

THE EFFECT OF SWIRL NUMBER ON DISCHARGE COEFFICIENT FOR VARIOUS ORIFICE SIZES IN A BURNER SYSTEM

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ABSTRACT

A liquid fuel burner system with different orifice plate sizes mounted at the exit plane of the air swirler outlet has been investigated using a 140 mm inside diameter combustor of 400 mm length. Three different orifice plate diameters of 45 mm, 40 mm and 35 mm were used with a 10° to 70° radial air swirler vane angles. The purpose of orifice plate insertion is to create the swirler pressure loss at the swirler outlet so that the swirler outlet shear layer turbulence could be maximized to assist in the mixing of fuel and air. Measurements of air flow rates, air inlet temperature, air inlet pressure and pressure drop across the combustor have been carried out to predict the isothermal characteristics inside the combustor, ie. discharge coefficient. The result shows that the discharge coefficient decreases as the swirl number increases for all three different orifice plate sizes.

Keywords : Swirler, orifice plate, pressure drop, swirl number, discharge coefficient

1.0 INTRODUCTION

There are several requirements that must be considered when designing a new combustor especially to meet the stringent regulation regarding emissions level from the exhaust. These requirements include stability limits, high combustion efficiency, high intensities of heat release, low burner pressure drop and low emissions production from the combustion. One-way to achieve this is the use of swirling air flow. Swirling flow is used for the stabilisation and control of the flame and to achieve a high intensity of combustion. Swirling flow induces a highly turbulent recirculation zone, which stabilises the flame resulting in better mixing and combustion [1]. It has been suggested that the large toroidal recirculation zone plays a major role in the flame stabilisation process by acting as a store for heat and chemically active species and, since it constitutes a well-mixed region, it serves to transport heat and mass to the fresh combustible mixture of air and fuel [2]. The common method of generating a swirling flow is

by using radial angled vanes in the passages of air. The characteristics of the swirling flow depend on the swirler vane angle [3].

Beer and Chigier [4], promoted the methods of inducing rotation in a stream of fluid and these can be divided into three principal categories:

- a. Tangential entry of the fluid stream, or of a part of it, into a cylindrical duct.
- b. The use of a guide vane in axial tube flow.
- c. Rotational of mechanical devices that impart swirling motion to the fluid passing through them. This include rotating vanes or grids and rotating tubes.

They concluded that method (a) and (b) are generally used in practice but method (c) is sometimes applied for experimental investigation of swirling flow. Most conventional liquid fuel burners employ the axial-flow type swirler. This swirl vanes are usually flat for easy manufacturing process, but curved blade may give better performance in aerodynamic properties [5].

Syred and Beer [6], showed that for tangential entry including radial vanes, the swirling flow may produce high efficiency for isothermal performance and low loss of pressure coefficient compared to that when using axial straight and profiled vane.

Al-Kabie [7], on the other hand found a significant improvement in performance as well as NO_x emissions when using radial swirler compared to the axial swirler due to the immediate contact of fuel with turbulence by swirled air as it leaves the fuel nozzle. Aerodynamic curved vanes allow the incoming axial flow force to turn gradually which inhibits flow separation on the suction side of the vane. This means that more complete turning and higher swirl and radial velocity component can be generated at the swirler mouth, with the added advantage of lower pressure loss.

Al-Kabie [7], also investigated the discharge coefficient (C_D) for various radial swirlers. He showed that, by using the radial swirler, the discharge coefficients were low, approximately 0.6 compared with the zero angle blade, approximately 0.9. Poor C_D is due to the vane angle and not just the 90° inlet and outlet blades. This led to a major consideration of the flow field inside the swirler vane passage as low C_D implied that flow separation occurred in the passage in spite of the curved vane.

2.0 SWIRL NUMBER

The swirl number is usually defined as the fluxes of angular and linear momentum [4] and it is used for characterising the intensity of swirl in enclosed and fully separated flows.

