F.5.2 APH Operation

In order to provide the means to effectively monitor and operate an APH system, the following design features (as applicable) are recommended.

a) Pressure and temperature connections should be provided upstream and downstream of the APH exchanger in both the combustion air and flue gas ducting for performance monitoring and troubleshooting.

b) Connections for flue gas analyzers should be provided upstream and downstream of the APH exchanger in the flue gas ducting for leak detection, system mass balances, and troubleshooting.

c) Pressure connections should be provided upstream and downstream of the fan(s).

d) Flow element(s) should be located downstream of the APH to measure combustion air flow.

e) Combustion air ducting to parallel fireboxes/cells should be hydraulically similar.

f) Combustion air ducting to multiple independently fired fireboxes/cells should contain a flow-control damper that permits O_2 control for each cell over the APH system's operating range.

g) Flue gas ducting from parallel fireboxes/cells should be hydraulically similar.

h) Flue gas ducting from multiple independently fired fireboxes/cells should contain a flow-control damper that permits arch/roof draft control for each cell over the APH system's operating range.

F.5.3 APH Maintenance

The most desirable location for duct blinds and dampers is near grade to limit work on or over an operating fired heater. When locating the fans and the APH, accessibility for maintenance should be considered.

Cleaning facilities are typically provided for APHs in heavy-fuel-oil-fired applications.

Refractory systems in existing heaters and ductwork should be inspected periodically for mechanical integrity and repaired, as required.

F.5.4 APH System Equipment Failure

It is usual to provide provisions for a secondary or fail-safe mode of heater operation. In most applications, the APH system is designed to permit stable fired-heater operation whenever the APH system experiences a mechanical failure. The two most common secondary operating modes are the following:

a) bypassing the APH system and defaulting to natural draft operation, and

b) activating a spare fan or alternative device.

The APH system should have the means to confirm that such a change has been safely and successfully executed. Refer to F.3.3 and F.4.5 for additional guidelines for natural-draft operations.

F.6 APH Performance Guidelines

F.6.1 Introduction

The common design objective of most APH systems is to maximize the fired-heater's efficiency. To achieve this objective, it is important to select a cold-end design (flue gas) temperature that maximizes flue gas heat recovery and minimizes fouling and corrosion. The flue gas temperature at which corrosion and fouling become excessive is affected by the following:

a) fuel sulfur, ash, and other contaminants,

b) fuel additives and flue gas additives,

c) flue gas oxygen and moisture content, and

d) air-preheater design.

F.6.2 Cold-end Temperatures

F.6.2.1 Recommended Minimum Metal Temperatures

Corrosion of air-preheater cold-end surfaces is generally caused by the condensation of sulfuric acid vapor formed from the products of combustion of a sulfur-laden fuel. The acidic deposits also provide a moist surface that is ideal for collecting solid particles that foul the APH's heat-transfer surface. Consequently, to obtain the preheater design life, it is imperative to measure and control the APH's cold-end surfaces above the acid-dew-point temperature.

Thermally aggressive APH Systems (i.e. those with metal temperatures at or below the FGADP temperatures) should mitigate such risks via the adoption of one or more of the following practices.

a) Separate the exchanger into a hot and cold module and make the cold module "easily replaceable."

b) Use corrosion-resistant materials: glass tubes, glass coated tubes, glass coated plates, coated tubes, stainless steel, or some other special corrosion-resistant material.

   NOTE 1 Glass tubes can break, which will reduce the efficiency gain from these tubes (most designs permit individual replacement of tubes).

   NOTE 2 Glass coatings can become porous and the tube/plate substrate will corrode (however, these tubes can be individually replaced).

   NOTE 3 Tube coatings are typically soft and subject to erosion.

c) Use thicker tubes and/or plates to provide additional corrosion allowance.

