TEXAS TECH UNIVERSITY

Department of Mechanical Engineering

ME 4251: Thermal Fluids Laboratory

Experiment 3

Determining the Discharge Coefficient of an Orifice Plate

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Abstract

The objective of this experiment was to determine the experimental discharge coefficient (C_d) of an orifice plate and establish an uncertainty bracket for the result. The orifice plate was tested in a fan-driven piping system in series with a Laminar Flow Element (LFE), which served as the primary measurement standard for actual volumetric flow rate. Data were collected across five fan speeds ranging from 20 Hz to 60 Hz. Theoretical ideal flow rates were calculated using the Bernoulli obstruction equation, while actual flow rates were derived from the LFE manufacturer's calibration constant and corrected for local air viscosity.

A linear curve fit of the actual volumetric flow rate as a function of the ideal volumetric flow rate yielded an experimental discharge coefficient of $C_d = 0.728$. Utilizing the Bevington and Robinson method for curve-fit uncertainty, the uncertainty of this coefficient was determined to be $u_{C_d} = \pm 0.0036$. Although the experimental C_d was higher than the values predicted by ASME standards, the low uncertainty indicates a highly repeatable and predictable relationship within the tested system. The results confirm that the orifice plate is an effective, simple device for volumetric flow measurement once calibrated for a specific piping arrangement.

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List of Symbols

A	Cross-sectional area of the pipe
B	LFE manufacturer's calibration constant
C_d	Experimental discharge coefficient
$C_{d,ASME}$	ASME standard discharge coefficient
D	Inner diameter of the pipe
d	Inner diameter of the orifice plate
F_1, F_2	Constants for pressure tap location correction
N	Number of data points
P, p	Pressure
p_{atm}	Atmospheric pressure
Δp_{fan}	Pressure increase provided by the fan
ΔP_{LFE}	Pressure drop across the laminar flow element
ΔP_{ORI}	Pressure drop across the orifice plate
Q	Volumetric flow rate
Q_{actual}	Actual volumetric flow rate
Q_{ideal}	Ideal volumetric flow rate
R	Gas constant for air
Re_D	Reynolds number based on pipe diameter
S	Standard deviation (standard error of the fit)
T_f	Temperature of the flowing air
T_s	Standard reference temperature
u	Uncertainty
β	Ratio of orifice diameter to pipe diameter (d/D)
Δ'	Spread of variables for uncertainty analysis
μ	Dynamic viscosity of the air
ρ	Local density of the air

1 Introduction

Fluid flow measurement is fundamental to the operation of industrial systems, municipal water distribution, and aerospace propulsion. Among various flow measurement technologies, the orifice plate remains one of the most widely utilized due to its simplicity, low cost, and lack of moving parts. However, because it relies on a sudden restriction of the flow area, the resulting pressure drop is influenced by complex phenomena such as fluid friction and the formation of a vena contracta. Therefore, the primary objective of this investigation is to determine the experimental discharge coefficient (C_d) of an orifice plate, establish its uncertainty using the Bevington and Robinson method, and compare the findings to established ASME standards.

The theoretical foundation of this experiment is rooted in two fundamental principles of fluid mechanics. First, the Law of Mass Conservation (the Continuity Equation) dictates that for a steady, incompressible flow, the volumetric flow rate (Q) remains constant. Consequently, a reduction in the cross-sectional area from the pipe (A_1) to the orifice (A_2) requires a proportional increase in fluid velocity (V):

$$Q = A_1 V_1 = A_2 V_2 \tag{1}$$

Second, Bernoulli's Principle (the Principle of Energy Conservation) states that for an ideal, inviscid fluid, an increase in kinetic energy must be balanced by a decrease in pressure energy. The energy balance between the upstream section and the orifice throat is expressed as:

$$p_1 + \frac{1}{2}\rho V_1^2 = p_2 + \frac{1}{2}\rho V_2^2 \tag{2}$$

By substituting the continuity relationship into the Bernoulli equation and accounting for the local air density (ρ), the ideal volumetric flow rate (Q_{ideal}) can be derived. The density calculation utilizes the Ideal Gas Law, incorporating ambient atmospheric pressure (p_{atm}), flow temperature (T_f), and the pressure increase provided by the system's fan (Δp_{fan}):

$$\rho = \frac{p_{atm} + \Delta p_{fan}}{RT_f} \tag{3}$$

Applying this density to the combined conservation equations yields the theoretical flow rate through the obstruction:

$$Q_{ideal} = \frac{\pi}{4} d^2 \left[\frac{2\Delta P_{ORI} R T_f}{(p_{atm} + \Delta p_{fan})(1 - \frac{d^4}{D^4})} \right]^{1/2} \tag{4}$$