The parameter can be given as [4]:

$$S = \frac{G_\phi}{G_x \cdot r_o} \quad (1)$$

where G_ϕ is the axial flux of angular momentum:

$$G_\phi = 2\pi \int_0^{r_o} \rho U_x U_\theta r^2 dr \quad (2)$$

and G_x is the axial flux of momentum (axial thrust):

$$G_x = 2\pi \int_0^{r_o} \rho U_x^2 r dr + 2\pi \int_0^{r_o} p r dr \quad (3)$$

In the above, r_o is the outer radius of the swirler and U_x and U_θ are the axial and tangential components of velocity at radius r respectively.

Since the pressure term in Equation (3) is difficult to calculate due to the fact that pressure varies with position in the swirling jet, the above definition for swirl number can be simplified by omitting this pressure term. Swirl number can be redefined as:

$$S' = \frac{G_\theta}{G'_x r_o} \quad (4)$$

where

$$G'_x = 2\pi \int_0^{r_o} \rho U_x r dr \quad (5)$$

The swirl number should, if possible, be determined from measured values of velocity and static pressure profiles. However, this is frequently not possible due to the lack of detailed experimental results. Therefore, it has been shown [4] that the swirl number may be satisfactorily calculated from geometry of most swirl generator. According to Claypole and Syred [5], if a perfect mixing and conservation of momentum is assumed, then the swirl number can be defined in term of the geometry of the combustor.

$$S_g = \frac{r_o \pi r_e}{A_t} \left[\frac{\text{tangential flow}}{\text{total flow}} \right]^2 \quad (6)$$

where

r_e is the radius of the swirler exit
 A_t is the total area of tangential inlet

Another form of geometric swirl number has been formulated by Al-Kabie [7] and given as:

$$S_a = \frac{\sin \theta}{1 + \frac{1}{\tan \theta}} \left[\frac{A_3}{C_c A_2} \right] \quad (7)$$

where

A_3 is the swirler exit area

A_2 is the swirler minimum throat area

C_c is the swirler contraction coefficient

Value for C_c , the swirler contraction coefficient, C_D , the swirler discharge coefficient and hence the swirl numbers were obtained using the following equations.

The pressure loss coefficient, K_{th} may also be expressed in term of mass flow rates as:

$$K_{th} = 2\rho\Delta P \left(\frac{A_2}{\dot{m}} \right) \quad (8)$$

where

ΔP is the pressure drop

\dot{m} is the volumetric air flow rates

The discharge coefficient, C_D is defined as:

$$C_D = \frac{1}{\sqrt{K_{th}}} \quad (9)$$

By combining Equations (8) and (9), an expression for the discharge coefficient in term of swirler pressure drop and air mass flow rates can be obtained as:

$$C_D = \frac{\dot{m}}{A_{th} \sqrt{2\rho\Delta P}} \quad (10)$$

The pressure loss coefficient can also be expressed in terms of contraction coefficient, C_c :

$$K_{th} = \left(\frac{1}{C_c} - \frac{A_{th}}{A_3} \right)^2 \quad (11)$$

By combining Equations (10) and (11), an expression for contraction coefficient in terms of the discharge coefficient, throat area and swirler exit area can be obtained as follows:

$$C_c = \frac{C_D}{1 + \left(\frac{C_D A_{th}}{A_3} \right)} \quad (12)$$

The value of C_c is dependent on the value of C_D and is obtained through experimental tests. These values of C_c are then used in Equation (7) to determine the geometric swirl number, S_a .

3.0 EXPERIMENTAL SET-UP

The drawing for radial air swirler is shown in Figure 1. Table 1 shows the various dimensions of radial air swirler used in the present work. The air swirlers are made from mild steel. They were manufactured in various angles to investigate the effect of pressure loss due to swirl number on the overall performance of the air swirler.

The general set-up for isothermal tests is shown in Figure 2. The rig is composed of a 140 mm inside diameter plenum chamber of 400 mm length and 140 mm inside diameter of flame tube with length of 400 mm. Tests were carried out at room temperature with varying air flow rates to produce different pressure drops. Industrial root blower was used for air supply at below 0.5% pressure loss.