   NOTE Forecasting or calculating the corrosion rate(s) for the several acid and cold-end material combinations is beyond the scope of this annex. Refer to the bibliography for additional sources of information on corrosion rates and acid condensation rates, and/or consult an authoritative source for application-specific guidance.

F.6.2.2 Recommended Minimum Flue Gas Temperatures

For APH applications in which the exchanger's minimum metal temperature is not measured or monitored, a common corrosion-avoidance practice is to control the cold flue gas temperature above a calculated minimum flue gas temperature. This minimum flue gas-temperature limit is usually the appropriate minimum metal temperature from Figure F.4 and Figure F.5 plus a small temperature allowance. Temperature allowances of 8 °C to 14 °C (15 °F to 25 °F) are typical.

F.6.2.3 Flue Gas Dew-point Monitoring

For APH systems with the capacity for reducing stack temperatures below the dew-point temperature, a program of dew-point testing can be helpful. The dew-point determinations can be used to adjust the APH's cold-end temperature. The cold-end metal temperature is lower than the cold flue gas temperature, so care should be exercised when the cold flue gas temperature is the only measurement available.

F.6.3 Hot-end Temperatures

F.6.3.1 General

The APH shall be designed to accommodate the full range of flue gas temperatures anticipated.

The temperature of the hot flue gas leaving a fired heater (hot-end temperature) is a function of heat transfer surface area, firing rate, and process temperature. The hot-end temperature increases as the heat transfer surfaces foul over time. The APH must be designed for the resulting increase in flue gas temperature.

The approach temperature is typically defined as the temperature difference between the flue gas leaving the convection section and the process temperature of the last convection section coil. Fired heater approach temperatures are typically in the range of 60 °C to 160 °C (100 °F to 300 °F).

F.6.3.2 Regenerative APH Exchangers

Regenerative APHs are generally suitable for maximum inlet flue gas temperatures up to 540 °C (1000 °F). Special materials and configurations allow regenerative APH use for flue gas temperatures up to 680 °C (1250 °F). The APH manufacturer should be consulted for specific recommendations.

F.6.3.3 Recuperative APH Exchangers

The standard cast-iron recuperative APH is generally suitable for maximum flue gas temperatures up to 540 °C (1000 °F). By using special materials and constructions, these APHs can be designed for maximum flue gas temperatures up to 980 °C (1800 °F). The exchanger manufacturer should be consulted for specific recommendations.