Because this ideal model ignores real-world losses, a Laminar Flow Element (LFE) is used in series as a primary standard to measure the actual volumetric flow rate (Q_{actual}). The LFE operates on the Hagen-Poiseuille relationship, where the flow is calculated via a manufacturer's constant (B) and a temperature-based viscosity correction:

$$Q_{actual} = B \left(\frac{T_s}{T_f} \right)^{0.7} \Delta P_{LFE} \tag{5}$$

To reconcile the theoretical prediction with the experimental truth, the discharge coefficient (C_d) is introduced. It represents the ratio of the actual flow to the ideal flow, effectively capturing the combined effects of the vena contracta and viscous friction:

$$Q_{actual} = C_d Q_{ideal} \tag{6}$$

By evaluating the system across multiple operating frequencies, the overall experimental discharge coefficient can be determined as the slope of the linear relationship between the actual and ideal volumetric flow rates.

2 Methods

The experimental system consists of a flow-generation unit and a series of measurement instruments used to evaluate the performance of an orifice plate. The primary equipment under test is a beveled-edge orifice plate with a measured inner diameter of 1.378 ± 0.002 in, installed in a pipe with an inner diameter of 3.143 ± 0.009 in.

Airflow is generated by a **Spencer Turbine Co. Model 1002 Centrifugal Blower**, driven by a 3 HP variable-speed motor capable of reaching 3500 RPM. The flow standard for the system is a **Meriam Instrument Model 50MC2-4F Laminar Flow Element (LFE)**, which is calibrated to 400 CFM at a differential pressure of 8 inH_2O under standard conditions.

As shown in Figure 1, the system is instrumented with Type T thermocouples and electronic pressure transducers connected to a National Instruments (NI) data acquisition system. A Fortin barometer is used to determine the local atmospheric pressure, and fluid manometers are used to perform two-point calibrations for the transducers prior to testing.

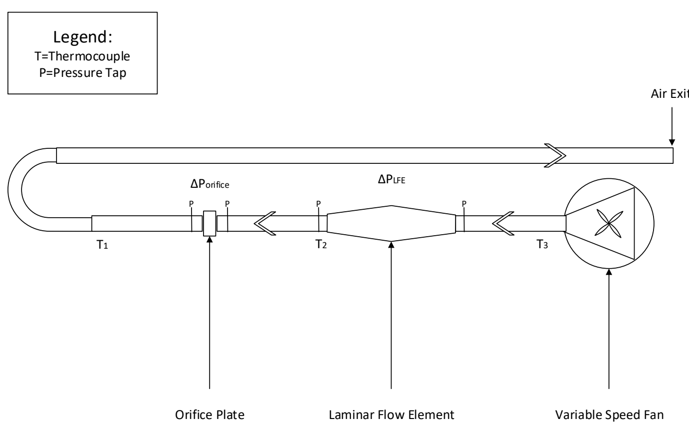

Figure 1: Schematic of the Experimental Flow Loop showing the Variable Speed Fan, Orifice Plate, and LFE locations.

The experimental procedure began by recording the ambient laboratory conditions using the Fortin barometer and thermocouples. The inner diameters of the pipe and orifice were measured using calipers to establish the diameter ratio (β). A two-point calibration was performed on the pressure transducers by comparing voltage outputs to known displacements on fluid manometers. The fan was then engaged at five discrete frequencies (20, 30, 40, 50, and 60 Hz). At each frequency, the system was allowed to reach steady state before the NI DAQ recorded the differential pressures across the orifice and LFE, as well as the flow temperatures. This process provided the necessary data to determine the relationship between the ideal and actual volumetric flow rates.