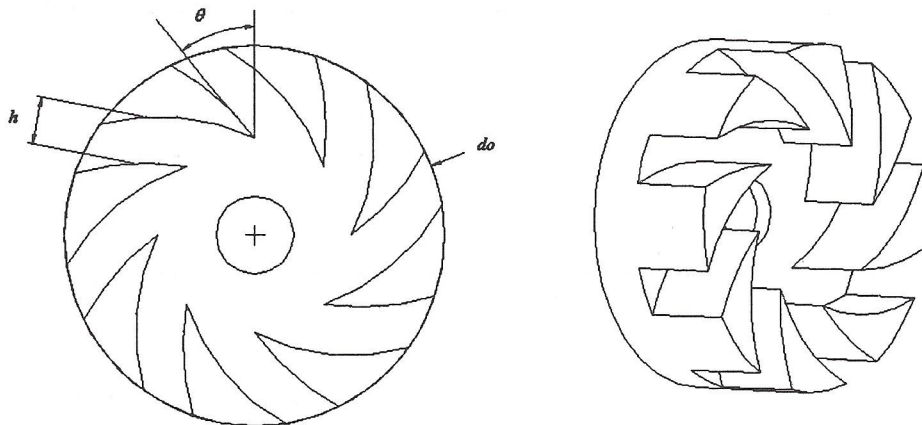


Figure 1 Schematic of Radial Air Swirler Design

Table 1 Dimensions of various radial swirler

Swirler angle	0°	10°	20°	30°	40°	45°	50°	60°	70°
Parameter									
Passage width, h (mm)	12	16	15	13.6	12.3	12	11.2	9.6	8.8
Swirl number, S_N	0	0.046	0.172	0.366	0.630	0.780	0.978	1.427	1.911
No. of vane, n	8								
Outlet diameter, d_o (mm)	98								
Inlet diameter, d_i (mm)	50								
Vane depth, L (mm)	25								

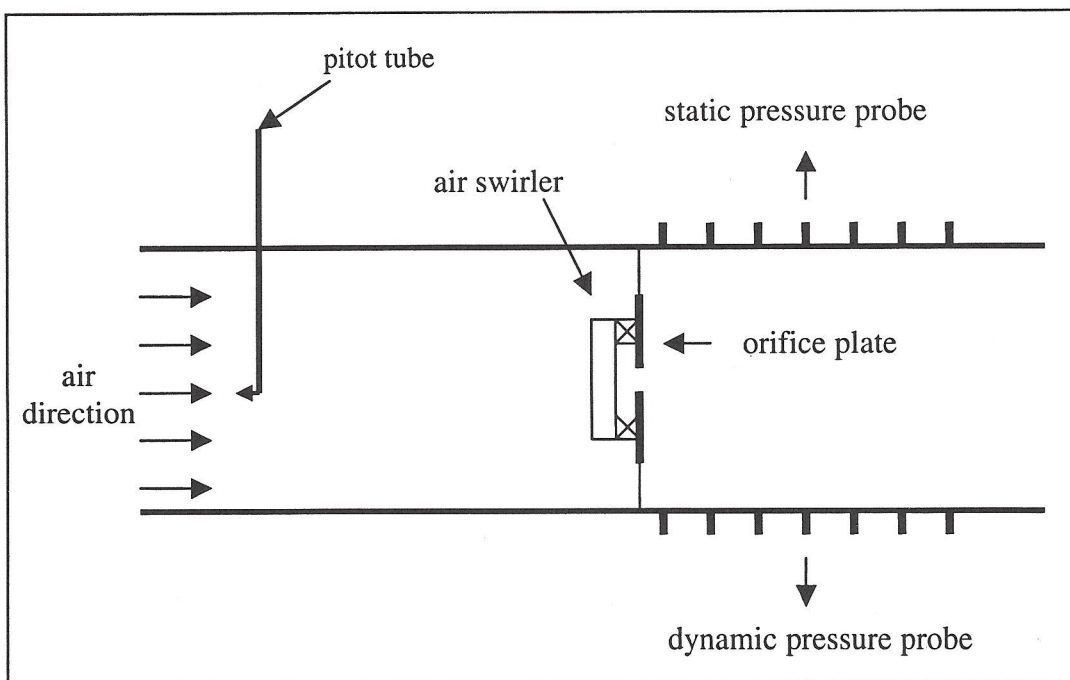


Figure 2 Schematic Diagram of Isothermal Test for Radial Swirler

Tests were also conducted using three different orifice plate sizes. The orifice was inserted at the exit plane of the radial swirler to enhance flame stabilisation and to provide better mixing of the air and fuel for combustion. The orifice plate also helps to prevent fuel from entraining into the corner recirculation zone that otherwise will create a local rich zone. The sizes of the orifice plate tested were 45 mm, 40 mm and 35 mm.