F.6.3.4 Heat Pipes and Indirect Systems

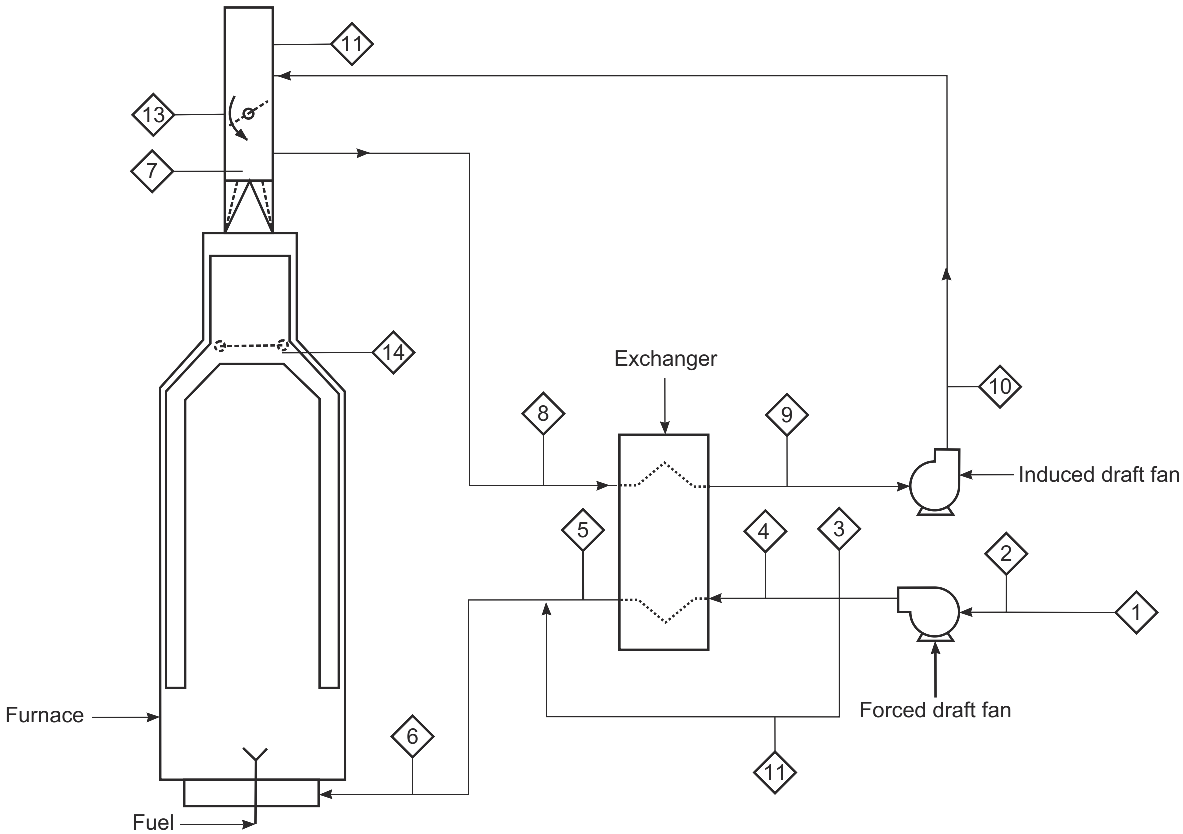
The coils of working fluid systems, whether heat pipes or indirect APH systems, are usually limited by the fluids' maximum allowable film temperatures, not the exchangers' coil material(s). For indirect systems containing a heat-transfer fluid, the fluid manufacturer's maximum allowable film-temperature limit should be followed. In the case of the heat-pipe preheater, the preheater manufacturer should be consulted for specific recommendations.

F.7 Ductwork Design and Analysis

F.7.1 Introduction

F.8 is intended to provide engineering procedures for the design and analysis of complex APH systems with regard to pressure drops and pressure profiles. It has been developed according to, and based on, commonly used correlations and procedures. While the individual correlations are relatively simple, their cumulative application to entire APH systems can become complicated. Comments on some specific applications have been included to provide guidance. F.8 is not intended as a primer on fluid flow; see the references in F.8.9 for additional information.

The basic assumption is that all of the pertinent design data, such as flows, temperatures, and pressure drops, for all components are available for integration into the APH-system design. These data should be compiled in a usable form (see Figure F.6 as an example). Additionally, it is necessary to know or to layout the spatial relationships between the basic pieces of equipment when developing the duct design.



Point Number	Flow Rate kg/h (lb/h)	Temperature °C (°F)	Pressure mm H_2O (in. H_2O)
1			
2			
3			
4			
5			
6			
7			
8			
9			
10			
11			
12			
13			
14			

Figure F.6—System Worksheet for Design and/or Analysis [16]

F.7.2 Velocity Guidelines

In the absence of project-specific values, the following design parameters should be used.

a) Straight duct velocity should be limited to 15 m/s (50 ft/s) at 100 % of design end-of-run conditions.

b) Turns or tee velocity should be limited to 15 m/s (50 ft/s) at 100 % of design end-of-run conditions.

c) Burner air-supply duct velocity should be based on the velocity head in these ducts equal to a maximum of 10 % of the burner-air side pressure drop. The resulting velocities should be no more than the following:

   1) 8 m/s (25 ft/s) for forced or balanced draft systems with natural draft capability;

   2) 9 m/s (30 ft/s) for forced or balanced draft systems without natural draft capability.

These guidelines can be altered to reflect the system's physical constraints and target efficiency. Lower velocities may be justified by lower power requirements.