Table 1: Experimental Equipment and Instrument Specifications

Name	**Manufacturer / Model**	**Range**	**Resolution**
Orifice Plate	TTU Shop / Custom	$d = 1.378$ in	± 0.001 in
Centrifugal Blower	Spencer Turbine / 1002	0–60 Hz	0.1 Hz
Laminar Flow Element	Meriam / 50MC2-4F	0–400 CFM	0.01 CFM
Barometer	Fortin / [Model]	25–31 inHg	0.01 inHg
Pressure Transducer	Validyne / DP15	± 20 inH_2O	0.001 V
Thermocouple	Omega / Type T	0–200 °F	0.1 °F
DAQ System	NI / USB-6211	± 10 V	16-bit
Digital Caliper	Mitutoyo / CD-6"CSX	0–6 in	0.0005 in

3 Results and Discussion

The performance of the orifice plate was evaluated by comparing the measured actual volumetric flow rate (Q_{actual}) against the theoretical ideal volumetric flow rate (Q_{ideal}).

To establish a baseline for comparison, the theoretical ASME standard discharge coefficient ($C_{d,ASME}$) was calculated for each operating point. Based on the system's use of corner pressure taps, the geometric correction factors are nullified ($L_1 = L_2' = 0$). Therefore, the empirical ASME formulation simplifies to a function of the diameter ratio (β) and the pipe Reynolds number (Re_D):

$$C_{d,ASME} = 0.5959 + 0.0312\beta^{2.1} - 0.184\beta^8 + 0.0029\beta^{2.5}\left(\frac{10^6}{Re_D}\right)^{0.75} \tag{7}$$

Using the measured $\beta = 0.438$, this equation yields expected values ranging from 0.604 to 0.608. The calculated experimental results for the five tested fan frequencies, including the Reynolds numbers and both experimental and ASME discharge coefficients, are presented in Table 2.

Table 2: Summary of Experimental Flow Results and ASME Comparisons

Fan Speed (Hz)	Re_D	Q_{ideal} **(CFM)**	Q_{actual} **(CFM)**	**Exp.** C_d	**ASME** C_d
20	19,718	62.60	44.68	0.714	0.608
30	31,926	100.53	71.91	0.715	0.606
40	44,648	138.62	100.32	0.724	0.605
50	57,202	176.13	128.70	0.731	0.604
60	68,797	213.14	155.76	0.731	0.604

To determine the overall characteristic of the orifice plate, Q_{actual} was plotted as a function of Q_{ideal} (Figure 2). A linear curve fit of the data, forced through the origin to satisfy the theoretical relationship $Q_{actual} = C_d Q_{ideal}$, yielded an overall experimental discharge coefficient of **0.728**. Utilizing the Bevington and Robinson uncertainty method, the uncertainty for this coefficient was calculated to be ± 0.0036.

The experimental data demonstrates a highly linear relationship, as evidenced by the minimal scatter in Figure 2. However, a significant discrepancy exists between the experimental C_d values and the ASME standards. While the ASME values for this Reynolds number range are approximately 0.604–0.608, the experimental values are consistently higher, ranging from 0.714–0.731. This represents a percent error of approximately 20%.

Furthermore, a contradictory trend was observed regarding the Reynolds number. The ASME standard predicts a slight decrease in C_d as Re_D increases. In contrast, the experimental data showed a slight increase in C_d with increasing flow rate. Because the calculated uncertainty (u_{C_D}) is extremely low, this discrepancy cannot be attributed to random measurement error. Instead, it suggests a systematic bias. Potential sources for this offset include the specific geometry of the pressure tap locations in the TTU lab setup compared to the standard flange or corner taps