4.0 RESULTS AND DISCUSSION

In order to achieve better mixing between fuel and air in a liquid fuel burner, turbulence flow must be generated to promote mixing. Turbulence energy is created from the pressure energy dissipated downstream of the flame stabilizer. In the radial swirler, turbulence can be generated by increasing the aerodynamic blockage or by increasing the pressure drop across the swirler.

The discharge coefficient for radial swirler were obtained by passing a metered air flow through the radial swirler and flame tube while monitoring the static pressure loss upstream of the radial swirler relative to the atmospheric pressure. The results were plotted as a function of Reynolds number and presented in Figures 3 to 6.

From Figure 3, it can be seen that generally all discharge coefficients were approximately constant with variation in Reynolds number. Thus, the value of the discharge coefficient may be concluded to be independent of Reynolds number. In the case of non-orifice plate for all radial swirlers, 0° vane angle swirler or the zero swirl number gave the highest C_D at approximately 0.95. The C_D values decreased with the increase in swirl number, with the highest swirl number of 1.911 having the C_D value of approximately 0.63. This may be attributed to the fact that the excessive swirl was generated by the restriction on the swirler passage width.

Figures 4 to 6 show the effect of inserting different sizes of orifice plate. It can be seen that inserting the orifice plate decreases the C_D values. The smallest orifice size demonstrated the lowest discharge coefficient. This very low discharge coefficient value implies that flow separation may have occurred in the passages despite of the curved blades.

Figure 4 shows the effect of inserting 45 mm orifice plate on C_D values. The highest swirl number demonstrated the lowest discharge coefficient of 0.623. The highest discharge coefficient when using this type of orifice plate is 0.69, which is still quite low compared to non-orifice plate test. Figure 5 and Figure 6 show the plotted discharge coefficient against Reynolds number when using 40 mm and 35 mm orifice plate diameters. Once again it is noted that the smaller size orifice plate demonstrated smaller number of discharge coefficients compared to the larger ones. For the swirl number of 0.046, the discharge coefficients dropped around 8% for 40 mm orifice plate diameter compared to that of the 35 mm size of orifice plate diameter.

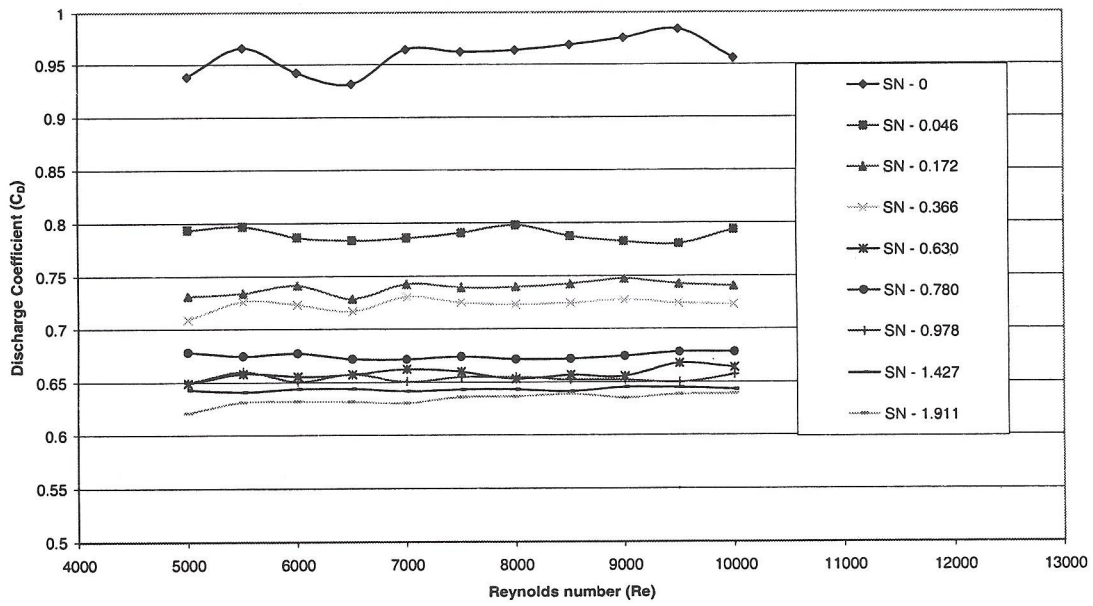


Figure 3 Discharge coefficient vs Reynolds number for various swirl numbers without orifice plate, cold test