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F.7.3 Friction Factor Calculations

F.7.3.1 General

Before performing any of the pressure-drop calculations contained in F.8.4, the flow elements' friction factors shall be obtained.

NOTE The correlations of F.8.4 are predicated on the use of Moody friction factors, not Fanning friction factors. The Moody friction factors for lined and unlined ducts can be read from Figure F.7. For the calculation of the Reynolds number (Re) in either SI or USC units, see F.8.3.2.

F.7.3.2 Reynolds Number

The Reynolds number, Re, is calculated in SI units as given in Equation (F.1) or Equation (F.2):

$$\mathrm{Re} = \rho \times v \times d / \mu \tag{F.1}$$

or

$$\mathrm{Re} = q_{\mathrm{m,a}} \times d / \mu \tag{F.2}$$

[16] Users of this Figure should not rely exclusively on the information contained in this document. Sound business, scientific, engineering, and safety judgment should be used in employing the information contained herein.

where

d is the duct inside diameter, in millimeters;

ρ is the flow density, in kilograms per cubic meter (kg/m^3);

v is the linear velocity, in meters per second;

μ is the viscosity, in millipascal seconds (mPa·s);

$q_{m,a}$ is the areic mass flow rate, in kilograms per square meter per second (kg/m^2·s).

The Reynolds number, Re, is calculated in USC units as given in Equation (F.3) or Equation (F.4):

$$\mathrm{Re} = 123.9 \times \rho \times v \times d / \mu \tag{F.3}$$

or

$$\mathrm{Re} = 123.9 \times q_{\mathrm{m,a}} \times d / \mu \tag{F.4}$$

where

d is the duct inside diameter, in inches;

ρ is the flow density, in pounds per cubic foot (lb/ft^3);

v is the linear velocity, in feet per second (ft/s);

μ is the viscosity, in centipoise (cP);

$q_{m,a}$ is the areic mass flow rate, in pounds per square foot per second (lb/ft^2·s).

NOTE "Areic" is the SI term for "per unit area," in this case "mass flow rate per unit area."

F.7.3.3 Flue Gas and Air Viscosity

If the viscosities, μ, of the combustion air and/or flue gas streams are not known at all pertinent locations within the system, μ, expressed in millipascal seconds (mPa·s) and μ, expressed in centipoise (cP), may be calculated using the generalized Equation (F.5) and Equation (F.6), respectively, for both air and flue gas without introducing any significant error into the pressure-drop calculations:

$$\mu = 0.0162\ (T/255.6)^{0.691} \tag{F.5}$$

where

T is the absolute temperature, in kelvin (K).

$$\mu = 0.0162\ (T/460)^{0.691} \tag{F.6}$$

where

T is the absolute temperature, in degrees Rankine (°R).

NOTE Rankine is a deprecated unit.

F.7.4 Pressure Drop Calculations

F.7.4.1 General

The following equations and figures are a synopsis of the large quantity of available literature on the subject of fluid flow. This material has been used successfully in the design of duct systems and it is thought to be particularly useful in that type of calculation. Two formats of each correlation are presented: linear velocity basis and mass velocity basis. Use of either format remains the preference of the designer, as both formats produce similar results.

F.7.4.2 Pressure Drop in a Straight Duct

F.7.4.2.1 Pressure Drop

The correlations in Equation (F.7) to Equation (F.11) may be applied to straight ducts, with or without internal refractory linings. Additionally, these correlations can be used to calculate fitting losses for any fitting with a hydraulic length. For example, Figure F.9 provides the equivalent lengths of various physical configurations of cylindrical mitered elbows. The mitered elbow's hydraulic length that is used with Equation (F.7) to Equation (F.11) can be obtained by multiplying the elbow's equivalent lengths (from Figure F.9) by its flow diameter.

