CHARACTERISTICS OF DISCHARGE COEFFICIENT IN INDUSTRIAL BURNER APPLICATIONS

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ABSTRACT
Experimental investigation is made to determine discharge coefficient of different types of burners used in industrial combustion applications. Tests were carried out to study the influence of burner geometry and flow variables, such as Reynolds number, porosity, length/diameter ratio and number of holes on discharge coefficient. Results have shown reasonable agreement when compared with data done by other researchers. Tests were carried out at Sudan University of Science and Technology-College of Engineering.

INTRODUCTION
There is a tendency of operating the combustion zone at lower equivalence ratio in order to reduce flame temperature, improve the temperature pattern and to reduce the combustion pollutants level at the exit of the combustion system. This necessitates a substantial increase of the air admitted to the combustion zone. There is a little information on $C_d$ for multi-holes orifice plate normal to the flow direction specially grid plate with $L/D$ ratio less than 0.3. Systems are suggested to be rapid mixing burners for combustion in industrial applications, which could reduce the combustion pollutant to level similar to the fully premixed systems (Aldabbagh, 1988).

Although industrial combustion engines contribution to total air pollution is very small compared to other types of heat engines (petrol, diesel, and coal fired),
increase demand and utilization in different applications make them significant source of pollution.

Aldabbagh, and Andrews, (1983) showed that flame stabilizer geometry has a major influence on combustion efficiency and flame stability but less influence on NOx.

Therefore, it is necessary to know burner pressure drop and hence pressure loss coefficient in order to design burner pressure drop at specific Mach number.

The present work investigates pressure loss characteristics for different types of burner by studying the use of these constructions in a number of different contexts to reduce flow non-uniformities in ducts, for boundary layer control applications, and as a mean of conveniently simulating the pressure drop characteristics of some other more complex component in a mock up of a real system. Knowledge of the flow characteristics of squared edged orifices of small diameters is important in a number of applications, such as fluid power engineering, pneumatics techniques of metrology and fluidics.

Experimental Equipment:

A schematic layout of the test rig, built at Sudan University of Science and Technology-College of Engineering, is shown in (Fig. 1). The system used to provide the air consists of a blower driven by an electric motor. A venturi meter was positioned at a distance to allow for fully developed flow and consequently accurate flow metering.

A manometer is used to measure the venturi static and differential pressure and static pressure loss. The temperatures of air at inlet and of the venturi and of the air upstream burner are measured by sensor thermocouples. Different type of burners has been tested to evaluate the geometric effect on pressure loss coefficient. (Fig.2) showed Venturi flow meter used in the present investigation, the fluid is accelerated through a converging cone of angle 15-20° and the pressure difference between the upstream side of the cone and the throat is measured and provides the signal for the rate of flow.

Theoretical Approach:

Calculation of Air Mass Flow: Air mass flow rate was calculated according to British Standard B.S. 1042. The basic equation of the mass flow rate is given by:

\[ \text{Mass Flow Rate} = \frac{C_d \cdot A \cdot \sqrt{2 \cdot \gamma \cdot (P_2 - P_1)}}{\rho} \]

where:
- \( C_d \) is the discharge coefficient
- \( A \) is the throat area
- \( \rho \) is the density of air
- \( P_1 \) and \( P_2 \) are the pressures upstream and downstream of the throat respectively
- \( \gamma \) is the specific heat ratio of air
\[ m' = C_D Z E \varepsilon A(2\rho \Delta P)^{0.5} \]  
\text{Where} \quad C_D = \text{Discharge Coefficient} \\
Z = \text{Correction factor} \\
\varepsilon = \text{Expansibility factor, B. S. 1042[10]. expressed as}

\text{Fig. (1): General Layout of the Test rig}

\text{Fig. (2): Venturi Flow Meter}
\[ \varepsilon = \left\{ \frac{\gamma^2}{\gamma - 1} \left[ \frac{1 - s^2}{1 - s^2 r^2} \right] \right\} \left( \frac{1 - r^\gamma}{1 - r} \right)^{\frac{1}{2}} \]  
\[ \text{(4-2)} \]

where:

\[ \gamma = \text{specific heats ratio } C_p/C_v, \quad C_p = \text{Specific heat at constant pressure, } \]
\[ C_v = \text{Specific heat at constant volume, } E = 1/(1-s^2)^{0.5} \text{ where } s = (d/D)^2, \quad d = \text{throat diameter, } m, \quad D = \text{pipe diameter, } m, \quad r = \text{Ratio of the absolute pressure at the upstream tapping to that at the venturi throat.} \]

From equation (1) Discharge Coefficient can be calculated: B. S. 1042:

\[ C_D = \frac{m}{A_2 (2 \rho \Delta P)^{0.5}} \]  
\[ \text{(2)} \]