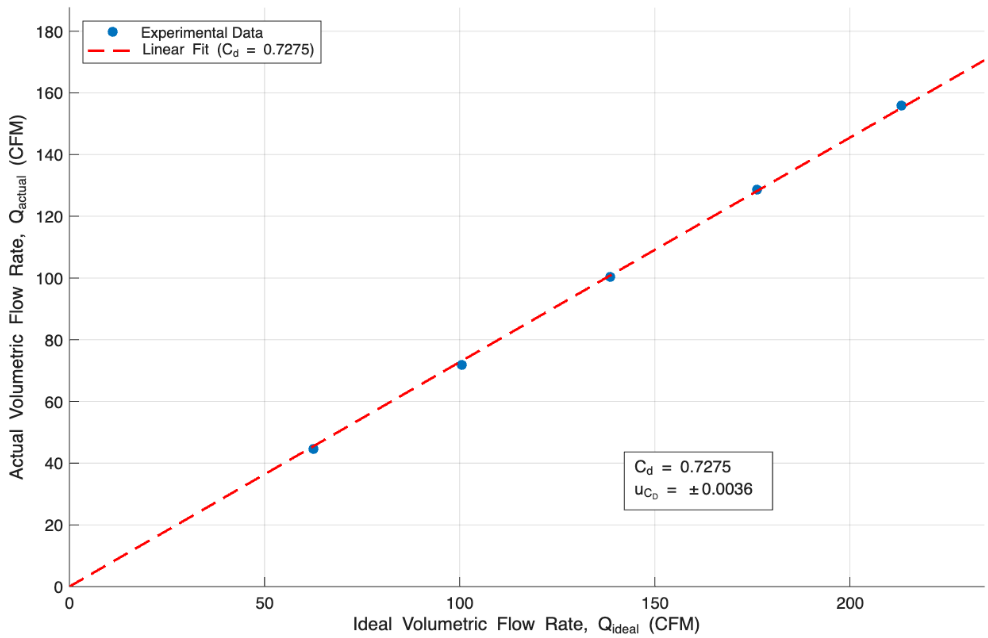

Figure 2: Actual vs. Ideal Volumetric Flow Rate with Linear Curve Fit (C_d).

used in ASME derivations, or a potential drift in the calibration of the Laminar Flow Element's manufacturer constant (B). Despite the absolute offset, the high repeatability of the results confirms that the orifice plate is a reliable flow measurement tool once its system-specific discharge coefficient is calibrated.

4 Conclusion

The experimental evaluation of the orifice plate has led to several significant conclusions regarding its performance and its suitability as a flow measurement device:

- The orifice plate exhibits a highly predictable and linear relationship between the theoretical ideal flow and the actual volumetric flow rate. This confirms that despite the complexities of the vena contracta and fluid friction, the device maintains a consistent performance characteristic across a wide operational range (20–60 Hz).

- The experimentally determined discharge coefficient ($C_d = 0.728$) is notably higher than the values predicted by ASME standards. While standardized coefficients are valuable for general estimates, this result underscores the necessity of system-specific calibration to ensure high accuracy in industrial applications.

- Statistical analysis of the flow relationship reveals an extremely low uncertainty bracket (± 0.0036). The precision of these results indicates that the discrepancy between the experimental and ASME values is not a product of random measurement error, but rather a systematic bias inherent to the experimental configuration.

- A contradictory trend was observed where the experimental discharge coefficient increased slightly with the Reynolds number, whereas ASME standards predict a subtle decrease. This suggests that the local piping geometry or the placement of pressure taps in this specific laboratory setup significantly influences the pressure-flow relationship differently than standardized theoretical models.

- It is recommended that future investigations include a multi-point calibration of the Laminar Flow Element (LFE) to verify the manufacturer's empirical constant (B). Additionally, testing the orifice plate with different pressure tap configurations (such as flange or corner taps) could help isolate the source of the systematic offset observed in this report.

Appendices

Appendix A: Experimental Data Sheets

The following tables summarize the raw measurements, calibration data, and processed results collected during the experiment.

Table 3: Ambient Conditions and System Geometry

Parameter	**Measurement / Value**	**Unit**
Indicated Atmospheric Pressure	27.01	inHg
Ambient Temperature	72.0	°F
Standard Corrected Pressure	26.90	inHg
Pipe Inner Diameter (D) Trials	3.130, 3.144, 3.155, 3.141	in
Average Pipe Diameter (D_{avg})	3.143	in
Orifice Inner Diameter (d) Trials	1.375, 1.378, 1.377, 1.381	in
Average Orifice Diameter (d_{avg})	1.378	in
Diameter Ratio (β)	0.438	–

Table 4: Two-Point Pressure Transducer Calibration Data

Transducer	V_1 **(No Flow)**	P_1 **(inH$_2$O)**	V_2 **(Max Flow)**	P_2 **(inH$_2$O)**
Fan	2.058 V	0.0	4.464 V	21.5
LFE	1.568 V	0.0	6.663 V	3.22
Orifice	2.100 V	0.0	4.692 V	23.2

Table 5: Raw Sensor Data across Fan Operating Frequencies

Fan Speed (Hz)	**Fan Transducer (V)**	**LFE Transducer (V)**	**Orifice Transducer (V)**	**Average T (°F)**
20	2.333	2.992	2.319	75.5
30	2.674	3.868	2.669	76.2
40	3.149	4.785	3.188	78.9
50	3.742	5.718	3.866	83.6
60	4.468	6.637	4.697	90.7