The pressure drop per 100 m, $\Delta P_{SI}/100$, expressed in millimeters of water column (mm H_2O), is given by Equation (F.7) and Equation (F.8):

$$\Delta P_{SI}/100 = (5.098 \times 10^3) f_{mF} \times \rho \times v^2 / d \quad \text{(F.7)}$$

$$\Delta P_{SI}/100 = (5.098 \times 10^3) f_{mF} \times q_{m,a}{}^2 / \rho \times d \quad \text{(F.8)}$$

where

f_{mF} is Moody's friction factor (see Figure F.7);

ρ is the flowing bulk density, in kilograms per cubic meter;

v is the linear velocity, in meters per second;

$q_{m,a}$ is the areic mass flow rate, in kilograms per square meter per second;

d is the duct inside diameter, in millimeters.

The pressure drop per 100 ft, $\Delta P_{USC}/100$, expressed in inches of water column (in. H_2O), is given by Equation (F.9) and Equation (F.10):

$$\Delta P_{USC}/100 = (3.587) f_{mF} \times \rho \times v^2 / d \quad \text{(F.9)}$$

$$\Delta P_{USC}/100 = (3.587) f_{mF} \times q_{m,a}{}^2 / \rho \times d \quad \text{(F.10)}$$

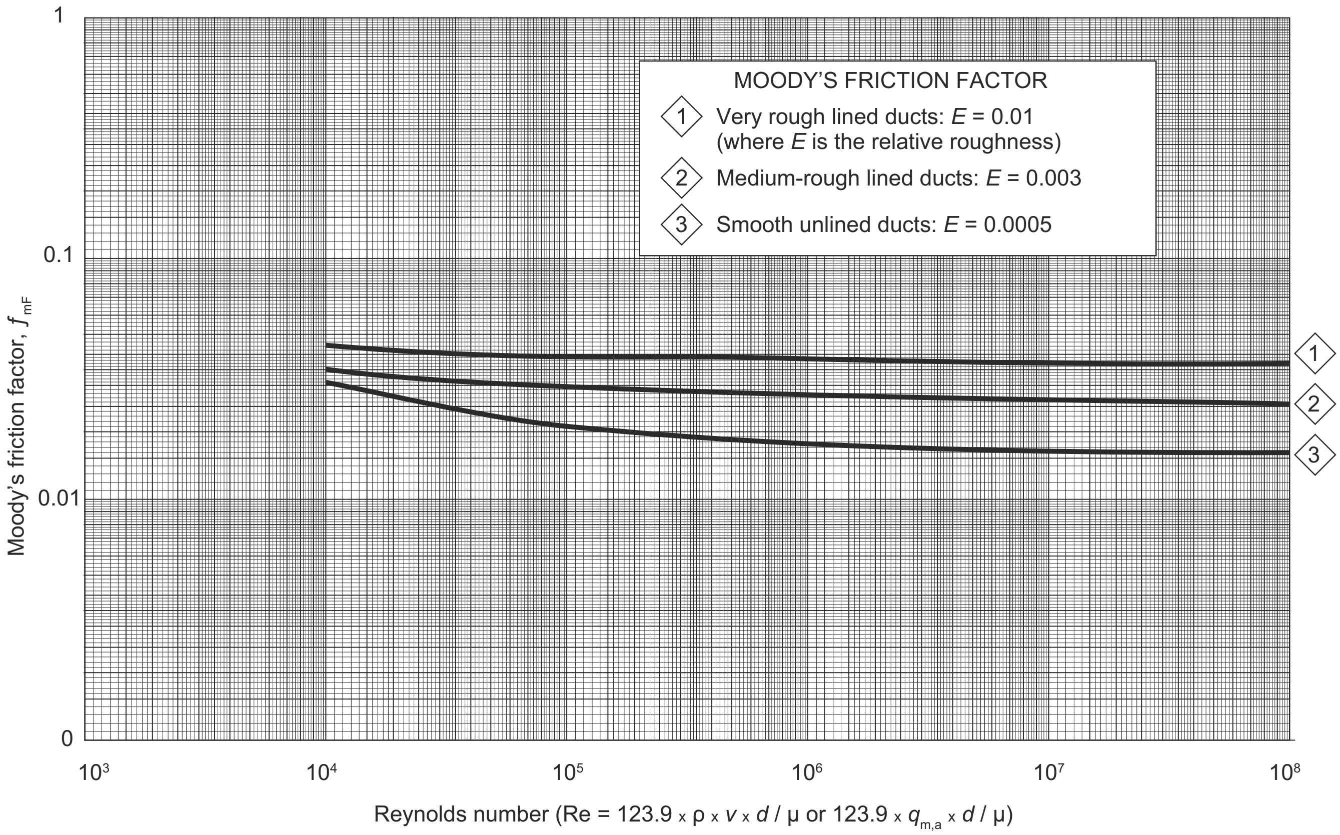


Figure F.7—Moody's Friction Factor vs Reynolds Number

where

f_{mF} is Moody's friction factor (see Figure F.7);

ρ is the flow density, in pounds per cubic foot;

v is the linear velocity, in feet per second;

$q_{m,a}$ is the areic mass flow rate, in pounds-mass per square foot per second;

d is the duct inside diameter, in inches.

F.7.4.2.2 Hydraulic Mean Diameter