Where:

\[ m = \text{actual mass flow rate from venturi meter, } \rho = \text{inlet density kg/m}^3, \quad A_2 = \text{open area, } C_D = \text{overall discharge coefficient.} \]

Pressure drop as percentage of upstream pressure is given by:

\[ \frac{\Delta P_{12}}{P} \% = \frac{(h_{1s} - h_{2s}) \times \rho_w \times g \times 100}{P_a + \rho_w \times g \times h_{1T}} \]  
\[ \text{(3)} \]

\[ \frac{\Delta P}{P} \% = \frac{\Delta P \times 100}{P_a + \Delta P} \]  
\[ \text{(4)} \]

Where:

\[ h = \text{pressure in H}_2\text{O, } \Delta P = \text{pressure loss, } \rho_w = \text{water density kg/m}^3, \quad g = \text{gravimetric acceleration (m/s}^2\text{), } \Delta h = \text{pressure difference in m H}_2\text{O, } P_a = \text{atmospheric pressure in N/m}^2. \]

Equation (2) and (4) may be combined to get:

\[ \frac{\Delta P}{P} = \frac{\gamma}{2} \left( \frac{M}{C_D} \times \frac{A_1}{A_2} \right)^2 \]  
\[ \text{(5)} \]
Correction of the Pressure Drop to Reference Mach number:

The pressure drop is a function of Mach no. (1). It is useful to correct the measured pressure drop of the stabilizer to the standard Mach no. of 0.0467 ($M_{ref}$) B. S. 1042[10].

$$\frac{\Delta P}{P}_{\text{corr}} = \left(\frac{\Delta P}{P}_{\text{meas}} \times \left(\frac{M_{ref}}{M_{meas}}\right)^2\right) \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldot
Influence of Porosity: The variations of discharge coefficient with porosity are shown (Fig. 5 and 6) for different types of burners. The results show that values of discharge coefficient are increased with increase in porosity and this can be explained by dependence of pressure recovery distance on area ratio results in improving discharge coefficient values.
Influence of Pressure Loss: Pressure loss can be shown to be related to area ratio by the following equation:

\[
\frac{\Delta P}{P} = 0.5 \frac{u^2}{C_D^2} \frac{1}{RT} \left[ \frac{A_1}{A_2} \right]^2
\]

Where, \(A_1/A_2\) is the combustor area to burner area ratio, \(C_D\) discharge coefficient.

Therefore an opposite trend can be seen in (Fig. 7) which shows that the higher the pressure loss the lower the discharge coefficient and this due to the inverse proportionality of pressure loss with area ratio. This is supported by the theoretical data plotted on the same graph based on sharp-edged orifice plate.

Influence of Wall Thickness/Diameter (L/D) Ratio:

Another important non-dimensional parameter beside the porosity is the L/D ratio, which can be formed from the basic dimensions and it is a convenient specification of the orifice geometry.

The influence of L/D ratio is shown in (Fig.8) for the same number of holes. The results are plotted together with data based on the same area ratio which obtained from, (Smith, C. F., 1982), and showed a reasonable agreement.
In both figures there is a trend of increase in discharge coefficient with increase in L/D ratio.

**Fig. (6):** Discharge Coefficient VS Area Ratio for Different Types of Burners

**Fig. (7):** Discharge coefficient VS Pressure Loss, for Different Types of Burners
Influence of Number of Holes:

(Fig. 8) shows the variation of discharge coefficient with number of holes for different burner pressure drop. It shows a trend of increasing discharge coefficient values with increase in number of holes. The larger recirculation zone may explain this for the lower number of holes causing an abstraction to the flow, results in lower discharge coefficient values.
COMPARISON OF RESULTS

Results have been compared with data done by, (Al-dabbagh, N. A. and G. E. Andrews, 1988) and showed a good agreement as illustration in (Fig. 10-14).

Fig. (10) Experimental $C_d$ Vs Re-compared with others

Fig. (11): Experimental $C_d$ Vs $A_2/A_1$ compared with others
CONCLUSIONS

- Discharge coefficient of different types of perforated plats used as a burner have been measured and showed a reasonable agreement when compared with data obtained by other researchers.
- Reynolds number has no significant influence on discharge coefficient in the region where Re, higher that 8000 especially for the lower pressure loss burner.
Discharge coefficient values increased for higher area ratio for the same L/D ratio.
- For the same area ratio and the same number of holes the discharge coefficient increased with increase in L/D ratio and the influence of number of holes increases as L/D ratio increases.

Fig. (14): Experimental $C_d$ Vs no. of holes compared with others

REFERENCES
2- British Standards Institution B. S. Methods of the measurement of fluid Flow in pipes, 1042 part 1, Section 1.1 and 1.2, 1964 .