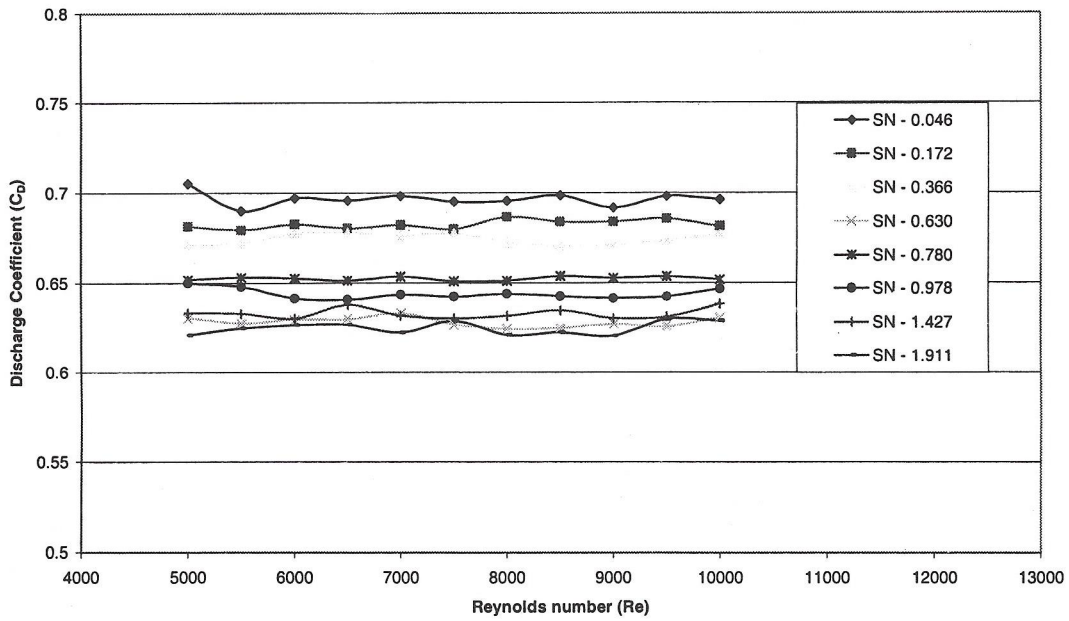


Figure 4 Discharge coefficient vs Reynolds number for various swirl numbers with 45 mm orifice plate, cold test