Equation (F.1) through Equation (F.4) and Equation (F.7) through Equation (F.10) employ a diameter dimension, d, and hence are applicable to round ducts. To use these equations for rectangular ducts, an equivalent circular duct diameter, also referred to as the hydraulic mean diameter, needs to be calculated. A useful correlation, in SI or USC units, for the hydraulic mean diameter, d_e, expressed in millimeters (inches), is given in Equation (F.11):

$$d_e = 2ab/(a + b) \quad \text{(F.11)}$$

where

a is the length of one side of rectangle, expressed in millimeters (inches);

b is the length of adjacent side of rectangle, expressed in millimeters (inches).

NOTE When using d in Equation (F.11), use the actual velocity calculated for the rectangular duct.

F.7.4.3 Pressure Drop Estimation in Straight Ducts

By making several assumptions, the calculation of pressure drop in straight ducts can be reduced to a simplifying chart, presented for convenience as Figure F.8. Any error introduced is not significant for most cases.

NOTE When the pressure drop, ΔP, as given in Equation (F.12), is determined from Figure F.8 using a hydraulic mean diameter, it is necessary to apply the correlation shown on the curve rather than the one in Equation (F.11).

$$\Delta P = \Delta P_1 \times C_1 \times C_2 \quad \text{(F.12)}$$

where

ΔP is the corrected pressure drop per 30 linear m (100 linear ft), expressed in mm H_2O (in. H_2O);

ΔP_1 is the uncorrected pressure drop taken from Figure 8 a);

C_1 is a pressure-drop correction factor for temperature taken from Figure F.8 b);

C_2 is a roughness correction factor, as follows:

- — very rough (e.g. brick): 1.0;
- — medium-rough (e.g. castable refractory): 0.68;
- — smooth (e.g. unlined steel): 0.45.

The calculation for rectangular ducts is as given in Equation (F.13):