Table 6: Processed Flow Rates, Reynolds Numbers, and Discharge Coefficients

Fan Speed (Hz)	Q_{ideal} **(CFM)**	Q_{actual} **(CFM)**	**Exp.** C_d	**Reynolds No.** (Re_D)	**ASME** C_d
20	62.6	44.68	0.714	19,700	0.608
30	101.0	71.90	0.715	31,900	0.606
40	139.0	100.00	0.724	44,600	0.605
50	176.0	129.00	0.731	57,200	0.604
60	213.0	156.00	0.731	68,800	0.604

Appendix B: Calibration Data

B.1 Barometer Correction

To determine the absolute atmospheric pressure for density calculations, the indicated laboratory pressure was corrected for temperature expansion using the Fortin barometer relationships for brass and mercury.

- Indicated Pressure ($h_{indicated}$): 27.01 inHg
- Ambient Temperature (T_o): 72.0 °F
- Standard Corrected Pressure ($h_{standard}$): 26.90 inHg (at 32 °F)

This standard pressure converts to an absolute baseline of 1895.0 psf.

B.2 Pressure Transducer Calibration Equations

Prior to the experiment, the three variable reluctance pressure transducers were calibrated using fluid manometers. A two-point calibration (no-flow and max-flow) was performed to establish a linear relationship between the DAQ voltage output (V) and the differential pressure (ΔP in $\mathrm{inH_2O}$). The general linear form is:

$$\Delta P = m(V - V_1) + P_1$$

Using the calibration points from Appendix A, the following conversion equations were derived and programmed into the data reduction script:

1. **Fan Pressure Transducer:**

$$\Delta P_{fan} = \frac{21.5 - 0.0}{4.464 - 2.058}(V - 2.058) = 8.9360 \cdot V - 18.3903$$

2. **Laminar Flow Element (LFE) Transducer:**

$$\Delta P_{LFE} = \frac{3.22 - 0.0}{6.663 - 1.568}(V - 1.568) = 0.6320 \cdot V - 0.9910$$

3. **Orifice Plate Transducer:**

$$\Delta P_{ORI} = \frac{23.2 - 0.0}{4.692 - 2.100}(V - 2.100) = 8.9506 \cdot V - 18.7963$$

Appendix C: Sample Calculations

This section details the calculation process for the 20 Hz fan speed operating condition.

C.1 Local Air Density (ρ)

Using the Ideal Gas Law:

$$\rho = \frac{p_{atm} + \Delta p_{fan}}{RT_f} = \frac{(1895.0 \text{ psf}) + (12.80 \text{ psf})}{1717 \frac{\text{ft}\cdot\text{lbf}}{\text{slug}\cdot{}^\circ\text{R}} \cdot (535.17^\circ\text{R})} = 0.0671 \text{ lb/ft}^3$$

C.2 Ideal Volumetric Flow Rate (Q_{ideal})

Substituting geometric parameters and measured differential pressure (ΔP_{ORI} = 1.96 inH_2O):

$$Q_{ideal} = \frac{\pi}{4} d^2 \left[\frac{2\Delta P_{ORI} R T_f}{(p_{atm} + \Delta p_{fan})(1 - \beta^4)} \right]^{1/2} = 62.60 \text{ CFM}$$

C.3 Actual Volumetric Flow Rate (Q_{actual})

Using the LFE manufacturer constant ($B = 50$) and temperature correction:

$$Q_{actual} = B \left(\frac{T_s}{T_f} \right)^{0.7} \Delta P_{LFE} = 50 \left(\frac{529.67}{535.17} \right)^{0.7} (0.90) = 44.68 \text{ CFM}$$

Appendix D: Uncertainty Analysis

The uncertainty analysis follows the Taylor Series Method (TSM) for propagation of error, as outlined in the laboratory documentation.

D.1 Propagation of Error for Derived Quantities

The total uncertainty for each calculated variable is determined by the square root of the sum of the squares of the partial derivatives multiplied by their respective elemental uncertainties.