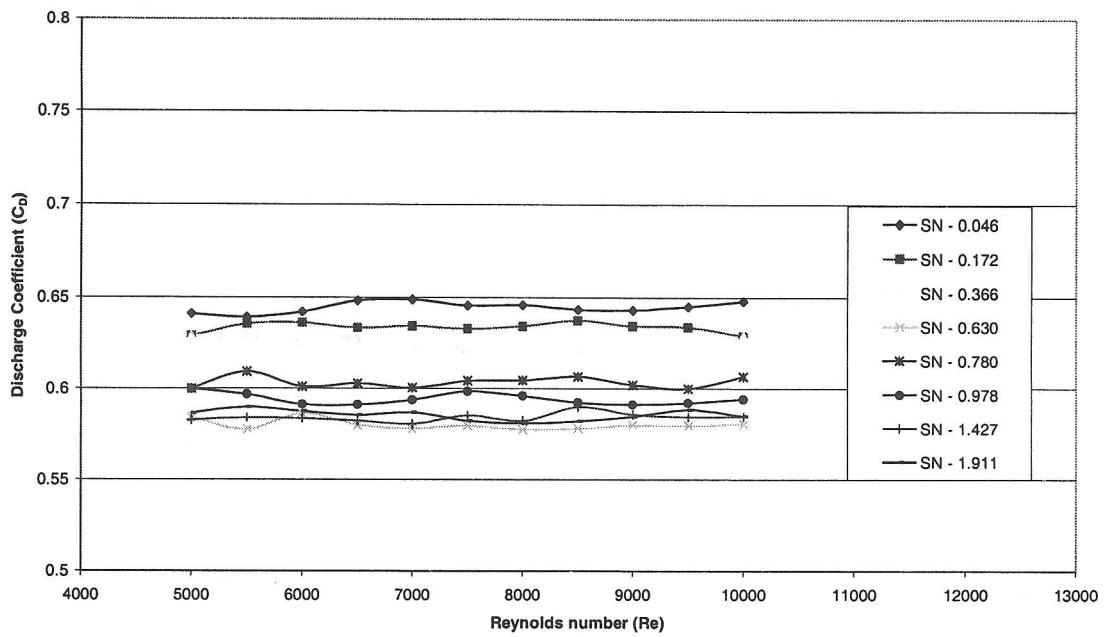


Figure 5 Discharge coefficient vs Reynolds number for various swirl numbers with 40 mm orifice plate, cold test

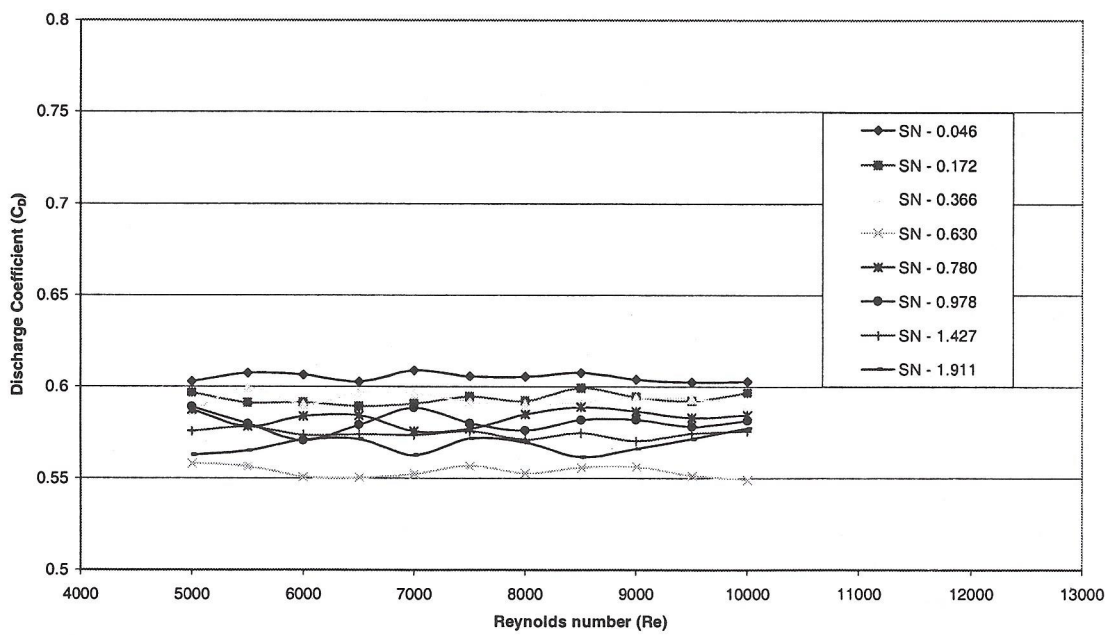


Figure 6 Discharge coefficient vs Reynolds number for various swirl numbers with 35 mm orifice plate, cold test

5.0 CONCLUSION

The characteristic of swirler discharge coefficient led to a major consideration of the flow field inside the radial swirler passages as low C_D implied that flow separation occurred in the passages in spite of the curved blade. The results of the experiment showed that, with no orifice assistant, radial swirler with 70° vane angle gave the smallest discharge coefficient. The reason behind this was probably due to the reduction of separation inside the passage at the swirler throat due to the reduction of swirler minimum area because of the increase in vane angle. Swirler with orifice assistant gives the smaller value of discharge coefficient compared to the non-orifice plate assistant. This shows that orifice plate increases the total pressure loss and for a specific orifice plate, the C_D reduces as the orifice diameter was reduced.

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