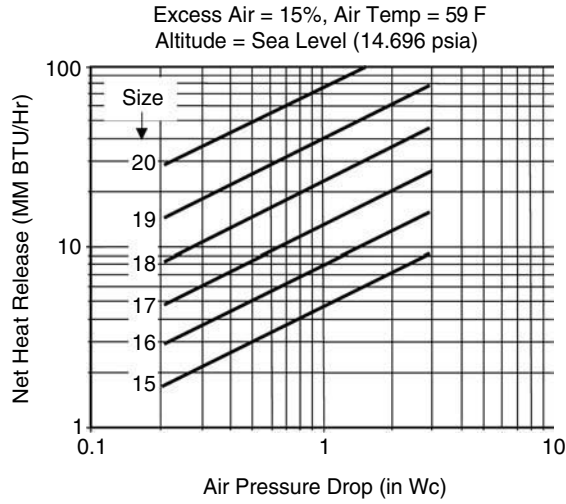


FIGURE 3.16 Capacity curves used by many burner manufacturers for sizing burners.

where dP is the pressure drop through the burner, ρ is the density of the combustion air, and V is the mean velocity of the air at a particular location in the burner. The density of the combustion air can be related to the combustion air temperature T and atmospheric pressure P using the ideal gas law:

$$\rho \propto \frac{P}{T} \quad (3.27)$$

The velocity V of the air through the burner is proportional to the mass flow of air going through the burner and the density. This can be written as:

$$V \propto \frac{\dot{m}}{\rho} \propto \frac{(100 + EA)}{\rho} \quad (3.28)$$

where $\dot{m}$ represents the mass flow rate and EA represents the percent excess air in the furnace. Substituting Equations 3.27 and 3.28 into Equation 3.26 gives:

$$\Delta P \propto (100 + EA)^2 \times \frac{P}{T} \quad (3.29)$$

Equation 3.29 can be used to write the following equation to correct for the airside pressure drop at actual firing conditions:

$$\Delta P_{\text{Actual}} = \Delta P_{CC} \times \left(\frac{100 + EA_{\text{Actual}}}{100 + EA_{CC}} \right)^2 \times \left(\frac{T_{\text{Actual}}}{T_{CC}} \right) \times \left(\frac{P_{CC}}{P_{\text{Actual}}} \right) \quad (3.30)$$

where subscript *Actual* represents the actual firing conditions and the subscript *CC* represents the variables from the capacity curves (i.e., $T_{CC} = 460 + 59$, $P_{CC} = 14.696$ psia, $EA_{CC} = 15\%$). Notice that as the temperature of the combustion air increases, the airside pressure drop through the burner also increases. This occurs because increasing the temperature reduces the density of the combustion air. A reduction in the density of the combustion air requires a higher volumetric airflow rate, through the burner, which results in an increase in the pressure drop. Similarly, if

the atmospheric pressure is reduced, the combustion air density is reduced and, hence, the airside pressure drop is increased.

Example 3.12

Using Figure 3.16, determine the pressure drop through a size 15 burner with a heat release of 3 MMBTU/hr operating at 13% excess air. The combustion air temperature is 100°F and the atmospheric pressure is 14.0 psia.

From the capacity curves in Figure 3.16, for a heat release of 3 MMBTU/hr at standard conditions, the pressure drop will be approximately 0.5 inches of water column. This will be the pressure drop if the burner is operating at 15% excess air with the combustion air temperature at 59°F and an atmospheric pressure of 14.696 psia. To correct for the actual firing conditions, we can use Equation 3.30.

$$\Delta P_1 = 0.5 \times \left(\frac{100 + 13}{100 + 15} \right)^2 \times \left(\frac{460 + 100}{460 + 59} \right) \times \left(\frac{14.696}{14.0} \right) = 0.547 \text{ (in WC)}$$

Although the percent excess air is reduced from 15 to 13%, the pressure drop through the burner has increased because the density of the combustion air is lower at actual conditions than at standard conditions. This results in a higher volumetric flow rate of combustion air through the burner and, hence, a larger pressure loss.

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