1. Uncertainty in Local Air Density (ρ):

$$u_\rho = \left[\left(\frac{\partial \rho}{\partial p_{atm}} u_{p_{atm}} \right)^2 + \left(\frac{\partial \rho}{\partial T_f} u_{T_f} \right)^2 \right]^{1/2}$$

Where:

$$\frac{\partial \rho}{\partial p_{atm}} = \frac{1}{R_{air} T_f}, \quad \frac{\partial \rho}{\partial T_f} = -\frac{p_{atm} + \Delta p_{fan}}{R_{air} T_f^2}$$

2. Uncertainty in Actual Volumetric Flow Rate (Q_{actual}):

$$u_{Q_{act}} = \left[\left(\frac{\partial Q_{act}}{\partial T_s} u_{T_s} \right)^2 + \left(\frac{\partial Q_{act}}{\partial T_f} u_{T_f} \right)^2 + \left(\frac{\partial Q_{act}}{\partial \Delta P_{LFE}} u_{\Delta P_{LFE}} \right)^2 \right]^{1/2}$$

Where:

$$\frac{\partial Q_{act}}{\partial T_s} = 0.7B \left(\frac{T_s}{T_f} \right)^{-0.3} \frac{\Delta P_{LFE}}{T_f}, \quad \frac{\partial Q_{act}}{\partial T_f} = -0.7 B T_s^{0.7} T_f^{-1.7} \Delta P_{LFE}$$

3. Uncertainty in Ideal Volumetric Flow Rate (Q_{ideal}): Given that $Q_{ideal} = \frac{\pi}{4} d^2 D^2 (D^4 - d^4)^{-1/2} \ldots$, the partial derivatives with respect to the geometry are significantly more complex:

$$u_{Q_{ideal}} = \left[\left(\frac{\partial Q_{ideal}}{\partial d} u_d \right)^2 + \left(\frac{\partial Q_{ideal}}{\partial D} u_D \right)^2 + \left(\frac{\partial Q_{ideal}}{\partial T_f} u_{T_f} \right)^2 + \left(\frac{\partial Q_{ideal}}{\partial p_{atm}} u_{p_{atm}} \right)^2 \right]^{1/2}$$

Where the critical geometric partial derivatives from the lab documentation are:

$$\frac{\partial Q_{ideal}}{\partial d} = \frac{\pi d D^6 \left[\frac{2 \Delta P_{ORI} R T_f}{p_{atm} + \Delta p_{fan}} \right]^{1/2}}{2(D^4 - d^4)^{3/2}}$$

$$\frac{\partial Q_{ideal}}{\partial D} = \frac{\pi d^6 D \left[\frac{2 \Delta P_{ORI} R T_f}{p_{atm} + \Delta p_{fan}} \right]^{1/2}}{2(D^4 - d^4)^{3/2}}$$

D.2 Discharge Coefficient Uncertainty (u_{C_D})

As specified by Bevington and Robinson, the uncertainty in the slope (C_d) of the linear relationship between Q_{ideal} and Q_{actual} is determined by:

$$S^2 = \frac{\sum_{i=1}^{N}(Q_{actual,i} - [c_1 Q_{ideal,i} + c_2])^2}{N-2}$$

$$\Delta' = N \sum Q_{ideal,i}^2 - \left(\sum Q_{ideal,i}\right)^2$$

$$u_{C_D} = \sqrt{\frac{N \cdot S^2}{\Delta'}}$$

For the data collected across five fan speeds ($N = 5$), the resulting uncertainty in the discharge coefficient was calculated to be ± 0.0036.