$$] d_e = 1.3[(ab)^{0.625} / (a + b)^{0.25}] \quad \text{(F.13)}$$

F.7.4.4 Pressure Drop in Fittings and Changes in Cross-section

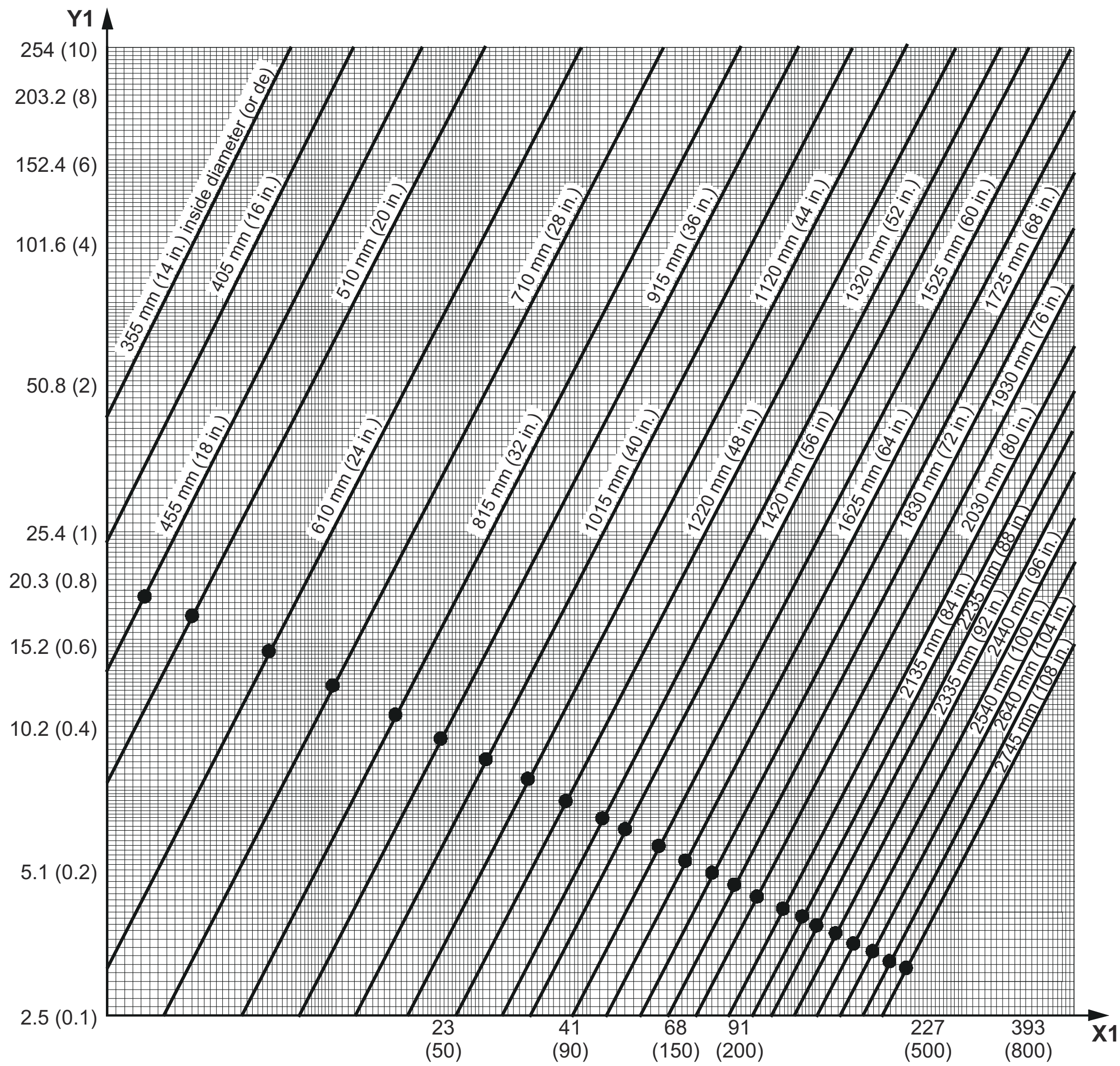
The pressure drop, Δp, of formed round elbows, various fittings, shape changes, and flow disturbances can be calculated with the loss coefficients provided in Table F.2 and Equation (F.14) and Equation (F.15) for SI units, with Δp expressed in millimeters of water column (mm H_2O), and Equation (F.16) and Equation (F.17) for USC units with Δp expressed in inches of water column (in. H_2O).