Appendix E: MATLAB Scripts

E.1 Orifice Plate Flow and Reynolds Number Calculations

```
%% Orifice Plate Flow and Reynolds Number Calculations
clear; clc;

% --- 1. USER INPUTS ---
% Measured values aligned with 20, 30, 40, 50, 60 Hz fan speeds.
freq_Hz           = [20; 30; 40; 50; 60];
T_f_F             = [75.5; 76.20; 78.90; 83.56; 90.72];
Delta_P_fan_inH2O = [2.46; 5.50; 9.75; 15.05; 21.54];
Delta_P_ORI_inH2O = [1.96; 5.09; 9.74; 15.81; 23.24];
Delta_P_LFE_inH2O = [0.90; 1.45; 2.03; 2.62; 3.20];

% System Geometry & Lab Pressure
p_atm_inHg = 26.90;  % Corrected lab atmospheric pressure
D_in       = 3.1425; % Average pipe inner diameter
d_in       = 1.3778; % Average orifice inner diameter

% --- 2. CONSTANTS & CONVERSIONS ---
Ts_R  = 70 + 459.67; % Standard temp in Rankine
B     = 400 / 8;     % LFE Manufacturer constant (SCFM/inH2O)
R_air = 1716;        % Gas constant for air in ft*lbf/(slug*R)

% Convert diameters to feet and calculate Beta
D_ft = D_in / 12;
d_ft = d_in / 12;
beta = d_in / D_in;

% Convert pressures to pounds per square foot (psf)
p_atm_psf       = p_atm_inHg * 70.7262;
Delta_P_fan_psf = Delta_P_fan_inH2O * 5.20233;
Delta_P_ORI_psf = Delta_P_ORI_inH2O * 5.20233;

% Convert temperatures to Rankine
T_f_R = T_f_F + 459.67;

% --- 3. CALCULATIONS ---
% Q_actual (in CFM)
Q_actual = B .* (Ts_R ./ T_f_R).^0.7 .* Delta_P_LFE_inH2O;

% Q_ideal (Calculate in ft^3/s first, then convert to ft^3/min)
term1 = (pi / 4) * d_ft^2;
term2 = (2 .* Delta_P_ORI_psf .* R_air .* T_f_R) ./ ...
```

```
        ((p_atm_psf + Delta_P_fan_psf) .* (1 - beta^4));
Q_ideal_cfs = term1 .* sqrt(term2);
Q_ideal = Q_ideal_cfs .* 60; % Convert to CFM

% Individual Cd for each run
Cd_per_run = Q_actual ./ Q_ideal;

% --- 4. REYNOLDS NUMBER CALCULATION ---
% Pipe cross-sectional area (ft^2)
A_pipe = (pi / 4) * D_ft^2;

% Actual flow velocity in the pipe (ft/s)
V_actual = (Q_actual ./ 60) ./ A_pipe;

% Local air density (slugs/ft^3) using Ideal Gas Law
rho_slugs = (p_atm_psf + Delta_P_fan_psf) ./ (R_air .* T_f_R);

% Dynamic viscosity of air (lb*s/ft^2) using Sutherland's Law
mu_slugs = (2.27e-8 .* T_f_R.^1.5) ./ (T_f_R + 198.6);

% Reynolds Number (based on Pipe Diameter D)
Re_D = (rho_slugs .* V_actual .* D_ft) ./ mu_slugs;

% --- 5. OUTPUT TABLE ---
ResultsTable = table(freq_Hz, T_f_F, Q_ideal, Q_actual, Cd_per_run, Re_D, ...
    'VariableNames', {'Fan_Speed_Hz', 'Temp_F', 'Q_ideal_CFM', ...
                      'Q_actual_CFM', 'Cd_per_run', 'Reynolds_Number'});
disp('--- Experimental Flow Rate Results ---')
disp(ResultsTable)
```

E.2 Barometer Pressure Correction

```
%% Barometer Pressure Correction
clc; clear all

% Inputs
h_indicated = 27.01; % inHg
T = 72;              % deg F

% Constants
alpha = 10.22e-6; % 1/F (brass linear expansion)
beta  = 101e-6;   % 1/F (mercury volumetric expansion)
```

```
% Temperature differences from 32 F reference
dT_scale = T - 62;
dT_Hg    = T - 32;

% Correction
delta_h = ((alpha*dT_scale - beta*dT_Hg) / (1 + beta*dT_Hg)) * h_indicated;

% Standard reading
h_standard = h_indicated + delta_h
```

Appendix F: Lab Handout

The original Spring 2026 ME 4251 Experiment 3 instructional handout is attached below.

Determining the Discharge Coefficient of an Orifice Plate, 2026 Spring

Objective

You will determine the discharge coefficient of an orifice plate. Knowing the discharge coefficient, you would be able to use this remarkably simple device with no moving parts together with a pressure measurement system to measure volume flowrate in a piping system.

You will also find an uncertainty bracket for your estimate of the discharge coefficient. You will also compare your result to an established value from the literature. You will also prepare your first formal and complete group lab report.

The lab report is due **2 weeks** from date of the lab in which you collect the data.

Test Procedure

1. Measure absolute pressure in the lab via the barometer.
2. Measure the inner diameters of the orifice plate and of the pipe.
3. Perform 2-point calibrations of the electronic pressure transducers using manometers. Will least squares fits be required? Would plots of the calibration data be helpful? Discuss with your lab instructor.
4. Measure all other required quantities (e.g., transducer voltages and **flow temperatures** via type T thermocouples using the NI data acquisition system) to determine actual volumetric flowrates and ideal volumetric flowrates for **five** different fan speeds.