In SI units:

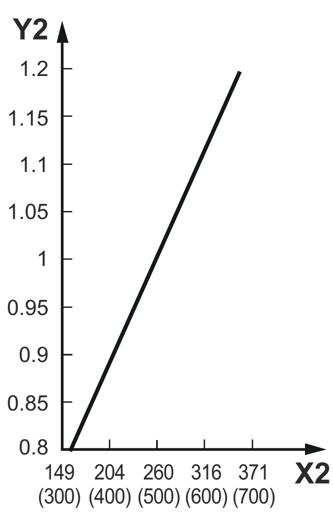
$$\Delta p = C(5.102 \times 10^{-2})\rho \times v^2 \quad \text{(F.14)}$$

or

$$\Delta p = C(5.102 \times 10^{-2})q_{\mathrm{m,a}}^2/\rho \quad \text{(F.15)}$$



a) Uncorrected Pressure Drop



b) Temperature Correction Factor

Key

X1 flue gas mass flow rate, 103 kg/h (102 lb-m/h)

Y1 pressure drop, Δp_1, expressed as millimeters H_2O per 30 linear m (inches H_2O per 100 linear ft)

X2 flue gas temperature, expressed in degrees Celsius (degrees Fahrenheit)

Y2 pressure-drop correction, C_1

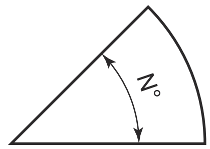
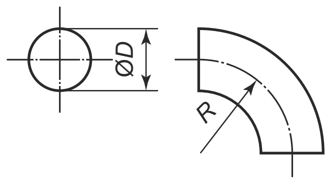
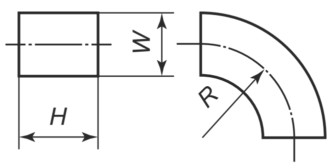
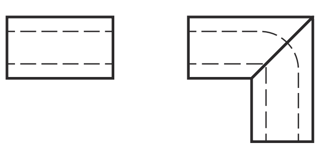
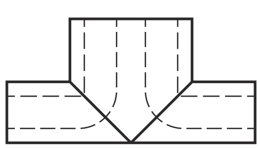
NOTE 1 Flue gas relative molecular mass is 28.

NOTE 2 Gauge pressure in duct is 100 kPa (14.5 psi).

NOTE 3 The bullet points in the figure coincide with a flue gas velocity of 15 m/s (50 ft/s).

Figure F.8—Duct Pressure Drop vs Mass Flow

Table F.2—Loss Coefficients for Common Fittings

Fitting Type	Fitting Illustration	Dimensional Condition	Loss Coefficient	*L/D* or *L/W*
Elbow of *N* degree turn (rectangular or round)		No vanes	*N*/90 times the value for a similar 90° elbow	
90° round section elbow		Miter [a]	1.30	65
		$R/D = 0.5$	0.90	45
		$R/D = 1.0$	0.33	17
		$R/D = 1.5$	0.24	12
		$R/D = 2.0$	0.19	10
90° rectangular section elbow		Miter $H/W = 0.25$	1.25	25
		$R/W = 0.5$	1.25	25
		$R/W = 1.0$	0.37	7
		$R/W = 1.5$	0.19	4
		Miter $H/W = 0.5$	1.47	49
		$R/W = 0.5$	1.10	40
		$R/W = 1.0$	0.28	9
		$R/W = 1.5$	0.13	4
		Miter $H/W = 1.0$	1.50	75
		$R/W = 0.5$	1.00	50
		$R/W = 1.0$	0.22	11
		$R/W = 1.5$	0.09	4.5
		Miter $H/W = 4.0$	1.35	110
		$R/W = 0.5$	0.96	85
		$R/W = 1.0$	0.19	17
		$R/W = 1.5$	0.07	6
90° miter elbow with vanes [a]			$C = 0.1$ to 0.25	
Mitered tee with vanes		Equal to an equivalent elbow (90°) (base loss on the entering velocity)		