Analysis Procedure

1. Compile a table of all **measured** data.
2. Create a PYTHON or MATLAB script or EXCEL file to perform all necessary computations, including calculations of the actual flowrate and the ideal flowrate for each test condition.
3. Compile a table of **calculated** quantities. Let the flowrates be expressed in ft^3/min. Note that your tables should not simply be filled in versions of the lab data sheet. Design the tables yourself in the most logical and straightforward manner possible. Format all quantities to the correct level of precision. Specify units, where appropriate, in column or row headings.
4. Plot the actual flowrate (vertical axis) as a function of the ideal flowrate (horizontal axis).
5. Perform a linear curvefit of this plot to determine your estimate of the discharge coefficient.

6. All measured and calculated data presented in your report should be displayed to three significant figures (or four, if the first digit is a one).

7. Compare your result to the ASME-established value for the discharge coefficient, as described in White, Fluid Mechanics, 7th edition. A good way to do this would be to compute the ASME C_D according to the equations given in White, 7th edition, p. 423, for each flow condition. Why would it be nice if the five values of C_D turn out to be approximately equal to one another?

8. Consult sections 3 and 4 especially of the Thermal Fluids Handbook, as posted on blackboard, to achieve your best possible work.

Uncertainty Analysis

Because a major goal of the lab is to determine the discharge coefficient, C_D, we must also determine an estimate in the uncertainty of C_D. This is quite challenging because we found C_D as a curve fit coefficient. So what the heck is the uncertainty of that? A method is outlined in Bevington and Robinson and I have posted an excerpt to blackboard. It is really just a laborious, detailed application of the error propagation method we have already learned in our first lab. The result is:

$$u_{C_D} = \sqrt{N \frac{S^2}{\Delta'}} \tag{1a}$$

where N is the number of data pairs in the curve fit and where S^2 is the square of the standard error

$$S^2 = \frac{\sum \left[Q_{actual,i} - (c_1 Q_{ideal,i} + c_2)\right]^2}{n-2} \tag{1b}$$

and

$$\Delta' = N \sum Q_{ideal,i}^2 - \left(\sum Q_{ideal,i}\right)^2 \tag{1c}$$

If you read the material from Bevington and Robinson, you may notice that our data does not perfectly meet the criteria for this analysis. So be it.

General Comments on Lab Reports

Remember that the purpose of the report is to document what you have done, providing enough information so that a future worker could reproduce your results without having to personally consult you. Your work should be self-contained and complete.

1. In the "Introduction" section you do not need to re-derive well-known results from fluid mechanics. However, you should go beyond simply listing the equations that I have presented in lecture. You should alert your reader to the physical principles behind each equation and perhaps state the assumptions and limitations relevant to the current experiment.

2. Use an equation editor. Do not simply cut-and-paste equations from my slides.

3. In the "Methods" section you should clearly distinguish between the equipment under test and measurement equipment. Also, use some judgment in determining what equipment deserves to be characterized in detail. For example, is it more

important to describe in detail the laminar flow element (LFE) or the computer workstation? Also, in characterizing measurement equipment try to convey its capabilities, e.g., range and resolution. Also, if equipment was present, but it was not used in your work, it should not be included in your report.

4. Any tabulated data anywhere in the report should be meticulously formatted. All columns of data should be labeled with a variable name and with units. All numerical entries should be displayed to an appropriate number of digits based upon the method of measurement or calculation. Data tables should not simply echo those supplied by us for your data collection but should be appropriate for their purpose in your report.

5. Short is good–provided the job is done.

A References

1 Texas Tech University, *Thermal Fluids Lab Handbook*, Department of Mechanical Engineering, Lubbock, TX.

2 Bevington, P. R., and Robinson, D. K., *Data Reduction and Error Analysis for the Physical Sciences*, 3rd ed., McGraw-Hill, New York, NY, 2003.

3 American Society of Mechanical Engineers, *Measurement of Fluid Flow in Pipes Using Orifice, Nozzle, and Venturi*, ASME MFC-3M-2004, New York, NY, 2004.

4 White, F. M., *Fluid Mechanics*, 8th ed., McGraw-Hill Education, New York, NY, 2015.